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Balance quality of rotating rigid bodies

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FOREWORD

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It has been approved by the Member Bodies of the following countries :

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Balance quality of rotating rigid bodies

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. These recommendations are concerned with the permissible residual unbalance.

By means of the measuring equipment available today, unbalance may now be reduced to rather low limits. However, it would be uneconomical to exaggerate the quality requirements. To what extent the unbalance must be reduced, and where the optimal economic and technical compromise on balance quality has to be struck, can, in individual cases, be correctly determined only by extensive measurement in the laboratory or in the field.

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 vibrations is influenced by many factors, such as the vibrating mass of the machine casing and its foundation, the bearing and the foundation stiffness, the proximity of the operating speed to the various resonance frequencies, etc. Moreover, the effect of the unbalances varies with their mutual angular position (see 3.2), and finally the machine vibrations may be due only in part to the presence of rotor unbalance.

1 SCOPE

This International Standard makes recommendations concerning the balance quality of rotating rigid bodies particularly as it relates to the permissible residual unbalance as a function of the maximum service speed.

It includes a tentative classification of various types of representative rotors in which the rotor groups are associated with ranges of recommended balance quality grades.

2 LIMITS OF APPLICABILITY

According to the table, page 5, various balance quality grades are assigned to different groups of rotors. Hence, it is possible by means of Figure 4 to determine the specific permissible residual unbalance of each rotor group as a function of the maximum service speed. These recommended balance quality grades are based on experience which was gained with rotors of various types, sizes, and service speeds. They are valid for rigid rotors, i.e. rigid from the point of view of balancing[1]. This "rigidity" is required not only at the rotational speed in the balancing machine but also throughout the whole operating range of the rotor under operational conditions. Later, similar recommendations will be proposed for "flexible" rotors, in which bending deflections occur as a function of rotational speed.

The recommendations 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; on the other hand, they may serve as a basis for more involved investigations, for example, when in special cases a more exact determination of the required balance quality 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.

The balance quality grades G are intended to afford a classification of balance quality in order to facilitate mutual understanding between the interested parties.

3 PERTINENT ASPECTS OF THE BALANCING PROBLEM

3.1 Representation of a state of unbalance

One and the same state of unbalance can be represented in various ways, as shown in Figures 1 a) to 1 g). In general, measurements yield analogue values corresponding to the state of unbalance illustrated in Figure 1 a) or 1 b). The unbalance correction process also takes

1) For definitions, see ISO 1925, *Balancing terminology*. (At present at the stage of draft.)

place in this way, except that in special cases a correction procedure corresponding to Figure 1 c) may be applied. If an investigation of the unbalance effect on the vibratory behaviour of a whole machine is intended, then an unbalance resolution according to Figure 1 d) may be useful, where S denotes the mass centre of the rotor or of the whole machine according to the purpose of the investigation. The representations in Figures 1 e) and 1 f) may be practical when transfer of the unbalances to different reference planes is to be carried out. The shortest and most general designation of quasi-static unbalance and couple unbalance is shown in Figure 1 f). The possibility of expressing the unbalance moment as couple unbalance in two arbitrary planes I and IIa is shown in Figure 1 g).

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 stiffness of the bearings and foundation.

In many cases, the static unbalance is of primary importance as compared with the couple unbalance, i.e. 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. Then the bearing load due to a couple unbalance is larger than that caused by a static unbalance, provided the sum of the opposite unbalances in the correction planes representing the couple unbalance exceeds a certain fraction of the static unbalance assumed to be located in the middle between the bearings. Denoting the bearing distance by l and that of the correction planes by a, then, if the permissible residual static unbalance is U_R, the permissible residual unbalances U_C, forming the couple unbalance, are reduced to $U_C = U_R \, l/2a$.

3.3 Rotors with one correction plane

For disk-shaped rotors, the use of only one correction plane may be sufficient, provided the bearing distance is sufficiently large and the disk rotates with sufficiently small axial run-out. Whether these conditions are fulfilled must be investigated in each individual case. After single-plane balancing has been carried out on a sufficient number of rotors of the particular type, the greatest residual unbalance moment is determined and divided by the bearing distance. If the unbalances found in this way are acceptable even in the worst case, i.e. if they are not larger than half the recommended value multiplied by the rotor mass, then it can be expected that single-plane balancing is sufficient.

3.4 Rotors with two correction planes

If the rotor does not satisfy the conditions stated above in 3.3 for the disk-shaped rotor, then 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 that the rotor rotate, since otherwise the residual couple unbalance (see 3.2) would remain undetected.

In the case of rotors for which the centre of gravity is located within the mid-third of the distance between the bearings, one-half the recommended value of the permissible residual unbalance in Figure 4 should be taken for each correction plane if these are equi-distant from the centre of gravity. For other rotors, it may be necessary to apportion the recommended value in accordance with the rotor's mass-distribution, so long as the principal part of the mass is situated between the correction planes. In unusual cases, the distribution of the recommended value must be specially investigated taking into account, say, the permissible bearing loads.

3.5 Assemblies

Rotors may be supplied for balancing as integral single components or as assemblies. For each assembly, the unbalances of the component parts must be added vectorially and any unbalances resulting from inaccuracies of assembly must be taken into account, giving particular attention to the fact that the parts may be assembled later in a different position from that in the balancing machine.

The maximum unbalance due to fit and geometrical tolerances is then the sum of the greatest possible radial displacements in both cases (in the balancing machine and assembled-for-service condition, respectively) multiplied by the mass of the component concerned. Such displacements may arise from both radial clearance and radial run-out as well as from axial run-out.

Therefore, the permissible residual unbalances of the individual components and the limits of fit, as well as the limits for radial and axial run-out, are determined by the condition that the sum of the unbalances due to these causes must not be larger than the recommended value for the rotor-type to which the assembly belongs. Of course, a sensible relationship should be observed between the magnitudes of the residual unbalances of the individual components and of the unbalances due to fit-inaccuracies. If the unbalance tolerance for an assembly can not be achieved by single-part balancing, then the assembled parts must be balanced as a unit.

If the individual components are balanced separately, then the connecting elements, such as bolts and keys, may be attached all to one part or distributed among the components. However, prior agreement should be reached as to the attachment of such elements.

4 ROTOR MASS AND PERMISSIBLE RESIDUAL UNBALANCE

In general, the larger the rotor mass, the greater the permissible unbalance. It is therefore appropriate to relate the permissible residual unbalance U to the rotor mass m. The specific unblance $e = U/m$ is equivalent to the displacement of the centre of gravity where this coincides with the plane of the static unbalance.

5 SERVICE SPEED AND PERMISSIBLE RESIDUAL UNBALANCE

Practical experience shows that for rotors of the same type, in general the permissible specific unbalance $e = U/m$ varies inversely as the speed n of the rotor in the limited range of velocities considered in Figure 4 for the respective balance quality grade. In this connection also, statistical empirical data, for rotors of the same type, point to the following relationship for circular velocity :

$$en = \text{constant}$$

or equivalently

$$e\omega = \text{constant}$$

where e may be taken as the eccentricity of the centre of gravity for the case of a static unbalance. This relationship follows also from practical considerations of mechanical similarity on the basis that, for geometrically similar rotors running at equal peripheral speeds, the stresses in rotors and rigid bearings are the same. The balance quality grades (of the table and Figure 4) are based on this relationship.

6 BALANCE QUALITY

6.1 Balance quality grades

On the basis of sections 4 and 5, balance quality grades have been established which permit a classification of the quality requirements. Each balance quality grade G comprises a range of permissible residual unbalances from an upper limit which is given by a certain magnitude of the product $e\omega$ to zero. Plotted against the maximum operating speed n, the upper limits of e are shown in Figure 4. The main balance quality grades G 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 balance quality grades are designated according to the upper limit of the product $e\omega$ where $\omega = 2\,\pi n/60 \approx n/10$, for n measured in revolutions per minute and ω in radians per second, and the product $e\omega$ is given in millimetres per second.

Example : For a rotor of the balance quality grade G 6,3 a recommended value $e = 20\ \mu$m is found if its maximum service speed is 3 000 rev/min. Hence, for a rotor of 40 kg, symmetrical in the sense of 3.4, the permissible residual unbalance in each of the two correction planes is 400 g·mm.

6.2 Balance quality grades G 1 and G 0,4

These extreme balance quality grades are most sensitive to the progressive development of balancing technology. In these ranges, the final balance quality selected is a compromise between technical requirements and the reality that they may be achieved. 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 (circularity, etc.) in the rotor bearings and/or the bearing accuracy are sufficiently restricted. For balancing at balance quality grade G 1, it may be necessary to balance the rotor in its own *service bearings*, while in order to satisfy balance quality grade G 0,4, balancing should be carried out with the rotor mounted in its own *housing and bearings and under service conditions*, and at the service operating temperature. For the balance quality grade G 1, at least for higher service speeds, a power transmission without universal joints is necessary. In general, for the balance quality grade G 0,4, self-drive is required.

6.3 Experimental determination of the required balance quality

In order to determine the permissible values of residual unbalance experimentally, rotors of the type under consideration are first balanced to the minimum achievable residual unbalance. Subsequently, artificial unbalances (test masses) of increasing magnitude are attached to the rotors (under service conditions) until the effect of the unbalances can be detected above the level of other existing disturbances, i.e. until these unbalances noticeably affect the vibration, the running smoothness, or the functioning of the machine. In two-plane balancing, the differing effects of equi-phased unbalances and of unbalance couples must be considered (see 3.2 and 3.3). If possible, this evaluation should be performed *in situ*. In addition. allowance should be made for changes which occur in service.

7 SOURCES OF ERROR IN BALANCING

7.1 Instrument "read-out" errors

In the balancing process performed by the manufacturer and during the balance check performed on delivery (that is, by the client), account must 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 maximum permissible unbalance, while in the second instance, a higher value may be allowed. The magnitude of the permissible deviations from the selected maximum permissible unbalance values will depend on the quality of the test equipment. The following deviation limits for each of the two cases may be used as examples :

Balance quality grades	Permissible deviation
G 2,5 – G 16	± 15 %
G 1	± 30 %
G 0,4	± 50 %

If the check on the residual unbalance of a balanced rotor is carried out with minimum possible deviation, the procedure outlined below may be followed (see Figure 2) :

A test unbalance mass equivalent to 5 to 10 times the magnitude of the suspected residual unbalance is attached to the rotor in different angular positions. In order to smooth out the scatter of individual measurements, it is advantageous to choose 8 equally spaced angular positions (that is, positions spaced 45° apart). The unbalance read-out values are then plotted at their respective angular positions (see Figure 2) and a curve passed through them; it should be approximate to a sinusoid. The arithmetic mean of the scale readings yields the horizontal line in Figure 2 which may be used as a measure of the test unbalance, while the amplitude of the sinusoidal curve is the measure of the actual residual unbalance. If no sinusoidal curve is obtained, it may be assumed that the existing residual unbalance is already below the limit of reproducibility.

If the linearity of the scale reading is questioned, then the test sequence may be repeated with a test unbalance which has been reduced (or augmented) by the amount of the suspected residual unbalance. The relation between the two sinusoidal curves (that is, the difference in the value recommended at each angular position) then yields a more reliable criterion. The check should be carried out separately for the two correction planes.

7.2 Errors due to the drive

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

In Figure 3, examples of the following sources of error are given :

a) unbalance effects attributable to the driving or supporting elements;

b) errors in concentricity and clearances of the driving or supporting elements;

c) clearance between driving or supporting element and rotor;

d) errors in concentricity of the rotor at the point of attachment relative to the journal.

The effects of the errors under a) and b) may be demonstrated by taking measurements at different angular positions of the couplings, for example, turning the cardan shaft through 180° after the first run.

The error under c) can be determined by two balancing runs in which the clearance is eliminated in two opposite directions.

The error under d), however, cannot be found by means of balancing. Here the only recourse is to extreme accuracy in manufacture or to a test under operating conditions without coupling elements as described above (Figure 2).

8 DATA ON DRAWINGS OR SCHEDULES

In addition to the value of the maximum permissible residual unbalance in each correction plane in gram millimetres (or statement of rotor mass, service speed, and the balance quality grade), design drawings or schedules should also contain precise data as to the type of bearings and their location in the balancing machine, drive arrangements, rotational speed of balancing, correction planes, location where the correction masses may be placed and such information as how much material may be removed safely, taking into account the required strength or other considerations. In some cases, an instruction may be given regarding the state of manufacture and degree of assembly of the rotor when ready for balancing (for example, with or without flywheel, key or the like).

9 BALANCE QUALITY GRADES AND TYPES OF ROTORS

In the table, a tentative classification of various types of rotor is given. Certain ranges of balance quality grades are associated with the various classes.

The types of prime movers, machines and rotors found in the table are examples based on present experience.

TABLE – **Balance quality grades for various groups of representative rigid rotors**

Balance quality grade G	$e\omega$ 1) 2) mm/s	**Rotor types – General examples**
G 4 000	4 000	Crankshaft-drives[3] of rigidly mounted slow marine diesel engines with uneven number of cylinders[4].
G 1 600	1 600	Crankshaft-drives of rigidly mounted large two-cycle engines.
G 630	630	Crankshaft-drives of rigidly mounted large four-cycle engines. Crankshaft-drives of elastically mounted marine diesel engines.
G 250	250	Crankshaft-drives of rigidly mounted fast four-cylinder diesel engines[4].
G 100	100	Crankshaft-drives of fast diesel engines with six or more cylinders[4]. Complete engines (gasoline or diesel) for cars, trucks and locomotives[5].
G 40	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[4]. Crankshaft-drives for engines of cars, trucks and locomotives.
G 16	16	Drive shafts (propeller shafts, cardan shafts) with special requirements. Parts of crushing machinery. 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.
G 6,3	6,3	Parts or process plant machines. Marine main turbine gears (merchant service). Centrifuge drums. Fans. Assembled aircraft gas turbine rotors. Fly wheels. Pump impellers. Machine-tool and general machinery parts. Normal electrical armatures. Individual components of engines under special requirements.
G 2,5	2,5	Gas and steam turbines, including marine main turbines (merchant service). Rigid turbo-generator rotors. Rotors. Turbo-compressors. Machine-tool drives. Medium and large electrical armatures with special requirements. Small electrical armatures. Turbine-driven pumps.
G 1	1	Tape recorder and phonograph (gramophone) drives. Grinding-machine drives. Small electrical armatures with special requirements.
G 0,4	0,4	Spindles, disks, and armatures of precision grinders. Gyroscopes.

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

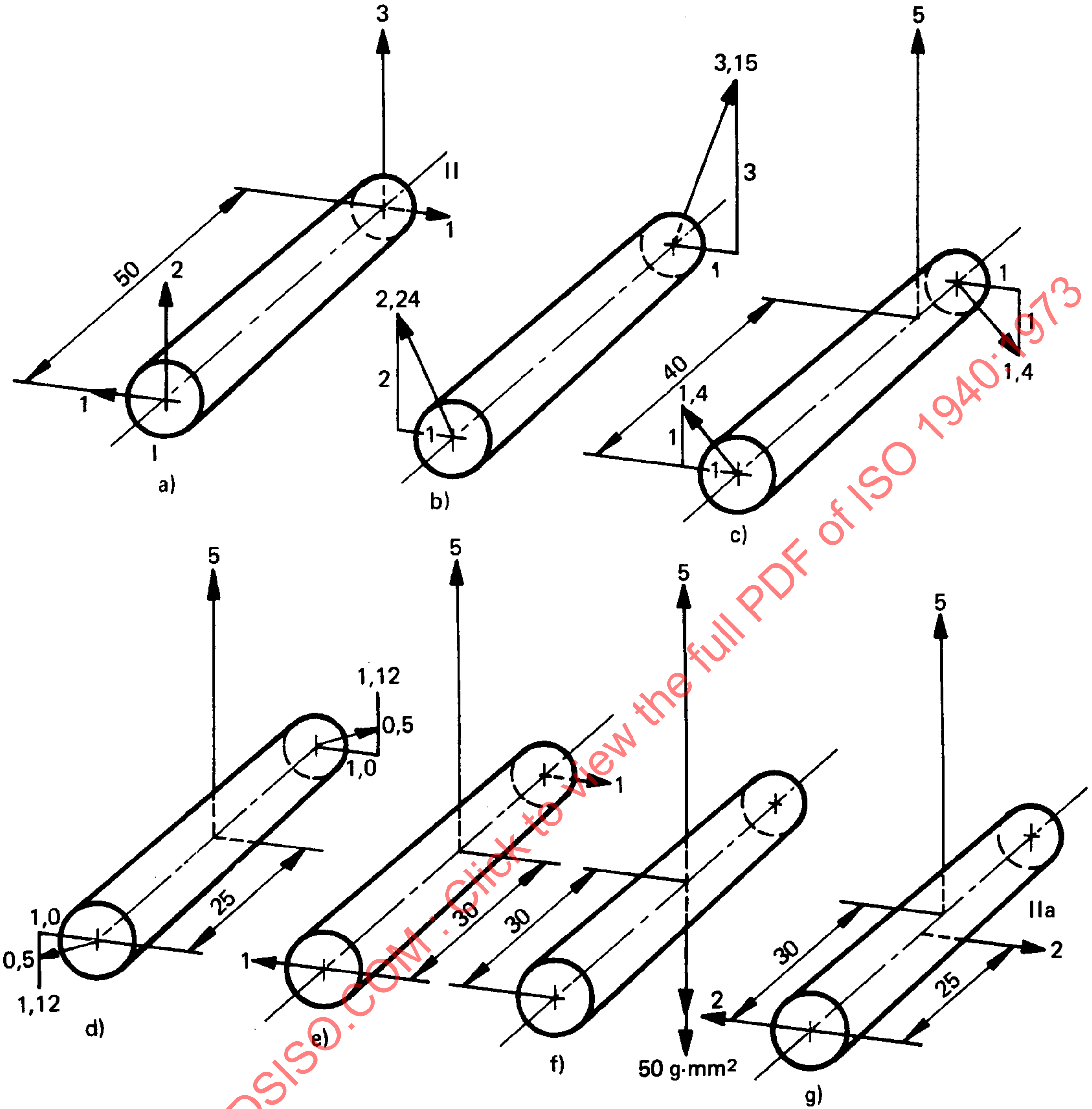
2) In general, for rigid rotors with two correction planes, one-half of the recommended residual unbalance is to be taken for each plane; these values apply usually for any two arbitrarily chosen planes, but the state of unbalance may be improved upon at the bearings. (See 3.2 and 3.4.) For disk-shaped rotors the full recommended value holds for one plane (see section 3).

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

4) For the purposes of this International Standard, 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 – Unless otherwise indicated, the units of vector amplitude or vector components are in gram millimetres. The units of length are in millimetres. The centre of gravity in all cases is located equidistant between correction planes I and II.



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

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

c) Quasi-static unbalance together with an associated couple unbalance referred to the two correction planes I and II. The quasi-static unbalance may be located anywhere, for example also in one of the correction planes.

d) Special case of c) : The line of action of the quasi-static unbalance passes through the centre of gravity. The unbalance therefore is static. There is an associated couple unbalance.

e) Another special case of c) : The quasi-static unbalance intersects the plane of the unbalance components. The couple unbalance then assumes its minimum value and lies in a plane perpendicular to the quasi-static unbalance.

f) Different representation of case e) : Quasi-static unbalance with a vector representation of the associated couple unbalance (right-hand rule).

g) Referred to planes I and IIa, the same minimum couple unbalance as in case e), together with the associated quasi-static unbalance.

FIGURE 1 – **Various representations of a given state of unbalance of a rigid rotor with correction planes in a rotor-fixed reference system**