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Mechanical vibration — Torsional vibration of rotating machinery —

Part 1: Evaluation of steam and gas turbine generator sets due to electrical excitation

Vibrations mécaniques — Vibration de torsion des machines tournantes —

Partie 1: Évaluation des groupes électrogènes à turbine à vapeur et à gaz due à l'excitation électrique

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

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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 of the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT), see www.iso.org/iso/foreword.html.

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

This second edition cancels and replaces the first edition (ISO 22266-1:2009), which has been technically revised.

The main changes are as follows:

— terms and definitions revised to account for definitions given in other standards;

— evaluation concept refined and substantiated, contradictory statements removed;

— guidance on modelling uncertainties added;

— annex enhanced to give guidance on measurement equipment for monitoring torsional vibration;

— wording at some instances revised in order to make the content unambiguous;

A list of all parts of the ISO 22266 series can be found on the ISO website.

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

This corrected version of ISO 22266-1:2022 incorporates the following correction:

— In the second paragraph of Annex B, the verbal form "shall" was reverted back to "must" in the third sentence to read: "To cause significant torsional vibration, the excitation frequency must be close to a torsional natural frequency of the shaft train for a sufficiently long time for the vibration mode to become established."

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 rotor torsional vibration. In those incidents, generator rotors and some of the long elastic turbine blades of the LP rotors were damaged. In general, the incidents were due to vibration modes of the coupled shaft and blade system that were resonant with the grid electrical excitation frequencies. Detailed investigations were carried out and it became apparent that the mathematical models used at that time to predict rotor torsional natural frequencies were not adequate. In particular, they did not take into account, with sufficient accuracy, the coupling between long elastic turbine blades and the shaft line. Therefore, advanced research work was carried out to analyse the blade-to-disc-to-shaft coupling effects more accurately and branch models were developed to account properly for these effects in shaft train torsional natural frequency calculations.

In the 1980s, torsional factory tests were developed to verify the predicted torsional natural frequencies of LP rotors. These factory tests were very useful in identifying any necessary corrective actions before the product went into service. However, it is not always possible to test all the elements that comprise the assembled rotor. Hence, unless testing is carried out on the shaft train on site, some discrepancies could still exist between the overall system model and the installed machine.

There is inevitably some uncertainty regarding the accuracy of the calculated and measured torsional natural frequencies. It is therefore necessary to design shaft train torsional natural frequencies with sufficient margin from the grid system frequencies to compensate for such inaccuracies, unless the modes are insensitive to excitation torques. Acceptable margins will vary depending on the extent to which any experimental validation of the calculated torsional frequencies is carried out. The margins should also take into account the sensitivity of the torsional natural frequencies and the modal excitability with respect to modelling uncertainties. The main objective of this document is to provide guidelines for the selection of frequency margins during the design stage and on the fully coupled shaft train on site.

In general, the presence of a torsional natural frequency is only of concern if it coincides with an excitation frequency and has a modal distribution allowing energy to be fed into the corresponding vibration mode (resonance). If either of these conditions is not satisfied, the presence of a natural frequency is of no practical consequence (e.g. a particular mode of vibration is of no concern if it cannot be excited). In the context of this document, the excitation is due to variations in the electromechanical torque, induced at the air gap of the generator. Any shaft train 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.

Mechanical vibration — Torsional vibration of rotating machinery —

Part 1: Evaluation of steam and gas turbine generator sets due to electrical excitation

1 Scope

This document provides guidelines for the assessment of torsional natural frequencies and component strength, under normal operating conditions, for the coupled shaft train, including long elastic rotor blades, of steam and gas turbine generator sets. In particular, the guidelines apply to the torsional responses of the coupled shaft train at grid and twice grid frequencies due to electrical excitation of the electrical network to which the turbine generator set is connected. Excitation at other frequencies (e.g. subharmonic frequencies) are not covered in this document.

No guidelines are given regarding the torsional vibration response caused by steam excitation or other excitation mechanisms not related to the electrical network.

Where the shaft cross sections and couplings do not fulfil the required strength criteria and/or torsional natural frequencies do not conform with defined frequency margins, other actions shall be defined to resolve the problem.

The requirements included in this document are applicable to

a) steam turbine generator sets connected to the electrical network, and

b) gas turbine generator sets connected to the electrical network.

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

NOTE Radial (lateral, transverse) and axial vibration of steam and/or gas turbine generator sets is dealt with in ISO 20816-2.

2 Normative references

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

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

ISO 11086, *Gas turbines — Vocabulary*

IEC 60050-602, *International Electrotechnical Vocabulary — Chapter 602: Generation, transmission and distribution of electricity – Generation*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 2041, ISO 11086, IEC 60050-602 and the following apply.

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

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

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

3.1
elastic blade
blade which is fastened to a shaft or disc and has properties which has at least one natural frequency affecting the calculation of the torsional natural frequencies of the shaft train

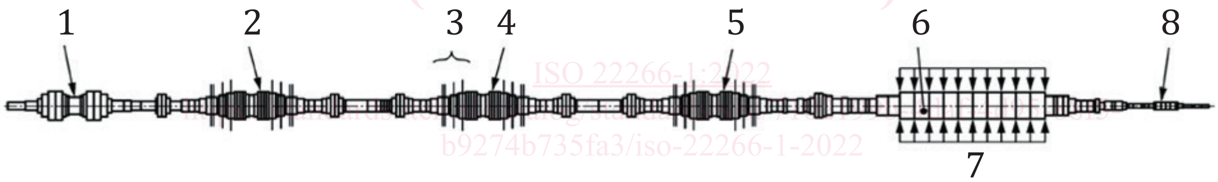
3.2
shaft
mainly cylindrical rotatable component carrying one or more elements (e.g. disc, coupling, blade)

3.3
rotor
rotating assembly (e.g. HP, IP, LP steam turbine, gas turbine, generator or exciter) comprising of one or more elements (e.g. shaft, disc, coupling, blade)

Note 1 to entry: Typically, several rotors are assembled onto one shaft train of the turbine generator set.

3.4
shaft train
fully connected assembly of all rotors typically comprising of at least one driving rotor and one generator rotor (see Figure 1)

Note 1 to entry: When the torsional natural frequencies are calculated, it is the complete shaft train that is considered.



Key

1 HP rotor
2 LP rotor 1
3 blades
4 LP rotor 2
5 LP rotor 3
6 generator rotor
7 excitation torque applied
8 exciter rotor

Figure 1 — Shaft train consisting of six rotors

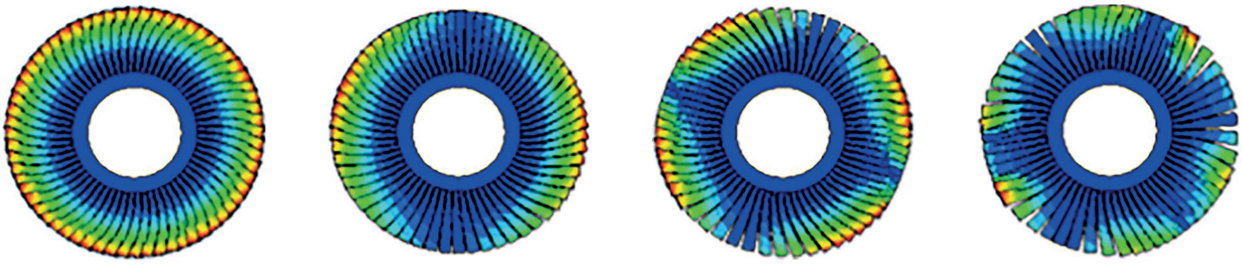
3.5
torsional vibration magnitude
maximum oscillatory angular displacement measured in a cross-section perpendicular to the rotation axis of the shaft train

3.6
excitation torque
torque produced by the generator, exciter or driven components that excites the torsional vibration mode(s) of the shaft train

3.7
zero-nodal diameter mode
mode of vibration in which all elastic blades in a particular row vibrate in phase with one another (see Figure 2)

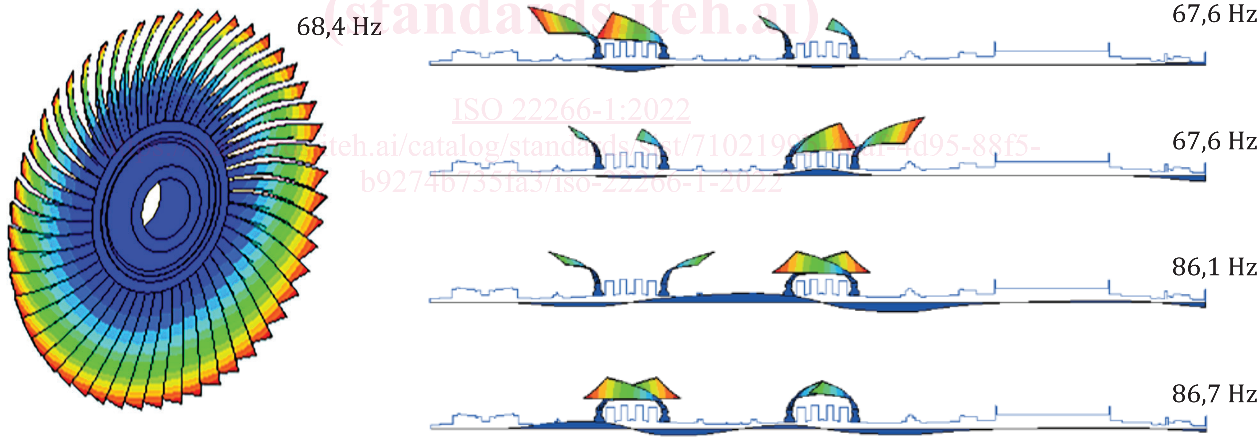
Note 1 to entry: When the shaft/disc and the elastic blades couple under dynamic conditions, the combined system produces several frequencies with zero-nodal diameter blade mode participation that are different from the individual shaft and blade frequencies (see Figure 3). These modes are often referred to as all-in-phase or umbrella modes.

Note 2 to entry: To calculate blade row natural frequencies, a section of the shaft or disc should be included in the blade row model.



Zero-nodal diameter **One-nodal diameter** **Two-nodal diameter** **Three-nodal diameter**

Figure 2 — Schematic illustration of different nodal diameters



a) Uncoupled zero- nodal diameter mode of separated bladed disc

b) Coupled modes of shaft-disc-blade assembly

Note 3 to entry In frequencies of the shaft-disc-blade assembly, the first two modes occur at the same frequency which is due to the given decimals. Identical natural frequencies are theoretically possible if the shaft line is totally symmetric from left to right. In practice this is never the case and there will always be a small difference in the frequencies.

Figure 3 — Schematic illustration of shaft-disc-blade dynamic coupling

3.8
static torsional stress
stress in the section of the shaft train being considered, due to the mean torque transmitted

3.9
dynamic torsional stress
stress in the section of the shaft train being considered, due to the torsional vibrations, being superimposed on the static torsional stress transmitted

4 Abbreviated terms and symbols

4.1 Abbreviated terms

AC	alternating current
DC	direct current
HP	high-pressure
IP	intermediate-pressure
LP	low-pressure
NF	natural frequency
OEM	original equipment manufacturer
SSR	sub-synchronous resonance
SSTI	sub-synchronous torsional interaction

4.2 Symbols

A_l	lower grid frequency variation
A_u	upper grid frequency variation
$B_{\mathrm{l},i}$	mode specific lower separation margin of mode i
$B_{\mathrm{u},i}$	mode specific upper separation margin of mode i
$C_{\mathrm{l},i}$	mode specific lower calculation uncertainty of mode i
$C_{\mathrm{u},i}$	mode specific upper calculation uncertainty of mode i
$D_{\mathrm{l},i}$	mode specific lower reduced calculation uncertainty of mode i
$D_{\mathrm{u},i}$	mode specific upper reduced calculation uncertainty of mode i
i	mode number
x	grid frequency multiplication factor
Ω	rotating frequency
Ω_n	nominal rotating frequency
Ω_e	nominal grid frequency
$\Omega_{\mathrm{e},1}$	grid frequency axis
$\Omega_{\mathrm{e},2}$	twice grid frequency axis
$\omega_i\,(\Omega)$	calculated natural frequency of mode i (can be speed depending)
$\tilde{\omega}_i\,(\Omega)$	measured natural frequency of mode i (can be speed depending)

5 Shaft train modelling and uncertainties

5.1 General

In view of the possible excitation from the electrical grid, it is necessary to design the overall system torsional natural frequencies with regard to both the grid and twice grid system frequencies. For those modes that can be excited by torsional oscillation of the generator and are evaluated to be critical to the integrity of the shaft train, there shall be sufficient frequency margin from both the grid and twice grid system frequencies. This is the primary consideration for avoiding any torsional vibration issues on large turbine generators.

These parameters shall be taken into account when defining the frequency margin:

a) calculation uncertainty due to inaccuracies in the mathematical models used;

b) experimental validation of the system torsional natural frequencies at nominal rotating frequency;

c) the required margin between the shaft train torsional natural frequencies and excitation frequencies (grid and twice grid frequencies);

d) any specified/experienced grid frequency excursions;

e) operating temperature effects.

Mechanical parts (e.g. shrunk-on couplings, coupling bolts and turbine blades) that are connected to the shaft can contribute to system torsional vibration if they are not adequately designed for strength and/or tuned to have natural frequencies away from grid frequencies. 5.2 gives details regarding the modelling of mechanical parts.

Severe torsional vibration can lead to plastic deformation in the shaft train resulting in material fatigue which, in the worst case, can lead to cracking in the rotor components (e.g. shaft, blades couplings). Depending on the extent of the deformation, the operating behaviour of the turbine generator set can be permanently affected.

5.2 Modelling of the shaft train and the electrical system

5.2.1 General

Torsional vibration in the shaft train is most commonly excited by variations in the electromechanical torque induced at the air gap of the generator but may also be induced by rotor-stator interactions in the turbine generator system and by fluid-structure interactions in the turbine.

In reality, the turbine generator set and the electrical system to which it is connected form a coupled electro-mechanical system. In order to calculate the electro-mechanical torque induced at the air gap of the generator, the coupled electro-mechanical system is split into separate mechanical and electrical systems, which are usually modelled independently.

The model of the electrical system typically contains only basic information of the mechanical system (e.g. total shaft train inertia or lumped mass model of the shaft train with a few degrees of freedom). With this model, the air gap torque acting on the generator rotor is calculated and used as the excitation input for the complete model of the mechanical system. The mechanical model is used to calculate the system natural frequencies and the stress and fatigue caused by the air gap torque excitation.

Separate modelling is suitable for load cases where the electrical and mechanical systems do not or only marginally interact with each other. This is the situation for load cases exciting the shaft train at grid and twice grid frequency (e.g. out-of-phase synchronization, load unbalance). However, it is not valid for load cases with strong interaction (e.g. sub-synchronous resonance) where the phenomenon cannot be modelled or modelled only with poor accuracy.

When the turbine generator set is operating under ideal steady state conditions involving balanced three-phase currents and voltages, the effects of higher harmonics are negligible and the electromagnetic torque applied to the rotor in the generator air gap is essentially a constant, non-varying torque that transfers the turbine mechanical power through the generator and electrically to the power system. Under such ideal conditions, there will typically be little or no rotor torsional vibration. Torsional vibrations occur as a result of transient or unbalanced steady state power system disturbances which act to induce variations in the generator air gap magnetic field and, hence, the output torque.

5.2.2 Elastic blade modelling

The zero-nodal diameter mode shape of elastic blade rows are such that all blades in a row vibrate in phase with one another. A tangential force acting on the shaft train can therefore excite blade modes having a tangential component. In addition, modal interaction takes place between the blades, discs and shaft such that the resulting natural frequencies of the assembled rotor or shaft train are different from those of the individual components (see Figure 3). It is important to note that for other blade modes with non-zero-nodal diameters, different sectors of the blade row vibrate in anti-phase to those of adjacent sectors and are therefore not excited by torsional oscillation of the shaft train.

For short- and medium-height blade rows (e.g. of HP/IP turbines, first several stages of LP turbines or last several stages of gas turbine compressors), the frequencies of the lowest zero-nodal diameter modes are generally far away from the frequencies of interest for torsional analysis. Therefore, when calculating the natural frequencies of the shaft train, such blades can be considered as rigid and only their torsional inertias need be taken into account.

For longer blades (e.g. the last and penultimate stages of the LP turbine or the first gas turbine compressor stage), the frequencies of the zero-nodal diameter modes can be within the range of, or sufficiently close to, the grid and/or the twice grid frequency in order to significantly affect the resulting system modes, which can then become critical as far as torsion is concerned. These modes interact with those of the other components in such a way that additional coupled modes are introduced with various combinations of blade vibration in phase and anti-phase with the shaft train. Under adverse conditions, such modes could amplify shaft/blade stresses due to external torques arising from grid disturbances. Consequently, when calculating the natural frequencies of the shaft train and blades, it is necessary to model the long blades elastically to fully replicate the zero-nodal diameter (all-in-phase) modes of them.

If the lowest zero-nodal diameter mode of the blade row and disc (or shaft section at the blade row location for drum type rotors) is less than 2,5 times the nominal grid frequency of the electrical grid system (e.g. 125 Hz in countries where the nominal grid frequency is 50 Hz and 150 Hz in countries where the nominal grid frequency is 60 Hz), consideration shall be given to modelling the blade elastically.

Otherwise, the blades can be modelled by their torsional inertia and it is only necessary to lump the total inertia of a blade row at the appropriate point in the shaft/disc model.

As the centrifugal loading of rotating blades is speed dependent, their natural frequencies are also speed dependent. Therefore, if long blades are modelled elastically, natural frequencies of the overall shaft train become speed dependent as well. Where the blade model does not consider their speed dependency, it should model the natural frequency at nominal rotating speed. In this case, margins B and C shall be applied to the calculated natural frequencies at nominal rotating speed (see 6.2).

5.2.3 Modelling generator rotor windings

Detailed knowledge of the generator rotor structural design is needed for accurately modelling its stiffness. Effects of the rotor body section with its copper windings and wedges shall also be taken into account.

5.2.4 Grid/excitation modelling

To calculate the excitation torque acting on the generator at the winding section, it is common practice to use analytical short circuit equations or numeric network models. Typically, and as long as all relevant system parameters are known and allow calculation of the air gap torque for load cases where no analytical equations are available, the numeric network models have a higher accuracy.

Based on the individual torsional mode shapes of the shaft train in the area of the generator rotor windings where the air gap torque acts on the shaft train, it is possible that some torsional natural frequencies can be excited by the SSR/SSTI phenomena during operation. This behaviour is based on the interaction between one or more natural frequencies of the mechanical system and one or more natural frequencies of the electrical system. To analyse SSR/SSTI phenomena a more detailed numerical model of the grid system, including an appropriate representation of the relevant shaft train modes (e.g. lumped mass model of the shaft train with a few degrees of freedom), is required. Modelling and assessment of SSR/SSTI phenomena is beyond the scope of this document.

5.2.5 Damping modelling

The overall damping of the electro-mechanical coupled system depends on a large number of parameters which are typically known only to a very limited extent. Consequently, damping ratios reported in literature vary considerably from approximately 0,01 % to 1,0 %[4]. However, calculations shall be performed with conservative damping ratios typically being smaller than 0,1 %. Within a vibration event damping values can vary with time and load[5].

Typically, damping is chosen to be proportional to mass and stiffness properties or modal damping values are used. No general guidance can be given on the modelling approach and damping values to be used as damping values depend on design of the rotor, manufacturing accuracy and electrical and grid conditions which can vary during operation.

5.2.6 Gear box modelling

Gear boxes couple the lateral, torsional and, for single helical gears, the axial vibrations of shaft trains resulting in interaction between lateral and torsional dynamics. In this case, journal bearings can provide considerable damping for torsional vibration. Due to gear teeth interaction, the stiffness of the gear is dependent on the angular position of the shafts and on the torque being transmitted. Taking these effects into account can lead to very complex non-linear models being needed resulting in a tremendous effort to evaluate the dynamic behaviour of the shaft train.

In many cases (e.g. if flexible couplings are used), the torsional-lateral-axial interaction can be ignored. In this case the gear box model shall take the gear ratio and stiffness and inertia properties of the gear into account.

5.2.7 Flexible coupling modelling

If there are large angular alignment differences between the two coupling flanges (e.g. with cardan joints for flexible couplings) the input rotation is non-linearly transformed into the output rotation. However, for shaft trains the alignment differences are very small and hence non-linearities can be neglected and linearly modelling flexible couplings is sufficient.

The stiffness of the flexible coupling can significantly affect torsional natural frequencies and shall be determined with care. Depending on mode shapes, tuning of torsional natural frequencies can be easily achieved by changing the stiffness of the flexible coupling. Typically, the inertia of the flexible coupling is significantly smaller when compared to the rest of the shaft train and changes to it have a relatively insignificant effect on the overall torsional natural frequencies.

5.3 Design element uncertainties

In the shaft train torsional model, there are some components and design elements that typically have larger modelling uncertainties than others. The list gives an overview of typical modelling uncertainties:

a) Rotor and shaft train joint and interface uncertainties:

   1) shrink fit values;