INTERNATIONAL STANDARD

ISO 22266-1

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Mechanical vibration — Torsional vibration of rotating machinery —

Part 1:
Land-based steam and gas turbine generator sets in excess of 50 MW

Vibrations mécaniques — Vibration de torsion des machines tournantes —

Partie 1: Groupes électrogènes à turbines à vapeur et à gaz situés sur terre et excédant 50 MW

Reference number
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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 22266-1 was prepared by Technical Committee ISO/TC 108, *Mechanical vibration, shock and condition monitoring*, Subcommittee SC 2, *Measurement and evaluation of mechanical vibration and shock as applied to machines, vehicles and structures*.

ISO 22266 consists of the following parts, under the general title *Mechanical vibration — Torsional vibration of rotating machinery*:

— *Part 1: Land-based steam and gas turbine generator sets in excess of 50 MW*

Introduction

During the 1970s, a number of major incidents occurred in power plants that were deemed to be caused by or that were attributed to torsional vibration. In those incidents, generator rotors and some of the long turbine blades of the low-pressure (LP) rotors were damaged. In general, they were due to modes of the coupled shaft and blade system that were resonant with the grid excitation frequencies. Detailed investigations were carried out and it became apparent that the mathematical models used at that time to predict the torsional natural frequencies were not adequate. In particular, they did not take into account with sufficient accuracy the coupling between long turbine blades and the shaft line. Therefore, advanced research work was carried out to analyse the blade-to-discs-to-shaft coupling effects more accurately, and branch models were developed to account properly for these effects in shaft system frequency calculations.

In the 1980s, dynamic torsional tests were also developed in the factory to verify the predicted dynamically coupled blade-disc frequencies for the low-pressure rotors. These factory tests were very useful in identifying any necessary corrective actions before the product went in service. However, it is not always possible to test all the rotor elements that comprise the assembly. Hence, unless testing is carried out on the fully assembled train on site, some discrepancy could still exist between the overall system models and the actual installed machine.

There is inevitably some uncertainty regarding the accuracy of the calculated and measured torsional natural frequencies. It is therefore necessary to design overall system torsional frequencies with sufficient margin from the grid system frequencies to compensate for such inaccuracies. The acceptable margins will vary depending on the extent to which any experimental validation of the calculated torsional frequencies is carried out. The main objective of this part of ISO 22266 is to provide guidelines for the selection of frequency margins in design and on the fully coupled machine on site.

In general, the presence of a natural frequency is only of concern if it coincides with an excitation frequency within the margins defined in this part of ISO 22266 and has a modal distribution allowing energy to be fed into the corresponding vibration mode. If either of these conditions is not satisfied, the presence of a natural frequency is of no practical consequence, i.e. a particular mode of vibration is of no concern if it cannot be excited. In the context of this part of ISO 22266, the excitation is due to variations in the electromechanical torque, which is induced at the air gap of the generator. Any shaft torsional modes that are insensitive to these induced excitation torques do not present a risk to the integrity of the turbine generator, regardless of the value of the natural frequency of that mode (see 4.2 and 5.2).

Mechanical vibration — Torsional vibration of rotating machinery —

Part 1: Land-based steam and gas turbine generator sets in excess of 50 MW

1 Scope

This part of ISO 22266 provides guidelines for applying shaft torsional vibration criteria, under normal operating conditions, for the coupled shaft system and long blades of a turbine generator set. In particular, these apply to the torsional natural frequencies of the coupled shaft system at line and twice line frequencies of the electrical network to which the turbine generator set is connected. In the event that torsional natural frequencies do not conform with defined frequency margins, other possible actions available to vendors are defined.

This part of ISO 22266 is applicable to

— land-based steam turbine generator sets for power stations with power outputs greater than 50 MW and normal operating speeds of 1 500 r/min, 1 800 r/min, 3 000 r/min and 3 600 r/min, and

— land-based gas turbine generator sets for power stations with power outputs greater than 50 MW and normal operating speeds of 3 000 r/min and 3 600 r/min.

Methods currently available for carrying out both analytical assessments and test validation of the shaft system torsional natural frequencies are also described.

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 2041:—[1], *Mechanical vibration, shock and condition monitoring — Vocabulary*

ISO 2710-1, *Reciprocating internal combustion engines — Vocabulary — Terms for engine design and operation*

ISO 2710-2, *Reciprocating internal combustion engines — Vocabulary — Terms for engine maintenance*

1) To be published. (Revision of ISO 2041:1990)

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 2041, ISO 2710-1 and ISO 2710-2 and the following apply.

3.1
set
assembly of one or more elements such as high-pressure, intermediate-pressure, low-pressure turbines and generator and exciter elements

3.2
shaft system
fully connected assembly of all the rotating components of a set

NOTE 1 Figure 1 shows an example.

NOTE 2 When the torsional natural frequencies are calculated, it is the complete shaft system that is considered.

3.3
torsional vibration
oscillatory angular deformation (twist) of a rotating shaft system

3.4
torsional vibration magnitude
maximum oscillatory angular displacement measured in a cross section perpendicular to the axis of the shaft system between the angular position considered and a given arbitrary reference position

3.5
natural frequency
frequency of free vibration of an undamped linear vibration system

NOTE 1 The same definition is given for natural frequency of a mechanical system in ISO 2041.

NOTE 2 It is usually not necessary to calculate the natural frequency for a damped system, which is

$$\omega_{\mathrm{d}} = \omega_{\mathrm{n}} \sqrt{1-\eta^2}$$

where η is the damping ratio.

3.6
modal vector
relative magnitude for the whole section, where the system is vibrating at its associated natural frequency and an arbitrary cross section of the system is chosen as a reference and given a magnitude of unity

3.7
torsional mode shape
shape produced by connecting the modal vector magnitudes at each section

3.8
vibratory node
point on a mode shape where the relative modal vector magnitude is equal to zero

3.9
natural mode of torsional vibration
torsional mode shape which is produced when the shaft is vibrating at its natural frequency

EXAMPLE First mode of vibration or one-node mode of vibration, second mode of vibration or two-node mode of vibration.

NOTE Figure 2 shows examples.

3.10
excitation torque
torsional torque produced by the generator, exciter or driven components that excites torsional vibration of the shaft system

3.11
harmonic
each term of the Fourier series of the excitation or response signal

3.12
all-in-phase mode
mode of vibration in which all blades in a particular row vibrate in phase with one another

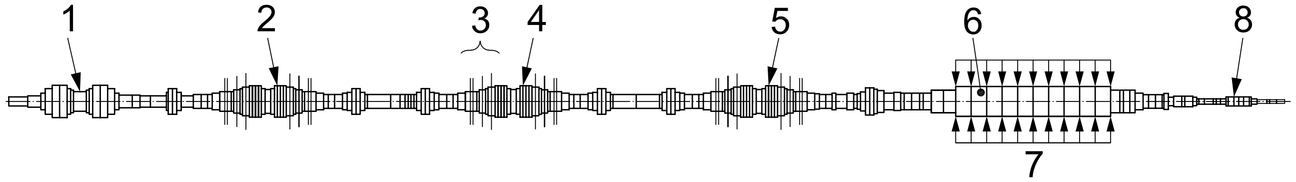
NOTE When the rotor disc and the blades couple under dynamic conditions, the combined system produces several new “all-in-phase” frequencies that are different from the individual disc and blade frequencies (see Figure 3). These modes are often referred to as *zero-nodal diameter* or “umbrella” modes.

3.13
resonant speed
characteristic speed at which resonances of the shaft system are excited

EXAMPLE The shaft speed at which the natural frequency of a torsional vibration mode equals the frequency of one of the harmonics of the excitation torques.

NOTE The same definition is given for resonant speed/critical speed in ISO 2041.

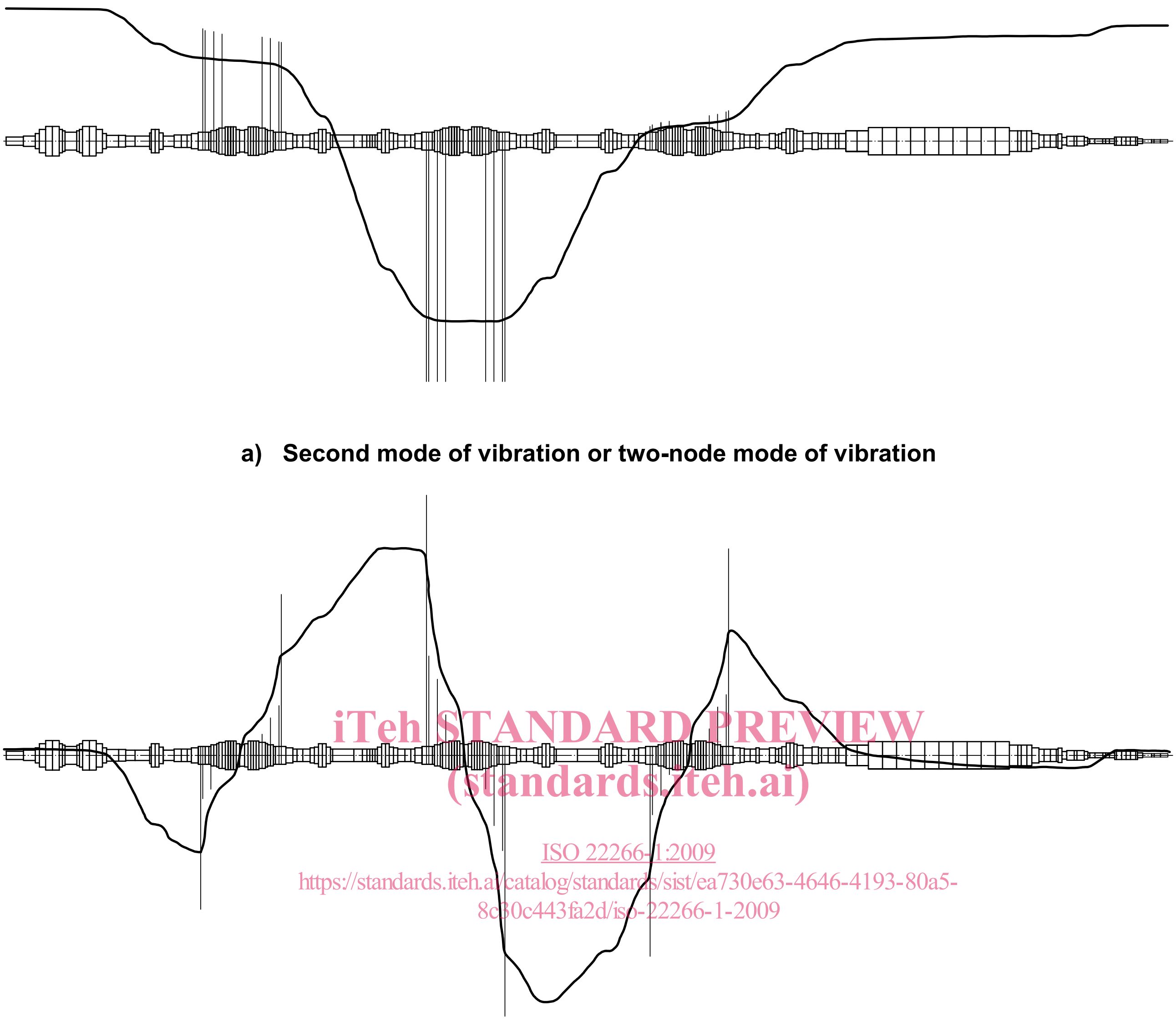
3.14
additional torsional stress
stress due to the torsional vibrations of a given excitation harmonic superimposed on the torsional stress corresponding to the mean torque transmitted in the given section of the shaft system being considered



Key

1 high-pressure (HP) rotor
2 low-pressure (LP) rotor 1
3 blades
4 LP rotor 2
5 LP rotor 3
6 generator rotor
7 excitation torque applied
8 exciter

Figure 1 — Six-rotor steam turbine generator system

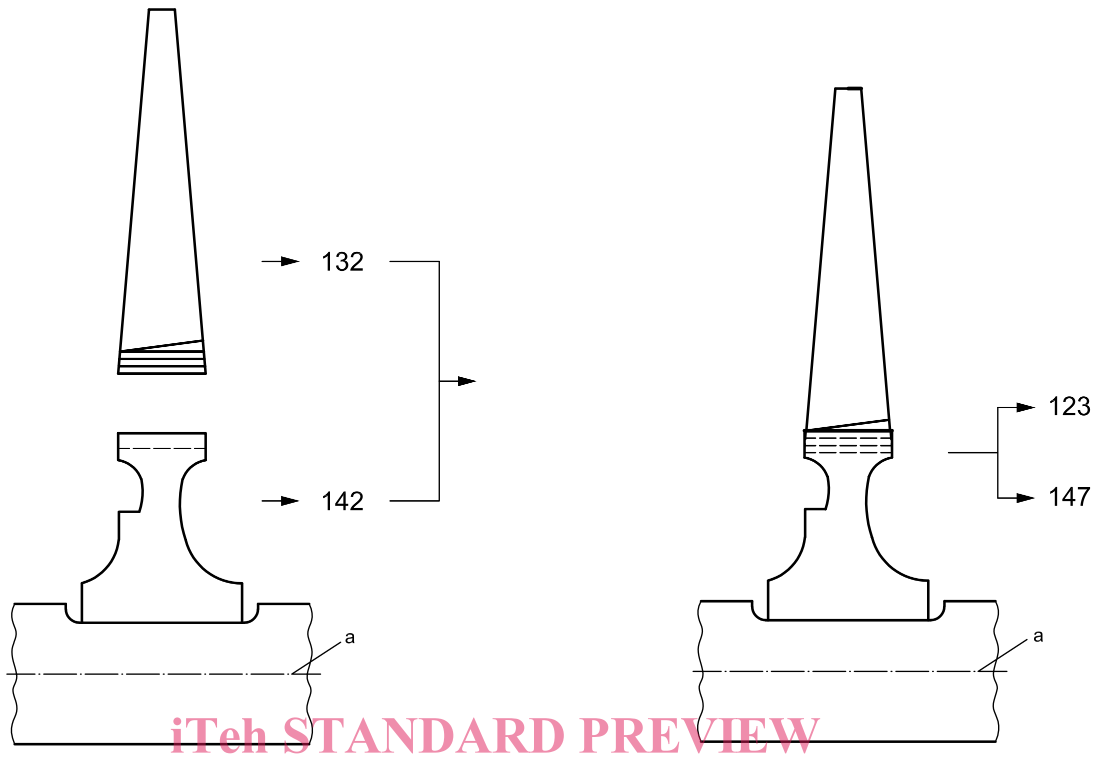


a) Second mode of vibration or two-node mode of vibration

b) Sixth mode of vibration or six-node mode of vibration

Figure 2 — Typical torsional mode shapes of the shaft system

Frequencies in hertz



a) Uncoupled frequencies of separated blade and disc **b) Coupled frequencies of blade-disc assembly**

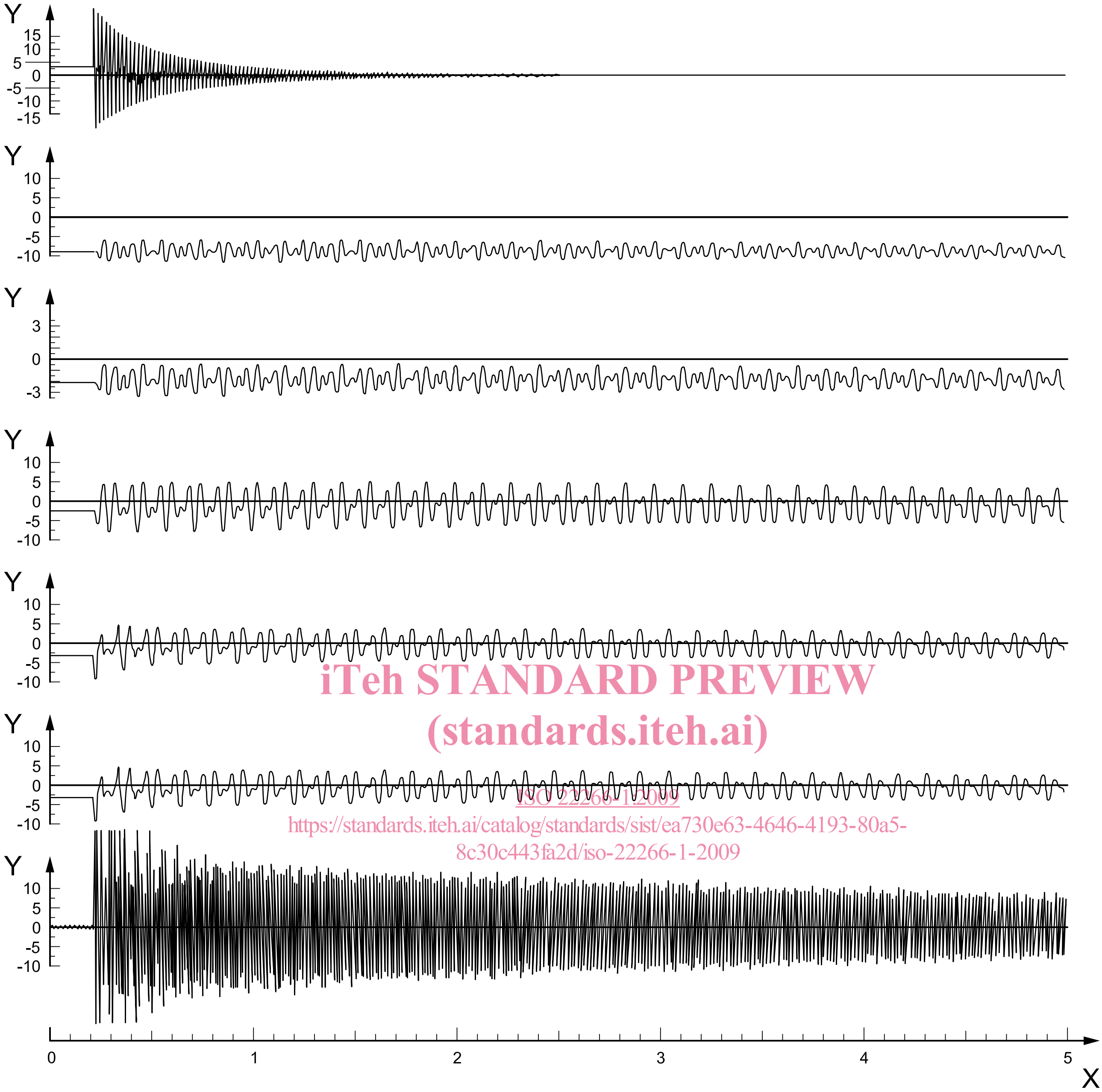
[a] Rotor central axis

Figure 3 — Schematic illustration of blade-disc dynamic coupling

3.15
synthesized torsional stress
dynamic torsional stress generated at a section of the shaft system given by the vector sum of all the harmonics of the excitation torques, taking into account both the magnitude and phase of the stress generated by each harmonic

NOTE 1 See Figure 4, in which the six upper plots show, for a particular point on the shaft, the time history of the additional torsional stress for each of the first six excitation harmonics. The lowest plot is the combined effect of vectorially adding all of the individual harmonics.

NOTE 2 Mean torque is not used when elaborating the synthesized torsional stress.



Key

X time, s

Y torsional stress

Figure 4 — Typical dynamic torsional stress

3.16
prohibited frequency range
frequency range over which the stress caused by the torsional vibration exceeds the stress value permitted for continuous operation

NOTE Although continuous operation in this frequency range is forbidden, passing through it in transient operation is permissible, provided that it offers no danger of accumulated damage to the shaft system.

