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Mechanical vibration — Methods and criteria for the mechanical balancing of flexible rotors

Vibrations mécaniques — Méthodes et critères

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Contents

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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.

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.

International Standard ISO 11342 was prepared by Technical Committee ISO/TC 108, *Mechanical vibration and shock*, Subcommittee SC 1, *Balancing, including balancing machines*.

This first edition of ISO 11342 cancels and replaces ISO 5406:1980 and ISO 5343:1983.

Annexes A, B, C, D, E, F, G, H and J of this International Standard are for information only.

Introduction

The aim of balancing any rotor is satisfactory running when installed on site. In this context "satisfactory running" means that no more than an acceptable level of vibration is caused by the unbalance remaining in the rotor. In the case of a flexible rotor, it also means that no more than an acceptable magnitude of deflection occurs in the rotor at any speed up to maximum service speed.

Most rotors are balanced by their manufacturers prior to machine assembly because afterwards, for example, there may 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 on site is the aim, the balance quality of the rotor is usually initially assessed in a balancing facility. Satisfactory running on site is in most cases judged in relation to vibration from all causes, while in the balancing facility primarily once-per-revolution effects are considered.

Section 2 of this International Standard classifies rotors into groups in accordance with their balancing requirements and establishes in Section 3 methods of assessment of residual unbalance.

This International Standard also shows in Section 3 how criteria for use in the balancing facility may 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 International Standard shows how they may be derived from ISO 10816-1 and parts 1 to 4 of ISO 7919, if desired in terms of vibration, or from ISO 1940-1 if desired in terms of permissible residual unbalance.

ISO 1940-1 is concerned with the balance quality of rotating rigid bodies and is thus not directly applicable to flexible rotors because they may undergo significant bending deflection. However, in subclauses 2.3 and 3.4 of this International Standard, methods are presented for adapting the criteria of ISO 1940-1 to flexible rotors.

As this International Standard is complementary in many details to parts 1 and 2 of ISO 1940, it is recommended that, where applicable, they should be considered together.

Mechanical vibration — Methods and criteria for the mechanical balancing of flexible rotors

Section 1: General

1.1 Scope

This International Standard classifies rotors into groups in accordance with their balancing requirements, describes balancing procedures, specifies methods of assessment of the final state of unbalance, and gives guidance on balance quality criteria.

All rotors are classified into those which can be balanced by rigid rotor, modified rigid rotor, or high-speed (flexible rotor) balancing techniques.

Two methods are specified for evaluating the balance quality of a flexible rotor in a balancing facility before machine assembly: the first assesses the vibration level, and the second assesses the rotor residual unbalance. If the rotor balance tolerances suggested herein are achieved during correction in a balancing facility, the specified vibration limits of the assembled machine in service (see ISO 10816-1 and parts 1 to 4 of ISO 7919) will most probably be achieved. Accordingly, the criteria specified are those to be met when the rotor is tested in the balancing facility, but they are derived from those specified for the complete machine, when installed, or from values known to ensure satisfactory running of the rotor when it is installed.

As in the case of parts 1 and 2 of ISO 1940, this International Standard is not intended to serve as an acceptance specification for any rotor group, but rather to give indications of how to avoid gross deficiences and/or unnecessarily restrictive requirements. This International Standard may also 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 methods of manufacture and limits of unbalance, satisfactory running conditions can most probably be excepted.

There are situations in which an otherwise acceptably balanced rotor experiences an unacceptable vibration level *in situ*, owing to resonances. A resonant or near-resonant condition in a lightly damped structure can result in excessive vibratory response to a small unbalance. In such cases, it may be necessary to alter the natural frequency or damping of the structure rather than to balance to very low levels, which may not be maintainable over time.

The subject of structural resonances and modifications thereof is outside the scope of this International Standard.

The methods and criteria given are the result of experience with general industrial machinery. They may not be directly applicable to specialized equipment or to special circumstances. Therefore, there may be cases where deviations from this International Standard may be necessary[1)].

1.2 Normative references

The following standards contain provisions which, through reference in this text, constitute provisions of this International Standard. At the time of publication, the editions indicated were valid. All standards

1) Information on such exceptions is welcomed and should be communicated to the national standards body in the country of origin for transmission to the secretariat of ISO/TC 108.

are subject to revision, and parties to agreements based on this International Standard are encouraged to investigate the possibility of applying the most recent editions of the standards indicated below. Members of IEC and ISO maintain registers of currently valid International Standards.

ISO 1925:1990, *Mechanical vibration — Balancing — Vocabulary.*

ISO 1940-1:1986, *Mechanical vibration — Balance quality requirements of rigid rotors — Part 1: Determination of permissible residual unbalance.*

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

ISO 2041:1990, *Vibration and shock — Vocabulary.*

ISO 2953:1985, *Balancing machines — Description and evaluation.*

ISO 7919-1:1986, *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation — Part 1: General guidelines.*

ISO 7919-2:—[2], *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation — Part 2: Large land-based steam turbine-generator sets.*

ISO 7919-3:—[2], *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation — Part 3: Guidelines for coupled industrial machines.*

ISO 7919-4:—[2], *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation — Part 4: Guidelines for gas turbines.*

ISO 8821:1989, *Mechanical vibration — Balancing — Shaft and fitment key convention.*

ISO 10816-1:—[2], *Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts — Part 1: General guidelines.*

1.3 Definitions

For the purposes of this International Standard, the definitions relating to mechanical balancing given in ISO 1925 and many of the definitions relating to vibration given in ISO 2041 apply.

Definitions given in ISO 1925 relating to flexible rotors are given for information in annex H.

2) To be published.

Section 2: Balancing methods

2.1 Fundamentals of flexible rotor dynamics and balancing

2.1.1 Unbalance distribution

The rotor design and method of construction can significantly influence the magnitude and distribution of unbalance along a rotor. 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 local unbalances arising from shrink-fitted discs, couplings, etc.

Since the unbalance distribution along a rotor 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 flexible rotor than in a rigid rotor because it determines the degree to which any flexural mode of vibration is excited. Moreover, the effect of unbalance at any point along a rotor depends on the bending deflection of the rotor at that point.

The correction of unbalance in transverse planes along a rotor, other than those in which the unbalance occurs, may induce vibrations at speeds other than that at which the rotor was originally corrected. These vibrations may exceed specified tolerances, particularly at or near the flexural critical speeds.

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

2.1.2 Flexible rotor mode shapes

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 isotropic bearings, are rotating plane curves. Typical curves for the three lowest principal modes for a simple rotor supported in flexible bearings near to its end are illustrated in figure 1.

For a damped rotor/bearing system, the flexural modes may be space curves rotating about the shaft axis, especially in the case of substantial damping, arising perhaps from fluid-film bearings. A possible substantially damped second mode is 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 must be stressed 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.

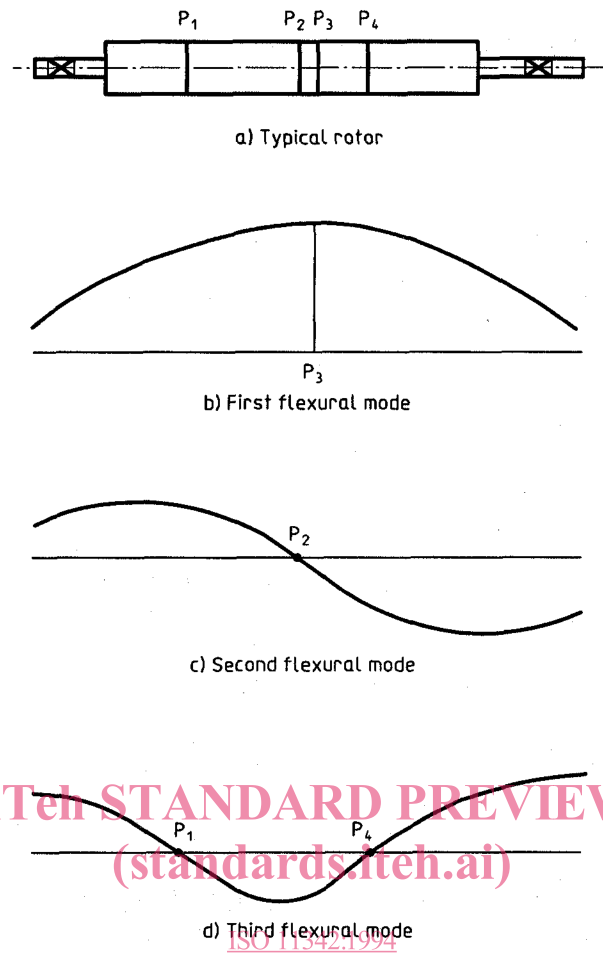
2.1.3 Response of a flexible rotor to unbalance

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 critical speed, it is usually the mode associated with this critical speed which dominates the deflection of the rotor. The degree to which large amplitudes of rotor deflection occur in these circumstances is determined by:

a) the magnitude of the modal unbalances;

b) the proximity of the associated critical speeds to the running speeds; and

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



NOTE — P_1 to P_4 are correction planes.

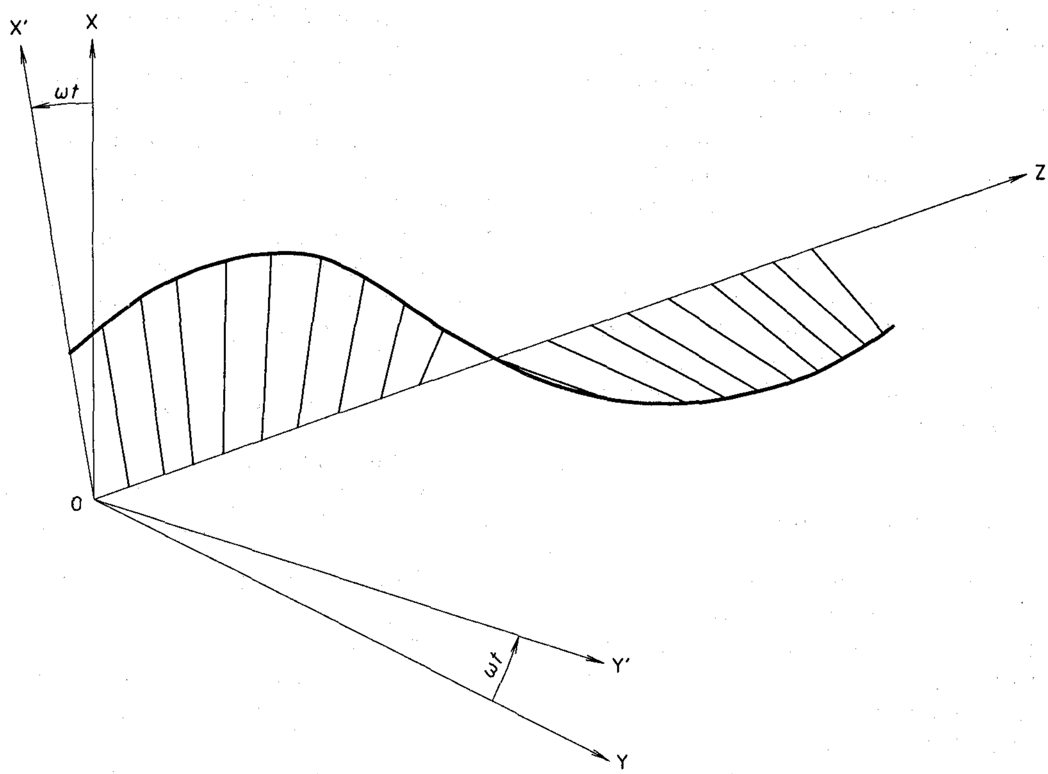
Figure 1 — Typical mode shapes for flexible rotors on flexible supports

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 International Standard.

The modal unbalances for a given unbalance distribution are a function of the flexible rotor modes. Moreover 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. Consider an example in which the curves of figure 1 b) to 1 d) are mode shapes for the rotor in figure 1 a). A correction mass attached to the rotor in figure 1 a) in the plane P_2 will produce no change in response in the second mode. Similarly, a correction mass attached in either plane P_1 or P_4 will not affect the response in the third mode. Conversely, a correction mass in plane P_3 will produce the maximum effect on the first mode.

2.1.4 Aims of flexible rotor balancing

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 economic, but still satisfy the needs of the user.



NOTE — OX, OY and OZ are fixed axes. OX' and OY' are axes rotating about OZ at speed ω.

Figure 2 — Possible damped second-mode shape

Balance criteria are specified to achieve the following:

a) acceptable values of machinery vibration and shaft deflection;

b) acceptable values of unbalance forces applied to the bearings.

The ideal aim in balancing flexible rotors 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 couple 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 necessary reduction in the unbalance forces is usually achieved by adding or removing masses in a limited number of correction planes. There will invariably be some distributed residual unbalance after balancing.

Vibrations or oscillatory forces due to the residual unbalance must be reduced to acceptable magnitudes over a range of speeds, including one or more critical speeds. Only in special cases is it sufficient to balance flexible rotors for a single speed. It should be noted that a rotor, balanced satisfactorily for a given service speed range, may still experience excessive vibration if it has to run through a critical speed to reach its service speed. Balancing a rotor according to its mode shapes is not an end in itself. Whatever balancing technique is used, the final goal is to apply unbalance correction distributions to minimize the unbalance effects up to the service speed, and possible overspeed.

2.1.5 Provision for correction planes

Rotors are often balanced mode by mode. In this process, correction masses are located along the rotor, so that at each stage in the balancing procedure the new correction masses do not significantly disturb modes already balanced.

The exact number of axial locations along the rotor that are needed for this process depends to some extent on the particular balancing procedure which is adopted. For example, centrifugal compressor rotors are sometimes assembly-balanced 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 approaches or exceeds its nth critical speed, then at least n and commonly $(n + 2)$ correction planes are needed along the rotor.

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

2.1.6 Rotors coupled together

When two rotors are coupled together, the complete unit will have a series of critical speeds and mode shapes. In general, these speeds are neither equal to nor simply related to the critical 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. In theory, 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, it is often necessary that each rotor be balanced separately as an uncoupled shaft. In many cases, this procedure will ensure satisfactory operation of the coupled rotors. The degree to which this technique is practicable depends, for example, on the mode shapes and the critical speeds of the uncoupled and coupled rotors, and the distribution of unbalance.

If further balancing on site is required, reference should be made to annex A.

2.2 Classification

For the purposes of this International Standard, rotors are divided into five main classes as shown in 2.2.1 to 2.2.5 and in table 1. Each class requires different balancing techniques. A procedure to determine if a rotor is rigid or flexible is given in annex E.

2.2.1 Class 1: Rigid rotors

A rotor is considered to be rigid when its unbalance can be corrected in any two (arbitrarily selected) planes. After the correction, its residual unbalance does not change significantly (relative to the shaft axis) at any speed up to the maximum service speed and when running under conditions which approximate closely to those of the final supporting system. Rotors of this type can be corrected by rigid-rotor balancing methods (see ISO 1940-1).

2.2.2 Class 2: Quasi-rigid rotors

A rotor that cannot be considered rigid but that can be balanced using modified rigid-rotor balancing techniques is considered to be a quasi-rigid rotor.

Class 2 rotors are subdivided (see table 1) into:

a) rotors in which the axial distribution of unbalance is known (classes 2a, 2b, 2c and 2d; also class 2e in which the axial distribution is partly known);

b) rotors in which the axial distribution of unbalance is not known (classes 2f, 2g and 2h).

The subdivision of class 2 rotors shows the many reasons why rotors can often be balanced satisfactorily at low speed as rigid rotors even though they are flexible. Some rotors will fit into more than one category of the subdivision.

2.2.3 Class 3: Flexible rotors

A rotor that cannot be balanced using modified rigid-rotor balancing techniques but instead requires the use of high-speed balancing methods is considered to be a flexible rotor.

Class 3 is subdivided (see table 1) because the balancing techniques, criteria and bearing requirements may differ substantially for different rotors.

2.2.4 Class 4

A rotor that could fall into class 1, 2 or 3 but has in addition one or more components that are themselves flexible or flexibly attached is considered to be a class 4 rotor.

A subdivision of class 4 rotors is indicated in 2.4.2.