International Standard ISO 5406

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The mechanical balancing of flexible rotors

Équilibrage mécanique des rotors flexibles

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Foreword

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Draft International Standards adopted by the technical committees are circulated to the member bodies for approval before their acceptance as International Standards by the ISO Council.

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It has been approved by the member bodies of the following countries :

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The member body of the following country expressed disapproval of the document on technical grounds :

France

Contents

The mechanical balancing of flexible rotors

0 Introduction

This International Standard classifies rotors into groups in accordance with their balancing requirements, establishes methods of assessment of final unbalance, and gives initial guidance on the establishment of balance quality grades so that, ultimately, balance quality grades can be established for all types of flexible rotors.

As the next stage in the development of these balance quality grades, the criteria for evaluating the unbalance of flexible rotors will be further described in an addendum to this International Standard.

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

1 Scope and field of application

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

All rotors are therefore classified to indicate which can be balanced by normal or modified rigid rotor balancing techniques and which require some form of high speed balancing. Classification of rotors into different categories permits the use of simplified balancing methods for some rotors and ensures that for others, where necessary, an adequate balancing operation is performed by a suitable method.

As in the case of ISO 1940, this International Standard is not intended to serve as an acceptance specification for any rotor group, but rather to give indications of how to avoid gross deficiencies as well as exaggerated or unattainable requirements. Nevertheless, it may 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 limits or methods of manufacture and balance, satisfactory running conditions can most probably be expected. However, there may be cases where deviations from this International Standard may be necessary.

2 References

ISO 1925, *Balancing — Vocabulary.*

ISO 1940, *Balance quality of rotating rigid bodies.*

ISO 2041, *Vibration and shock — Vocabulary.*

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

3 Definitions

3.1 The definitions relating to mechanical balancing in International Standard ISO 1925 and many of the definitions relating to vibration and shock in ISO 2041 are applicable to this International Standard.

3.2 For the convenience of users of this International Standard, the following terms and definitions are repeated from ISO 1925 (in the case of 3.4 and 3.15 the entries are adapted from ISO 1925).

3.3 rigid rotor : A rotor is considered rigid when it can be corrected in any two (arbitrarily selected) planes and, after that correction, its unbalance does not significantly exceed the balancing tolerances (relative to the shaft axis) at any speed up to maximum service speed and when running under conditions which approximate closely to those of the final supporting system.

3.4 flexible rotor : A rotor not satisfying definition 3.3 due to elastic deflection.

3.5 bearing support : The part, or series of parts, that transmits the load from the bearing to the main body of the structure.

3.6 foundation : A structure that supports the mechanical system.

NOTES

1 The foundation may be fixed in space or may undergo a motion that provides excitation for the supported system.

2 In the context of the balancing and vibration of rotating machines, the term "foundation" is usually applied to the heavy base structure on which the whole machine is mounted.

3.7 controlled initial unbalance : Initial unbalance which has been minimized by individual balancing of components and/or careful attention to design, manufacture and assembly of the rotor.

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

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

3.10 modal balancing : A procedure for balancing flexible rotors in which balance corrections are made to reduce the amplitude of vibration in the separate significant principal flexural modes to within specified limits.

3.11 n^{th} modal unbalance : That unbalance which affects only the n^{th} principal mode of the deflection configuration of a rotor/bearing system.

NOTE — This n^{th} modal unbalance is not a single unbalance but an unbalance distribution $u_n(z)$ in the n^{th} principal mode. It can be mathematically represented with respect to its effect on the n^{th} principal mode by a single unbalance vector $\vec{U}_n$ obtained from the formula :

$$\vec{U}_n = \int_0^1 \vec{u}_n(z)\, \phi_n(z)\, dz$$

where $\phi_n(z)$ is the mode function.

3.12 equivalent n^{th} modal unbalance : The minimum single unbalance $\vec{U}_{ne}$, equivalent to the n^{th} modal unbalance in its effect upon the n^{th} principal mode of the deflection configuration.

NOTES

1 There exists the relation $\vec{U}_n = \vec{U}_{ne}\, \phi_n(z_e)$ where $\phi_n(z_e)$ is the mode function value for $z = z_e$, the axial co-ordinate of the transverse plane where $\vec{U}_{ne}$ is applied.

2 A set of balance masses distributed in an appropriate number of correction planes and so proportioned that the mode under consideration will be affected, may be called the equivalent n^{th} modal unbalance set.

3 An equivalent n^{th} modal unbalance will affect some modes other than the n^{th} mode.

3.13 modal unbalance tolerance : With respect to a mode, that amount of modal unbalance that is specified as the maximum below which the state of unbalance in that mode is considered acceptable.

3.14 multiple-frequency vibration : A vibration at a frequency corresponding to an integral multiple of the rotational speed.

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

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

NOTE — The change of condition may be permanent or temporary.

3.16 low speed balancing (relating to flexible rotors) : A procedure of balancing at a speed where the rotor to be balanced can be considered rigid.

3.17 high speed balancing (relating to flexible rotors) : A procedure of balancing at speeds where the rotor to be balanced cannot be considered rigid.

4 Fundamentals of flexible rotor dynamics with respect to balancing

4.1 The motion of a flexible rotor

Consider a thin slice of a shaft perpendicular to the shaft axis (see figure 1 where for simplicity of illustration the cross-section of the shaft is shown to be circular). Assume that, when the shaft is not rotating, the shaft axis intersects the slice at its geometric centre E (it is assumed throughout this International Standard that the deflection of the shaft due to gravity is ignored). The mass centre C of the slice is in general offset from E by a small distance e due to the small imperfections unavoidably produced in the shaft during manufacture (from errors in casting, machining tolerances and so on). The mass m of the slice and the offset distance e form a measure of unbalance in the slice, namely $m \times e$.

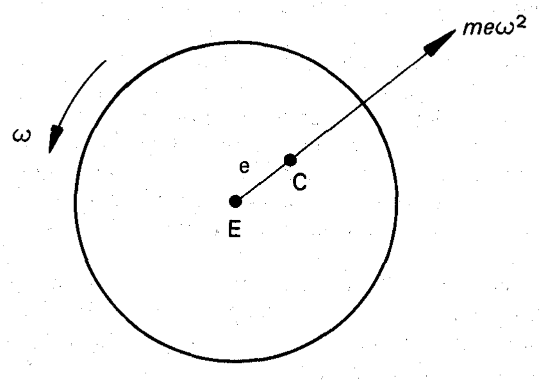


Figure 1 — Centrifugal force acting on an elementary slice of a shaft rotating about its mid-point

If the shaft starts to rotate about the shaft axis with an angular velocity ω, the thin slice starts to rotate in its own plane with speed ω about an axis through E. A centrifugal force $me\omega^2$ is thus experienced by the slice. This force is transverse to the shaft axis and may be accompanied at other cross sections along the shaft by similar forces which are likely to vary in magnitude and direction along the shaft. These forces cause the shaft to bend and the deflection modifies the resultant forces experienced by the shaft.

Satisfactory operation of the shaft can be specified in terms of one of the following :

a) vibration induced by the unbalance forces;

b) limits on the resultant forces applied by the shaft to the bearings;

c) residual unbalance.

In all cases in which it is necessary to reduce the unbalance forces, this is usually achieved by attaching a suitable axial distribution of correction masses along the shaft. It is not practical, and indeed not necessary, to balance the shaft exactly (that is to make e zero at all cross-sections along the shaft), so that there will invariably be some residual unbalance distributed along the shaft.

4.2 Unbalance distribution

Apart from any special design features, the axial distribution of unbalance along the rotor is likely to be random. The distribution may be significantly influenced by the presence of large local unbalances arising from shrink-fitted discs, couplings, etc.

The method of construction can significantly influence the magnitude and distribution of unbalance along the rotor. Rotors may be machined from a single forging or they may be constructed by fitting several components together. For example, jet engine rotors are constructed by joining many shell and disc components, whereas alternator rotors are usually manufactured from a single piece of material, though they may still have additional components fitted.

Since the unbalance distribution along a rotor is likely to be random, the unbalance distribution along two nominally identical rotors may be similar, but they will rarely be identical. Indeed, significant differences in initial unbalance and residual unbalance are common in otherwise identical rotors. The distribution of unbalance is of greater significance in a flexible rotor than in a rigid rotor because it determines the degree to which any flexural mode of vibration is excited. Moreover, the magnitude of the unbalance force at any point along the rotor depends on the bending deflection of the rotor at that point.

The correction of unbalance in axial planes along the rotor other than those in which the unbalance occurs may induce vibrations at speeds other than that at which the rotor was originally corrected. In many circumstances the vibrations may exceed specified tolerances, particularly at critical speeds.

Rotors which become heated during operation are susceptible to thermal distortions which can lead to variations in the unbalance.

4.3 Flexible rotor mode shapes

If damping is neglected, the modes of a rotor are the flexural principal modes and, for a rotor supported in "isotropic" bearings, are plane curves rotating about the shaft axis. Typical curves for the three lowest principal modes for a simple rotor supported in flexible bearings near to its ends are illustrated in figure 2.

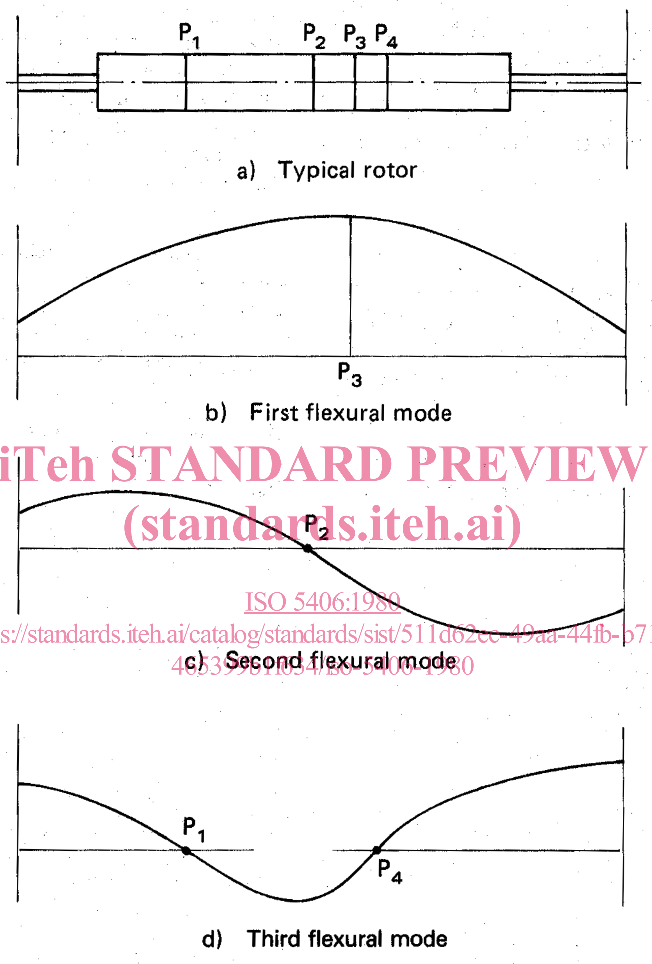


Figure 2 — Typical mode shapes for flexible rotors on flexible supports

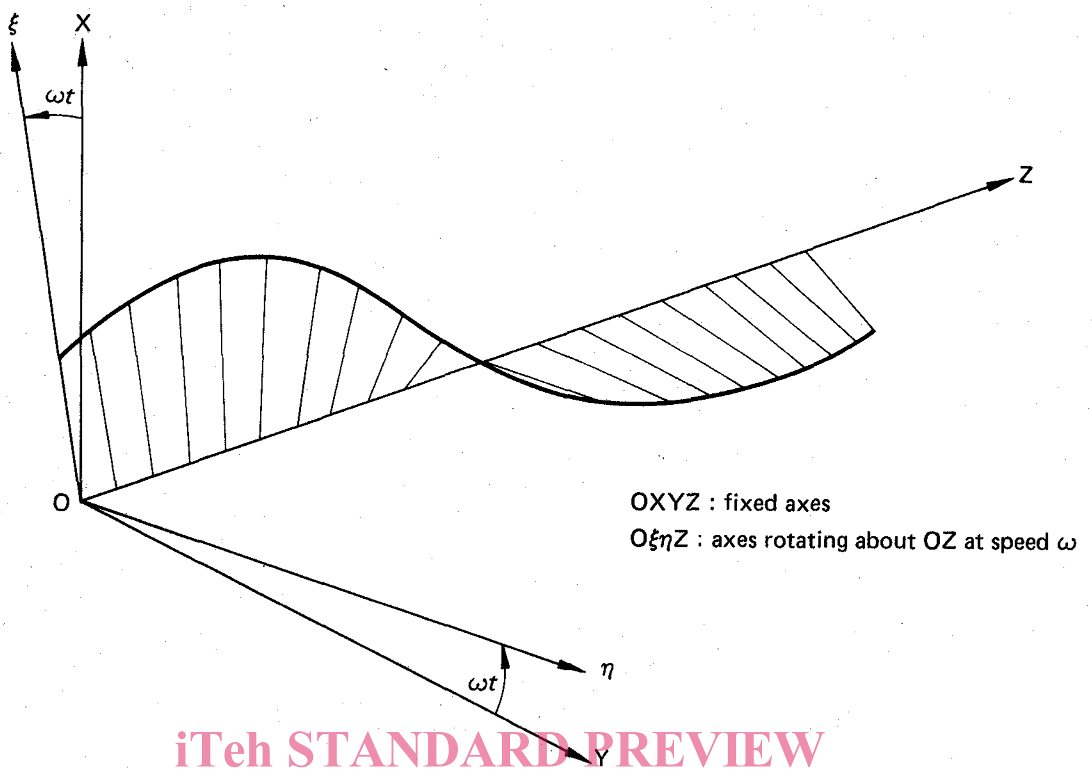


Figure 3 — Possible damped second mode shapes

For a damped rotor bearing system the flexural modes may be space curves rotating about the shaft axis, especially in the case of substantial damping, arising perhaps from fluid-film bearings. A possible substantially damped second mode is illustrated in figure 3. 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 are strongly influenced by the dynamic properties and axial locations of the bearings and their supports.

4.4 Response of a flexible rotor to unbalance

The unbalance distribution can be expressed in terms of modal components and the deflection in each mode is caused by the corresponding modal component of unbalance. Moreover, the response of the rotor in the vicinity of a critical speed is usually predominantly in the associated mode. The rotor modal response is a maximum at the rotor critical speed corresponding to that mode. Thus, when a rotor rotates at a speed near to a critical speed, it is disposed to adopt a deflection shape corresponding to the mode associated with this critical speed. The degree to which large amplitudes of rotor deflection occur in these circumstances is determined both by the component of unbalance in the mode in question and by the amount of damping experienced by the rotor system in this condition.

If the component of unbalance in a particular mode is reduced by a number of discrete masses, then the corresponding modal component of deflection is similarly reduced. The reduction of the modal components in this way forms the basis of two of the balancing procedures described in annex A.

If the rotor has a speed close to its first flexural critical value, then the deflection shape of the rotor tends to approximate to that shown in figure 2 b). Similarly, the deflection shapes of the rotor when rotating at speeds in the vicinity of its second or third critical speeds resemble those shown in figure 2 c) and 2 d). Similar comments apply to the higher modes.

Principal modes of the type shown in figures 2 b) to 2 d) determine the modal components of unbalance. Moreover the balancing effect produced by a given correction in a particular mode depends on the ordinate on the mode shape curve at the axial location of the correction. Thus a balancing mass attached to the rotor in figure 2 a) in the plane P_2 will produce no change in the response in the second mode. Similarly a correction mass attached in either P_1 or P_4 will not affect the response in the third mode. Conversely, a balancing mass in plane P_3 will produce the maximum effect on the first mode. If the rotor-bearing system has substantial damping, the rotor deflection will form space curves, which are related to the damped mode shapes mentioned above. A typical deflection curve under such circumstances for speeds near the second critical speed would resemble that shown in figure 3.

4.5 Objectives of flexible rotor balancing

It has already been observed that it is not practical to balance a rotor exactly, that is, to ensure that the offset e is zero at all cross-sections along the rotor. Indeed, the aims of balancing are many and are primarily determined by the operational requirements of the machine. Before balancing any particular rotor it is desirable to decide what criteria of balance can be regarded as adequate. In this way the balancing process can be made efficient and economic and still satisfy the needs of the user.

Balancing in general is usually a process whereby rotor vibration or bearing forces are reduced to within appropriate specified tolerances. For some applications it is only necessary to balance rotors at one speed, but in many cases the vibrations or oscillatory forces due to unbalance must be reduced to low levels over a range of speed, including several critical speeds.

It should also be remembered that the eventual aim of balancing is to ensure satisfactory running of the rotor in its operating environment and not only in the balancing facility. To this end it may be desirable to simulate service support conditions in specifying bearings for the balancing facility. Thus the bearings and pedestals used for balancing should reproduce to the necessary extent the mass and stiffness of the service bearings.

4.6 Provision for correction planes

Correction masses are attached to a rotor to counteract the effect of an initial lack of balance. Although the unbalance invariably has a random distribution along the rotor, the correction masses are discrete in magnitude, in axial location along the rotor, and in angular location around the rotor. Rotors are often balanced in a modal sequence and in this process correction masses are situated along the rotor so that at each stage in the balancing procedure the new correction masses do not disturb modes already balanced (see annex A). The exact number of axial locations along the rotor that are needed for this process depends to some extent on the particular balancing procedure which is adopted. Generally, however, if the operating speed of the rotor exceeds its n^{th} critical speed, then at least $(n + 2)$ correction planes (transverse to the rotor axis) are likely to be needed along the rotor.

In practice the number of axial locations that are available for use as correction planes is often limited by design considerations (and in field balancing by limitations on accessibility). An adequate number of correction planes should be included at the design stage. For turbine rotors, usually two end planes and a mid-span plane are available. For generator rotors, a minimum of two end planes and a mid-span plane have customarily been available in the balancing facility. For larger machines (with more flexible rotors and more critical speeds below the maximum operating speed), two additional planes or multiple planes have been used by some manufacturers. Centrifugal compressor rotors are usually assembly-balanced in the end planes only after each disc and the shaft have been separately balanced in a low speed balancing machine. With such restrictions, considerable ingenuity is often required from the balancing engineer.

4.7 Rotors coupled together

When assessing coupled rotors, the nomenclature of the critical speeds requires some clarification for the following reason. Consider two rotors. Each rotor has a series of critical speeds and mode shapes that are usually different from those of the other rotor. When the two rotors are coupled together, the complete unit will also have a series of critical speeds and mode shapes. However, these speeds are neither equal to nor simply related to the critical speeds of the uncoupled rotors. Moreover, the deflection shape of each part of the coupled unit, when vibrating in one of the coupled principal modes, need not be simply related to any mode shape of the corresponding uncoupled rotor. In theory, therefore, the unbalance distribution along two or more coupled rotors should be treated in terms of modal components with respect to the coupled system and not to the modes of the uncoupled rotors.

In practice, however, it is desirable for simplicity of production processes that each rotor should be balanced separately as an uncoupled shaft. Although no simple general indications can be formulated, it is often possible to make approximate comparisons between the coupled and uncoupled mode shapes and critical speeds, and in most cases such an approximate is adequate to ensure satisfactory operation of the coupled rotors. The degree to which this simple technique is practicable depends on the mode shapes and the critical speeds of the uncoupled and coupled rotors, the stiffness of the coupling and coupling shaft sections, the distribution of unbalance (which is not known) and the unbalance and especially the machining errors in the coupling assembly. The success of the technique is assisted if the coupling if flexible. it must, however be emphasized that, strictly speaking, each rotor may only be considered separately for balancing purposes provided that, when forming part of the coupled system, its modal deflection shapes do not differ significantly from its uncoupled mode shapes.

On the other hand, balancing a single-span rotor according to its mode shapes is not an aim in itself. If modal balancing techniques are used, the final goal is to gain information, as accurately as possible, about the unbalance and its distribution along the rotor, and as far as possible to correct the unbalance over the speed range. if this goal is reached it is not necessary that the modal shapes or natural frequencies should be the same when balancing and when the rotor is running in situ.

When two rotors, each separately supported in its own bearings, are coupled together, provided the coupling does not form a significant overhung mass on either rotor by comparison with the rotor mass, it is probable that each rotor may be balanced separately as an independent rotor.

5 Classification

5.1 For the purposes of this International Standard, rotors are divided into five main classes as shown below and in the table. Each class requires different balancing techniques.

Class 1 — A rotor whose unbalance can be corrected in two (arbitrarily selected) planes so that, after the correction, its unbalance does not change significantly at any speed up to the maximum service speed. Rotors of this type can be corrected by rigid rotor balancing methods.[1)]

Class 2 — A rotor that cannot be considered rigid but that can be balanced using modified rigid rotor balancing techniques.

1) Recommendations for the balancing of rigid rotors are given in ISO 1940.

Class 3 — A rotor that cannot be balanced using modified rigid rotor balancing techniques but instead requires the use of high speed balancing methods.

Class 4 — A rotor that could fall into classes 1, 2 or 3 but has in addition one or more components that are themselves flexible of flexibly attached.

Class 5 — A rotor that could fall into class 3 but for some reason, for example economy, is balanced for one speed of operation only.

NOTE — The number of modes that are considered in the balancing operation is not necessarily an indication of the number of critical speeds through which a rotor passes as it is run up to maximum service speed.

5.2 Class 2 rotors are subdivided (see the table) into :

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

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

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

5.3 Class 3 is sub-divided (see the table) because the balancing techniques, criteria and bearing requirements may differ substantially for different rotors.

5.4 A sub-division of class 4 rotors is indicated in 7.4.

6 Factors governing the classification of class 2 rotors

6.1 General

A low speed balancing machine considers only the static and couple unbalances in a rotor and does not evaluate the effect of deflection due to modal components of unbalance. Some rotors that are balanced in a low speed machine may therefore vibrate excessively both when running through critical speeds and at service speed. However, it is possible in some circumstances to balance a rotor in a low speed machine so that not only are the static and couple unbalances cancelled but also the remaining modal unbalances are sufficiently small to ensure satisfactory running when the rotor is installed in its final environment.

The amount of modal unbalance remaining in a rotor after the static and couple unbalances are corrected will depend partially on the modal shapes of the rotor and the axial positions of the unbalances relative to the correction planes used.

To assess to what extent it is likely that low speed balancing will be successful, it is necessary to consider the following factors.

6.2 Mass distribution of the rotor

No general rule can be laid down regarding the mass distribution of the rotor except that, if the axial positions of the unbalances are known, the balancing planes should be provided in the most suitable axial positions to cancel the effect of the unbalances.

6.3 Rotors made up of individual components

When a rotor is composed of separate components that are distributed axially and mounted concentrically on a shaft, it would considerably increase the probability of a satisfactory balance if one or both of the following methods of manufacture were adopted :

a) Each component and the shaft should be individually balanced as a rigid rotor to specified tolerances before assembly. In addition, the concentricities of the shaft diameters or other location features that position the individual components on the shaft should be held to close tolerances relative to the shaft axis.

The concentricities of the mandrel diameters or other locating features that position each individual component on the mandrel should likewise be held within close tolerances relative to the shaft axis of the mandrel. Mandrel concentricity may be checked by turning the workpiece on the mandrel through 180°.

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

It is advisable to check by calculation the unbalance produced by the eccentricities for the minimum practicable manufacturing tolerances.

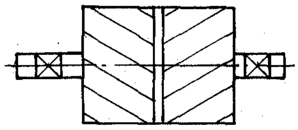
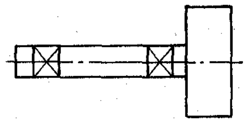
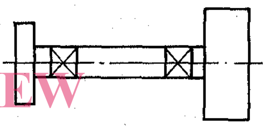
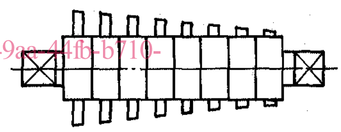
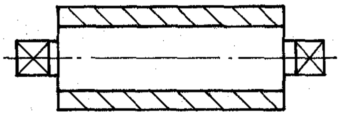
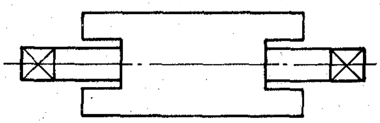
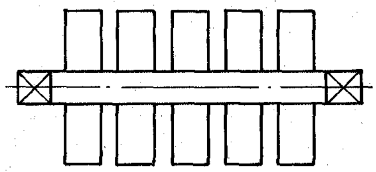
When calculating the effect of the eccentricities of these location features on the mandrel and on the shaft, it is important to note that the effect of the eccentricities can be cumulative on the final assembly. If there are many effects to be taken into account, some statistical approach may be suitable. It may be found that the correction which may finally be necessary to compensate for these probable eccentricities in the mounting is large compared to the correction that is likely to be required in the component itself.

In such cases, the pre-balancing of the component on a separate mandrel is of relatively little value and it may then be considered preferable to proceed by the method described in b) below.

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

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

Table — Classification of rotors

Class of rotor	Description	Example
Class 1 (rigid rotors)	A rotor whose unbalance can be corrected in any two (arbitrarily selected) planes so that, after that correction, its unbalance does not change significantly at any speed up to maximum service speed.	 Gear wheel
Class 2 (quasi-rigid rotor)	*A rotor that cannot be considered rigid but that can be balanced using modified rigid rotor balancing techniques.*	–
Rotors in which the axial distribution of unbalance is known		
Class 2a	A rotor with a single transverse plane of unbalance, for example a single mass on a light flexible shaft whose unbalance can be neglected.	 Grinding wheel
Class 2b	A rotor with two transverse planes of unbalance for example two masses on a light shaft whose unbalance can be neglected.	 Grinding wheel with pulley
Class 2c	A rotor with more than two transverse planes of unbalance.	 Compressor rotor
Class 2d	A rotor with uniformly or linearly varying unbalance.	 Printing press roller
Class 2e	A rotor consisting of a rigid mass of significant axial length supported by flexible shafts whose unbalance can be neglected.	 Computer memory drum
Rotors in which the axial distribution of unbalance is not known		
Class 2f	A symmetrical rotor with two end correction planes; whose maximum speed does not significantly approach second critical speed; whose service speed range does not contain first critical speed, and which has a controlled initial unbalance.	 Multi-stage centrifugal pump