INTERNATIONAL STANDARD

ISO 10814

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Mechanical vibration — Susceptibility and sensitivity of machines to unbalance

Vibrations mécaniques — Susceptibilité et sensibilité des machines aux balourds

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Reference number
ISO 10814:1996(E)

Foreword

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

International Standard ISO 10814 was prepared by Technical Committee ISO/TC 108, *Mechanical vibration and shock*, Subcommittee SC 1, *Balancing, including balancing machines*.

Annexes A to C of this International Standard are for information only.

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Introduction

Rotor balancing during manufacture (as described in ISO 1940-1 and ISO 11342, for example) is normally sufficient to attain acceptable vibration levels in service if other sources of vibration are absent. There are exceptions, however, where additional balancing during commissioning becomes necessary. Furthermore, after commissioning, some machines may require occasional or even frequent field rebalancing.

If the vibration levels are unsatisfactory during commissioning, the reason may be inadequate balancing or assembly errors. Another important cause may be that an assembled machine is especially sensitive to relatively small residual unbalances which are well within normal balance tolerances.

If vibration magnitudes are unsatisfactory, the first step often is an attempt to reduce the vibration by field balancing. If high vibration can be reduced by relatively small correction masses, high sensitivity to unbalance is indicated. This can arise, for example, if a resonant speed is close to the normal service speed and the damping in the system is low.

A sensitive machine which is also highly susceptible to its unbalance changing, may require frequent rebalancing *in situ*. This may be caused, for example, by changes in wear, temperature, mass, stiffness and damping during operation.

If the unbalance and other conditions of the machine are essentially constant, occasional trim balancing may be sufficient. Otherwise it may be necessary to modify the machine to change the resonant speed, damping or other parameters. Therefore, there is a need to consider permissible sensitivity values of the machine.

The repeatability of the sensitivity of a machine is influenced by several factors and may change during operation. Some thermal machines, especially those with sleeve bearings, have modal vibration characteristics which vary with certain operational parameters such as steam pressure and temperature, partial steam admission or oil temperature. For electrical machines, other parameters such as the excitation current may influence the vibration behaviour. In general, the machine vibration characteristics are influenced by the design features of the machine, including coupling of the rotor and the support conditions including the foundation. It should be noted that the latter may vary with time, for example owing to wear and tear.

This International Standard is only concerned with once-per-revolution vibration caused by unbalance; however it should be recognized that unbalance is not the only cause of once-per-revolution vibration.

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Mechanical vibration — Susceptibility and sensitivity of machines to unbalance

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1 Scope

1.1 This International Standard defines methods for determining machine vibration sensitivity to unbalance and provides evaluation guidelines as a function of the proximity of relevant resonant speeds to the service speed.

It includes a classification of certain machines in groups associated with the susceptibility to a change in unbalance.

This International Standard also makes recommendations on how to apply the numerical sensitivity values in some particular cases.

1.2 Machines are classified into three types in clause 4, and sensitivity values assigned to different groups of machines are shown in clause 5. The sensitivity values should be used on simple machine systems, preferably with rotors having only one resonant speed over the entire service speed range. They may also be used for machines that have more resonant speeds in the service speed range if the resonant speeds are widely separated (e.g. more than 20 % spaced).

The proposed sensitivity values are not intended to serve as acceptance specifications for any machine group but rather to give indications of how to avoid gross deficiencies as well as exaggerated or unattainable requirements. They may also serve as a basis for more involved investigations, for example, when in special cases a more exact determination of the required sensitivity is necessary. If due regard is paid to the proposed values, satisfactory running conditions can be expected in most cases.

The consideration of these values alone does not guarantee that a given magnitude of vibration in service is not exceeded. Many other sources of vibration can occur which are not the subject of this International Standard.

2 Normative references

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

ISO 1925:1990, *Mechanical vibration — Balancing — Vocabulary*.

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

3 Definitions

For the purposes of this International Standard, the definitions given in ISO 1925 and ISO 2041 and the following definitions apply.

3.1 susceptibility to unbalance: An indication of the likelihood of a machine having a significant change of unbalance over a certain period of operation.

3.2 sensitivity to unbalance: A measure of the vibration response of a machine to a change in unbalance.

NOTE 1 It is usually numerically expressed in the two ways shown in 3.2.1 and 3.2.2 below.

3.2.1 local sensitivity: Ratio of the magnitude of the change of the displacement or velocity vector in a specified measuring plane to the magnitude of a change in the unbalance in a specified plane in the rotor at a specified speed.

In technical terms, the local sensitivity can be expressed as:

$$S_{k,r} = \frac{\left|\Delta \vec{S}_k\right|}{\left|\Delta \vec{U}_r\right|} \qquad \ldots (1)$$

where

$\left|\Delta \vec{S}_k\right|$ is the change in once-per-revolution vibration in plane k;

$\left|\Delta \vec{U}_r\right|$ is the change in trial unbalance attached to plane r in the rotor (or change in trial unbalance set).

NOTE 2 The local sensitivity is frequently referred to as the "influence coefficient". It is a dimensional quantity.

3.2.2 modal sensitivity, M_n: Ratio of the change of the modal displacement vector to a change of the modal eccentricity (modal unbalance divided by modal mass). It is a non-dimensional quantity.

In practical determinations of modal sensitivity, care should be taken to extract the relevant modal components.

Modal sensitivity for excitation of the machine by unbalance for mode n can be shown to be:

$$M_n = \frac{\left(\dfrac{\Omega}{\omega_n}\right)^2}{\sqrt{\left[1-\left(\dfrac{\Omega}{\omega_n}\right)^2\right]^2 + 4\,\zeta_n^2\left(\dfrac{\Omega}{\omega_n}\right)^2}} \qquad \ldots (2)$$

where

Ω is the rotational speed;

ω_n is the nth undamped resonant speed;

ζ_n is the damping ratio of the nth mode.

Under conditions where the rotational speed equals a resonant speed, M_n becomes approximately $1/(2\zeta_n)$. For light damping, this is the maximum amplification at resonance denoted by Q_n; it is influenced only by the level of damping in the system.

NOTE 3 Modal sensitivity is sometimes referred to as the vibration magnification factor for mode n.

4 Machine susceptibility classification

4.1 Type I: Low susceptibility

Machines of this type have a low likelihood of experiencing significant unbalance changes during operation. Typically, they have large rotor masses in comparison to the support housing and operate in a clean environment, have negligible wear and exhibit minimal rotor distortion caused by temperature changes.

EXAMPLES:

Paper machine rolls, printing rolls and high-speed vacuum pumps.

4.2 Type II: Moderate susceptibility

Machines of this type have a moderate likelihood of experiencing significant unbalance changes during operation, such as rotors in an environment with large temperature changes and/or moderate wear.

EXAMPLES:

Pumps in clean media, electric armatures, gas and steam turbines, small turbo generators for industrial applications and turbo compressors.

4.3 Type III: High susceptibility

Machines of this type have a high likelihood of experiencing significant unbalance changes during operation, such as blowers running in deposit producing environments, pumps operating in sludge, rotors with high wear or running in corrosive environments.

EXAMPLES:

Centrifuges, fans, screw conveyors and hammer mills.

5 Modal sensitivity values

5.1 Modal sensitivity ranges A to E

In figures 1 to 3, modal sensitivity is classified for service speeds below and above a resonant speed of the machine system. The range limits are chosen in such a way that, close to a resonant speed, the magnification factor is constant. The curves in figures 1 to 3 are derived from equation (2). The range limits depend on the type of machine in the susceptibility classification (the damping ratio being lower, in the same sensitivity range, for type III than for type I machines).

Generally, for the sensitivity ranges below, the corresponding running conditions given in table 1 can be expected.

Examples of how to use figures 1 to 3 are given in annex B.

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Table 1

Modal sensitivity range	Expected running conditions
A: Very low sensitivity	Very smooth resonant speed; difficult to detect
B: Low sensitivity	Smooth, low and stable vibrations
C: Moderate sensitivity	Acceptable, moderate and slightly unsteady vibrations
D: High sensitivity	Sensitive to unbalance; regular field balancing may be required
E: Very high sensitivity	Too sensitive to unbalance; to be avoided

5.2 Characteristics of modal sensitivity

5.2.1 While range A will theoretically appear to be the most desirable, considerations of cost and feasibility may often make it necessary to operate with higher modal sensitivities.

5.2.2 For high-performance machines (e.g. those that have a short period between planned maintenance cycles), it may be permissible to allow higher values of modal sensitivity.

5.2.3 For machines for which field balancing is not practical or not economical, smaller values of modal sensitivity may have to be selected.

5.2.4 Consideration of the sensitivity does not always give sufficient assurance that, at all parts of the machine, vibration limits are not exceeded (see clauses 7 and 8).

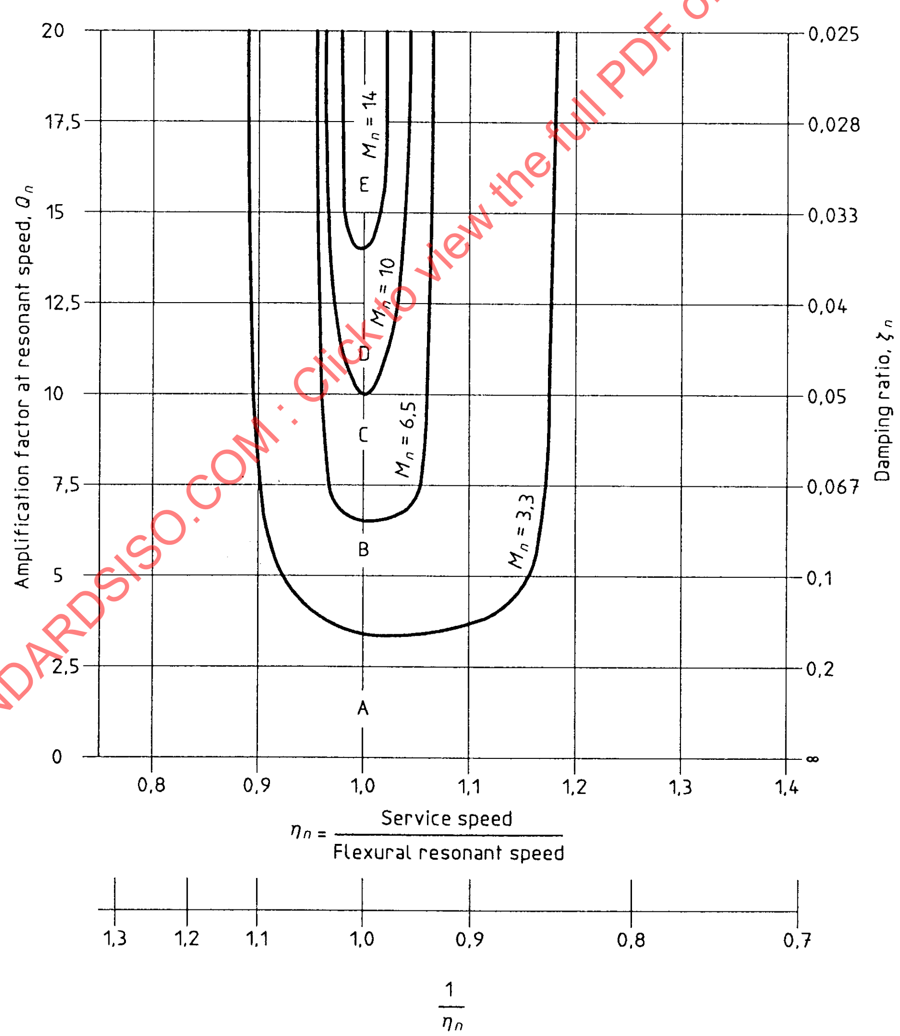

Figure 1 — Modal sensitivity ranges A to E for type I machines

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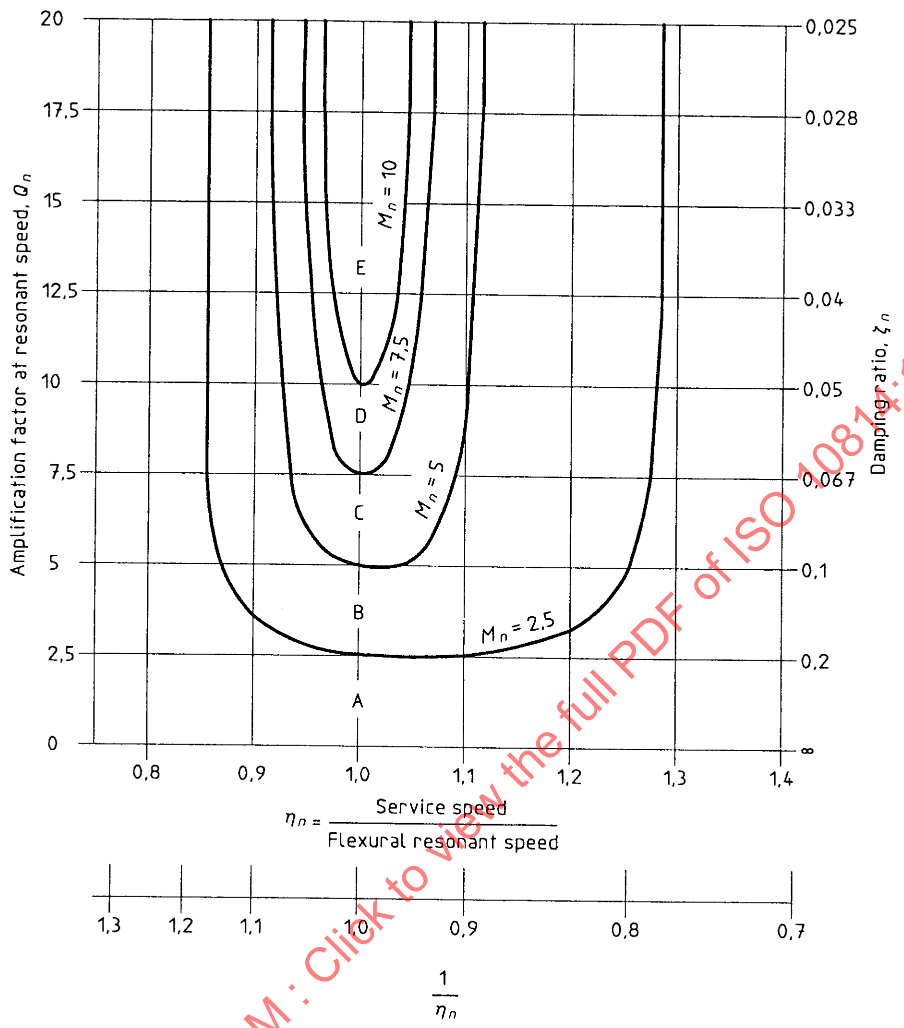

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Figure 2 — Modal sensitivity ranges A to E for type II machines

5.3 Accelerating rotors

Higher values of the modal sensitivity are permissible for machines which, in service, always accelerate rapidly through all resonant speeds because steady-state response has insufficient time to develop. Machines which are only started and stopped infrequently may also be acceptable with higher values of modal sensitivity.

Figure 4 illustrates, for a single degree of freedom system, the reduction in the modal sensitivity as a function of the rotational acceleration. However, for most practical acceleration rates, this effect is small and can be neglected.

Such a reduction in the modal sensitivity may be calculated before referring to figures 1 to 3 (as these figures are only applicable for slowly accelerating rotors), taking the following points into account:

a) passing through a resonant speed is assumed to happen with constant rotational acceleration;

b) the modal sensitivity shall be tested (where possible) under steady-state conditions as close to the resonant speed as possible;

c) the maximum amplitudes during rapid acceleration or deceleration will occur at rotor speeds different from the resonant speeds because of the delay in response.

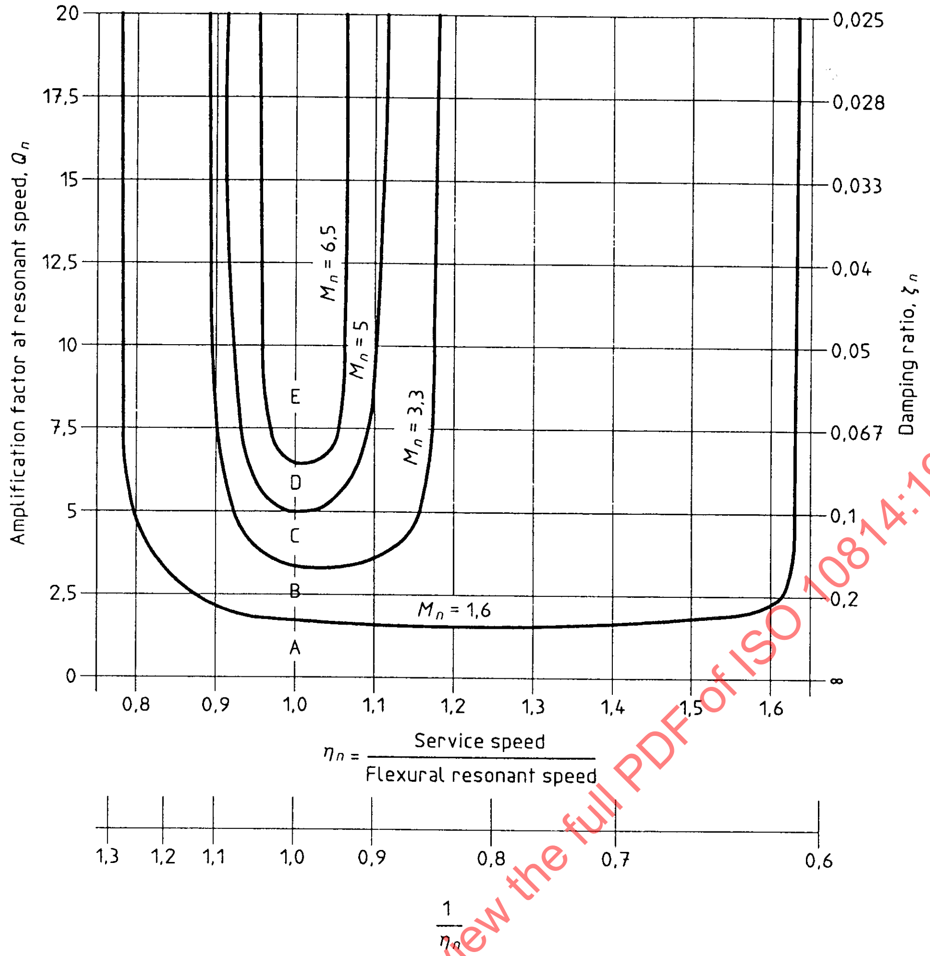

Figure 3 — Modal sensitivity ranges A to E for type III machines

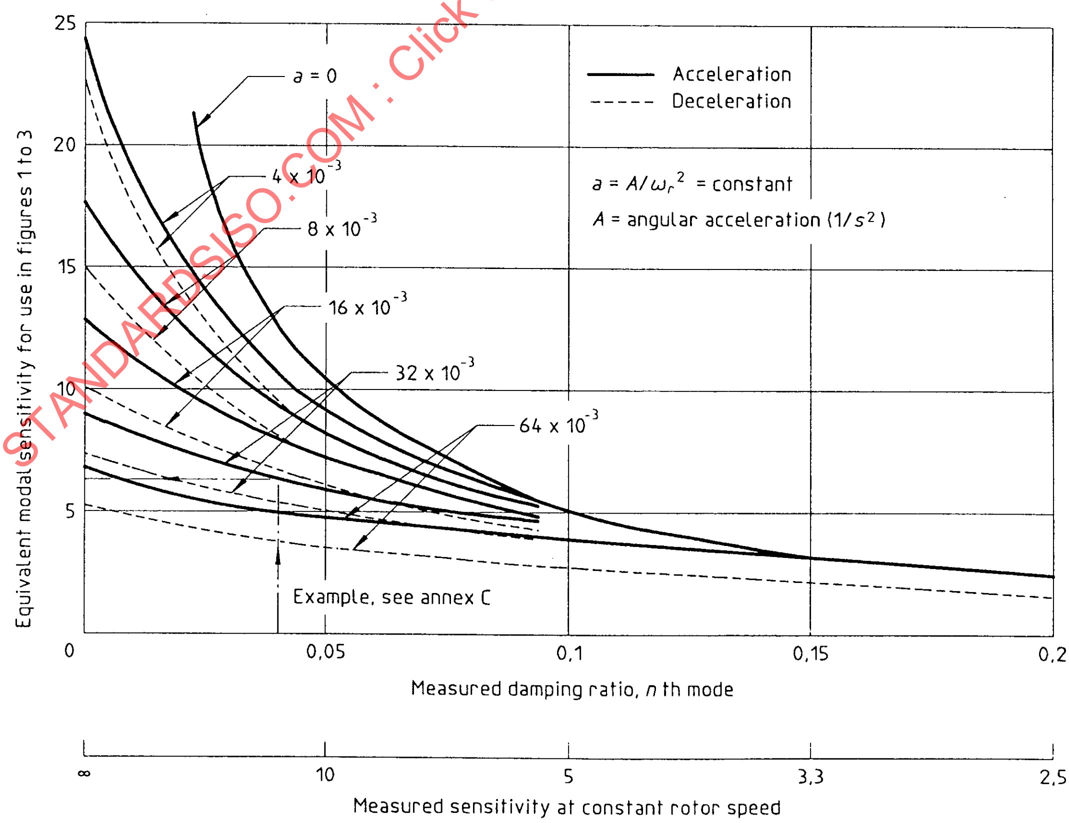

Figure 4 — Reduction of modal sensitivity as a function of the damping ratio and the rotational acceleration

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6 Experimental determination of the modal sensitivity near a resonant speed under operational conditions

Once-per-revolution vibration is normally measured in amplitude and phase so that the Nyquist diagram procedure given in 6.1 can be used. If only amplitude measurements are available, the procedure given in 6.2 should be used.

6.1 Nyquist diagram procedure

At resonant speeds the sensitivity to unbalance is highly dependent on the damping that is present in the system. As the damping itself may depend on many parameters, it is recommended that sensitivity tests be performed with the machine under as close to normal operating conditions as possible (e.g. at normal operating temperatures).

In many cases the response of the system close to a given resonant speed occurs predominantly in the corresponding mode only, so that its behaviour can be modelled by an equivalent single degree of freedom system. In these circumstances, the damping and the flexural resonant speed can be found from measurements during slow run-up or coast-down, where the rate of change of speed is small.

Such a single degree of freedom system describes the vibration in the nth mode, and the following relationship is applicable:

$$Q_n \approx \frac{\omega_n \times \Omega_{45}}{\omega_n^2 - \Omega_{45}^2} \qquad \ldots (3)$$

where

Q_n is the maximum modal sensitivity and equals the value of M_n when the rotor speed equals the resonant speed ($\Omega = \omega_n$);

Ω_{45} (or Ω_{135}) is the speed where the phase has shifted $\pm 45°$ from that at the resonant speed.

An example of the procedure is shown in annex A.

Under certain circumstances, the magnitude and phase pattern on the Nyquist plot may be irregular because there are several modes in proximity to each other. In such cases, an evaluation of the modal sensitivity can be made if a trial weight set is added to the rotor which preferentially excites the mode of interest. The procedure is then applied to the difference vectors which describe the response to the trial mass set.

With the exception of sensors located close to the nodes of the flexural principal modes, any once-per-revolution vibration data can be used for the procedure, but data from sensors installed at the location of relatively high amplitudes will give more accurate results.

6.2 Bode diagram procedure

If only a plot of the once-per-revolution displacement amplitude versus speed is available, it may still be possible to find the vibration magnification factor Q_n by the procedure illustrated in figure 5.

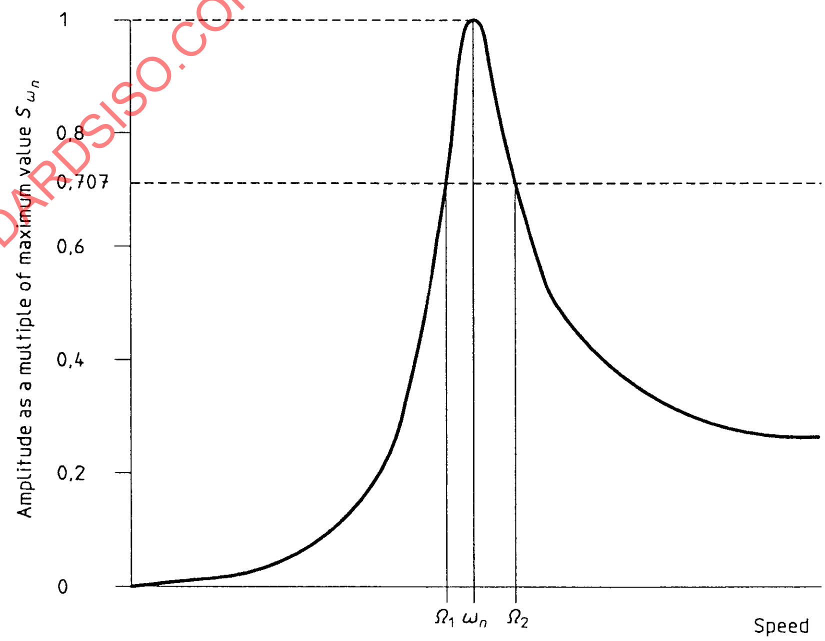

Figure 5 — Bode diagram procedure for estimating the modal sensitivity

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If ω_n is the rotor speed corresponding to the maximum vibration amplitude S_{ω_n}, and Ω_1 and Ω_2 the speeds where the displacement is 0,707 of the maximum amplitude, then the amplification factor is:

$$Q_n \approx \frac{\omega_n}{\Omega_2 - \Omega_1} \qquad \ldots (4)$$

This approach also has accuracy limitations, if the damping is low or the shape of the resonance curve is significantly influenced by adjacent modes or other factors. In such cases, the use of a trial weight set as explained in 6.1 may be applicable.

The speed of response of the measuring apparatus and an insufficient number of sampling points can also be a problem.

7 Numerical values for the local sensitivity

In many cases (e.g. on rotors with an overhung portion, rotors with limited clearance on certain rotor parts, or on rotors which in service run close to resonant speeds) the local sensitivity (influence coefficient) may be of interest at a variety of speeds in the operational speed range including the resonant speeds.

The magnitude of the local sensitivity that is measured on a machine is, amongst other things, a function of the location of the measurement plane and the axial location of the test unbalance. It therefore differs from the modal sensitivity which has a single value at a given rotor speed.

It is generally only necessary to measure local sensitivity at measuring points where at speeds of interest the amplitudes must be limited. Then the measured sensitivity at a location and speed of interest should be evaluated from the ratio:

$$\frac{\text{Permissible amplitude}}{\text{Permissible unbalance during operation}}$$

NOTE 4 Permissible amplitude and permissible unbalance can be obtained from ISO 10816-1 and ISO 11342, respectively.

Alternatively, values mutually agreed upon for this purpose between the machine manufacturer and the user may be substituted.

Depending on the operational parameters of a machine (speed, speed/resonant speed, etc.), it might be advisable to limit the maximum permissible local sensitivity by applying a factor ($\leqslant$ 1,0) to the above formula.

It is common practice to accept higher vibration during run-up and run-down than at service speed. It is therefore possible to accept higher local sensitivity at other than service speeds if the vibrations do not exceed agreed limits.

8 Experimental determination of the local sensitivity

For measuring local sensitivity values, it is recommended that where possible rotor planes be used which produce maximum vibration response for the modes and speeds of interest, and where it is easily possible to add test weights.

8.1 Prepare the machine for normal operation.

8.2 Run the machine to the desired data collection speed Ω. This speed is often chosen as that speed in the service speed range that is closest to a resonant speed. Wait until vibrations and other relevant parameters are steady and measure the once-per-revolution vibration in the agreed planes (k). During measurement, the speed, load and other parameters of the machine which could influence the state of vibration should be held as constant as possible.

8.3 Attach a single trial weight, $\Delta\vec{U}_r$, to the rotor in the agreed plane (r). This should be big enough to produce a clearly measurable change in the state of vibration from that found in 8.2. This should not be so large that dangerous vibrations are developed at any speed that the machine will run through or operate at. Sometimes it may be necessary to attach a trial weight set.

8.4 Measure the vibrations under the same conditions as in 8.2.

8.5 Calculate for each measuring plane, the vectorial difference between the measured values and those found in 8.2 and 8.4. This is the value $\Delta\vec{S}_k$. The amount of this value divided by the amount of the trial weight, $\Delta\vec{U}_r$, is the local sensitivity to unbalance:

$$S_{k,r} = \frac{\left|\Delta\vec{S}_k\right|}{\left|\Delta\vec{U}_r\right|}$$

This value, $S_{k,r}$, is the value for the chosen data collection speed.

The linearity of the system and repeatability of the measured values shall be taken into account.

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9 Damped unbalance sensitivity analysis

If the relevant experimental data are not available, it may be useful to make a numerical analysis for machines which pass through or approach resonant speeds during run-up or service.

Such analyses may include the following in the mathematical model:

a) stiffness, mass and damping characteristics of the rotor and support system;

b) bearing and seal stiffness and damping as a function of the rotor speed;

c) identification of the natural frequencies and corresponding mode shapes;

d) calculation of the modal damping;

e) calculation of the local sensitivities at specified rotor axial locations and for specified unbalance planes.

In the calculation, unbalance should be placed in appropriate locations for modes of interest. For example, unbalance should be placed near mid-span for the first mode and in anti-phase near the ends for the second mode. In addition, values should be calculated for the local sensitivities for comparison with values which can be measured on site.

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