INTERNATIONAL STANDARD

ISO 11342

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Mechanical vibration — Methods and criteria for the mechanical balancing of flexible rotors

Vibrations mécaniques — Méthodes et critères

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Contents

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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.

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 11342 was prepared by Technical Committee ISO/TC 108, *Mechanical vibration and shock*, Subcommittee SC 1, *Balancing, including balancing machines.*

This first edition of ISO 11342 cancels and replaces ISO 5406:1980 and ISO 5343:1983.

Annexes A, B, C, D, E, F, G, H and J of this International Standard are for information only.

Introduction

The aim of balancing any rotor is satisfactory running when installed on site. In this context "satisfactory running" means that no more than an acceptable level of vibration is caused by the unbalance remaining in the rotor. In the case of a flexible rotor, it also means that no more than an acceptable magnitude of deflection occurs in the rotor at any speed up to maximum service speed.

Most rotors are balanced by their manufacturers prior to machine assembly because afterwards, for example, there may be only limited access to the rotor. Furthermore, balancing of the rotor is often the stage at which a rotor is approved by the purchaser. Thus, while satisfactory running on site is the aim, the balance quality of the rotor is usually initially assessed in a balancing facility. Satisfactory running on site is in most cases judged in relation to vibration from all causes, while in the balancing facility primarily once-per-revolution effects are considered.

Section 2 of this International Standard classifies rotors into groups in accordance with their balancing requirements and establishes in Section 3 methods of assessment of residual unbalance.

This International Standard also shows in Section 3 how criteria for use in the balancing facility may be derived from either vibration limits specified for the assembled and installed machine or unbalance limits specified for the rotor. If such limits are not available, this International Standard shows how they may be derived from ISO 10816-1 and parts 1 to 4 of ISO 7919, if desired in terms of vibration, or from ISO 1940-1 if desired in terms of permissible residual unbalance.

ISO 1940-1 is concerned with the balance quality of rotating rigid bodies and is thus not directly applicable to flexible rotors because they may undergo significant bending deflection. However, in subclauses 2.3 and 3.4 of this International Standard, methods are presented for adapting the criteria of ISO 1940-1 to flexible rotors.

As this International Standard is complementary in many details to parts 1 and 2 of ISO 1940, it is recommended that, where applicable, they should be considered together.

Mechanical vibration — Methods and criteria for the mechanical balancing of flexible rotors

Section 1: General

1.1 Scope

This International Standard classifies rotors into groups in accordance with their balancing requirements, describes balancing procedures, specifies methods of assessment of the final state of unbalance, and gives guidance on balance quality criteria.

All rotors are classified into those which can be balanced by rigid rotor, modified rigid rotor, or high-speed (flexible rotor) balancing techniques.

Two methods are specified for evaluating the balance quality of a flexible rotor in a balancing facility before machine assembly: the first assesses the vibration level, and the second assesses the rotor residual unbalance. If the rotor balance tolerances suggested herein are achieved during correction in a balancing facility, the specified vibration limits of the assembled machine in service (see ISO 10816-1 and parts 1 to 4 of ISO 7919) will most probably be achieved. Accordingly, the criteria specified are those to be met when the rotor is tested in the balancing facility, but they are derived from those specified for the complete machine, when installed, or from values known to ensure satisfactory running of the rotor when it is installed.

As in the case of parts 1 and 2 of ISO 1940, this International Standard is not intended to serve as an acceptance specification for any rotor group, but rather to give indications of how to avoid gross deficiences and/or unnecessarily restrictive requirements. This International Standard may also serve as a basis for more involved investigations, for example when a more exact determination of the required balance quality is necessary. If due regard is paid to the specified methods of manufacture and limits of unbalance, satisfactory running conditions can most probably be excepted.

There are situations in which an otherwise acceptably balanced rotor experiences an unacceptable vibration level *in situ*, owing to resonances. A resonant or near-resonant condition in a lightly damped structure can result in excessive vibratory response to a small unbalance. In such cases, it may be necessary to alter the natural frequency or damping of the structure rather than to balance to very low levels, which may not be maintainable over time.

The subject of structural resonances and modifications thereof is outside the scope of this International Standard.

The methods and criteria given are the result of experience with general industrial machinery. They may not be directly applicable to specialized equipment or to special circumstances. Therefore, there may be cases where deviations from this International Standard may be necessary[1].

1.2 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

1) Information on such exceptions is welcomed and should be communicated to the national standards body in the country of origin for transmission to the secretariat of ISO/TC 108.

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 1940-1:1986, *Mechanical vibration — Balance quality requirements of rigid rotors — Part 1: Determination of permissible residual unbalance.*

ISO 1940-2:—[2], *Mechanical vibration — Balance quality requirements of rigid rotors — Part 2: Balance errors.*

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

ISO 2953:1985, *Balancing machines — Description and evaluation.*

ISO 7919-1:1986, *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation — Part 1: General guidelines.*

ISO 7919-2:—[2], *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation — Part 2: Large land-based steam turbine-generator sets.*

ISO 7919-3:—[2], *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation — Part 3: Guidelines for coupled industrial machines.*

ISO 7919-4:—[2], *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation — Part 4: Guidelines for gas turbines.*

ISO 8821:1989, *Mechanical vibration — Balancing — Shaft and fitment key convention.*

ISO 10816-1:—[2], *Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts — Part 1: General guidelines.*

1.3 Definitions

For the purposes of this International Standard, the definitions relating to mechanical balancing given in ISO 1925 and many of the definitions relating to vibration given in ISO 2041 apply.

Definitions given in ISO 1925 relating to flexible rotors are given for information in annex H.

2) To be published.

Section 2: Balancing methods

2.1 Fundamentals of flexible rotor dynamics and balancing

2.1.1 Unbalance distribution

The rotor design and method of construction can significantly influence the magnitude and distribution of unbalance along a rotor. Rotors may be machined from a single forging or they may be constructed by fitting several components together. For example, jet engine rotors are constructed by joining many shell, disc and blade components. Generator rotors, however, are usually manufactured from a single forging, but will have additional components fitted. The distribution of unbalance may also be significantly influenced by the presence of large local unbalances arising from shrink-fitted discs, couplings, etc.

Since the unbalance distribution along a rotor is likely to be random, the distribution along two rotors of identical design will be different. The distribution of unbalance is of greater significance in a flexible rotor than in a rigid rotor because it determines the degree to which any flexural mode of vibration is excited. Moreover, the effect of unbalance at any point along a rotor depends on the bending deflection of the rotor at that point.

The correction of unbalance in transverse planes along a rotor, other than those in which the unbalance occurs, may induce vibrations at speeds other than that at which the rotor was originally corrected. These vibrations may exceed specified tolerances, particularly at or near the flexural critical speeds.

In addtion, some rotors which become heated during operation are susceptible to thermal distortions which can lead to changes in the unbalance. If the rotor unbalance changes significantly from run to run, it may be impossible to balance the rotor within tolerance.

2.1.2 Flexible rotor mode shapes

If the effect of damping is neglected, the modes of a rotor are the flexural principal modes and, in the special case of a rotor supported in isotropic bearings, are rotating plane curves. Typical curves for the three lowest principal modes for a simple rotor supported in flexible bearings near to its end are illustrated in figure 1.

For a damped rotor/bearing system, the flexural modes may be space curves rotating about the shaft axis, especially in the case of substantial damping, arising perhaps from fluid-film bearings. A possible substantially damped second mode is illustrated in figure 2. In many cases, the damped modes can be treated approximately as principal modes and hence regarded as rotating plane curves.

It must be stressed that the form of the mode shapes and the response of the rotor to unbalances are strongly influenced by the dynamic properties and axial locations of the bearings and their supports.

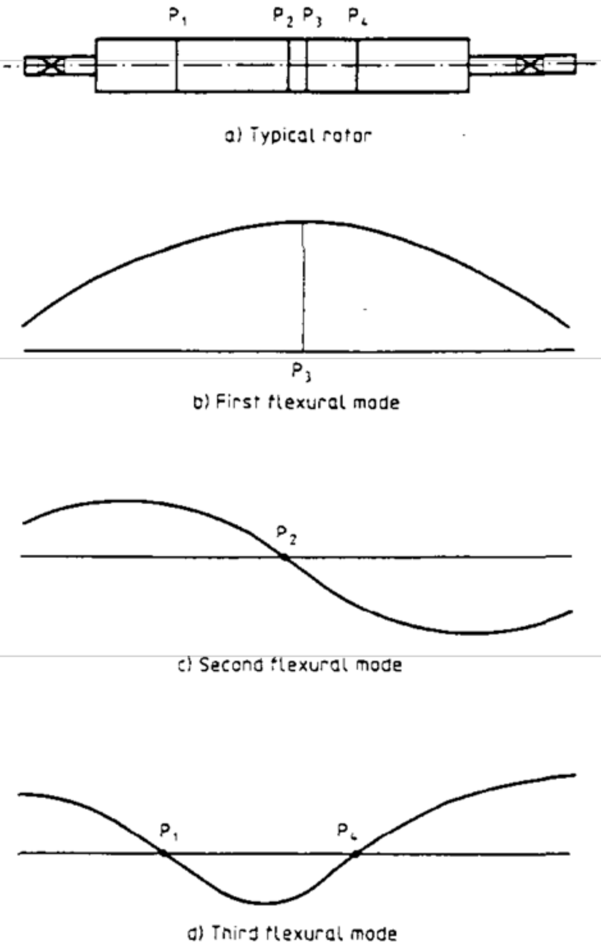
2.1.3 Response of a flexible rotor to unbalance

The unbalance distribution can be expressed in terms of modal unbalances. The deflection in each mode is caused by the corresponding modal unbalance. When a rotor rotates at a speed near a critical speed, it is usually the mode associated with this critical speed which dominates the deflection of the rotor. The degree to which large amplitudes of rotor deflection occur in these circumstances is determined by:

a) the magnitude of the modal unbalances;

b) the proximity of the associated critical speeds to the running speeds; and

c) the amount of damping in the rotor/support system.



NOTE — P_1 to P_4 are correction planes.

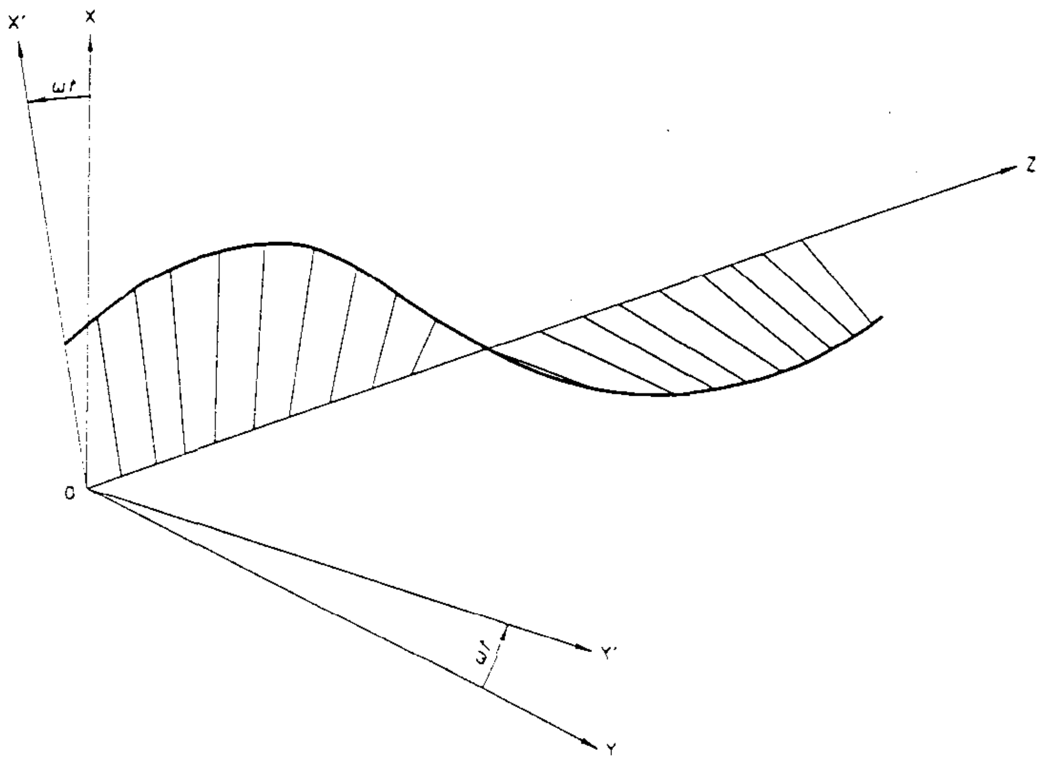
Figure 1 — Typical mode shapes for flexible rotors on flexible supports

If a particular modal unbalance is reduced by the addition of a number of discrete correction masses, then the corresponding modal component of deflection is similarly reduced. The reduction of the modal unbalances in this way forms the basis of the balancing procedures described in this International Standard.

The modal unbalances for a given unbalance distribution are a function of the flexible rotor modes. Moreover the effect produced in a particular mode by a given correction depends on the ordinate of the mode shape curve at the axial location of the correction. Consider an example in which the curves of figure 1 b) to 1 d) are mode shapes for the rotor in figure 1 a). A correction mass attached to the rotor in figure 1 a) in the plane P_2 will produce no change in response in the second mode. Similarly, a correction mass attached in either plane P_1 or P_4 will not affect the response in the third mode. Conversely, a correction mass in plane P_3 will produce the maximum effect on the first mode.

2.1.4 Aims of flexible rotor balancing

The aims of balancing are determined by the operational requirements of the machine. Before balancing any particular rotor, it is desirable to decide what balance criteria can be regarded as satisfactory. In this way the balancing process can be made efficient and economic, but still satisfy the needs of the user.



NOTE — OX, OY and OZ are fixed axes. OX' and OY' are axes rotating about OZ at speed ω.

Figure 2 — Possible damped second-mode shape

Balance criteria are specified to achieve the following:

a) acceptable values of machinery vibration and shaft deflection;

b) acceptable values of unbalance forces applied to the bearings.

The ideal aim in balancing flexible rotors would be to correct the local unbalance occurring at each elemental length by means of unbalance corrections at the element itself. This would result in a rotor in which the centre of mass of each elemental length lies on the shaft axis.

A rotor balanced in this ideal way would have no static and couple unbalance and no modal components of unbalance. Such a perfectly balanced rotor would then run satisfactorily at all speeds in so far as unbalance is concerned.

In practice, the necessary reduction in the unbalance forces is usually achieved by adding or removing masses in a limited number of correction planes. There will invariably be some distributed residual unbalance after balancing.

Vibrations or oscillatory forces due to the residual unbalance must be reduced to acceptable magnitudes over a range of speeds, including one or more critical speeds. Only in special cases is it sufficient to balance flexible rotors for a single speed. It should be noted that a rotor, balanced satisfactorily for a given service speed range, may still experience excessive vibration if it has to run through a critical speed to reach its service speed. Balancing a rotor according to its mode shapes is not an end in itself. Whatever balancing technique is used, the final goal is to apply unbalance correction distributions to minimize the unbalance effects up to the service speed, and possible overspeed.

2.1.5 Provision for correction planes

Rotors are often balanced mode by mode. In this process, correction masses are located along the rotor, so that at each stage in the balancing procedure the new correction masses do not significantly disturb modes already balanced.

The exact number of axial locations along the rotor that are needed for this process depends to some extent on the particular balancing procedure which is adopted. For example, centrifugal compressor rotors are sometimes assembly-balanced in the end planes only, after each disc and the shaft have been separately balanced in a low-speed balancing machine. Generally, however, if the speed of the rotor approaches or exceeds its nth critical speed, then at least n and commonly $(n + 2)$ correction planes are needed along the rotor.

An adequate number of correction planes at suitable axial positions should be included at the design stage. In practice, the number of correction planes is often limited by design considerations and in field balancing by limitations on accessibility.

2.1.6 Rotors coupled together

When two rotors are coupled together, the complete unit will have a series of critical speeds and mode shapes. In general, these speeds are neither equal to nor simply related to the critical speeds of the individual, uncoupled rotors. Moreover, the deflection shape of each part of the coupled unit need not be simply related to any mode shape of the corresponding uncoupled rotor. In theory, therefore, the unbalance distribution along two or more coupled rotors should be evaluated in terms of modal unbalances with respect to the coupled system and not to the modes of the uncoupled rotors.

For practical purposes, it is often necessary that each rotor be balanced separately as an uncoupled shaft. In many cases, this procedure will ensure satisfactory operation of the coupled rotors. The degree to which this technique is practicable depends, for example, on the mode shapes and the critical speeds of the uncoupled and coupled rotors, and the distribution of unbalance.

If further balancing on site is required, reference should be made to annex A.

2.2 Classification

For the purposes of this International Standard, rotors are divided into five main classes as shown in 2.2.1 to 2.2.5 and in table 1. Each class requires different balancing techniques. A procedure to determine if a rotor is rigid or flexible is given in annex E.

2.2.1 Class 1: Rigid rotors

A rotor is considered to be rigid when its unbalance can be corrected in any two (arbitrarily selected) planes. After the correction, its residual unbalance does not change significantly (relative to the shaft axis) at any speed up to the maximum service speed and when running under conditions which approximate closely to those of the final supporting system. Rotors of this type can be corrected by rigid-rotor balancing methods (see ISO 1940-1).

2.2.2 Class 2: Quasi-rigid rotors

A rotor that cannot be considered rigid but that can be balanced using modified rigid-rotor balancing techniques is considered to be a quasi-rigid rotor.

Class 2 rotors are subdivided (see table 1) into:

a) rotors in which the axial distribution of unbalance is known (classes 2a, 2b, 2c and 2d; also class 2e in which the axial distribution is partly known);

b) rotors in which the axial distribution of unbalance is not known (classes 2f, 2g and 2h).

The subdivision of class 2 rotors shows the many reasons why rotors can often be balanced satisfactorily at low speed as rigid rotors even though they are flexible. Some rotors will fit into more than one category of the subdivision.

2.2.3 Class 3: Flexible rotors

A rotor that cannot be balanced using modified rigid-rotor balancing techniques but instead requires the use of high-speed balancing methods is considered to be a flexible rotor.

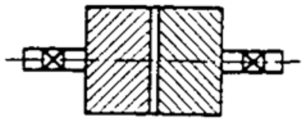
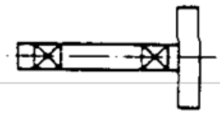
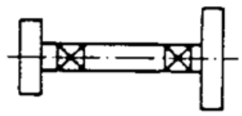
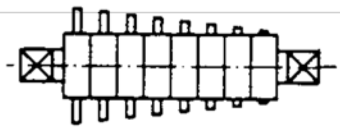
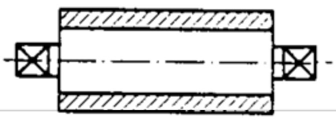
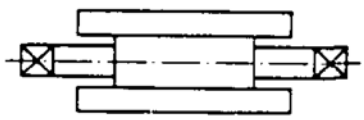
Class 3 is subdivided (see table 1) because the balancing techniques, criteria and bearing requirements may differ substantially for different rotors.

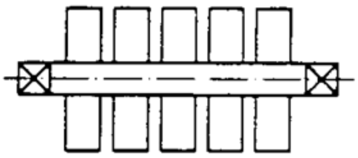
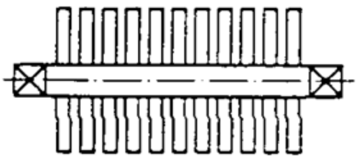
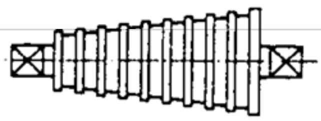
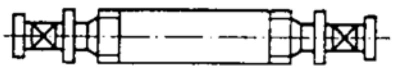
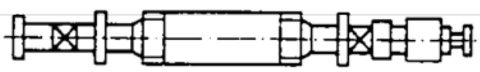
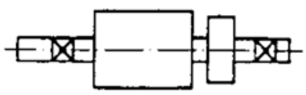
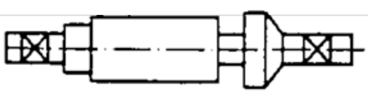
2.2.4 Class 4

A rotor that could fall into class 1, 2 or 3 but has in addition one or more components that are themselves flexible or flexibly attached is considered to be a class 4 rotor.

A subdivision of class 4 rotors is indicated in 2.4.2.

Table 1 — Classification of rotors

Class of rotor	Description	Example
Class 1: Rigid rotors	A rotor whose unbalance can be corrected in any two (arbitrarily selected) planes so that after that correction, its unbalance does not change significantly at any speed up to maximum service speed (see 2.2.1).	 Gear wheel
Class 2: Quasi-rigid rotors	A rotor that cannot be considered rigid but that can be balanced using modified rigid-rotor balancing techniques see (2.2.2).	
Rotors in which the axial distribution of unbalance is known		
Classe 2a	A rotor with a single transverse plane of unbalance, for example a single mass on a light flexible shaft whose unbalance can be neglected.	 Grinding wheel
Class 2b	A rotor with two transverse planes of unbalance, for example two masses on a light shaft whose unbalance can be neglected.	 Grinding wheel with pulley
Class 2c	A rotor with more than two transverse planes of unbalance.	 Axial compressor rotor
Class 2d	A rotor with uniformly or linearly varying unbalance.	 Printing press roller
Class 2e	A rotor consisting of a rigid mass of significant axial length supported by flexible shafts whose unbalance can be neglected.	 Computer memory drum

Class of rotor	Description	Example
Rotors in which the axial distribution of unbalance is not known		
Class 2f	A symmetrical rotor (with two end correction planes) whose maximum speed does not significantly approach second critical speed, whose service speed range does not contain first critical speed, and which has a controlled initial unbalance.	 Multistage centrifugal pump/compressor
Class 2g	A symmetrical rotor (with two end correction planes and a central correction plane) whose maximum speed does not significantly approach second critical speed and which has a controlled initial unbalance.	 Hig-speed centrifugal pump/compressor
Class 2h	As unsymmetrical rotor which has a controlled initial unbalance treated in a similar manner to a class 2f rotor.	 I.P. steam turbine rotor
Class 3: Flexible rotors	A rotor that cannot be balanced using modified rigid-rotor balancing techniques but instead requires the use of high-speed balancing methods (see 2.2.3).	
Class 3a	A rotor that, for any unbalance distribution, is significantly affected by only the first mode unbalance.	 Four-pole generator rotor
Class 3b	A rotor that, for any unbalance distribution, is significantly affected by only the first and second mode unbalance.	 Small two-pole generator rotor
Class 3c	A rotor significantly affected by more than the first and second mode unbalance.	 Large two-pole generator rotor
Class 4	A rotor that could fall into class 1, 2 or 3 but has in addition one or more components that are themselves flexible or flexibly attached (see 2.2.4).	 Rotor with centrifugal switch
Class 5	A rotor that could fall into class 3 but for some reason, for example economy, is balanced for one speed of operation only (see 2.2.5).	 High-speed motor

2.2.5 Class 5

A rotor that could fall into class 3 but in some circumstances is balanced for one speed of operation only is considered to be a class 5 rotor. For rotors to be treated as class 5, the conditions stated in 2.4.3 shall be met.

2.3 Balancing of class 2 rotors

2.3.1 General

Class 2 rotors are flexible rotors, however, they can be satisfactorily balanced at low speed by modified rigid-rotor balancing techniques. As such, they represent the borderline case between rigid rotors for which low-speed balancing is sufficient and flexible rotors that demand the high-speed balancing procedure given in 2.4.

A low-speed balancing machine generally measures only the dynamic unbalance of a rigid rotor. However, it is possible in some circumstances with the use of appropriate procedures to balance a flexible rotor at low speed so as to ensure satisfactory running when the rotor is installed in its final environment.

The amount of modal unbalance remaining in a rotor after low-speed balancing will depend on the mode shapes of the rotor and the axial positions of the unbalances relative to the correction planes used.

2.3.2 Selection of correction planes

If the axial positions of the unbalances are known, the correction planes should be provided as closely as possible to these positions. If the axial positions of the unbalances are not known, then see 2.1.5 for general guidance.

2.3.3 Rotors made up of individual components

When a rotor is composed of separate components that are distributed axially and mounted concentrically on a shaft, the adoption of one of the following procedures will considerably increase the probability that low-speed balancing is sufficient.

NOTE 1 Certain rotors in all classes contain a number of individual components which are mounted concentrically (for example, blades, coupling bolts, pole pieces, etc.). These components may be arranged according to their individual mass or mass moment to achieve some or all of the required unbalance correction described in any of the methods. If these components must be fitted or replaced after balancing, they should be arranged in balanced sets.

2.3.3.1 Assembly of individually balanced components

Each component, including the shaft, should be individually balanced as a rigid rotor in accordance with ISO 1940-1 before assembly. In addition, the fits and concentricities of the shaft diameters or other locating features that position the individual components on the shaft should be held to close tolerances.

The fits and concentricities of the balancing mandrel diameters or other location features that position each individual component on the mandrel should likewise be held within close tolerances relative to the shaft axis of the mandrel. Errors in unbalance and concentricity of the mandrel may be compensated by index balancing (see ISO 1940-1).

When balancing the components and the shaft individually, due allowances should be made for any unsymmetrical feature such as keys (see ISO 8821) that form part of the complete rotor but are not used in the individual balancing of the separate components.

It is advisable to check by calculation the unbalance produced by errors, such as eccentricities and assembly tolerances on the mandrel and on the shaft, to evaluate their effects.

When calculating the effect of these errors, it is important to note that the effect of the errors can be cumulative on the final assembly. Procedures for dealing with such errors can be found in ISO 1940-2.

2.3.3.2 Sequential-assembly balancing

The shaft should first be balanced. The rotor should then be balanced as each component is mounted, correction being made only on the latest component added. This method avoids the necessity for close control of concentricities of the locating diameters or other features that position the individual components on the shaft.

If this method is adopted, it is important to ensure that the balance of the parts of the rotor already treated is not changed by the addition of successive components.

In some cases, it may be possible to add two single-plane components at a time and perform dynamic unbalance correction in those two components. In cases where several components form a short stiff unit or sub-assembly or core section that is normally balanced in two planes only, one such unit or sub-assembly may be added at a time and corrected by two-plane (dynamic) balancing.

In some cases where a gross unbalance may occur in a single component, it may be advantageous to balance this component separately before mounting it on the rotor, in addition to carrying out the balancing procedure after it is mounted.

2.3.4 Service speed of the rotor

If the service speed range includes, or is close to, a flexural critical speed, then class 2 low-speed balancing methods should only be used with caution.

2.3.5 Initial unbalance

The process of balancing a flexible rotor in a low-speed balancing machine is a compromise. The magnitude and distribution of initial unbalance is a major factor determining the degree of residual modal unbalance that can be expected.

For rotors in which the axial distribution of initial unbalance is known and appropriate correction planes are available (classes 2a to 2e), the permissible initial unbalance is limited only by the amount of correction possible in the correction planes. Methods of low-speed balancing of such rotors are given in 2.3.6.1 to 2.3.6.5.

For rotors in which the actual distribution of the initial unbalance is not known, there are no generally applicable low-speed balancing methods. However, for rotors of classes 2f to 2h, although the exact distribution of initial unbalance cannot be known, the magnitude can be controlled by the prebalancing of individual components. In these cases, the low-speed initial unbalance can be used as a measure of the distribution of unbalance. The maximum initial unbalance that can be tolerated will depend on the allowable bearing load and the detail characteristics of the rotor. Balancing procedures for such rotors are given in 2.3.6.6 and 2.3.6.7.

2.3.6 Balancing procedures for class 2 rotors

2.3.6.1 Class 2a: Rotors with a single transverse plane of unbalance

If the initial unbalance is known to be wholly contained in one transverse plane and the correction is made in this plane, then the rotor will be balanced for all speeds.

In these circumstances, the unbalance can be corrected by low-speed balancing as effectively as at service speed.

2.3.6.2 Class 2b: Rotors with two transverse planes of unbalance

If the initial unbalance is known to be wholly contained in two transverse planes and the corrections are made in these planes, then the rotors will be balanced for all speeds.

In these circumstances, the unbalance can be measured and corrected by low-speed balancing as effectively as at service speed.

2.3.6.3 Class 2c: Rotors with more than two transverse planes of unbalance

When a rotor is composed of two or more separate components that are distributed axially, it is likely that there will be more than two transverse planes of unbalance. A satisfactory state of balance may be achieved by low-speed balancing, provided that the methods of manufacture and the precautions suggested in 2.3.3 are followed.

It is important to recognize that the assembly process may produce changes in the shaft geometry (e.g. shaft runout), and further changes may occur during high-speed service.

2.3.6.4 Class 2d: Rotors with uniformly distributed or linearly varying unbalance

If, because of design or method of construction, a rotor has unbalances that are distributed uniformly along its entire length (e.g. a tube), it may be possible by selecting suitable axial positions of two correction planes to achieve satisfactory running over the entire speed range by low-speed balancing. It is likely that the optimum position of the two correction planes producing the best overall running conditions can only be determined by experimentation on a number of rotors of similar type.

For a simple rotor system that satisfies the following conditions:

a) single-span rotor with no significant overhang,

b) uniform or linear distribution of unbalance,

c) uniform bending flexibility of rotor along its length,

d) symmetrical position of end correction planes about midspan, and

e) continuous service speeds below and not significantly approaching second critical speed, the optimum position for the two correction planes is 22 % of the bearing span inboard of each bearing. If correction in these planes does not produce satisfactory results, it may still be possible to balance the rotor at low speed by utilizing correction planes in the middle and at the rotor ends as shown in annex B. To do this, it is necessary to assess what proportion of the total initial unbalance is to be corrected at the centre plane.

2.3.6.5 Class 2e: Rotors with a rigid core

If the unbalance in the rotor is known to be contained wholly within a substantially rigid section of the rotor and the unbalance correction is also made within this section, then the unbalance will be zero in all modes. Such a rotor that has flexibility derived solely from its flexible shaft can be balanced at low speed.

2.3.6.6 Class 2f: Symmetrical rotors with controlled initial unbalance (two end correction planes)

When a rotor is composed of separate components that are individually balanced before assembly, as outlined in 2.3.3.1, a satisfactory state of balance may be achieved in a low-speed balancing machine provided the initial unbalance of the assembled rotor does not exceed specified tolerances.

For such rotors, the axial distribution and magnitude of unbalance of the complete assembly will not be known. Since the maximum speed of this class of rotor does not significantly approach the second critical speed, the most unfavourable case that will occur with a given distribution of unbalance is when the individual contributions of the assembled components to the resultant unbalance have the same angular position. The maximum initial unbalance that may be corrected in two planes will have to be determined by experience.

If realistic data on shaft and bearing flexibility, etc. are available, analysis of response to unbalance using mathematical models will be useful initially.

2.3.6.7 Class 2g and 2h rotors with controlled initial unbalance

Experience has shown that symmetrical rotors (class 2g) that conform to the requirements given in 2.3.6.6, but have an additional central correction plane, may be balanced at low speed as a rigid rotor, with a higher permissible initial unbalance for the complete rotor than the value arrived at in 2.3.6.6. Experience has shown that between 30 % and 60 % of the initial static unbalance should be corrected in the central plane.

For unsymmetrical rotors (class 2h) that do not conform to the configuration defined in 2.3.6.6. (for example, as regards symmetry or overhangs), it may be possible to use a similar procedure and hence to arrive at the maximum permissible initial unbalance that may be corrected satisfactorily at any given correction plane.

However, in extreme cases, the permissible initial unbalance arrived at in this way may be too small to make this method of balancing practicable and in these cases some other method of balancing the rotor will have to be adopted, for example, sequential-assembly balancing (2.3.3.2).

2.4 Balancing of class 3, 4 and 5 rotors

2.4.1 Balancing procedures for class 3 rotors

2.4.1.1 General

Traditionally, two different methods have been formulated for achieving a satisfactory state of balance, namely modal balancing and the influence coefficient approach. The basic theory behind both of these methods and their relative merits are described widely in the literature and therefore no further detailed description will be given here. In most practical balancing applications, the method adopted will normally be a combination of both approaches. The degree to which the method is weighted to one or the other will be dependent on the particular circumstances. One such balancing method is described in 2.4.1.5. However, it should be recognized that for particular cases some variation of the procedure may be used.

2.4.1.2 Rotor supports

For balancing purposes the rotor should be mounted on suitable bearings. In some cases it is desirable that the bearing supports in the balancing facility are chosen to provide similar conditions to those at site, so that the modes obtained during site operation willl be adequately represented during the balancing process and hence reduce the necessity for subsequent field balancing.

If the rotor to be balanced has an overhang of significant mass, or flexibility, that would normally be supported when installed on site, it may be necessary to provide an additional bearing (steady bearing) to limit its deflection during the balancing operation.

2.4.1.3 Measuring system

Transducers should be positioned to measure shaft, bearing or support vibration or bearing force as ppropriate. The system should be capable of measuring the amplitude of the once-per-revolution component of the signal, together with the phase angle relative to some fixed angular reference on the rotor. Alternatively, it is possible to use a measuring system which resolves the once-per-revolution measurements into orthogonal components.

It is recommended that the transducers and their supports be selected such that they do not undergo resonant vibration at any test speed, to avoid errors in interpretation of the unbalance response.

2.4.1.4 Low-speed (rigid-rotor) balancing

Experience has shown that it may be advantageous to carry out rigid-rotor balancing at low speed, prior to balancing at higher speeds. This may be particularly advantageous for rotors significantly affected by only the first flexural critical speed.

If desired, therefore, balance the rotor at low speed when it is not affected by modal unbalances. Alternatively, this stage can be omitted by proceeding directly to 2.4.1.5.

NOTE 2 Low-speed balancing may avoid the need for carrying out the procedure given in 2.4.1.5.10 .

2.4.1.5 High-speed (flexible-rotor) balancing

All vibration (or force) measurements in this subclause relate to the once-per-revolution vectors.

The procedure for high-speed (flexible-rotor) balancing is given in 2.4.1.5.1 to 2.4.1.5.10.

2.4.1.5.1 Run the rotor to some safe speed approaching the first flexural critical speed. This will be termed the first flexural balancing speed.

NOTE 3 For some rotor types (e.g. turbine rotors with shrunk on stages, or generator rotors) it is advisable to make only preliminary corrections near the flexural critical speeds to get the rotor to its service speed, or overspeed, where components may move into their final position.

Record the readings of vibration (or force) under steady-state conditions. Before proceeding, it is essential to confirm that the readings are repeatable. Several runs may be necessary for this purpose.

2.4.1.5.2 Add a set of trial masses to the rotor, which should be selected to produce a significant vector change in vibration (or force) at the first flexural balancing speed.

If low-speed (rigid-rotor) balancing has been omitted, the trial mass set usually comprises only one mass, which for rotors which are essentially symmetric about midspan will be placed towards the middle of the rotor.

If low-speed balancing has been performed, then the trial mass set will usually consist of masses located at three distinct correction planes. In this case, the masses are proportioned so that the low-speed (rigid-rotor) balancing is not upset.

2.4.1.5.3 Run the rotor to the same speed and under the same conditions as in 2.4.1.5.1, and record the new readings of vibration (or force).

2.4.1.5.4 From the vectorial changes of the readings between 2.4.1.5.1 and 2.4.1.5.3, compute the relevant influence coefficients at the first flexural balancing speed. Hence, compute the magnitude and angular position of the correction to be applied to cancel the effects of unbalance at the first flexural balancing speed. Add this correction.

NOTE 4 A graphical illustration of the vectorial subtraction underlying this calculation is shown in annex G.

The rotor should now run at any speed up to and through the first flexural critical speed without any significant amplification of vibration (or force). If this is not the case, refine the correction or repeat the procedure given in 2.4.1.5.1 to 2.4.1.5.4 using a new balancing speed, possibly closer to the first flexural critical speed.

2.4.1.5.5 Run the rotor to some safe speed approaching the second flexural critical speed. This will be the second flexural balancing speed. Record readings of vibration (or force) under steady-state conditions at this speed.

2.4.1.5.6 Add a set of trial masses to the rotor. These should be located along the rotor, so that a significant vector change in vibration (or force) is produced at the second flexural balancing speed, without significantly affecting the first mode and, if relevant, the low-speed balance.

2.4.1.5.7 Run the rotor to the same speed as in 2.4.1.5.5 and record the new readings of vibration (or force).

2.4.1.5.8 From the vectorial changes in the readings between 2.4.1.5.5 and 2.4.1.5.7, compute the influence coefficients at the second flexural balancing speed for this set of trial masses. Use these values to compute a set of correction masses which cancel the effects of unbalances at the second flexural balancing speed. Attach this set of correction masses.

The rotor should now run at any speed up to and through the second flexural critical speed without any significant amplification of vibration (or force). If this is not the case, refine the correction or repeat the procedure given in 2.4.1.5.5 to 2.4.1.5.8 using a balancing speed, possibly closer to the second flexural critical speed.

2.4.1.5.9 Continue the above operations for balancing speeds, close to each flexural critical speed in turn within the permissible speed range. Each new set of trial masses should be chosen so that they have a significant effect on the appropriate mode, but do not significantly affect the balance which has already been achieved at lower speeds. The trial mass distribution can be obtained from experience or a computer simulation. For each case, a set of correction masses should be computed from the relevant influence coefficient and attached to the rotor. Each set of correction masses will compensate for the unbalance at the current balancing speed.

2.4.1.5.10 If, after correction at all flexural balancing speeds, significant vibrations (or forces) still occur within the service speed range, the procedure given 2.4.1.5.9 should be repeated at a balancing speed close to the maximum permissible test speed. In this case, it may not be possible to magnify the effect of the remaining (higher) modal unbalance components by running close to their associated flexural critical speeds.

NOTES

5 For some rotors it may be possible to run safely through some or all of the critical speeds before completing the balancing. In that case, the number of runs required to determine the influence coefficients can be reduced.

6 It should be noted that the method described above assumes that there is a linear relationship between the unbalance vector and the vibration (or force) response vector. In certain cases this may not be so, particularly, for example, where there is a high initial unbalance and the rotor is supported by fluid film bearings. In these cases it may be necessary to redetermine the influence coefficients as the vibration (or force) response vector is reduced in magnitude.

7 In practice, the above procedure or variations can be automated as a computer-aided balancing method.

2.4.2 Balancing procedures for class 4 rotors

Rotors in this class may have a basic shaft and body construction that would fall into class 1, 2 or 3. In addition, they have one or more components that are either flexible or are flexibly mounted so that the unbalance of the whole system may change with speed.

Rotors in this class may fall into two categories:

a) rotors whose unbalance changes continuously with speed, for example, rubber-bladed fans;

b) rotors whose unbalance changes up to a certain speed and remains constant above that speed, for example, rotors of single-phase induction motors with a centrifugal starting switch.

It is sometimes possible to balance these rotors with counterbalances of similar characteristics. If not, the following procedures should be used.

2.4.2.1 Rotors that fall into category a) should be balanced in a balancing machine at the speed at which it is specified that the rotor should be in balance.

2.4.2.2 Rotors that fall into category b) should be balanced at a speed above that at which the unbalance ceases to change.

NOTE 8 It may be possible to minimize or counterbalance the effects of the flexible components by careful design and by attention to their locations, but it should be appreciated that rotors in this class are likely to be in balance at one speed only or within a limited range of speed.

2.4.3 Balancing procedures for class 5 rotors

Some rotors that are flexible and pass through one or more critical speeds on their way to service speed may, under special circumstances, be balanced for one speed only (usually service speed). However, rotors having critical speeds close to service speed or those coupled to other flexible rotors should be excluded from this class. In general, rotors that fall into this class fulfil one or more of the following conditions:

a) the acceleration and deceleration up to and from service speed is so rapid that the amplitude of vibration at the critical speeds will not build up beyond acceptable limits;

b) the damping of the system is sufficiently high to keep vibrations at the critical speeds within acceptable limits;

c) the rotor is supported in such a manner that objectionable vibrations are avoided;

d) a high level of vibration at the critical speeds is acceptable;

e) the rotor runs at service speed for such long periods that otherwise unacceptable starting/stopping conditions can be tolerated.

A rotor that fulfils any of the conditions above should be balanced in a high-speed balancing machine or equivalent facility at the speed at which it is determined that the rotor should be in balance.

For these rotors, it is especially important that the stiffness of the balancing machine support system be sufficiently close to site conditions to ensure that at the balancing speed the predominant modes are the same as those that will be experienced on site.

Some consideration should be given to the axial correction mass distribution. It may be possible to choose optimum axial positions for the correction planes so that two planes may be sufficient. This may produce a minimum residual unbalance in the lower modes and thus minimize the vibrations when running through critical speeds.

Section 3: Criteria

3.1 Evaluation of final state of unbalance

Depending on the class and purpose of the rotor being assessed, the final state of unbalance may be evaluated either in terms of vibration at specified measuring points, or by residual unbalance in specified correction planes.

NOTE 9 In the case of small mass-produced rotors, simpler assessment procedures than those detailed in this International Standard may suffice.

3.1.1 Rotors whose final state of unbalance is evaluated by means of vibration measurement in a high-speed balancing or overspeed facility

3.1.1.1 Installation

In cases where the mode shapes of the rotor depend significantly on the dynamic properties of the supports, the rotor should have supports whose dynamic properties as nearly as possible represent those of site conditions, thus permitting, where appropriate, a range of modes to be examined in the balancing facility.

When special-purpose balancing machine pedestals that have variable stiffness are used, they should be locked in the appropriate state of adjustment for the evaluation.

If the rotor has an overhung mass that would normally be supported when installed on site, a steady bearing may be used to limit its deflection during the test.

If the rotor has an overhung mass that is not supported in any way when installed on site, it should also be left unsupported during the test.

In some cases, two vibration transducers should be installed on each bearing pedestal 90° apart at the same transverse plane to permit resolution of the vertical and horizontal transverse vibrations, when such resolution is required.

As an alternative, or additionally, shaft-riding probes or non-contacting proximity transducers may be used to measure shaft vibrations. Generally, these shall be installed 90° apart in the same transverse plane to permit resolution of the vertical and horizontal transverse vibrations if such resolution is required.

In all cases, there should be no resonances of the transducer and/or mounting within the speed range of the test.

However, it may be necessary in the early stage of balancing to provide support with a steady bearing to enable the rotor to get safely to service speed or overspeed to allow the rotor components to move into their final position (see note 3 in 2.4.1.5.1).

The output from all transducers should be read on equipment that is able to differentiate between the synchronous component due to unbalance, the slow-speed runout where relevant, and other components of the vibration.

The drive for the rotor should be such as to impose negligible restraint on the vibration of the rotor and introduce negligible unbalance into the system.

NOTE 10 To establish that the drive coupling introduces negligible balance error, the coupling should be indexed balanced as described in ISO 1940-2.

Before the residual unbalance of the rotor is assessed, it should be run at some convenient low speed or speeds to remove any temporary bend it may have taken. Once this has been achieved, the remaining repeatable low-speed runout values should be measured and, where necessary, subtracted vectorially from any subsequent shaft measurements at the speed of interest.

3.1.1.2 Procedure

When the above conditions have been satisfied, the rotor should be run up to speed at a low acceleration rate to ensure that vibration peaks are suppressed. All significant peaks of vibration should then be measured between 70 % of the observed first flexural critical speed and the maximum service speed if measurement over the whole speed range is not possible.

The rotor should be held at maximum service speed long enough to eliminate any transient effects. Synchronous vibration measurements should then be taken.

At the conclusion of the test at maximum service speed, the rotor should be run to a specified overspeed, if demanded by the specification (see note 11).

After the rotor has been held for a specified time at full overspeed, if overspeed is demanded by the specification, it should be decelerated to maximum service speed and the synchronous vibrations again measured. If permanent changes of residual unbalance, caused by overspeeding, are expected to occur, the final balancing and evaluation of vibration should be carried out after overspeeding.

The rotor should then be decelerated and measurements of synchronous vibration should be taken at decrements of speed of not more than 5 % of the maximum service speed between the maximum service speed and 70 % of the first flexural critical speed if measurement is not available for the whole speed range. During the test, the rate of deceleration should be slow enough to ensure that the peaks of vibration at the critical speeds are not significantly suppressed.

NOTE 11 The overspeed to be attained will depend upon the overall test specification for the particular type of rotor. Where no specification exists, agreement should be reached between the manufacturer and user.

3.1.2 Rotors whose final state of unbalance is assessed by means of vibration measurement on the test bed

Rotors whose final state of unbalance is evaluated on the test bed should have an instrumentation and test procedure as stated in 3.1.1, but it should be appreciated that different procedures may be necessary in some cases, for example:

a) the rotor is assembled as a complete machine driven by its own power;

b) rotors for which only full-speed readings can be obtained, such as an induction motor;

c) vibration transducers cannot be placed at the bearings; in these cases, the points where vibrations should be measured should be agreed between the manufacturer and user;

d) the state of unbalance depends on load, in which case the range of load over which the residual unbalance is assessed should be agreed between the manufacturer and user.

3.1.3 Rotors whose final state of unbalance is assessed by means of vibration measurement at site

3.1.3.1 Machines that have their state of balance assessed after final installation at site are subject to many factors that can produce vibration. Some of this vibration may be at shaft rotational frequency from sources other than mechanical unbalance. Some of the factors that can produce such vibrations, together with some of the precautions that should be taken, are mentioned in annex A.

3.1.3.2 If any of the stationary parts of the machine or the supporting foundation structure are in resonance at the service speed, high levels of vibration are sometimes produced even though the rotor residual unbalance is well within normally accepted tolerances.

In such circumstances, balancing within exceptionally fine limits may be required to reduce the vibration level. Such improvements may be only useful if the machine is not highly susceptible to unbalance. If, in operation, there is a high probability that new unbalances will occur, consideration should be given to the practicality of eliminating the structural resonances or increasing the damping in the system or other measures, so that satisfactory operation can be obtained.

3.1.3.3 In the final installation at site, there may be logistical factors during commissioning that will conflict with obtaining the steady-state conditions needed to assess the state of balance. It may then be necessary to combine the result of balancing runs with tests for other purposes. If the preliminary running of the installed machine shows the result of balancing to be in doubt, special runs should be arranged specifically for confirming the adequacy of the balance.

In many installations (for example, where the prime mover is a "direct to line start" induction motor) it may impossible to control the speed of rotation during run-up, and steady conditions can only be achieved at full speed. Agreement should therefore be reached between the manufacturer and user on the speed range over which the state of balance quality should be checked.

The balance check is normally made with the machine unloaded. If the machine is loaded, the load at which the state of balance is to be checked should be agreed between the manufacturer and user.

3.1.3.4 Vibration measuring equipment should be installed as specified in 3.1.1.1. Where suitable monitoring equipment is provided in the installation, this may be used instead. Alternatively, vibrations may be read on portable apparatus using a hand-held vibration transducer.

3.1.4 Rotors whose final state of unbalance is evaluated in a low-speed balancing machine as residual unbalance in specified correction planes

Class 2 rotors usually have their balance quality assessed in a low-speed balancing machine. In most cases, a subsequent high-speed check will be made on the test bed or on site. In specific cases, by agreement between the manufacturer and user, the high-speed assessment may be dispensed with and the rotor accepted on the basis of the residual unbalance at low speed. This applies particularly to class 2 rotors sold as spares, where a final assessment at site may be delayed for a considerable time.

The rotor should be complete and all attachments such as half-couplings, gear wheels, etc. should be fitted.

The balancing machine should be one that conforms to ISO 2953. See ISO 1940-1 for the procedure for assessing residual unbalance and for cautionary comments.

Before the residual unbalance of the rotor is assessed, it should be run at some suitable speed to remove any temporary bend.

When the above conditions have been satisfied, the rotor should be run at the balancing speed and readings taken of amount and angle of unbalance remaining in each measuring plane.

For class 2f and 2g rotors, the initial unbalance after assembly should also be stated, in addition to the measured residual unbalance. For rotors that have been balanced in several stages during assembly or that have been made up of balanced components (see 2.3.3), the residual unbalance achieved at each stage should be stated.

3.1.5 Rotors whose final state of unbalance is evaluated at high speed based on residual unbalance in specified correction planes

The axial position of the correction planes and the balancing speed should be stated for each mode.

If the rotor is assessed in a balancing machine having its own instrumentation, this should be used throughout the test.

If the rotor is assessed in an overspeed or similar facility, the instrumentation and general installation of the rotor into the facility should be as stated in 3.1.1

When the above conditions have been met, the rotor should be run to speeds where it is possible to evaluate the magnitude and angular position of the residual unbalance distributions. (See 3.4.)

If requested by the customer, the rotor may be checked by a similar method to that referred to in 2.4.1.5. The check should be carried out for each balancing speed.

3.2 Choice of criteria

It is a usual practice when assessing the balance quality of a flexible rotor in the factory to consider the once-per-revolution vibration of the bearing pedestals or shaft in a balancing facility or test bed that reasonably approximates to the site conditions for which the rotor is intended (see, however, 3.3.4). This is the method described in 3.3.

Another practice is to assess the balance quality by considering the unbalance remaining in specified correction planes. This is the method described in 3.4. For class 2 rotors, this form of assessment may be made at low speed, without any necessity to use a high-speed balancing facility.

When employing either practice, it is sometimes possible, based on experience, to adjust acceptance levels to permit the use of facilities or installations that do not closely imitate site conditions and/or to allow for the final effect of coupling to another rotor in site.

The choice between the use of the method described in 3.3 and the use of that described in 3.4 shall be at the discretion of the manufacturer of the rotor.

3.3 Recommendations for vibration criteria in the balancing facility

If the final state of unbalance is to be assessed in terms of vibration criteria in the balancing facility, then these shall be chosen to ensure that the relevant vibration limits are satisfied on site.

There is a complex relationship between vibrations measured in the balancing facility and those obtained in the fully assembled machine at site, which is dependent on a number of factors. It should be noted that acceptance of machines on site is usually based on vibration criteria given, for example, in parts 1 to 4 of ISO 7919 or in ISO 10816-1. In most cases, this relationship has been derived for specific machine types by experience of balancing typical rotors in the same facility. Where such experience exists, it should be used as the basis for defining the permissible vibration in the balancing facility.

There may, however, be cases where such experience does not exist (e.g. a new balancing facility or rotors of substantially different design). The remainder of 3.3 relates to such cases and explains the permissible levels on once-per-revolution vibration which can be derived from the vibration severity specified in the product specification. If no product specification describing the acceptable running conditions on site exists, reference should be made as appropriate to ISO 10816-1 or parts 1 to 4 of ISO 7919.

3.3.1 General

Numerical values derived according to this subclause are not intended to serve as acceptance specifications but as guidelines. When used in this manner, gross deficiencies or unrealistic requirements may be avoided.

If due regard is paid to the recommended values, satisfactory running conditions can be expected. However, there may be cases when deviation from these recommendations become necessary.

These recommendations may also serve as the basis for more detailed investigations, for example when, in special cases, a more exact determination of the required balance quality is necessary.

3.3.2 Special cases and exceptions

There are exceptional cases where machinery is designed for special purposes and of necessity embodies features which inherently affect the vibration characteristics. Aircraft jet engines and derivatives of such engines for industrial purposes are one example. As engines of this type are designed to minimize weight, their main structures and bearing supports are considerably more flexible than in general industrial machinery. Special steps are taken in the design to accommodate undesirable effects resulting from such support flexibility, and extensive development testing is carried out to ensure that the vibration levels are safe and acceptable for the intended use of the engine.

For such cases as this, where the vibration characteristics have been shown to be acceptable by extensive testing before production units are delivered, it is not intended that the recommendations of 3.3 should apply.

3.3.3 Factors influencing machine vibration

The vibration resulting from the unbalance of the rotor is influenced by many factors, such as the mounting of the machine and the distortion of the rotor.

Were maximum permissible levels of vibration are stated in product specifications, they usually refer to total vibration *in situ* arising from all sources. The value quoted could therefore include the vibrations arising from a multiplicity of sources with different frequencies, and the manufacturer should consider what levels of vibration can be permitted from unbalance alone in order to keep within the permissible overall level of vibration.

3.3.4 Critical clearances and complex machine systems

Special attention should be paid to the levels of vibration and static displacement occurring at points of minimum clearance (for example, at process fluid seals) because of the greater likelihood of damage at these points than at others. It should be appreciated that the conditions on site may modify the mode shapes and thus the vibration levels at the points of measurements. (See 2.1.3.)

Rotors that are to be assembled in rigidly coupled multibearing systems (for example, steam turbine sets) need particular consideration in this respect. The magnitude of the unbalance and its distribution are important factors in such applications. (See annex A.)

3.3.5 Permissible vibrations in the balancing facility

Permissible vibration in the balancing facility can be expressed in two ways:

a) vibration on the bearing pedestal calculated from the permissible bearing vibration on site; or

b) shaft vibration calculated from the permisible shaft vibration on site.

In either case, the appropriate value can be determined using the following expression:

$$Y = K_2 \cdot K_1 \cdot K_0 \cdot X$$

where

X is the permissible total bearing or shaft vibration in the transverse horizontal or vertical direction for measurements taken on site in the service speed range as given in the product specification or the appropriate standard (e.g. ISO 10816-1 or parts 1 to 4 of ISO 7919);

Y is the corresponding permissible once-per-revolution bearing pedestal or shaft vibration in the balancing facility;

K_0 is the ratio of the permissible once-per-revolution vibration to the permissible total vibration ($K_0 \leqslant 1$);

K_1 is a conversion factor used if the rotor support and/or coupling systems differ from site conditions; it is defined as the ratio of the once-per-revolution measurements in the balancing facility (shaft and/or bearing pedestals) to similar measurements taken on the assembled machine on site (if not applicable, $K_1 = 1$);

K_2 is a conversion factor used if, in the balancing facility shaft, measurements are taken at locations other than those for which X is specified; its value depends on the modal characteristics of the rotor; if the measurement locations are the same, then $K_2 = 1$.

NOTES

12 The conversion relationship gives units for Y which are the same as those for X. In practice, it may subsequently be convenient to express Y in different units. For example, displacement instead of velocity.

13 The value of K_1 will often depend upon the direction of measurement.

14 For cases in which the measurement cannot be made at the same locations, K_2 may be determined analytically using a rotor dynamics model of the system. In general, the values for K_1 and K_2 vary significantly as functions of rotor speed.

The values of K_1 and K_2 may vary widely between one installation and another and will be speed-dependent. Some suggested values for K_0 and K_1 are shown in annex C. The value of K_2 needs to be established for each specific application. If a critical speed of a particular configuration of the rotor-bearing system coincides with the service speed, higher values of the relevant conversion factors have to be used.

It should be noted that, in practice, it is not essential that these conversion factors are determined in isolation, provided that a composite factor is available.

NOTE 15 Two examples of the use of these conversion factors are given in annex F.

In addition, it should be noted that modal amplification of the vibration will occur at critical speeds. Balancing practice is therefore usually directed not only towards satisfactory limitation of vibration within the service speed range, but also towards smooth passage through critical speeds below the maximum service speed. For critical speeds, it is especially difficult to establish quantitative criteria because of the need to allow for many important factors such as damping.

When the bending deflection during run-up is of concern, due to rotor/stator clearances or stresses, the bending of a rotor at critical speeds below service speed should be considered in terms of peak-to-peak displacement of that part of the rotor at which the displacement is of consequence.

3.4 Recommendations for establishing criteria for permissible residual unbalance in specified correction planes

3.4.1 General

The material in this subclause is an adaptation of criteria presented in parts 1 and 2 of ISO 1940, as it is not possible to derive the permissible unbalances for flexible rotors directly from existing documents concerned with the assessment of vibrations in rotating machinery. Usually, there is no simple relationship between rotor unbalance and machine vibration under service conditions. The amplitude of vibrations is influenced by many factors such as the vibrating mass of the machine casing and its foundations, the stiffness of the bearing and of the foundation, the proximity of the service speed to the various resonance frequencies, and the damping.

NOTE 16 Further details are given in ISO 10814.

Subclauses 3.4.2 and 3.4.3 establish guidelines for the required balance quality of flexible rotors. The values given are based on a limited amount of documented practical experience with the various classes of rotors. However, if due regard is paid to the recommended values, satisfactory running conditions can be expected. Nonetheless, the suggested levels and classifications are not yet completely verified, and deviations from these recommendations may be necessary in certain cases[3].

For a particular flexible rotor, the unbalance criteria used in this clause are derived from the criteria in parts 1 and 2 of ISO 1940 for an equivalent rigid rotor of the same general type.

Numerical values derived according to this subclause are not intended to serve as acceptance specifications but as guidelines. When used in this manner, gross deficiencies or unrealistic requirements may be avoided.

3.4.2 Criteria for the permissible residual unbalance of class 2 rotors

The residual unbalance for any completely assembled class 2 rotor should not exceed the residual unbalance recommended for an equivalent rigid rotor in ISO 1940-1.

In addition, for rotors of subclasses 2f, 2g and 2h, each component or, when applicable, each subassembly of components should be balanced to limits based on experience or those recommended in ISO 1940-1, applied to each component.

3.4.3 Criteria for the permissible residual unbalance of class 3 rotors

3.4.3.1 Class 3a

For rotors that are significantly affected by only the first modal residual unbalance, whatever their unbalance distribution, the residual unbalance should not exceed the following limits, expressed as percentages of the total residual unbalance recommended for an equivalent rigid rotor in parts 1 and 2 of ISO 1940 and based upon the highest service speed of the rotors:

a) the equivalent first modal residual unbalance should not exceed 60 %; and

b) if low-speed balancing is carried out initially, the total residual unbalance as a rigid rotor should not exceed 100 %.

3.4.3.2 Class 3b

For rotors that are significantly affected by only the first and second modal unbalances, whatever their unbalance distribution, the residual unbalance should not exceed the following limits, expressed as percentages of the total residual unbalance recommended for an equivalent rigid rotor in parts 1 and 2 of ISO 1940 and based upon the highest service speed of the rotor:

a) the equivalent first modal residual unbalance should not exceed 100 %; and

b) the equivalent second modal residual unbalance should not exceed 60 %; and

c) if low-speed balancing is carried out initially, the total residual unbalance as a rigid rotor should not exceed 100 %.

NOTE 17 Example 3 in annex F illustrates the calculation of these limits.

3.4.3.3 Class 3c

For rotors which are significantly affected by more than the first and second modal unbalances, no recommendations are available.

NOTES

18 A method for the experimental determination of the equivalent modal residual unbalances is described in annex D.

19 If the influence of overhung masses is significant, then the percentages given should be reviewed.

20 If, *in situ*, the service speed or service speed range is close to either the first or second flexural critical speed, these figures may require modification.

21 In the balancing facility, the proposed limits will not necessarily result in vibration magnitudes within normal limits in the speed range from 80 % to 120 % of any critical speed. If such amplified vibrations occur, it does not necessarily mean that more refined balancing is needed because, for example, damping in the balancing facility is often smaller than *in situ*.

22 When all relevant rotor flexural modes cannot be taken into account in the balancing facility (e.g. due to an insufficient number of correction planes), a decision has to be reached concerning which modes should be emphasized for balancing.

3) Reports of any such deviations are welcome. Comments should be directed to the national standards body in the country of origin for transmission to the secretariat of ISO/TC 108, and will be taken into account when preparing subsequent editions of this International Standard.

Annex A
(informative)

Cautionary notes concerning multispan rotors on site

A.1 Introduction

Unbalance is not the only cause of vibration, not even once-per-revolution vibration. Before undertaking balancing or related operations, due consideration should be given to identify the factors other than unbalance that are influencing the levels of vibration of the machine. Such factors include those given in A.2 to A.4.

This is particularly true in installations where two or more rotors are coupled together, such as a large steam turbine generator.

A.2 Bearing misalignment

Small parallel or angular misalignment of the rotor bearings can produce effects which cannot be removed by balancing. If these effects are present, the misalignment should be corrected prior to further assessment of the vibration of the machine (see also the second paragraph of A.3).

A.3 Radial and axial runout of coupling faces

There is no practical means of ensuring that large rotors can be coupled together without a small amount of radial and axial runout of the coupling faces between the mating halves of the coupling. These runouts can also produce vibration effects which cannot be satisfactorily corrected by balancing. Therefore, if the machine is not responding to balancing operations, the radial and axial runout of the coupling faces should be checked.

Where appropriate, errors should be corrected to lie within the tolerances which have been found to be satisfactory in practice for the size and type of machine under consideration, before attempting further balancing.

A.4 Bearing instability

Various forms of instability (e.g. fluid film whirl) may take place in the types of hydrodynamically lubricated bearings which are normally used in multispan flexible rotor systems.

The symptoms of these phenomena are well known, and it is necessary to ascertain whether such symptoms are present before attempting to improve the quality of running by balancing.

Discussions of such effects and possible remedial measures are outside the scope of this International Standard, beyond noting that the frequency is normally less than once per revolution.

Annex B
(informative)

Low-speed three-plane balancing of class 2d rotors

B.1 This annex is concerned with the low-speed balancing of rotors which have one central and two end correction planes, and in which the initial unbalance is uniformly distributed or increases linearly along the shaft.

Such rotors can be satisfactorily balanced on a low-speed balancing machine provided that an assessment can be made of the proportion of the total unbalance of the rotor which should be corrected at the central plane.

This annex provides a method whereby the balance correction in three planes may be calculated from the initial unbalance measured in two planes. The vector sum of the forces and moments created by the three unbalance corrections $\vec{U}_1$, $\vec{U}_2$ and $\vec{U}_3$ about a given point on the rotor must equal those caused by the initial unbalances, $\vec{U}_L$ and $\vec{U}_R$, about the same point.

This annex describes a method for three-plane balancing of rotors which meet all of the following conditions:

a) single-span rotor with no significant overhang;

b) uniform or linear distribution of unbalance;

c) uniform bending flexibility or the rotor along its length;

d) symmetrical position of end correction planes about midspan;

e) continuous service speeds below and not significantly approaching the second critical speed.

B.2 It can be shown that the initial unbalance will be completely corrected up to, and including, its first modal component when the vector relationships

$$\vec{U}_1 = \vec{U}_L - 0{,}5H\left(\vec{U}_R + \vec{U}_L\right)$$

$$\vec{U}_2 = H\left(\vec{U}_L + \vec{U}_R\right)$$

$$\vec{U}_3 = \vec{U}_R - 0{,}5H\left(\vec{U}_R + \vec{U}_L\right)$$

are satisfied, where H is the ratio of the central correction to the initial static unbalance.

It should be noted that $\vec{U}_L$, $\vec{U}_R$, $\vec{U}_1$, $\vec{U}_2$ and $\vec{U}_3$ are vectors.

Values of H are presented graphically in figure B.1 as a function of z/l, where z is the distance from the left-hand bearing of the correction plane 1 and l is the bearing span (shaft length).

It should be noted that H is zero when $z/l = 0{,}22$, which indicates that in this case the centre plane is no longer needed and the procedure has become a two-plane balancing procedure, usually called "quarter-point balancing". For values of z/l greater than 0,22, the correction in the centre plane is on the opposite side of the shaft.

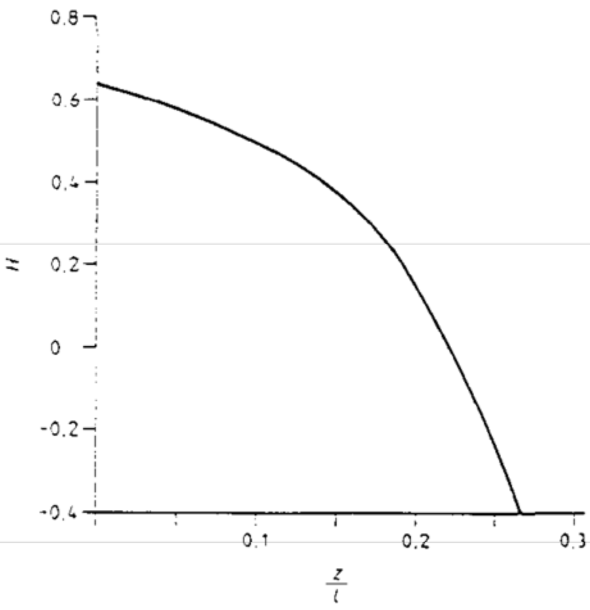


Figure B.1 — Graphical presentation for determination of H

Annex C
(informative)

Conversion factors[4]

Machinery is classified as follows (see table C.1).

I: Individual parts of engines and machines, integrally connected to the complete machine in its normal operating condition.

II: Medium-sized machines without special foundations and rigidly mounted engines or machines (up to 300 kW) on special foundations.

III: Large prime movers and other large machines with rotating masses mounted on rigid and heavy foundations that are relatively stiff in the direction of vibration measurement.

IV: Large prime movers and other large machines with rotating masses mounted on foundations that are relatively soft in the direction of vibration measurement.

Table C.1 — Suggested conversion factor ranges (see 3.3.5)

Machinery classification	Typical machines	K_0	K_1		
			Bearing support absolute	Shaft absolute	Shaft relative
I	Superchargers	1,0	0,6 to 1,6	1,6 to 5,0	1,0 to 3,0
	Small electric motors up to 15 kW	1,0			
II	Paper-making machines	0,7 to 1,0			
	Medium-sized electric machines, 15 kW to 75 kW	0,7 to 1,0			
	Electrical machines up to 300 kW on special foundations	0,7 to 1,0			
	Compressors	0,7 to 1,0			
	Small turbines	1,0			
III	Large electric motors	0,7 to 1,0			
	Pumps	0,7 to 1,0			
	2-Pole generators	0,8 to 1,0			
	Turbines and multipole generators	0,9 to 1,0			
IV	Gas turbines (but see 3.3.2)	1,0			
	2-Pole generators	0,8 to 1,0			
	Turbines and multipole generators	0,9 to 1,0			

4) Users are recommended to compare the above values with their own experience. Comments on the results or such comparisons are welcome and should be directed to the national standards body in the country of origin for transmission to the secretariat of ISO/TC 108.

Annex D
(informative)

Experimental determination of equivalent modal residual unbalances

The following procedure may be used to determine the effect of unbalance in specific modes in order to assess the equivalent modal residual unbalance remaining in the rotor.

D.1 Mount the rotor whose unbalance is to be measured in a high-speed hard-bearing balancing machine or other high-speed test facility.

D.2 Run the rotor to some safe speed approaching the first flexural critical speed and note readings of bearing vibrations or forces.

D.3 Add a trial unbalance to the rotor. The unbalance should be sufficient to show a significant effect and should be placed axially where it will have the maximum effect on the first mode. This will generally be towards the middle of the rotor. Take readings of bearing vibration or forces at the same speed as in D.2.

D.4 From the readings obtained in D.2 and D.3, compute vectorially the equivalent first modal unbalance. This can be done graphically by the construction given in annex G, in this case with the single unbalance mass forming the trial mass set. The magnitude of the equivalent first modal unbalance is

$$\text{trial unbalance} \times \frac{AO}{AB}$$

D.5 Remove the trial unbalance.

D.6 Run the rotor to some safe speed approaching the second flexural critical speed, provided this is lower than the maximum safe operating speed. Note readings of bearing vibrations or forces.

D.7 Add a trial unbalance to the rotor. This should be sufficicient to show a significant effect and should be placed axially where it will have the maximum effect on the second mode. Take readings of bearing vibrations or forces at the same speed as in D.6.

D.8 From the readings obtained in D.6 and D.7, compute vectorially the equivalent second modal unbalance. The graphical procedure of D.4 may be used in this case.

D.9 Remove the trial unbalance.

D.10 Continue the above operations for successive modes until the equivalent modal unbalances in all significant modes have been determined.

NOTES

23 The procedure given in this annex assumes that the vibration measured at a speed close to a critical speed is predominantly in the corresponding mode. Sometimes it may not be possible to run close to the critical speeds of some of the significant modes and thus readily identify the corresponding modal components. In these cases, further procedures are necessary to separate the individual modal components.

24 When determining the equivalent modal unbalances, it may be desirable to use a set of trail unbalances in order to pass safely through the low critical speeds.

Annex E
(informative)

Procedure to determine if a rotor is rigid or flexible

This annex describes procedures that may be employed to determine whether a rotor is rigid or flexible. If it is found that a rotor falls into the rigid category, then it can be balanced using a low-speed balancing procedure. In general, flexible rotors need to be balanced at high speeds using procedures such as those given in 2.4. Exceptions include class 2 rotors, which by definition although flexible are on the borderline, and low-speed balancing may be adequate using the special procedures given in 2.3.

The physical appearance of a rotor is insufficient to determine whether a rotor falls into the rigid or flexible categories for balancing purposes. For example, a rotor with a rigid appearance may, if it operates at high speeds, approach or pass through a critical speed involving significant bending of the rotor and therefore requires high-speed balancing. Conversely, a rotor that looks flexible is a rigid rotor for balancing purposes if its highest service speed is well below the first flexural critical speed.

E.1 Determination of whether a rotor is rigid or flexible

One or more of the following may be used to ascertain whether a rotor is rigid or flexible, and thereby determine the balancing method to be adopted.

E.1.1 Consult the rotor manufacturer or the user for a definition of the rotor class and a recommended balancing procedure.

E.1.2 If the first flexural critical speed exceeds the maximum service speed by at least 30 %, then the rotor can often be considered rigid for balancing purposes.

E.1.3 Alternatively, the following test sequence may be performed. Balance the rotor at low speed in two correction planes in accordance with the procedures specified in ISO 1940-1.

Mount the rotor in a facility that is capable of rotating the rotor to at least service speed and that has stiffness and damping of the bearings and their supports similar to the service installation. Accelerate the rotor gradually to service speed, taking care that vibration at all times stays within safe limits. Record vibration readings as a function of speed during the acceleration and subsequent deceleration.

If no significant change in vibration occurs as a function of speed, then the rotor is either rigid, or is flexible with low levels of modal unbalance. To determine which of these possibilities apply, perform the flexibility test defined in E.2.

If a significant change in vibration occurs during acceleration or deceleration, one or more of the following possibilities exist:

a) the rotor is flexible;

b) the rotor is rigid but flexibly supported;

c) the rotor has components that shift location significantly as a function of speed or temperature.

To help discriminate between these possibilities, accelerate the rotor again to service speed and then check if the readings during deceleration to zero speed repeat those of the prior deceleration run. If they do, the rotor has settled. Next perform the flexibility test given in E.2 to determine if the rotor is rigid or flexible.

NOTE 25 Settlement may have occurred by taking the rotor to its service speed or beyond by permanently "seating" components due to centrifugal force. For example, generator and motor rotors frequently require a settlement run to enable the copper windings and support systems to move radially outward to their final position.

If the readings do not repeat, the rotor's unbalance is variable and the rotor cannot generally be balanced within tolerance until this problem is corrected.

E.2 Rotor flexibility test

Add a mass to the centre of the rotor, or to an available position where it may be expected to cause high rotor vibration. Accelerate the rotor to service speed, taking care that vibration at all times stays within safe limits. If the vibration magnitudes become excessive during the acceleration, reduce the magnitude of the mass and repeat the process. Measure the vibration vector at the same speed and location as in E.1.3. Determine the effect of the mass on the vibration level by vectorially subtracting the vibration vector recorded in E.1.3 from the new reading. Denote the result by $\vec{A}$.

Stop the rotor and remove the mass. Install two masses at the same angular position as the central mass that was removed. These masses should be placed at the rotor end planes near the rotor journals. The masses should be chosen to provide the same quasi-static unbalance in the plane of the single test mass without introducing any additional couple unbalance. Accelerate the rotor to service speed again, take another reading and determine the effect of the two masses on the rotor by subtracting the vector from E.1.3 from the reading. Denote this vector by $\vec{B}$.

E.3 Evaluation of flexibility test data

Compute the magnitude of the vector $(\vec{A} - \vec{B})$. If this magnitude when divided by the magnitude of vector $\vec{A}$ is less than 0,2, the rotor can usually be considered rigid for balancing purposes. Conversely, if this ratio is 0,2 or greater, then the rotor should be treated as a quasi-rigid or a flexible rotor.

If sufficient rotor system modelling data are available, it is possible to generate analytically the data needed for calculating the ratio in E.3, thereby avoiding the need to perform the flexibility test. Particular care must be taken with this approach to model accurately the stiffness and damping characteristics of the rotor/support system.

Annex F
(informative)

Examples of the use of conversion factors

Example 1 (refers to 3.3)

Rotor	Steam turbine rotor (Machinery classification IV)
Service speed	3 000 rev/min
Permissible vibration severity (rms) on site, X	2,5 mm/s [value for bearing pedestal vibration from the product specification[1]]
Location in factory	Balancing facility with less-stiff supports than on site.
Measurement locations	On bearing pedestals
Conversion factors, values from table C.1 according to the experience of the manufacturer	$K_0 = 0{,}9$ $K_1 = 1{,}3$ $K_2 = 1{,}0$ (not applicable)
Permissible once-per-revolution vibration (rms) on the bearing pedestal in the balancing facility	$Y = K_2 \cdot K_1 \cdot K_0 \cdot X$ $= 1{,}0 \times 1{,}3 \times 0{,}9 \times 2{,}5 = 2{,}93$ mm/s
At the service speed, this will be equivalent to a once-per-revolution displacement (peak-to-peak) of:	$2{,}93 \times \frac{60}{2\pi} \times \frac{2\sqrt{2}}{3\,000} \times 10^3 = 26\ \mu m$

1) If the product specification is not stated in terms of "vibration severity" or no product specification is available, then see, for example, ISO 10816-1.

Example 2 (refers to 3.3)

Rotor	660 MW generator rotor
Service speed	3 000 rev/min
Permissible shaft relative vibration (peak-to-peak displacement) on site, X	80 µm close to the journals (limit of zone A from ISO 7919-2)
Location in factory	Balancing facility with less-stiff support than on site
Measurement locations	Close to the journals and at the coupling
Conversion factors according to the experience of the manufacturer	$K_0 = 0,9$ $K_1 = 1,3$ $K_2 = 1,0$ for measurements close to the journals $K_2 = 4,0$ for measurements at the coupling (obtained by consideration of rotor modal characteristics)
Permissible once-per-revolution shaft relative vibration (peak-to-peak displacement) in the balancing facility: a) close to the journals: b) at the coupling:	 $Y = 1,0 \times 1,3 \times 0,9 \times 80 = 94$ µm $Y = 4,0 \times 1,3 \times 0,9 \times 80 = 374$ µm

Example 3 (refers to 3.4.3.2)

Rotor	Turbo compressor (G 2,5 from ISO 1940-1)
Rotor class	3b
Service speed	15 000 rev/min
Rotor mass	1 000 kg
Assume low-speed balancing in two planes close to bearings: total residual unbalance for an equivalent rigid rotor in accordance with ISO 1940-1 permissible equivalent first modal unbalance (100 %) permissible equivalent second modal unbalance (60 %) total residual unbalance for the rigid rotor	 $1,60 \ \frac{\text{g·mm}}{\text{kg}} \times 1\ 000 \text{ kg} = 1\ 600 \text{ g·mm}$ 1 600 g·m 960 g·mm 1 600 g·mm (800 g·mm per plane)

Annex G
(informative)

Method of computation of unbalance correction

This annex gives a suggested method of computation of unbalance correction by observation of the effect of a trial mass set.

Let vector OA in figure G.1 represent the initial vibration plotted to some arbitrary reference angle.

Let vector OB represent the resultant vibration, at the same speed and plotted to the same reference, when a trial mass set is added to the rotor.

Then the "effect" of the trial mass set is represented in amplitude and angle by the vector AB.

Therefore in order to nullify the original vibration, the trial mass set should be moved through the angle BAO and each mass in the set adjusted in size in the ratio AO/AB.

Figure G.1 — Vectorial effect of a trial mass set

Annex H
(informative)

Definitions from ISO 1925:1990 relating to flexible rotors

H.1 (rotor) flexural critical speed: Speed of a rotor at which there is maximum flexure of the rotor and where that flexure is significantly greater than the motion of the journals.

H.2 rigid-rotor-mode critical speed: Speed of a rotor at which there is maximum motion of the journals and where that motion is significantly greater than the flexure of the rotor.

H.3 (rotor) flexural principal mode: For undamped rotor/bearing systems, that mode shape which the rotor takes up at one of the (rotor) flexural critical speeds.

H.4 multiplane balancing: As applied to the balancing of flexible rotors, any balancing procedure that requires unbalance correction in more than two correction planes.

H.5 modal balancing: Procedure for balancing flexible rotors in which unbalance corrections are made to reduce the amplitude of vibration in the separate significant principal flexural modes to within specified limits.

H.6 nth modal unbalance: That unbalance which affects only the nth principal mode of the deflection configuration of a rotor/bearing system.

NOTE 26 The nth modal unbalance is not a single unbalance but an unbalance distribution $u(z)$ in the nth principal mode. It can be mathematically represented with respect to its effect on the nth principal mode by a single unbalance vector $\bar{U}_n$ obtained from the formula

$$\bar{U}_n = \int_0^L u(z)\phi_n(z)\mathrm{d}z$$

where

$\phi_n(z)$ is the mode function;

L is the rotor length;

$u(z)$ is the unbalance distribution.

H.7 equivalent nth modal unbalance: The minimum single unbalance $\bar{U}_{ne}$, equivalent to the nth modal unbalance in its effect on the nth principal mode of the deflection configuration.

NOTES

27 There exists the relation $\bar{U}_n = \bar{U}_{ne}\phi_n(z_e)$, where $\phi_n(z_e)$ is the mode function value $z = z_e$, the axial coordinate of the transverse plane where $\bar{U}_{ne}$ is applied.

28 A set of masses distributed in an appropriate number of correction planes and so proportioned that the mode under consideration will be affected, may be called the equivalent nth modal unbalance set.

29 An equivalent nth modal unbalance will affect some modes other than the nth mode.

H.8 modal balance tolerance: With respect to mode, that amount of equivalent modal unbalance that is specified as the maximum below which the state of unbalance in that mode is considered to be acceptable.

H.9 multiple-frequency vibration: Vibration at a frequency corresponding to an integral multiple of the rotational speed.

NOTE 30 This vibration may be caused by anisotropy of the rotor, non-linear characteristics of the rotor/bearing system, or other causes.

H.10 thermally induced unbalance: That change in condition exhibited by a rotor if its state of unbalance is significantly altered by its changes in temperature.

NOTE 31 The change in condition may be permanent or temporary.

H.11 low-speed balancing (relating to flexible rotors): Procedure of balancing at a speed where the rotor to be balanced can be considered to be rigid.

H.12 high-speed balancing (relating to flexible rotors): Procedure of balancing at a speed where the rotor to be balanced cannot be considered to be rigid.

Annex J
(informative)

Bibliography

[1] ISO 3945:1985, *Mechanical vibration of large rotating machines with speed range from 10 to 200 r/s — Measurement and evaluation of vibration severity in situ.*

[2] ISO 9001:1994, *Quality systems — Model for quality assurance in design, development, production, installation and servicing.*

[3] ISO 10814:—[5], *Susceptibility and sensitivity of machines to unbalance.*

5) To be published.