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Mechanical vibration — Rotor balancing —

Part 11: Procedures and tolerances for rotors with rigid behaviour

Vibrations mécaniques — Équilibrage des rotors —
Partie 11: Modes opératoires et tolérances pour rotors à comportement rigide

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Ch. de Blandonnet 8 • CP 401
CH-1214 Vernier, Geneva, Switzerland
Tel. +41 22 749 01 11
Fax +41 22 749 09 47
copyright@iso.org
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Contents

Page

Foreword ..... **v**

Introduction ..... **vi**

1 Scope **1**

2 Normative references **1**

3 Terms and definitions **1**

4 Pertinent aspects of balancing **1**
4.1 General 1
4.2 Representation of the unbalance 1
4.3 Unbalance effects 2
4.4 Reference planes for unbalance tolerances 2
4.5 Correction planes 4
4.5.1 General 4
4.5.2 Rotors which need one correction plane only 4
4.5.3 Rotors which need two correction planes 4
4.5.4 Rotors with more than two correction planes 4
4.6 Permissible residual unbalance 4

5 Similarity considerations **5**
5.1 General 5
5.2 Permissible residual unbalance and rotor mass 5
5.3 Permissible residual specific unbalance and service speed 6

6 Specification of unbalance tolerances **6**
6.1 General 6
6.2 Derivation of the unbalance tolerances 6
6.3 Balance quality grade G 7
6.3.1 Classification 7
6.3.2 Special designs 7
6.3.3 Permissible residual unbalance 10
6.4 Experimental evaluation 10
6.5 Unbalance tolerances based on bearing forces or vibrations 10
6.5.1 Bearing forces 10
6.5.2 Vibrations 11
6.6 Methods based on established experience 11

7 Allocation of permissible residual unbalance to tolerance planes **11**
7.1 Single plane 11
7.2 Two planes 11
7.2.1 General 11
7.2.2 Limitations for inboard rotors 12
7.2.3 Limitations for outboard rotors 12

8 Allocation of unbalance tolerances to correction planes **13**
8.1 General 13
8.2 Single plane 14
8.3 Two planes 14

9 Assembled rotors **14**
9.1 General 14
9.2 Balanced as a unit 14
9.3 Balanced on component level 14

10 Accounting for errors in the verification of permissible residual unbalances **15**
10.1 General 15
10.2 Unbalance tolerance 15
10.3 Combined error of unbalance measurements 15

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(https://ecommerce.sist.si)
ISO 21940-11:2016
https://ecommerce.sist.si/catalog/standards/sist/...-48ac-be52-58ab181c4fa2/iso-21940-11-2016

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10.4 Verification of the permissible residual unbalance 15
10.4.1 General 15
10.4.2 Unbalance readings within tolerance 16
10.4.3 Unbalance readings out of tolerance 16
10.4.4 Region of uncertainly 16

Annex A (informative) **Example of the specification of permissible residual unbalance based on balance quality grade G and allocation to the tolerance planes** **17**

Annex B (informative) **Specification of unbalance tolerances based on bearing force limits** **21**

Annex C (informative) **Specification of unbalance tolerances based on established experience** **23**

Annex D (informative) **Rules for allocating unbalance tolerances from tolerance planes to correction planes** **25**

Bibliography **28**

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ISO 21940-11:2016
https://ecommerce.sist.si/catalog/standards/sist/484b9a61-2f36-48ac-be52-58ab181c4fa2/iso-21940-11-2016

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Foreword

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

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

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For an explanation on the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT) see the following URL: www.iso.org/iso/foreword.html.

The committee responsible for this document is ISO/TC 108, *Mechanical vibration, shock and condition monitoring*, Subcommittee SC 2, *Measurement and evaluation of mechanical vibration and shock as applied to machines, vehicles and structures*.

This first edition cancels and replaces ISO 1940-1:2003, which has been technically revised. The main changes are deletion of the terms and definitions which were transferred to ISO 21940-2 and a more pronounced explanation of the application of permissible residual unbalances for the processes of balancing a rotor and verifying its residual unbalance. Information on specification of unbalance tolerances based on vibration limits has been removed.

It also incorporates the Technical Corrigendum ISO 1940-1:2003/Cor 1:2005.

A list of parts in the ISO 21940 series can be found on the ISO website.

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Introduction

Rotor balancing is a procedure by which the mass distribution of a rotor (or part or module) is checked and, if necessary, adjusted to ensure the unbalance tolerance is met. This document covers the balancing of rotors with rigid behaviour. A rotor is said to be rigid when the flexure of the rotor caused by its unbalance distribution can be neglected with respect to the agreed unbalance tolerance at any speed up to the maximum service speed. For these rotors, the resultant unbalance, and often moment unbalance, are of interest, which when combined are expressed as a dynamic unbalance of the rotor.

The balancing machines available today enable residual unbalances to be reduced to very low limits. Therefore, it is necessary to specify an unbalance quality requirement for a balancing task, as in most cases it would not be cost-effective to reduce the unbalance to the limits of the balancing machine.

In addition to specifying an unbalance tolerance, it is necessary to consider the errors introduced by the balancing process. This document takes into account the influence of these errors to distinguish clearly between the specified permissible residual unbalance and the reduced residual unbalance values to be achieved during the balancing process.

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Mechanical vibration — Rotor balancing —

Part 11: Procedures and tolerances for rotors with rigid behaviour

1 Scope

This document establishes procedures and unbalance tolerances for balancing rotors with rigid behaviour. It specifies

a) the magnitude of the permissible residual unbalance,

b) the necessary number of correction planes,

c) the allocation of the permissible residual unbalance to the tolerance planes, and

d) how to account for errors in the balancing process.

NOTE In ISO 21940-14, the assessment of balancing errors is considered in detail. Fundamentals of rotor balancing are contained in ISO 19499 which gives an introduction to balancing.

This document does not cover the balancing of rotors with flexible behaviour. Procedures and tolerances for rotors with flexible behaviour are dealt with in ISO 21940-12.

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2 Normative references

There are no normative references in this document.

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3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 21940-2 apply.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— IEC Electropedia: available at http://www.electropedia.org/

— ISO Online browsing platform: available at http://www.iso.org/obp

4 Pertinent aspects of balancing

4.1 General

Rotor balancing is a procedure by which the mass distribution of a rotor is examined and, if necessary, adjusted to ensure that the residual unbalance or vibration in service is within specified limits. It should be noted that the vibration in service can originate from sources other than unbalance.

Rotor unbalance can be caused by design, material, manufacturing and assembly. Every rotor has an individual unbalance distribution along its length, even in series production.

4.2 Representation of the unbalance

For a rotor with rigid behaviour, different vectorial quantities can be used to represent the same unbalance as shown in Figure 1.

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Figure 1 a) to c) shows different representations in terms of resultant unbalance and resultant couple unbalance, whereas Figure 1 d) to f) shows different representations in terms of a dynamic unbalance in two planes.

NOTE 1 The resultant unbalance vector can be located in any radial plane (without changing magnitude and angle), but the associated resultant couple unbalance is dependent on the location of the resultant unbalance vector.

NOTE 2 The centre of unbalance is that location on the shaft axis for the resultant unbalance, where the resultant couple unbalance is a minimum.

If single-plane balancing is sufficient (see 4.5.2) or when considerations are made in terms of resultant unbalance and resultant couple unbalance (see 4.5.4), the representation in Figure 1 a) to c) is preferable.

In the case of typical two-plane considerations, the representation in Figure 1 d) to f) is advantageous.

4.3 Unbalance effects

Resultant unbalance and resultant moment unbalance (the latter can also be expressed as resultant couple unbalance) have different effects on forces on the bearings and on the vibration of the machine. In practice, therefore, both unbalances are often considered separately. Even if the unbalance is stated as a dynamic unbalance in two planes, it should be noted that in most cases there is a difference in effects if the unbalances predominantly form either a resultant unbalance or a resultant moment unbalance.

4.4 Reference planes for unbalance tolerances

It is recommended to use reference planes to state the unbalance tolerances. For these planes, only the magnitude of each residual unbalance needs to stay within the respective balance tolerance whatever the angular position may be.

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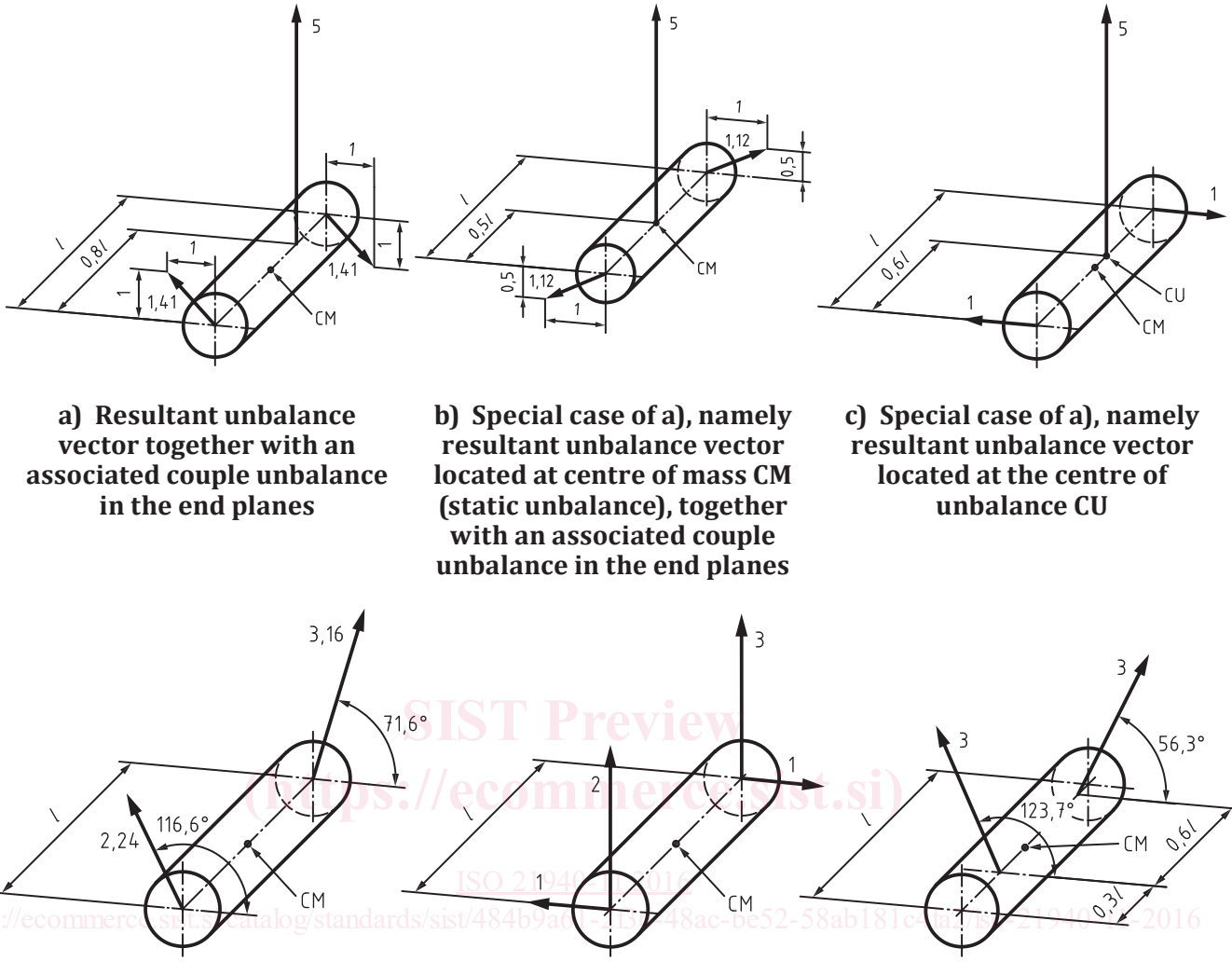
The aim of balancing is usually to reduce vibrations and forces transmitted through the bearings to the environment. For the purposes of this document, the reference planes for unbalance tolerances are taken to be the bearing planes. However, this use of bearing planes does not always apply.

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NOTE For a component without a shaft (e.g. a disc shaped element), but where the final bearing positions are known (or can be estimated), these planes can be used.

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Unbalances in gram millimetres



a) Resultant unbalance vector together with an associated couple unbalance in the end planes

b) Special case of a), namely resultant unbalance vector located at centre of mass CM (static unbalance), together with an associated couple unbalance in the end planes

c) Special case of a), namely resultant unbalance vector located at the centre of unbalance CU

d) Unbalance vector in each of the end planes

e) Two 90° unbalance components in each of the end planes

f) Unbalance vector in each of any two other planes

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Key

CM	centre of mass
CU	centre of unbalance
l	rotor length

NOTE For Figure 1 c), the associated couple unbalance is a minimum and lays in a plane orthogonal to the resultant unbalance vector.

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

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4.5 Correction planes

4.5.1 General

Rotors that are out of unbalance tolerance need correction. These unbalance corrections often cannot be performed in the planes where the unbalance tolerances were set, but need to be performed where material can be added, removed or relocated.

The number of necessary correction planes depends on the magnitude and distribution of the initial unbalance, as well as on the design of the rotor, e.g. the shape of the correction planes and their location relative to the tolerance planes.

4.5.2 Rotors which need one correction plane only

For some rotors, only the resultant unbalance is out of tolerance but the resultant moment unbalance is in tolerance. This typically happens with rotors having a single disc, provided that

a) the bearing distance is sufficiently large,

b) the disc rotates with sufficiently small axial runout, and

c) the correction plane for the resultant unbalance is properly chosen.

After single-plane balancing has been carried out on a sufficient number of rotors, the largest residual moment unbalance is determined and divided by the bearing distance, yielding a couple unbalance.

If, even in the worst case, the couple unbalance found this way is acceptable, it can be expected that single-plane balancing is sufficient.

For single-plane balancing, the rotor does not need to rotate but, for sensitivity and accuracy reasons, in most cases, rotational balancing machines are used.

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4.5.3 Rotors which need two correction planes

If a rotor with rigid behaviour does not comply with the conditions specified in 4.5.2, the moment unbalance needs to be reduced as well. In most cases, resultant unbalance and resultant moment unbalance are assembled into a dynamic unbalance: two unbalance vectors in two planes; see Figure 1 d).

For two-plane balancing, it is necessary for the rotor to rotate, since otherwise the moment unbalance would remain undetected.

4.5.4 Rotors with more than two correction planes

Although all rotors with rigid behaviour theoretically can be balanced in two planes, sometimes more than two correction planes are used, e.g.

a) in the case of separate corrections of resultant unbalance and couple unbalance, if the correction of the resultant unbalance is not performed in one (or both) of the couple planes, and

b) if the correction is spread along the rotor.

In special cases, spreading the correction along the rotor can be necessary due to restrictions in the correction planes (e.g. correction of crankshafts by drilling into the counterweights) or advisable in order to keep the function and component strength.

4.6 Permissible residual unbalance

In the simple case of an inboard rotor for which the couple unbalance may be ignored (see 4.5.2), its unbalance state can then be described as a single vectorial quantity, the unbalance, *U*.

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To obtain a satisfactory running of the rotor, the magnitude of this unbalance, i.e. the residual unbalance, U_{res}, shall not be higher than a permissible value, U_{per}:

$$U_{res} \leq U_{per} \quad (1)$$

More generally, the same applies to any type of a rotor with rigid behaviour, but then U_{per} covers the resultant unbalance and the resultant moment unbalance, see also 5.2.

NOTE The SI unit for U_{per} is kg·m (kilogram metres), but for balancing purposes, more practical units are g·mm (gram millimetres), kg·mm (kilogram millimetres) or mg·mm (milligram millimetres).

U_{per} is defined as the total tolerance in the plane of the centre of mass. In the case of two-plane balancing, this tolerance shall be allocated to the tolerance planes (see Clause 7).

5 Similarity considerations

5.1 General

Some considerations on similarity can help in the understanding and calculation of the influences of rotor mass and service speed on the permissible residual unbalance.

5.2 Permissible residual unbalance and rotor mass

In general, for rotors of the same type, the permissible residual unbalance, U_{per}, is proportional to the rotor mass, m, as given in Formula (2):

$$U_{per} \sim m \quad (2)$$

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The ratio of U_{per} to the rotor mass, m, is the permissible residual specific unbalance, e_{per}, as given in Formula (3):

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$$e_{per} = U_{per}/m \quad (3)$$

NOTE 1 The SI unit for U_{per}/m is kg·m/kg (kilogram metres per kilogram) or m (metres), but a more practical unit is g·mm/kg (gram millimetres per kilogram), which corresponds to µm (micrometres) because many permissible residual specific unbalances are between 0,1 µm and 10 µm.

NOTE 2 The term e_{per} is useful especially if geometric tolerances (e.g. runout, play) are related to unbalance tolerances.

NOTE 3 In the case of a rotor with only a resultant unbalance (see 4.5.2), e_{per} is the distance of the centre of mass from the shaft axis. However, in the case of a general rotor with both resultant unbalance and resultant moment unbalance present, e_{per} is an artificial quantity containing the effects of the resultant unbalance as well as of the resultant moment unbalance.

NOTE 4 There are limits for achievable residual specific unbalance, e_{per}, depending on the setup conditions in the balancing machine (e.g. centring, bearings and drive).

NOTE 5 Small values of e_{per} can only be achieved in practice if the accuracy of shaft journals (roundness, straightness, etc.) is adequate. In some cases, it can be necessary to balance the rotor in its own service bearings, using belt-, air- or self-drive. In other cases, balancing needs to be carried out with the rotor completely assembled in its own housing with bearings and self-drive, under service condition and temperature.

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5.3 Permissible residual specific unbalance and service speed

For rotors of the same type, experience shows that, in general, the permissible residual specific unbalance value, e_{per}, varies inversely with the service speed, n, of the rotor:

$$e_{per} \sim 1/n \tag{4}$$

Differently expressed, this relationship is given in Formula (5):

$$e_{per}\,\Omega = c \tag{5}$$

where

Ω is the angular velocity of the service speed, in rad/s (radians per second), with $\Omega = 2\pi\, n/(60$ s/min) and the service speed, n, in r/min (revolutions per minute);

c is a constant.

This relationship follows also from the fact that for geometrically similar rotors running at equal peripheral velocities, the stresses in the rotors and the bearing specific loads (due to centrifugal forces) are the same. The balance quality grade G (see 6.3) is based on this relationship.

6 Specification of unbalance tolerances

6.1 General

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The first step in the balancing process is to establish the magnitude of permissible residual unbalance of the rotor and to allocate it to the tolerance plane(s). In order to meet these unbalance tolerances reliably, reduced residual unbalance tolerances shall take account of errors as detailed in Clause 10.

NOTE 1 The ideal target value of the unbalance typically is zero (i.e. in a vector diagram, the unbalance tolerance is the radius of the circular tolerance region around the origin).

NOTE 2 Sometimes the target unbalance has a specified quantity, given by amount and angle (e.g. removed keys, asymmetric crank shafts, compensating shafts or rotational vibration exciter). In these cases, the unbalance tolerance is the radius of a circle around the specified target unbalance vector.

6.2 Derivation of the unbalance tolerances

The magnitude of permissible residual unbalance can be determined by five different methods. The methods are based on

a) balance quality grades, derived from long-term practical experience with a large number of different rotors (see 6.3),

b) experimental evaluation of permissible residual unbalances (see 6.4),

c) limited bearing forces due to unbalance (see 6.5.1),

d) limited vibrations due to unbalance (see 6.5.2), and

e) established experience with unbalance tolerances (see 6.6).

Both the choice of method and the permissible unbalance tolerance are recommended to be part of the agreement between the manufacturer and customer.

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