BS ISO 21940-12:2016

BSI Standards Publication

Mechanical vibration — Rotor balancing

Part 12: Procedures and tolerances for rotors with flexible behaviour

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National foreword

This British Standard is the UK implementation of ISO 21940-12:2016. It supersedes BS ISO 11342:1998 which is withdrawn.

The UK participation in its preparation was entrusted to Technical Committee GME/21/5, Mechanical vibration, shock and condition monitoring - Vibration of machines.

A list of organizations represented on this committee can be obtained on request to its secretary.

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Mechanical vibration — Rotor balancing —

Part 12:
Procedures and tolerances for rotors with flexible behaviour

Vibrations mécaniques — Équilibrage des rotors —

Partie 12: Modes opératoires et tolérances pour les rotors à comportement flexible

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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.

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation on the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the WTO principles in the Technical Barriers to Trade (TBT) see the following URL: Foreword - Supplementary information

The committee responsible for this document is ISO/TC 108, *Mechanical vibration, shock and condition monitoring*, Subcommittee SC 2, *Measurement and evaluation of mechanical vibration and shock as applied to machines, vehicles and structures*.

This first edition of ISO 21940-12 cancels and replaces ISO 11342:1998, which has been technically revised. The main changes are deletion of the terms and definitions which were transferred to ISO 21940-2 and deletion of former Annex F which is a duplication of a part of D.1. It also incorporates the Technical Corrigendum ISO 11342:1998/Cor.1:2000.

ISO 21940 consists of the following parts, under the general title *Mechanical vibration — Rotor balancing*:

— *Part 11: Procedures and tolerances for rotors with rigid behaviour*[1)]

— *Part 12: Procedures and tolerances for rotors with flexible behaviour*[2)]

— *Part 13: Criteria and safeguards for the in-situ balancing of medium and large rotors*[3)]

— *Part 14: Procedures for assessing balance errors*[4)]

— *Part 21: Description and evaluation of balancing machines*[5)]

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1) Revision of ISO 1940-1:2003 + Cor.1:2005, *Mechanical vibration — Balance quality requirements for rotors in a constant (rigid) state — Part 1: Specification and verification of balance tolerances*

2) Revision of ISO 11342:1998 + Cor.1:2000, *Mechanical vibration — Methods and criteria for the mechanical balancing of flexible rotors*

3) Revision of ISO 20806:2009, *Mechanical vibration — Criteria and safeguards for the in-situ balancing of medium and large rotors*

4) Revision of ISO 1940-2:1997, *Mechanical vibration — Balance quality requirements of rigid rotors — Part 2: Balance errors*

5) Revision of ISO 2953:1999, *Mechanical vibration — Balancing machines — Description and evaluation*

— *Part 23: Enclosures and other protective measures for the measuring station of balancing machines*[6)]

— *Part 31: Susceptibility and sensitivity of machines to unbalance*[7)]

— *Part 32: Shaft and fitment key convention*[8)]

The following part is under preparation:

— *Part 2: Vocabulary*[9)]

6) Revision of ISO 7475:2002, *Mechanical vibration — Balancing machines — Enclosures and other protective measures for the measuring station*

7) Revision of ISO 10814:1996, *Mechanical vibration — Susceptibility and sensitivity of machines to unbalance*

8) Revision of ISO 8821:1989, *Mechanical vibration — Balancing — Shaft and fitment key convention*

9) Revision of ISO 1925:2001, *Mechanical vibration — Balancing — Vocabulary*

Introduction

The aim of balancing any rotor is to achieve satisfactory running when installed *in-situ*. In this context, "satisfactory running" means that not more than an acceptable magnitude of vibration is caused by the unbalance remaining in the rotor. In the case of a rotor with flexible behaviour, it also means that not more than an acceptable magnitude of deflection occurs in the rotor at any speed up to the maximum service speed.

Most rotors are balanced in manufacture prior to machine assembly because afterwards, for example, there might 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 *in-situ* is the aim, the balance quality of the rotor is usually initially assessed in a balancing machine. Satisfactory running *in-situ* is, in most cases, judged in relation to vibration from all causes, while in the balancing machine, primarily, once-per-revolution effects are considered.

This part of ISO 21940 classifies rotors in accordance with their balancing requirements and establishes methods of assessment of residual unbalance.

This part of ISO 21940 also shows how criteria for use in the balancing machine can 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 part of ISO 21940 shows how they can be derived from ISO 10816 and ISO 7919 if desired in terms of vibration, or from ISO 21940-11, if desired in terms of permissible residual unbalance. ISO 21940-11 is concerned with the balance quality of rotating rigid bodies and is not directly applicable to rotors with flexible behaviour because rotors with flexible behaviour can undergo significant bending deflection. However, in this part of ISO 21940, methods are presented for adapting the criteria of ISO 21940-11 to rotors with flexible behaviour.

There are situations in which an otherwise acceptably balanced rotor experiences an unacceptable vibration level *in situ*, owing to resonances in the support structure. A resonance or near resonance condition in a lightly damped structure can result in excessive vibratory response to a small unbalance. In such cases, it can be more practicable to alter the natural frequency or damping of the structure rather than to balance to very low levels, which might not be maintainable over time (see also ISO 21940-31).

Mechanical vibration — Rotor balancing —

Part 12: Procedures and tolerances for rotors with flexible behaviour

1 Scope

This part of ISO 21940 presents typical configurations of rotors with flexible behaviour in accordance with their characteristics and balancing requirements, describes balancing procedures, specifies methods of assessment of the final state of balance, and establishes guidelines for balance quality criteria.

This part of ISO 21940 can also serve as a basis for more involved investigations, e.g. when a more exact determination of the required balance quality is necessary. If due regard is paid to the specified methods of manufacture and balance tolerances, satisfactory running conditions can be expected.

This part of ISO 21940 is not intended to serve as an acceptance specification for any rotor, but rather to give indications of how to avoid gross deficiencies and unnecessarily restrictive requirements.

Structural resonances and modifications thereof lie outside the scope of this part of ISO 21940.

The methods and criteria given are the result of experience with general industrial machinery. It is possible that they are not directly applicable to specialized equipment or to special circumstances. Therefore, in some cases, deviations from this part of ISO 21940 are possible.

2 Normative references

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

ISO 1925[10)], *Mechanical vibration — Balancing — Vocabulary*

ISO 2041, *Mechanical vibration, shock and condition monitoring — Vocabulary*

ISO 21940-11[11)], *Mechanical vibration — Rotor balancing — Part 11: Procedures and tolerances for rotors with rigid behaviour*

ISO 21940-14, *Mechanical vibration — Rotor balancing — Part 14: Procedures for assessing balance errors*

ISO 21940-32, *Mechanical vibration — Rotor balancing — Part 32: Shaft and fitment key convention*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 1925 and ISO 2041 apply.

10) To become ISO 21940-2 when revised.

11) To be published.

4 Fundamentals of dynamics and balancing of rotors with flexible behaviour

4.1 General

Rotors with flexible behaviour normally require multiplane balancing at high speed. Nevertheless, under certain conditions, a rotor with flexible behaviour can also be balanced at low speed. For high-speed balancing, 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 is given here. In most practical balancing applications, the method adopted is normally a combination of both approaches, often incorporated into a computer package.

4.2 Unbalance distribution

The rotor design and method of construction can significantly influence the magnitude and distribution of unbalance along the rotor axis. 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 unbalances arising from shrink-fitted discs, couplings, etc.

Since the unbalance distribution along a rotor axis 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 rotor with flexible behaviour than in a rotor with rigid behaviour because it determines the degree to which any flexural mode is excited. The effect of unbalance at any point along a rotor depends on the mode shapes of the rotor.

The correction of unbalance in transverse planes along a rotor other than those in which the unbalance occurs can induce vibrations at speeds other than that at which the rotor was originally balanced. These vibrations can exceed specified tolerances, particularly at, or near, the flexural resonance speeds. Even at the same speed, such correction can induce vibrations if the flexural mode shapes *in-situ* differ from those dominating during the balancing process.

Rotors should be checked for straightness, and where necessary corrected prior to high-speed balancing, since a rotor with an excessive bend or bow will result in a compromise balance, which can lead to poor performance in service.

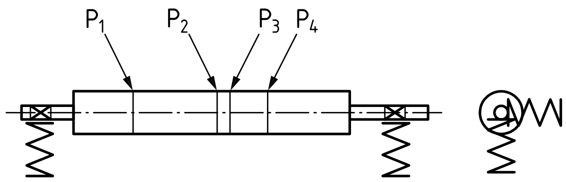
In addition, some rotors which become heated during operation are susceptible to thermal bows which can lead to changes in the unbalance. If the rotor unbalance changes significantly from run to run, it might be impossible to balance the rotor within tolerance.

4.3 Mode shapes of rotors with flexible behaviour

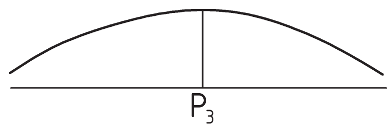
If the effect of damping is neglected, the modes of a rotor are the flexural principal modes and, in the special case of a rotor supported in bearings which have the same stiffness in all radial directions, are rotating plane curves. Typical shapes for the three lowest principal modes for a simple rotor supported in flexible bearings near to its ends are illustrated in Figure 1.

For a damped rotor and bearing system, the flexural modes can be space curves rotating about the shaft axis, especially in the case of substantial damping, arising perhaps from fluid-film bearings. Possible damped first and second modes are illustrated in Figure 2. In many cases, the damped modes can be treated approximately as principal modes and, hence, regarded as rotating plane curves.

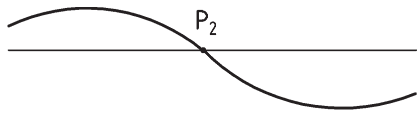
It is important to note that the form of the mode shapes and the response of the rotor to unbalances are strongly influenced by the dynamic properties and axial locations of the bearings and their supports.



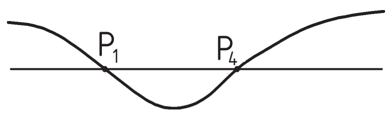
a) Typical rotor



b) First flexural mode



c) Second flexural mode



d) Third flexural mode

Key

P_1, P_2, P_4 nodes

P_3 antinode

Figure 1 — Simplified mode shapes for rotors with flexible behaviour on flexible supports

4.4 Response of a rotor with flexible behaviour 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 resonance speed, it is usually the mode associated with this resonance speed which dominates the deflection of the rotor. The degree to which large amplitudes of rotor deflection occur under these circumstances is influenced mainly by the following:

a) the magnitude of the modal unbalances;

b) the proximity of the associated resonance speeds to the running speeds;

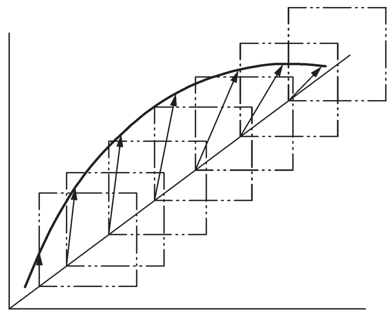
c) the amount of damping in the rotor and support system.

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 part of ISO 21940.

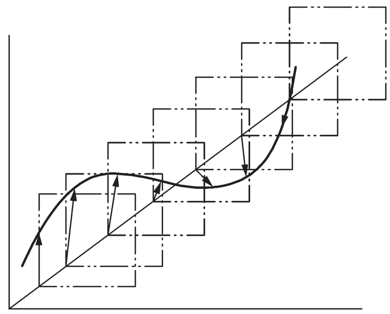
The modal unbalances for a given unbalance distribution are a function of the rotor modes. Moreover, for the simplified rotor shown in Figure 1, the effect produced in a particular mode by a given correction depends on the ordinate of the mode shape curve at the axial location of the correction: maximum effect near the antinodes, minimum effect near the nodes. Consider an example in which the curves of Figure 1 b) to d) are mode shapes for the rotor in Figure 1 a). A correction mass in plane P_3 has the maximum effect on the first mode, while its effect on the second mode is small.

A correction mass in plane P_2 will produce no response at all on the second mode, but will influence both the other modes.

Correction masses in planes P_1 and P_4 will not affect the third mode, but will influence both the other modes.



a) First mode



b) Second mode

Figure 2 — Examples of possible damped mode shapes

4.5 Aims of balancing rotors with flexible behaviour

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 economical, but still satisfies the needs of the user.

Balancing is intended to achieve acceptable magnitudes of machinery vibration, shaft deflection and forces applied to the bearings caused by unbalance.

The ideal aim in balancing rotors with flexible behaviour 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 moment 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 unbalance can be distributed along the length of the rotor, but the balancing process is usually achieved by adding or removing masses in a limited number of correction planes. Thus, there is invariably some distributed residual unbalance after balancing, which is assumed to be within tolerance for the affected mode shapes.

It is necessary to reduce vibrations or oscillatory forces caused by the residual unbalance to acceptable magnitudes over the service speed range. Only in special cases is it sufficient to balance rotors with flexible behaviour for a single speed. It should be noted that a rotor, balanced satisfactorily for a given service speed range, can still experience excessive vibration if it has to run through a resonance speed to reach its service speed. Therefore, for passing through resonance speeds, the allowable vibration may be greater than that permissible at service speed.

Whatever balancing technique is used, the final goal is to apply unbalance correction distributions to minimize the unbalance effects at all speeds up to the maximum service speed, including start up and shut down and possible overspeed. In meeting this objective, it might be necessary to allow for the influence of modes with resonance speeds above the service speed range.

4.6 Provision for correction planes

The exact number of axial locations along the rotor that are needed depends to some extent on the particular balancing procedure which is adopted. For example, centrifugal compressor rotors are sometimes balanced as an assembly 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 is influenced by n flexural resonance speeds, which possibly include resonance speeds above the maximum service speed, then usually if low-speed balancing is carried out, $n + 2$ correction planes are needed along the rotor, if not, n planes can be used.

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

4.7 Coupled rotors

When two rotors are coupled together, the complete unit has a series of resonance speeds and mode shapes. In general, these speeds are neither equal nor simply related to the resonance 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. Ideally, 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.

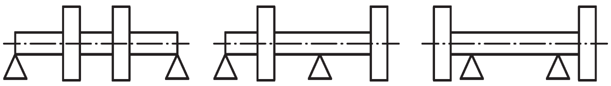
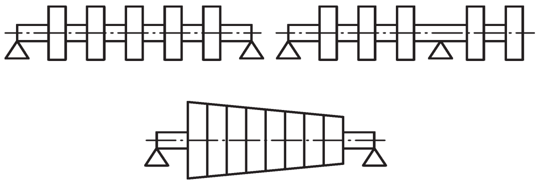
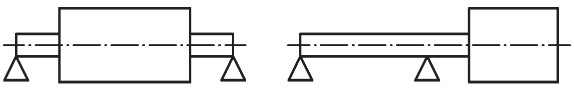
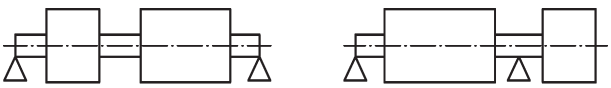
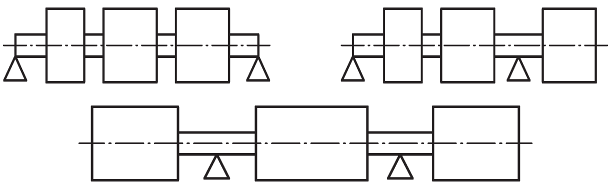
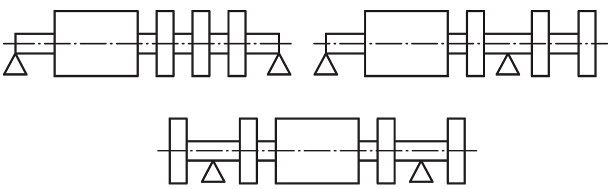
For practical purposes, in most cases, each rotor is balanced separately as an uncoupled shaft and this procedure normally ensures satisfactory operation of the coupled rotors. The degree to which this technique is practicable depends, for example, on the mode shapes and the resonance speeds of the uncoupled and coupled rotors, the distribution of unbalance, the type of coupling and on the bearing arrangement of the shaft train. If further balancing *in-situ* is required, refer to Annex A.

5 Rotor configurations

Typical rotor configurations are shown in Table 1, their characteristics outlined and the recommended balancing procedures listed. Table 1 gives concise descriptions of the rotor characteristics. Full descriptions of these characteristics and requirements are given in the corresponding procedures in Clauses 6 and 7. These procedures are listed in Table 2.

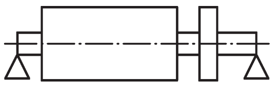
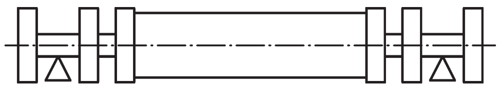
Sometimes, a combination of balancing procedures can be advisable. If more than one balancing procedure could be used, they are listed in the sequence of increasing time and cost. Rotors of any configuration can always be balanced at multiple speeds (see 7.3) or sometimes, under special conditions, be balanced at service speed (see 7.4) or at a fixed speed (see 7.5).

Table 1 — Rotors with flexible behaviour

Configuration	Rotor characteristics	Recommended balancing procedure[a]
1.1 Discs	**Elastic shaft without unbalance, rigid disc(s)**	
	**Single disc** — perpendicular to shaft axis — with axial runout	 A; C B; C
	**Two discs** — perpendicular to shaft axis — with axial runout — at least one removable — integral	 B; C B + C, E G
	**More than two discs** — all discs removable, except one — integral	 B + C, D, E G
1.2 Rigid sections	**Elastic shafts without unbalances, rigid sections**	
	**Single rigid section** — removable — integral	 B; C; E B
	**Two rigid sections** — at least one removable — integral	 B + C; E G
	**More (than two) rigid sections** — all discs removable, except one — integral	 B + C; E G
1.3 Discs and rigid sections	**Elastic shaft without unbalance, rigid discs and sections**	
	**One each** — at least one part removable — integral	 B + C; E G
	**More parts** — all discs removable, except one — integral	 B + C; E G

[a] See Table 2 for explanations of procedures A to G; two additional balancing procedures H and I can be used under special circumstances, see 7.4 and 7.5.

Table 1 *(continued)*

Configuration	Rotor characteristics	Recommended balancing procedure[a]
1.4 Rolls	**Mass, elasticity and unbalance distribution along the rotor**	
	— under special conditions (see 6.5.6) — in general	F G
1.5 Rolls and discs/rigid sections	**Flexible roll, rigid discs, rigid sections**	
 	— discs/rigid sections removable — under special conditions (see 6.5.6) — in general — integral	 C + F; E + F G G
1.6 Integral rotor	**Mass, elasticity and unbalance distribution along the rotor**	
	Main parts with unbalances not detachable	G

[a] See Table 2 for explanations of procedures A to G; two additional balancing procedures H and I can be used under special circumstances, see 7.4 and 7.5.

Table 2 — Balancing procedures

Procedure	Description	Subclause
Low-speed balancing		
A	Single-plane balancing	6.5.1
B	Two-plane balancing	6.5.2
C	Individual component balancing prior to assembly	6.5.3
D	Balancing subsequent to controlling initial unbalance	6.5.4
E	Balancing in stages during assembly	6.5.5
F	Balancing in optimum planes	6.5.6
High-speed balancing		
G	Multiple speed balancing	7.3
H	Service speed balancing	7.4
I	Fixed speed balancing	7.5

6 Procedures for balancing rotors with flexible behaviour at low speed

6.1 General

Low-speed balancing is generally used for rotors with rigid behaviour and high-speed balancing is generally used for rotors with flexible behaviour. Procedures to determine whether a rotor shows rigid or flexible behaviour are described in Annex E. However, with the use of appropriate procedures, it is possible under some circumstances to balance rotors with flexible behaviour at low speed so as to ensure satisfactory running when the rotor is installed in its final environment. Otherwise, rotors with flexible behaviour require the use of a high-speed balancing procedure.

Most of the procedures explained in this subclause require some information regarding the axial distribution of unbalance.

In some cases where a gross unbalance can occur in a single component, it can 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.

Certain rotors contain a number of individual parts which are mounted concentrically (e.g. blades, coupling bolts, pole pieces). These parts can 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 procedures. If these parts need to be assembled after balancing, they should be arranged in balanced sets.

Some rotors are made of individual components (e.g. turbine discs). In these cases, it is important to recognize that the assembly process can produce changes in the shaft geometry (e.g. shaft runout) and further changes can occur during high-speed service.

6.2 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. When a rotor is composed of two or more separate components that are distributed axially, there can be more than two transverse planes of unbalance.

6.3 Service speed of the rotor

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

6.4 Initial unbalance

The process of balancing a rotor with flexible behaviour in a low-speed balancing machine is an approximate one. The magnitude and distribution of initial unbalance are major factors determining the degree of success that can be expected.

For rotors in which the axial distribution of initial unbalance is known and appropriate correction planes are available, the permissible initial unbalance is limited only by the amount of correction possible in the correction planes.

For rotors in which the distribution of the initial unbalance is not known, there are no generally applicable low-speed balancing methods. However, sometimes the magnitude can be controlled by the pre-balancing of individual components. In these cases, the low-speed initial unbalance can be used as a measure of the distribution of unbalance.

6.5 Low-speed balancing procedures

6.5.1 Procedure A — Single-plane balancing

If the initial unbalance is principally contained in one transverse plane and the correction is made in this plane, then the rotor is balanced for all speeds.

6.5.2 Procedure B — Two-plane balancing

If the initial unbalance is principally concentrated in two transverse planes and the corrections are made in these planes, then the rotor is balanced for all speeds.

If the unbalance in the rotor is distributed within a substantially rigid section of the rotor and the unbalance correction is also made within this section, then the rotor is balanced for all speeds.

6.5.3 Procedure C — Individual component balancing prior to assembly

Each component, including the shaft, shall be low-speed balanced before assembly in accordance with ISO 21940-11. In addition, the concentricities of the shaft diameters or other locating features that position the individual components on the shaft shall be held to close tolerances relative to the shaft axis (see ISO 21940-14).

The concentricities of the balancing mandrel diameters or other locating features that position each individual component on the mandrel shall likewise be held within close tolerance relative to the axis of the mandrel. Errors in unbalance and concentricity of the mandrel can be compensated by index balancing (see ISO 21940-14).

When balancing the components and the shaft individually, make due allowances for any unsymmetrical feature such as keys (see ISO 21940-32) 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 assembly errors, e.g. eccentricities and assembly tolerances to evaluate their effects. When calculating the effect of these errors on the mandrel and on the shaft, 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 21940-14.

6.5.4 Procedure D — Balancing subsequent to controlling initial unbalance

When a rotor is composed of separate components that are balanced individually before assembly (see 6.5.3), the state of unbalance might still be unsatisfactory. Subsequent balancing of the assembly at low speed is permissible only if the initial unbalance of the assembly does not exceed specified values.

If reliable data on shaft and bearing flexibility, etc. are available, analysis of response to unbalance using mathematical models is useful to assess the unbalance correction distribution.

Experience has shown that symmetrical rotors that conform to the requirements above, but have an additional central correction plane can be balanced at low speed with higher initial unbalances of the assembly. Experience has shown that between 30 % and 60 % of the initial resultant, unbalance should be corrected in the central plane.

For unsymmetrical rotors that do not conform to the configuration defined above, e.g. as regards symmetry or overhangs, it might be possible to use a similar procedure using different percentages in the correction planes based on experience.

However, in extreme cases, the initial shaft unbalance can be so large that some other method of balancing the rotor is required, e.g. Procedure E.

6.5.5 Procedure E — Balancing in stages during assembly

The shaft shall first be balanced. The rotor shall 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 can be possible to add two single-plane components at a time and perform two-plane balancing on the assembly by using one correction plane in each of the two components. In cases where several components form a rigid section, e.g. a sub-assembly or core section which is normally balanced in two planes only, one such section can be added at a time and corrected by two-plane balancing.

6.5.6 Procedure F — Balancing in optimum planes

If, because of the design or method of construction, a series of rotors has unbalances that are distributed uniformly along their entire length (e.g. tubes), it can 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 conditions a) to e) in the following, the optimum position for the two correction planes is 22 % of the bearing span inboard of each bearing:

a) single-span rotor with end bearings;

b) uniform mass distribution with no significant overhangs;

c) uniform bending flexibility of the shaft along its length;

d) continuous service speeds not significantly approaching second resonance speed;

e) uniform or linear distribution of unbalance.

If this correction method does not produce satisfactory results, it can 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.

7 Procedures for balancing rotors with flexible behaviour at high speed

7.1 General

Generally, high-speed balancing is required for rotors with flexible behaviour. However, with the use of appropriate procedures, it is possible, under some circumstances, to balance rotors with flexible behaviour at low speed (see Clause 6).

7.2 Installation for balancing

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

If a rotor has an overhung mass that would normally be supported when installed *in-situ*, a steady bearing may be used to limit its deflection during balancing.

If a rotor has an overhung mass that is not supported in any way when installed *in-situ*, it shall also be left unsupported during balancing. However, it can be necessary in the early stage of balancing to provide support with a steady bearing to enable the rotor to be run safely to service speed or overspeed to allow the rotor components to move into their final position.

Transducers shall be positioned to measure shaft, bearing or support vibration, or bearing force as appropriate. The system shall be capable of measuring the once-per-revolution component of the signal. The measurement can be expressed either as amplitude and phase angle or in terms of orthogonal components relative to some fixed angular reference on the rotor.

In some cases, two vibration transducers may be installed 90° apart at the same transverse plane to permit resolution of the transverse vibrations when such resolution is required.

In all cases, there shall be no resonances of the transducer and mountings, which significantly influence vibration measurement within the speed range of the balancing process.

The output from all transducers shall be read on equipment that can differentiate between the synchronous component caused by unbalance, the slow-speed runout when significant, 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. Alternatively, if known unbalance is introduced by the drive system, then it should be compensated for in the vibration evaluation.

To confirm that the drive coupling introduces negligible balance error, the coupling shall be index balanced as described in ISO 21940-14.

7.3 Procedure G — Multiple speed balancing

7.3.1 General

This subclause sets out the basic principles of high-speed balancing in a very simple form. The rotor is balanced successively on a modal basis at a series of balancing speeds in turn, which are selected so that there is a balancing speed close to each resonance speed within the service speed range. There may also be a balancing speed close to the maximum permissible test speed. In essence, each mode with a resonance speed within the service speed range is corrected in turn, followed by a final balance of the remaining (higher) modes at the highest balancing speed.

The procedures used in practice may be packaged in the form of computer-aided balancing methods, which permit automated or, otherwise, simplified techniques, e.g. the influence coefficient method. In the simplest versions, on-line computer-aided balancing guides the operator through the process and performs, for example, the vector subtraction listed in 7.3.3.6, 7.3.3.10 and 7.3.3.11. In other cases, prior knowledge of the relevant influence coefficients can be available which can be incorporated in the computer-aided package so that tests with trial mass sets are not required. Under appropriate circumstances, vibration data for the unbalanced response can be safely acquired at several balancing speeds during one run of the rotor, rather than at a single balancing speed, so that the necessary corrections for several modes can be computed in one operation.

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

7.3.2 Initial low-speed balancing

Experience has shown that it can be advantageous to carry out initial balancing at low speed, prior to balancing at higher speeds. This can be particularly advantageous for rotors significantly affected by only the first flexural resonance 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 7.3.3.

NOTE Low-speed balancing can avoid the need for carrying out the final balancing of the remaining (higher) modes as described in 7.3.3.12.

7.3.3 General procedure

7.3.3.1 Throughout this procedure, correction planes should be chosen according to the relevant mode shapes (see also Clause 4).

7.3.3.2 If necessary, the rotor has to be run at some convenient low speed or speeds to remove any temporary bend. If shaft measuring transducers are used, the remaining repeatable low-speed runout values should be measured and, where necessary, subtracted vectorially from any subsequent shaft measurements at the balancing speeds.

7.3.3.3 Run the rotor to some safe speed approaching the first flexural resonance speed. This is termed the "first flexural balancing speed".

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

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

If low-speed balancing has been omitted and for rotors which are essentially symmetrical about mid-span, the trial mass set usually comprises of only one mass placed near the middle of the rotor span.

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

7.3.3.5 Run the rotor to the same speed and under the same conditions as in 7.3.3.3 and record the new readings of vibration (or force).

7.3.3.6 From the vectorial changes of the readings between 7.3.3.3 and 7.3.3.5, compute the effect of the trial mass set 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 1 A graphical illustration of the vectorial subtraction underlying this calculation is shown in Annex F.

NOTE 2 In this description, it is assumed that the effects on the measurements of unbalances in other modes can be neglected or are eliminated by appropriate procedures.

The rotor shall now run through the first flexural resonance speed with acceptable vibration (or force). If this is not the case, refine the correction or repeat the procedure in 7.3.3.3 to 7.3.3.6 using a new balancing speed, possibly closer to the first flexural resonance speed, but not so close as to affect phase and amplitude stability.

7.3.3.7 Run the rotor to some safe speed approaching the second flexural resonance speed. This is the "second flexural balancing speed". Record readings of vibration (or force) at this speed.

7.3.3.8 Add a set of trial masses to the rotor, which should be selected and positioned along the rotor to produce a significant vector change in vibration (or force) at the second flexural balancing speed without significantly affecting the first mode and, if relevant, the low-speed balance.

7.3.3.9 Run the rotor to the same speed as in 7.3.3.7 and record the new readings of vibration (or force).

7.3.3.10 From the vectorial changes in the readings between 7.3.3.7 and 7.3.3.9, compute the effect of the trial mass set 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 unbalance at the second flexural balancing speed. Attach this set of correction masses.

The rotor shall now run through the first and second flexural resonance speeds with acceptable vibration (or force). If this is not the case, refine the correction or repeat the procedure in 7.3.3.7 to 7.3.3.10 using a different balancing speed possibly closer to the second flexural resonance speed, but no so close as to affect phase and magnitude stability.

See also the notes in 7.3.3.6.

7.3.3.11 Continue the above operations for balancing speeds close to each flexural resonance speed in turn within the permissible speed range. Each new set of trial masses should be chosen so that it has a significant effect on the appropriate mode, but does 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 and attached to the rotor. Each set of correction masses compensates for the unbalance at the current balancing speed.

7.3.3.12 If, after correction at all flexural balancing speeds, significant vibrations (or forces) still occur within the service speed range, the procedure in 7.3.3.10 should be repeated at a balancing speed close to the maximum permissible test speed. In this case, it might not be possible to magnify the effect of the remaining (higher) modal unbalance components by running close to their associated flexural resonance speeds.

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 resonance speeds to enable the rotor to run to its service speed or overspeed, where components can move into their final position. For some rotors, it can be possible to run safely through some or all of the resonance speeds before completing the balancing. In that case, the number of runs required to determine the influence coefficients can be reduced.

The method described above assumes that there is a linear relationship between the unbalance vector and the vibration (or force) response vector. This might 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 can be necessary to redetermine the effects of the trial mass sets as the vibration (or force) response vector is reduced in magnitude.

As explained in 7.3.1, the high-speed balancing procedure is presented in a very simple form. In particular, the flexural resonance speeds are assumed to be sufficiently widely spaced so that the vibration measured at a flexural balancing speed is predominantly in the mode associated with the corresponding resonance speed. If two flexural resonance speeds are close together, then more refined procedures (which are beyond the scope of this simple outline) are necessary to uncouple the individual modal components of vibration.

For machines that have non-isotropic bearing support systems, each mode (see Figure 1) splits into two modes, often of similar shape, with resonances appearing at different speeds. Reducing the unbalance in one of these modes often reduces the unbalance in the other one also, avoiding the need to balance each mode separately.

7.4 Procedure H — Service speed balancing

Some rotors that are flexible and pass through one or more resonance speeds on their way up to service speed may, under special circumstances, be balanced for one speed only (usually service speed). However, rotors having resonance speeds close to service speed or those coupled to other flexible rotors are excluded. In general, these rotors should 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 resonance speeds does not build up beyond acceptable limits;

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

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

d) the measured vibration at the resonance speeds is acceptable;

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

A rotor that fulfils any of the above conditions may 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.

If the rotor falls into category c), it is especially important that the stiffness of the balancing machine support system be sufficiently close to site conditions to ensure that, at service speed in the balancing machine, the predominant modes are the same as those that are experienced *in-situ*.

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

7.5 Procedure I — Fixed speed balancing

7.5.1 General

These rotors have a basic shaft and body construction that either allows for low-speed balancing or requires high-speed balancing procedures. In addition, they have one or more components that are either flexible or are flexibly mounted so that the unbalance of the whole system can change with speed, which is an indication of a rotor with body-elastic behaviour.

Rotors in this case can fall into the following two categories:

a) rotors whose unbalance changes continuously with speed, e.g. rubber-bladed fans;

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

7.5.2 Procedure

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

Rotors that fall into category 7.5.1 a) shall be balanced in a balancing machine at the speed at which it is specified that the rotor should be in balance.

Rotors that fall into category 7.5.1 b) shall be balanced at a speed above that at which the unbalance ceases to change.

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

8 Evaluation criteria

8.1 Choice of criteria

It is a usual practice when evaluating the balance quality of a rotor with flexible behaviour in the factory to consider the once-per-revolution vibration of the bearing pedestals or shaft in a balancing machine or test facility that reasonably approximates to the site conditions for which the rotor is intended (see, however, 8.2.5). This is the method described in 8.2.

Another practice is to evaluate the balance quality by considering the residual unbalance. This is the method described in 8.3. For rotors with flexible behaviour balanced using low-speed balancing procedures (Procedures A to F in 6.5), this form of assessment may be made at low speed without any necessity to use a high-speed balancing machine.

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 or allow for the final effect of coupling to another rotor *in-situ*.

Evaluation criteria are, therefore, established either in terms of vibration limits or permissible residual unbalances.

It is not possible to derive the permissible unbalances for rotors with flexible behaviour 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 See also ISO 21940-31.

8.2 Vibration limits in the balancing machine

8.2.1 Overview

If the final state of unbalance is to be evaluated in terms of vibration criteria in the balancing machine, 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 machine and those obtained in the fully assembled machine on 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 in, for example, ISO 7919 or ISO 10816. In most cases, this relationship has been derived for specific machine types by experience of balancing typical rotors in the same balancing machine. Where such experience exists, it should be used as the basis for defining the permissible vibration in the balancing machine.

There can, however, be cases where such experience does not exist (e.g. a new balancing machine or rotors of substantially different design). Details of such cases are given in 8.2.6, which explains the permissible levels of once-per-revolution vibration that 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 7919 or ISO 10816.

8.2.2 General

Numerical values derived according to Clause 8 are not intended to serve as acceptance specifications, but as guidelines. When used in this manner, gross deficiencies or unrealistic requirements can be avoided.

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

These recommendations can also serve as the basis for more detailed investigations, e.g. when a more exact determination of the required balance quality is necessary.

8.2.3 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 Clause 8 apply.

8.2.4 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.

Where 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.

8.2.5 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, e.g. 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 can modify the mode shapes and, thus, the vibration levels at the points of measurements (see 4.3).

Rotors that are to be assembled in rigidly coupled multi-bearing systems, e.g. 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).

8.2.6 Permissible vibrations in the balancing machine

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

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

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

In either case, the corresponding permissible once-per-revolution bearing pedestal or shaft vibration in the balancing machine, y, can be expressed as:

$$y = x\, K_0\, K_1\, K_2 \qquad (1)$$

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 7919 or ISO 10816);

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

K_1 is a conversion factor used if the rotor support or coupling systems differ from site conditions. It is defined as the ratio of the once-per-revolution measurements in the balancing machine (shaft 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, which is used if in the balancing machine 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 $K_2 = 1$.

NOTE 1 The conversion relationship gives units for y which are the same as those for x. In practice, it can subsequently be convenient to express y in different units, e.g. displacement instead of velocity.

NOTE 2 The value of K_1 often depends upon the direction of measurement.

NOTE 3 For cases in which the measurement cannot be made at the same locations, K_2 can be determined analytically using a rotor dynamics model of the system.

The values of K_1 and K_2 can vary widely between one installation and another and are speed dependent. Some typical values for K_0 and K_1 are given in Annex C. The value of K_2 needs to be established for each specific application. If a resonance 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 be determined in isolation provided that a composite factor is available.

In addition, it should be noted that modal amplification of the vibration occurs at resonance 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 resonance speeds below the maximum service speed. For resonance speeds, it is especially difficult to establish quantitative criteria because it is almost impossible to arrange the same support conditions in the balancing machine as *in situ* (especially damping).

When the bending deflection during run up is of concern, because of rotor and stator clearances or stresses, the bending of a rotor at resonance speeds below service speed should be considered in terms of displacement of that part of the rotor at which the displacement is of consequence.

8.3 Residual unbalance tolerances

8.3.1 Overview

This subclause provides recommendations for rigid-body unbalance and modal unbalances for a rotor with flexible behaviour based on criteria given in ISO 21940-11.

8.3.2 General

The following establishes guidelines for the required balance quality of rotors with flexible behaviour. The values given are based on practical experience with the various types of rotor. If due regard is paid to the recommended values, satisfactory running conditions can be expected. Nonetheless, deviations from these recommendations can be necessary in certain cases.

For rotors with flexible behaviour balanced at low speed, permissible residual unbalances in specified correction planes are used to state the balance quality. For rotors balanced at high speed, permissible residual modal unbalances are applied.

The residual unbalance tolerances are based on those recommended in ISO 21940-11 for rotors with rigid behaviour and percentages of these values for the bending modes of rotors with shaft-elastic behaviour.

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

NOTE 1 A method for the experimental determination of the equivalent modal residual unbalances is described in 9.2.3.

NOTE 2 If the influence of overhung masses is significant, then it is possible that the percentages given are not applicable.

NOTE 3 If, *in situ*, the service speed or service speed range is close to either the first or second flexural resonance speed, it is possible that the percentages given require modification.

NOTE 4 In the balancing machine, the specified limits do not necessarily result in vibration magnitudes within normal limits in the speed range from 80 % to 120 % of any resonance speed. If such amplified vibrations occur, it does not necessarily mean that more refined balancing is needed because, for example, damping in the balancing machine is often smaller than *in situ*.

8.3.3 Limits for low-speed balancing

The residual unbalance for any completely assembled rotor shall not exceed the residual unbalance recommended for an equivalent rigid body in ISO 21940-11.

In addition, for rotors which are balanced in accordance with procedure C, D or E (see Table 2) each component, or when applicable, each sub-assembly of components shall be balanced to tolerances based on experience or those recommended in ISO 21940-11, applied to each component.

8.3.4 Limits for multiple speed balancing

8.3.4.1 First bending mode

For a rotor that is significantly affected by only the first modal residual unbalance, then, whatever its unbalance distribution, the residual unbalance shall not exceed the following limits, expressed as percentages of the total residual unbalance recommended for an equivalent rigid body in ISO 21940-11 and based upon the highest service speed of the rotor:

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

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

8.3.4.2 First and second bending modes

For a rotor that is significantly affected by only the first and second modal unbalances, then, whatever its unbalance distribution, the residual unbalance shall not exceed the following limits, expressed as percentages of the total residual unbalance recommended for an equivalent rigid body in ISO 21940-11 and based upon the highest service speed of the rotor:

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

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

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

In cases when one of the modes is less significant than the other, the corresponding limit can be relaxed, but shall not exceed 100 %.

NOTE The example in D.1 illustrates the calculation of these limits.

8.3.4.3 More than two bending modes

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

9 Evaluation procedures

NOTE Depending on the type and purpose of the rotor being assessed, the final state of balance can be evaluated either in terms of vibration at specified measuring planes or by residual unbalances. In cases of small mass-produced rotors, assessment procedures simpler than those detailed in this part of ISO 21940 can suffice.

9.1 Evaluation procedures based on vibration limits

9.1.1 Vibration assessed in a high-speed balancing machine

The installation of the rotor in the balancing machine shall conform to the guidelines given in 7.2.

When the above conditions have been satisfied, the rotor shall be run up to speed at a low acceleration rate to ensure that vibration peaks are not suppressed. If measurement over the whole speed range is not possible, all significant peaks of vibration should be measured between 70 % of the observed first flexural resonance speed and the maximum service speed. Alternatively, this could be achieved by a comparable run down.

The rotor shall be held at maximum service speed long enough to eliminate any transient effects. Once-per-revolution vibration measurements shall then be taken.

9.1.2 Vibration assessed on a test facility

Rotors whose final state of balance is evaluated on a test facility should have instrumentation as stated in 7.2, but it should be noted that different procedures can be necessary in some cases when

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

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

c) vibration transducers cannot be placed at the bearings, in this case, the points where vibrations should be measured should be agreed between the manufacturer and user, or

d) the state of balance 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.

9.1.3 Vibration assessed on site

9.1.3.1 Machines that have their state of balance assessed after final installation on site are subject to many factors that can produce vibration. Some of this vibration can 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.

9.1.3.2 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 can be required to reduce the vibration level. Such improvements can only be useful if the machine is not highly susceptible to unbalance (for details, see ISO 21940-31). If, in operation, there is a high probability that new unbalances 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.

9.1.3.3 In the final installation on site, there might be factors during commissioning that conflict with obtaining the steady-state conditions needed to assess the state of balance. It can 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 (e.g. where the prime mover is a “direct to line start” induction motor), it can be 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 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 and be constant for the duration of the balancing procedure.

9.1.3.4 Vibration measuring equipment should be installed as specified in 7.2. Where suitable monitoring equipment is provided in the installation, this may be used instead. Alternatively, vibrations may be read on portable apparatus using, for instance, a hand-held vibration transducer.

9.2 Evaluation based on residual unbalance tolerances

9.2.1 General

Three different approaches are outlined in 9.2.2 to 9.2.4.

9.2.2 Evaluation at low speed

The evaluation at low speed is based on the unbalance tolerances for rotors with rigid behaviour as stated in ISO 21940-11.

Rotors in this category usually have their balance quality assessed in a low-speed balancing machine. In most cases, a subsequent high-speed check is made on the test facility 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 rotors sold as spares where a final assessment on site may be delayed for a considerable time.

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

The balancing machine should be one that conforms to ISO 21940-21. See ISO 21940-11 for the procedure to assess the residual unbalance and ISO 21940-14 for typical errors in the unbalance measuring process.

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

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

For rotors with controlled initial unbalance, the initial unbalance after assembly shall 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 (Procedure E), the residual unbalance achieved at each stage shall be stated.

9.2.3 Evaluation at multiple speeds based on modal unbalances

Multiple speeds give an insight into the unbalance distribution of the rotor and its expected flexible behaviour.

To assess the state of balance, the residual equivalent modal unbalances are calculated for the respective modes. The equivalent modal unbalance is defined as the smallest unbalance in an individual plane which has the same effect as the modal unbalance. This means that, for each respective mode, the residual unbalance is calculated for the most sensitive plane. This assumes that correction planes are located in suitable positions.

Special care should be taken when selecting influence coefficients for the equivalent residual modal unbalance calculation to avoid assessment errors associated with the unrepeatability of vibration readings and the nonlinearity of the rotor's response to unbalance.

Balance planes in couplings which are to be rigidly coupled *in situ* should not be used for residual modal unbalance assessment since they may excite a mode that does not exist in the coupled rotor system.

The procedure is as follows:

a) mount the rotor in a high-speed balancing machine or other high-speed test facility;

b) if low-speed balancing is performed, the residual unbalance for the rigid body may be assessed either by using the influence coefficient method or by using the balancing machine and its capability to indicate unbalances in two planes;

c) run the rotor to some safe speed approaching first flexural resonance speed and note readings of vibration or force;

d) add a trial mass to the rotor. The unbalance caused by this mass should be sufficient to show a significant effect and should be placed axially where it has the maximum effect on the first mode. Take readings of vibration or force at the same speed as in c);

e) From the readings obtained in c) and d), compute vectorially the equivalent first modal unbalance. For example, this can be done graphically by the construction in Annex F, in this case, with the single trial mass forming the trial mass set. Then, the magnitude of the equivalent first modal unbalance is:

$$\frac{\mathrm{OA}}{\mathrm{AB}} U_{\mathrm{t}} \tag{2}$$

where U_{t} is the unbalance caused by the trial mass;

f) remove the trial mass;

g) run the rotor to some safe speed approaching second flexural resonance speed provided this is lower than the maximum safe service speed. Note readings of vibration or force;

h) add a trial mass to the rotor. The unbalance caused by this mass should be sufficient to show a significant effect and should be placed axially where it has maximum effect on the second mode. Take readings of vibration or force at the same speed as in g);

i) from the readings obtained in g) and h), compute vectorially the equivalent second modal unbalance. The graphical procedure of e) may be used in this case;

j) remove the trial mass;

k) continue the operations for successive modes until the equivalent modal unbalances in all significant modes have been determined.

An example is shown in Annex D.

When determining the equivalent modal unbalances, it can be necessary to use a set of trial masses in order to pass safely through the low resonance speeds. Finally, all calculated residual unbalances are summarized into an equivalent residual modal unbalance.

NOTE 1 The procedure given assumes that the vibration measured at a speed close to a resonance speed is predominantly in the corresponding mode and, therefore, usually gives a close approximation to the equivalent residual modal unbalances.

NOTE 2 Sometimes, it is not possible to run the rotor close to the resonance speeds of some of the significant modes. In these cases, further procedures are necessary to separate the individual modal components.

NOTE 3 If the rotor remains in the balancing machine after a balancing procedure, in accordance with 7.3, the information obtained during balancing can be used directly, without the need for further test runs.

NOTE 4 Trial mass sets (unbalance combinations selected to excite a certain mode) can be utilized for residual modal unbalance assessment instead of the most sensitive plane procedure.

9.2.4 Evaluation at service speed in two specified test planes

If the service speed is used, special care is needed to choose the test planes properly.

The axial position of the correction planes and the balancing speed shall be stated.

If the rotor is assessed in a balancing machine having its own instrumentation, this shall 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 7.2.

Annex A
(informative)

Cautionary notes concerning rotors when installed *in-situ*

A.1 General

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 identifying the factors other than unbalance that are influencing the magnitude of vibration of the machine. Such factors include those mentioned below.

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

A.2 Bearing misalignment

Small parallel or angular misalignment of the rotor bearings can produce effects which are not caused by unbalance. If these effects are present, the misalignment might need to be corrected prior to further assessment of the vibration of the machine (see also the last paragraph of A.3).

A.3 Radial and axial runout of coupling faces

There is no practical way 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 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 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 whirl or whip) can take place in the types of hydrodynamically lubricated bearings which are normally used in multi-span 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 part of ISO 21940.

Annex B
(informative)

Optimum planes balancing — Low-speed three-plane balancing

B.1 This Annex is concerned with the low-speed balancing of rotors which have one central and two end correction planes if all of the following conditions are met:

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;

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

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 unbalance correction in three planes can be calculated from the initial unbalance measured in two planes, $\boldsymbol{U}_\mathrm{L}$ and $\boldsymbol{U}_\mathrm{R}$. The vector sum of the forces and the vector sum of the moments created by the three unbalance corrections $\boldsymbol{U}_1$, $\boldsymbol{U}_2$ and $\boldsymbol{U}_3$ about a given point on the rotor should compensate those caused by the initial unbalances, $\boldsymbol{U}_\mathrm{L}$ and $\boldsymbol{U}_\mathrm{R}$, about the same point.

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

$$\boldsymbol{U}_1 = \boldsymbol{U}_\mathrm{L} - \frac{1}{2} H (\boldsymbol{U}_\mathrm{L} + \boldsymbol{U}_\mathrm{R}) \tag{B.1}$$

$$\boldsymbol{U}_2 = H (\boldsymbol{U}_\mathrm{L} + \boldsymbol{U}_\mathrm{R}) \tag{B.2}$$

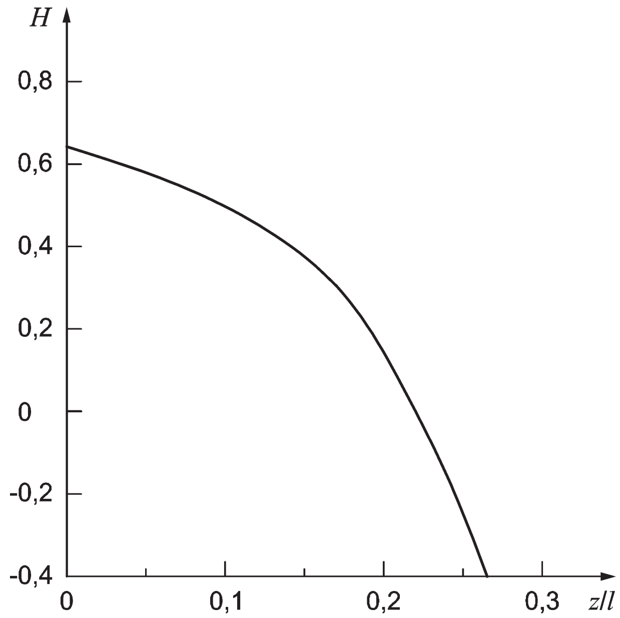
$$\boldsymbol{U}_3 = \boldsymbol{U}_\mathrm{R} - \frac{1}{2} H (\boldsymbol{U}_\mathrm{L} + \boldsymbol{U}_\mathrm{R}) \tag{B.3}$$

where H is the central correction divided by the initial static unbalance.

The bold type indicates that $\boldsymbol{U}_\mathrm{L}$, $\boldsymbol{U}_\mathrm{R}$, $\boldsymbol{U}_1$, $\boldsymbol{U}_2$ and $\boldsymbol{U}_3$ 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 to 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.



Key

H central correction divided by the initial static unbalance

l bearing span (shaft length)

z distance from the left-hand bearing to correction plane 1

Figure B.1 — Graphical presentation for determination of *H*

Annex C
(informative)

Conversion factors

Table C.1 gives conversion factors for machinery which is classified as follows.

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 — Typical conversion factors (see 8.2.6)

<table>
<tr><th rowspan="2">Machinery classification</th><th rowspan="2">Typical machines</th><th rowspan="2">K_0</th><th colspan="3">K_1</th></tr>
<tr><th>Bearing support absolute</th><th>Shaft absolute</th><th>Shaft relative</th></tr>
<tr><td rowspan="2">I</td><td>Superchargers</td><td>1,0</td><td rowspan="14">0,6 to 1,6</td><td rowspan="14">1,6 to 5,0</td><td rowspan="14">1,0 to 3,0</td></tr>
<tr><td>Small electric motors up to 15 kW</td><td>1,0</td></tr>
<tr><td rowspan="5">II</td><td>Paper-making machines</td><td>0,7 to 1,0</td></tr>
<tr><td>Medium-sized electric machines 15 kW to 75 kW</td><td>0,7 to 1,0</td></tr>
<tr><td>Electrical machines up to 300 kW on special foundations</td><td>0,7 to 1,0</td></tr>
<tr><td>Compressors</td><td>0,7 to 1,0</td></tr>
<tr><td>Small turbines</td><td>1,0</td></tr>
<tr><td rowspan="4">III</td><td>Large electric motors</td><td>0,7 to 1,0</td></tr>
<tr><td>Pumps</td><td>0,7 to 1,0</td></tr>
<tr><td>2-pole generators</td><td>0,8 to 1,0</td></tr>
<tr><td>Turbines and multipole generators</td><td>0,9 to 1,0</td></tr>
<tr><td rowspan="3">IV</td><td>Gas turbines (see also 8.2.3)</td><td>1,0</td></tr>
<tr><td>2-pole generators</td><td>0,8 to 1,0</td></tr>
<tr><td>Turbines and multipole generators</td><td>0,9 to 1,0</td></tr>
</table>

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

K_1 is the ratio of the once-per-revolution measurements in the balancing machine (shaft or bearing pedestals) to similar measurements taken on the assembled machine on site (if not applicable, $K_1 = 1$).

NOTE In relation to the entries for K_1, "absolute" refers to measurement of vibration with reference to an inertial reference frame and "relative" refers to measurement relative to an appropriate structure, such as a bearing housing. A full discussion of these terms is given in ISO 20816-1.

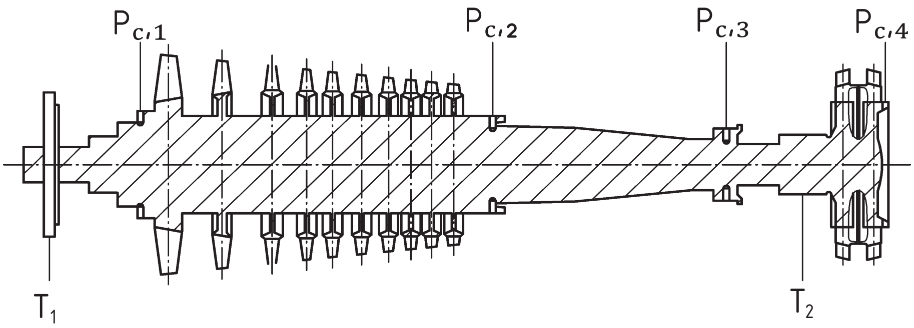
Annex D
(informative)

Example calculation of equivalent residual modal unbalances

D.1 Residual unbalance calculation

The principles of residual unbalance calculation are shown in the following example. A recommended procedure is outlined in 9.2.3.

The rotor is a gas turbine rotor with four correction planes $P_{c,1}$ to $P_{c,4}$ (see Figure D.1). The balancing calculations are based on vibration measurements at the two bearings (transducers T_1 and T_2).



Key

$P_{c,1}$, $P_{c,2}$, $P_{c,3}$, $P_{c,4}$ correction planes

T_1, T_2 transducers

Figure D.1 — Example gas turbine rotor

The service speed of the rotor is 10 125 r/min.

The rotor mass is 1 625 kg.

The permissible total unbalance for an equivalent rigid body according to balance quality grade G 2,5 is taken from ISO 21940-11 to be 2,37 g·mm/kg.

The total residual unbalance for an equivalent rigid body is therefore

$$2{,}37\ \frac{\mathrm{g}\cdot\mathrm{mm}}{\mathrm{kg}} \times 1\ 625\ \mathrm{kg} = 3\ 850\ \mathrm{g}\cdot\mathrm{mm} \tag{D.1}$$

The permissible equivalent first modal unbalance (60 % thereof, see 8.3.4.2) is 2 311 g·mm.

The permissible equivalent second modal unbalance (60 %) is 2 311 g·mm.

As given by Formula (D.1), the total permissible residual unbalance for the rigid body (low-speed balancing) is 3 850 g·mm, i.e. 1 925 g·mm per correction plane $P_{c,1}$ and $P_{c,3}$.

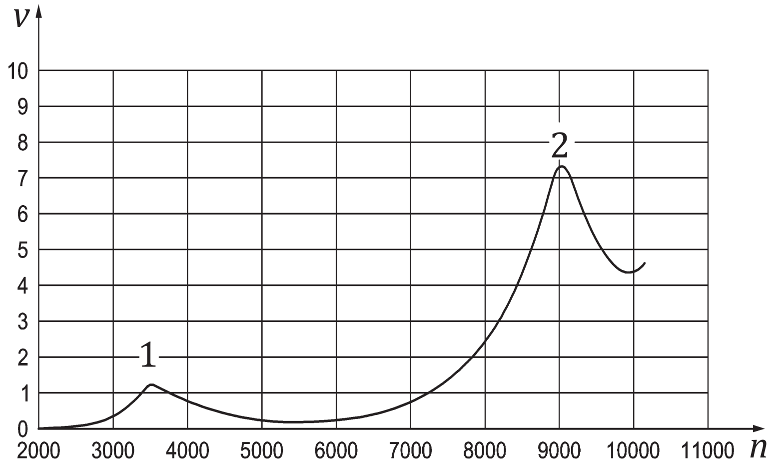
D.2 Influence coefficients

The balancing speeds for this rotor are the following (see Figure D.2):

— 1 000 r/min (low speed);

— 3 400 r/min (just below rotor resonance 1);

— 9 000 r/min (just below rotor resonance 2).



Key

v vibration velocity in mm/s

n rotational speed in r/min

1 rotor resonance 1

2 rotor resonance 2

Figure D.2 — Run-up curve (before balancing)

The influence coefficients given in Table D.1, which are used for residual unbalance calculation, have been calculated from runs with trial masses. The influence coefficients are given in (mm/s)/(kg·mm) and at an angle relative to a reference system on the rotor.

Table D.1 — Influence coefficients

Measurement point	Correction plane				Speed
	$P_{c,1}$	$P_{c,2}$	$P_{c,3}$	$P_{c,4}$	r/min
Transducer 1	0,059 4/3° [a]	0,033 0/1°	0,009 12/333° [a]	0,004 90/233°	1 000
Transducer 2	0,002 16/35° [a]	0,022 7/14°	0,033 4/11° [a]	0,042 5/9°	
Transducer 1	0,249/82°	0,343/94°	0,055/222°	0,360/265° [a]	3 400
Transducer 2	0,087/107°	0,157/87°	0,102/34°	0,224/6° [a]	
Transducer 1	1,99/146°	2,29/285° [a]	1,56/293°	2,07/176°	9 000
Transducer 2	1,92/353°	1,99/134° [a]	1,16/109°	0,595/281°	

[a] Influence coefficients used for residual unbalance calculation.

Planes $P_{c,1}$ and $P_{c,3}$, which are nearest to the bearings, are selected for the low speed. For other speeds, the most sensitive plane for each transducer is selected.

D.3 Final vibration readings

The vibration vectors given in Table D.2 were measured during the run in the final balance condition.

Table D.2 — Final vibration readings

Speed r/min	Transducer T_1	Transducer T_2	Unit
1 000	0,01/237°	0,022/147°	mm/s
3 400	0,55/52°	0,22/125°	mm/s
9 000	2,35/305°	1,44/139°	mm/s

D.4 Residual unbalance at low speed, 1 000 r/min

The calculation is carried out in accordance with the influence coefficient method for correction planes $P_{c,1}$ and $P_{c,3}$ (planes nearest to the bearings) and transducers 1 and 2 (see Table D.3).

Table D.3 — Residual unbalance at 1 000 r/min

Plane	Calculated g·mm	Permissible g·mm
$P_{c,1}$	246	1 925
$P_{c,3}$	671	1 925

D.5 Residual unbalance at 3 400 r/min

At the other balancing speeds, the residual unbalance is obtained by dividing the amount of the vibration vector (see Table D.2) by the absolute value of the influence coefficient (see Table D.1). This means that no consideration needs to be given to the phase angle of the vibration or the phase information of the receptance (see Table D.4).

Table D.4 — Residual unbalance at 3 400 r/min

Transducer	Calculated	Permissible g·mm
T_1	(0,55/0,36) 1 000 = 1 530 g·mm	2 311
T_2	(0,22/0,224) 1 000 = 982 g·mm	2 311

D.6 Residual unbalance at 9 000 r/min

See Table D.5.

Table D.5 — Residual unbalance at 9 000 r/min

Transducer	Calculated	Permissible g·mm
T_1	(2,35/2,29) 1 000 = 1 026 g·mm	2 311
T_2	(1,44/1,99) 1 000 = 723 g·mm	2 311

Annex E
(informative)

Procedures to determine whether a rotor shows rigid or flexible behaviour

E.1 General

This Annex describes procedures that may be used to determine whether a rotor shows rigid or flexible behaviour. The test according to E.2 reveals whether a rotor resonance below the maximum service speed exists, while the tests according to E.3 and E.4 show the influence of the first rotor resonance if it is above the maximum service speed.

If it is determined that a rotor falls into the rigid category, then it can be balanced using a low-speed balancing procedure. In general, rotors with flexible behaviour need to be balanced at high speeds using procedures such as those in Clause 7. There are, however, rotors which, by definition, are flexible, but are borderline and for which low-speed balancing can be adequate using the special procedures given in Clause 6.

E.2 Determination of whether a rotor shows rigid or flexible behaviour

E.2.1 One or more of the information sources given in E.2.2 to E.2.4 may be used to ascertain whether a rotor falls into the rigid or flexible category and, thereby, determine the balancing method to be adopted.

E.2.2 Consult the rotor manufacturer or the user for a definition of the rotor configuration and characteristics and a recommended balancing procedure (see Clause 5).

E.2.3 If the first flexural resonance speed exceeds the maximum service speed by at least 50 %, then the rotor can often be considered rigid for balancing purposes.

E.2.4 Alternatively, the following test sequence may be performed if a rotor resonance exists below the service speed.

Balance the rotor at low speed in two correction planes in accordance with the procedures specified in ISO 21940-11.

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 one of these possibilities applies, perform the flexibility test described in E.3.

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.3 to determine if the rotor falls into the rigid or flexible category.

NOTE Settlement can occur by running the rotor to its service speed or beyond by permanently settling 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.3 Rotor flexibility test

Add a mass to the centre of the rotor or to an available position where it can 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 magnitude becomes excessive during the rotor acceleration, reduce the magnitude of the mass and repeat the process. Measure the vibration vector at service speed and at the same location as in E.2.4. Determine the effect of the mass on the vibration level by vectorially subtracting the vibration vector recorded in E.2.4 from the new reading. Denote the result as vector $\boldsymbol{A}$.

Stop the rotor and remove the mass. Install two masses close to the rotor end planes in the same angular position as the central mass that was removed. These masses should be chosen to provide the same static unbalance in the plane of the single mass without introducing any additional moment 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.2.4 from the reading. Denote this vector as $\boldsymbol{B}$.

These tests can be used to show the influence of the first resonance if it is below the service speed.

E.4 Evaluation of flexibility test data

Compute the magnitude of the vector $\boldsymbol{A} - \boldsymbol{B}$. If this magnitude, when divided by the magnitude of vector $\boldsymbol{A}$ 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 flexible.

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

Annex F
(informative)

Method of computation of unbalance correction

The following is a possible method of computation of unbalance correction by observation of the effect of a trial mass set.

Let vector $\overrightarrow{OA}$ in Figure F.1 represent the initial vibration plotted to some arbitrary reference angle.

Let vector $\overrightarrow{OB}$ 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 amount and phase angle by the vector $\overrightarrow{AB}$.

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 by the ratio OA/AB.

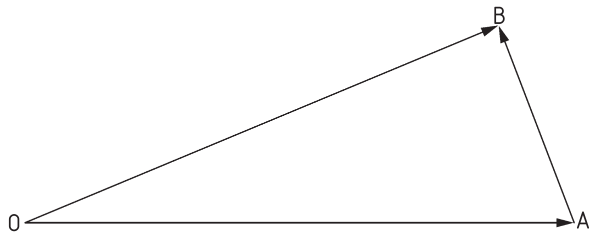


Figure F.1 — Vectorial effect of a trial mass set

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[2] ISO 10816 (all parts), *Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts*

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12) To be published. Revision and amalgamation of ISO 7919-1 and ISO 10816-1.

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