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Mechanical vibration — Rotor balancing —

Part 12:
Procedures and tolerances for rotors with flexible behaviour

Vibrations mécaniques — Équilibrage des rotors —

Partie 12: Modes opératoires et tolérances pour les rotors à comportement flexible

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Foreword

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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).

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For an explanation on the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the WTO principles in the Technical Barriers to Trade (TBT) see the following URL: Foreword - Supplementary information

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

This first edition of ISO 21940-12 cancels and replaces ISO 11342:1998, which has been technically revised. The main changes are deletion of the terms and definitions which were transferred to ISO 21940-2 and deletion of former Annex F which is a duplication of a part of D.1. It also incorporates the Technical Corrigendum ISO 11342:1998/Cor.1:2000.

ISO 21940 consists of the following parts, under the general title *Mechanical vibration — Rotor balancing*:

— *Part 11: Procedures and tolerances for rotors with rigid behaviour*[1)]

— *Part 12: Procedures and tolerances for rotors with flexible behaviour*[2)]

— *Part 13: Criteria and safeguards for the in-situ balancing of medium and large rotors*[3)]

— *Part 14: Procedures for assessing balance errors*[4)]

— *Part 21: Description and evaluation of balancing machines*[5)]

1) Revision of ISO 1940-1:2003 + Cor.1:2005, *Mechanical vibration — Balance quality requirements for rotors in a constant (rigid) state — Part 1: Specification and verification of balance tolerances*

2) Revision of ISO 11342:1998 + Cor.1:2000, *Mechanical vibration — Methods and criteria for the mechanical balancing of flexible rotors*

3) Revision of ISO 20806:2009, *Mechanical vibration — Criteria and safeguards for the in-situ balancing of medium and large rotors*

4) Revision of ISO 1940-2:1997, *Mechanical vibration — Balance quality requirements of rigid rotors — Part 2: Balance errors*

5) Revision of ISO 2953:1999, *Mechanical vibration — Balancing machines — Description and evaluation*

— *Part 23: Enclosures and other protective measures for the measuring station of balancing machines*[6)]

— *Part 31: Susceptibility and sensitivity of machines to unbalance*[7)]

— *Part 32: Shaft and fitment key convention*[8)]

The following part is under preparation:

— *Part 2: Vocabulary*[9)]

6) Revision of ISO 7475:2002, *Mechanical vibration — Balancing machines — Enclosures and other protective measures for the measuring station*

7) Revision of ISO 10814:1996, *Mechanical vibration — Susceptibility and sensitivity of machines to unbalance*

8) Revision of ISO 8821:1989, *Mechanical vibration — Balancing — Shaft and fitment key convention*

9) Revision of ISO 1925:2001, *Mechanical vibration — Balancing — Vocabulary*

Introduction

The aim of balancing any rotor is to achieve satisfactory running when installed *in-situ*. In this context, “satisfactory running” means that not more than an acceptable magnitude of vibration is caused by the unbalance remaining in the rotor. In the case of a rotor with flexible behaviour, it also means that not more than an acceptable magnitude of deflection occurs in the rotor at any speed up to the maximum service speed.

Most rotors are balanced in manufacture prior to machine assembly because afterwards, for example, there might be only limited access to the rotor. Furthermore, balancing of the rotor is often the stage at which a rotor is approved by the purchaser. Thus, while satisfactory running *in-situ* is the aim, the balance quality of the rotor is usually initially assessed in a balancing machine. Satisfactory running *in-situ* is, in most cases, judged in relation to vibration from all causes, while in the balancing machine, primarily, once-per-revolution effects are considered.

This part of ISO 21940 classifies rotors in accordance with their balancing requirements and establishes methods of assessment of residual unbalance.

This part of ISO 21940 also shows how criteria for use in the balancing machine can be derived from either vibration limits specified for the assembled and installed machine or unbalance limits specified for the rotor. If such limits are not available, this part of ISO 21940 shows how they can be derived from ISO 10816 and ISO 7919 if desired in terms of vibration, or from ISO 21940-11, if desired in terms of permissible residual unbalance. ISO 21940-11 is concerned with the balance quality of rotating rigid bodies and is not directly applicable to rotors with flexible behaviour because rotors with flexible behaviour can undergo significant bending deflection. However, in this part of ISO 21940, methods are presented for adapting the criteria of ISO 21940-11 to rotors with flexible behaviour.

There are situations in which an otherwise acceptably balanced rotor experiences an unacceptable vibration level *in situ*, owing to resonances in the support structure. A resonance or near resonance condition in a lightly damped structure can result in excessive vibratory response to a small unbalance. In such cases, it can be more practicable to alter the natural frequency or damping of the structure rather than to balance to very low levels, which might not be maintainable over time (see also ISO 21940-31).

Mechanical vibration — Rotor balancing —

Part 12:
Procedures and tolerances for rotors with flexible behaviour

1 Scope

This part of ISO 21940 presents typical configurations of rotors with flexible behaviour in accordance with their characteristics and balancing requirements, describes balancing procedures, specifies methods of assessment of the final state of balance, and establishes guidelines for balance quality criteria.

This part of ISO 21940 can also serve as a basis for more involved investigations, e.g. when a more exact determination of the required balance quality is necessary. If due regard is paid to the specified methods of manufacture and balance tolerances, satisfactory running conditions can be expected.

This part of ISO 21940 is not intended to serve as an acceptance specification for any rotor, but rather to give indications of how to avoid gross deficiencies and unnecessarily restrictive requirements.

Structural resonances and modifications thereof lie outside the scope of this part of ISO 21940.

The methods and criteria given are the result of experience with general industrial machinery. It is possible that they are not directly applicable to specialized equipment or to special circumstances. Therefore, in some cases, deviations from this part of ISO 21940 are possible.

2 Normative references

The following documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 1925[10)], *Mechanical vibration — Balancing — Vocabulary*

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

ISO 21940-11[11)], *Mechanical vibration — Rotor balancing — Part 11: Procedures and tolerances for rotors with rigid behaviour*

ISO 21940-14, *Mechanical vibration — Rotor balancing — Part 14: Procedures for assessing balance errors*

ISO 21940-32, *Mechanical vibration — Rotor balancing — Part 32: Shaft and fitment key convention*

3 Terms and definitions

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

10) To become ISO 21940-2 when revised.

11) To be published.

4 Fundamentals of dynamics and balancing of rotors with flexible behaviour

4.1 General

Rotors with flexible behaviour normally require multiplane balancing at high speed. Nevertheless, under certain conditions, a rotor with flexible behaviour can also be balanced at low speed. For high-speed balancing, two different methods have been formulated for achieving a satisfactory state of balance, namely modal balancing and the influence coefficient approach. The basic theory behind both of these methods and their relative merits are described widely in the literature and therefore, no further detailed description is given here. In most practical balancing applications, the method adopted is normally a combination of both approaches, often incorporated into a computer package.

4.2 Unbalance distribution

The rotor design and method of construction can significantly influence the magnitude and distribution of unbalance along the rotor axis. Rotors may be machined from a single forging or they may be constructed by fitting several components together. For example, jet engine rotors are constructed by joining many shell, disc and blade components. Generator rotors, however, are usually manufactured from a single forging, but will have additional components fitted. The distribution of unbalance may also be significantly influenced by the presence of large unbalances arising from shrink-fitted discs, couplings, etc.

Since the unbalance distribution along a rotor axis is likely to be random, the distribution along two rotors of identical design will be different. The distribution of unbalance is of greater significance in a rotor with flexible behaviour than in a rotor with rigid behaviour because it determines the degree to which any flexural mode is excited. The effect of unbalance at any point along a rotor depends on the mode shapes of the rotor.

The correction of unbalance in transverse planes along a rotor other than those in which the unbalance occurs can induce vibrations at speeds other than that at which the rotor was originally balanced. These vibrations can exceed specified tolerances, particularly at, or near, the flexural resonance speeds. Even at the same speed, such correction can induce vibrations if the flexural mode shapes *in-situ* differ from those dominating during the balancing process.

Rotors should be checked for straightness, and where necessary corrected prior to high-speed balancing, since a rotor with an excessive bend or bow will result in a compromise balance, which can lead to poor performance in service.

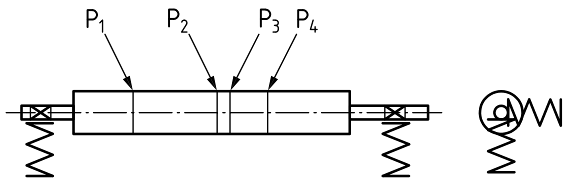
In addition, some rotors which become heated during operation are susceptible to thermal bows which can lead to changes in the unbalance. If the rotor unbalance changes significantly from run to run, it might be impossible to balance the rotor within tolerance.

4.3 Mode shapes of rotors with flexible behaviour

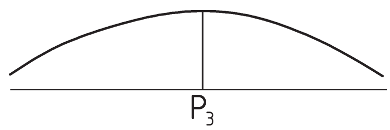
If the effect of damping is neglected, the modes of a rotor are the flexural principal modes and, in the special case of a rotor supported in bearings which have the same stiffness in all radial directions, are rotating plane curves. Typical shapes for the three lowest principal modes for a simple rotor supported in flexible bearings near to its ends are illustrated in Figure 1.

For a damped rotor and bearing system, the flexural modes can be space curves rotating about the shaft axis, especially in the case of substantial damping, arising perhaps from fluid-film bearings. Possible damped first and second modes are illustrated in Figure 2. In many cases, the damped modes can be treated approximately as principal modes and, hence, regarded as rotating plane curves.

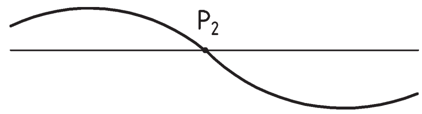
It is important to note that the form of the mode shapes and the response of the rotor to unbalances are strongly influenced by the dynamic properties and axial locations of the bearings and their supports.



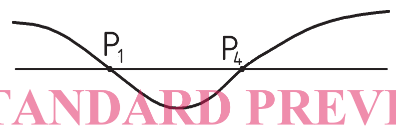
a) Typical rotor



b) First flexural mode



c) Second flexural mode



d) Third flexural mode

Key

P_1, P_2, P_4 nodes

P_3 antinode

Figure 1 — Simplified mode shapes for rotors with flexible behaviour on flexible supports

4.4 Response of a rotor with flexible behaviour to unbalance

The unbalance distribution can be expressed in terms of modal unbalances. The deflection in each mode is caused by the corresponding modal unbalance. When a rotor rotates at a speed near a resonance speed, it is usually the mode associated with this resonance speed which dominates the deflection of the rotor. The degree to which large amplitudes of rotor deflection occur under these circumstances is influenced mainly by the following:

a) the magnitude of the modal unbalances;

b) the proximity of the associated resonance speeds to the running speeds;

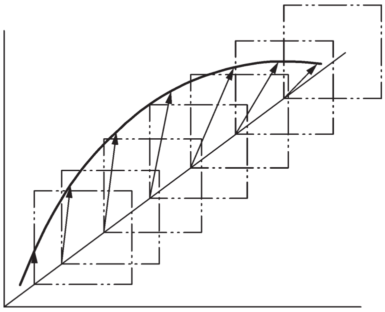
c) the amount of damping in the rotor and support system.

If a particular modal unbalance is reduced by the addition of a number of discrete correction masses, then the corresponding modal component of deflection is similarly reduced. The reduction of the modal unbalances in this way forms the basis of the balancing procedures described in this part of ISO 21940.

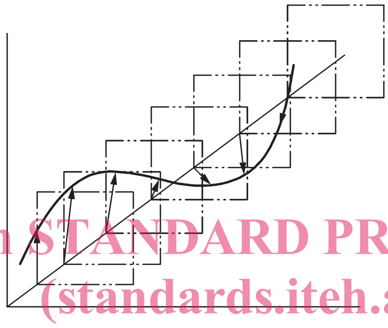
The modal unbalances for a given unbalance distribution are a function of the rotor modes. Moreover, for the simplified rotor shown in Figure 1, the effect produced in a particular mode by a given correction depends on the ordinate of the mode shape curve at the axial location of the correction: maximum effect near the antinodes, minimum effect near the nodes. Consider an example in which the curves of Figure 1 b) to d) are mode shapes for the rotor in Figure 1 a). A correction mass in plane P_3 has the maximum effect on the first mode, while its effect on the second mode is small.

A correction mass in plane P_2 will produce no response at all on the second mode, but will influence both the other modes.

Correction masses in planes P_1 and P_4 will not affect the third mode, but will influence both the other modes.



a) First mode



b) Second mode

Figure 2 — Examples of possible damped mode shapes

4.5 Aims of balancing rotors with flexible behaviour

The aims of balancing are determined by the operational requirements of the machine. Before balancing any particular rotor, it is desirable to decide what balance criteria can be regarded as satisfactory. In this way, the balancing process can be made efficient and economical, but still satisfies the needs of the user.

Balancing is intended to achieve acceptable magnitudes of machinery vibration, shaft deflection and forces applied to the bearings caused by unbalance.

The ideal aim in balancing rotors with flexible behaviour would be to correct the local unbalance occurring at each elemental length by means of unbalance corrections at the element itself. This would result in a rotor in which the centre of mass of each elemental length lies on the shaft axis.

A rotor balanced in this ideal way would have no static and moment unbalance and no modal components of unbalance. Such a perfectly balanced rotor would then run satisfactorily at all speeds in so far as unbalance is concerned.

In practice, the unbalance can be distributed along the length of the rotor, but the balancing process is usually achieved by adding or removing masses in a limited number of correction planes. Thus, there is invariably some distributed residual unbalance after balancing, which is assumed to be within tolerance for the affected mode shapes.

It is necessary to reduce vibrations or oscillatory forces caused by the residual unbalance to acceptable magnitudes over the service speed range. Only in special cases is it sufficient to balance rotors with flexible behaviour for a single speed. It should be noted that a rotor, balanced satisfactorily for a given service speed range, can still experience excessive vibration if it has to run through a resonance speed to reach its service speed. Therefore, for passing through resonance speeds, the allowable vibration may be greater than that permissible at service speed.

Whatever balancing technique is used, the final goal is to apply unbalance correction distributions to minimize the unbalance effects at all speeds up to the maximum service speed, including start up and shut down and possible overspeed. In meeting this objective, it might be necessary to allow for the influence of modes with resonance speeds above the service speed range.

4.6 Provision for correction planes

The exact number of axial locations along the rotor that are needed depends to some extent on the particular balancing procedure which is adopted. For example, centrifugal compressor rotors are sometimes balanced as an assembly in the end planes only after each disc and the shaft have been separately balanced in a low-speed balancing machine. Generally, however, if the speed of the rotor is influenced by n flexural resonance speeds, which possibly include resonance speeds above the maximum service speed, then usually if low-speed balancing is carried out, $n + 2$ correction planes are needed along the rotor, if not, n planes can be used.

An adequate number of correction planes at suitable axial positions shall be included at the design stage. In practice, the number of correction planes is often limited by design considerations and *in-situ* balancing by limitations on accessibility.

4.7 Coupled rotors

When two rotors are coupled together, the complete unit has a series of resonance speeds and mode shapes. In general, these speeds are neither equal nor simply related to the resonance speeds of the individual, uncoupled rotors. Moreover, the deflection shape of each part of the coupled unit need not be simply related to any mode shape of the corresponding uncoupled rotor. Ideally, therefore, the unbalance distribution along two or more coupled rotors should be evaluated in terms of modal unbalances with respect to the coupled system and not to the modes of the uncoupled rotors.

For practical purposes, in most cases, each rotor is balanced separately as an uncoupled shaft and this procedure normally ensures satisfactory operation of the coupled rotors. The degree to which this technique is practicable depends, for example, on the mode shapes and the resonance speeds of the uncoupled and coupled rotors, the distribution of unbalance, the type of coupling and on the bearing arrangement of the shaft train. If further balancing *in-situ* is required, refer to Annex A.

5 Rotor configurations

Typical rotor configurations are shown in Table 1, their characteristics outlined and the recommended balancing procedures listed. Table 1 gives concise descriptions of the rotor characteristics. Full descriptions of these characteristics and requirements are given in the corresponding procedures in Clauses 6 and 7. These procedures are listed in Table 2.

Sometimes, a combination of balancing procedures can be advisable. If more than one balancing procedure could be used, they are listed in the sequence of increasing time and cost. Rotors of any configuration can always be balanced at multiple speeds (see 7.3) or sometimes, under special conditions, be balanced at service speed (see 7.4) or at a fixed speed (see 7.5).