

British Standard

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Balance quality requirements of rigid rotors —

**Part 1: Method for determination of
permissible residual unbalance —**

**[ISO title: Mechanical vibration —
Balance quality requirements of rigid
rotors — Part 1: Determination of
permissible residual unbalance]**

Committees responsible for this British Standard

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British Engine Group of SMMT
 British Maritime Technology
 British Steel Corporation
 Department of Trade and Industry (National Engineering Laboratory)
 Electricity Supply Industry in England and Wales
 Institute of Sound and Vibration Research
 Institution of Electronic and Radio Engineers
 Institution of Mechanical Engineers
 Lloyds Register of Shipping
 Ministry of Defence
 Motor Industry Research Association
 Power Generation Association (BEAMA Ltd.)
 Society of British Aerospace Companies Limited
 Society of Environmental Engineers
 United Kingdom Atomic Energy Authority

The following bodies were also represented in the drafting of the standard, through subcommittees and panels:

Engineering Equipment and Materials Users Association
 Coopted member

This British Standard, having been prepared under the direction of the General Mechanical Engineering Standards Committee, was published under the authority of the Board of BSI and comes into effect on 30 June 1987

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National foreword

This Part of BS 6861 has been prepared under the direction of the General Mechanical Engineering Standards Committee. It is identical with ISO 1940/1-1986 “*Mechanical vibration — Balance quality requirements of rigid motors — Part 1: Determination of permissible residual unbalance*” which was prepared by Technical Committee ISO/TC 108 of the International Organization for Standardization (ISO) and in the development of which the United Kingdom played an active part. A further Part of this British Standard, concerned with errors associated with the determination of residual unbalance, is to be published in due course. BS 6861-1 supersedes BS 5265-1, which has been withdrawn.

Terminology and conventions. The text of the international standard has been approved as suitable for publication as a British Standard without deviation. Some terminology and certain conventions are not identical with those used in British Standards; attention is drawn especially to the following.

The comma has been used as the decimal marker. In British Standards it is current practice to use a full point on the baseline as the decimal marker.

Wherever the words “International Standard” or “part of ISO 1940” appear, referring to this standard, they should be read as “British Standard” or “Part of BS 6861” respectively.

Cross-references

International standard	Corresponding British Standard
ISO 1925-1981	BS 3851:1982 Glossary of terms used in the mechanical balancing of rotating machinery (Identical)
ISO 5343-1983	BS 5265 Mechanical balancing of rotating bodies Part 3:1984 Recommendations for criteria for evaluating flexible rotor balance (Identical)
ISO 5406-1980	Part 2:1981 Methods for the mechanical balancing of flexible rotors (Identical)

The Technical Committee has reviewed the provisions of ISO 2371 and ISO 2953, to which reference is made in the text, and has decided that they are acceptable for use in conjunction with this standard.

A related British Standard to ISO 2371-1974 is BS 4924:1973 “Specification. Description and evaluation of field balancing equipment”.

A related British Standard to ISO 2953-1985 is BS 3852 “Balancing machines” Part 1:1979 “Method of description and evaluation”.

Textual errors. When adopting the text of the international standard, the textual errors listed below were discovered. They have been marked in the text and have been reported to ISO in a proposal to amend the text of the international standard.

- a) In **3.1**, paragraph 2, line 2, 3.15 g mm should read 3.16 g mm.
- b) In Figure 1(a) the value 3.15 should read 3.16.
- c) In **7.3.3.1**, line 12, *h* should be read *b*.

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Summary of pages

This document comprises a front cover, an inside front cover, pages i to iv, pages 1 to 16, an inside back cover and a back cover.

This standard has been updated (see copyright date) and may have had amendments incorporated. This will be indicated in the amendment table on the inside front cover.





0 Introduction

Balancing is the process of attempting to improve the mass distribution of a body so that it rotates in its bearings without unbalanced centrifugal forces. Of course, this aim can be attained only to a certain degree; even after balancing, the rotor will possess residual unbalance.

The measuring equipment available today enables unbalance to be reduced to low limits. However, it would be uneconomical to exaggerate the quality requirements. It has therefore become necessary to determine to what extent the unbalance should be reduced, and where the optimum economic and technical compromise on balance quality requirements would be struck.

It is not readily possible to draw conclusions as to the permissible residual unbalances from any existing recommendations on the assessment of the vibratory state of machinery, since there is often no easily recognizable relation between the rotor unbalance and the machine vibrations under operating conditions. The amplitude of the once-per-revolution vibrations is influenced by characteristics of the rotor, of the machine, of the structure and of the foundation, and by the proximity of the service speed to the various resonance frequencies, etc. Moreover, the machine vibrations may be due only in part to the presence of rotor unbalance.

1 Scope and field of application

This part of ISO 1940 gives recommendations for determining unbalance and for specifying related quality requirements of rigid rotors; it specifies

- a) a representation of unbalance in one or two planes;
- b) methods for determining permissible residual unbalance;
- c) methods for allocating it to the correction planes;
- d) methods for identifying the residual unbalance state of a rotor by measurement;
- e) a summary of errors associated with the residual unbalance identification.

In Table 1 and Figure 2 recommendations are given, based on worldwide experience, concerning the balance quality requirements of rigid rotors, according to their type, mass and maximum service speed.

This part of ISO 1940 is also intended to facilitate the relations between manufacturer and user of machines. Terminology specified in this part of ISO 1940 may be used for establishing technical specifications. (For definitions, see ISO 1925.)

Detailed consideration of errors associated with the determination of residual unbalance is not included in this part of ISO 1940 (ISO 1940/2 will deal with these errors). This part of ISO 1940 does not define permissible residual unbalances for flexible rotors; these are covered in ISO 5343. The methods for balancing are not described.

The recommended balance quality grades are not intended to serve as acceptance specifications for any rotor group, but rather to give indications of how to avoid gross deficiencies as well as exaggerated or unattainable requirements; they may also serve as a basis for more involved investigations, for example, when a more exact determination of the required balance quality by measurement in the laboratory or in the field is necessary. If due regard is paid to the recommended limits, satisfactory running conditions can most probably be expected. However, there may be cases when deviations from these recommendations become necessary, e.g. because of unusual construction or geometry.

2 References

- ISO 1925, *Balancing — Vocabulary*.
 ISO 2371, *Field balancing equipment — Description and evaluation*.
 ISO 2953, *Balancing machines — Description and evaluation*.
 ISO 5343, *Criteria for evaluating flexible rotor balance*.
 ISO 5406, *The mechanical balancing of flexible rotors*.

3 Pertinent aspects of the balancing problem

3.1 Representation of a state of unbalance

One and the same state of unbalance of a rigid rotor can be represented by vectorial quantities in various ways, as shown in Figure 1 a) to Figure 1 f). For most rotors, unbalances are measured in two planes as illustrated in Figure 1 a) to Figure 1 c). The unbalance correction process generally takes place in this way.

For example, if the representation of Figure 1a) is adopted, unbalance corrections of 2,24 g·mm and 3,15 g·mm¹⁾ would be applied in planes I and II respectively, at the appropriate angular orientation around the rotor. Comparison of Figure 1 a) and Figure 1 c) shows that a shorter distance between the correction planes normally leads to a larger sum of corrections and increases the angular difference between their vectors.

¹⁾ See national foreword for details of textual errors.

Alternatively, a representation similar to Figure 1 d) may be used; in such a case, correction would have to be made in the plane of the resultant unbalance vector, together with corrections in each of planes I and II.

If an investigation of the unbalance effect on the vibratory behaviour of a whole machine is intended, an unbalance resolution according to Figure 1 e) may be useful, where the point S denotes the mass centre of the rotor.

The representation in Figure 1 f), with the resultant unbalance vector passing through the centre of unbalance, C, may be useful for some rotors. The associated couple unbalance is a minimum in this case and the couple vectors lie in a plane normal to the vector of the resultant unbalance.

3.2 Unbalance effects

An unbalanced rotating body will cause not only forces on its bearings and foundation, but also vibrations of the machine. At any given speed, both effects depend essentially on the geometric proportions and mass distribution of rotor and machine, as well as on the dynamic stiffness of the bearings and the foundation.

In many cases, the static unbalance is of primary importance as compared with the couple unbalance. Two unbalances in different planes in the same direction usually cause a greater disturbance than two equal unbalances in opposite directions.

Similarly, there are cases in which couple unbalance is especially disturbing. For example, consider a rotor where the distance between the bearings is smaller than the distance between the correction planes (a situation encountered in a rotor with overhung disks at both ends); under such circumstances, the bearing load due to a couple unbalance may be larger than that caused by a static unbalance.

3.3 Rotors with one correction plane

For disc-shaped rotors, the use of only one correction plane may be sufficient, provided that the bearing distance is sufficiently large and the disc rotates with sufficiently small axial run-out. Whether these conditions are fulfilled shall be investigated in each individual case. After single-plane balancing has been carried out on a sufficient number of rotors of a particular type, the greatest residual couple unbalance is determined and divided by the bearing distance. If, even in the worst case, the unbalances found in this way are acceptable, it can be expected that single-plane balancing is sufficient.

3.4 Rotors with two correction planes

If a rigid rotor does not satisfy the conditions specified in 3.3 for the disc-shaped rotor, two correction planes are needed. This type of balancing is called two-plane (“dynamic”) balancing in contrast to the single-plane (“static”) balancing described in 3.3. For single-plane balancing, only static equilibrium in any angular position of the rotor is required. For two-plane balancing, it is necessary for the rotor to rotate, since otherwise the residual couple unbalance would remain undetected.

The permissible residual unbalance at each of two correction planes will be dependent on the position of the correction planes and the bearings and also on the relative phase angle between the two residual unbalances. Three ways for determining the permissible residual unbalance are given in clause 6. Methods for allocating the permissible residual unbalance to the correction planes are described in clause 7.

3.5 Assemblies

Some rotors may be balanced as integral single components, others as assemblies. For each assembly, the unbalances of the component parts shall be added vectorially and any unbalances resulting from inaccuracies of assembly shall be taken into account, particular attention being paid to the fact that the parts may be assembled later in a different position from that in the balancing machine.

If the unbalance tolerance for an assembly cannot be achieved by balancing each component separately, the assembly shall be balanced as a unit.

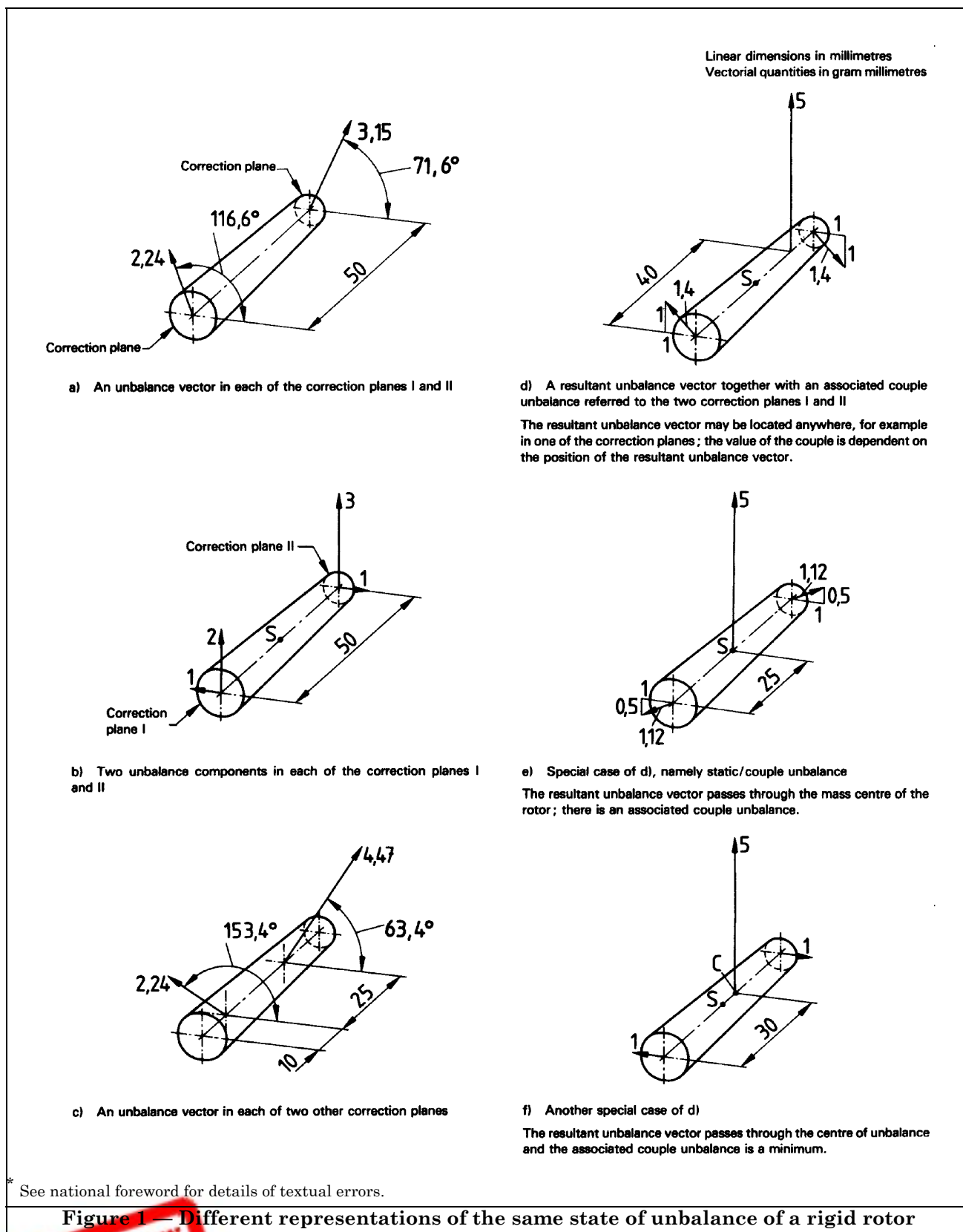
If the individual components are balanced separately, prior agreement should be reached as to the attachment of connecting elements such as bolts and keys. (A future International Standard dealing with the handling of shaft keys during balancing is in preparation.)

4 Permissible unbalance related to rotor mass

In general, the larger the rotor mass, the greater the permissible residual unbalance. It is, therefore, appropriate to relate the value of the permissible residual unbalance, U_{per} , to the rotor mass, m , in terms of permissible residual specific unbalance value, e_{per} , as given by the following formula:

$$e_{\text{per}} = \frac{U_{\text{per}}}{m}$$

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In the special case where all unbalances present in a rotor can be reduced to the equivalent system of a single unbalance located uniquely in one transverse plane along the shaft axis such that the couple unbalance is zero, the permissible residual specific unbalance value, e_{per} , can be considered as equivalent to a permissible displacement of the mass centre of the rotor from the shaft axis. In all other cases, one of which is shown in Figure 1, the equivalent residual displacement, e , of the mass centre, after two-plane balancing to the permissible values, is smaller than the specific permissible residual unbalance value, e_{per} .

5 Quality grades relating service speed and permissible residual specific unbalance

Experience shows that, in general, for rotors of the same type, the permissible residual specific unbalance value, e_{per} , varies inversely as the speed of the rotor in the speed range shown in Figure 2 for a given balance quality grade; this relationship is given by the following formula:

$$e_{\text{per}} \times v = \text{constant}$$

where v is the angular velocity of the rotor at maximum service speed.

This relationship follows also from the fact that, for geometrically similar rotors running at equal peripheral velocities, the stresses in rotors and bearings are the same. The balance quality grades (given in Table 1 and illustrated in Figure 2) are based on this relationship.

6 Determination of balance quality requirements

6.1 General

The balance quality requirements can be determined by three methods as described in 6.2 to 6.4. The first method is based on empirical quality grades derived from long-term practical experience with a large number of different rotors (see 6.2). The second method is an experimental one and is often used in mass production balancing (see 6.3). The third method is used if permissible bearing forces due to unbalance are specified (see 6.4).

Choice of method should be agreed between the manufacturer and user.

6.2 Balance quality requirements based on established grades

On the basis of clauses 4 and 5, balance quality grades have been established which permit a classification of the quality requirements. Each balance quality grade in Table 1 comprises a range of permissible residual specific unbalance from an upper limit to zero, the upper limit being given by a certain magnitude of the product of the relationship ($e_{\text{per}} \times v$), expressed in millimetres per second; balance quality grades are designated according to the product of the relationship, i.e. if the product of $e_{\text{per}} \times v$ is equal to 630 mm/s, the balance quality grade is designated **G630**.

Balance quality grades are separated from each other by a factor of 2,5. A finer grading may be necessary in some cases, especially when high precision balancing is required.

The upper limits of e_{per} are plotted against the maximum service speed in Figure 2. The permissible residual unbalance given by $U_{\text{per}} = e_{\text{per}} \times m$, where m is the rotor mass.

NOTE For balance quality grades **G1** and **G0,4**, the final balance quality requirements selected is a compromise between technical requirements and what is practically possible. The selected limit is usually associated with the minimal state of unbalance which can reasonably be repeated.

The recommended values in these balance quality grades can only be achieved in practice if the accuracy of shaft journals (roundness, etc.) in the rotor bearings and/or the bearing accuracy are sufficient. In order to achieve a balance quality grade **G1**, it is usually necessary to balance the rotor in its own service bearings, using belt, air or self-drives. In order to achieve balance quality grade **G0,4**, balancing usually needs to be carried out with the rotor mounted in its own housing and bearing and under service conditions and temperature; self-drive is generally required.

6.3 Balance quality requirements based on experimental determination

Experimental determination of the balance quality requirements is often carried out for mass production applications. Tests are commonly run *in situ*, although occasionally they may be carried out in balancing machines, provided that the characteristics of the balancing machine are essentially those of the service condition of the machine in which the rotor will be used.

The value of permissible residual unbalance in each correction plane is determined experimentally by introducing various test unbalances successively in each plane; the criterion to be chosen is given by the one most representative (e.g. vibrations, forces or noise caused by unbalance).

In two-plane balancing, the differing effects of unbalances with the same phase angle and of unbalance couples have to be taken into account. In addition, changes in the local environment, and/or in the rotor, which may occur in service, have to be taken into consideration.

6.4 Balance quality requirements based on specified permissible bearing forces

Where the effect of unbalance forces transmitted at the bearings into the supporting structure is of major concern and limits to these forces are specified, this has to be taken into consideration in the determination of permissible residual unbalance.

The value of permissible unbalance in each bearing plane can then be derived directly from the maximum permissible forces due to unbalance at each bearing. If the rotor is balanced in a balancing machine which measures the residual unbalance in the bearing planes, these values may be applied directly. However, if the residual unbalance is measured at other planes, the permissible residual unbalance at these planes can be calculated, using the methods described in clause 7, by defining U_{per} as the sum of the permissible residual unbalances in the bearing planes.

NOTE The derivation of the permissible residual unbalance in each bearing plane in terms of the maximum permissible forces due to unbalance at each bearing is dependent on many factors, including the service speed, the rotor mass distribution and the bearing support stiffness. However, in the special case of a rigid rotor supported by rigid bearings, the permissible residual unbalance at each bearing plane is equal to the maximum permissible force due to unbalance at that bearing divided by the square of the angular velocity at the maximum service speed.

7 Allocation of permissible residual unbalance to each correction plane on the basis of U_{per}

7.1 General

The balance quality requirements can be determined by one of the three methods described in clause 6. In the method described in 6.2 the balance quality requirements are determined as maximum permissible values of residual unbalance in each correction plane, and therefore no further allocation is required. However, the use of the method described in 6.1 (and in some cases the use of the method described in 6.3) leads to the total value of the permissible residual unbalance, U_{per} , and therefore implies allocation to each correction plane if more than one correction plane is used.

As a general rule, U_{per} should be allocated to the correction planes in such a way that the ratio of the residual unbalances referred to the bearing planes is in the same proportion as the permissible dynamic loads for the service bearings. Consequently, if the rotor is balanced in a balancing machine which measures unbalance in the planes of the service bearings, the above rule may be applied directly. In general, however, the residual unbalance is measured in planes other than the service bearing planes. Furthermore, there may be special requirements (e.g. vibration emission, noise, fatigue limitations) which require the permissible residual unbalance to be split between the service bearing planes in a different ratio. Therefore this clause describes a number of alternative methods for determining the permissible residual unbalance at each correction plane in terms of U_{per} .

NOTE Permissible dynamic loads for the service bearings may be deduced from bearing catalogues or from a knowledge of the allowable specific load, the length and the diameter of the bearing.

7.2 Single-plane balancing

For rotors with one correction plane, the permissible residual unbalance as measured in this plane is equal to U_{per} .

7.3 Two-plane balancing

7.3.1 General

A number of methods for determining the permissible residual unbalance in a rotor having two correction planes are given in 7.3.2 and 7.3.3.

Three simple methods are described in 7.3.2 which will lead, in practically all cases, to reasonable and justifiable permissible residual unbalances in each correction plane, so that, for any phase relationship between them, the maximum dynamic loads on both bearings are in good accordance with the static bearing load ratio by weight. These simple methods, although approximate, have been successfully applied to many rotors.

The method described in 7.3.3.1 is completely general and can be applied to all types of rotor. This method takes into account the position of the correction planes and the most unfavourable case of phase angle relationship between the residual unbalances in the correction planes.

Table 1 — Balance quality grades for various groups of representative rigid rotors

Balance quality grade	Product of the relationship ($e_{\text{per}} \times v$) ^{ab} mm/s	Rotor types – General examples
G4 000	4 000	Crankshaft/drives ^c of rigidly mounted slow marine diesel engines with uneven number of cylinders ^d
G1 600	1 600	Crankshaft/drives of rigidly mounted large two-cycle engines
G630	630	Crankshaft/drives of rigidly mounted large four-cycle engines Crankshaft/drives of elastically mounted marine diesel engines
G250	250	Crankshaft/drives of rigidly mounted fast four-cylinder diesel engines ^d
G100	100	Crankshaft/drives of fast diesel engines with six or more cylinders ^d Complete engines (gasoline or diesel) for cars, trucks and locomotives ^e
G40	40	Car wheels, wheel rims, wheel sets, drive shafts Crankshaft/drives of elastically mounted fast four-cycle engines (gasoline or diesel) with six or more cylinders ^d Crankshaft/drives of engines of cars, trucks and locomotives
G16	16	Drive shafts (propeller shafts, cardan shafts) with special requirements Parts of crushing machines Parts of agricultural machinery Individual components of engines (gasoline or diesel) for cars, trucks and locomotives Crankshaft/drives of engines with six or more cylinders under special requirements
G6,3	6,3	Parts of process plant machines Marine main turbine gears (merchant service) Centrifuge drums Paper machinery rolls; print rolls Fans Assembled aircraft gas turbine rotors Flywheels Pump impellers Machine-tool and general machinery parts Medium and large electric armatures (of electric motors having at least 80 mm shaft height) without special requirements Small electric armatures, often mass produced, in vibration insensitive applications and/or with vibration-isolating mountings Individual components of engines under special requirements
G2,5	2,5	Gas and steam turbines, including marine main turbines (merchant service) Rigid turbo-generator rotors Computer memory drums and discs Turbo-compressors Machine-tool drives Medium and large electric armatures with special requirements Small electric armatures not qualifying for one or both of the conditions specified for small electric armatures of balance quality grade G6,3 Turbine-driven pumps
G1	1	Tape recorder and phonograph (gramophone) drives Grinding-machine drives Small electric armatures with special requirements
G0,4	0,4	Spindles, discs, and armatures of precision grinders Gyroscopes

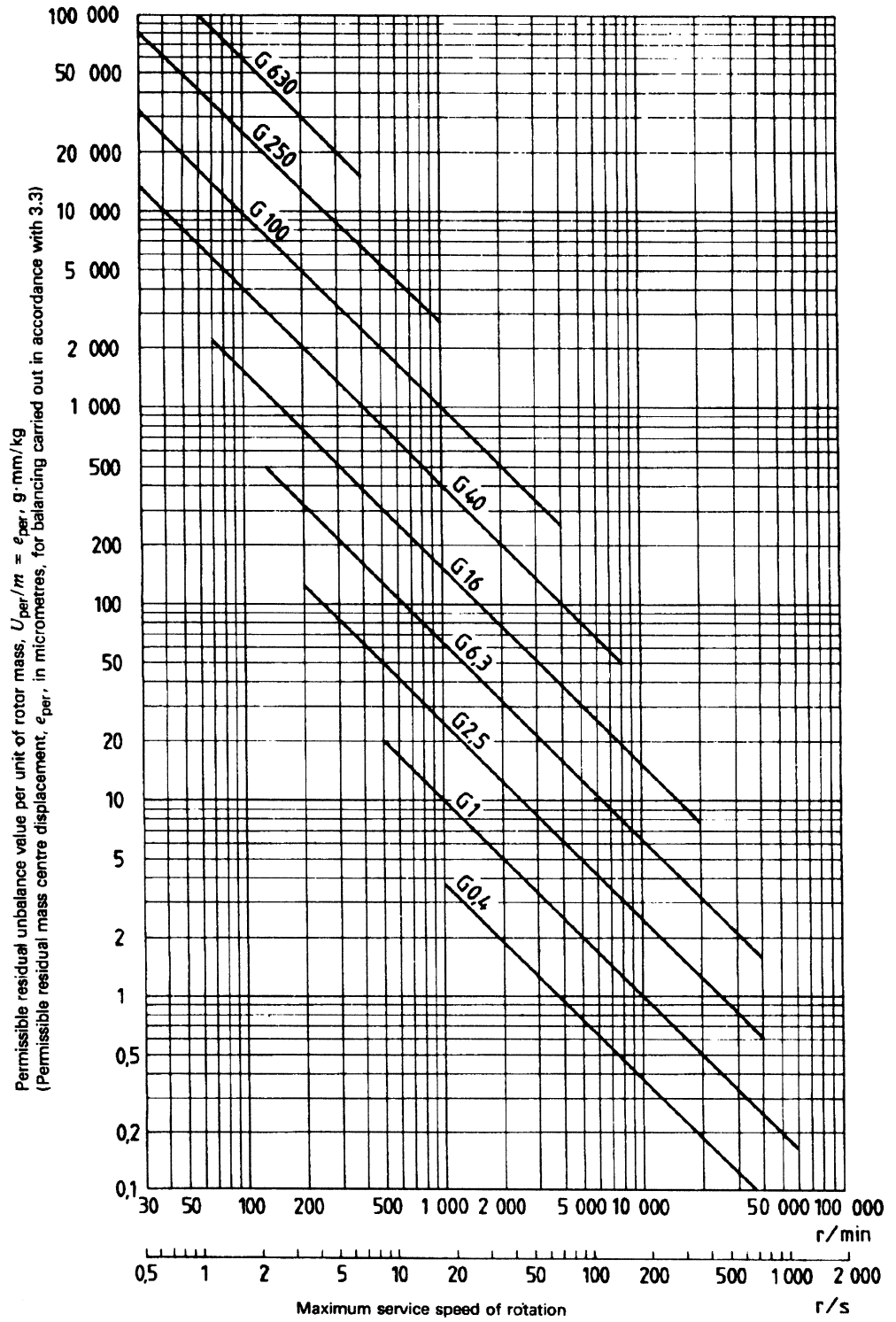
^a $v = 2\pi n/60 \approx n/10$, if n is measured in revolutions per minute and v in radians per second.

^b For allocating the permissible residual unbalance to correction planes, see clause 7.

^c A crankshaft/drive is an assembly which includes a crankshaft, flywheel, clutch, pulley, vibration damper, rotating portion of connecting rod, etc. (see 3.5).

^d For the purposes of this part of ISO 1940, slow diesel engines are those with a piston velocity of less than 9 m/s; fast diesel engines are those with a piston velocity of greater than 9 m/s.

^e In complete engines, the rotor mass comprises the sum of all masses belonging to the crankshaft/drive described in note 3 above.



NOTE — The numerical value after the letter G is equal to the product of ($e_{per} \times \omega$), expressed in millimetres per second.

Figure 2 — Maximum permissible residual specific unbalance value corresponding to various balance quality grades



There are certain types of rotors, such as overhung rotors with both correction planes located on the same overhang, where the bearing spacing is significantly greater than the distance between the correction planes. For these rotors, there may be a significant difference between the permissible residual unbalance for the case when the residual unbalance in both correction planes is either in phase or 180° out of phase. Although the method described in 7.3.3.1 may be applied to such rotors, it can, in certain circumstances, lead to the rotor being more finely balanced than is necessary. For this reason, a method in which the residual unbalance is measured in alternative planes which are not coincident with the correction planes is described in 7.3.3.2. This ensures that advantage can be taken of a favourable phase relationship without the need to have to measure the phase angles directly.

The methods described will give acceptable results for most rotors, but in a few special cases the calculations may result in balancing tolerances so fine that they are unrealistic. In these cases, consideration shall be given to the extent to which these tolerances can be relaxed without the load capacities of the bearings or other limiting parameters being exceeded.

NOTE Choice of method shall be at the discretion of the rotor manufacturer.

7.3.2 Simplified approximate methods

For most rotors, the methods described in 7.3.2.1 to 7.3.2.3 should yield satisfactory results if they are applied in accordance with the recommendations.

7.3.2.1 Distance between correction planes less than the bearing span (see Figure 3)

This simplified method is applicable to rotors which satisfy the following service conditions:

- the centre of gravity is located within the mid third of the bearing span;
- the distance between the correction planes is greater than one third of and less than the bearing span;
- the correction planes are essentially equidistant from the centre of gravity of the rotor.

For these rotors, allocate half of the permissible residual unbalance, U_{per} , to each correction plane, i.e.

$$U_{\text{perI}} = U_{\text{perII}} = \frac{U_{\text{per}}}{2}$$

If the service conditions described in a) and b) are met, but the correction planes are not essentially equidistant from the centre of gravity, allocate the permissible residual unbalance in each correction plane such that

a) the sum of the permissible residual unbalances in the correction planes is equal to U_{per} ;

b) the ratio of the permissible residual unbalance in each correction plane is the same as the ratio of the distance from each correction plane to the mass centre of the rotor with the larger residual unbalance being permitted in the correction plane closest to the mass centre. However, notwithstanding the above, the ratio of the larger to smaller permissible residual unbalance should never exceed 0,7/0,3, i.e.

$$0,3 U_{\text{per}} < U_{\text{perI}} = U_{\text{per}} \frac{h_{\text{II}}}{b} < (0,7 U_{\text{per}})$$

$$0,3 U_{\text{per}} < U_{\text{perII}} = U_{\text{per}} \frac{h_{\text{I}}}{b} < (0,7 U_{\text{per}})$$

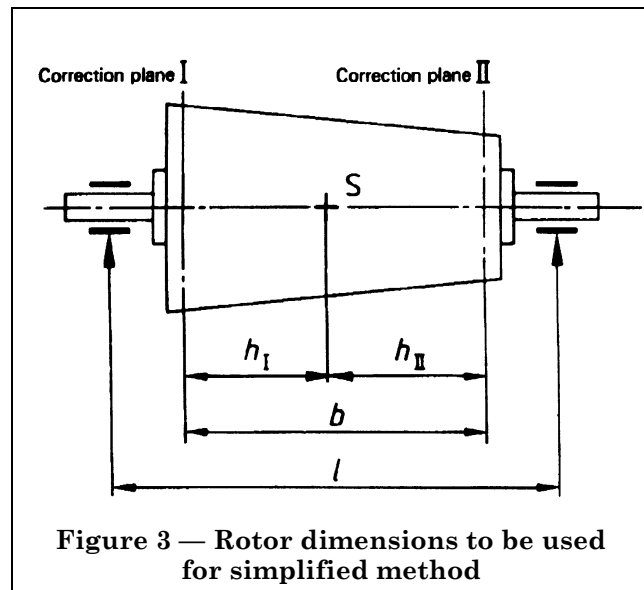


Figure 3 — Rotor dimensions to be used for simplified method

7.3.2.2 Distance between correction planes greater than the bearing span (see Figure 4)

For rotors in which the distance between the correction planes, b , is greater than the bearing span, l , the effect of couple unbalance becomes increasingly more important. In this case, reduce the permissible residual unbalance value for the rotor, U_{per}^* , in the ratio l/b as follows:

$$U_{\text{per}}^* = U_{\text{per}} \times \frac{l}{b}$$

and allocate this value for U_{per}^* to the correction planes in the same way as described in 7.3.2.1.

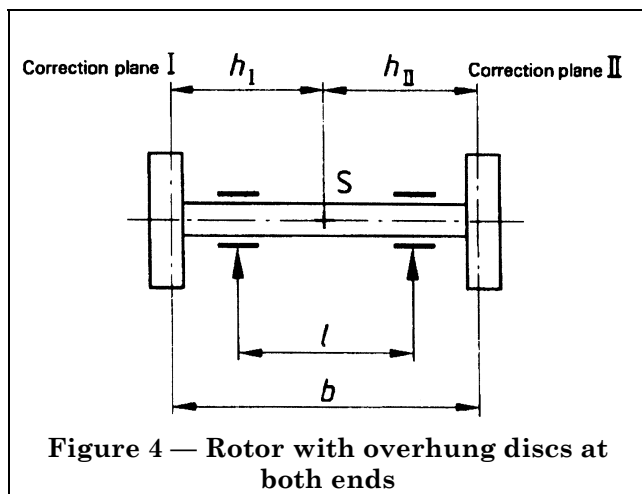


Figure 4 — Rotor with overhung discs at both ends

7.3.2.3 Distance between correction planes smaller than one-third of the bearing span (see Figure 5)

For rotors in which the distance between the correction planes is small, the effect of the residual unbalance in each correction plane is sensitive to the phase angle between the unbalance in each plane. Consequently, the use of the method described in 7.3.2.1 may lead to a finer state of balance than is necessary. It is therefore recommended that for these rotors, U_{per} be allocated to correction planes in terms of static and couple residual unbalance as described below.

Select an arbitrary plane III for correction of static (or quasistatic) unbalance. (Plane III may be coincident with either plane I or plane II.) Let c be the distance from plane III to the more distant bearing, as shown in Figure 5.

Make the following adjustment to U_{per} for allocation to the correction planes:

$$U_{\text{perIII}} = \frac{U_{\text{per}}}{2} \times \frac{l}{2c}$$

$$U_{\text{perI}} = U_{\text{perII}} = \frac{U_{\text{per}}}{2} \times \frac{3l}{4b}$$

for couple corrections made in planes I and II, 180° apart.

The allocation of U_{per} specified above

- assumes equal permissible dynamic bearing loads,
- takes into account the most unfavourable phase relationship between residual static and couple unbalance,
- applies for any position of the static correction plane, and
- is valid for $0 < b < \frac{l}{3}$.

²⁾ See national foreword for details of textual errors.

7.3.3 General methods

The methods described in 7.3.3.1 and 7.3.3.2 may be applied to all rotors regardless of their mass distribution and the position of the correction planes.

7.3.3.1 General method for all rotors (independent of phase angle)

Let U_{perI} and U_{perII} be the permissible residual unbalances in correction planes I and II, respectively, to be determined in accordance with the following procedure.

Choose one bearing as the reference; measure all distances positive from this reference bearing to the other bearing.

Determine the following parameters:

- the distance between the bearings: l
- the distance from the reference bearing to correction plane I: a
- the distance between the correction planes: $h^2)$

See Figure 6.

Define the ratio, k , of the permissible residual unbalance at the reference bearing to the total permissible residual unbalance, U_{per} (see note 1).

The permissible residual unbalance at the other bearing will be $(1 - k) U_{\text{per}}$ so that the sum of the permissible residual unbalances at the bearings is equal to U_{per} .

Define the ratio, R , of the permissible residual unbalance in correction plane II to that in correction plane I, i.e. $R = U_{\text{perII}}/U_{\text{perI}}$ (see note 2).

Calculate four values of U_{perI} from the following equations:

$$U_{\text{perI}} = U_{\text{per}} \frac{kl}{[(l-a) + R(l-a-b)]} \quad \dots (1)$$

$$U_{\text{perI}} = U_{\text{per}} \frac{kl}{[(l-a) - R(l-a-b)]} \quad \dots (2)$$

$$U_{\text{perI}} = U_{\text{per}} \frac{(1-k)l}{[a + R(a+b)]} \quad \dots (3)$$

$$U_{\text{perI}} = U_{\text{per}} \frac{(1-k)l}{[a - R(a+b)]} \quad \dots (4)$$

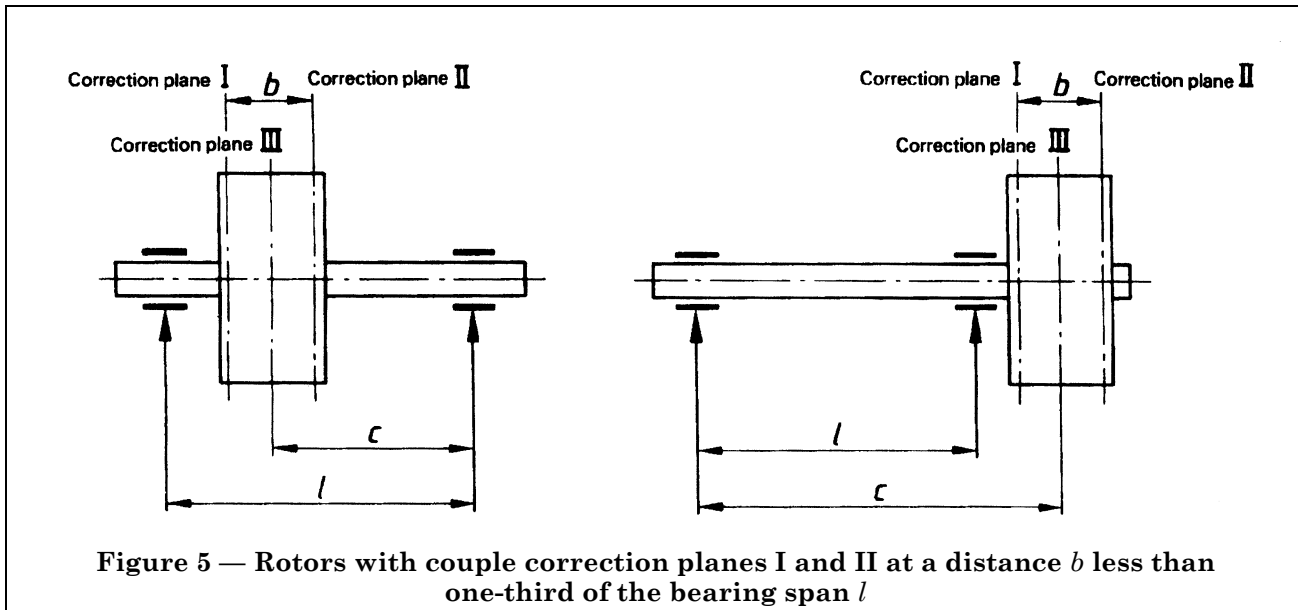


Figure 5 — Rotors with couple correction planes I and II at a distance b less than one-third of the bearing span l

Select the smallest absolute value obtained from equations (1) to (4) as the permissible residual unbalance in correction plane I, U_{perI} . Calculate the value of the corresponding permissible residual unbalance in correction plane II, U_{perII} , from the following equation:

$$U_{\text{perII}} = R \times U_{\text{perI}}$$

The rotor will fulfil the balance quality requirements if the residual unbalance does not exceed U_{perI} in correction plane I and U_{perII} in correction plane II.

NOTE 1 The value of k will be dependent on various design and operational considerations, but for most applications it will be 0,5. However, there are cases where it is desirable or necessary to accept a different residual unbalance referred to one bearing than to the other (such as, for example, if load capacities or stiffness of the service bearings are different) and, in these cases, its value may be permitted to vary between 0,3 and 0,7.

NOTE 2 For most practical applications, the ratio R would be chosen as unity. However, in some cases a different value of R would be more appropriate, such as, for example, if the expected unbalance is significantly different for each correction plane. The residual unbalance at the planes of the bearings are independent of the value of R . Values of R outside the range 0,5 to 2,0 may be impracticable.

See the annex for an example of the application of this method.

7.3.3.2 General method for rotors where the distance between the correction planes is significantly smaller than the bearing span

This method is particularly useful where there is a significant difference between the permissible residual unbalances, depending upon whether the unbalances in the two correction planes are in phase or 180° out of phase. Rotors in which the axial distance between the correction planes is significantly smaller than the bearing span fall into this category.

This method is particularly useful for overhung rotors in which both correction planes are located on the same overhang.

The basis of the method is that the state of unbalance is measured and judged at the service bearing planes so that the general rule outlined in 7.1 applies directly, i.e. U_{per} should be allocated to the bearing planes in such a way that the ratio of the residual unbalances referred to the bearing planes is in the same proportion as the permissible dynamic loads for the service bearings. If it is not possible to measure at the service bearing planes, planes should be chosen which are as close as possible to the service bearings.

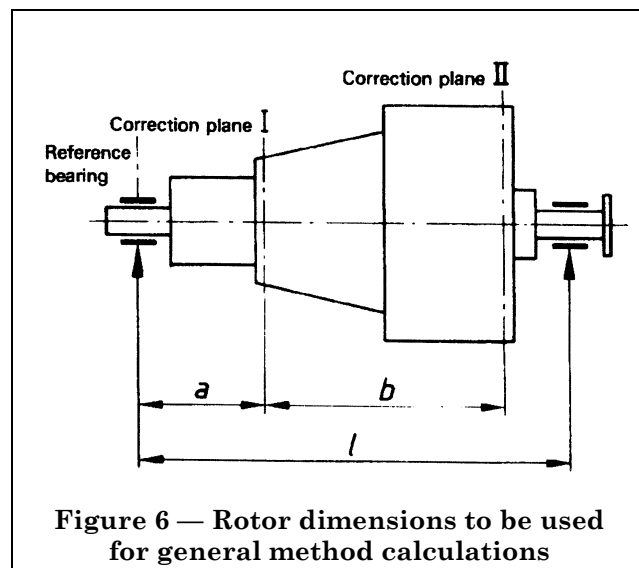


Figure 6 — Rotor dimensions to be used for general method calculations

8 Determination of the residual unbalances

The errors summarized in clause 9 should be taken into account in the determination of the residual unbalances.

8.1 Determination with a balancing machine

When the determination is carried out using a balancing machine, the residual unbalances are determined directly. The accuracy of the machine calibration and the minimum achievable residual unbalance should be sufficient for the task (see ISO 2953).

8.2 Determination by means of an amplitude- and phase-measuring device

The determination can be carried out by means of an amplitude- and phase-measuring device of sufficient sensitivity and accuracy (e.g. as defined in ISO 2371). Place a trial mass of about five times the permissible residual unbalance in one plane and determine the response to it on the measuring plane (or on more measuring planes respectively) (represented for one plane in Figure 7 by point 1). If two-plane balancing is necessary, repeat the same procedure for the other correction plane. From these measurements, the residual unbalance can be evaluated using the influence coefficient method or any other equivalent procedure.

If the measurement accuracy, especially the linearity of the system is in doubt, it is recommended that another set of readings is taken with the same trial mass(es) but displaced by 180° (shown in Figure 7 on point 2).

The initial reading before applying the trial masses is designated as point R (◀) and the mid-point of the line connecting points 1 and 2 is designated as point M (+). If the distance between the points M and R represents an unbalance smaller than the permissible residual unbalance, the system may be considered sufficiently linear. Otherwise, alternative procedures have to be applied in accordance with methods for determining unbalance in two correction planes of a rigid rotor (a future International Standard on this subject is currently in preparation).

8.3 Determination by measurements taken with a measuring device without phase indication

If no adequate balancing machine for the reliable determination of residual unbalances in accordance with 8.1 or no measuring device of sufficient accuracy and sensitivity in accordance with 8.2 is available, this method should be used.

This method is mainly used to determine the residual unbalance after single-plane balancing. It only requires the use of a device for measuring the magnitude of the response to the combined residual unbalance and known trial masses. Attach a trial mass M equivalent to five to ten times the suspected residual unbalance U_r to the correction plane in different angular positions in random sequence. Three different positions are theoretically sufficient, but to smooth out the scatter of individual measurements, it is advantageous to choose 8 or 12 equally spaced angular positions. Then plot the indicated values V (taken at one bearing or a point which responds well to unbalance variation in the correction plane) at the respective angular positions of the trial mass and a curve is drawn through them (see Figure 8 as an example with 8 positions). The curve should be approximately sinusoidal. The arithmetic mean of the scale readings yields the horizontal line in Figure 8, which may be used as the measure V_e of the trial-mass unbalance, while the amplitude of the curve is the measure V_r of the actual residual unbalance amount.

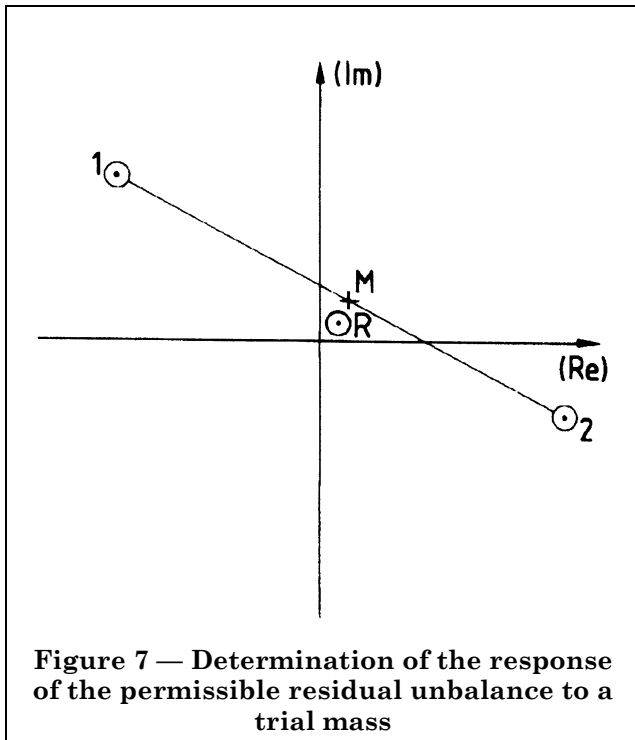
The amount of the residual unbalance is then given by the equation

$$|U_r| = \frac{V_r}{V_e} |M|$$

and its angular position in the correction plane is given by the angle w in the graph in Figure 8.

If no sinusoidal curve or a straight line is obtained, it may be assumed that the existing residual unbalance is already below the limit of reproducibility, or that too small a trial mass has been chosen, or that the measuring sensitivity is inadequate.

The simple method, described for single-plane balancing, yields a reliable result only if measurements are taken where they are not affected by a residual couple unbalance in the rotor. In all other cases, the reliable determination of residual unbalances requires the measurement of responses on two bearings with one trial mass attached in different angular positions in two correction planes. A procedure for determining the residual unbalances in two planes will be described in a future International Standard.



9 Sources of errors in balancing

9.1 Instrument “read-out” errors

In the balancing process carried out by the manufacturer and during the balance check carried out on delivery (i.e. by the client), account shall be taken of possible errors arising from inaccuracies inherent in the measuring methods and equipment. In the first instance, it is necessary to ensure that the residual unbalance is less than the appropriate permissible residual unbalance, while in the second instance, a higher value may be allowed. The magnitude of the permissible deviations from the selected permissible residual unbalance values, U_{per} , will depend on the accuracy of the test equipment.

Typical examples of recommended deviations are given in Table 2.

Table 2 — Typical examples of recommended deviations

Balance quality grades	Reduction of U_{per} recommended to the manufacturer	Increase of U_{per} recommended for the check by the client
G2,5 to G16	10 %	15 %
G1	20 %	25 %
G0,4	25 %	35 %

Methods for the determination of errors will be dealt with in a future International Standard.

9.2 Errors due to the drive and auxiliary equipment

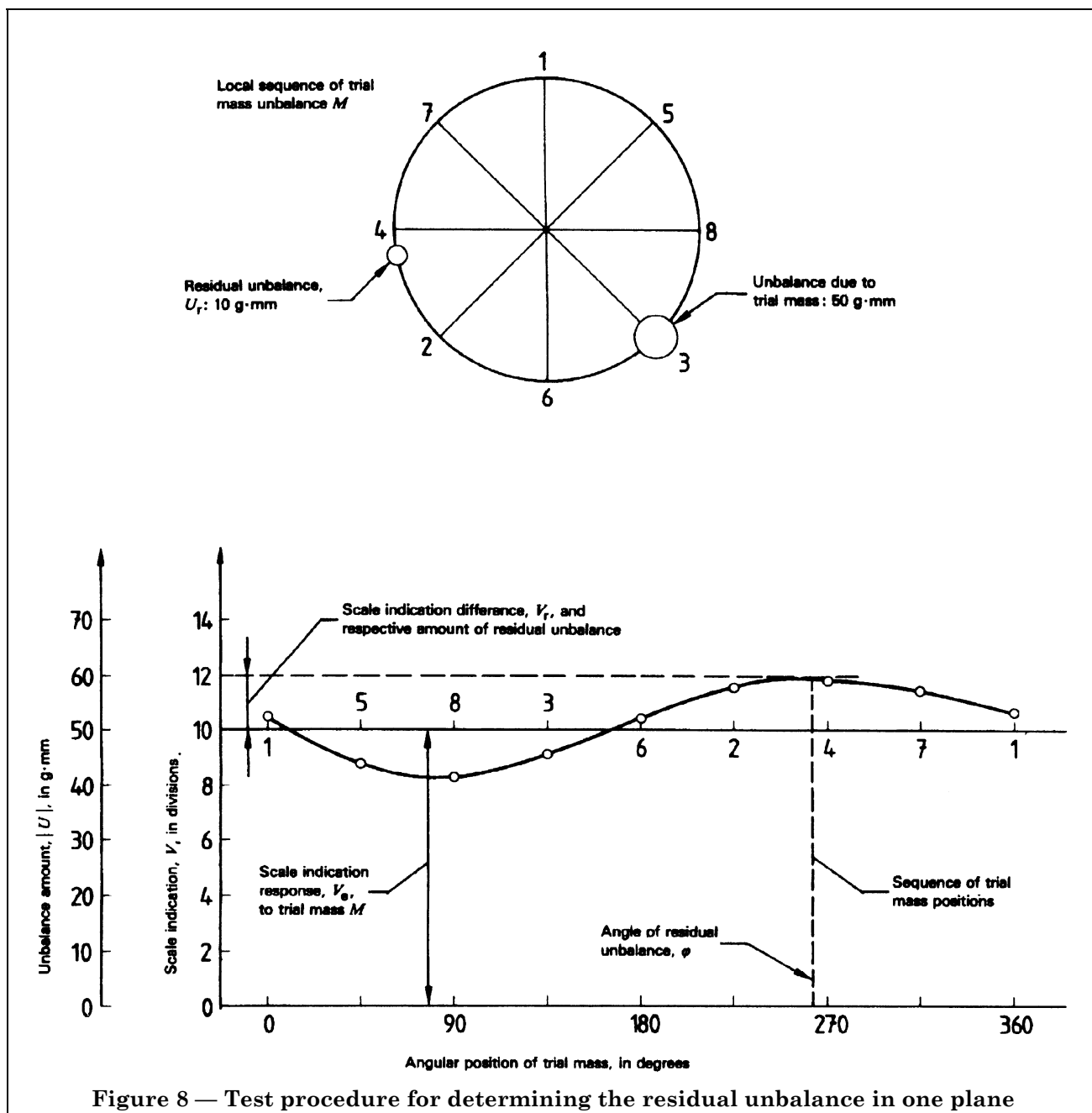
In the balancing process in general, and in the check on residual unbalance in particular, it has to be borne in mind that serious errors can occur due to the fact that driving elements are coupled to the rotor, or due to devices used to support rotors without their own bearings.

Examples of errors which may occur are given in the following list and some of these are illustrated in Figure 9:

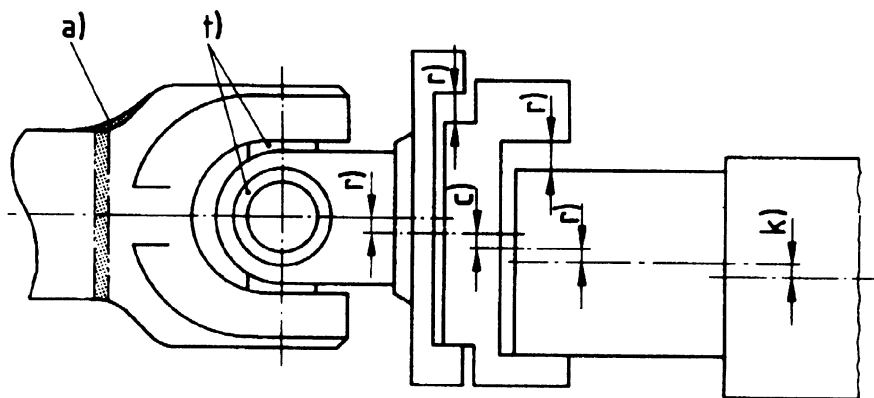
- a) inherent unbalance in drive shaft;
- b) inherent unbalance in mandrel;
- c) eccentricity and axial run-out in drive element;
- d) eccentricity and axial run-out in mandrel;
- e) eccentricity of slave rolling element bearings;
- f) misalignment of bearings;
- g) lack of concentricity of journal and bearing surfaces;
- h) eccentricity and non-squareness of inner runs of rolling element service bearings fitted after the balancing procedure;
- i) rotors with more than two bearings;
- j) keys and keyways;
- k) axial and radial run-out of drive attachment interface;
- l) errors due to stripping and reassembling;
- m) loose parts;
- n) presence of entrapped liquids or solids;
- o) thermal and gravitational effects;
- p) windage effects;
- q) magnetic effects;
- r) clearance in fits;
- s) non-squareness of rolling element bearings;
- t) excessive clearance or tightness in universal joints.

The effect of errors on the determination of residual unbalance will be dealt with in the future part 2 of this International Standard.

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NOTE — Letters refer to list, given in 9.2, of examples of errors which may occur.

Figure 9 — Error sources in end-drive elements (see 9.2)

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Annex Example for the general method described in 7.3.3.1 for two-plane balancing

(This annex does not form an integral part of the standard.)

Turbine rotor

Rotor mass: $m = 3\,600$ kg

Service speed: $n = 4\,950$ r/min

Balance quality grade required: **G2,5**

Permissible specific unbalance, e_{per} :

$$e_{\text{per}} = 2,5 \left[\frac{30}{\pi \times 4\,950} \times 10^3 \right]$$

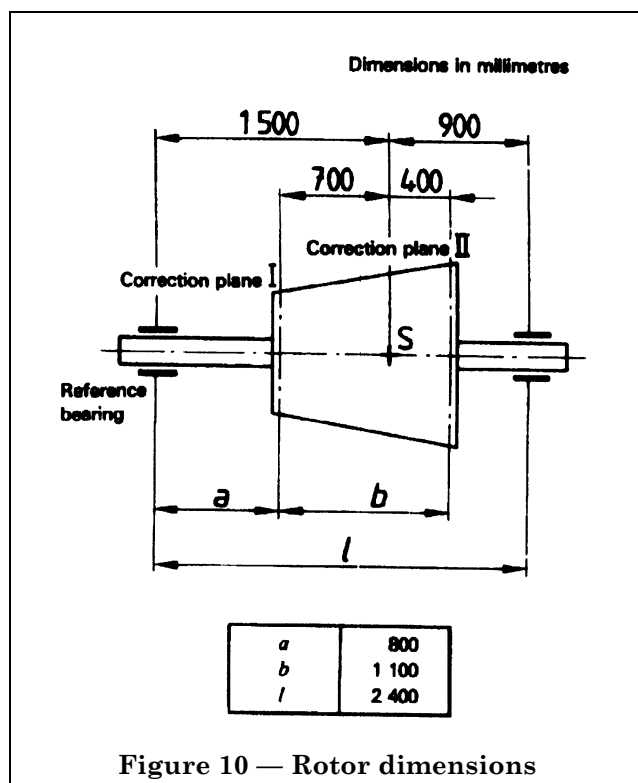
$$= 4,8 \text{ g.mm/Kg}$$

Permissible unbalance, U_{per} :

$$U_{\text{per}} = m \times e_{\text{per}}$$

$$= 3\,600 \times 4,8$$

$$= 17,3 \times 10^3 \text{ g mm}$$



Case 1

$k = 0,5$ (factor of permissible residual unbalance at the reference bearing)

$R = 1$ (ratio of the permissible residual unbalance in the two correction planes II and I)

Therefore using

$$\text{equation (1): } U_{\text{perI}} = 9,9 \times 10^3 \text{ g mm}$$

$$\text{equation (2): } U_{\text{perI}} = 18,9 \times 10^3 \text{ g mm}$$

$$\text{equation (3): } U_{\text{perI}} = 7,7 \times 10^3 \text{ g mm}$$

$$\text{equation (4): } U_{\text{perI}} = 18,9 \times 10^3 \text{ g mm}$$

Smallest absolute value of U_{perI} : $U_{\text{perI}} = 7,7 \times 10^3 \text{ g mm}$

$$U_{\text{perII}} = R \times U_{\text{perI}} = 1 \times U_{\text{perI}}: U_{\text{perII}} = 7,7 \times 10^3 \text{ g mm}$$

Possible maximum residual unbalance: $U_{\text{perI}} + U_{\text{perII}} = 15,4 \times 10^3 \text{ g mm} < U_{\text{per}}$

Case 2 (using different assumptions for the parameters k and R)

$$k = \frac{900}{2\,400} = 0,38 \left(\frac{\text{static load on reference bearing}}{\text{total static load or rotor mass}} \right)$$

$$R = \frac{700}{400} = 1,75 \left(\frac{\text{distance of correction plane I from centre of gravity}}{\text{distance of correction plane II from centre of gravity}} \right)$$

Therefore using

$$\text{equation (1): } U_{\text{perI}} = 6,3 \times 10^3 \text{ g mm}$$

$$\text{equation (2): } U_{\text{perI}} = 21,8 \times 10^3 \text{ g mm}$$

$$\text{equation (3): } U_{\text{perI}} = 6,3 \times 10^3 \text{ g mm}$$

$$\text{equation (4): } U_{\text{perI}} = -10,2 \times 10^3 \text{ g mm}$$

Smallest absolute value of U_{perI} : $U_{\text{perI}} = 6,3 \times 10^3 \text{ g mm}$

$$U_{\text{perII}} = R \times U_{\text{perI}} = 1,75 \times U_{\text{perI}}: U_{\text{perII}} = 11 \times 10^3 \text{ g mm}$$

Possible maximum residual unbalance: $U_{\text{perI}} + U_{\text{perII}} = 17,3 \times 10^3 \text{ g mm} < U_{\text{per}}$

Publications referred to

See national foreword.



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