INTERNATIONAL STANDARD

ISO 21940-1

First edition
2019-02

Mechanical vibration — Rotor balancing —

Part 1: Introduction

Vibrations mécaniques — Équilibrage des rotors —

Partie 1: Introduction

Reference number
ISO 21940-1:2019(E)

ISO copyright office
CP 401 • Ch. de Blandonnet 8
CH-1214 Vernier, Geneva
Phone: +41 22 749 01 11
Fax: +41 22 749 09 47
Email: copyright@iso.org
Website: www.iso.org

Published in Switzerland

Contents

Page

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).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation on the voluntary nature of standards, 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.

This document was prepared by Technical Committee 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 of ISO 21940-1 cancels and replaces ISO 19499:2007, which has been technically revised. The main changes are as follows:

— reference made to all International Standards in the ISO 21940 series;

— deletion of former Table 2 "Guidelines for balancing procedures";

— deletion of former Annex C "How to determine rotor flexibility based on an estimation from its geometric design".

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

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

Vibration caused by rotor unbalance is one of the most critical issues in the design and maintenance of rotating machines. It gives rise to dynamic forces which adversely affect both machine and human health and well-being. The purpose of this document is to give guidance on the usage of the other parts of the ISO 21940 series.

Balancing is explained in a general manner, using the specific terms and definitions, to help readers to select the appropriate balancing approach for their application.

Mechanical vibration — Rotor balancing —

Part 1: **Introduction**

1 Scope

This document provides a general background to balancing technology, as used in the ISO 21940 series, and directs the reader to the appropriate parts of the series that include vocabulary, balancing procedures and tolerances, balancing machines and machine design for balancing.

Individual procedures are not included here as these can be found in the appropriate parts of ISO 21940.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 2041, *Mechanical vibration, shock and condition monitoring — Vocabulary*

ISO 21940-2, *Mechanical vibration — Rotor balancing — Part 2: Vocabulary*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 2041 and ISO 21940-2 apply.

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

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

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

4 Fundamentals of balancing

4.1 General

Balancing is a procedure by which the mass distribution of a rotor (or part of a rotor or module) is measured and adjusted to ensure that unbalance tolerances are met.

Many factors can cause rotor unbalance, e.g. non-homogenous material, manufacture, assembly, wear during operation, debris or an operational event. It is important to understand that every rotor, even in series production, has a unique individual unbalance distribution.

New rotors are commonly balanced by the manufacturer in balancing machines before installation into their operational environment. Following rework or repair, rotors can be rebalanced in a balancing machine or, if appropriate facilities are not available, the rotor can be balanced *in situ* (for details, see ISO 21940-13). For *in-situ* balancing, the rotor is held in its service bearings and support structure and rotated within its operational drive train.

When rotated, unbalance generates forces that can be directly measured by force gauges mounted on the structures supporting the bearings or indirectly by measuring either the motion of the bearing or the shaft. The unbalance vector can be calculated from these measurements and balancing achieved by adding, removing or moving correction masses on the rotor. Depending on the balancing task, the mass corrections are performed in one, two or more correction planes.

Inertia forces due to unbalances or correction masses added during the balancing process induce an excitation of the rotor and support system, which is observed as once-per-revolution vibration. Once-per-revolution vibration and vibration at other frequencies can also be excited by other effects, e.g. asymmetric stiffness, magnetic or fluid forces, but it is only the once-per-revolution effects that can be compensated for by balancing. Non-linear systems can also cause frequencies other than at once per revolution to be generated but these are usually a second order effect.

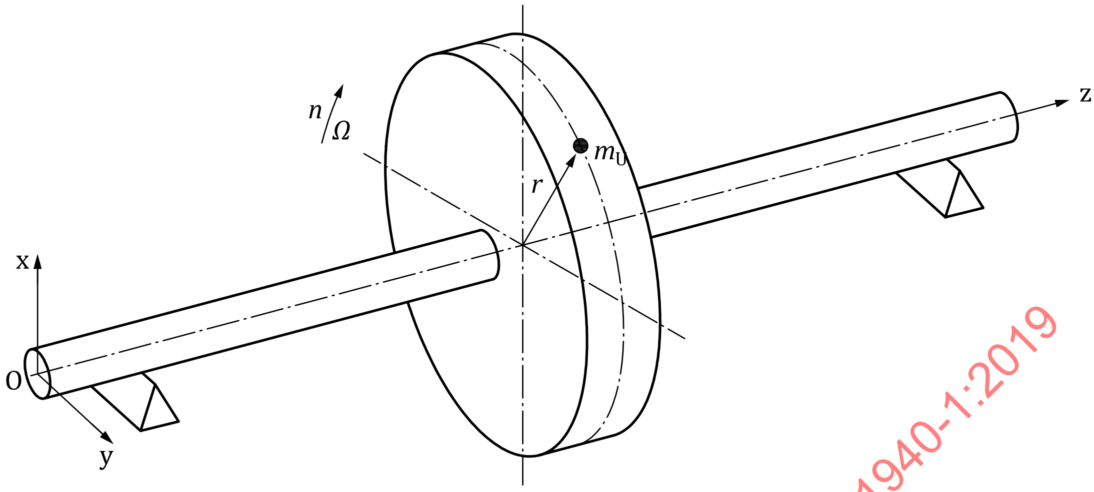
The theory of balancing is widely described in the literature (see e.g. References [11], [12]), and therefore only the basics are presented here to aid the understanding of the terms used in balancing standards and to direct the user towards the appropriate parts of ISO 21940.

4.2 Unbalance of a single disc

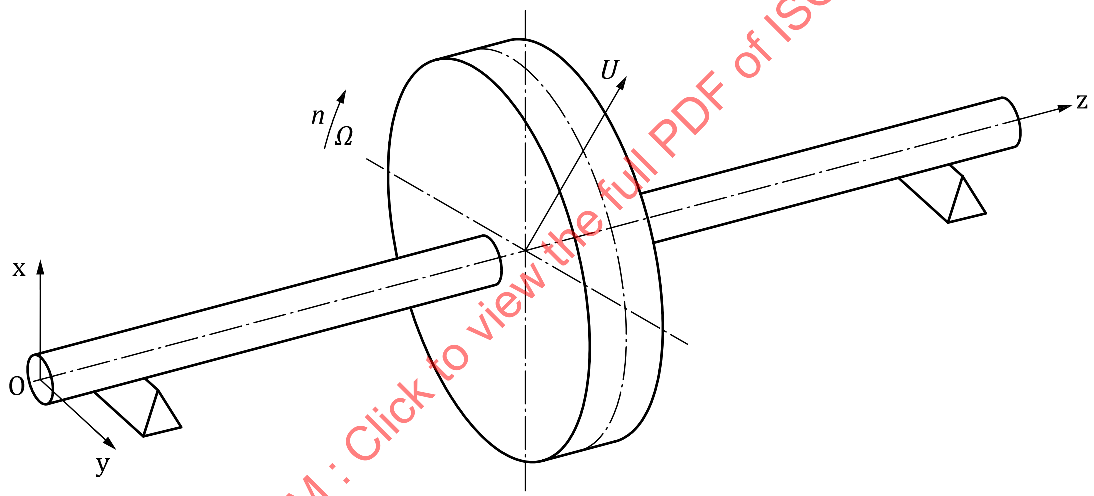
The simplest mechanical model of a rotor consists of a single disc supported on two bearings by a massless shaft as shown in Figure 1. An unbalance mass, m_U, on the disc with a radial distance from the shaft axis, $\boldsymbol{r}$, generates the unbalance vector, $\boldsymbol{U}$, whereby $\boldsymbol{U} = m_U\, \boldsymbol{r}$. The unbalance vector $\boldsymbol{U}$ is expressed in the unit of mass times length, usually kg·m, but for practical reasons, smaller units are generally used, e.g. kg mm, g mm or, for very small unbalances, mg mm.

NOTE Bold font indicates vector quantities.

At a rotational speed n (angular velocity Ω), the unbalance causes a centrifugal force $\boldsymbol{F} = \boldsymbol{U}\, \Omega^2$. When expressing the unbalance, $\boldsymbol{U}$, in kg·m, and the angular velocity, Ω, in rad/s, $\boldsymbol{F}$ is expressed in newtons, N.



a) Unbalance of a disc as unbalance mass m_U at radius *r*



b) Unbalance of a disc as unbalance vector *U*

Figure 1 — Unbalance of a disc

The unbalance, ***U***, can be expressed as the eccentricity, ***e***, of the disc mass, *M*, from the shaft axis, given by the expression ***U*** = *M* ***e***. See Figure 2.

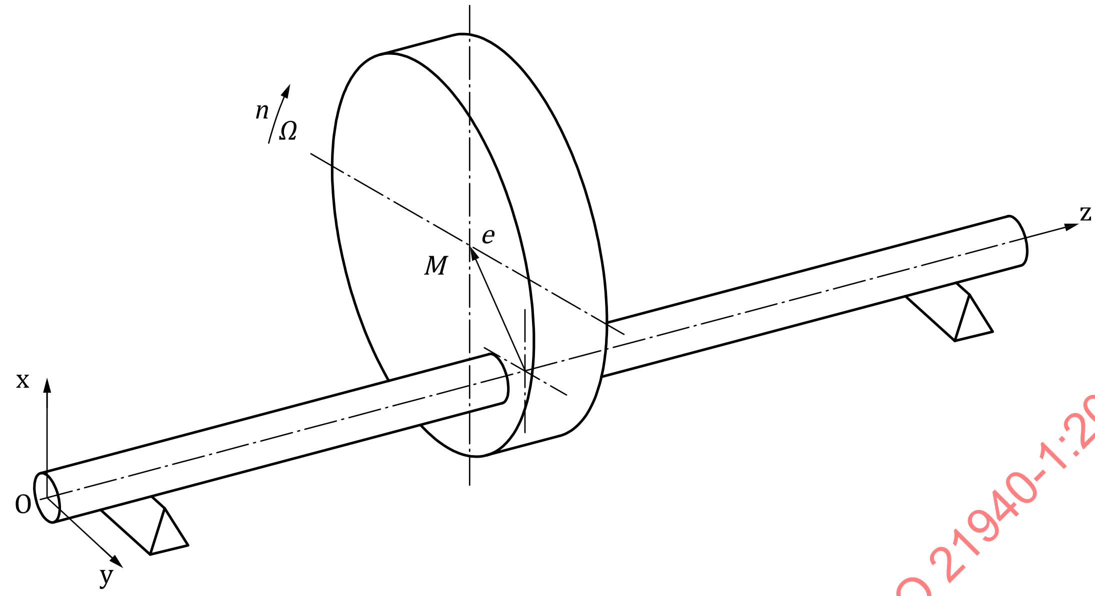


Figure 2 — Unbalance of a disc, expressed as the eccentricity of the mass centre from the shaft axis

4.3 Unbalance distribution

For a general rotor, with a certain axial length, unbalance is made up of an infinite number of unbalance vectors, distributed along the shaft axis. If a lumped-mass model is used to simulate the rotor behaviour, the unbalance can be represented by a finite number of unbalance vectors of different amplitudes and angular directions, as illustrated in Figure 3.

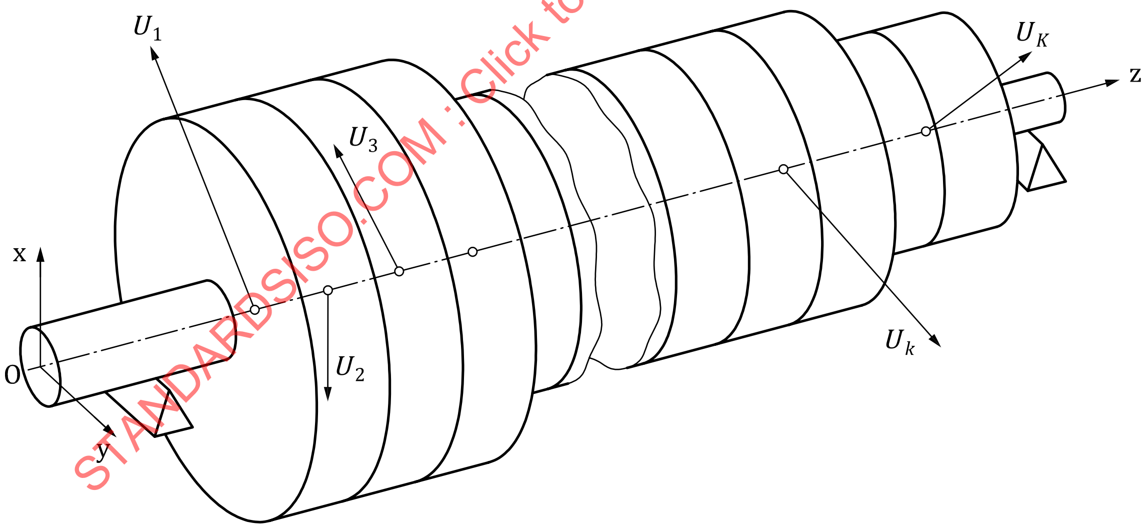


Figure 3 — Unbalance distribution in a rotor modelled as *K* disc elements perpendicular to the *z* axis

4.4 Unbalance representation

If all unbalance vectors were corrected in their respective planes, the rotor would be perfectly balanced but, in practice, it is neither possible nor necessary to measure and correct for all the individual unbalances. Throughout the ISO 21940 series, the following representations are used to specify rotor unbalance:

a) resultant unbalance $\boldsymbol{U}_{\mathrm{r}}$, vector sum of all unbalance vectors distributed along the rotor;

NOTE 1 The plane to state the resultant unbalance can be arbitrarily chosen.

NOTE 2 If the plane for the resultant unbalance is the plane of the mass centre, the unbalance is called static unbalance.

b) resultant moment unbalance $\boldsymbol{P}_{\mathrm{r}}$, the vector sum of the moments of all the unbalance vectors distributed along the rotor with respect to an arbitrarily selected plane perpendicular to the shaft axis;

c) resultant equivalent modal unbalance values $\boldsymbol{U}_{n\mathrm{e,r}}$, the unbalance distribution which affects each of the nth natural modes of the rotor system.

Mathematical and graphical representations of unbalance are described in Annex A.

NOTE 3 The resultant unbalance [see a)] and resultant moment unbalance [see b)] can be combined. The combination is called “dynamic unbalance” and is represented by two unbalance vectors in two arbitrarily chosen planes perpendicular to the shaft axis.

NOTE 4 The balancing procedures described in the ISO 21940 series assume the rotor system is linear and the modes of vibration are orthogonal. For example, adding 2 g mm mass correction has twice the effect of 1 g mm and the mode shape of one mode is not affected by other modes. Fluid-film bearings, which are often used in high-speed balancing machines, can introduce non-linearities and cross coupling between modes but generally the effects are small and the balancing procedures described in the ISO 21940 series can be adopted.

5 Factors to consider when balancing

5.1 General

For the purpose of balancing, it is normal to refer to rotors as rigid or flexible, i.e. they have rigid or flexible behaviour, respectively. However, the terms "rigid" or "flexible" are a gross simplification which can lead to a misinterpretation by suggesting that the balance classification of the rotor is only dependent on its physical construction. Unbalance is an intrinsic property of the rotor, but the dynamics of the bearings and support structure and the rotational speed of the rotor can affect the rotor’s response to unbalance. The balance quality to which the rotor is expected to run and the magnitude and distribution of the initial unbalance along the rotor also influence the chosen balancing procedure to be used. As a result, a rotor that behaves as rigid under one set of conditions (service speed, initial unbalance, unbalance tolerances, etc.) can behave as flexible under another set of conditions.

Guidance on rigid and flexible rotor behaviours is given in 5.2 and 5.3.

There are special cases of rotors with unbalance indications that change with speed or time in a way that cannot be explained with a bending shaft. These are considered in 5.3.3 and 5.3.4.

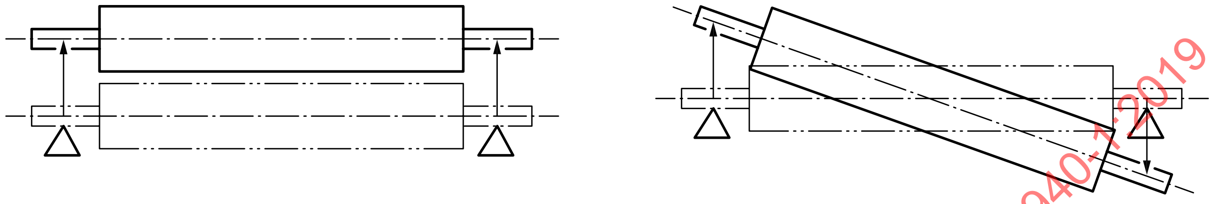
5.2 Rotors with rigid behaviour

Rigid rotor behaviour is where the flexure of a 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. The majority of these rotors operate way below the rotor-support system resonance speed.

Rotors with rigid behaviour can be balanced in accordance with the requirements of ISO 21940-11. The aim of such balancing is to correct for the resultant unbalance, with at least a single-plane balance correction for static unbalance, or with at least a two-plane balance correction for the dynamic unbalance.

Rotors designated to have rigid behaviour in the operating environment can be balanced at any speed on the balancing machine provided the speed is sufficiently low to ensure the rotor behaviour remains rigid, with no significant flexure, but sufficiently high to generate an unbalance force that can be accurately measured.

A rotor with unbalance, rotating with rigid behaviour on elastic supports, undergoes displacements that are combinations of the rigid-body modes, as shown in Figure 4, which can be related to static and moment unbalance. There is no significant flexure of the rotor and all displacements of the rotor arise from movements of the bearings and their support structure.



a) Parallel mode due to static unbalance **b) Tilting mode due to moment unbalance**

Figure 4 — Rotating rigid-body modes of a symmetric rotor on a symmetric elastic support structure

In the example shown in Figure 4 a), the principal axis of inertia of the rotor is offset parallel to the shaft axis and is defined as static unbalance.

The example shown in Figure 4 b), where the principal axis of inertia is inclined and crossing the shaft axis at the rotor's centre of mass, is defined as moment unbalance.

The addition of both static and moment unbalance, where the principal axis of inertia of the rotor is both inclined and offset from the shaft axis, is defined as dynamic unbalance.

In practice, every rotor has some flexural deflections in relation to the gross rigid-body motion of the rotor, but provided this flexure is small, the rotor can normally be considered to behave as rigid. ISO 21940-12:2016, Annex E, describes an experimental method to measure the rotor's flexibility and gives criteria for the degree of flexibility below which the rotor would normally be considered to be rigid.

NOTE Even with a rotor that is considered to behave as rigid and operates at rotational speeds well below its first flexural resonance speed, it can be necessary to consider the rotor's flexural behaviour when a low unbalance tolerance is required (as in 7.5.2).

5.3 Rotors with flexible behaviour

5.3.1 General

Flexible rotor behaviour classifies all rotors where the rotor's mass can move as a function of the rotor's rotational speed and is fully described in ISO 21940-12, which includes the following types:

a) shaft-elastic behaviour (see 5.3.2);

b) component-elastic behaviour (see 5.3.3); and

c) settling behaviour (see 5.3.4).

5.3.2 Shaft-elastic behaviour

A rotor is considered to have flexible behaviour when the unbalance causes the body of the rotor to bend in addition to the rigid-body modes described in 5.2. Figure 5 shows typical flexural mode shapes for a symmetric rotor. Most rotors with flexible behaviour can be balanced in accordance with the requirements of ISO 21940-12.

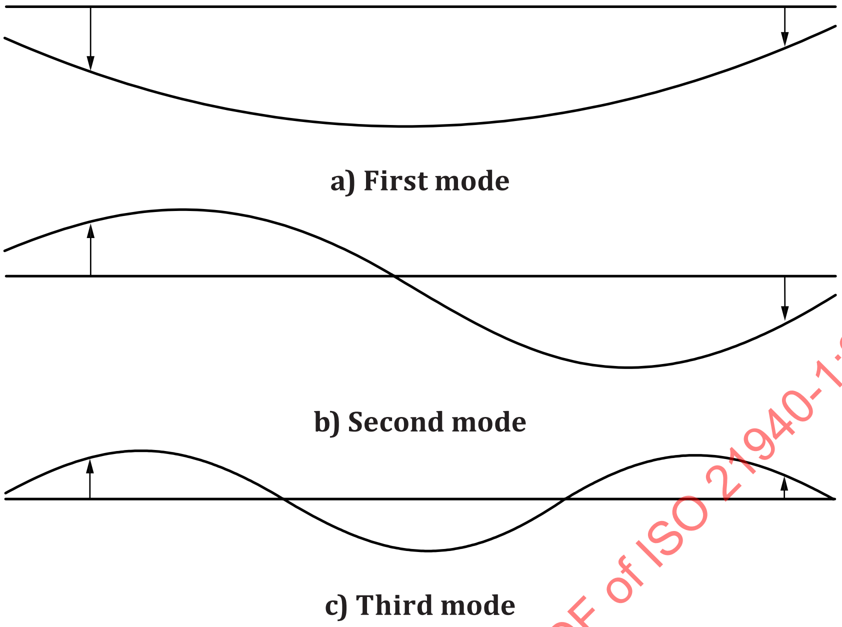


Figure 5 — Schematic representation of the first three flexural modes of a rotor with flexible behaviour on an elastic support structure

For the rotor with rigid behaviour described in 5.2, if the speed is increased, the unbalance tolerance reduced or the support structure changed, it can be necessary to take flexible behaviour into account, since the rigid-body balancing procedures might not be sufficient to achieve the desired balance condition.

5.3.3 Component-elastic behaviour

Rotors can have one or more components that are themselves either flexible or flexibly mounted so that the unbalance of the whole system might consistently change with speed (e.g. rotors with tie bars that deflect at high speed, rubber-bladed fans and single-phase induction motors with a centrifugal switch).

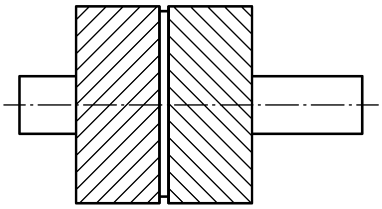
5.3.4 Settling behaviour

Rotors can have a method of construction where components settle after reaching a certain rotational speed or other condition. This movement then becomes stable after one or just a few events. Once components reach their final position and settle, the rotor can require further balancing (e.g. shrunk-on turbine discs, built-up rotors, copper winding in generators and generator retaining rings).

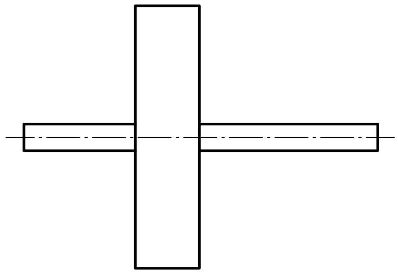
Once the component has settled, the rotor shall be balanced using the appropriate balancing procedure as discussed in 5.2 and 5.3.

5.4 Examples of rotor behaviours

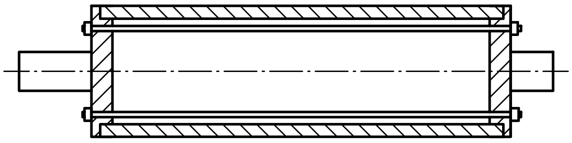
Some examples of rotor behaviours are illustrated in Figure 6 and details of these types of behaviour are further explained in Annex B.



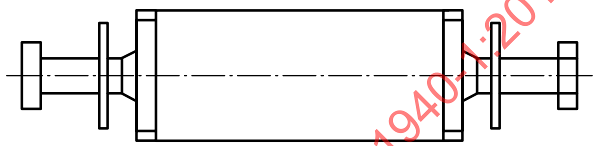
a) Rigid behaviour (e.g. a solid gear wheel at low speed)



b) Flexible behaviour (e.g. a disc on an elastic shaft at high speed)



c) Component elastic behaviour (e.g. a drum with tie bars, elastically deflecting under the centrifugal load)



d) Settling behaviour (e.g. a generator rotor with windings, once for all settling under a certain centrifugal load)

Figure 6 — Examples of rotor types that demonstrate particular rotor behaviours

6 Selection of a balancing procedure

At a given rotational speed, the physical properties of the rotor and those of its supporting structure control the rotor's response to a distributed unbalance along its length. Balancing to a required tolerance depends on these parameters and changing them or the unbalance tolerance specified can change the procedure needed (for further information, see Clause 7).

Since different balancing procedures require the use of different types of balancing machine and resource input, it is important to select an appropriate procedure to optimize the balancing process in order to meet the required unbalance tolerances.

There are two International Standards dealing with balancing these rotors:

— ISO 21940-11 for rotors with rigid behaviour,

— ISO 21940-12 for rotors with flexible behaviour.

Table 1 gives information on descriptions, stated in this document, as well as in the other relevant parts of ISO 21940, listing the balancing tasks and procedures.

Where possible, the rotor manufacturer or user should be consulted to establish the most suitable balancing procedure.

Table 1 — Overview of rotor behaviours, the description in this document and in the relevant parts of ISO 21940

Description in this document			Description in the ISO 21940 series	
Rotor behaviour		Example	Relevant part	Balancing task or procedure
Rigid behaviour (as described in 5.2)		Figure 6 a)	ISO 21940-11	Single-plane and two-plane balancing[a]
Flexible behaviour (see 5.3)	Shaft-elastic behaviour (as described in 5.3.2)	Figure 6 b)	ISO 21940-12	Six low-speed balancing procedures A to F Balancing procedure G: Multiple speed balancing Balancing procedure H: Service speed balancing
	Component-elastic behaviour (as described in 5.3.3)	Figure 6 c)		Procedure I: Fixed-speed balancing
	Settling behaviour (as described in 5.3.4)	Figure 6 d)		Settling of components at high speed, before (final) balancing[b]

[a] Single-plane balancing can correct for the resultant unbalance and two-plane balancing can correct for the resultant unbalance and the resultant moment unbalance.

[b] This procedure is mentioned in ISO 21940-12:2016, 7.3.3.12, but no designated letter is given.

7 Unbalance tolerances

7.1 General

Modern balancing machines enable residual unbalances to be reduced to low limits but it is uneconomic to over specify unbalance tolerance requirements. It is generally only necessary to define the unbalance tolerance needed for a rotor to operate with acceptable vibration and dynamic forces in its normal service environment.

7.2 Permissible residual unbalance

7.2.1 General

To achieve acceptable vibration magnitudes and dynamic forces for the rotor operating *in situ*, the permissible values of residual unbalance should be stated by the rotor's manufacturer. However, where this information is not available, the ISO 21940 series can provide guidance.

7.2.2 Permissible residual unbalance for rotors with rigid behaviour

When balancing rotors with rigid behaviour, the measured force and vibration are directly related to the permissible residual unbalance, $\boldsymbol{U}_{\mathrm{per}}$, which can be defined in different ways, as described in ISO 21940-11. In this way, both the resultant unbalance, $\boldsymbol{U}_{\mathrm{r}}$, and resultant moment unbalance, $\boldsymbol{P}_{\mathrm{r}}$, can be balanced to the required tolerance for the rotor.

7.2.3 Permissible residual unbalance for rotors with flexible behaviour

When balancing rotors with flexible behaviour, the measured force and vibration are dependent on the dynamics of the whole rotor system that includes its bearings and support structure. ISO 21940-12 describes methods to calculate the unbalance tolerance based on vibration values or permissible residual unbalances, derived from the permissible residual unbalances calculated in ISO 21940-11.

Balancing rotors with flexible behaviour requires the residual static unbalance, $\boldsymbol{U}_{\mathrm{r}}$, residual moment unbalance, $\boldsymbol{P}_{\mathrm{r}}$, and residual equivalent modal unbalance values, $\boldsymbol{U}_{n\mathrm{e,r}}$ (n = 1, 2, ...), to be reduced to the required tolerance for the rotor. This may include the consideration of vibration modes above service speed (as described in 7.4).

The procedures to evaluate modal unbalance and methods to determine permissible residual modal unbalance are described in ISO 21940-12.

7.3 Vibration limits

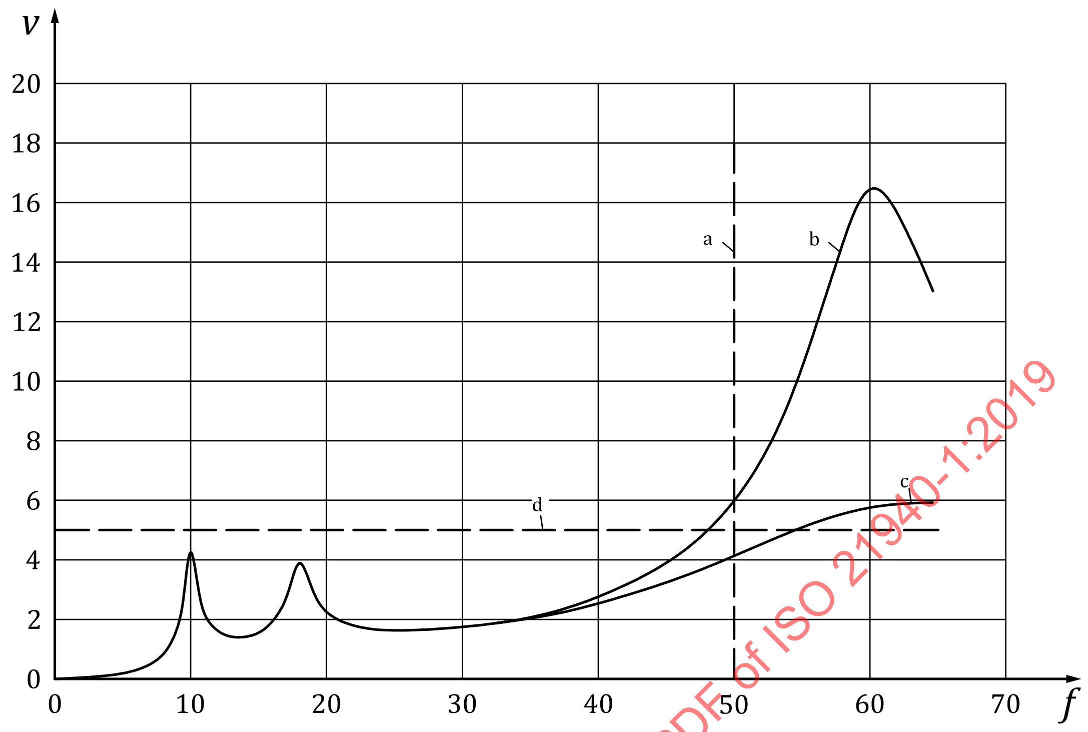
Vibration limits are commonly used when balancing rotors with flexible behaviour in high-speed balancing machines. The balancing machine operator may supply vibration limits where it can be demonstrated that similar rotors balanced to these limits operated successfully *in situ*.

NOTE Balancing machines can have different pedestal system dynamic properties and therefore vibration limits for a given rotor can be set individually for each balancing machine and for a selected speed range.

Where detailed information is available concerning the relationship of support stiffness and measurement positions between the balancing machine and the site conditions, a method to estimate vibration limits is presented in ISO 21940-12.

7.4 Influence of modes above service speed

When a low residual unbalance or vibration magnitude is required, it can be necessary to consider shaft flexural modes that occur at frequencies above the service speed. The effects of these higher modes can become important depending upon their proximity to the service speed, the unbalance distribution which can be affected by preceding balancing steps and the amount of damping in the rotor system (see ISO 21940-31). For example, the rotor response schematically shown in Figure 7 is significantly affected at the service speed (3 000 r/min, i.e. 50 Hz) by the higher mode, even though the lower modes have been balanced within limits. Here, depending on the amount of damping, the rotor would not meet the vibration limits based on an RMS vibration limit of 5 mm/s at the normal service speed due to the higher mode, even though the lower modes are acceptable.



Key

f	frequency, Hz	a	Service speed 50 Hz.
v	RMS vibration magnitude, mm/s	b	Low damping.
		c	High damping.
		d	Vibration tolerance.

Figure 7 — Influence of a rotor mode above service speed

7.5 Factors influencing balancing procedures

7.5.1 General

The rotor's response to a distributed unbalance along its length is controlled at all speeds by the physical properties of the rotor and its supporting structure. Balancing to a required quality depends on these parameters, and changing them or the unbalance tolerance specified can change the procedure needed to meet the balance requirements.

7.5.2 Tolerances

By simply reducing the unbalance tolerance, it can be necessary to reconsider the behaviour of the rotor and adopt a different procedure to bring the rotor within tolerance. Examples include:

a) A rotor with rigid behaviour, balanced using a single-plane procedure to reduce resultant unbalance, can simply require a single-plane balancing with this reduced tolerance;

b) A rotor with rigid behaviour, balanced using a single-plane procedure to reduce resultant unbalance can additionally require a two-plane procedure to account for the resultant moment unbalance;

c) A rotor with rigid behaviour, balanced in two planes to reduce both resultant and moment unbalance (dynamic unbalance) can additionally require flexible-behaviour procedures to reduce contributions from the modal unbalances, even though the rotor is running below its first flexural resonance speed;

d) A rotor with flexible behaviour, which has been balanced to reduce the dynamic unbalance and a number of modal unbalances, can require additional flexible rotor balancing procedures to reduce modal unbalances of even more (higher) flexural modes of the rotor, even though the rotor is running below the flexural resonance speeds of these higher modes (see 7.4);

e) It is possible that a rotor with either rigid or flexible behaviour, successfully balanced using the appropriate procedure, needs to be balanced further to take account of component-elastic or component settling behaviours;

f) Where a tighter tolerance can only be achieved at a single speed, the service speed balancing procedure (ISO 21940-12, procedure I) sometimes needs to be considered.

7.5.3 Speed and support conditions

Other changes of rotor behaviour can occur if operational conditions are changed (e.g. by changing the service speed or bearing design or support structure).

7.5.4 Initial unbalance

The initial unbalance distribution has an influence on the vibration response of the rotor system. It determines which unbalance (see Clause 4) is out of tolerance and therefore needs attention. Different manufacturing and assembling procedures can lead to different magnitudes of initial unbalance.

8 Selection of a balancing machine

8.1 General

Rotor behaviour together with the chosen balancing procedure dictate whether a low- or high-speed balancing machine is required. Some examples of balancing machine requirements are provided in Table 2.

Table 2 — Examples of balancing machine requirements

	Balancing machine requirements
1	Rotor mass
2	Bearing types, size and centre distance
3	Rotor length
4	Rotor diameter
5	Minimum achievable residual unbalance
6	Unbalance reduction ratio
7	Bearing support stiffness to match the installed environment
8	Vacuum facilities for high-speed bladed rotors
9	Allowable maximum speed for bladed rotors on low-speed machines
10	Induced electrical current for high-speed electrical rotors
11	Influence of rotor overhangs in the balancing facility
NOTE 1 This list is not exhaustive and it is possible that other requirements are appropriate for special machines.	
NOTE 2 ISO 21940-21 gives detailed requirements for low-speed balancing machines.	

Different types of unbalance require the use of different types of balancing machine, for example:

a) resultant unbalance: a single-plane balancing machine (low-speed balancing machine) is sufficient;

b) resultant moment unbalance: a two-plane balancing machine (low-speed balancing machine) is needed;

c) modal unbalances: a high-speed balancing machine is often needed.

NOTE 1 Single-plane balancing can also be performed on a two-plane balancing machine.

NOTE 2 A high-speed balancing machine can usually handle both low-speed balancing and high-speed balancing.

NOTE 3 Rotors with flexible behaviour as classified in ISO 21940-12, procedures A to F, can be adequately balanced at low speed.

8.2 Special requirements

While the rotor is in the balancing machine, additional tests can be undertaken to ensure that it is fit for purpose. Table 3 gives examples of additional tests that can be performed (e.g. on a large electrical generator rotor) while the rotor is still in the balancing machine. Similarly, the need for additional tests should be considered for specialized rotors whilst in the balancing machine.

Table 3 — Examples of tests that can be undertaken in the balancing machine for an electrical generator rotor

	Additional test requirements
1	Overspeed the rotor to settle the end rings
2	Undertake thermal stability checks
3	Perform electrical tests to check the integrity of the windings
These tests should only be carried out provided the appropriate facilities are available and safety requirements are satisfied.	

9 International Standards on balancing

9.1 General

In the field of balancing, most International Standards are collated in the ISO 21940 series, which to aid the user are divided into five main areas, as shown in Table 4. Some application-specific balancing standards for rotating machines which are not in the ISO 21940 series are included in the Bibliography.

Table 4 — Topic coverage in International Standards on balancing

Topic	ISO number	Topic
Introduction	ISO 21940-1	Introduction
Vocabulary	ISO 21940-2	Balancing vocabulary
	ISO 2041	Vibration and shock vocabulary
Balancing procedures and tolerances	ISO 21940-11	Procedures and tolerances for rotors with rigid behaviour
	ISO 21940-12	Procedures and tolerances for rotors with flexible behaviour
	ISO 21940-13	Criteria and safeguards for the *in-situ* balancing of medium and large rotors
	ISO 21940-14	Procedures for addressing balancing errors
Balancing machines	ISO 21940-21	Description and evaluation of balancing machines
	ISO 21940-23	Enclosures and other protective measures for the measuring station of balancing machines
Machine design for balancing	ISO 21940-31	Susceptibility and sensitivity of machines to unbalance
	ISO 21940-32	Shaft and fitment key convention

9.2 Vocabulary

9.2.1 ISO 21940-2 — Balancing vocabulary

ISO 21940-2 defines the vocabulary used for balancing. It also

a) provides an alphabetical index of balancing vocabulary; and

b) gives an illustrated guide to balancing machines terminology.

9.2.2 ISO 2041 — Vibration and shock vocabulary

ISO 2041 defines the general vocabulary describing the terms used in vibration, shock and condition monitoring.

9.3 Balancing procedures and tolerances

9.3.1 General

ISO 21940-11, ISO 21940-12, ISO 21940-13 and ISO 21940-14 give indications of procedures and tolerances for different rotor behaviours, as well as how to avoid gross deficiencies and unnecessarily restrictive requirements. They are not intended to be used as acceptance specifications.

9.3.2 ISO 21940-11 — Procedures and tolerances for rotors with rigid behaviour

ISO 21940-11 describes the procedures and tolerances to be used for 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.

9.3.3 ISO 21940-12 — Procedures and tolerances for rotors with flexible behaviour

ISO 21940-12 describes the procedures and tolerances for rotors with flexible behaviour and

a) provides typical configurations of rotors with flexible behaviour;

b) specifies balancing requirements in accordance with their characteristics;

c) lists balancing procedures;

d) provides methods of assessment of the final state of balance; and

e) gives guidelines for the unbalance tolerance.

Guidance given in ISO 21940-12 for acceptable unbalance tolerances is normally applicable for machines that satisfy the susceptibility and sensitivity requirements outlined in ISO 21940-31 (for further information, see 9.5.1).

9.3.4 ISO 21940-13 — Criteria and safeguards for the *in-situ* balancing of medium and large rotors

ISO 21940-13 specifies procedures to be adopted when balancing medium and large rotors installed in their own bearings on site. It addresses the conditions under which it is appropriate to undertake *in-situ* balancing, the instrumentation required, the safety implications and the requirements for reporting and maintaining records.

The requirements of ISO 21940-13 may be used as a basis for drafting a contract to undertake *in-situ* balancing but it does not provide guidance on the methods to be used to calculate the correction masses from measured vibration data.

NOTE The procedures covered in ISO 21940-13 are suitable for medium and large machines, which includes small main boiler feed pumps below 1 MW up to large steam and gas turbines in excess of 40 MW. However, many of the principles are equally applicable to machines of a smaller size, where it is necessary to maintain good records of the vibration behaviour and the correction mass configurations.

9.3.5 ISO 21940-14 — Procedures for addressing balancing errors

ISO 21940-14 describes the procedures to be used for assessing balancing errors and specifies the requirements for

a) identifying errors in the unbalance measuring process of a rotor;

b) assessing the identified errors; and

c) taking the errors into account when determining and verifying residual unbalances.

ISO 21940-14 specifies balance acceptance criteria, in terms of residual unbalance, for use both directly after balancing and for a subsequent check of the unbalance tolerance by the user.

9.4 Balancing machines

9.4.1 ISO 21940-21 — Description and evaluation of balancing machines

ISO 21940-21 specifies the requirements for evaluating the performance of machines for balancing rotating components, which are suitable for

a) rotors with rigid behaviour at the balancing speed (see 5.2); or

b) rotors with flexible behaviour balanced in accordance with low-speed procedures.

NOTE Generally, low-speed balancing is used for rotors with rigid behaviour and high-speed balancing is used for rotors with flexible behaviour. However, with the use of appropriate procedures, it is possible, under some circumstances, to balance rotors with flexible behaviour at low speed so as to ensure satisfactory running when the rotor is installed in its final environment. The details of these special requirements are described in ISO 21940-12.

9.4.2 ISO 21940-23 — Enclosures and other protective measures for the measuring station of balancing machines

ISO 21940-23 is applicable to the measuring station of a balancing machine and

a) specifies the requirements for enclosures and other protective measures used to minimize mechanical hazards produced by the rotor in the unbalance measuring station of rotational balancing machines;

b) describes the hazards which are associated with the operation of balancing machines under a variety of rotor and balancing conditions; and

c) defines different classes of protection that enclosures and other protective measures provide and describes the limits of applicability for each class of protection.

ISO 21940-23 does not, even if they are combined with the measuring station, include requirements covering

— devices used for adjusting the mass distribution of a rotor;

— devices to transfer the rotor to another station;

— special enclosure features, e.g. noise reduction, windage reduction or vacuum (which can be required to spin bladed rotors at balancing speed).

9.5 Machine design for balancing

9.5.1 ISO 21940-31 — Susceptibility and sensitivity of machines to unbalance

ISO 21940-31 specifies

a) how to determine machine vibration sensitivity due to unbalance;

b) evaluation guidelines for the vibration sensitivity considering the proximity of the resonance speeds to the service speed;

c) a classification method that groups machines into one of three types based on the likelihood that the machine will experience unbalance changes during normal operation; and

d) methods to apply the numerical sensitivity values using simple sample cases.

9.5.2 ISO 21940-32 — Shaft and fitment key convention

ISO 21940-32 specifies

a) a convention for balancing the individual components (e.g. shaft and fitments) of a keyed rotor assembly to ensure the compatibility of all balanced components so that, when they have been assembled, the overall unbalance tolerance and/or vibration limit for the rotor assembly is met;

b) that half-keys be used when balancing the individual components of a keyed rotor assembly; and

c) a way of marking the components balanced in accordance with the key convention used.

The requirements in ISO 21940-32 apply to rotors balanced in a balancing machine, in their own bearings or *in situ*. The key convention can also be applied when measuring the residual unbalance and/or vibration of rotors with keyways, but to which fitments have not yet been assembled.

In addition to specifying that keys of constant rectangular or square cross-section mounted parallel to the shaft centreline are used, ISO 21940-32 applies to keys mounted on tapered shaft surfaces (e.g. woodruff and gibhead keys, dowels and other types of key). The principle of the half-key convention is applied as appropriate to the shape and location of the key.

Annex A
(informative)

Mathematical and graphical representation of unbalance

A.1 Notation

NOTE Bold font indicates vector quantities.

Table A.1 — Balancing notation as used in this annex

b	distance between the two couple unbalance vectors (mm)
$\mathbf{C}_k$, $-\mathbf{C}_k$	both unbalance vectors of the couple unbalance (g·mm), based on moment unbalance $\mathbf{P}_k$
$\mathbf{C}_\mathrm{r}$, $-\mathbf{C}_\mathrm{r}$	both unbalance vectors of a resultant couple unbalance (g·mm), based on resultant moment unbalance $\mathbf{P}_\mathrm{r}$
k	counter for the rotor elements, from 1 to K
$\mathbf{l}_k = (\mathbf{z}_k - \mathbf{z}_\mathrm{r})$	distance (mm) from the chosen plane of the resultant unbalance to the unbalance in the kth rotor element
n	counter for mode functions, 1, 2, 3, ...
O	datum mark for coordinates x, y, z
$\mathbf{P}_k$	moment unbalance (g·mm^2) of an unbalance $\mathbf{U}_k$
$\mathbf{P}_\mathrm{r}$	resultant moment unbalance (g·mm^2)
R	radial plane
$\mathbf{U}_k$	unbalance vector (g·mm) in an individual element
$\mathbf{U}_{n,k}$	individual nth modal unbalance (g·mm) in rotor element k
$\mathbf{U}_{n,\mathrm{r}}$	resultant nth modal unbalance (g·mm)
$\mathbf{U}_{n\mathrm{e},k}$	individual nth equivalent modal unbalance (g·mm), based on unbalance in rotor element k
$\mathbf{U}_{n\mathrm{e,r}}$	resultant nth equivalent modal unbalance (g·mm)
$\mathbf{U}_\mathrm{r}$	resultant unbalance (g·mm)
z	coordinate of rotor axis (mm)
$\mathbf{z}_k$	distance (mm) from a datum mark O to the plane of $\mathbf{U}_k$
$\mathbf{z}_\mathrm{r}$	distance (mm) from a datum mark O to the chosen plane of the resultant unbalance $\mathbf{U}_\mathrm{r}$
$\mathbf{z}_\mathrm{C}$, $\mathbf{z}_{-\mathrm{C}}$	distances of both couple unbalance vectors from a datum mark O (mm)
$\phi_n(z)$	shape function of the nth flexural mode
$\phi_n(z_k)$	ordinate of nth mode function at rotor element k
$\phi_{n,\max}$	maximum ordinate of nth mode function

A.2 Objective

This annex shows how a set of unbalance vectors, distributed along the rotor (as shown in Figure 1), contributes to

a) the resultant unbalance,

b) the resultant moment unbalance and the resultant couple unbalance, and

c) the modal unbalance of the rotor mode shapes.

A.3 Unbalance vectors

It is assumed that the rotor can be split into a number of slices (discs), each with a single unbalance vector, while their respective couple unbalance is neglectable. Then Figure A.1 shows the rotor consisting of K elements with associated unbalance vectors $\boldsymbol{U}_k$ (k = 1, 2, ..., K). The origin of each unbalance vector $\boldsymbol{U}_k$ is located a distance $\boldsymbol{z}_k$ from a datum mark O.

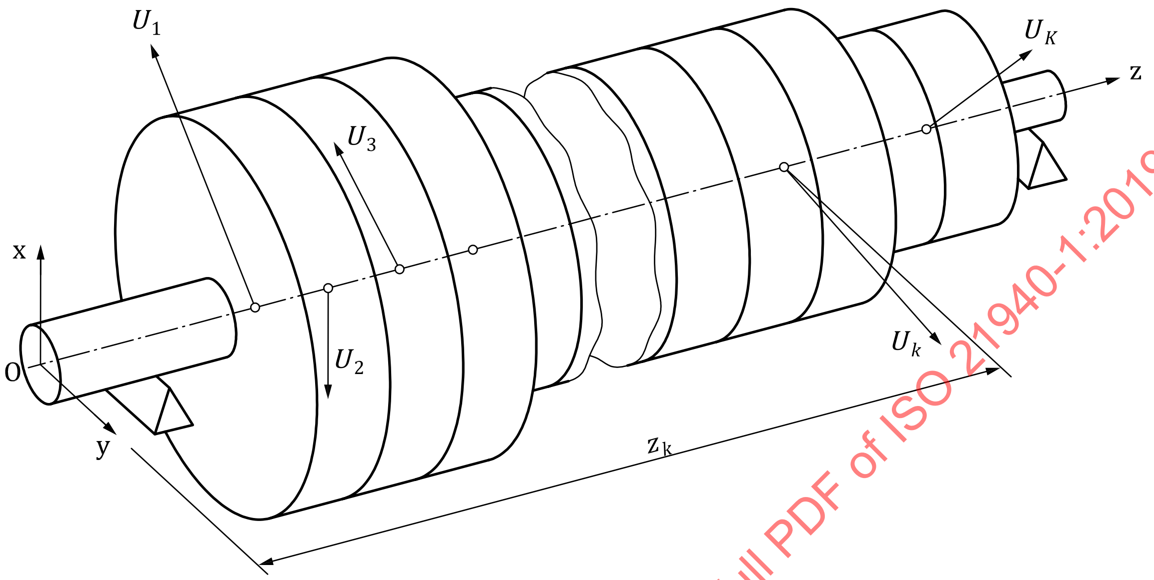


Figure A.1 — Unbalance distribution in the rotor modelled as *K* elements

The vector sum of all unbalance vectors, represented by the resultant unbalance, $\boldsymbol{U}_r$, is expressed by Formula (A.1).

$$\boldsymbol{U}_r = \sum_{k=1}^{K} \boldsymbol{U}_k \tag{A.1}$$

NOTE 1 The resultant unbalance is not linked to a certain radial plane. The choice of the plane is entirely arbitrary.

NOTE 2 The moment unbalance (see Clause A.4) is dependent on the position of the resultant unbalance.

The resultant unbalance can also be determined graphically. This method is illustrated using an example with K = 10. An exploded view of the centre planes of the 10 rotor elements and associated individual unbalance vectors is shown in Figure A.2.

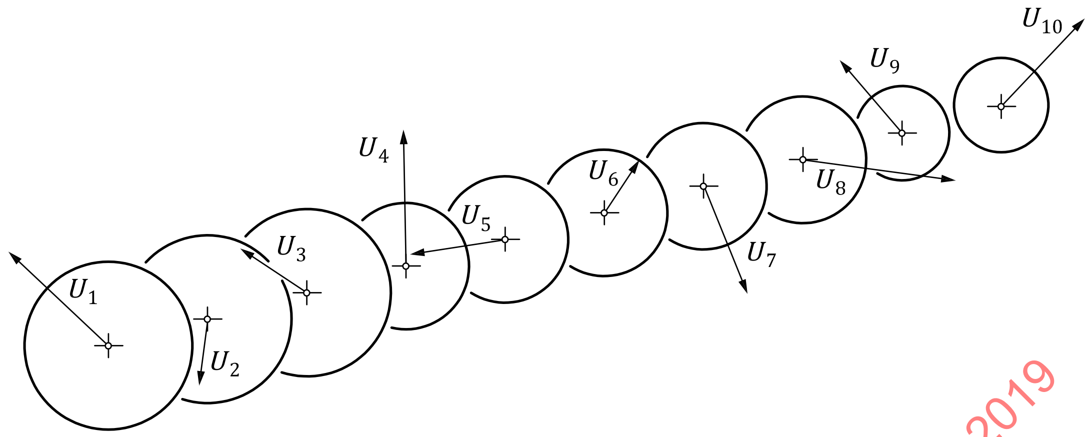
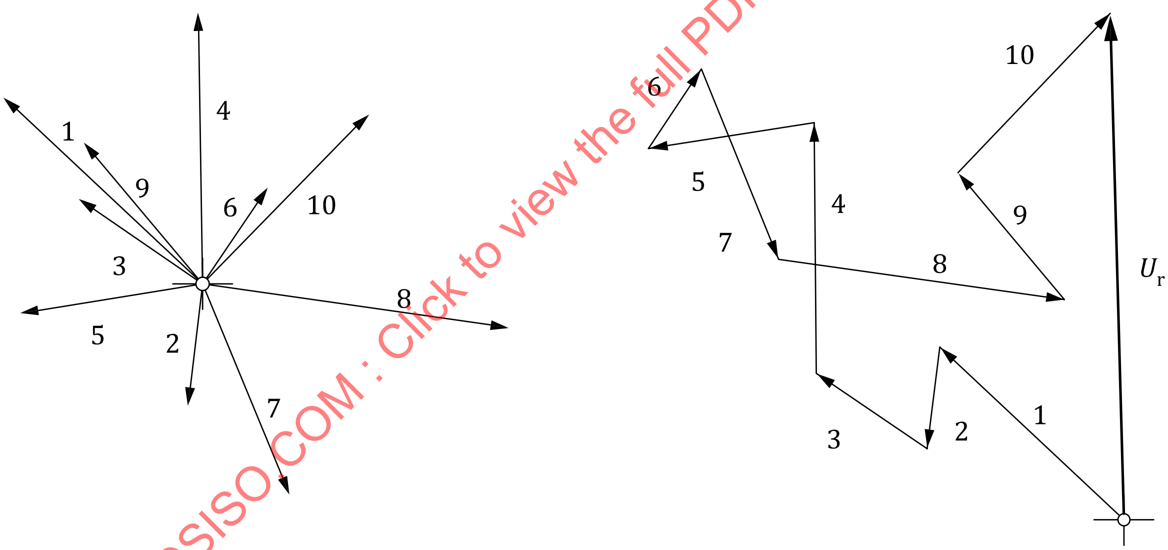


Figure A.2 — Exploded view of unbalance vectors from Figure A.1 for $K = 10$ individual rotor elements

The unbalance vectors, projected onto an arbitrary plane perpendicular to the shaft axis, are shown in Figure A.3 a). The resultant unbalance vector, $\boldsymbol{U}_{\mathrm{r}}$, is equal to the vector sum of the individual unbalance vectors as shown in Figure A.3 b).



a) **Unbalance vectors with the same origin (view along the *z* axis)**

b) **Graphical representation of the resultant unbalance**

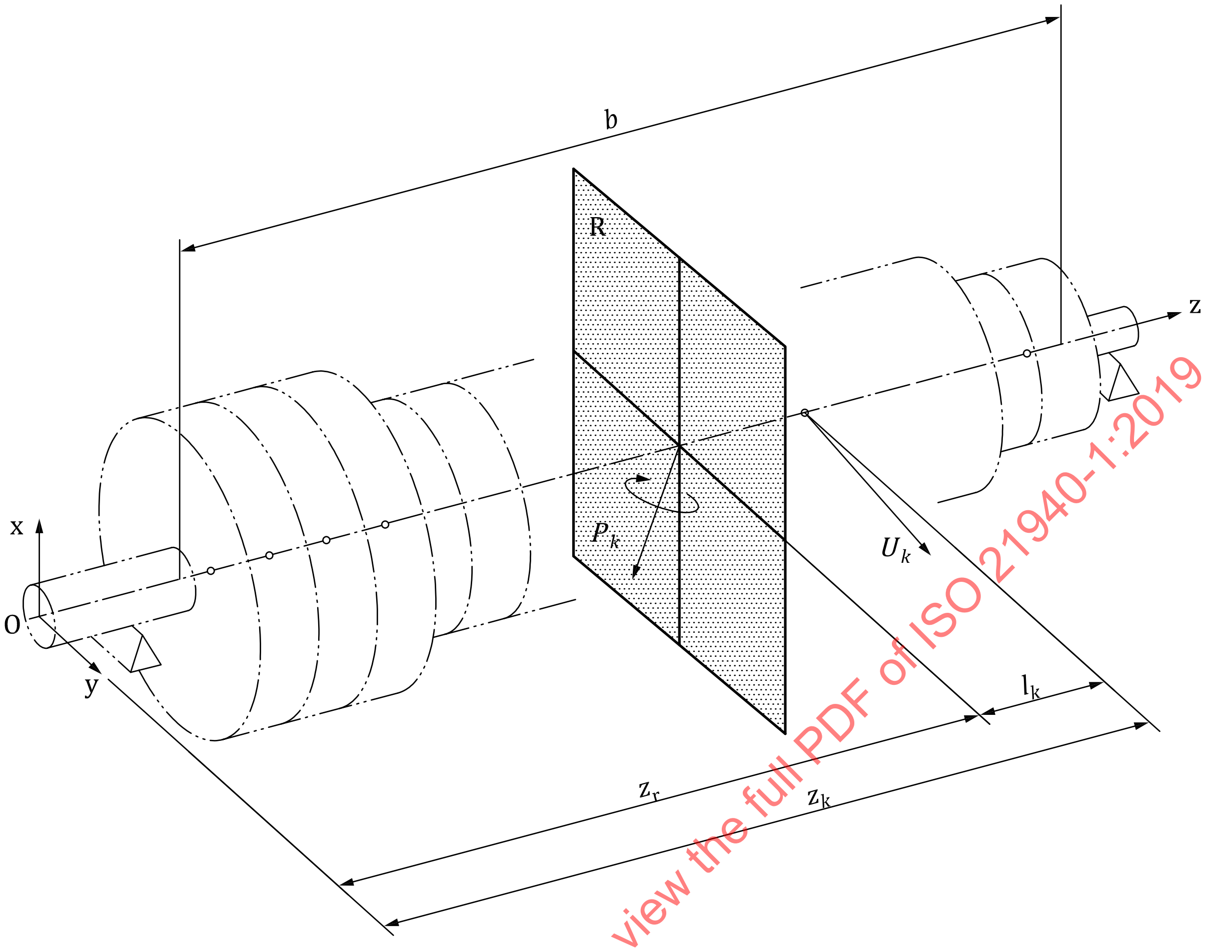
Figure A.3 — Graphical determination of the resultant unbalance vector, $\boldsymbol{U}_{\mathrm{r}}$

A.4 Moment unbalance

A moment unbalance can only be defined with respect to a plane perpendicular to the shaft axis as shown in Figure A.4. This arbitrarily chosen plane then becomes the plane of the resultant unbalance, since only in this position does the resultant unbalance not affect the moment unbalance. If a different plane is chosen, the value of moment unbalance changes but the resultant unbalance remains unchanged.

The moment unbalance, $\boldsymbol{P}_k$, can be calculated for an unbalance vector, $\boldsymbol{U}_k$, and the distance, $\boldsymbol{l}_k$, by using Formula (A.2).

$$\boldsymbol{P}_k = \boldsymbol{l}_k \times \boldsymbol{U}_k \quad \text{(A.2)}$$



NOTE Plane R is perpendicular to the shaft axis *z*.

Figure A.4 — Kinematic details of moment unbalance, created by $\boldsymbol{U}_k$

The resultant moment unbalance, $\boldsymbol{P}_r$, about the chosen plane, $\boldsymbol{z}_r$, from the origin, is equal to the vector sum of the moment unbalances of all individual unbalance vectors, $\boldsymbol{U}_k$, with the distance, $\boldsymbol{l}_k$, as expressed by Formula (A.3).

$$\boldsymbol{P}_r = \sum_{k=1}^{K} \boldsymbol{l}_k \times \boldsymbol{U}_k \qquad \text{(A.3)}$$

A.5 Couple unbalance

In some cases, it is more convenient to express the moment unbalance, $\boldsymbol{P}_k$, by the vector product of a couple unbalance (two unbalance vectors, $\boldsymbol{C}_k$ and $-\boldsymbol{C}_k$, with the related plane positions, $\boldsymbol{z}_C$ and $\boldsymbol{z}_{-C}$) as shown in Figure A.5. The planes for $\boldsymbol{C}_k$ and $-\boldsymbol{C}_k$ are chosen arbitrarily. The couple $\boldsymbol{C}_k$ is directly related to $\boldsymbol{P}_k$ by Formula (A.4).

$$\boldsymbol{P}_k = \boldsymbol{z}_C \times \boldsymbol{C}_k + \boldsymbol{z}_{-C} \times (-\boldsymbol{C}_k) \qquad \text{(A.4)}$$

NOTE 1 The couple unbalance consists of a pair of unbalance vectors, $\boldsymbol{C}_k$ and $-\boldsymbol{C}_k$, with equal magnitude, but opposite direction.

NOTE 2 Formula (A.4) can be rewritten as $\boldsymbol{P}_k = (\boldsymbol{z}_C - \boldsymbol{z}_{-C}) \times \boldsymbol{C}_k$. The modulus $\left|\boldsymbol{z}_C - \boldsymbol{z}_{-C}\right|$ is the distance *b* between the two couple unbalance vectors.

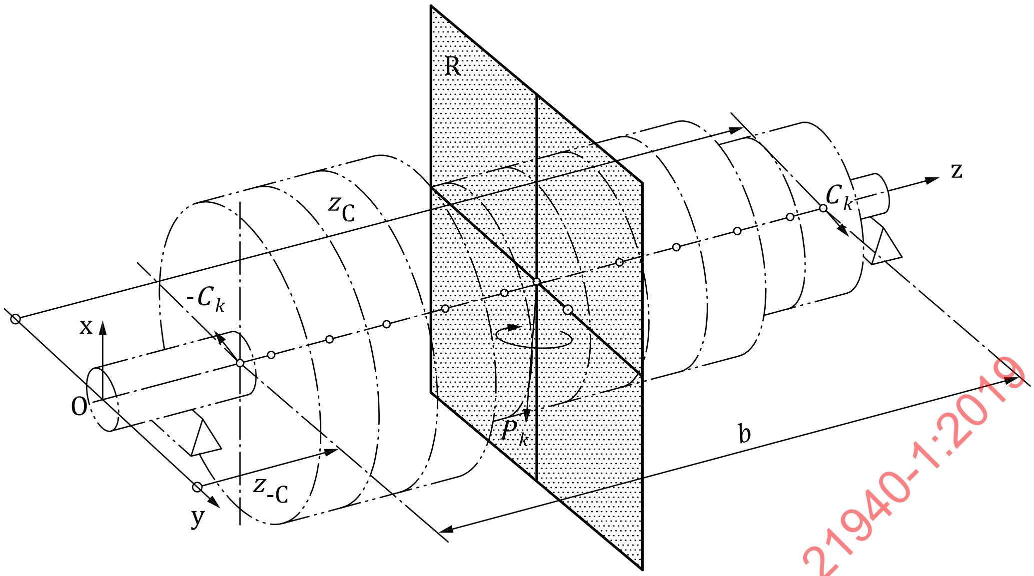
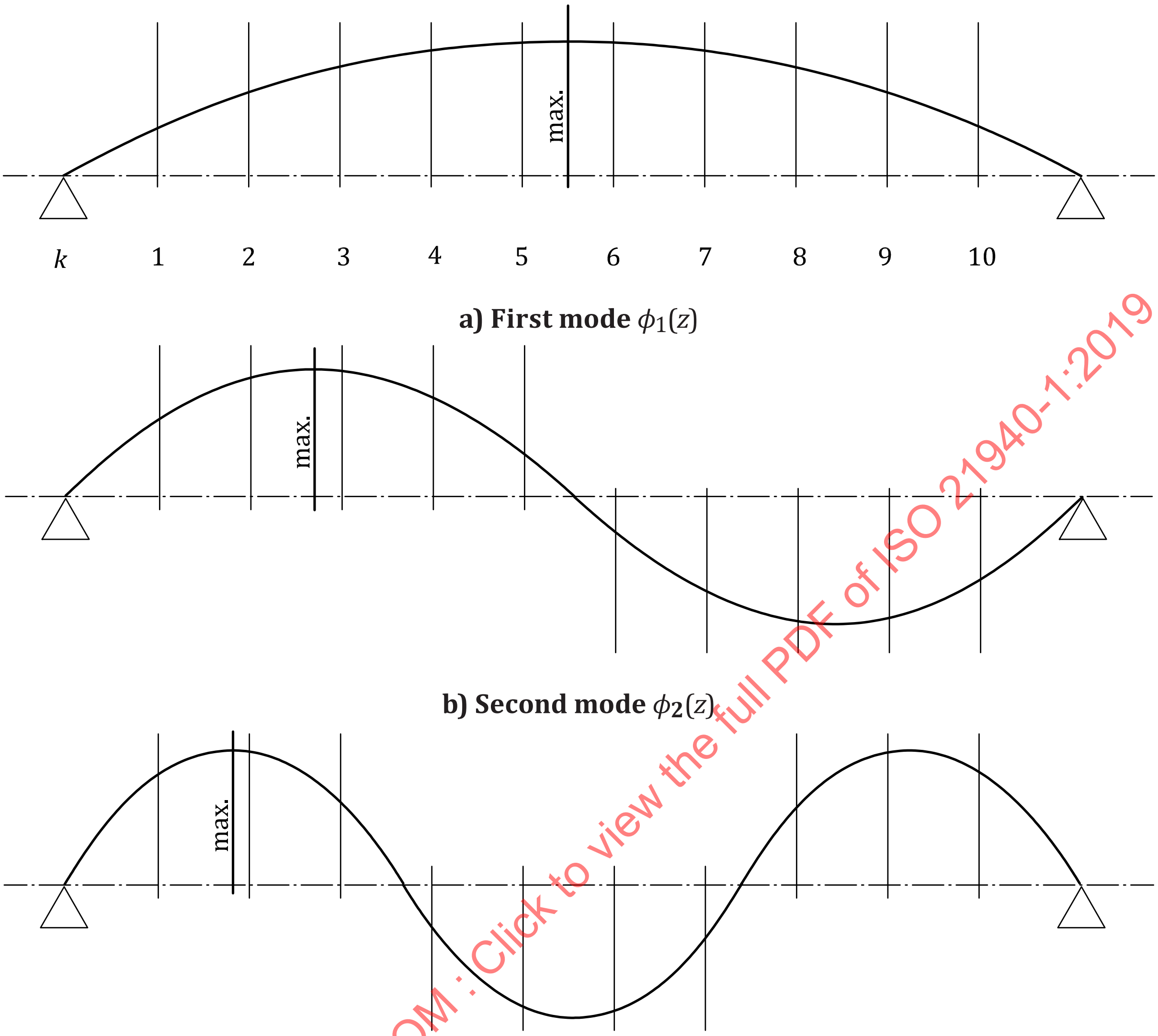


Figure A.5 — Process of determining the couple unbalance $-C_k$ and C_k from the moment unbalance P_k

A.6 Modal unbalances

A.6.1 Mode shapes

Modal unbalances become especially important when the operating speed is in the vicinity of a resonance speed of the rotor. Modal unbalance is based on bending deflections (mode shape functions $\phi_n(z)$, $n = 1, 2, 3, ...$) of the individual modes. Figure A.6 shows (idealized) modal deflections for the first three flexural modes of a symmetric rotor with equally distributed mass and stiffness and with no deflections at the bearings.



a) First mode $\phi_1(z)$

b) Second mode $\phi_2(z)$

c) Third mode $\phi_3(z)$

Figure A.6 — First three flexural mode shapes of a symmetric rotor with equally distributed mass and stiffness on rigid supports

A.6.2 Modal unbalance representation

Referring to the rotor model in Figure A.1, the individual unbalance vectors, $\boldsymbol{U}_k$, are weighted by the respective modal function $\phi_n(z_k)$. The unbalance distribution of the individual nth modal unbalance, $\boldsymbol{U}_{n,k}$, is expressed by Formula (A.5).

$$\boldsymbol{U}_{n,k} = \boldsymbol{U}_k\,\phi_n(z_k),\ k = 1, 2, 3, \ldots \tag{A.5}$$

The sum of these distributed unbalance vectors yields the resultant nth modal unbalance, $\boldsymbol{U}_{n,\mathrm{r}}$, which is given by Formula (A.6).

$$\boldsymbol{U}_{n,\mathrm{r}} = \sum_{k=1}^{K} \boldsymbol{U}_{n,k} \tag{A.6}$$

NOTE In case of plane bending modes, the ordinate values are just real numbers but for three-dimensional bending modes they can be complex numbers.

A clear distinction should be made between the individual nth modal unbalance as shown by Formula (A.5) and the resultant nth modal unbalance as shown by Formula (A.6).

A.6.3 Equivalent modal unbalance

Modal unbalances are not unique since different ways of scaling or normalizing the ordinate values of the mode exist. If the individual modal unbalances, $\boldsymbol{U}_{n,k}$, are divided by the maximum ordinate value of the mode, $\phi_{n,\max}$, it becomes the individual nth equivalent modal unbalance, $\boldsymbol{U}_{\mathrm{ne},k}$, acting at the plane of the chosen maximum ordinate.

A general mathematical representation for the equivalent modal unbalance, $\boldsymbol{U}_{\mathrm{ne},k}$, of the nth mode is given by Formula (A.7).

$$\boldsymbol{U}_{\mathrm{ne},k} = \boldsymbol{U}_k \frac{\phi_n(z_k)}{\phi_{n,\max}}, \; k = 1, 2, 3, \ldots \tag{A.7}$$

The sum of these vectors along the rotor is the resultant nth equivalent modal unbalance, $\boldsymbol{U}_{\mathrm{ne,r}}$, as given by Formula (A.8).

$$\boldsymbol{U}_{\mathrm{ne,r}} = \sum_{k=1}^{K} \boldsymbol{U}_{\mathrm{ne},k} \tag{A.8}$$

The equivalent modal unbalance (for different flexural modes) can be used to state rotor unbalances and rotor unbalance tolerances. However, it cannot be used to correct modal unbalances, since the single correction, although correct for the mode calculated, can have a detrimental effect for other modes.

NOTE 1 $\boldsymbol{U}_{\mathrm{ne,r}}$ is a single unbalance vector acting at the axial position of $\phi_{n,\max}$ with the same effect on the bending of the rotor in this mode as the combined effect of all the individual unbalances.

NOTE 2 If the equivalent modal unbalance was used as a test unbalance or an unbalance correction, it would also influence all other flexural modes, since it is a single unbalance vector.

Annex B
(informative)

Examples of different rotor behaviours as indicated on a typical hard-bearing balancing machine

B.1 General

Clause 5 describes four different types of rotor behaviour, rigid and flexible, together with the special conditions of component-elastic and settling component. This annex describes these behaviours in more detail observing how the rotor's response varies as a function of rotor speed for an unbalanced and balanced condition. In order to demonstrate these idealized behaviours, it is assumed that

a) the rotor is run in a balancing machine between a low speed n_1 (a typical balancing speed on a hard-bearing balancing machine) and high speed n_2;

b) the balancing machine bearing supports are sufficiently rigid such that there is no significant change in their dynamic characteristics;

c) the indication is calibrated over the speed range in terms of unbalances.

NOTE 1 Hard-bearing balancing machines are typically calibrated to indicate unbalance masses (or correction masses). For many rotors, the indication is constant within the working speed range of the machine, provided the unbalance does not change.

NOTE 2 In the case of a rotor with flexible behaviour, the unbalance indication changes with speed due to the influence of the flexural modes, even though the unbalances remain constant. Hence a change of unbalance indication can be due either to flexible rotor behaviour or to a change of unbalance.

NOTE 3 In special cases, hysteretic effects can cause a variation of the indication between a run-up and a run-down, but this is not shown here.

For the example presented in Clauses B.2 to B.5, the unbalance, as indicated on the balance machine and typically expressed in g·mm, is plotted as a function of rotor speed. In addition, a polar diagram is shown of the unbalance over the same speed range, where the unbalance amplitude is the radius and the unbalance indicated angle is the circumferential angle.

A simple unbalance tolerance limit for each case is indicated as a horizontal line on the linear graph and a circle on the polar diagram. However, at speeds close to the resonance speeds, the tolerances can be more complex, since they need to meet the permissible residual modal unbalance tolerance requirements.

B.2 Rigid behaviour

For idealized rotors with rigid behaviour, the unbalance response indication remains constant with speed (as shown in Figure B.1). Therefore, there is a negligible change in the unbalance indication over the speed range relative to the tolerance and the balanced rotor simply shows an improved level of unbalance.

