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**International Standard****1940 / 1**

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

*Vibrations mécaniques — Exigences en matière de qualité dans l'équilibrage des rotors rigides — Partie 1: Détermination du balourd résiduel admissible*

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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.

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. They are approved in accordance with ISO procedures requiring at least 75 % approval by the member bodies voting.

International Standard ISO 1940/1 was prepared by Technical Committee ISO/TC 108, *Mechanical vibration and shock*.

This part of ISO 1940 cancels and replaces, in part, ISO 1940-1973, of which it constitutes a minor revision.

Users should note that all International Standards undergo revision from time to time and that any reference made herein to any other International Standard implies its latest edition, unless otherwise stated.

# Mechanical vibration — Balance quality requirements of rigid rotors —

## Part 1: Determination of permissible residual unbalance

### 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 figures 1a) to 1f). For most rotors, unbalances are measured in two planes as illustrated in figures 1a) to 1c). 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·mm would be applied in planes I and II respectively, at the appropriate angular orientation around the rotor. Comparison of figures 1a) and 1c) 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.

Alternatively, a representation similar to figure 1d) 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 1e) may be useful, where the point S denotes the mass centre of the rotor.

The representation in figure 1f), 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.)

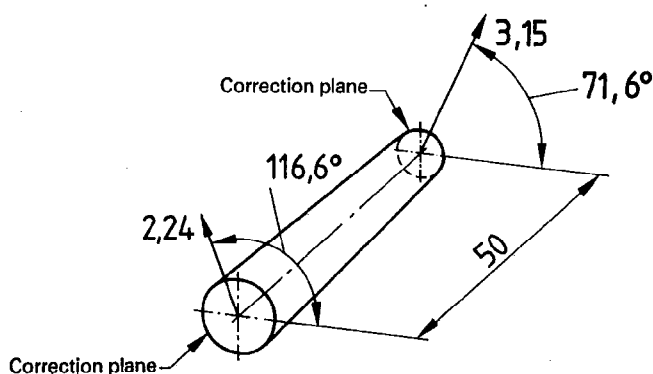
### 4 Permissible unbalance related to rotor mass

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,  $U_{\text{per}}$ , to the rotor mass,  $m$ , in terms of permissible residual specific unbalance value,  $e_{\text{per}}$ , as given by the following formula:

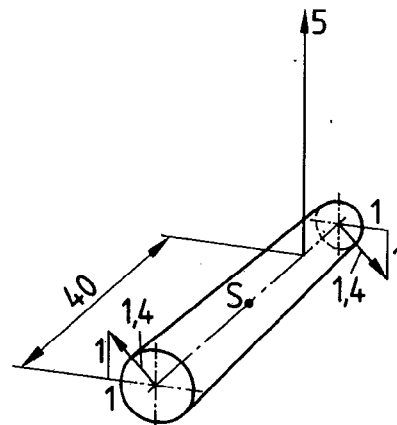
$$e_{\text{per}} = \frac{U_{\text{per}}}{m}$$

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

Linear dimensions in millimetres  
Vectorial quantities in gram millimetres

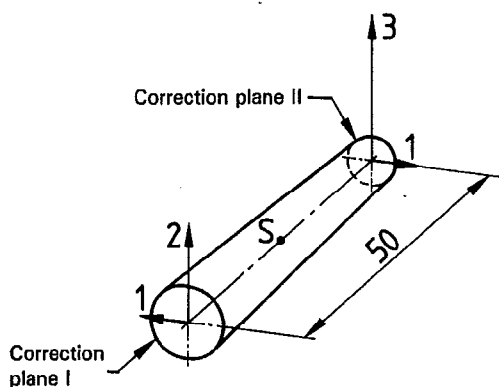


a) An unbalance vector in each of the correction planes I and II

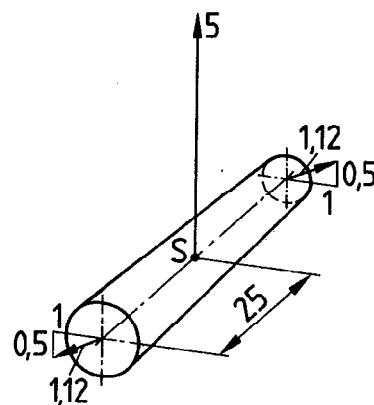


d) A resultant unbalance vector together with an associated couple unbalance referred to the two correction planes I and II

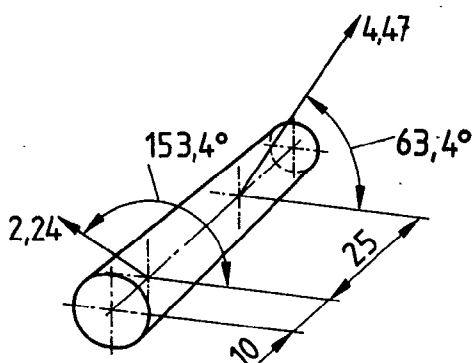
The resultant unbalance vector may be located anywhere, for example in one of the correction planes; the value of the couple is dependent on the position of the resultant unbalance vector.



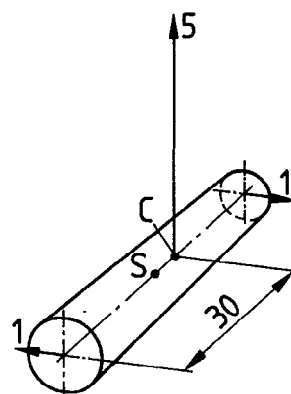
b) Two unbalance components in each of the correction planes I and II



e) Special case of d), namely static/couple unbalance  
The resultant unbalance vector passes through the mass centre of the rotor; there is an associated couple unbalance.



c) An unbalance vector in each of two other correction planes



f) Another special case of d)  
The resultant unbalance vector passes through the centre of unbalance and the associated couple unbalance is a minimum.

Figure 1 — Different representations of the same state of unbalance of a rigid rotor

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such that the couple unbalance is zero, the permissible residual specific unbalance value,  $e_{\text{per}}$ , 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,  $e_{\text{per}}$ .

## 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,  $e_{\text{per}}$ , 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:

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

where  $\omega$  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 ( $e_{\text{per}} \times \omega$ ), expressed in millimetres per second; balance quality grades are designated according to the product of the relationship, i.e. if the product of  $e_{\text{per}} \times \omega$  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  $e_{\text{per}}$  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

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

Balance quality grade	Product of the relationship ( $e_{\text{per}} \times \omega$ ) <sup>1), 2)</sup> mm/s	Rotor types — General examples
<b>G4 000</b>	4 000	Crankshaft/drives <sup>3)</sup> of rigidly mounted slow marine diesel engines with uneven number of cylinders <sup>4)</sup>
<b>G1 600</b>	1 600	Crankshaft/drives of rigidly mounted large two-cycle engines
<b>G630</b>	630	Crankshaft/drives of rigidly mounted large four-cycle engines Crankshaft/drives of elastically mounted marine diesel engines
<b>G250</b>	250	Crankshaft/drives of rigidly mounted fast four-cylinder diesel engines <sup>4)</sup>
<b>G100</b>	100	Crankshaft/drives of fast diesel engines with six or more cylinders <sup>4)</sup> Complete engines (gasoline or diesel) for cars, trucks and locomotives <sup>5)</sup>
<b>G40</b>	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 <sup>4)</sup> Crankshaft/drives of engines of cars, trucks and locomotives
<b>G16</b>	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
<b>G6,3</b>	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
<b>G2,5</b>	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 <b>G6,3</b> Turbine-driven pumps
<b>G1</b>	1	Tape recorder and phonograph (gramophone) drives Grinding-machine drives Small electric armatures with special requirements
<b>G0,4</b>	0,4	Spindles, discs, and armatures of precision grinders Gyroscopes

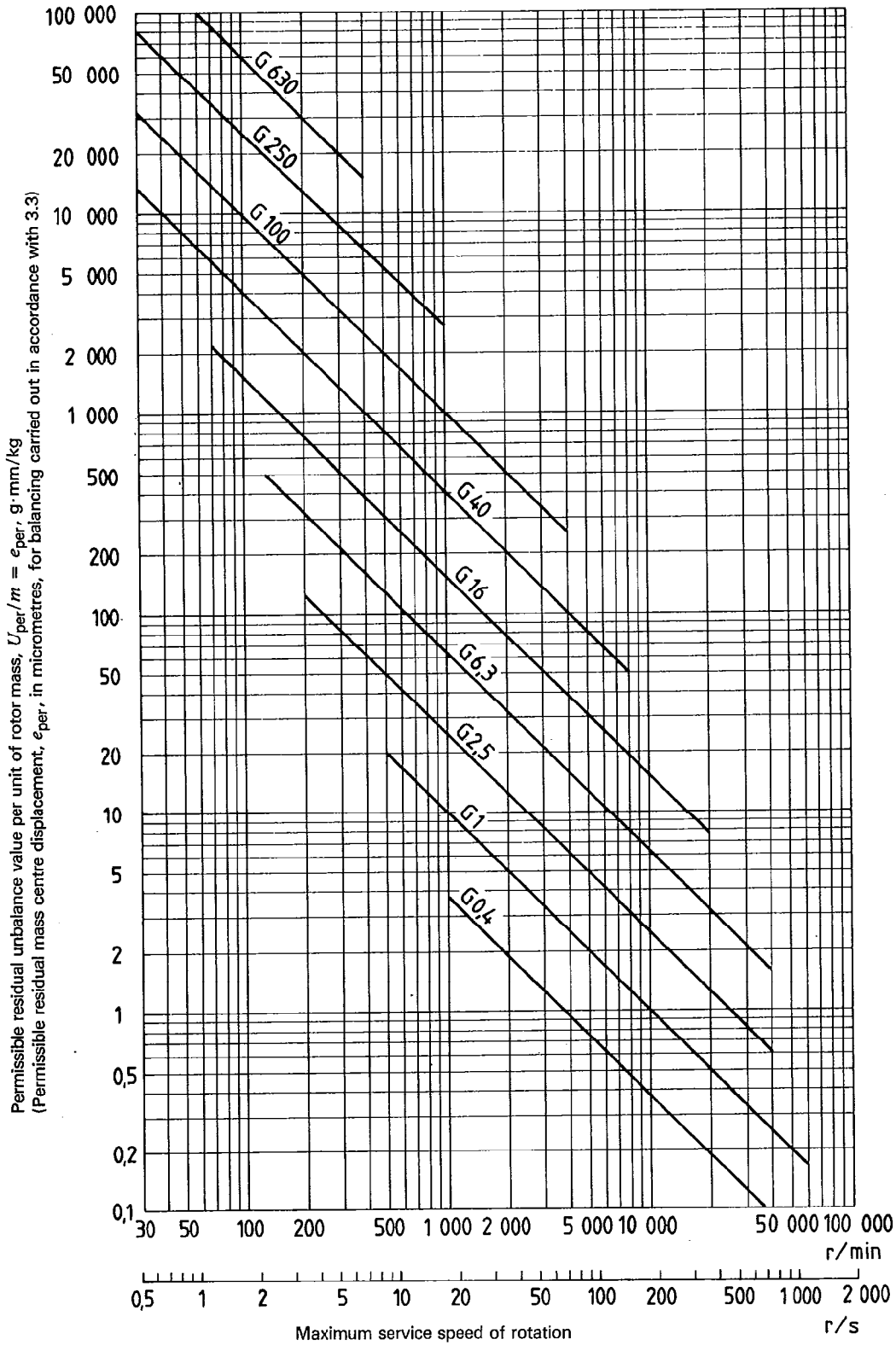
1)  $\omega = 2\pi n/60 \approx n/10$ , if  $n$  is measured in revolutions per minute and  $\omega$  in radians per second.

2) For allocating the permissible residual unbalance to correction planes, see clause 7.

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

4) 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.

5) 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



calculated, using the methods described in clause 7, by defining  $U_{\text{per}}$  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 $U_{\text{per}}$

### 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,  $U_{\text{per}}$ , and therefore implies allocation to each correction plane if more than one correction plane is used.

As a general rule,  $U_{\text{per}}$  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  $U_{\text{per}}$ .

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  $U_{\text{per}}$ .

### 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.

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  $180^\circ$  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:

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

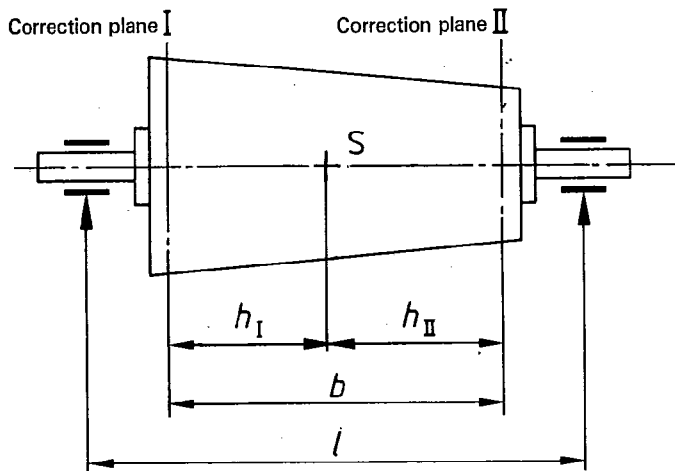


Figure 3 – Rotor dimensions to be used for simplified method

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

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

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.

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

$$0,3 U_{per} < U_{perII} = U_{per} \frac{h_I}{b} < 0,7 U_{per}$$

**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,

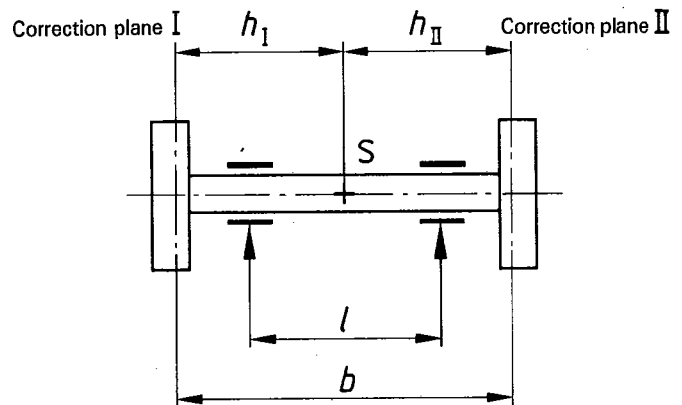


Figure 4 – Rotor with overhung discs at both ends

reduce the permissible residual unbalance value for the rotor,  $U_{per}^*$ , in the ratio  $l/b$  as follows:

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

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

**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 quasi-static) 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:

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

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

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

The allocation of  $U_{per}$  specified above

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

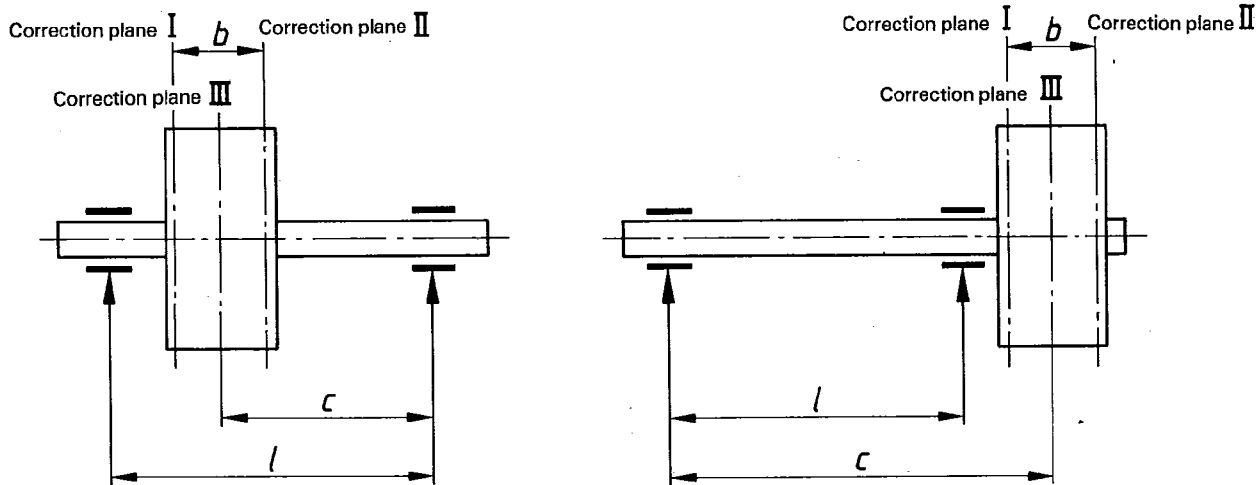


Figure 5 — Rotors with couple correction planes I and II at a distance  $b$  less than one-third of the bearing span  $l$

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:  $h$

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  $(1 - k) U_{per}$  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.  $R = U_{perII} / U_{perI}$  (see note 2).

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

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

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

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

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

Select the smallest absolute value obtained from equations (1) to (4) as the permissible residual unbalance in correction

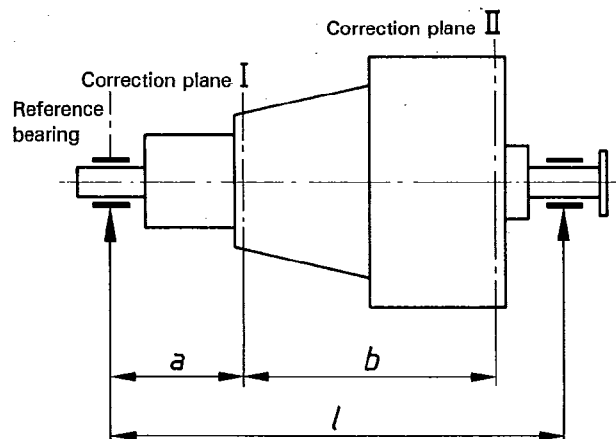


Figure 6 — Rotor dimensions to be used for general method calculations

## ISO 1940/1-1986 (E)

plane I,  $U_{perI}$ . Calculate the value of the corresponding permissible residual unbalance in correction plane II,  $U_{perII}$ , from the following equation:

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

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.

## NOTES

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.

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.

## 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.

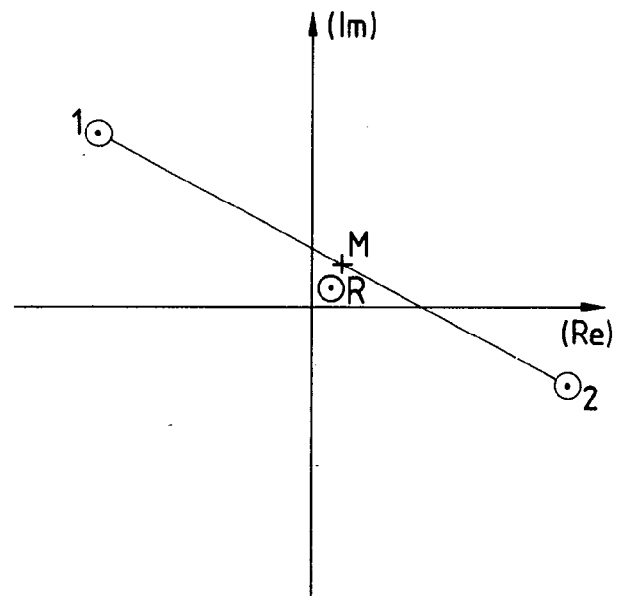


Figure 7 — Determination of the response of the permissible residual unbalance to a trial mass

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  $\varphi$  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.

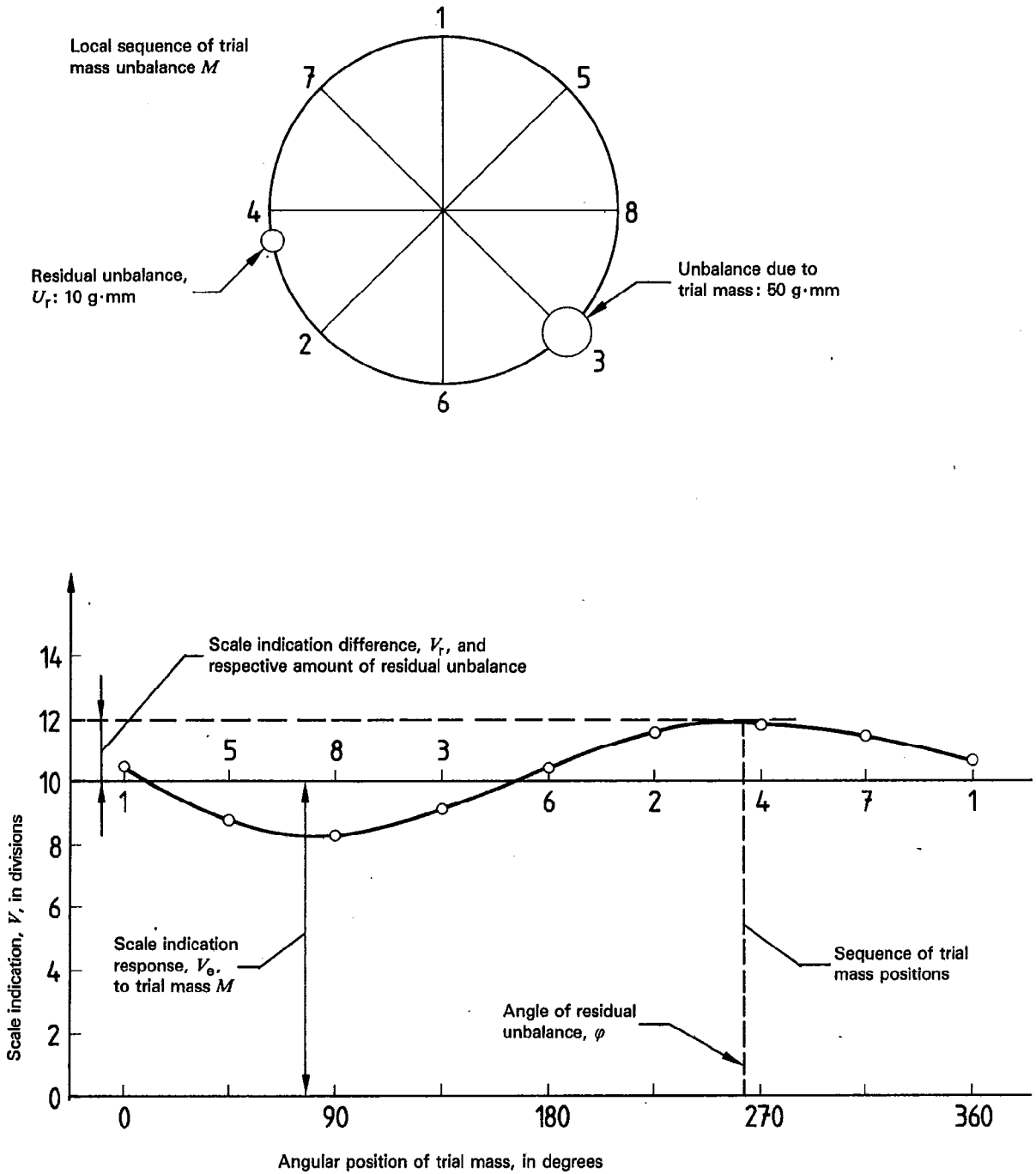
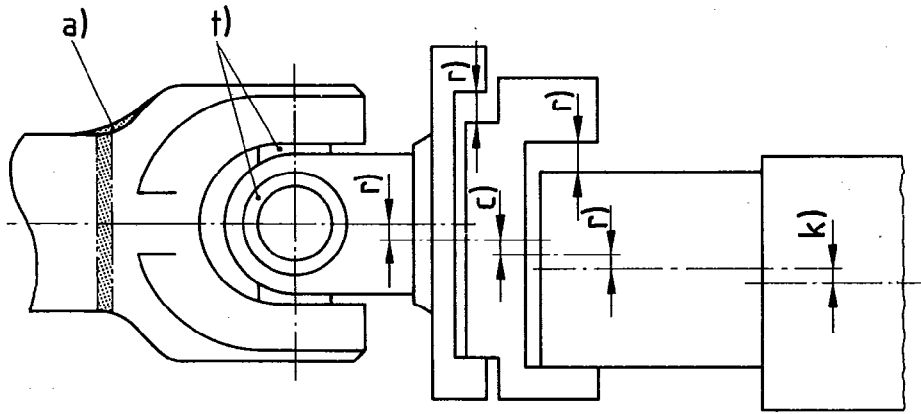


Figure 8 — Test procedure for determining the residual unbalance in one plane



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)

### 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:  $m = 3\,600\text{ kg}$

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

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

Permissible specific unbalance,  $e_{\text{per}}$ :

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

$$= 4,8\text{ g}\cdot\text{mm/kg}$$

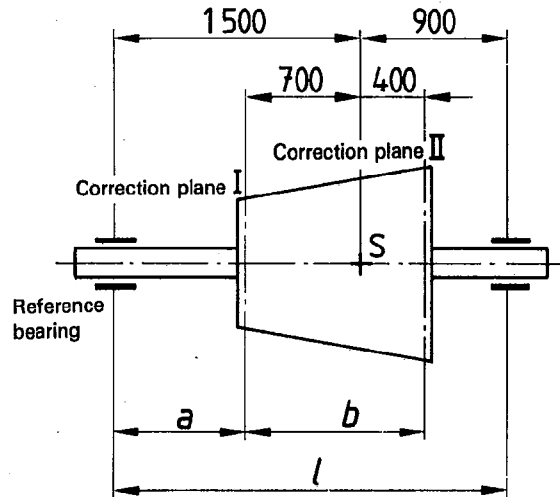
Permissible unbalance,  $U_{\text{per}}$ :

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

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

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

Dimensions in millimetres



$a$	800
$b$	1 100
$l$	2 400

Figure 10 — Rotor dimensions

#### 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

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

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

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

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

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

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

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

#### 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

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

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

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

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

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

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

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