

Fans for general purposes — Method of measurement of fan vibration

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National foreword

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**Industrial fans — Method of
measurement of fan vibration**

Ventilateurs industriels — Méthode de mesure des vibrations des ventilateurs



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Foreword

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

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

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

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO 14695 was prepared by Technical Committee ISO/TC 117, *Industrial fans*.

Introduction

ISO 14695 is a part of a series of standards covering important aspects of fans which affect their design, manufacture and use. This series includes ISO 5801, ISO 5802, ISO 12499, ISO 13347, ISO 13349, ISO 13350, ISO 13351 and ISO 14694.

Vibration is recognized as an important parameter in the description of the mechanical performance of fans. It gives an indication of how well the fan has been designed and constructed and can forewarn of possible operation problems. The problems may be associated with inadequacies in support structures and machine deterioration, etc.

Vibration measurements may be required for a variety of reasons of which the following are the most important:

- a) design/development evaluations;
- b) *in situ* testing;
- c) as information for a condition-monitoring or machinery health programme (ISO 14694 and Annex C gives recommended measuring positions for machinery health measurement);
- d) to inform the designer of associated supporting structures, foundations, ducting systems, etc., of the residual vibration which will be transmitted by the fan into the associated structure;
- e) as a quality assessment at the final inspection stage;
- f) to be sure of acceptability of specific dynamic loading.

All the information which can be obtained from tests conducted in accordance with this International Standard (see Clause 10) is neither necessary nor appropriate for quality-grading purposes. Reference should be made to ISO 14694 for this purpose. Vibration as a consequence of unbalance should be measured at the fan bearings and, in this connection, the recommendations given in ISO 1940-1 should be followed.

Whilst an open inlet/open outlet test may be useful as a quality guide, this International Standard recognizes that the vibration of a fan will be dependent upon the specified aerodynamic duty, which determines the rotational speed and position on the fan characteristic curve.

Although alternative standards exist which deal with the vibration of machines generally (e.g. ISO 10816), they presently have limitations because of their universal nature when considering a specific family of machines such as fans.

This International Standard describes the methods of measurement which will give consistent results and which may be used as a basis for comparison between products. The amount of information which needs to be presented and the preferred units are given in ISO 14694. Such information is dependent on the purpose for which the test has been conducted, the type of fan, its application and its method of mounting in service.

Industrial fans — Method of measurement of fan vibration

1 Scope

This International Standard describes a method of measuring the vibrational characteristics of fans of all types, except those designed solely for air circulation, for example, ceiling fans and table fans. However, it is limited to fans of all types installed with a power of less than 300 kW. For fans of greater power than this, the methods described in ISO 10816-1 and the applicable limits given in ISO 10816-3 may be used. This International Standard gives a general method only and does not give criteria for interpretation of data (see ISO 14694).

This International Standard specifies the measurement of vibration that may be recorded as overall root-mean-square r.m.s. velocity, acceleration or displacement, or in terms of a frequency spectrum, within the appropriate frequency range. Methods of testing when suspended on elastic ropes or when installed on resilient mountings are included.

It is recognized that the oscillatory forces at mounting points can be a useful measurement for analysing the effects on support structures; but such measurements are outside the scope of this International Standard.

Annexes are given for information. Annex B gives secondary measurement methods which, whilst not recommended for accurate measurements, may be used for assessing the balance of series-produced fans or for comparative site measurements.

2 Normative references

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 1940-1, *Mechanical vibration — Balance quality requirements of rigid rotors — Part 1: Specification and verification balance tolerances*

ISO 2041:1990, *Vibration and shock — Vocabulary*

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

ISO 5801:1997, *Industrial fans — Performance testing using standardised airways*

ISO 10816-3, *Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts — Part 3: Industrial machines with nominal power above 15 kW and nominal speeds between 120 r/min and 15 000 r/min when measured in situ*

ISO 14694:2003, *Industrial fans — Specification for balance quality and vibration levels*

IEC 60034-14, *Rotating electrical machines — Part 14: Mechanical vibration of certain machines with shaft heights 56 mm and higher — Measurement, evaluation and limits of vibration*

IEC 60651, *Sound level meters*

IEC 61260, *Electroacoustics — Octave-band and fractional- octave band filters*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 2041 and the following apply.

- 3.1 background vibration**
all sources of vibration independent of the source
- 3.2 duty point**
(aerodynamic duty) point on the fan performance curve at which a fan operates
- 3.3 fan performance curve**
(fan characteristic) plot of pressure rise developed by the fan against the airflow through a fan
- 3.4 radius of gyration**
measure of the distribution of mass about a chosen axis, given as the square root of the moment of inertia about that axis divided by the mass
- 3.5 resilient mount**
mount with elastic characteristics, and measurable deflection, but no permanent deformation under normal load conditions

4 Symbols and units

For the purposes of this International Standard, the following symbols and units apply.

Symbol	Term	Unit
a	Instantaneous vibration acceleration	m/s^2
a_0	Reference vibration acceleration	m/s^2
A	Vibration acceleration amplitude of peak	m/s^2
A_{dB}	r.m.s. vibration acceleration level above a reference of 10^{-6} m/s^2	dB
	$A_{\text{dB}} = 20 \log_{10} \left(\frac{A_{\text{r.m.s.}}}{10^{-6}} \right)$	
$A_{\text{r.m.s.}}$	r.m.s. vibration-acceleration amplitude	m/s^2
d	Instantaneous vibration displacement	μm , mm or m
D	Vibration displacement amplitude of peak	μm , mm or m

Symbol	Term	Unit
f	Frequency = $\omega/2\pi$	Hz
f_H	Frequency of sway mode of vibration	Hz
f_R	Frequency of rotational mode of vibration in zx plane	Hz
f_T	Frequency of torsion/yaw mode of vibration in xy plane	Hz
f_V	Frequency of vertical mode of vibration	Hz
f_1	Frequency of coupled rocking mode where sway is dominant	Hz
f_2	Frequency of coupled rocking mode where rotation is dominant	Hz
I_R	Moment of inertia of system about y -axis through centre of gravity of system	$\text{kg}\cdot\text{m}^2$
I_T	Moment of inertia of system about z -axis through centre of gravity of system located at X_G, Y_G, Z_G	$\text{kg}\cdot\text{m}^2$
$I_{zz,1}$	Moment of inertia of fan about z -axis through centre of gravity of fan located at x_1, y_1, z_1	$\text{kg}\cdot\text{m}^2$
$I_{zz,2}$	Moment of inertia of motor about z -axis through centre of gravity of motor located at x_2, y_2, z_2	$\text{kg}\cdot\text{m}^2$
$I_{zz,3}$	Moment of inertia of base about z -axis through centre of gravity of base located at x_3, y_3, z_3	$\text{kg}\cdot\text{m}^2$
$I_{xx,1}$	Moment of inertia of fan about x -axis through centre of gravity of fan located at x_1, y_1, z_1	$\text{kg}\cdot\text{m}^2$
$I_{xx,2}$	Moment of inertia of motor about x -axis through centre of gravity of motor located at x_2, y_2, z_2	$\text{kg}\cdot\text{m}^2$
$I_{xx,3}$	Moment of inertia of base about x -axis through centre of gravity of base located at x_3, y_3, z_3	$\text{kg}\cdot\text{m}^2$
$I_{yy,1}$	Moment of inertia of fan about y -axis through centre of gravity of fan located at x_1, y_1, z_1	$\text{kg}\cdot\text{m}^2$
$I_{yy,2}$	Moment of inertia of motor about y -axis through centre of gravity of motor located at x_2, y_2, z_2	$\text{kg}\cdot\text{m}^2$
$I_{yy,3}$	Moment of inertia of base about y -axis through centre of gravity of base located at x_3, y_3, z_3	$\text{kg}\cdot\text{m}^2$
k_H	Horizontal stiffness of resilient mount	N/m
k_V	Vertical stiffness of resilient mount	N/m
L_a	Vibratory acceleration level	dB
L_v	Vibratory velocity level	dB
m	Total mass of assembly	kg

Symbol	Term	Unit
m_1	Mass of fan	kg
m_2	Mass of motor	kg
m_3	Mass of base	kg
r	Radius of gyration	m
t	Time	s
T	Period of vibration	s
V	Instantaneous vibration velocity	mm/s or m/s
v_0	Reference vibration velocity	mm/s or m/s
V	Vibration velocity amplitude of peak	mm/s or m/s
V_{dB}	r.m.s. vibration velocity above a reference level of 10^{-9} m/s	dB
	$V_{dB} = 20 \log_{10} \left(\frac{V_{r.m.s.}}{10^{-9}} \right)$	
$V_{r.m.s.}$	Overall root-mean-square velocity	mm/s or m/s
x, y, z	Cartesian coordinates	m
\bar{z}	Effective vertical offset of resilient mounts	m
X_G, Y_G, Z_G	Positions of centre of gravity of fans assembly relative to arbitrary origin	m
$\delta_{1,2,n}$	Individual deflections of resilient mounts	m
Σ	Summation	—
ω	Angular frequency = $2\pi f$	rad·s ⁻¹

5 Mounting of test rig

5.1 General

Fans shall be mounted by one of the methods described in 5.2, 5.3 or 5.4, as appropriate.

Resilient mountings shall be chosen to give a uniform static deflection within $\begin{matrix} +20 \\ -35 \end{matrix}$ % of their nominal deflection. Fans which are resiliently mounted in normal applications shall be tested using the same type and number of mounts as used in that application. The support arrangement used, or existing at the time of the test, shall be described in the test report [see Clause 11 e)].

NOTE 1 The fan support arrangement may significantly affect the vibration levels measured on the fan structure. Mounting arrangements may be generally classified as stiff or resilient. In order to minimize the influence of the support condition on measured vibration levels, so that results from different tests may be readily compared, a resilient or soft-mounting arrangement for the fan is preferred. There is a risk of high-amplitude vibration when mounting stiffness results in the natural frequency f_n being near the frequency equivalent to the operating speed. For the purposes of this International Standard, a stiff mounting is considered to have a natural frequency f_n greater than 1,5 times the operating frequency. A resilient mounting is considered to have a natural frequency f_n less than 0,25 times the operating frequency.

NOTE 2 Tests using the rubber-rope method of suspension are recommended for design and development evaluation and/or as a grading exercise.

The test-rig arrangement will generally be determined by the nature and location of the test. For design/development evolution, sophisticated purpose-designed rigs are likely to be available. However, any arrangement of the test rig which does not readily permit the aerodynamic duty of the fan to be controlled, or in which the duty is otherwise unknown, may be unsuitable for tests in accordance with this specification where the vibration produced by the fan varies with duty. Wherever practicable, a standardized airway, as defined in ISO 5801, should be used.

NOTE 3 Two typical arrangements of fan-vibration test rigs are shown in Figures 1 and 2.

5.2 Resilient base mounting of fan

The natural frequencies of the fan on its mountings in the six possible degrees of freedom (see Figure 3) shall not be greater than 0,25 times of the slowest operational rotation frequency of the fan under test.

NOTE 1 A guide to calculating the natural frequencies is given in Annex A.

The mass of any additional attachments shall not exceed one-tenth of the normal mass of the unit under test, to reduce the influence of the mass and the moments of inertia of these parts on the vibration level.

NOTE 2 Where a fan is not normally equipped with resilient mountings, it may be necessary to fit special mounting brackets and isolators or other attachments to the fan in order to accommodate them.

No major flexural resonances of the support structure shall coincide with either the rotational frequencies of the fan under test or the rigid-body natural frequencies of the support arrangement, except when the fan and support are being investigated together.

NOTE 3 Fan reaction torque of the fan may be significant on a resiliently mounted arrangement (see A.3.2 for advice on inertia bases).

5.3 Mounting of fans with resilient elastic rope

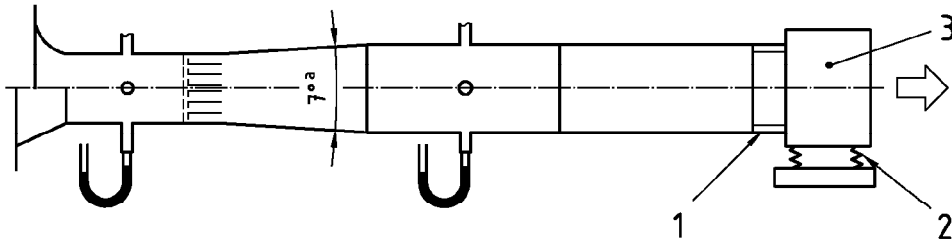
When testing fans with an integral motor which are to be supplied as single units without a mounting frame, these fans shall be mounted in an elastic-rope test rig (examples are shown in Figure 4).

NOTE 1 Low-frequency mounting permits consistent readings to be taken for comparison purposes under free-air conditions only.

When a duct connection is required to obtain the vibration characteristics at the duty point, a suitable flexible connection shall be used with horizontal elastic restraint if necessary.

The fan shall be supported by nylon slings attached to an assembly of braided rubber cords of a suitable number, length and diameter, arranged symmetrically with respect to the centre of gravity, to provide and overall deflection in the range 200 mm to 400 mm under the fan mass.

NOTE 2 This soft mounting may be conveniently suspended from an "A" frame structure.



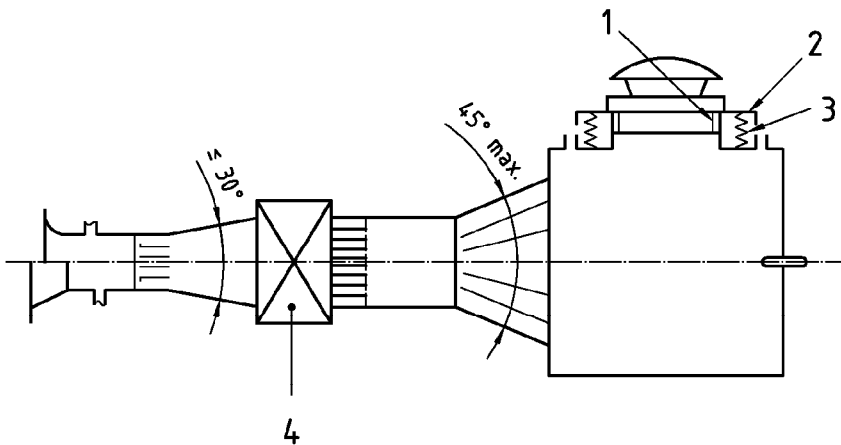
Key

- 1 flexible connection (diagrammatic)
- 2 resilient mounts
- 3 fan

NOTE This figure shows an arrangement suitable for a centrifugal axial fan or an in-line fan loaded by resistance screens on the inlet side. In some cases, it may be preferable to use a damper on the air-outlet side. This figure is generally in accordance with Figure 74 a) of ISO 5801:1997 with the addition of a flexible connection and resilient mounts between the fan and ducting.

^a Angle at the top.

Figure 1 — Example of resilient mounting arrangement of a fan-vibration test rig

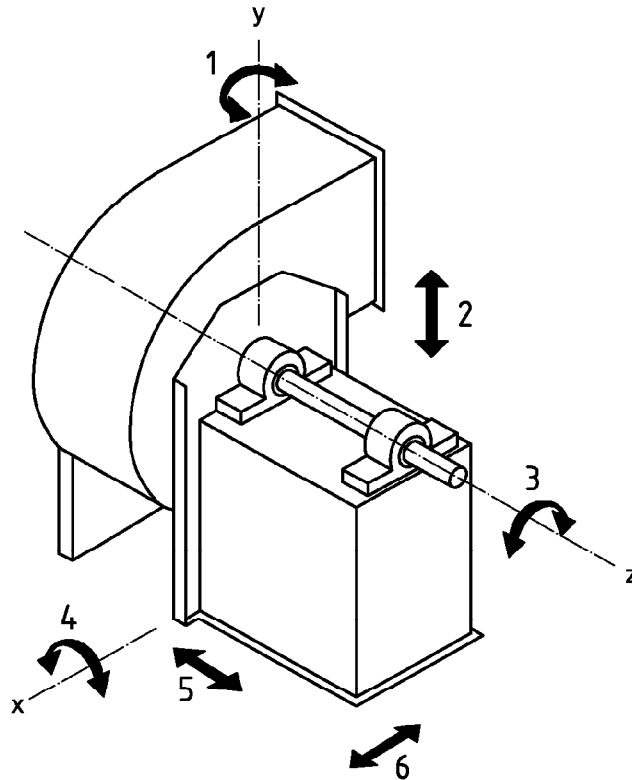


Key

- 1 flexible connection (diagrammatic) for tests
- 2 resilient mounted curb
- 3 resilient mounts
- 4 booster fan

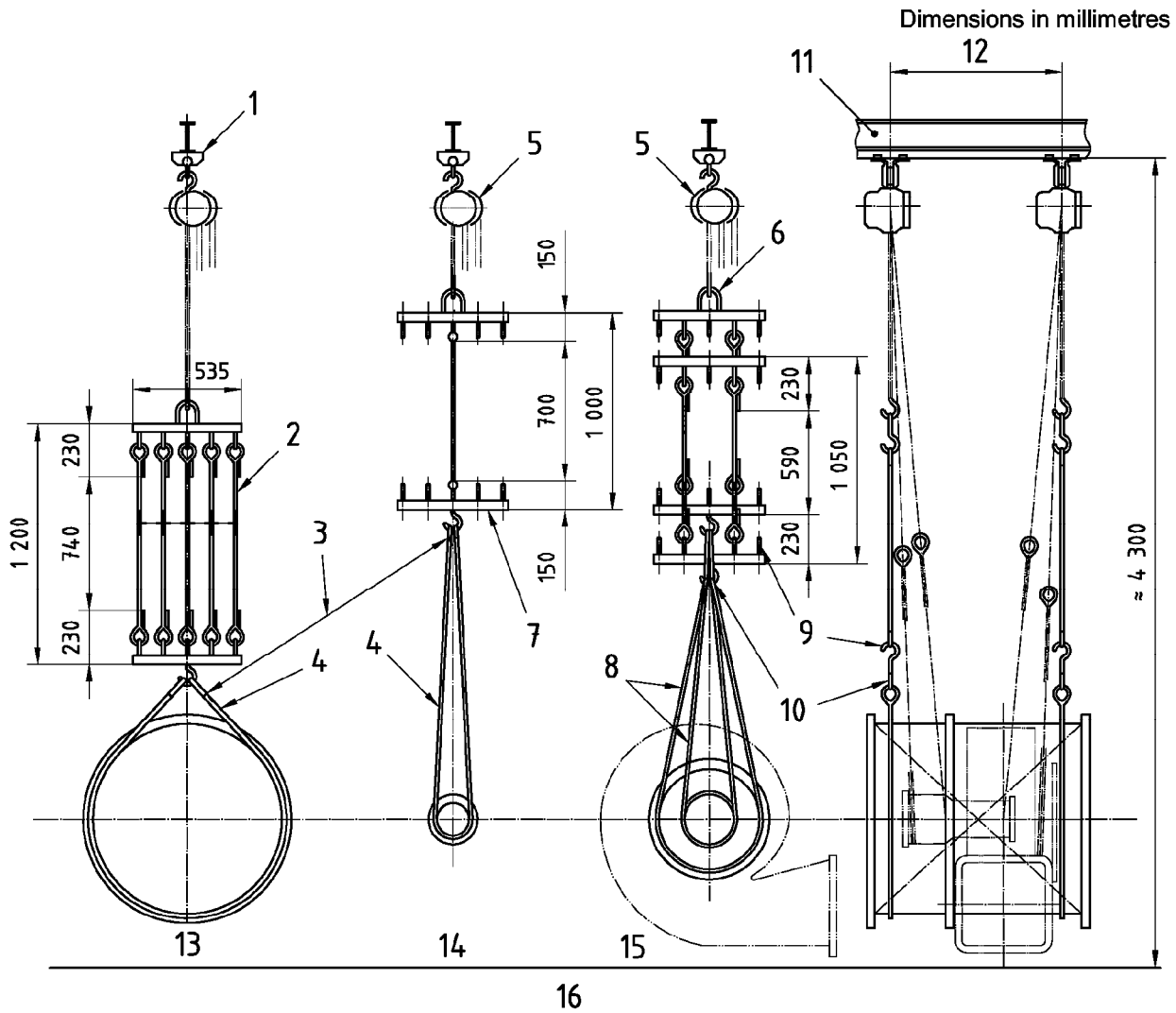
NOTE This figure shows a roof extraction fan arranged for vibration testing purposes. The fan has been mounted on a curb which is resiliently mounted. The mass of the curb and natural frequency of the mounted test fan should be in accordance with 5.2. This figure is generally in accordance with that given in Figure B.2 of ISO 5801:1997, but with the addition of a flexible connection and resilient mounts between the fan and the outlet cowl.

Figure 2 — Example of arrangement for testing roof extraction fan

**Key**

- 1 yawing couple (rotation around y axis)
- 2 vertical inertia force (y axis)
- 3 rocking couple (rotation around z axis)
- 4 pitching couple (rotation around x axis)
- 5 longitudinal (axial) inertia force (z axis)
- 6 horizontal (transverse) inertia force (x axis)

Figure 3 — Degrees of freedom



Key

- 1 shackle clamp or similar
- 2 heavy braided rubber cords
- 3 standard-pattern thimbles
- 4 nylon ropes for slinging fans
- 5 lifting tackle
- 6 round bars, $\varnothing 15$
- 7 flat bars, 38×19
- 8 nylon ropes required between centres of thimbles for slinging fans
- 9 round bars, $\varnothing 13$
- 10 round bars, $\varnothing 15$
- 11 RSJ
- 12 900 centres
- 13 axial fan, mass 600 kg
- 14 axial fan, mass 7 kg
- 15 centrifugal fan, mass 265 kg
- 16 ground level

NOTE For use with fans operating above 10 Hz (600 r/min).

Figure 4 — Examples of fans mounted with resilient elastic rope

5.4 Stiff mounting of fans

Prior to undertaking vibration measurements, the mounting of the fan shall be inspected to ensure that the fixing of foundation bolts are sound and the bolts are properly tightened.

NOTE Fans that either cannot readily be resiliently mounted or that are installed on site on a relatively stiff foundation, for example, terminal flange mounted or foot mounted, require special consideration. It may be difficult to comply with 5.5 in such circumstances. A valid comparison of the vibration magnitude for such fans with others of the same type can only be made if the foundation arrangements have similar dynamic characteristics. This implies that the dynamic stiffness, damping and effective mass of the structural mounting and, where applicable, the supporting soil, should be similar for the cases being compared.

5.5 Flexible connections

The stiffness of a duct connection or any service connection shall be less than 10 % of the dynamic stiffness of the mounting. The maximum natural frequency, as given in 5.2, shall be met with the flexible connections in position and the fan operating at the maximum pressure at which it is to be tested. The fan and duct shall be aligned and levelled before testing, so that the flexible connections are not strained. Where noise-barrier-type flexible connections, which are relatively stiff, are installed *in situ*, details shall be appended to the test report [see Clause 11 e)].

The fan shall be run in its normal operating attitude.

All connections to the fan under test should be sufficiently flexible, so that the overall dynamic stiffness of the mounting system is not significantly increased.

For the grading of general purpose fans, they may be run in an "open" condition without ducting. It should be borne in mind that vibration levels may be different than those under operating conditions. The test conditions should be clearly stated (see Clause 11).

6 Measuring equipment

6.1 General

Uncertainties, which are a function of instrumentation and the fan application, should normally be within 25 % or 2 dB of the reading.

NOTE Transducer attachment is a critical factor which may add substantially to the uncertainty, unless carried out strictly in accordance with the manufacturer's recommendations. This is especially true at frequencies above 5 kHz.

Additional uncertainties can arise due to the use of probes, magnets and other transducer attachments not covered in 7.2, which are difficult to ascertain. Uncertainties caused by the setting-up of the fan under running conditions and the mounting of the fan are not specified in this International Standard (see ISO 14694).

6.2 Calibration

Whichever type of vibration-measuring instrument is used, it shall have a valid calibration certificate less than 12 months old and calibration shall be traceable to a national standard.

Calibration certificates should include response over the frequency band in which the transducer is used, as given in 10.2.

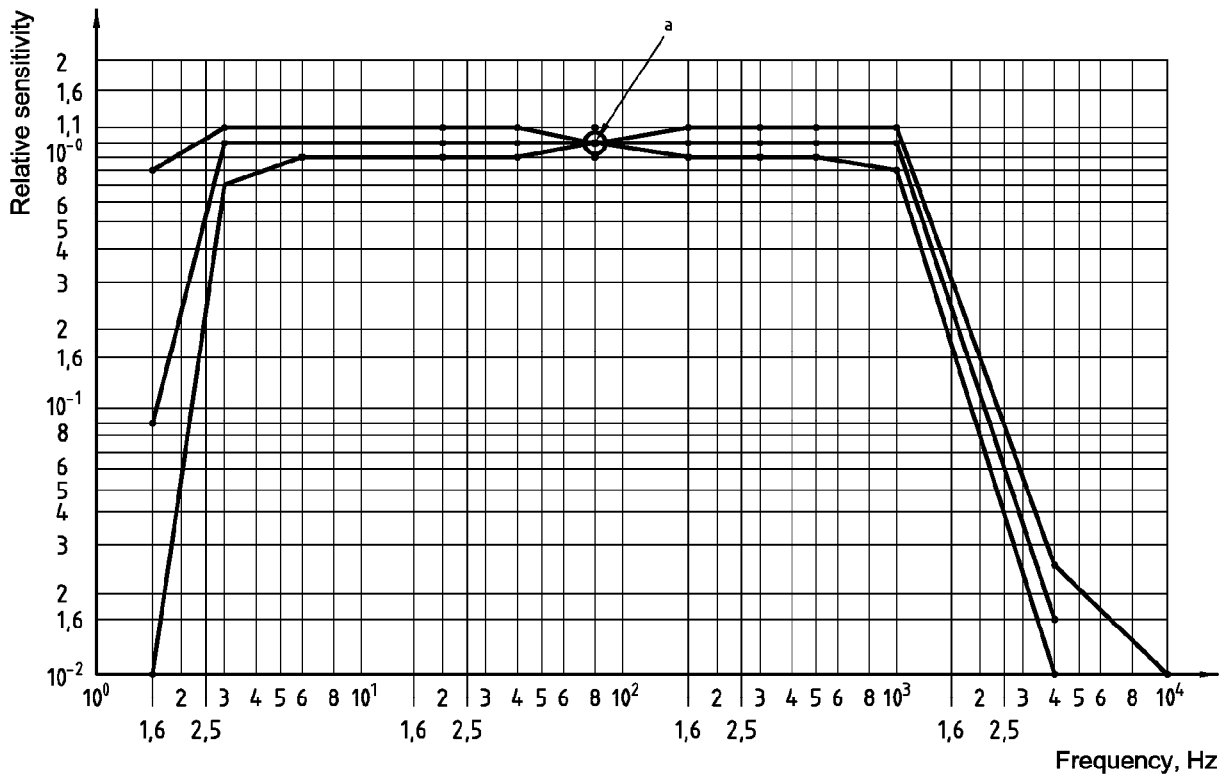
6.3 Instrument system

6.3.1 Instruments measuring r.m.s. velocity

For a fan running at rotational speeds of 10 Hz to 200 Hz, the instrument system shall comply with ISO 2954.

For a fan running at lower speeds down to 3 Hz, the instrument system shall comply with ISO 2954, but the low-frequency response shall be in accordance with Figure 5.

The reference frequency shall be either 80 Hz, as specified in ISO 2954, or 160 Hz.



a Reference point of calibration.

Figure 5 — Nominal value of relative sensitivity and limits of permissible deviation as a function of frequency

6.3.2 Instruments measuring r.m.s. acceleration

For measuring r.m.s. acceleration, the instruments shall have the same frequency responses as given in 6.3.1, but all tolerances shall be in accordance with Figure 5.

NOTE True r.m.s. implies an error of less than 5 % or 0,5 dB for signals with crest factors of 3.

6.4 Transducers

To provide a flat response over the full frequency range and to avoid transducer resonance at high frequencies, the transducer shall be a small accelerometer, preferably less than 30 g in mass, mounted as described in Clause 7.

NOTE 1 For heavier fans, it may be possible to increase the mass of the accelerometer in relation to the mass of the fan.

NOTE 2 The frequency range of interest will determine the selection of the transducer. If the maximum frequency of interest is significantly less than 10 kHz, slightly larger accelerometers may be required (typically less than 60 g in mass).

Care should be taken to reduce the bandwidth of the associated instrumentation to filter out resonance which would otherwise be included in the overall measurement.

6.5 Piezoelectric accelerometers

Piezoelectric accelerometers shall have high-input impedance preamplifiers, for example, charge-signal conditioning amplifiers, with a flat frequency response from 3 Hz to 10 kHz within ± 1 dB.

NOTE It is recommended that the preamplifier be equipped with a low pass filter with a slope of 12 dB per octave or greater, with attenuation of 6 dB or more at 20 kHz.

6.6 Preamplifiers

The integrating network (filter) of the preamplifier shall be correct to ± 1 dB at 5 Hz and within $\pm 0,5$ dB from 10 Hz upwards.

NOTE For the measurement of velocity, integration of the acceleration signal may be performed in the preamplifier.

Preamplifiers may be built into accelerometers, but the requirements given in 6.4 for size, mass and resonance frequency shall be met.

6.7 Analysers

6.7.1 General

Analysers shall perform in real time to at least 1 kHz or, when analysers are used outside real time, the time averaging shall be appropriately increased, i.e. multiplied by the ratio of frequency-analysis range used over the real-time range.

6.7.2 Narrow-band analysis

The narrow-band method is recommended for identifying harmonic and side-band families of vibration components, such as rotational speed, because of the constant absolute bandwidth. When constant-bandwidth analysers are used for narrow-band analysis, the effective analysis bandwidth for fans with a rotational speed above 10 r/s shall be 0,2 Hz, or less for machines above 50 r/s below 1 kHz. For machines lower than 50 r/s and higher than 10 r/s, 0,5 Hz bandwidth is sufficient below 1 kHz. In the frequency range between 1 kHz and 10 kHz, 10 Hz bandwidth is sufficient.

For fans running at speeds lower than 10 r/s and higher than 2 r/s, for constant bandwidth analysis below 100 Hz, the bandwidth shall be 0,5 Hz or less, and above 400 Hz, 2 Hz or less. A Hanning window shall be used in the case of FFT analysers for the best frequency accuracy. For these devices and the best amplitude accuracy, a flat-top window shall be used.

6.7.3 One-third octave analysis

For one-third octave analysis, one-third octave filters or analysers complying with IEC 61260 shall be used. The r.m.s. detector performance and its linearity shall be that required for type 1 sound level meters complying with IEC 60651.

NOTE When using a fast Fourier transform (FFT) analyser for the purpose of obtaining one-third octave bands, a substantial increase in averaging time (number of spectra) is normally required.

When such an analysis is used, then the user has to be aware of the effect, of relative bandwidth. Families of harmonics and sidebands of vibration components do not have an expected constant frequency difference (for harmonic and sideband cursor functions).

6.7.4 Less than one-third octave analysis

Fractional octave filters or analysers narrower than one-third octave shall comply with IEC 61260.

6.8 Indicators

Indicators, whether analogue or digital, shall indicate the r.m.s. value of the signal when measuring the velocity or acceleration and, when measuring displacement, shall indicate the peak-to-peak value.

6.9 Outputs

Outputs, if provided, shall have low output impedance, preferably below 50 Ω .

7 Transducer attachment

7.1 General

7.1.1 The resonant response of any transducer mounting bracket shall be outside the frequency range of interest.

The method of attachment chosen shall guarantee the integrity of measurement over the frequency range of interest. Any form of mounting (bracket, probe, adaptor, etc.) used between the transducers and the measurement surface may modify the response of the transducer and lead to an erroneous result. For further information, see ISO 5348.

NOTE The recommended method, and by far the most reliable mounting technique, is to fasten the transducer directly to the measured surface by stud mounting. The stud may be fastened to the surface of the machine by drilling and tapping (see also 7.2). For vibration severity and balancing, see also Annex B.

7.1.2 The mounting surface shall be either

- a) flat, with the entire face of the transducer in contact with the machine surface or, if this proves to be impractical,
- b) a metal block, contoured to suit the machine surface, which shall be secured directly to the machine by either bolting, welding or brazing, etc. Then the transducer shall be attached to the mounting block, as described in 7.1.1.

It is recommended that a thin coating of silicone grease on the threads and mating surfaces be used to improve coupling where sensitivity and accuracy are of utmost importance.

7.1.3 For triaxial measurements, the accelerometer may be mounted on a welded block, but its size and weight shall not be greater than that of the block.

NOTE 1 It is possible to make use of manufactured triaxial units comprising three orthogonally mounted transducers.

NOTE 2 In general, the following factors are also considered when the transducer attachment is being made:

- a) vibration frequency limitations of transducer mounting;
- b) electrical grounding problems;
- c) hazardous products (explosives).

NOTE 3 In some instances, a combination of transducer and instrument grounding results in "ground loop" pick-up of mains-frequency-related voltages. This may normally be spotted by the presence of high levels in the 50 Hz region, whose frequency does not change with changes in operational speed. This problem may be reduced by isolating the transducer from the metal housing on which it is located.

Electromagnetic pick-up may also occur in transducers and cables caused by stray fields associated with heavy current flows, sparking, contact breaking, etc. Relocation of the transducer mounting or cable may help.

7.2 Attachments

Transducers shall be attached by one of the following methods:

- a) screwed directly into a tapped hole in the fan which has a machined surface;
- b) screwed into a block welded or brazed directly to the fan;
- c) attached to the machined surface of a steel block by means of a fastener, the block in turn being secured to the fan by a bolt;
- d) screwed into a steel block which is attached to the fan by an adhesive (see Figure 6);
- e) screwed onto a stud bonded to the surface;
- f) bonded directly to the surface of lightweight fans.

NOTE 1 The methods of attachment shown are given in order of preference.

NOTE 2 When measurements are being taken for the quality grading of production fans, permanent transducer attachments may not be desirable. In such cases, hand-held or magnetically held transducers may be used, provided that readings are limited to the maximum frequencies given in Table B.1.

8 Selection of measuring positions

8.1 General

Measurement positions shall be selected at the following locations.

- a) locations that give a true representation of the equipment vibration characteristics with due regard to the positions at which the fan is supported;
- b) locations that
 - 1) indicate vibration transmitted to connecting ducting, i.e. fan inlet and/or outlet,
 - 2) indicate vibration transmitted to any supporting structures or foundations, i.e. fan mounting feet, and
 - 3) are at points which indicate the fan health (see ISO 14694 and Annex C), i.e. fan/motor shaft-bearing housings;
- c) locations that will be accessible after installation.

Locations that are unduly flexible should be avoided.

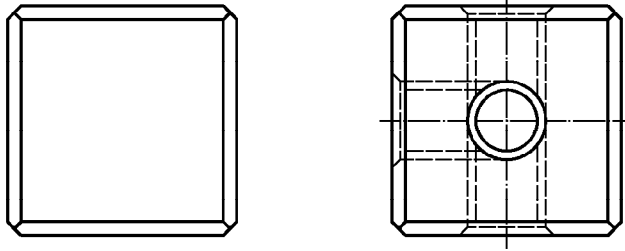
Each measuring position shall permit the measurement of vibration in three mutually perpendicular axes at a single point, e.g. longitudinally (axially), transversally (horizontally), and vertically.

NOTE 1 This may be facilitated by the creation of machined spot faces on the equipment at measuring positions which will allow

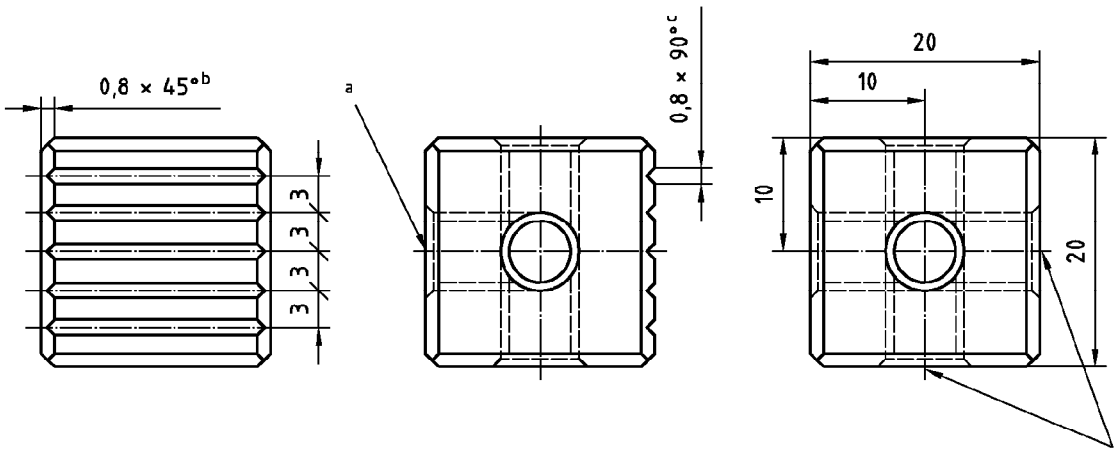
- i) the transducers to be directly attached in the three principal directions, and
- ii) a triaxial block to be attached to which the transducers can be fitted.

NOTE 2 Recommended measuring positions are shown in Figures 7 and 8.

Linear dimensions in millimetres



a) Base for thin adhesive



b) Base for thick adhesive

NOTE 1 When mounting a velocity transducer, a larger block will be required.

NOTE 2 When attaching a block to a fan by adhesive, care should be taken to ensure that the film is as thin as possible and the block is set true, (i.e. vertically and horizontally) to within 2° . Adequate time should be allowed for the adhesive to cure. Typical adhesives are

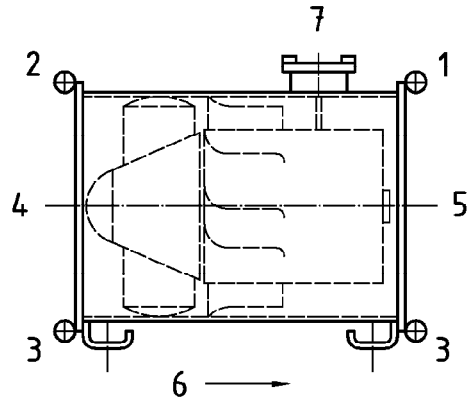
- epoxy resin;
- plastic padding (hard);
- cyanoacrylate;
- dimethacrylate ethylurethane diol;
- wax.

a Drill and tap to suit transducer mounting.

b $0,8 = 45^\circ$ on all edges.

c 5 grooves.

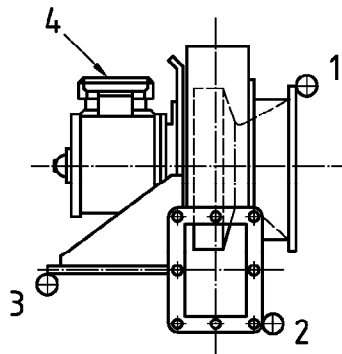
Figure 6 — Typical mounting block for transducer



Key

- 1 casing-flange motor end
- 2 casing-flange impeller end
- 3 support bracket (if applicable)
- 4 inlet
- 5 outlet
- 6 airflow
- 7 motor terminal box

Figure 7 — Recommended measuring positions for axial-flow fans



Key

- 1 flat inlet flange
- 2 fan outlet flange
- 3 support bracket
- 4 motor terminal box

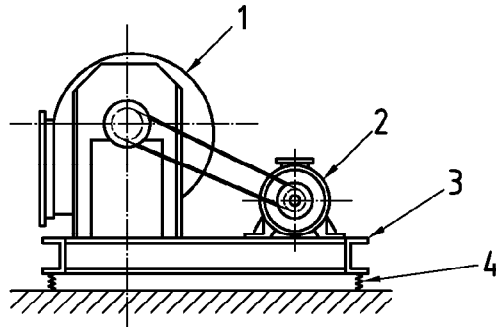
Figure 8 — Recommended measuring positions for centrifugal fans

8.2 Mounting on combination base frame

When a fan is indirectly driven by an electric motor or other prime mover through pulleys and belts or chains and the complete assembly shall be mounted on a combination base frame (see Figure 9), vibration readings shall, additionally, be taken at the position of the base support immediately adjacent to each resilient mount.

8.3 Mounting of transducers for other fans

Apart from the types of fan detailed in 8.1 and 8.2, there is a number of other arrangements which are detailed Figures 1 to 4 inclusive of ISO 14694:2003. The transducer positions shown in these figures are especially relevant for the determination of balance quality.



Key

- 1 centrifugal fan
- 2 motor
- 3 base support
- 4 resilient mounts

Figure 9 — Example showing an indirectly driven centrifugal fan assembly on resilient mounts

9 Test environment and operating data

The test operating data, i.e. airflow rate, fan pressure, installation types, fan speed, ambient temperature and regulation of airflow and its measurement, shall be in accordance with ISO 5801.

Background vibration levels shall be measured at the selected measuring positions (see Clause 8) with the fan inoperative. They shall be either not more than 30 % or 10 dB lower than the absolute overall level recorded when the fan is on test.

10 Procedure

10.1 General

Before any measurements are undertaken, run the fan for a sufficient length of time to ensure that the fan motor has reached its working temperature and that stable operating conditions have been achieved at normal operating speed.

In the case of variable-speed fans, carry out a preliminary sweep over the full speed range to identify the variation magnitude. Make detailed measurements at the speed corresponding to the maximum vibration magnitude previously obtained. Throughout the tests, supply the fan with its normal electrical supply in terms of voltage, frequency and phase.

Where the test rig (see the last paragraph of 5.1) has a standardized airway in accordance with ISO 5801, make vibration measurements at a minimum of three points on the fan performance curve within its operating range. These shall be undertaken at the following points:

- a) corresponding to the maximum-volume airflow rate that can be obtained from the test rig;
- b) the duty point (or that corresponding to maximum efficiency);
- c) the maximum operating pressure.

For general *in situ* testing, make measurements at points corresponding to the minimum and maximum loading levels that may exist in operational service and also include the rated output of the fan.

For either standardized airway tests or *in situ* tests, vary the fan operating conditions continuously from maximum operating volume to maximum operating pressure and observe the fan to determine if there are any points on the operating curve at which vibrations are particularly severe. If any such points are found, include these among the measurement points on the performance curve.

For measurements concerned with condition monitoring or machinery health programmes, the running condition will normally be that naturally prevailing at the time of measurement. Wherever possible, details of the operating conditions should be recorded along with the measurements themselves.

NOTE For the "test rig without standardized airway" configuration, the fan performance operating point(s) may be difficult to standardize and are therefore agreed between the purchaser and manufacturer.

10.2 Measurement parameters

Make broad-band measurements of vibration magnitude using one or more of the following parameters:

- a) vibration velocity, expressed in millimetres per second r.m.s., over the frequency range 10 Hz to 1 kHz (see ISO 10816-3, except for fans running below 10 r/s when the lower limit shall be 3 Hz);
- b) vibration acceleration, expressed in metres per second squared r.m.s., over the frequency range 10 Hz to 10 kHz;
- c) vibration displacement expressed in micrometers, peak-to-peak, over the frequency range 3 Hz to 1 kHz.

NOTE 1 The vibration measurements can be adequately characterized in terms of magnitude and frequency (or frequency range).

NOTE 2 In certain cases, velocity and acceleration may be expressed in logarithmic terms in decibels above a reference level of 10^{-9} m/s and 10^{-6} m/s² respectively (see ISO 1683 and Annex D).

NOTE 3 The preferred descriptor for overall measurement is vibration velocity but acceleration and displacement may be more appropriate in cases where the dominant frequencies are high or low respectively.

The relationship between these three vibration magnitudes are shown for a harmonic-free sinusoidal condition in Annex E along with useful relationships for broad-band vibration consisting of a series of known harmonics.

In order to avoid confusion when interpreting results, it is important at all times to identify clearly the units which are being used for the vibration-magnitude measurements.

In cases where it is necessary to supplement the broad-band measurement (see above) with frequency analysis of the vibration, carry out frequency analysis and report it over the same frequency range as described in this subclause.

10.3 Frequency analysis

Where spectral information is required, perform frequency analysis by means of a constant bandwidth FFT analyser giving an effective bandwidth of not more than 1 Hz below 100 Hz, 4 Hz below 1 kHz and 40 Hz from 1 kHz upwards, or by means of a constant percentage analyser, with bands not wider than one-third octave.

If discrete frequency measurements are required (using the instrumentation described in Clause 6), take them at one position only, corresponding to that location at which the overall r.m.s. level was a maximum.

NOTE Frequency analysis need only be carried out and reported for the transducer location where the highest overall value was measured.

When performing a spectral analysis, averaging should be appropriate for the characteristics of the signal measured.

When using a swept filter analyser the sweep rate, averaging time and pen writing speed should be adjusted in accordance with the manufacturer's instructions.

11 Presentation of results

Recorded measurement results shall be accompanied by essential data for the machine and the instruments used.

The following information shall be given in whole or in part (see ISO 14694):

- a) the fan manufacturer, the type and definition of unit being tested, including all distinguishing features such as capacity, power, voltage, actual mounting orientation, serial number, number of blades on fan impeller, first lateral shaft critical speed, design balance quality (see ISO 1940-1);
- b) the fan driver, the type, duty, installed power, speed, serial number, vibration-severity quality (see IEC 60034-14, where applicable);
- c) the total mass of the unit, and mass of the rotating assembly;
- d) the date of the test and the site at which the test was carried out, and the names of personnel involved in the testing;
- e) the assembly, resilient mounting, the type, location, deflection at each mounting, vertical natural frequency either measured or calculated; structural mounting, description of mounting arrangement and soil (see Clause 5);
- f) a description of the test arrangement, preferably including a dimensioned sketch, including the positions of the resilient mounts (see Clause 5);
- g) the measuring equipment including the make, type and serial number (see Clause 6);
- h) a description and the results of calibration checks on measuring instruments (see 6.2);
- i) the method of attachment of the measuring transducers (see Clause 7);
- j) the test operating data (see Clause 9);
- k) the position, axes and measurement data from each individual measurement point and upper and lower frequency limits of the overall measurement (see Clause 8);
- l) a description and measurements of the test environment, i.e. background vibration, ambient temperature, etc. (see Clause 9);
- m) a reference to this International Standard, and a statement of any deviations from the method described;
- n) maximum allowable vibration levels for the test;
- o) statement that the unit tested passed, or failed, the test.

Annex A (informative)

Guide to calculating resilient mount positions and rigid-body natural frequencies

A.1 General

The preferred method for obtaining the six rigid-body natural frequencies associated with a fan and motor assembly supported on a base with four resilient mounts is by testing. In cases where this is not possible, the natural frequencies may be obtained by analysis. This annex outlines, by example, the analysis procedure and is intended for guidance only. Although simplifying assumptions are made in the analysis, the procedure provides sufficiently accurate values of the natural frequencies of the system for the purpose of selecting mounts (see Clause 8). Where there is a need for more than four separate mounts, the complexity of the analysis increases.

A.2 Example of a calculation for a centrifugal fan assembly

A.2.1 General

Consider the example of a resiliently-mounted centrifugal fan assembly as shown in Figure A.1. The centre of gravity mount locations and rigid-body natural frequencies of this assembly are calculated in A.2.2 and A.2.3. A similar analysis could also be applied to axial or mixed-flow fans.

A.2.2 Location of centre of gravity of the fan assembly

The position of:

m_1 relative to an arbitrary origin is x'_1, y'_1, z'_1

m_2 relative to an arbitrary origin is x'_2, y'_2, z'_2

m_3 relative to an arbitrary origin is x'_3, y'_3, z'_3

Therefore, the position of the centre of gravity of the fan assembly relative to this arbitrary origin is given by X_G, Y_G, Z_G where

$$X_G = \frac{(m_1 x'_1) + (m_2 x'_2) + (m_3 x'_3)}{\sum m} \quad (\text{A.1})$$

$$Y_G = \frac{(m_1 y'_1) + (m_2 y'_2) + (m_3 y'_3)}{\sum m} \quad (\text{A.2})$$

$$Z_G = \frac{(m_1 z'_1) + (m_2 z'_2) + (m_3 z'_3)}{\sum m} \quad (\text{A.3})$$

and the position of

m_1 relative to the centre of gravity is x_1, y_1, z_1

m_2 relative to the centre of gravity is x_2, y_2, z_2

m_3 relative to the centre of gravity is x_3, y_3, z_3

where

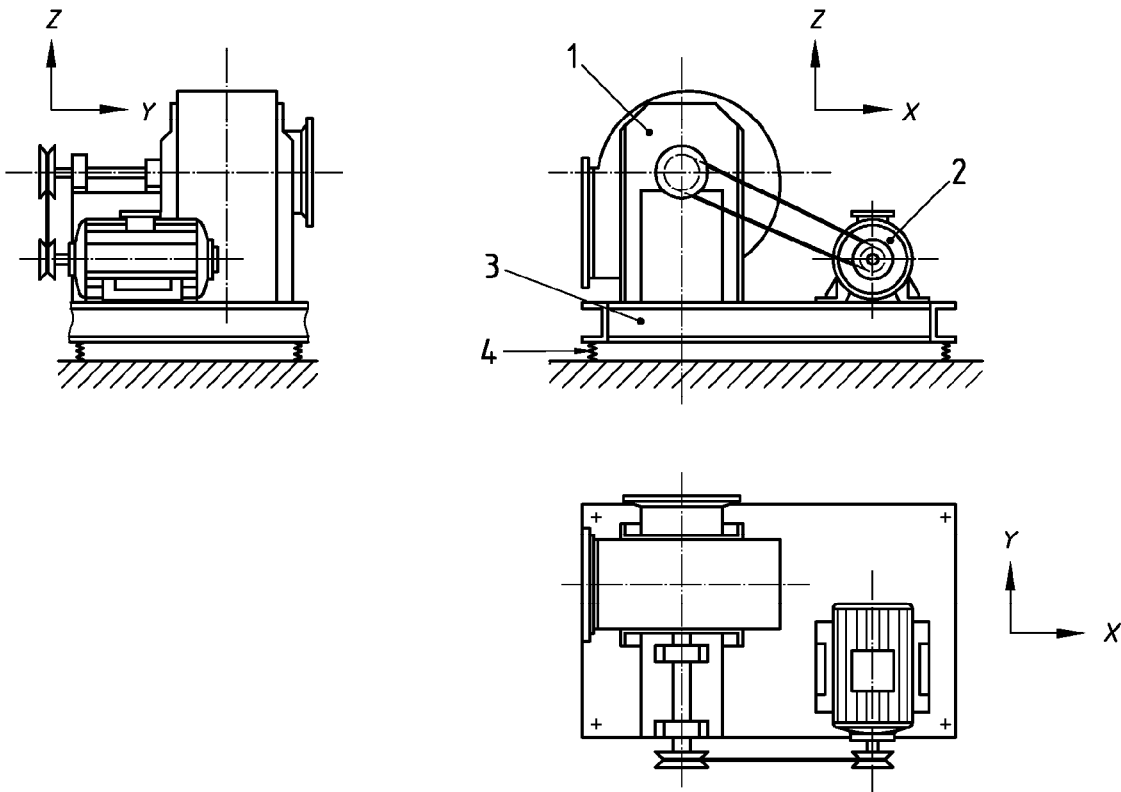
m is the total mass of assembly, in kilograms (kg);

m_1 is the mass of the fan, in kilograms (kg);

m_2 is the mass of the moto, in kilograms (kg);

m_3 is the mass of the base, in kilograms (kg);

$x_1 = x'_1 - X_G; y_1 = y'_1 - Y_G$, etc.



Key

- 1 fan, m_1
- 2 motor, m_2
- 3 base support, m_3
- 4 resilient mounting

Figure A.1 — General layout of a centrifugal fan assembly on resilient mountings

A.2.3 Positioning of resilient mounts

As described in Clause 5, resilient mounts are to be positioned such that each mount has the same nominal static deflection. Thus, if mount locations relative to the centre of gravity are given by (x, y, z) and the vertical and horizontal stiffnesses of each mount are represented by k_V and k_H , in newtons per metre, respectively mounts have to be positioned such that:

$$\sum_{n=1}^4 k_V \delta_n x = 0$$

$$\sum_{n=1}^4 k_H \delta_n y = 0$$

where δ_n is the individual deflection, in millimetres (mm), and

where mount deflections δ_1 to δ_4 are nominally equal to within tolerance limits of $\begin{matrix} +20 \\ -35 \end{matrix}$ % (see Clause 5).

NOTE Fan thrust may affect the nominal deflection under running conditions.

A.3 Calculation of rigid-body natural frequencies

A.3.1 General

In general, fan assemblies are designed such that the vertical height of the centre of gravity above the mounts is less than one-third of the smallest horizontal distance between mounts. If this condition is satisfied, it is sufficiently accurate for the purpose of 5.2 to calculate the rigid-body natural frequencies of the fan assembly assuming pure modes of vibration, i.e. that there is no coupling between translational and rotational motions. For cases where the one-third ratio is exceeded, coupled mode analysis is recommended (see A.3.2).

NOTE If a large degree of asymmetry exists, a more complex analysis is required, in which case cross-products of inertia and stiffness are included. In this case, it is advisable to consult a specialist.

A.3.2 Coupled mode analysis

If the vertical offset between the overall centre of gravity and the elastic centre of the mounting configuration is not negligible, two rocking modes will occur whose frequencies f_1 and f_2 , in hertz (Hz), are given by the following equations:

$$f_1 = C_1 f_H \tag{A.4}$$

$$f_2 = C_2 f_H \tag{A.5}$$

where

$$C_1 = \left[\frac{1}{2} \left((1+B) - \sqrt{(1-B)^2 + \left(\frac{4\bar{z}^2}{r^2} \right)} \right) \right]^{1/2} \tag{A.6}$$

$$C_2 = \left[\frac{1}{2} \left((1+B) + \sqrt{(1-B)^2 + \left(\frac{4\bar{z}^2}{r^2} \right)} \right) \right]^{1/2} \tag{A.7}$$

where

$$B = \frac{\sum (k_{Hz}^2 + k_V x^2)}{r^2 \sum k_H} \quad (\text{A.8})$$

$$r \text{ is the radius of gyration, in millimetres (mm)} = \left(\frac{I_R}{\sum m} \right)^{1/2} \quad (\text{A.9})$$

$$\bar{z} \text{ is the effective vertical offset of resilient mounts, in metres (m)} = \frac{\sum (k_{Hz})}{\sum k_H} \quad (\text{A.10})$$

I_R is the moment of inertia about the y -axis through the centre of gravity of the system, in kilograms per square metre ($\text{kg}\cdot\text{m}^2$).

In general, the effect of coupling is to increase the separation in frequency between the decoupled rocking and sway modes. Thus, the decoupled mode approach will tend to underpredict and overpredict, respectively, the coupled mode frequencies which result when the vertical offset is not negligible.

The vertical offset can be minimized by using an inertia base to support the fan and motor assembly in a vibration test. The mass of the inertia base should be at least equal to the combined mass of the fan and motor assembly, and preferably greater.

It is advantageous to select a plan size of the inertia base which is at least 15 % greater than the fan-base frame size. The fan/motor assembly should then be located on the base such that the overall centre of gravity fan/motor is as close as possible to the centre of gravity inertia base. This approach should allow the $\begin{matrix} +20 \\ -30 \end{matrix}$ % tolerance on each static deflection of the mounting to be achieved.

A.3.3 Calculation

The rigid body natural frequencies, assuming the following:

- a) uncoupled mode of vibration;
- b) the horizontal stiffness of each isolator is the same in all directions;
- c) all natural frequencies are undamped;

are calculated as follows.

- 1) Vertical mode. The frequency of the vertical mode of vibration, f_V , in hertz (Hz), is given by the following equation:

$$f_V = \frac{1}{2\pi} \left(\frac{\sum k_V}{\sum m} \right) \quad (\text{A.11})$$

where m is the total mass of the assembly, in kilograms (kg).

- 2) Horizontal sway mode. The frequency of sway mode of vibration, f_H , in hertz (Hz), given in the following equation is conservative in that it is always greater than the corresponding frequency, f_1 , obtained by coupled mode analysis (see A.3.2):

$$f_H = \frac{1}{2\pi} \left(\frac{\sum k_H}{\sum m} \right)^{1/2} \quad (\text{A.12})$$

- 3) Torsion/yaw mode. The frequency of torsion/yaw mode of vibration in the zy plane, f_T , in hertz (Hz) is given by the following equation:

$$f_T = \frac{1}{2\pi} \left(\frac{\sum k_H (x^2 + y^2)}{I_T} \right)^{1/2} \quad (\text{A.13})$$

where I_T is the moment of inertia of system about the z -axis through the centre of gravity, in kilograms per square metre ($\text{kg}\cdot\text{m}^2$), given by the following equation:

$$I_T = I_{zz,3} + m_1 (x_1^2 + y_1^2) + m_2 (x_2^2 + y_2^2) + m_3 (x_3^2 + y_3^2) \quad (\text{A.14})$$

where $I_{zz,3}$ is the moment of inertia of the base about the z -axis through the centre of gravity of the base located at x_3, y_3, z_3 (in $\text{kg}\cdot\text{m}^2$).

NOTE The correct equation for I_T is given as follows:

$$I_T = I_{zz,1} + I_{zz,2} + I_{zz,3} + m_1 (x_1^2 + y_1^2) + m_2 (x_2^2 + y_2^2) + m_3 (x_3^2 + y_3^2) \quad (\text{A.15})$$

However, $I_{zz,1}$ and $I_{zz,2}$ (i.e. the moments of inertia of the fan and motor about the z -axis of their own centre of gravity respectively) are not always readily available since they have to be measured by experiment. An alternative, which is usually acceptable unless the fan/motor centres of gravity coincide with the overall centre of gravity, is to use Equation (A.14), which neglects $I_{zz,1}$ and $I_{zz,2}$, and derive a conservative over-estimate of the natural frequency f_T in Equation (A.13).

- 4) Rocking mode (zx plane). If the one-third ratio between the vertical centre of gravity and the mount separation is not exceeded then the frequency of the rotational mode of vibration in the zy plane, f_R , in hertz (Hz), given below is approximately 5 % less than the corresponding frequency f_2 obtained by coupled mode analysis (see A.3.2):

$$f_R = \frac{1}{2\pi} \left(\frac{\sum (k_H z^2 + k_V x^2)}{I_R} \right)^{1/2} \quad (\text{A.16})$$

where I_R is the moment of inertia of the system about the y -axis through the centre of gravity, in kilograms per square metre ($\text{kg}\cdot\text{m}^2$), given by the following equation:

$$I_R = I_{yy,3} + m_1 (x_1^2 + z_1^2) + m_2 (x_2^2 + z_2^2) + m_3 (x_3^2 + z_3^2) \quad (\text{A.17})$$

where $I_{yy,3}$ is the moment of inertia of the base about the y -axis through the centre of gravity of the base, in kilograms per square metre ($\text{kg}\cdot\text{m}^2$).

NOTE The correct equation for I_R is as follows:

$$I_R = I_{yy,1} + I_{yy,2} + I_{yy,3} + m_1(x_1^2 + z_1^2) + m_2(x_2^2 + z_2^2) + m_3(x_3^2 + z_3^2) \quad (\text{A.18})$$

where

$I_{yy,1}$ is the moment of inertia of the fan about the y -axis through the centre of gravity of the fan, in kilograms per square metre ($\text{kg}\cdot\text{m}^2$);

$I_{yy,2}$ is the moment of inertia of the motor about the y -axis through the centre of gravity of the motor, in kilograms per square metre ($\text{kg}\cdot\text{m}^2$);

$I_{yy,3}$ is the moment of inertia of the base about the y -axis through the centre of gravity of the base, in kilograms per square metre ($\text{kg}\cdot\text{m}^2$);

However, for reasons similar to those given in the note to item (3), it is acceptable to use Equation (A.17) which neglects $I_{yy,1}$ and $I_{yy,2}$.

- 5) Rocking mode (zy plane). The frequency of coupled rocking mode where sway is dominant, f_1 , in hertz (Hz), and the frequency of coupled rocking mode where rotation is dominant, f_2 , in hertz (Hz), is as given by the following equation:

$$f_1, f_2 = I_{xx,3} + m_1(y_1^2 + z_1^2) + m_2(y_2^2 + z_2^2) + m_3(y_3^2 + z_3^2) \quad (\text{A.19})$$

where $I_{xx,3}$ is the moment of inertia of the base about the x -axis through centre of gravity of the base, in kilograms per square metre ($\text{kg}\cdot\text{m}^2$);

NOTE Publications [12] and [13] give additional information on the calculation of frequencies.

Annex B (informative)

Secondary measurement methods for *in situ* testing or quality grading

B.1 General

This annex applies to vibration measurement which can be readily performed, but because of increased uncertainties in the measurement are not recommended for establishing accurate levels of vibration. However, where it is required to obtain vibration levels as a measurement of finished balance quality on machines produced in series, or for comparative site measurement, the method given in this annex may be used.

B.2 Bearing-housing measurements

When performing vibration analysis, it is always recommended that the transducer be securely mounted to the machine surface using one of the methods given in 7.2. However, other methods can be considered where the machines' operating speeds are low enough that those vibration frequencies of interest fall below the frequency limits imposed on such attachment methods by the transducer manufacturers. When such mounting methods are used, the contact should be made on a heavy part of the machine such as the machine frame or bearing housing, not at a compliant appendage or on lightweight sheet metal. Applications should be firm so that the pick-up maintains surface contact throughout the measurement operation. Table B.1 gives guidance on the maximum frequency at which the transducer mounting should be used.

Table B.1 — Typical maximum frequencies for transducer mounting

Transducer mounting method	Maximum frequency
Hand-held with 200 mm probe	200
Hand-held without probe	600
Magnetic transducer holder	1 500
NOTE 1 See also 7.2.	
NOTE 2 When measurements are being taken for the quality grading of production fans, permanent transducer attachments may not be desirable. In such cases, hand or magnetically held transducers may be used, provided that readings are limited to the maximum frequencies given in this Table.	
NOTE 3 See also ISO 5348.	
NOTE The values given are based on a particular manufacturer's recommendations.	

B.3 Shaft measurements

Many high-speed machines consist of relatively lightweight rotors mounted in heavy cases with rigid bearing supports. Due to the weight and stiffness of the machine case and bearings, there is often little outward evidence of rotor or shaft vibration, even though they may be vibrating excessively within the clearances of the bearings. On this type of machine, it becomes necessary to measure shaft vibration in order to know when seal and bearing clearances are in danger. Various pick-ups and their attachments can be used for this purpose as follows:

- a) non-contact pick-up (proximity);
- b) shaft-rider pick-up.

When undertaking shaft vibration measurements, it is strongly recommended that the equipment manufacturer's advice be sought in all areas related to selection, installation, application and safety procedures.

It is acknowledged that "shaft stick" accessories for use with a seismic velocity transducer are sometimes used, but this practice in general is not recommended for safety reasons.

NOTE For further information on shaft measurement, refer to ISO 7919-1.

Annex C (informative)

Recommended measuring positions for machinery health measurements

C.1 Machinery health

Within any machine, the various rotating components are located within some form of bearing arrangement. Any defect, e.g. unbalance, may lead to a vibration which is transmitted to the machine bearing housings via the bearings. For example, as the bearings wear or as the impeller balance deteriorates, there will be a corresponding increase in vibration magnitude. The characteristic of this change of magnitude with time is a useful indication of the machine's health. Measurement of shaft vibration can also be taken (see Annex B and ISO 7919-1).

C.2 Selection of measuring positions

Before selecting the final fixed location of a transducer for in-service condition checking, in the absence of the manufacturer's recommended locations, the site should be surveyed in the vicinity of the proposed transducer location, with a magnetically attached transducer (or similar means). This is to ensure that the proposed transducer location will best describe the vibrations of interest, for example, an antinodal position where the greatest magnitude occurs in either the most significant or "worst-case" direction. If stray signals at frequencies of 50 Hz, 60 Hz, 100 Hz or 120 Hz appear when the fan is running, the source of the signal may be electrical pick-up and, if so, when determined from a test using a soft rubber suspension for the transducer, this should be traced and eliminated. Recommended measuring positions are given in Figures C.1 and C.2.

C.3 Taking measurements

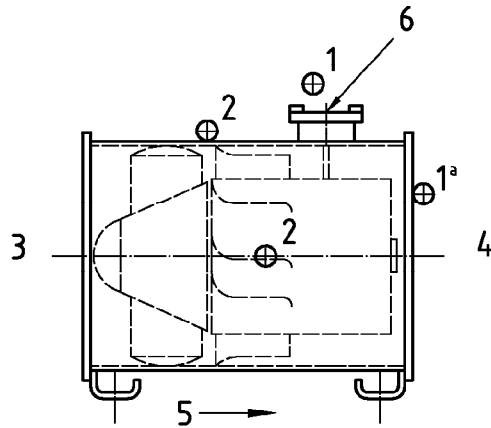
It is essential, in machinery health monitoring, that the measuring position remain unaltered. The bearing housings represent the ideal positions to make machine measurements and monitor the changes in vibration levels that directly reflect the health of the machine. Taking measurements at bearing housings on fans is not always possible once a fan has been installed. For example, for axial-flow fans, once the fan is installed, the supports for the stator/rotor, cable entry and bearing lubrication pipes may be the only direct connections to casing.

To ensure consistency of measurements, it is recommended that these positions be marked. This will ensure that vibrations checks and analysis are made at the same points. A machined spot face is of particular value here.

Whenever possible, the measuring positions should permit measurements to be made in the following three orthogonal directions, i.e.:

- a) vertical, longitudinal and transverse where the axis of the shaft is horizontal;
- b) vertical and in two transverse directions where the axis of the shaft is vertical.

Once measuring positions have been determined, they should be incorporated onto the production drawings to ensure that the same positions are used on every subsequent occasion.

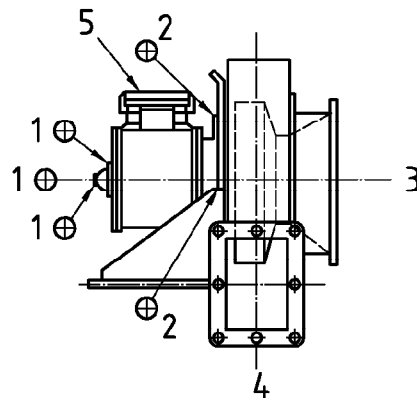


Key

- 1 motor free end
- 2 motor supports fan end
- 3 inlet
- 4 outlet
- 5 airflow
- 6 terminal box

^a Axis measurements may not be possible when installed.

Figure C.1 — Recommended measuring position for machinery health measurement of axial-flow fans



Key

- 1 motor free end
- 2 motor fan end
- 3 inlet
- 4 outlet
- 5 terminal box

Figure C.2 — Recommended measuring position for machinery health measurement of centrifugal fans

Annex D (informative)

Relationship between absolute vibration levels and levels expressed in decibels of vibration characteristics

Logarithmic scales are sometimes used to plot vibration amplitudes such as displacement, velocity and acceleration. This enables the decibel scale to be used in comparing levels and compressing the scale where the range of values is large.

The decibel (dB) is the ratio of one level with respect to a reference level and therefore has no dimensions. In order to quote absolute vibration levels, the reference level has to be stated.

For example, we can say that one vibration level is 10 dB greater than another level without any further explanation. If, however, we wish to say that a vibration level is, for example, 85 dB, we have to refer it to a reference level. We should say, therefore, that the vibratory velocity is 85 dB referred to 10^{-9} m/s.

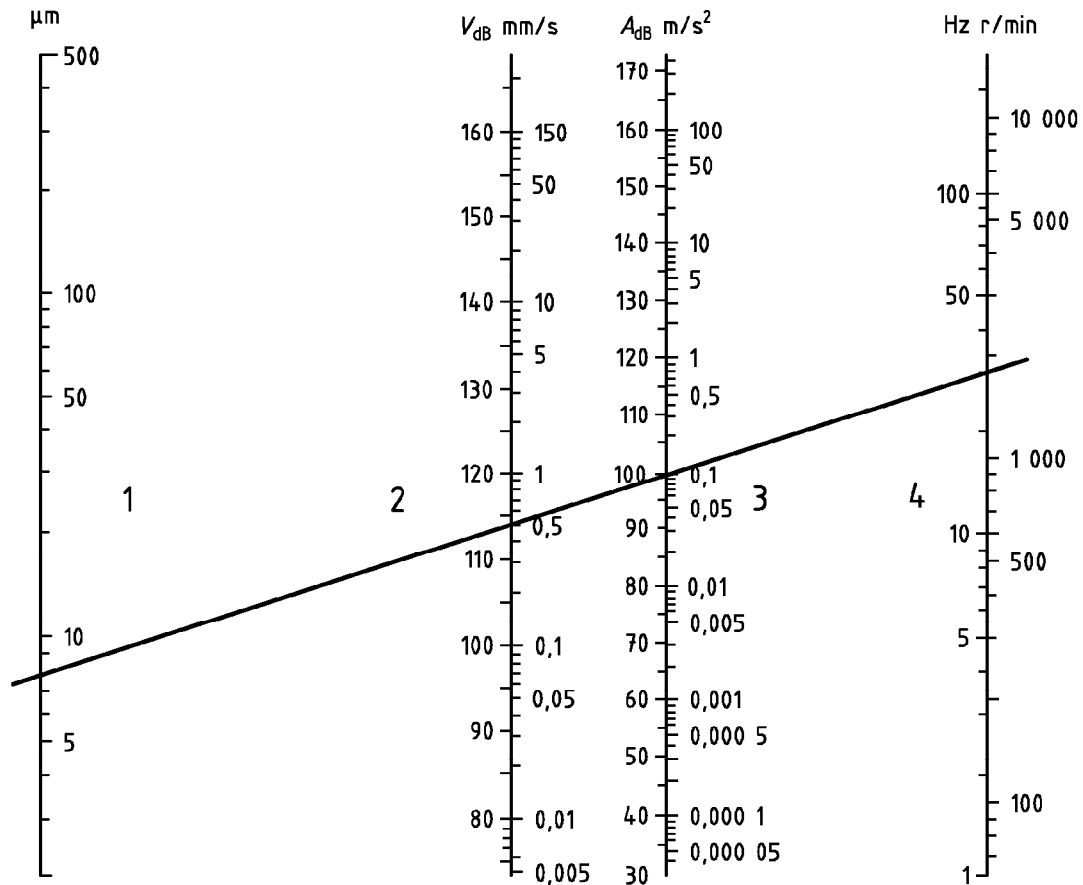
The nomogram given in Figure D.1 gives an easy method for converting from one property to another and from absolute to decibel levels, where the vibration is sinusoidal in character. The reference levels given in Table D.1 are in accordance with ISO 1683.

Table D.1 — Reference levels

Quantity	Definition	Reference
Vibratory acceleration level, L_a , dB	$L_a = 20 \log_{10} (a/a_0)$	$a_0 = 10^{-6} \text{ m/s}^2$
Vibratory velocity level, L_v , dB	$L_v = 20 \log_{10} (v/v_0)$	$v_0 = 10^{-9} \text{ m/s}$
Where		
a	is the instantaneous vibration acceleration (m/s^2);	
a_0	is the reference vibration (m/s^2);	
v	is the instantaneous vibration velocity (m/s);	
v_0	is the reference vibration velocity (m/s).	

When preparing results in a graphical form, frequency may also be plotted on a logarithmic scale. This has the effect of expanding the lower end, where important values such as rotational frequency are usually present, and compressing the higher end which usually encompasses those frequencies relating to bearing fatigue wear, etc. It does, however, give a constant percentage resolution over the whole width of the chart and keeps its size down to reasonable proportions.

NOTE Instruments exist where acceleration levels are measured in decibels relative to a reference value of 10^{-5} m/s^2 resulting in reported values 20 dB lower than those defined above.



Key

- 1 displacement (peak to peak)
- 2 velocity r.m.s
- 3 acceleration r.m.s
- 4 frequency

NOTE 1 Displacement $1 \mu\text{m} = 10^{-3} \text{ mm}$.

NOTE 2 When expressed in decibels (reference value: 10^{-5} m/s^2), subtract 20 dB.

EXAMPLE Displacement = $7,62 \mu\text{m}$ at fan velocity = $1\ 780 \text{ r/min}$ (30 Hz).

Draw a straight line between the scales and read-off the other values:

$$\begin{aligned} \text{velocity (r.m.s.)} &= 0,5 \text{ mm/s} \\ &= 114 V_{\text{dB}} \end{aligned}$$

$$\begin{aligned} \text{acceleration (r.m.s.)} &= 0,095 \text{ m/s}^2 \text{ r.m.s.} \\ &= 99,5 A_{\text{dB}} \end{aligned}$$

Figure D.1 — Machine vibration nomogram to convert absolute values to decibel levels (sinusoidal wave form)

Annex E (informative)

Relationship between vibration magnitude of single frequency signal

Any vibration consisting of a periodic sinusoidal waveform can be defined in terms of displacement, velocity and/or acceleration magnitudes at the appropriate waveform frequency. Velocity is the first derivative of displacement while acceleration is the first derivative of velocity and therefore the second derivative of displacement. For example, for a sinusoidal displacement, d , in metres (m), expressed as:

$$d = D \sin \omega t$$

where

D is the peak displacement amplitude, in metres (m);

ω is the angular frequency ($= 2\pi f$), in hertz (Hz);

t is the time, in seconds (s).

The velocity, v , in millimetres per second (mm/s) or in metres per second (m/s), expressed as:

$$v = \omega D \cos \omega t = V \cos \omega t$$

The acceleration, a , in metres per second squared (m/s^2), expressed as:

$$a = -\omega^2 D \sin \omega t = A \sin \omega t$$

where V and A are the peak amplitudes of velocity and acceleration, respectively.

The r.m.s. value of each parameter is equal to the peak value divided by $\sqrt{2}$.

If the vibration consists of a series of known amplitudes of d_j , v_j , a_j (where $j = 1, 2, 3, \dots, n$) for displacement, velocity and acceleration respectively, at known angular frequencies ($\omega_1, \omega_2, \omega_3, \dots, \omega_n$), the overall r.m.s. velocity $V_{\text{r.m.s.}}$, in millimetres per second (mm/s) or in metres per second (m/s), is given by the equation

$$V_{\text{r.m.s.}} = \sqrt{1/2 \left(D_1^2 \omega_1^2 + D_2^2 \omega_2^2 + D_3^2 \omega_3^2 + \dots + D_n^2 \omega_n^2 \right)}$$

$$V_{\text{r.m.s.}} = \sqrt{1/2 \left(V_1^2 + V_2^2 + V_3^2 + \dots + V_n^2 \right)}$$

$$V_{\text{r.m.s.}} = \sqrt{1/2 \left(\frac{A_1^2}{\omega_1^2} + \frac{A_2^2}{\omega_2^2} + \frac{A_3^2}{\omega_3^2} + \dots + \frac{A_n^2}{\omega_n^2} \right)}$$

If time records of the vibration measurement are available, the r.m.s. value can be derived from the square root of the integral, with respect to time, of the square of the vibration, e.g. for overall r.m.s. velocity, $V_{r.m.s.}$:

$$V_{r.m.s.} = \sqrt{\frac{1}{T} \int_0^T v^2 dt}$$

where T is the period of vibration, in seconds (s).

The operation of directly converting between magnitudes of displacement, velocity and acceleration can only be accomplished for single frequency components of vibration. As an example of such a conversion, if the overall r.m.s. vibration velocity $V_{r.m.s.}$ is known, the peak-to-peak displacement, D , in micrometres (μm), millimetres (mm) or metres (m), may be calculated from the following equation:

$$\begin{aligned} D &= 2\,000 \sqrt{2 \left(\frac{V_{r.m.s.}}{\omega_f} \right)} \\ &= 450 \left(\frac{V_{r.m.s.}}{f} \right) \end{aligned}$$

where f is the frequency, in hertz (Hz)

$$f = \omega/2\pi$$

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