INTERNATIONAL STANDARD

ISO 19499

First edition
2007-07-15

Mechanical vibration — Balancing — Guidance on the use and application of balancing standards

Vibrations mécaniques — Équilibrage — Lignes directrices pour l'utilisation et l'application de normes d'équilibrage

Reference number
ISO 19499:2007(E)

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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.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

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.

ISO 19499 was prepared by Technical Committee ISO/TC 108, *Mechanical vibration, shock and condition monitoring*.

Introduction

Vibration caused by rotor unbalance is one of the most critical issues in the design and maintenance of machines. It gives rise to dynamic forces which adversely impact both machine and human health and well-being. The purpose of this International Standard is to provide a common framework for balancing rotors so that appropriate methods will be used. This standard serves essentially as guidance on the usage of other International Standards on balancing in that it categorizes types of machine unbalance. As such, it can be viewed as an introductory standard to the series of International Standards on balancing developed by ISO/TC 108.

Balancing is explained in a general manner, as well as the unbalance of a rotor. A certain representation of the unbalance is recommended for an easier understanding of the necessary unbalance corrections.

Mechanical vibration — Balancing — Guidance on the use and application of balancing standards

1 Scope

This International Standard provides an introduction to balancing and directs the user through the available International Standards associated with rotor balancing. It gives guidance on which of these standards should be used. Individual procedures are not included here as these will be found in the appropriate International Standards.

2 Normative references

The following referenced documents are indispensable for the application 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 1925:2001, *Mechanical vibration — Balancing — Vocabulary*

ISO 2041, *Vibration and shock — Vocabulary*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 1925 and ISO 2041 apply.

4 Fundamentals of balancing

4.1 General

Balancing is a procedure by which the mass distribution of a rotor (or part or module) is checked and, if necessary, adjusted to ensure that balance tolerances are met.

Rotor unbalance may be caused by many factors, including material, manufacture and assembly, wear during operation, debris or an operational event. It is important to understand that every rotor, even in series production, has an individual unbalance distribution.

New rotors are commonly balanced by the manufacturer in specially designed balancing machines before installation into their operational environment. Following rework or repair, rotors may be rebalanced in a balancing machine or, if appropriate facilities are not available, the rotor may be balanced *in situ* (see ISO 20806 for details). In the latter case, the rotor is held in its normal service bearings and support structure and installed within its operational drive train.

The unbalance on the rotor generates centrifugal forces when it is rotated in a balancing machine or *in situ*. These forces may be directly measured by force gauges mounted on the structures supporting the bearings or indirectly by measuring either the motion of the pedestal or the shaft. From these measurements, the unbalance can be calculated and balancing achieved by adding, removing or shifting of correction masses on the rotor. Depending on the particular balancing task, the corrections are performed in one, two or more correction planes.

4.2 Unbalance distribution

In reality, unbalance is made up of an infinite number of unbalance vectors, distributed along the shaft axis of the rotor. If a lumped-mass model is used to represent the rotor, unbalance may be represented by a finite number of unbalance vectors of different magnitude and angular direction as illustrated in Figure 1.

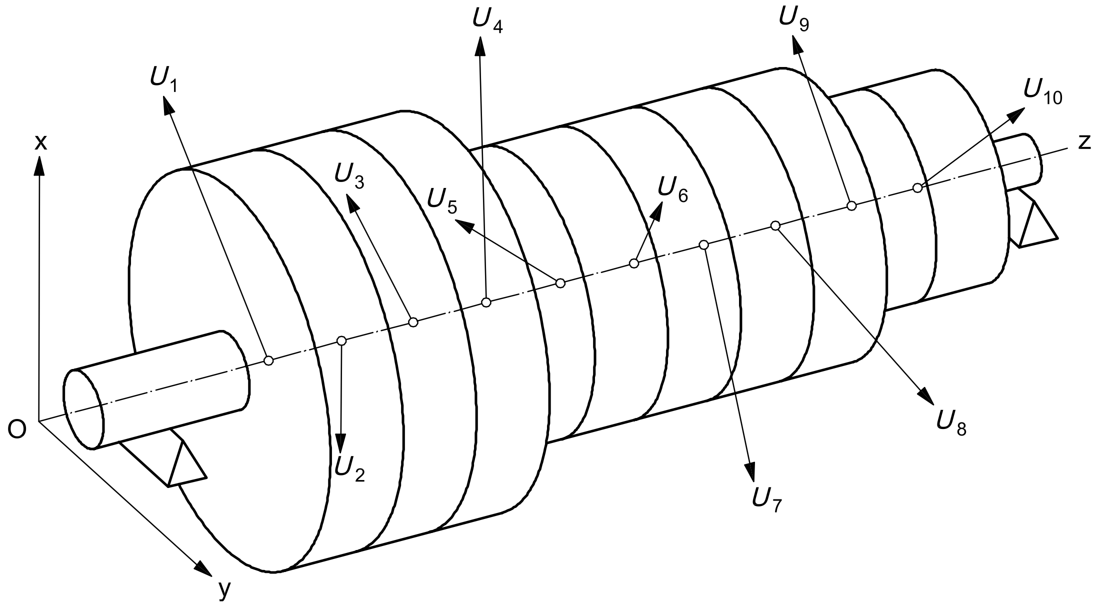


Figure 1 — Unbalance distribution in a rotor modelled as 10 elements perpendicular to the z-axis

If all unbalance vectors were corrected in their respective planes, then the rotor would be perfectly balanced. In practice, it is not possible to measure these individual unbalances and it is not necessary. A more condensed description is needed, leading to practical balancing procedures.

4.3 Unbalance representation

Rotor unbalance can be expressed by a combination of the following three kinds of unbalance representations:

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

b) resultant moment unbalance, $\vec{P}_{\mathrm{r}}$, the vector sum of the moments of all the unbalance vectors distributed along the rotor about the arbitrarily selected plane of the resultant unbalance;

c) modal unbalance, $\vec{U}_n$, that unbalance distribution which affects only the nth natural mode of a rotor/bearing system.

Mathematical and graphical representations of unbalances are shown in Annex A.

NOTE Resultant unbalance [see 4.3 a)] and resultant moment unbalance [see 4.3 b)] can be combined. The combination is called “dynamic unbalance” and is represented by two unbalances in two arbitrarily chosen planes perpendicular to the shaft axis.

5 Balancing considerations

5.1 General

In the past, International Standards classified all rotors to be either rigid or flexible, and balancing procedures for these two main classes of rotors are given in ISO 1940-1 and ISO 11342, respectively (see Table 1). However, the simple rigid/flexible classification is a gross simplification, which can lead to a misinterpretation and suggests that the balance classification of the rotor is only dependent on its physical construction. Unbalance is an intrinsic property of the rotor, but the behaviour of the rotor and its response to unbalance in its normal operating environment are affected by the dynamics of the bearings and support structure, and by its operating speed. Furthermore, the balance quality to which the rotor is expected to run and the magnitude and distribution of the initial unbalance along the rotor will dictate which balancing procedure is necessary; see Table 1.

Table 1 — Overview of rotor behaviour, related International Standards and balancing procedures

Rotor behaviour (Numbers refer to subclauses in this International Standard)	**Example**	**Related International Standard**	**Balancing task or procedure** (Letters as used in ISO 11342:1998)
Rigid behaviour (5.2)	Figure 4 a)	ISO 1940-1	One- and two-plane balancing [a]
Flexible behaviour (5.3)	Figure 4 b)	ISO 11342 [b]	Six low-speed balancing procedures (A to F) Balancing procedure at multiple speeds (G) Balancing procedure for one speed only (usually service speed) (H)
Component elastic behaviour (5.4.2)	Figure 4 c)		Fixed-speed balancing procedure (I)
Component seating behaviour (5.4.3)	Figure 4 d)		Settling of components at high speed [c]

[a] One- and two-plane balancing includes balancing the resultant unbalance and the resultant moment unbalance.

[b] ISO 11342:1998 uses “flexible” as a generic term that includes flexible, component elastic and component seating behaviours.

[c] This procedure is mentioned in Clause 7 of ISO 11342:1998, but no designated letter is given.

5.2 Rotors with rigid behaviour

An ideal rotor when rotating, with rigid behaviour on elastic supports, will undergo displacements that are combinations of the two dynamic rigid-body modes, as seen in Figure 2 for a simple symmetric rotor with unbalance. There is no flexure of the rotor and all displacements of the rotor arise from movements of the bearings and their support structure.

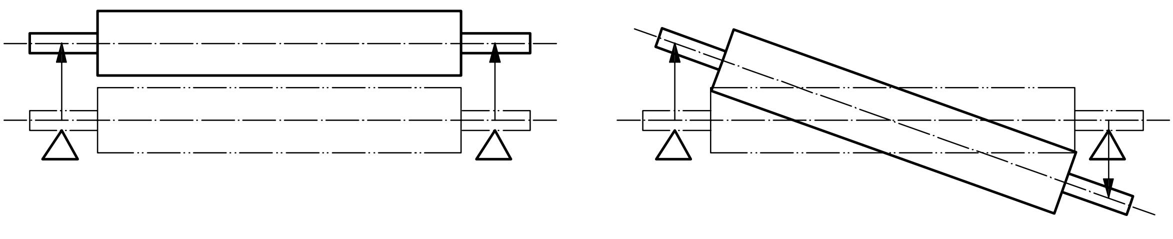


Figure 2 — Rigid-body modes of a symmetric rotor on a symmetric elastic support structure

In reality, no rotor will be totally rigid and will have small flexural deflections in relation to the gross rigid-body motion of the rotor. However, the rotor may be regarded as rigid provided these deflections caused by a given unbalance distribution are below the required tolerances at any speed up to the maximum service speed. The majority of such rotors, and indeed many manufactured rotors, can be balanced as rigid rotors, in accordance with the requirements of ISO 1940-1. This aims at balancing the resultant unbalance with at least a single-plane balance correction, or the dynamic unbalance with a two-plane balance correction.

NOTE 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 still operates with a rigid behaviour.

5.3 Rotors with flexible behaviour

5.3.1 General

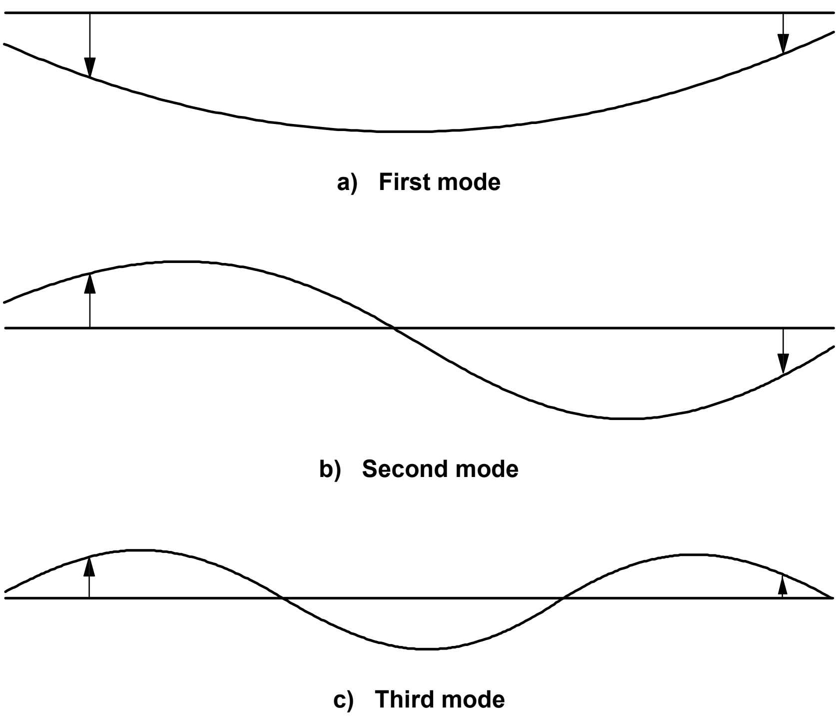
If the speed is increased or the tolerance reduced for the same rotor described in 5.2, it may become necessary to take flexible behaviour into account. Here the deflection of the rotor is significant, and rigid-body balancing procedures are not sufficient to achieve a desired balance condition. Figure 3 shows typical flexural mode shapes for a symmetric rotor. For these rotors that exhibit flexural behaviour, the balancing procedures in ISO 11342 should be adopted.

5.3.2 Low-speed balancing

In special circumstances, even rotors with flexible behaviour may be balanced satisfactorily at low speed. ISO 11342:1998 describes procedures A to F, which, as far as possible, all aim to correct the unbalance in their planes of origin.

5.3.3 Multiple-speed balancing

This procedure should be used to balance the resultant unbalance, the resultant moment unbalance and the relevant modal unbalances, according to ISO 11342:1998, procedure G.



a) First mode

b) Second mode

c) Third mode

Figure 3 — Schematic representation of the first three flexural modes of a rotor with flexible behaviour on an elastic support structure

5.3.4 Service speed balancing

These rotors are flexible and pass through one or more critical speeds on their way up to service speed. However, due to operating conditions or machine construction, high levels of vibration can be tolerated at the critical speeds and the rotor is only balanced at the service speed according to ISO 11342:1998, procedure H.

5.4 Rotors with special behaviour

5.4.1 General

The majority of rotors will exhibit either rigid or flexible behaviours, but the following special behaviours can exist and must be considered to achieve a successful balancing of the rotor.

5.4.2 Elastic behaviour of components

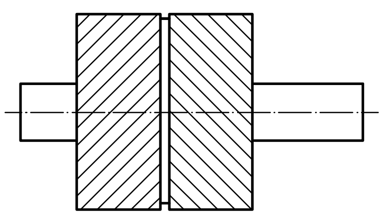
These rotors can have a shaft and body construction that either requires low-speed or high-speed balancing procedures. However, in addition, they have one or more components that themselves are either flexible or are flexibly mounted so that the unbalance of the whole system might consistently change with speed. Examples of such rotors are a rotor with tie bars that deflect at high speed, rubber-bladed fans and single-phase induction motors with a centrifugal switch. These should be balanced in accordance with ISO 11342:1998, procedure I.

5.4.3 Seating behaviour of components

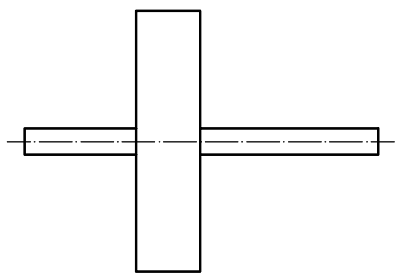
These rotors can have a construction where components settle after reaching a certain speed or other condition. This movement will then become stable after one or just a few events. The components will reach a final position and become re-seated, after which the rotor may require further balancing. Examples are shrunk-on turbine discs, built-up rotors, copper winding in generators and generator retaining rings. Subsequent behaviour of the rotor will then dictate the balancing procedure required. This rotor behaviour is mentioned in Clause 7 of ISO 11342:1998, but no procedure is specified.

5.5 Examples of rotor behaviours

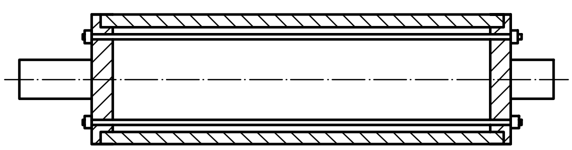
The different behaviours may be represented by the following rotors (see Figure 4):



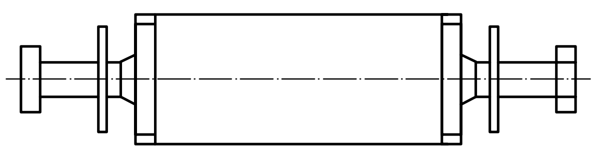
a) Rigid behaviour (a solid gear wheel)



b) Flexible behaviour (a disc on an elastic shaft, for example a Laval rotor)



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



d) Seating behaviour (a generator rotor with windings, once for all seating under a certain centrifugal load)

Figure 4 — Examples of rotor types that demonstrate particular rotor behaviours

Such illustrations are not sufficient for a full description: obviously, rotors such as those shown in Figures 4 c) and 4 d) may also show flexible behaviour; on the contrary, a rotor such as Figure 4 b) could be a low-speed fan without considerable flexible behaviour, i.e. a rotor with a rigid behaviour. Details of these types of behaviour are further explained in Annex B.

5.6 Influencing factors

5.6.1 General

A rotor's response characteristic is defined by its physical properties and those of its supporting structure. The vibration response measured at the supporting structure or on the rotor will depend on these physical properties plus the magnitude of unbalance and its distribution along the rotor, as well as the speed of rotation. Balancing to a required tolerance will therefore depend on all these parameters, and changing these or the tolerance specified may change the procedure needed to meet that tolerance.

5.6.2 Tolerances

By simply reducing the balance tolerance, it may be necessary to reconsider the behaviour of the rotor and adopt a different procedure to bring the rotor within tolerance, as given in the following examples.

a) A rotor with rigid behaviour, balanced using a single-plane procedure to reduce resultant unbalance, may simply require additional more accurate single-plane balancing.

b) A rotor with rigid behaviour, balanced using a single-plane procedure to reduce resultant unbalance, may also require a two-plane procedure to take into account the moment unbalance (or both resultant and moment unbalance together as a dynamic unbalance).

c) A rotor with rigid behaviour, balanced in two planes to reduce both resultant and moment unbalance (or both resultant and moment unbalance together as a dynamic unbalance), may additionally require flexible behaviour procedures to reduce contributions from the modal unbalances, even though the rotor is running below its first flexural critical speed.

d) A rotor with flexible behaviour, for which a flexible rotor behaviour procedure has been carried out to reduce the dynamic unbalance and a number of modal unbalances, may require additional flexible rotor behaviour procedures to reduce modal unbalances of even more (higher) flexural modes of the rotor, even though the rotor is running below the flexural critical speeds of the higher modes.

e) A rotor with either rigid or flexible behaviour, successfully balanced using the appropriate procedure, may need to consider special procedures to take into account component elastic or component seating behaviours.

f) Where a tighter tolerance can only be achieved at a single speed, the service speed balancing procedure may need to be considered.

5.6.3 Speed and support conditions

Other changes of rotor behaviour may occur if operational conditions are changed (e.g. by changing speed or support conditions).

5.6.4 Initial unbalance

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

6 Balance tolerances

6.1 General

The balancing equipment and techniques available enable unbalances to be reduced to low limits. However, it would be uneconomic to over-specify the quality requirements. It is therefore necessary to define the optimum balance quality for the rotor to operate with acceptable vibration and dynamic forces in its normal service environment.

6.2 Permissible residual unbalances

There is a direct relation between rotor unbalance and the once-per-revolution vibration under service conditions. The relationship depends on the machine's dynamic characteristics (i.e. rotor, structure and bearing dynamic properties). However, the overall machine vibration may be due only in part to the presence of rotor unbalance. Other sources of vibration could be magnetic or fluid forces.

Guidance on the derivation of permissible residual unbalance tolerances is given in

— ISO 1940-1 for a rigid rotor behaviour, and

— ISO 11342, using tolerance data from ISO 1940-1, for other rotor behaviours.

6.3 Vibration limits

There is no easily recognizable relationship between the machine vibration under service conditions and vibration in the balancing machine. The relation depends on the differences between the bearings and the support structure used in the balancing machine and installed condition. Further, the rotor in the balancing machine is tested in isolation and does not include effects from other rotors in the shaft line, as experienced when installed in its operational environment. It should be noted that different balancing machines may have different pedestal stiffness and therefore vibration limits have to be set individually for each balancing machine.

Where detailed information is available concerning these parameters, a method to estimate these limits is presented in ISO 11342. *In-situ* vibration limits are presented in the appropriate parts of ISO 7919 for rotating shafts and ISO 10816 for non-rotating parts.

Where insufficient detail is available to obtain these parameters, guidance should be taken from the balancing machine facility from which rotors have operated satisfactorily *in situ*.

7 Selection of a balancing procedure

7.1 General

Since different balancing procedures require different types of balancing machines and resource input, it is important to select an appropriate procedure (see Table 1) to optimize the balancing process to meet the required balance tolerances.

Rotors with a **rigid behaviour** (see 5.2) can be balanced using the single- and two-plane balancing guidelines provided in ISO 1940-1.

In general, rotors with a **flexible behaviour** (see 5.3) should be dealt with in accordance with the guidelines given in ISO 11342, where a number of procedures are defined for different rotor configurations:

— the general procedure is that for multiple-speed balancing, see 5.3.3 (procedure G of ISO 11342:1998);

— rotors that spend most of their time at a single speed can use service speed balancing (see 5.3.4; (procedure H of ISO 11342:1998);

— rotors that have flexible behaviour that can be adequately balanced at low speed (see 5.3.2; procedures A to F of ISO 11342:1998).

The procedure for rotors with **component elastic behaviour**, see 5.4.2, is given in ISO 11342:1998 (procedure I).

For rotors with **component seating behaviour** (see 5.4.3), it is first necessary for the rotor to be taken to the speed (or condition) at which the unbalance is stabilized and then balanced using the appropriate procedure for the rigid rotor behaviour (see ISO 1940-1) or flexible rotor behaviour (see ISO 11342).

Where possible, consult the rotor manufacturer or the user for a recommended balancing procedure. In the case of the rotor with flexible behaviour, a definition of the rotor configuration, as defined by ISO 11342, would suffice.

7.2 Selection of a balancing procedure when none is specified

7.2.1 Identify the rotor behaviour

Clause 5 introduces the major types of rotor behaviour and Table 2 gives guidance on selection of the balancing procedure together with the expected balancing machine to be used. A consideration of the balance tolerances must be taken into account when identifying the rotor behaviour, as discussed in 7.2.2.

WARNING — Guidance given in Table 2 should be used with care as the ratio of rotor speed to first flexural resonance *in situ* is only a typical value. This will be highly dependent on the dynamic characteristics of the bearings and their supports, initial unbalance, the balance quality required, and the detailed design of the rotor.

Although the behaviour of many rotors can be easily identified from the simple guidance given in both Clause 5 and Table 2, others will require a full analysis, particularly when making the distinction between rigid and flexible behaviours. Table 2 requires knowledge of the critical speeds, which are often unknown. Annex C presents an analytical method to obtain estimates of rotor flexibility in the operating environment.

Table 2 — Guidelines for balancing procedures

Guidance given in this table should be used with care as the ratio of rotor speed to first flexural critical speed *in situ* is only a typical value. This will be highly dependent on the dynamic characteristics of the bearings and their supports, initial unbalance, the balance quality required and the detailed design of the rotor.

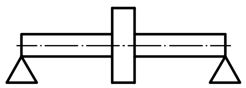
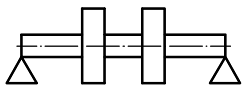
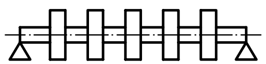
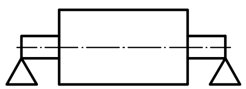
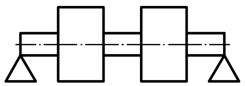
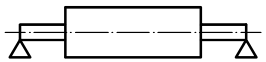
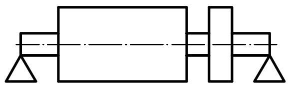
Schematic rotor representation	Rotor description		Rotor speed/first flexural critical speed *in situ*	Balancing process and International Standard
	Any rotor configuration		$< 0{,}30$	Single- or two-plane balancing ISO 1940-1
	Single rigid disc installed on an elastic shaft whose unbalance is negligible	Disc perpendicular to shaft axis	All values	Single-plane balancing ISO 1940-1
		Disc with axial runout	All values	Two-plane balancing ISO 1940-1
	Two rigid discs installed on an elastic shaft whose unbalance is negligible	Both discs perpendicular to shaft axis	All values	Two-plane balancing ISO 1940-1
		One or both disc(s) with axial runout	$< 0{,}7$	Two-plane balancing ISO 1940-1
			$\geqslant 0{,}7$	Multiple-speed balancing ISO 11342:1998, procedure G
	More than two rigid discs installed on an elastic shaft whose unbalance is negligible		$< 0{,}7$	Two-plane balancing ISO 1940-1
			$\geqslant 0{,}7$	Multiple-speed balancing ISO 11342:1998, procedure G
	Single rigid section installed on an elastic shaft whose unbalance is negligible		All values	Two-plane balancing ISO 1940-1

Table 2 (*continued*)

Guidance given in this table should be used with care as the ratio of rotor speed to first flexural critical speed *in situ* is only a typical value. This will be highly dependent on the dynamic characteristics of the bearings and their supports, the initial unbalance, the balance quality and the detailed design of the rotor.

Schematic rotor representation	Rotor description		Rotor speed/first flexural critical speed *in situ*	Balancing process and International Standard
	Two or more rigid sections installed on an elastic shaft whose unbalance is negligible		< 0,7	Two-plane balancing ISO 1940-1
			⩾ 0,7	Multiple-speed balancing ISO 11342:1998, procedure G
	Cylindrical roll	Assumed unbalance distribution according to machining tolerances (generally uniform unbalance distribution)	< 0,6	Two-plane balancing ISO 1940-1
			⩾ 0,6	Two-plane balancing in two optimum correction planes ISO 11342:1998, procedure F
		No known unbalance distribution	< 0,6	Two-plane balancing ISO 1940-1
			⩾ 0,6	Multiple-speed balancing ISO 11342:1998, procedure G
	Rotor including roll(s) and rigid sections and/or discs Integral rotors		< 0,6	Two-plane balancing ISO 1940-1
			⩾ 0,6	Multiple-speed balancing ISO 11342:1998, procedure G
	Any other configuration		No factor available	ISO 11342

NOTE 1 Factors stated are based on typical balance tolerances with limited initial unbalance.

NOTE 2 These guidelines apply to assembled rotors where the individual components have not been balanced prior to assembly.

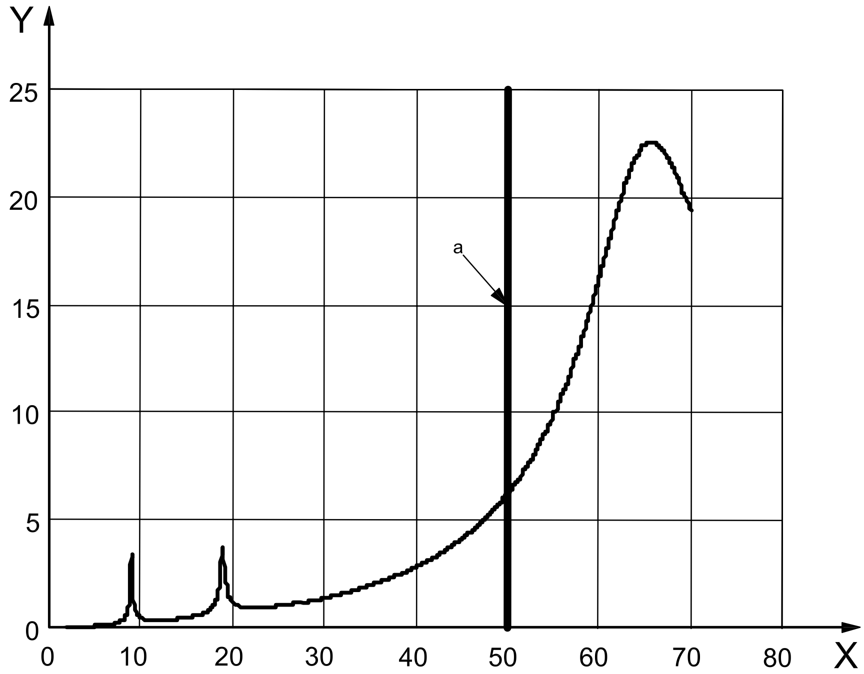
NOTE 3 In the fourth column of the table, it is the first flexural critical speed of the rotor that is considered, not the rigid-body critical speeds.

NOTE 4 There are examples where the first critical speed observed is associated with rigid-body modes of the rotor. This should not be confused with the first flexural.

7.2.2 Select the required balance tolerances for the rotor

Select the balance tolerance as recommended in Clause 6.

When low balancing tolerances or response levels are required, it may be necessary to consider shaft flexural modes that occur at frequencies above the operating speed range of the rotor. For example, the rotor response shown in Figure 5 is significantly affected at the service speed (50 Hz) by the higher first flexural mode, even though the first two (rigid-body) modes have been balanced and are at a much lower level. Here a machine would fail a balance tolerance based on r.m.s. vibration of 5 mm/s at a normal service speed of 50 Hz due to the higher mode, even though the lower modes would be acceptable. The level of excitation of this higher mode and the balance tolerance required will determine the rotor behaviour and balancing procedure adopted. In this case, the influence of the higher mode will require a flexible rotor-balancing procedure to be used to achieve the required balancing tolerance at the operational speed.



Key

X frequency, Hz

Y r.m.s. vibration level, mm/s

[a] Operating speed of 50 Hz.

Figure 5 — Influence of rotor modes above service speed

7.2.3 Select the appropriate balancing procedure

An introduction to the available balancing procedures is presented in Table 2. For details, see the appropriate International Standards.

7.2.4 Choose the appropriate balancing machine

Rotor behaviour, together with the chosen balancing procedure, will dictate whether a low- or high-speed balancing machine is required. Examples of balancing machine requirements are provided in Table 3 and these are covered in more detail in ISO 2953.

The different types of unbalance call for different types of balancing machines, as follows:

a) resultant unbalance:

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

b) resultant moment unbalance:

a two-plane balancing machine (low-speed) is needed;

c) modal unbalances:

a high-speed balancing machine will generally be 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 single-plane and two-plane balancing.

NOTE 3 Flexible rotors classified by procedures A to F of ISO 11342:1998 can be adequately balanced at low speed.

Table 3 — Examples of 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 test facility
NOTE This list is not exhaustive and other requirements could be appropriate for special machines.	

7.2.5 Selecting specialized rotor requirements

While the rotor is in the balancing machine, additional tests may be undertaken to ensure the rotor is fit for purpose. Table 4 gives examples of additional tests that could be performed on a large electrical generator rotor while in the balancing machine. Similarly, the need for additional tests should be considered for other specialized rotors whilst in the balancing machine.

Table 4 — Example of tests that may be undertaken in the balancing machine for an electrical generator rotor

1	Over speed the rotor to settle the end rings
2	Balance the rotor while passing through the shaft critical speeds, service speed and during over speed
3	Undertake thermal stability checks
4	Perform electrical tests to check the integrity of the windings
NOTE These tests should only be carried out provided the appropriate facilities are available and safety requirements are satisfied.	

8 International Standards on balancing

8.1 General

A suite of International Standards is available to aid the user in the field of balancing; see Table 5. These fall into six main areas: introduction (this International Standard), vocabulary, balancing procedures and tolerances, balancing machines, machine design for balancing, and machine vibration.

Table 5 — The suite of International Standards on balancing

Topic	International Standard		
Introduction	Introduction ISO 19499		
Vocabulary	Balancing vocabulary ISO 1925	Vibration and shock vocabulary ISO 2041	
Balancing procedures and tolerances	Rigid behaviour ISO 1940-1 Balance tolerances and balance errors ISO 1940-2	Flexible behaviour ISO 11342	*In-situ* balancing ISO 20806
Balancing machines	Description and evaluation ISO 2953	Symbols and instrumentation ISO 3719	Enclosures and protective measures ISO 7475
Machine design for balancing	Susceptibility and sensitivity of machines to unbalance ISO 10814	Shaft and fitment key convention ISO 8821	
Machine vibration	Evaluation of machine vibration by measurements on non-rotating parts ISO 10816 (all parts)	Evaluation of machine vibration by measurements on rotating shafts ISO 7919 (all parts)	

8.2 Vocabulary

8.2.1 ISO 1925, *Mechanical vibration — Balancing — Vocabulary*

This International Standard establishes a vocabulary on balancing in English and French. An alphabetical index is provided for each of the two languages.

Annex A of ISO 1925:2001 gives an illustrated guide to balancing machines terminology and includes equivalent terms in English, French and German.

8.2.2 ISO 2041, *Vibration and shock — Vocabulary*

A general vocabulary on vibration and shock, in English and French, is given in ISO 2041. An alphabetical index is provided for each of the two languages.

8.3 Balancing procedures and tolerances

8.3.1 General

These International Standards are not intended to serve as an acceptance specification for any rotor, but rather to give indications of how to avoid gross deficiencies and/or unnecessarily restrictive requirements.

8.3.2 ISO 1940-1, *Mechanical vibration — Balance quality requirements of rigid rotors in a constant (rigid) state — Part 1: Specification and verification of balance tolerances*

ISO 1940-1 gives recommendations for determining unbalance and for specifying related quality requirements of rigid rotors. It specifies

a) a representation of unbalance in one and two planes,

b) methods for determining permissible residual unbalance,

c) methods for allocating permissible residual unbalance to correction planes,

d) methods for identifying the residual unbalance state of a rotor by measurement, and

e) a summary of errors associated with the residual unbalance identification.

8.3.3 ISO 1940-2, *Mechanical vibration — Balance quality requirements of rigid rotors — Part 2: Balance errors*

ISO 1940-2 covers

a) identification of errors in the balancing process of rigid rotors,

b) the assessment of errors,

c) guidelines for taking errors into account, and

d) the evaluation of residual unbalance in any two correction planes.

Many of the errors associated with balancing of rotors with a rigid behaviour, as outlined in ISO 1940-2, apply equally to balancing of rotors that have a flexible behaviour.

8.3.4 ISO 11342, *Mechanical vibration — Methods and criteria for the mechanical balancing of flexible rotors*

ISO 11342 presents typical flexible rotor configurations according to their characteristics and balancing requirements, describes balancing procedures, specifies methods of assessment of the final state of unbalance, and gives guidance on balance quality criteria in terms of both vibration and residual unbalance. It includes low-speed balancing procedures which can be applied successfully for some rotors with a flexible behaviour.

ISO 11342 may also be applicable to serve as a basis for more involved investigations, for example when a more exact determination of the required balance quality is necessary. If due regard is paid to the specified method of manufacture and limits of unbalance, satisfactory running conditions can probably be expected.

The influence of structural resonances is outside the scope of ISO 11342 but is covered by ISO 10814 (see 8.5.2).

8.3.5 ISO 20806, *Mechanical vibration — Criteria and safeguards for the* in-situ *balancing of medium and large rotors*

ISO 20806 specifies procedures be adopted when balancing 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 standard may be used as a basis for a contract to undertake *in-situ* balancing.

ISO 20806 does not provide guidance on the methods used to calculate the correction masses from measured vibration data.

8.4 Balancing machines

8.4.1 ISO 2953, *Mechanical vibration — Balancing machines — Description and evaluation*

ISO 2953 gives requirements for the evaluation of performance and characteristics of machines for balancing rotating components where correction is required in one or more planes. It stresses the importance attached to the form in which the balancing machine characteristics should be specified by the manufacturer, and also outlines methods of evaluating balancing machines.

8.4.2 ISO 3719, *Mechanical vibration — Symbols for balancing machines and associated instrumentation*

ISO 3719 establishes symbols for use on balancing machines, including instrumentation. They are intended to complement (but not replace) those already standardized in documents such as ISO 7000. The primary purpose of symbols in ISO 3719 is to explain the functions and uses of the indicators and controls, etc. which are an integral part of a balancing machine.

8.4.3 ISO 7475, *Mechanical vibration — Balancing machines — Enclosures and other protective measures for the measuring station*

This International Standard specifies requirements for enclosures and other safety measures used to minimize hazards associated with the operation of balancing machines. It defines different classes of protection that enclosures and other protective features have to provide, and describes the limits of applicability for each class of protection.

Special enclosure features, such as noise reduction, windage reduction or vacuum (which is required to spin some rotors at balancing speed), are not covered by ISO 7475.

8.5 Machine design for balancing

8.5.1 ISO 8821, *Mechanical vibration — Balancing — Shaft and fitment key convention*

ISO 8821 defines the convention for balancing the individual components of a keyed assembly. It is intended to provide compatibility of all balanced components, so that when they are assembled they will meet the overall balance or vibration tolerance levels for the assembled rotor.

8.5.2 ISO 10814, *Mechanical vibration — Susceptibility and sensitivity of machines to unbalance*

ISO 10814 defines methods for determining machine vibration sensitivity to unbalance and provides evaluation guidelines as a function of the proximity of relevant resonant speeds to the service speed.

It includes a classification of certain machines in groups associated with the susceptibility to a change of unbalance and/or other machine parameters.

ISO 10814 also makes recommendations on how to apply the numerical sensitivity values in some particular cases. The recommended values that are shown are intended to be applied only to relatively simple systems and not, for example, to large power-generation equipment.

8.6 Machine vibration

8.6.1 ISO 7919 (all parts), *Mechanical vibration — Evaluation of machine vibration by measurements on rotating shafts*

ISO 7919-1 sets out general guidelines for measuring and evaluating machinery vibration by means of measurements made directly on the rotating shaft for the purpose of determining shaft vibration with regard to

a) changes in vibration behaviour,

b) excessive kinetic load, and

c) the monitoring of radial clearances.

The subsequent parts of ISO 7919 provide specific guidance for assessing the severity of vibration measurement on different classes of machines, and give figures of vibration limits.

ISO 7919 is applicable to measurements of absolute and relative radial shaft vibration. The procedures are applicable to the operational/condition monitoring of machines, and to acceptance testing on the test stand and after installation. Guidelines are also presented for establishing operational limits.

8.6.2 ISO 10816 (all parts), *Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts*

ISO 10816-1 establishes general conditions and procedures for the measurement and evaluation of vibration using measurements made on non-rotating and, where applicable, non-reciprocating parts of complete machines. The subsequent parts provide specific guidance for assessing the severity of vibration measurement on different classes of machines, and give figures of vibration limits.

The general evaluation criteria, which are presented in terms of both vibration magnitude and change of vibration magnitude, are related to both condition monitoring and acceptance testing. They have been provided primarily with regard to securing reliable, safe, long-term operation of a machine, while minimizing adverse effects on associated equipment. Guidelines are also presented for setting operational limits. The evaluation criteria relate only to the vibration produced by the machine itself and not to vibration transmitted to it from outside.

Annex A
(informative)

Mathematical and graphical representation of unbalance

A.1 Objective

This annex shows how a set of unbalance vectors, distributed along the rotor (see Figure A.1), contributes to

— resultant unbalance,

— resultant moment unbalance or the couple unbalance, and

— modal unbalances of the mode shapes of the rotor.

NOTE A balancing machine does not show the distributed unbalances, but shows the necessary unbalance corrections in terms of resultant unbalance and couple unbalance, or dynamic unbalance and (in the case of a high-speed balancing machine) even of modal unbalances.

A.2 Notation

$\vec{U}_k$	unbalance vector in an individual plane (g·mm)
$\vec{U}_\mathrm{r}$	resultant unbalance (g·mm)
$\vec{P}_k$	moment unbalance of an unbalance $\vec{U}_k$ (g·mm^2)
$\vec{P}_\mathrm{r}$	resultant moment unbalance (g·mm^2)
$\vec{C}_k , -\vec{C}_k$	both unbalance vectors of the couple unbalance (g·mm), based on moment unbalance $\vec{P}_k$
$\vec{C}_\mathrm{r} , -\vec{C}_\mathrm{r}$	both unbalance vectors of a resultant couple unbalance (g·mm), based on resultant moment unbalance $\vec{P}_\mathrm{r}$
$\vec{U}_{n,k}$	individual nth modal unbalance in plane k
$\vec{U}_{n,\mathrm{r}}$	resultant nth modal unbalance
$\vec{U}_{n\mathrm{e},k}$	individual nth equivalent modal unbalance, based on unbalance in plane k (g·mm)
$\vec{U}_{n\mathrm{e,r}}$	resultant nth equivalent modal unbalance (g·mm)
z	coordinate of rotor axis
$\vec{z}_k$	distance from a datum mark O to the plane of $\vec{U}_k$ (mm)
$\vec{z}_r$	distance from a datum mark O to the chosen plane of the resultant unbalance $\vec{U}_\mathrm{r}$ (mm)
$\vec{z}_C$, $\vec{z}_{-C}$	distances of both couple unbalance vectors from a datum mark O (mm)
$\vec{l}_k = (\vec{z}_k - \vec{z}_r)$	distance from the chosen plane of the resultant unbalance $\vec{U}_\mathrm{r}$ to the plane of the unbalance $\vec{U}_k$ (mm).

O	datum mark for coordinates x, y, z
b	distance between the two couple unbalance vectors (mm)
k	counter for the planes, from 1 to K
n	counter for mode functions 1, 2, 3, ...
$\phi_n(z)$	mode function, n = 1, 2, 3, ...
$\phi_n(z_k)$	ordinate of nth mode function at plane k
$\phi_{n,\max}$	maximum ordinate of nth mode function

A.3 Set of unbalance vectors

Figure A.1 shows a rotor consisting of K = 10 elements with an unbalance vector $\vec{U}_k$ (k = 1, 2, ...K) at each element.

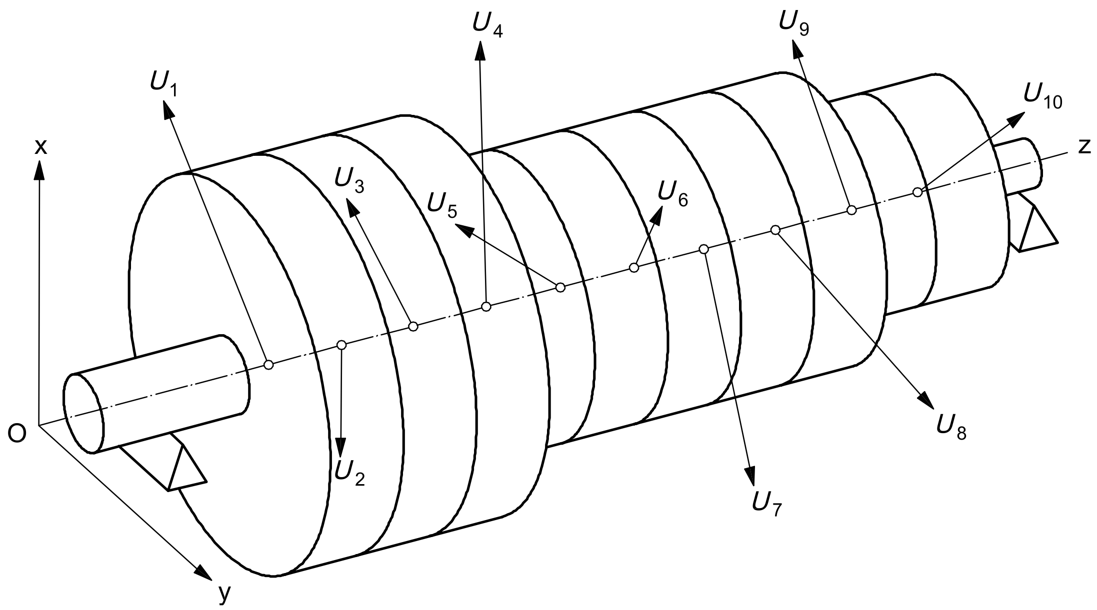


Figure A.1 — Unbalance distribution in the rotor modelled as 10 elements perpendicular to the z-axis

For a mathematical representation and evaluation, each of the unbalance vectors can be expressed, in a more general manner, by an unbalance vector $\vec{U}_k$ at a position $\vec{z}_k$ from a datum mark O (see Figure A.2).

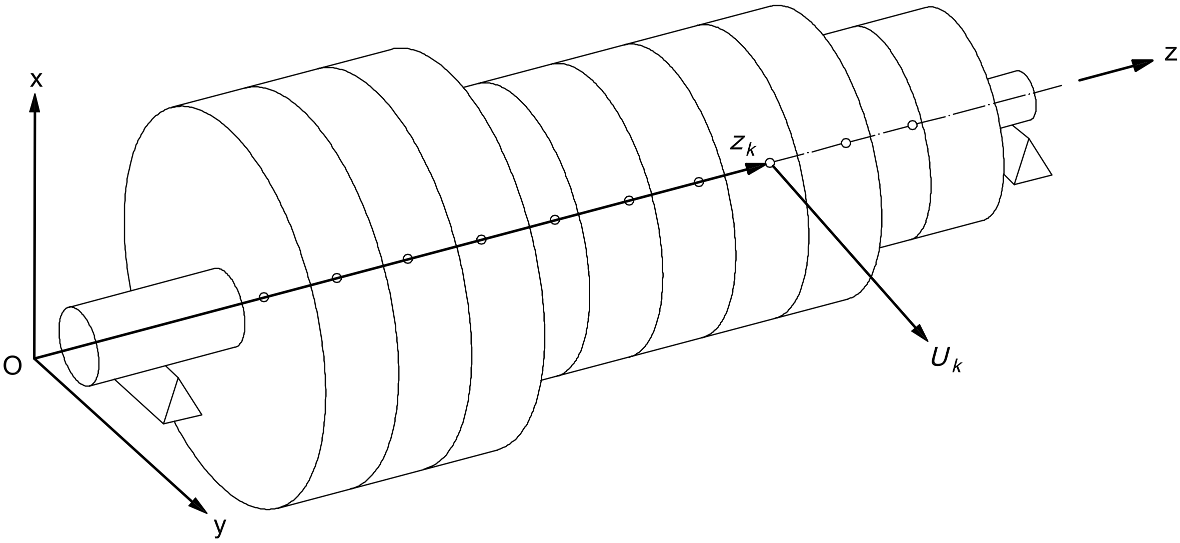


Figure A.2 — Unbalance vector $\vec{U}_k$ at axial position $\vec{z}_k$

Looking along the rotor axis, the three-dimensional drawings (Figures A.1 and A.2) can be represented by a two-dimensional view for each element. The unbalance vectors of Figure A.1 can be shown in individual planes as in Figure A.3.

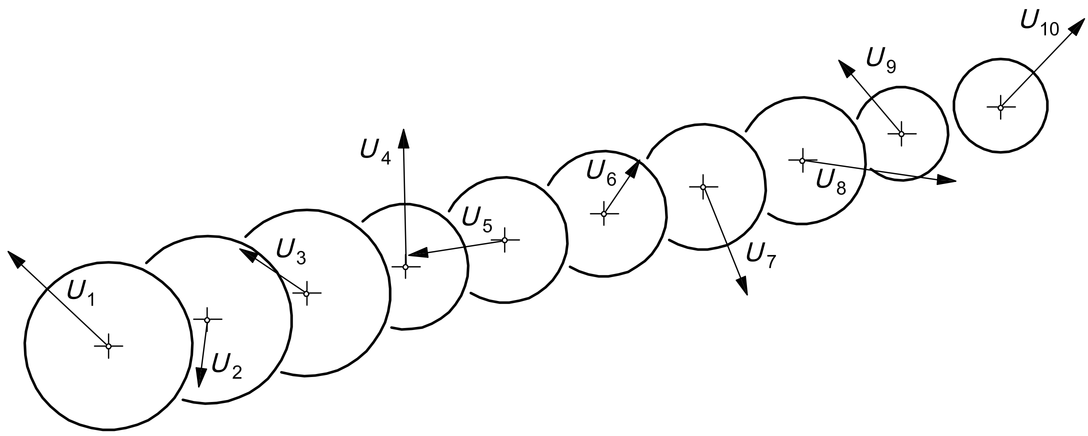


Figure A.3 — Unbalance vectors from Figure A.1 in individual planes, two-dimensional view

Alternatively, the unbalance vectors can be shown with the same origin, see Figure A.4 a), when projected onto an arbitrary plane that is normal to the shaft axis.

A.4 Resultant unbalance

The vector sum of all unbalance vectors, which represents the resultant unbalance $\vec{U}_{\mathrm{r}}$, in gram millimetres, can be shown graphically as in Figure A.4 b), or as the mathematical representation:

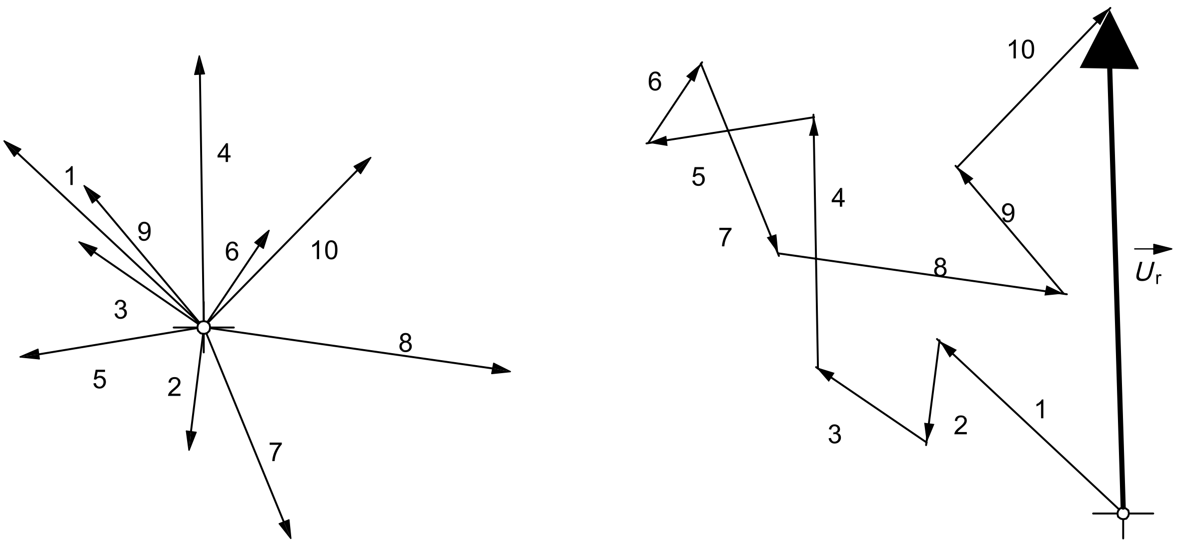
$$\vec{U}_{\mathrm{r}} = \sum_{k=1}^{K} \vec{U}_k \tag{A.1}$$

where

$\vec{U}_k$ are the individual unbalance vectors (g·mm);

k is the counter for the plane, from 1 to K.

The resultant unbalance is not linked to a certain radial plane, i.e. any plane may be chosen.



a) **Unbalance vectors $\vec{U}_k$ with the same origin**

b) **Graphical representation of the resultant unbalance $\vec{U}_{\mathrm{r}}$**

NOTE The numbers on the vectors in Figure A.4 represent the plane numbers, k.

Figure A.4 — Unbalance vectors $\vec{U}_k$, (k = 1, 2, ...K): Evaluation of resultant unbalance

A.5 Moment unbalance

A moment unbalance can only be defined with respect to a plane perpendicular to the shaft axis. This arbitrarily chosen plane then becomes the plane of the resultant unbalance, since only in this position does the resultant unbalance not effect the moment unbalance. See Figure A.5, where plane R, for example, is often chosen to be the centre plane.

For a mathematical representation and evaluation, the moment unbalance $\vec{P}_k$ (product of inertia) can be calculated for an unbalance vector $\vec{U}_k$ by the following vector product (see Figure A.5):

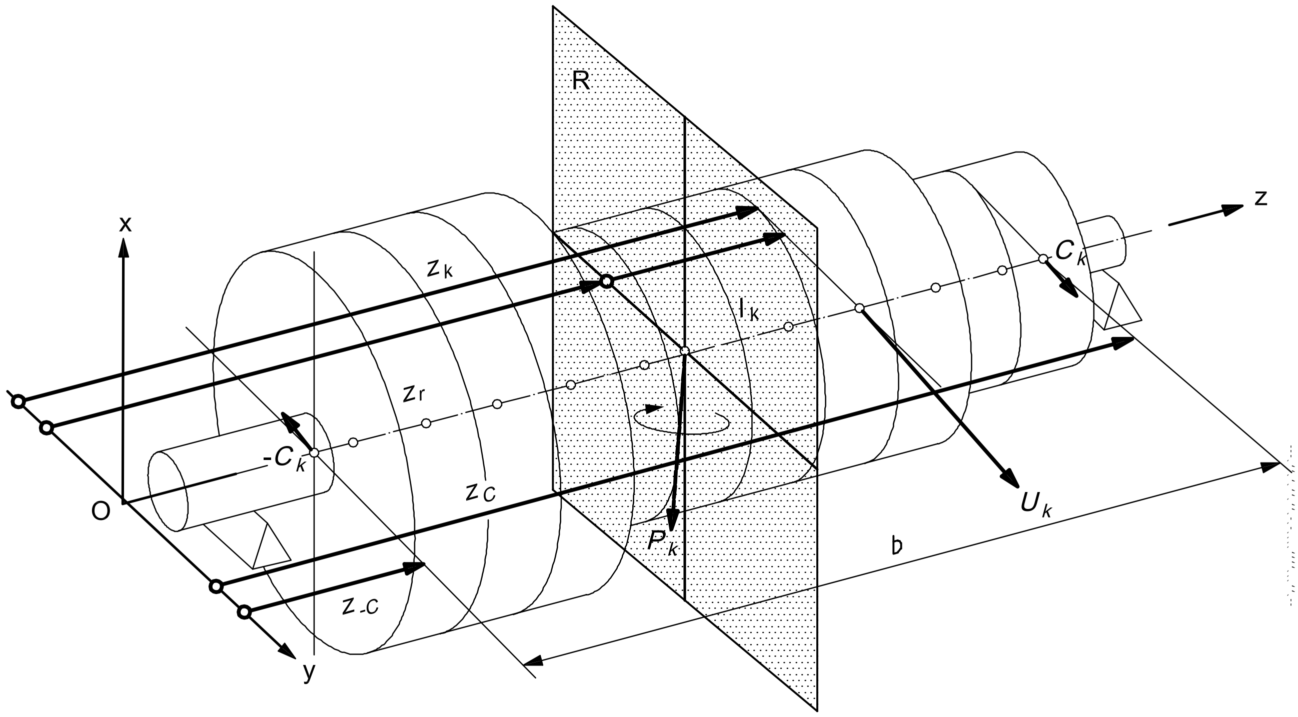
$$\vec{P}_k = \vec{l}_k \times \vec{U}_k \tag{A.2}$$

where

$\vec{P}_k$	is the moment unbalance, depending on the chosen position of $\vec{U}_r$ (g·mm^2);
$\vec{U}_k$	is the individual unbalance vector (g·mm);
$\vec{l}_k = (\vec{z}_k - \vec{z}_r)$	is the distance from the chosen plane R of the resultant unbalance $\vec{U}_r$ to the plane of the unbalance $\vec{U}_k$ (mm);
$\vec{z}_k$	is the distance from a datum mark O to the plane of $\vec{U}_k$ (mm);
$\vec{z}_r$	is the distance from a datum mark O to the plane of the resultant unbalance $\vec{U}_r$ (mm).

NOTE 1 The moment unbalance $\vec{P}_k$ is a vector different from the other unbalance vectors; it has magnitude, a direction and a direction of rotation.

NOTE 2 A balanced component that has been assembled skew to the axis of rotation creates a moment on the assembled rotor, because the axis of rotation is not a principal axis of inertia.



Key

R radial plane

NOTE b is the distance between the end planes of the rotor, and z_r is chosen to be in the centre plane (b/2).

Figure A.5 — With plane R chosen for the resultant unbalance $\vec{U}_r$: Representation of a moment unbalance $\vec{P}_r$ and the couple unbalance vectors $\vec{C}_k$ and $-\vec{C}_k$, together with the relevant distances

A.6 Resultant moment unbalance

The resultant moment unbalance $\vec{P}_\mathrm{r}$ is equal to the vector sum of the moment unbalances of all individual unbalance vectors $\vec{U}_k$ or has the mathematical representation:

$$\vec{P}_\mathrm{r} = \sum_{k=1}^{K} \vec{l}_k \times \vec{U}_k \tag{A.3}$$

where k is the counter for the plane, from 1 to K.

NOTE The resultant moment unbalance is needed for a clear distinction between a moment unbalance of an individual unbalance vector [see Equation (A.2)] and that of an unbalance distribution [see Equation (A.3)]. In practice, the term “moment unbalance” often means the unbalance distribution of the complete rotor.

A.7 Couple unbalance

In some cases, it is more convenient to express the moment unbalance $\vec{P}_k$ by the vector product of a couple unbalance (two unbalance vectors $\vec{C}_k$ and $-\vec{C}_k$) with the related plane positions $\vec{z}_C$ and $\vec{z}_{-C}$.

The planes for $\vec{C}_k$ and $-\vec{C}_k$ can be arbitrarily chosen. The index k is used here to state the relationship to $\vec{U}_k$ and $\vec{P}_k$.

For a graphical representation see Figure A.6 b). For the mathematical representation, the equation is

$$\vec{P}_k = \left(\vec{z}_C \times \vec{C}_k\right) + \left[\vec{z}_{-C} \times \left(-\vec{C}_k\right)\right] \tag{A.4}$$

where

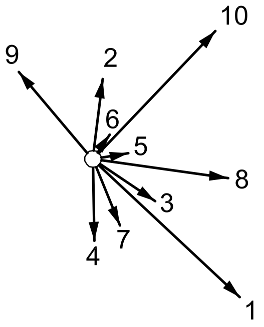
$\vec{P}_k$	is the moment unbalance (g·mm^2);
$\vec{C}_k$ and $-\vec{C}_k$	are the vectors of the couple unbalance (g·mm) related to moment unbalance $\vec{P}_k$; in this case they are chosen to be at the end planes, a distance b apart;
$\vec{z}_C$ and $\vec{z}_{-C}$	are the distances of these two vectors from a datum mark O (mm).

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

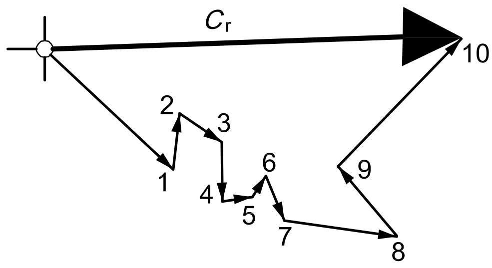
NOTE 2 This equation can be transformed to: $\vec{P}_k = (\vec{z}_C - \vec{z}_{-C}) \times \vec{C}_k$. The term $\left|(\vec{z}_C - \vec{z}_{-C})\right|$ is the distance b between the two couple unbalance vectors.

If the centre plane is chosen for the resultant unbalance and if the plane distance b equals the distance between both end planes (see Figure A.5), the individual couple unbalance vectors in the left-hand end plane are shown in Figure A.6 a).

The vector sum of all couple unbalances equals the resulting couple unbalance in the right-hand end plane, see Figure A.6 b) (in the left-hand end plane, it has the same magnitude but opposite direction).



a) Couple unbalance vectors in the right-hand end plane ($\vec{C}_k$, $k = 1$ through 10), calculated by scaling each $\vec{U}_k$ with its associated distance ratio $(z_k - z_r)/b$



b) Resulting couple unbalance $\vec{C}_r$ in the right-hand end plane
(in the left-hand end plane, it has the same magnitude but opposite direction)

NOTE The numbers on the vectors represent the plane numbers k.

Figure A.6 — Graphical representation and evaluation of unbalances, resultant couple unbalance

A.8 Modal unbalances

A.8.1 Mode shapes

Modal unbalances are based on bending deflections (mode shape functions $\phi_n(z)$, $n = 1, 2, 3, ...$) of the individual modes. Figure A.7 shows (idealized) modal deflections for the first three flexural modes of a symmetric rotor with equally distributed mass and stiffness.

A.8.2 Modal unbalance representation

Unbalances are assumed to be identical to Figure A.1. If each individual unbalance vector $\vec{U}_k$ is scaled with the relevant ordinate value of the respective mode $\phi_n(z_k)$, the individual nth modal unbalances $\vec{U}_{n,k}$ are obtained:

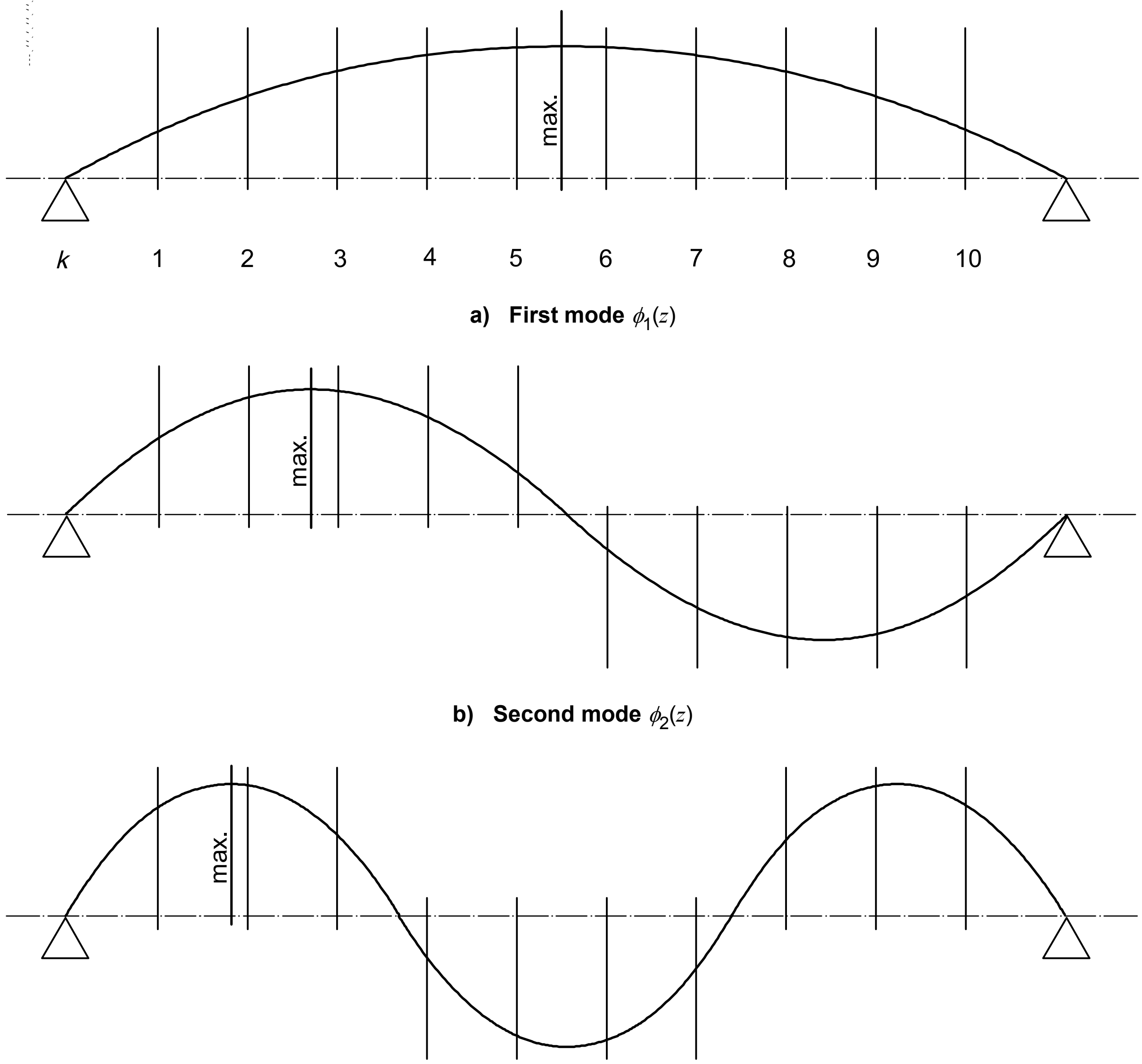
$$\vec{U}_{n,k} = \vec{U}_k \, \phi_n(z_k), \; k = 1, 2, 3, ...10 \tag{A.5}$$

All these unbalances for a particular mode together depict the modal unbalance distribution. The sum of these unbalances yields the resultant nth modal unbalance $\vec{U}_{n,\mathrm{r}}$:

$$\vec{U}_{n,\mathrm{r}} = \sum_{k=1}^{K} \vec{U}_k \, \phi_n(z_k), \; k = 1, 2, 3, ...10 \tag{A.6}$$

NOTE In the 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 model unbalance, see Equation (A.5), and the resultant nth modal unbalance, see Equation (A.6).



a) **First mode** $\phi_1(z)$

b) **Second mode** $\phi_2(z)$

c) **Third mode** $\phi_3(z)$

Figure A.7 — First three flexural mode shapes of a symmetric rotor with equally distributed mass and stiffness on rigid supports

A.9 Equivalent modal unbalance

Modal unbalances unfortunately do not lead to a common understanding since different ways of scaling the ordinate values of the mode exist. If the individual modal unbalances $\vec{U}_{n,k}$ are divided by the maximum ordinate value of the mode $\phi_{n,\text{max}}$, it becomes the individual nth equivalent modal unbalance $\vec{U}_{n\text{e},k}$ acting at the plane of the chosen maximum ordinate.

For a graphical representation of the first mode, see Figure A.8 d). For the general mathematical representation, the equation is

$$\vec{U}_{n\text{e},k} = \vec{U}_k \frac{\phi_n(z_k)}{\phi_{n,\text{max}}}, \qquad k = 1, 2, 3, ...10 \tag{A.7}$$

The sum of these vectors is the resultant nth equivalent modal unbalance $\vec{U}_{n\text{e,r}}$:

$$\vec{U}_{n\text{e,r}} = \sum_{k=1}^{K} \vec{U}_k \frac{\phi_n(z_k)}{\phi_{n,\max}} \tag{A.8}$$

The equivalent modal unbalance (for different flexural modes) is used to state rotor unbalances and rotor balance tolerances. However, it cannot be used to correct modal unbalances.

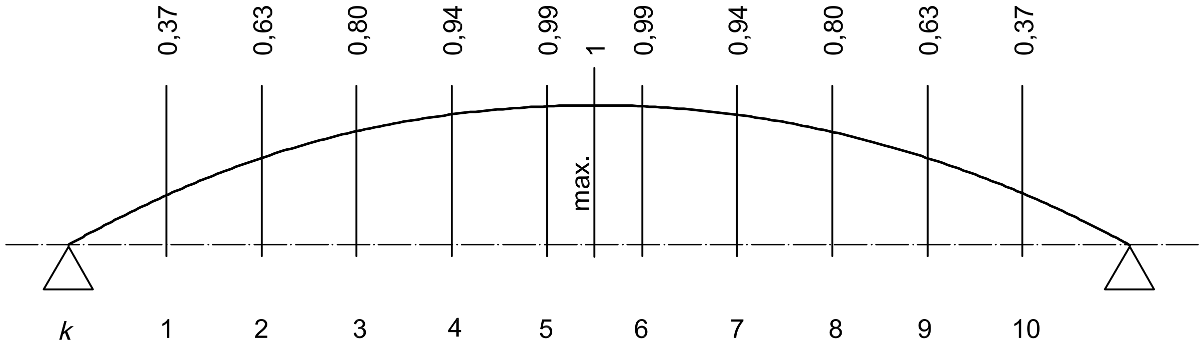
NOTE 1 This $\vec{U}_{n\text{e,r}}$ is a single unbalance vector acting at the axial position of $\phi_{n,\max}$ with the same effect on the bending of the rotor in this mode as the combined effect of all the individual unbalances.

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

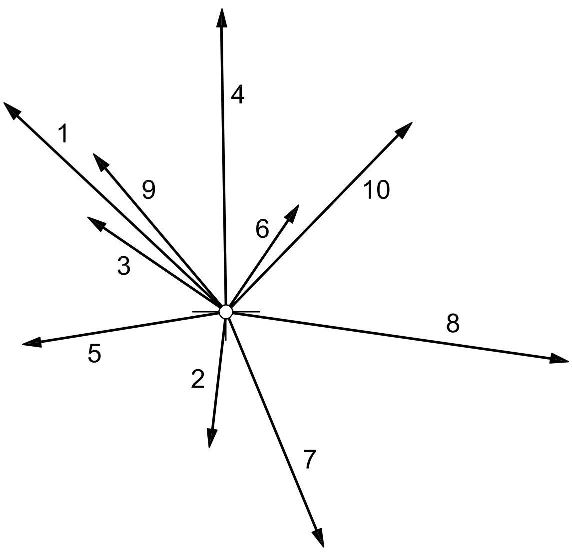
NOTE 3 In the case of plane-bending modes, the ordinate values are just real numbers, but for three-dimensional bending modes they can be complex numbers.

A clear distinction should be made between the individual nth model equivalent unbalance, see Equation (A.7), and the resultant nth modal equivalent unbalance, see Equation (A.8).

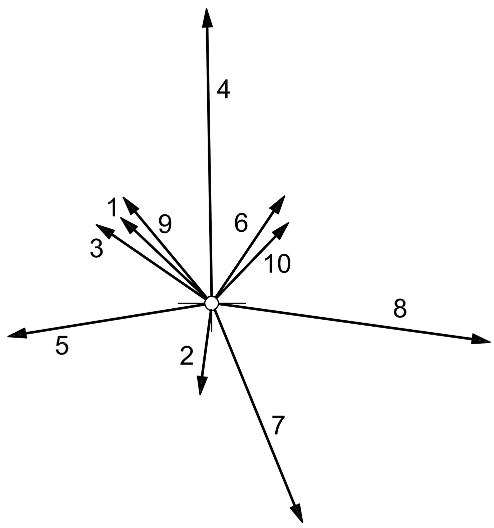
For the first mode, Figure A.8 a) shows the ordinate values for the different planes. Figure A.8 b) shows the unbalance vectors, $\vec{U}_k$, seen along the axis. Figure A.8 c) depicts the individual equivalent modal unbalances, $\vec{U}_{1\text{e},k}$, see Equation (A.7). Figure A.8 d) shows how the vector sum constitutes the resultant equivalent modal unbalance $\vec{U}_{1\text{e,r}}$, see Equation (A.8).



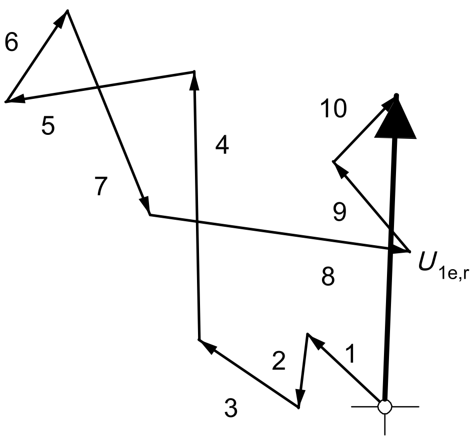
a) First mode with ordinate values (idealized)



b) Unbalances $\vec{U}_k$ in 10 planes



c) Unbalances weighted with the ratio of the ordinates in these planes and the maximum absolute ordinate value $\vec{U}_{1e,k} = \vec{U}_k \left[\phi_1(z_k) / \phi_{1,max} \right]$, $k = 1, 2, 3, ...10$



d) Vector sum of the weighted vectors, equals the resultant first equivalent modal unbalance $\vec{U}_{1e,r}$

NOTE The numbers on the vectors represent the plane numbers, k.

Figure A.8 — Graphical representation of first equivalent modal unbalance of a rotor on rigid supports

Annex B
(informative)

Examples of different rotor behaviours
(Indication on a typical hard-bearing balancing machine)

B.1 General

As the rotor speed varies, a rotor may show one of the following rotor behaviours or a combination thereof. Based on this behaviour, the indicated unbalance and the response to unbalance will be different. Observing how the rotor response varies with speed for two levels of unbalance (rotor unbalanced and balanced) may give hints about the behaviour of a particular rotor. Idealized characteristics of different rotor behaviours are described in B.2 to B.5.

In order to show this as clearly as possible, 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 its operational speed n_2,

b) the balancing machine bearing supports are sufficiently rigid that there is no significant change in their dynamic characteristics which would change the unbalance indication, and

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

NOTE 1 Modern hard-bearing balancing machines are typically calibrated to indicate unbalance masses (or correction masses). For many rotors, the indication will be constant within the working speed range of the machine, as long as the unbalance does not change.

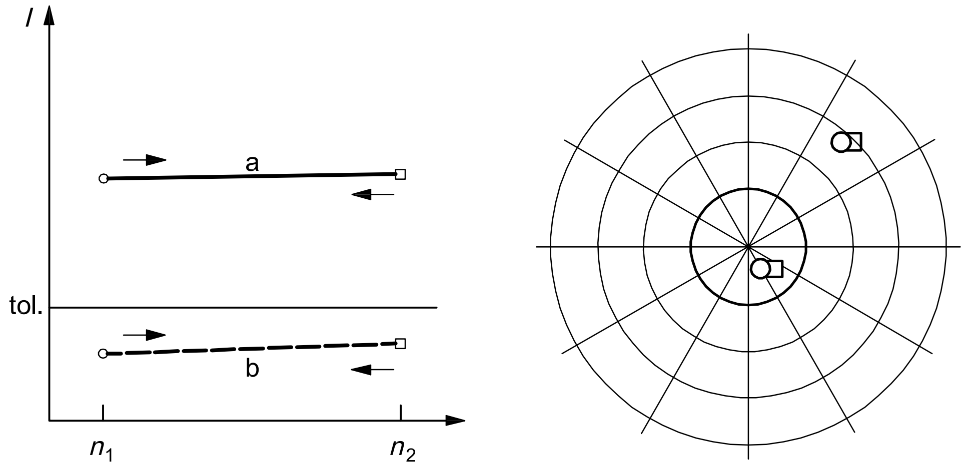
NOTE 2 In the case of a rotor with flexible behaviour, the unbalance indication will change with speed due to the influence of the flexural modes, even though the unbalances remain constant. Hence a change of unbalance indication could be due either to flexible rotor behaviour or to a change of unbalance.

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 following examples shown in Figures B.1 to B.5, two different representations of the indications are used. On the left, the indicated unbalance response is plotted against speed, whereas on the right the variation of the indicated unbalance response vectors (amplitude and phase) throughout the speed range are shown in a polar display. The balance tolerance is shown as a horizontal line on the left-hand diagram and as a circle on the right-hand diagram (in Figures B.2 and B.3 this is only applicable at low speed; at higher speeds the tolerances are much more complex, since they address different equivalent modal unbalances).

B.2 Rigid behaviour (see Figure B.1)

For rotors which exhibit rigid behaviour, there is a negligible change in the unbalance response indication over the speed range in both the balanced and unbalanced condition (in comparison to the balance tolerance).



Key

a unbalanced

b balanced

I indication on a balancing machine, see B.1

tol. balance tolerance

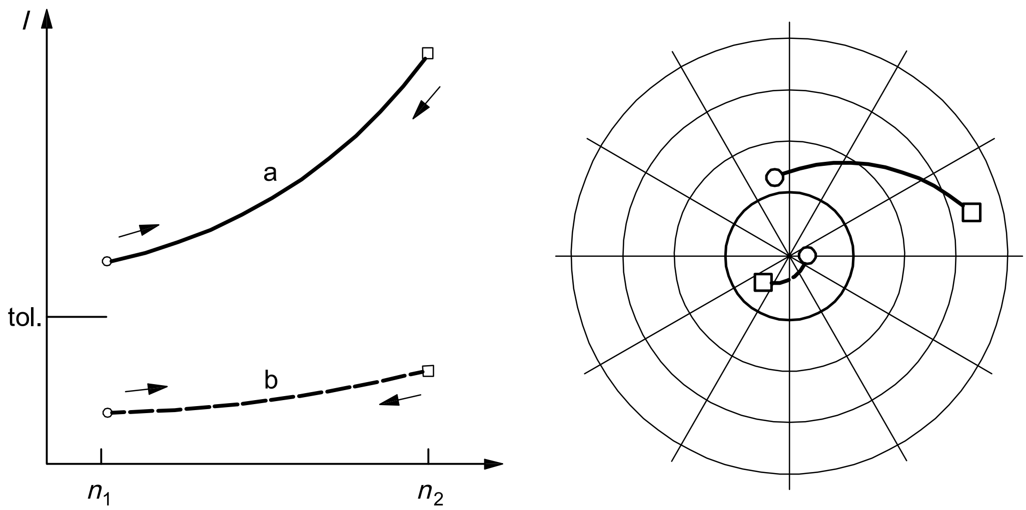
Figure B.1 — Rigid behaviour: Negligible change in indication over the speed range compared to the balance tolerance

B.3 Flexible behaviour (see Figures B.2 and B.3)

For rotors with flexible behaviour, the unbalance response indication changes with speed. But, in this case, it is not due to a change in unbalance but is a consequence of the rotor undergoing bending deflection due to modal unbalance (see Note 2 in B.1).

The change in indication is reversible; i.e. the indication will follow the same curve when increasing and reducing the speed.

If the operational speed is below the first bending resonance, the indication will be as shown Figure B.2.



Key

a unbalanced

b balanced

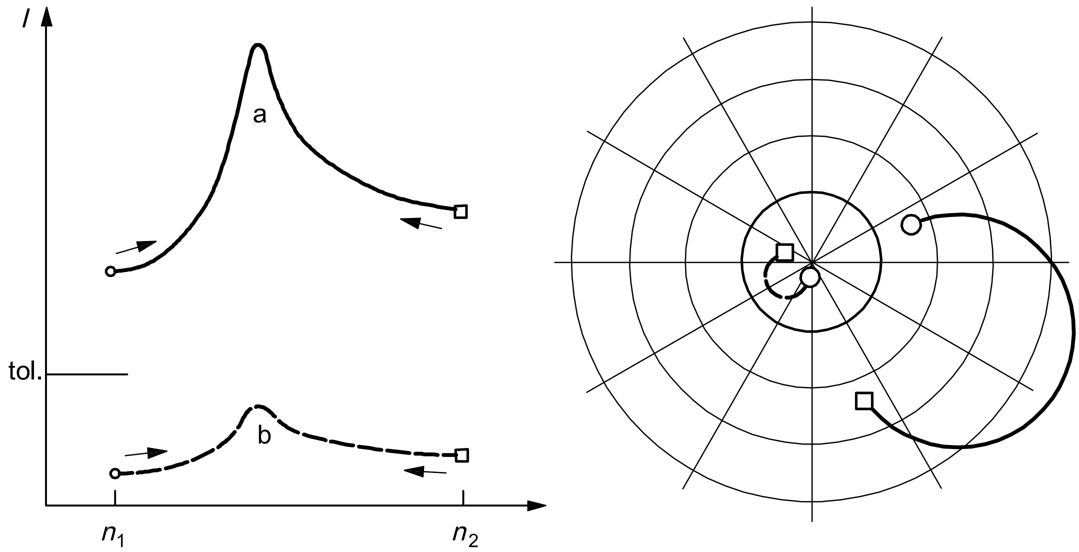
I indication on a balancing machine, see B.1

tol. balance tolerance, see Note in B.3

Figure B.2 — Flexible behaviour: Operational speed is below the first bending resonance

If the operational speed is above the first bending resonance, the indication will be as shown in Figure B.3.

Using a balancing procedure which addresses the flexible behaviour properly, the indications and the change in indications will become smaller and will meet the tolerances within the speed range.



Key

a unbalanced

b balanced

I indication on a balancing machine, see B.1

tol. balance tolerance, see Note in B.3

Figure B.3 — Flexible behaviour: Operational speed is above the first bending resonance

NOTE The balance tolerance in all figures is shown as a horizontal line on the left-hand diagram and as a circle on the right-hand diagram. In Figures B.2 and B.3, this is only applicable at low speed; at higher speeds the tolerances are much more complex, since they can address different equivalent modal unbalances.

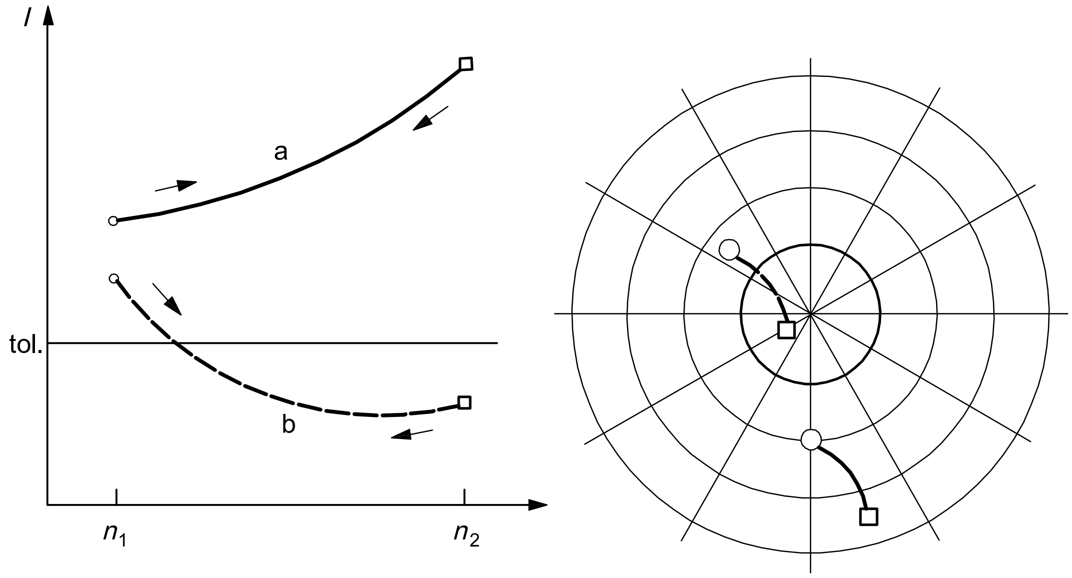
B.4 Elastic behaviour of components

For rotors with component elastic behaviour, the unbalance response indication changes with speed due to a change in unbalance, see Figure B.4.

The change in indication is reversible; i.e. the indication will follow the same curve when increasing and reducing the speed.

It is not possible to compensate for this type of unbalance change with correction masses. Therefore, if the component elastic behaviour cannot be reduced by a modified manufacturing or assembly process, balancing is only a compromise to minimize the unbalance level over the operational speed range. The datum position can be changed and this may be sufficient to achieve a compromise balance condition for which the unbalance remains within tolerance at all speeds. In some cases, this may not be possible and, in such cases, a rotor with component elastic behaviour would typically be balanced to be within tolerance at higher speeds, with a risk of being out of tolerance at lower speeds, as shown in Figure B.4.

Typically, such a rotor is balanced to be within tolerance at higher speeds, with the risk of being out of tolerance at lower speeds.



Key

a unbalanced

b balanced

I indication on a balancing machine, see B.1

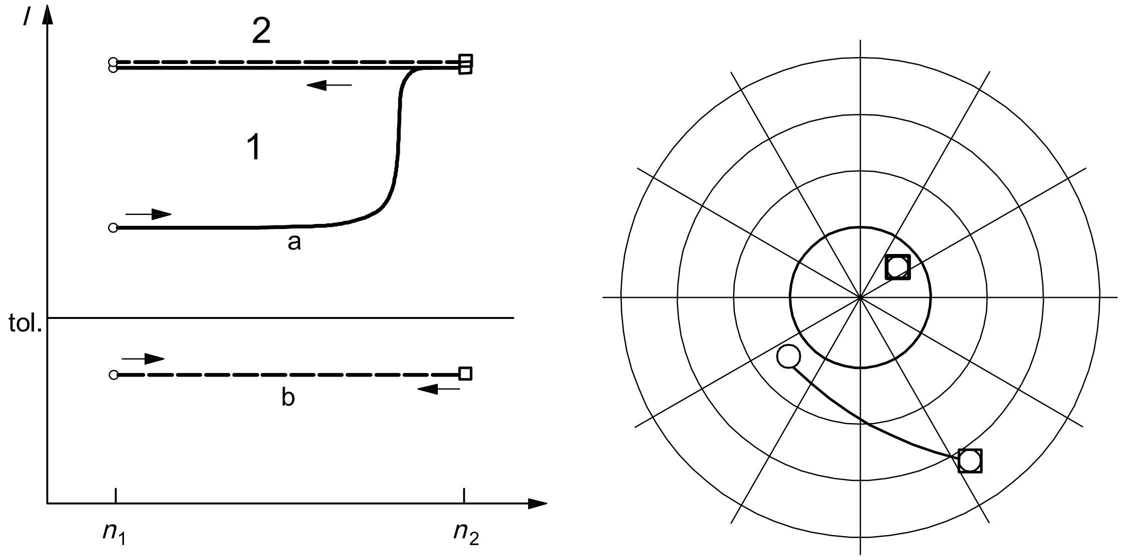
tol. balance tolerance

NOTE Component elastic behaviour is shown here in combination with a rigid rotor behaviour, but it may be combined also with other rotor behaviours.

Figure B.4 — Elastic behaviour of components

B.5 Seating behaviour

In this case, there is a sudden change in the unbalance response indication at a particular speed, as shown in Figure B.5.



Key

1 1st run

2 2nd run

a unbalanced

b balanced

I indication on a balancing machine, see B.1

tol. balance tolerance

Figure B.5 — Seating behaviour

The rotor settles with the first run. Here the second run shows a rigid behaviour, but seating behaviour can superimpose any rotor behaviour. This change in indication is not reversible; i.e. the indication does not follow the same curve when increasing and reducing the speed.

However, typically the rotor settles in its final position after running to a sufficient speed, usually over speed. Thus the seating behaviour will not occur again for all following runs.

NOTE 1 In the example shown in Figure B.5, the settled rotor shows a rigid behaviour. Seating behaviour can be combined also with one of the other rotor behaviours.

NOTE 2 As soon as the rotor has settled, it can be balanced using the appropriate procedure for that particular rotor behaviour.

Annex C
(informative)

How to determine rotor flexibility based on an estimation from its geometric design

C.1 General

The path taken to balance the rotor depends on rotor flexibility when running in its service environment. This annex provides a method to make judgements as to the flexibility of the rotor in its operating environment based on the geometric design of the rotor and an approximate knowledge of its supporting structure.

If the full details of the design of the rotor plus the dynamic characteristics of the bearings and their support structures are defined, the mathematical simulation will give a more exact estimate of rotor behaviour. This will enable a correct assessment of rotor behaviour to be made.

C.2 Mathematically modelling the rotor

The rotor can easily be mathematically modelled from its geometric design to estimate its natural frequencies. Basic parameters, such as the bearing centre distance, total rotor mass, the external diameters defining the stiffness and mass distribution, are required. Care should be taken when assessing the increase in stiffness due to shrunk-on components. To gain further confidence, simulating the first free-free flexural mode of the rotor and comparing this with experimental data can check the rotor model. A good approximation for a free-free mode is to support the rotor from a crane and excite it in the horizontal direction.

C.3 Procedure

Calculate the first pinned-pinned (i.e. hinged) critical speed of the rotor by using the mathematical representation developed in C.2 and with the rotor simply supported at the bearings with infinite radial stiffness. Estimate the stiffness of the rotor, S_r, in newtons per metre, according to the following equation:

$$S_\mathrm{r} = m_\mathrm{r} \cdot \omega_{\mathrm{r}1}{}^2 \qquad \text{(C.1)}$$

where

m_r is the rotor mass (kg);

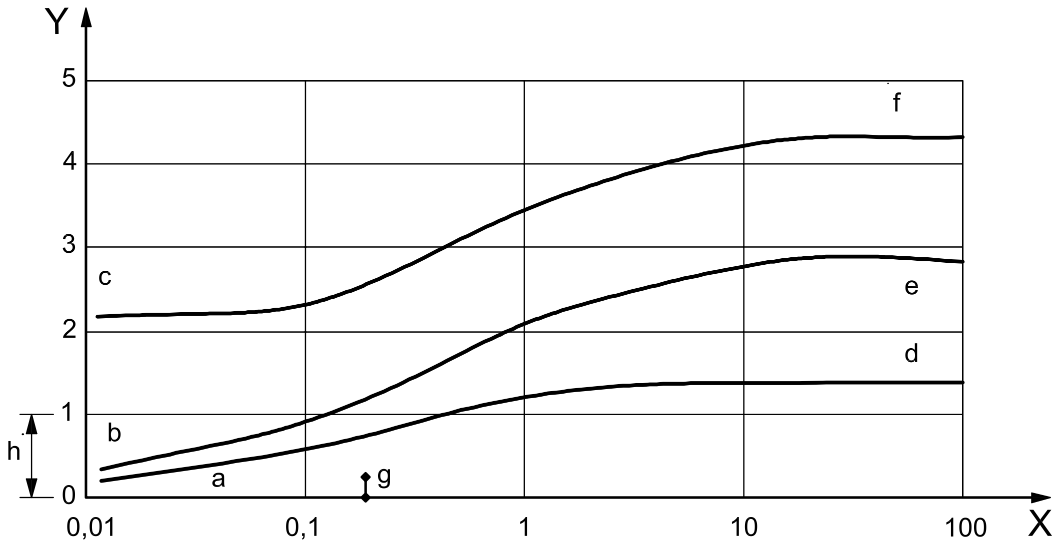
$\omega_{\mathrm{r}1}$ is the first pinned-pinned critical speed of the rotor (rad/s).

Recalculate the first, second and third critical speeds over a range of total support stiffness (i.e. bearing and support stiffness). Normalize the support stiffness relative to that calculated for the rotor, normalize the critical speeds relative to normal or maximum operating speed, and generate a graph as shown in the examples, see Figures C.1 and C.2.

In Figures C.1 and C.2, the first, second and third critical speeds are represented by the lines denoted by a–d, b–e and c–f, respectively. Due to the normal stiffness asymmetry of a supporting oil film and bearing structures, these critical speeds would usually be seen in pairs corresponding to the two principal stiffness directions, but to simplify the analysis here, the supports are assumed to be symmetrical. With low support stiffness, the two lower modes are the rigid-body modes and the third is the first flexural mode. Rotors on stiff supports will only see the flexural modes. If an approximate value for oil film stiffness is available this can be added to the diagram, since this will represent the maximum support stiffness available. On many machines the rotor critical peaks seen in the run-up and run-down can be related to those shown on the figure and will provide an approximate indication of the support stiffness.

C.4 Assignment of rigid or flexible behaviour to rotors

Figure C.1 is an example where the rotor is operating with rigid behaviour. At high and low support stiffness, c and d respectively, the first flexural critical speed is above the operating speed range. There may be a stiffness range where the supports have a value just below that of the rotor, where the rotor could be influenced by its flexible state. For the example given, this is unlikely since the stiffness would be above that of the oil film. In practice, the rotor is much stiffer than its supports and will operate as a rigid rotor with minimal bending.

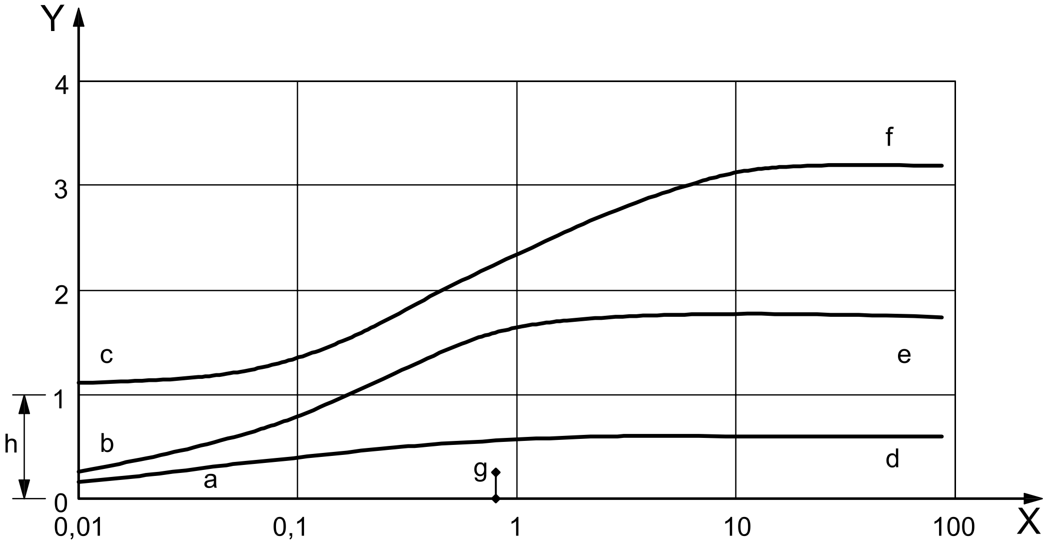


Key

X	ratio of total support stiffness over rotor stiffness	d	1st flexural mode (low support stiffness)
Y	ratio of critical speed over service speed	e	2nd flexural mode
a	1st rigid-body mode	f	3rd flexural mode
b	2nd rigid-body mode	g	estimated stiffness of the bearing oil films
c	1st flexural mode (high support stiffness)	h	range 0 to 1 for normal operating speed

Figure C.1 — Critical speed map of a rotor with rigid behaviour

The data shown in Figure C.2 clearly indicate that this rotor is operating with a flexible behaviour. For machines with high support stiffness, the first critical speed (d) is below the service speed range and, on low support stiffness, the first flexural mode (c) is sufficiently close to the service speed to become important.



Key

X	ratio of total support stiffness to rotor stiffness	d	1st flexural mode
Y	ratio of critical speed to service speed	e	2nd flexural mode
a	1st rigid-body mode	f	3rd flexural mode
b	2nd rigid-body mode	g	estimated stiffness of the bearing oil films
c	1st flexural mode	h	range 0 to 1 for normal operating speed

Figure C.2 — Critical speed map of a rotor with flexible behaviour

C.5 Special circumstances

The method shown considers the normal situation for large rotor systems, where the rotors are supported on stiffness-controlled support structures. For situations where the support structure is mass controlled or even in resonance, the above method could give misleading results. However, when the method is applied to large land-based gas turbines, where the bearing support structures carry the mass of the casings, the method correctly indicates that most of these rotors operate with a flexible behaviour at normal operating speed.

C.6 Marginal cases

This method is useful for identifying the majority of clearly flexible or rigid rotors, but there will be marginal rotors in between. In these situations, and where the dynamics of the support structure can be estimated, the procedure outlined in Annex E of ISO 11342:1998 can be mathematically simulated, where the relative response of trial masses added at the mid- and quarter-spans is considered.

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ICS 21.120.40

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