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

*Vibrations mécaniques — Méthodes et critères pour l'équilibrage  
mécanique des rotors flexibles*

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## Foreword

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

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

International Standard ISO 11342 was prepared by technical committee ISO/TC 108, *Mechanical vibration and shock*, Subcommittee SC 1, *Balancing, including balancing machines*.

This second edition cancels and replaces the first edition (ISO 11342:1994), of which it constitutes a technical revision.

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

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## Introduction

The aim of balancing any rotor is to achieve satisfactory running when installed on site. 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 flexible rotor, 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 may be only limited access to the rotor. Furthermore, balancing of the rotor is often the stage at which a rotor is approved by the purchaser. Thus, while satisfactory running on site is the aim, the balance quality of the rotor is usually initially assessed in a balancing facility. Satisfactory running on site is in most cases judged in relation to vibration from all causes, while in the balancing facility primarily once-per-revolution effects are considered.

This International Standard classifies rotors in accordance with their balancing requirements and establishes methods of assessment of residual unbalance.

This International Standard also shows how criteria for use in the balancing facility may be derived from either vibration limits specified for the assembled and installed machine or unbalance limits specified for the rotor. If such limits are not available, this International Standard shows how they may be derived from ISO 10816 and ISO 7919 if desired in terms of vibration, or from ISO 1940-1 if desired in terms of permissible residual balance. ISO 1940 is concerned with the unbalance quality of rotating rigid bodies and is not directly applicable to flexible rotors because flexible rotors may undergo significant bending deflection. However, in subclause 8.3 of this International Standard, methods are presented for adapting the criteria of ISO 1940-1 to flexible rotors.

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As this International Standard is complementary in many details to ISO 1940, it is recommended that, where applicable, the two should be considered together.

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 resonant or near resonant condition in a lightly damped structure can result in excessive vibratory response to a small unbalance. In such cases it may be more practicable to alter the natural frequency or damping of the structure rather than to balance to very low levels, which may not be maintainable over time. (See also ISO 10814.)

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

## 1 Scope

This International Standard presents typical flexible rotor configurations in accordance with their characteristics and balancing requirements, describes balancing procedures, specifies methods of assessment of the final state of unbalance, and gives guidance on balance quality criteria.

This International Standard may also be applicable to serve as a basis for more involved investigations, for example when a more exact determination of the required balance quality is necessary. If due regard is paid to the specified methods of manufacture and limits of unbalance, satisfactory running conditions can be expected.

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

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

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

## 2 Normative references

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

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

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

<sup>1)</sup> Information on such exceptions will be welcomed and should be communicated to the national standards body in the country of origin for transmission to the secretariat of ISO/TC 108/SC1.

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

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

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

### 3 Definitions

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

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

## 4 Fundamentals of flexible rotor dynamics and balancing

### 4.1 General

Flexible rotors normally require multiplane blancing at high speed. Nevertheless, under certain conditions a flexible rotor 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 will be given here. In most practical balancing applications, the method adopted will normally be 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 flexible rotor than in a rigid rotor 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 may induce vibrations at speeds other than that at which the rotor was originally corrected. These vibrations may exceed specified tolerances, particularly at, or near, the flexural critical speeds. Even at the same speed such correction can induce vibrations if the flexural mode shapes on site differ from those dominating during the balancing process.

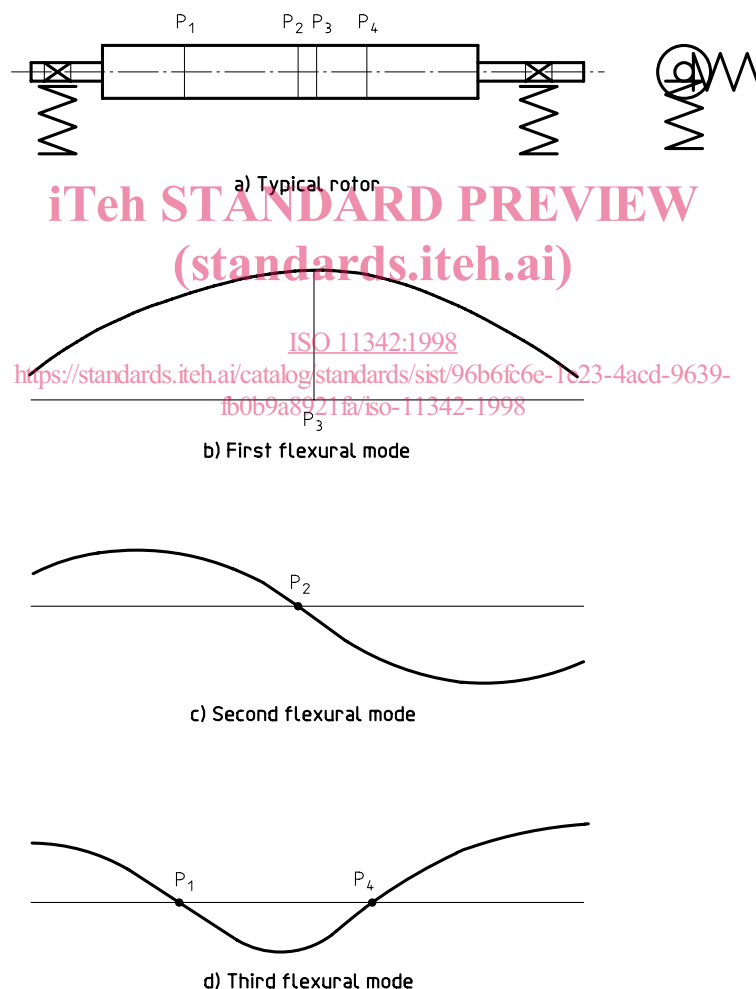
In addition, some rotors which become heated during operation are susceptible to thermal bows which can lead to changes in the unbalance. If the rotor unbalance changes significantly from run to run it may be impossible to balance the rotor within tolerance.

### 4.3 Flexible rotor mode shapes

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

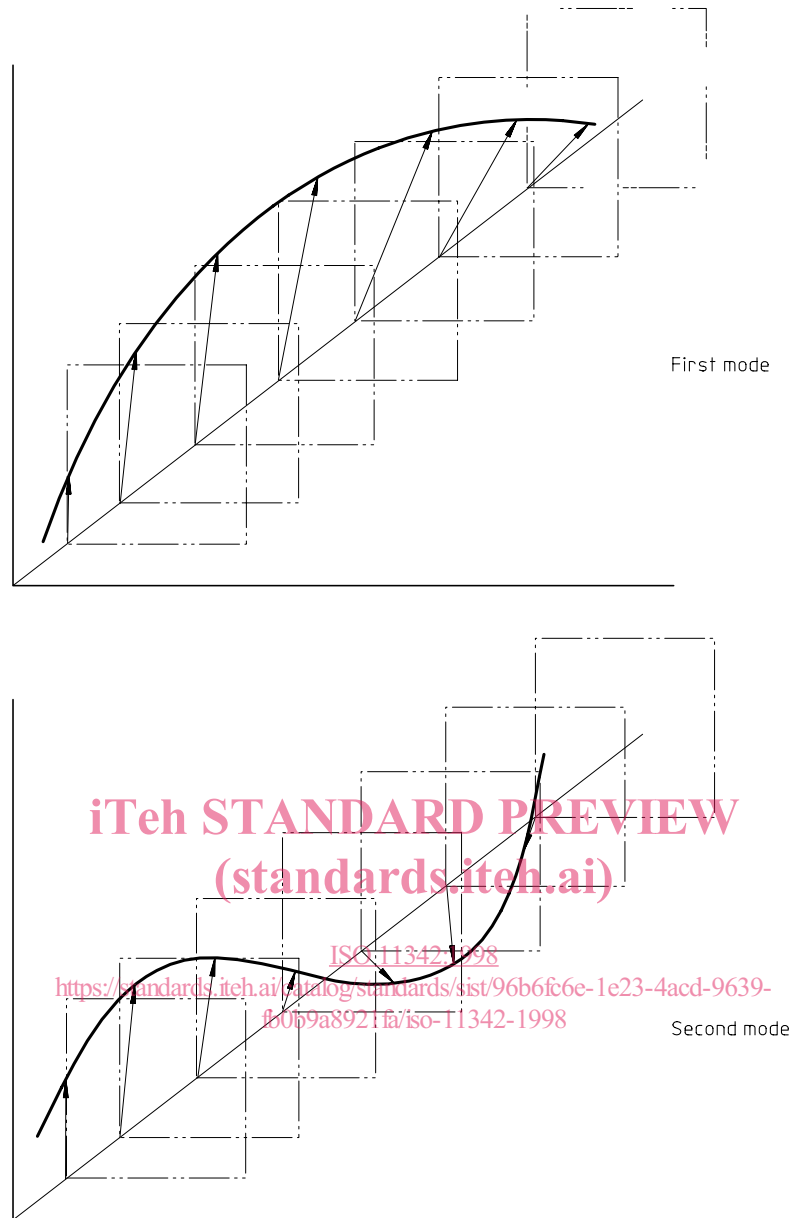
For a damped rotor/bearing system the flexural modes may be space curves rotating about the shaft axis, especially in the case of substantial damping, arising perhaps from fluid-film bearings. Possible damped first and second modes are illustrated in figure 2. In many cases the damped modes can be treated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

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



NOTE —  $P_1$ ,  $P_2$ , and  $P_4$  are nodes.  $P_3$  is an antinode.

**Figure 1 — Simplified mode shapes for flexible rotors on flexible supports**



**Figure 2 — Examples of possible damped mode shapes**

#### 4.4 Response of a flexible rotor to unbalance

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

- a) the magnitude of the modal unbalances;
- b) the proximity of the associated critical speeds to the running speeds; and
- c) the amount of damping in the rotor/support system.



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

The modal unbalances for a given unbalance distribution are a function of the flexible rotor modes. Moreover, 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 1 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, whilst 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.

#### 4.5 Aims of flexible rotor balancing

The aims of balancing are determined by the operational requirements of the machine. Before balancing any particular rotor, it is desirable to decide what balance criteria can be regarded as satisfactory. In this way the balancing process can be made efficient and economical, but still satisfy 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 flexible rotors would be to correct the local unbalance occurring at each elemental length by means of unbalance corrections at the element itself. This would result in a rotor in which the centre of mass of each elemental length lies on the shaft axis.

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

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

Vibrations or oscillatory forces caused by the residual unbalance must be reduced to acceptable magnitudes over the service speed range. Only in special cases is it sufficient to balance flexible rotors for a single speed. It should be noted that a rotor, balanced satisfactorily for a given service speed range, may still experience excessive vibration if it has to run through a critical speed to reach its service speed. However, for passing through critical 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 may be necessary to allow for the influence of modes with critical 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 assembly-balanced in the end planes only, after each disc and the shaft have been separately balanced in a low-speed balancing machine. Generally, however, if the speed of the rotor approaches or exceeds its  $n^{\text{th}}$  flexural critical speed, then at least  $n$  and usually  $(n + 2)$  correction planes are needed along the rotor.

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

#### 4.7 Rotors coupled together

When two rotors are coupled together, the complete unit will have a series of critical speeds and mode shapes. In general, these speeds are neither equal to nor simply related to the critical speeds of the individual, uncoupled rotors. Moreover, the deflection shape of each part of the coupled unit need not be simply related to any mode shape of the corresponding uncoupled rotor. 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 will normally ensure satisfactory operation of the coupled rotors. The degree to which this technique is practicable depends, for example, on the mode shapes and the critical speeds of the uncoupled and coupled rotors, and the distribution of unbalance and the type of coupling and on the bearing arrangement of the shaft train.

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

### 5 Rotor configurations

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

Sometimes a combination of balancing procedures may be advisable. If more than one balancing procedure could be used, they are listed in the sequence of increasing time/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 — Flexible rotors


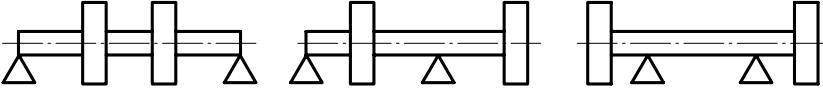
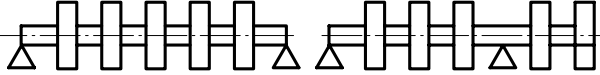
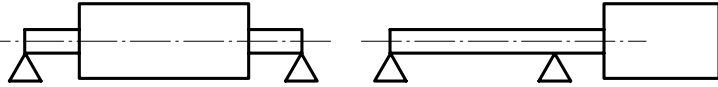


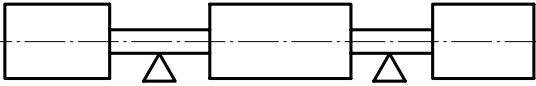
Configuration	Rotor characteristics	Recommended balancing procedure
<p><b>1.1 Discs</b></p>	<p>Elastic shaft without unbalance, rigid disc(s)</p>	<p>(see table 2) (see next page for key to A-G)</p>
	<p><b>Single disc</b></p> <ul style="list-style-type: none"> <li>- perpendicular to shaft axis</li> <li>- with axial runout</li> </ul>	<p>A; C B; C</p>
	<p><b>Two discs</b></p> <ul style="list-style-type: none"> <li>- perpendicular to shaft axis</li> <li>- with axial runout</li> <li>• at least one removable</li> <li>• integral</li> </ul>	<p>B; C B + C, E G</p>
 <p style="text-align: center; color: red; font-weight: bold;">iTeh STANDARD PREVIEW (standards.iteh.ai)</p> <p style="text-align: center; color: red;">ISO 11342:1998</p>	<p><b>More than two discs</b></p> <ul style="list-style-type: none"> <li>- all (but one) removable</li> <li>- integral</li> </ul>	<p>B + C, D, E G</p>
<p><b>1.2 Rigid sections</b></p> <p style="text-align: center; color: red;"> <a href="https://standards.iteh.ai/catalog/standards/sist/96b6fc6e-1e23-4acd-9639-fb0b9a8921fa/iso-11342-1998">https://standards.iteh.ai/catalog/standards/sist/96b6fc6e-1e23-4acd-9639-fb0b9a8921fa/iso-11342-1998</a> </p>	<p>Elastic shafts without unbalances, rigid sections</p>	
	<p><b>Single rigid section</b></p> <ul style="list-style-type: none"> <li>- removable</li> <li>- integral</li> </ul>	<p>B; C; E B</p>
		
 	<p><b>More (than two) rigid section</b></p> <ul style="list-style-type: none"> <li>- all (but one) removable</li> <li>- integral</li> </ul>	<p>B + C; E G</p>

Table 1 — Flexible rotors (concluded)

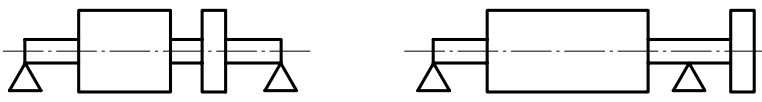
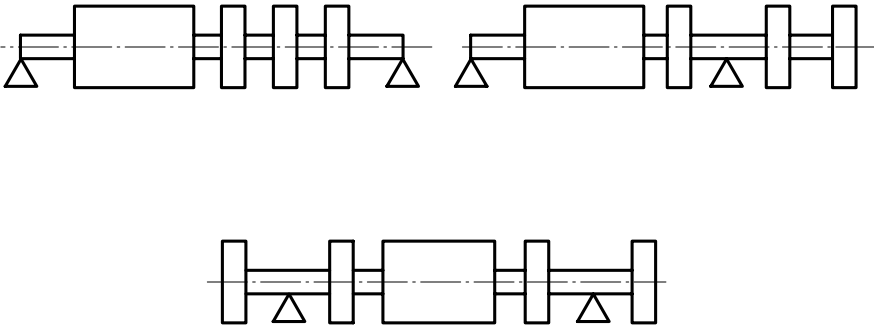

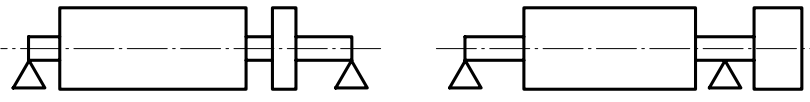
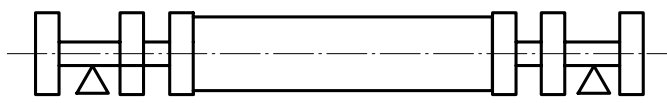
Configuration	Rotor characteristics	Recommended balancing procedure
<p><b>1.3 Discs and rigid sections</b></p>	<p>Elastic shaft without unbalance, rigid discs and sections</p>	<p>(see table 2) <sup>1)</sup></p>
	<p><b>One each</b></p> <ul style="list-style-type: none"> <li>- at least one part removable</li> <li>- integral</li> </ul>	<p>B + C; E G</p>
	<p><b>More parts</b></p> <ul style="list-style-type: none"> <li>- all (but one) removable</li> <li>- integral</li> </ul>	<p>B + C; E G</p>
<p><b>1.4 Rolls</b></p>	<p>Mass, elasticity and unbalance distribution along the rotor</p>	<p>F G</p>
	<ul style="list-style-type: none"> <li>- under special conditions</li> <li>- in general</li> </ul>	<p>F G</p>
<p><b>1.5 Rolls and discs/rigid sections</b></p>	<p>Flexible roll, rigid discs, rigid sections</p>	<p>C + F; E + F G G</p>
	<ul style="list-style-type: none"> <li>- discs/rigid sections/removable</li> <li>- under special conditions</li> <li>- in general</li> <li>- integral</li> </ul>	<p>C + F; E + F G G</p>
<p><b>1.6 Integral rotor</b></p>	<p>Mass, elasticity and unbalance distribution along the rotor</p>	<p>G</p>
	<p>Main parts with unbalances not detachable</p>	<p>G</p>
<p>1) A = Single-plane balancing                  B = Two-plane balancing                  C = Individual component balancing prior to assembly                  E = Balancing in stages during assembly                  F = Balancing in optimum planes                  G = Multiple speed balancing</p> <p>Two additional balancing procedures H and I can be used in special circumstances, see 7.4 and 7.5.</p>		

Table 2 — Balancing procedures

Procedure	Description	Subclause
<b>Low-speed balancing</b>		
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
<b>High-speed balancing</b>		
G	Multiple speed balancing	7.3
H	Service speed balancing	7.4
I	Fixed speed balancing	7.5

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## 6 Procedures for balancing flexible rotors at low speed

### 6.1 General

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

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

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

NOTE — Certain rotors contain a number of individual parts which are mounted concentrically (for example blades, coupling bolts, pole pieces, etc.). These parts may be arranged according to their individual mass or mass moment to achieve some or all of the required unbalance correction described in any of the 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 may produce changes in the shaft geometry (e.g. shaft run out) and further changes may occur during high-speed service.