INTERNATIONAL STANDARD

ISO 22266-1

Second edition
2022-05

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

Reference number
ISO 22266-1:2022(E)

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Published in Switzerland

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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.

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

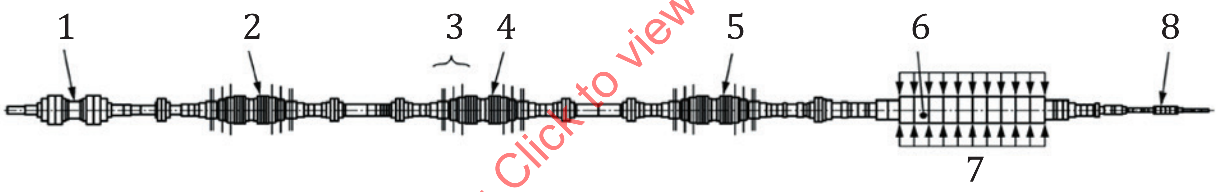
3.2
shaft
mainly cylindrical rotatable component carrying one or more elements (e.g. disc, coupling, blade)

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

Figure 1 — Shaft train consisting of six rotors

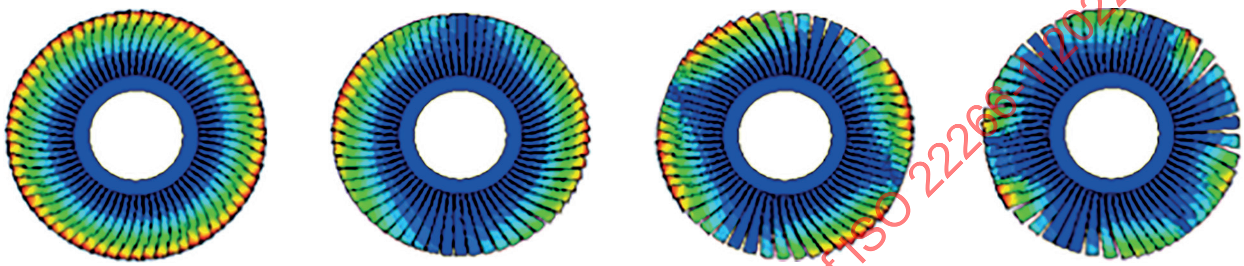
3.5
torsional vibration magnitude
maximum oscillatory angular displacement measured in a cross-section perpendicular to the rotation axis of the shaft train

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)

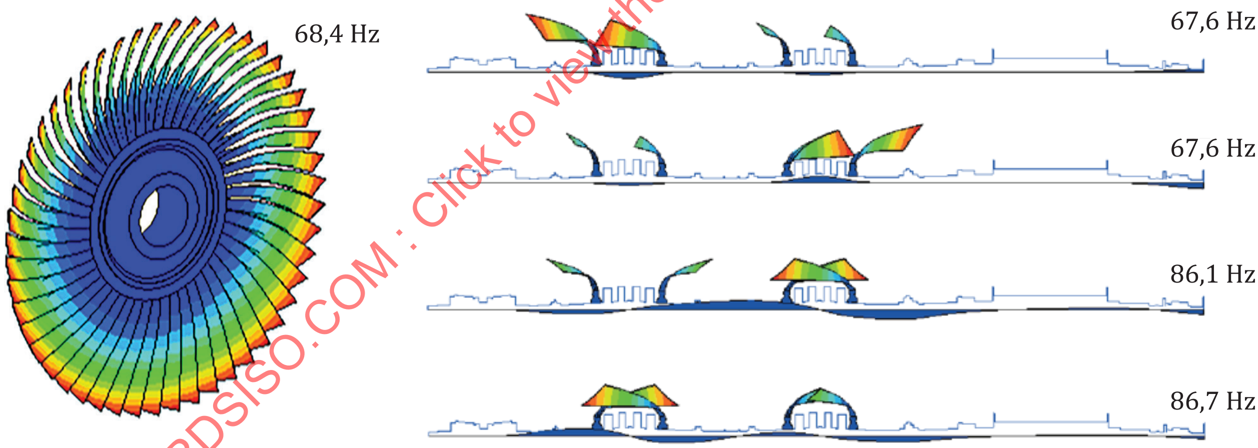
Note 1 to entry: When the shaft/disc and the elastic blades couple under dynamic conditions, the combined system produces several frequencies with zero-nodal diameter blade mode participation that are different from the individual shaft and blade frequencies (see Figure 3). These modes are often referred 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
$B_{l,i}$	mode specific lower separation margin of mode i
$B_{u,i}$	mode specific upper separation margin of mode i
$C_{l,i}$	mode specific lower calculation uncertainty of mode i
$C_{u,i}$	mode specific upper calculation uncertainty of mode i
$D_{l,i}$	mode specific lower reduced calculation uncertainty of mode i
$D_{u,i}$	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
$\Omega_{e,1}$	grid frequency axis
$\Omega_{e,2}$	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;

2) blade connection contact stiffness;

3) coupling stiffness.

The manufacturing tolerances (e.g. shrink fit diameters) and the finally achieved production quality (e.g. surface quality) of these joints and interfaces can have a significant influence on the stiffness properties of the model. The mass/inertia properties are generally well known for these design elements and are only marginally affected by manufacturing and assembly tolerances.

b) Large blade uncertainties:

There can be a scatter in the blade dynamic properties (natural frequencies) for large elastic blades of gas turbine compressors and LP steam turbines. In some cases, the scatter is partly intended to counteract any flow-induced vibration excitation. Consequently, for blades manufactured within their manufacturing tolerances the natural frequencies at standstill can vary by several Hertz. Models of large elastic blades shall take into account that the stiffness properties of the blades vary with speed due to the stiffening effect caused by centrifugal loading (see 5.2.2).

c) Generator winding uncertainties:

Similar to shrink fits and other joint types, the stiffness properties of the rotor model can be significantly affected by the interaction between generator shaft and the windings, depending on manufacturing tolerances and the finally achieved technical production quality. The mass/inertia properties are generally well known and are only marginally affected by manufacturing and assembly tolerances.

d) Material property uncertainties:

Typically, component surface temperatures are known with sufficient accuracy from other calculations (e.g. thermodynamic) or measurements so that body temperatures can be determined accurately. However, temperature distribution can vary significantly during transient events (e.g. load changes) or where there is a malfunction (e.g. differential cooling of generator windings). Consequently, the material properties introduce some uncertainty affecting the stiffness and mass/inertia properties of the shaft train.

5.4 Determination of calculation uncertainties

All known modelling uncertainties shall be assessed as, typically, they do not cancel each other out and will affect individual shaft train modes differently. To determine the nominal model initial calculation uncertainties the three methods described here are suitable and in addition, and when necessary, a combination of the methods can be used. Particular attention shall be paid to the uncertainties related to frequencies close to grid and twice grid frequency. Uncertainty can be based on:

a) Calculation:

For this approach all individual rotor modelling uncertainties are combined leading to higher/lower natural frequencies being calculated compared to the nominal model. Then all rotor models leading to higher/lower natural frequencies can be combined into shaft train models leading to higher/lower natural frequencies. Based on the difference in natural frequencies of the shaft train calculated with minimal/maximal and nominal model the uncertainty for each mode can be determined.

b) Blade natural frequency measurement:

Measurement of blade natural frequencies taken at rotor standstill can be used to populate the blade standstill model and the fact that blade standstill frequencies vary from blade to blade can be taken into account. With increasing rotational speed, the frequency variation between blades will reduce and so the uncertainties determined for the blades at standstill can only, to a limited extent, be transferred to the results of the shaft train calculation.

c) Rotor natural frequency measurements:

The uncertainties determined for the as manufactured rotor can only, to a limited extent, be transferred to the results of the shaft train calculations. For vibration modes of the shaft train dominated by one rotor, the uncertainties from the shop test on that rotor can be used with acceptable accuracy. Preferably the shop measurements are used to validate the uncertainties of the design elements. These values can then be used to determine the natural frequency uncertainties of other rotors by calculation.

NOTE 1 A statistically relevant number of measurements on different rotors with (almost) identical designs is required to determine the design elements uncertainties.

IMPORTANT — Caution shall be exercised when interpreting the natural frequencies obtained from static rotor tests where stiffness properties significantly depend on the rotating speed (e.g. bladed rotors and see A.4).

If, for a specific project, a test at standstill or a full speed shop measurement has been performed for a rotor, then the model of the rotor can directly be updated to match the shop measured results. Therefore, the modelling uncertainties for the specific rotor can be neglected (for the actual project where the rotor is going to be used) when determining the uncertainties of the shaft train based on calculation.

In order to properly determine natural frequency uncertainties, it is important to perform any calculations at the same temperature (typically room temperature) as when the shop measurements were taken so that a direct comparison can be made between the two sets of results and the model can be updated with confidence.

d) Measurements taken from similar shaft trains:

Where there are sufficient measured torsional natural frequencies taken from shaft trains having very similar designs (e.g. the same design elements, manufacturing tolerances, manufacturer), statistical information regarding the measured torsional natural frequencies can be used directly to determine the calculation uncertainty for each mode.

Caution shall be exercised where very similar rotors are installed in different combinations into different shaft trains. If sufficient measured torsional natural frequencies from similar shaft trains are available, only information for modes dominated by very similar rotors can be used to determine the calculation uncertainty. For example, where a very similar gas turbine rotor is installed in different turbine generator sets with different generator rotor designs, calculation uncertainties can be determined only for the torsional natural frequencies where the associated modes are dominated by the gas turbine rotor.

NOTE 2 A statistically relevant number of measurements on different shaft trains with (almost) identical designs is required to determine the modelling uncertainties for the shaft train.

6 Shaft train evaluation

6.1 General

Up to three analyses shall be completed in order to evaluate the shaft train torsional vibration characteristics to ensure that

a) the frequency margin between the calculated natural frequencies and the relevant electrical grid system frequencies is sufficient (see 6.2), and/or

b) at critical sections of the shaft train, the peak stresses induced by transient fault conditions are satisfactory (see 6.3), and

c) steady state excitation at the relevant electrical grid system frequencies due to the continuous allowable line unbalance loading does not cause any high-cycle fatigue failures (see 6.3).

The objective is to ensure that there are no shaft train vibration modes with natural frequencies in close proximity to the grid and twice grid frequencies of the electrical grid system that can be excited by variations in the generator air gap torque. Figure 4 gives guidance regarding which type of analysis is to be carried out, dependent on meeting the criteria defined in 6.2 and 6.3. The shaft train design is acceptable if the stress levels/fatigue damage caused by electrical excitation are within admissible limits. Having separation between the shaft torsional natural frequencies and the grid excitation frequencies is a major contributor to ensuring acceptable stress/fatigue margins.

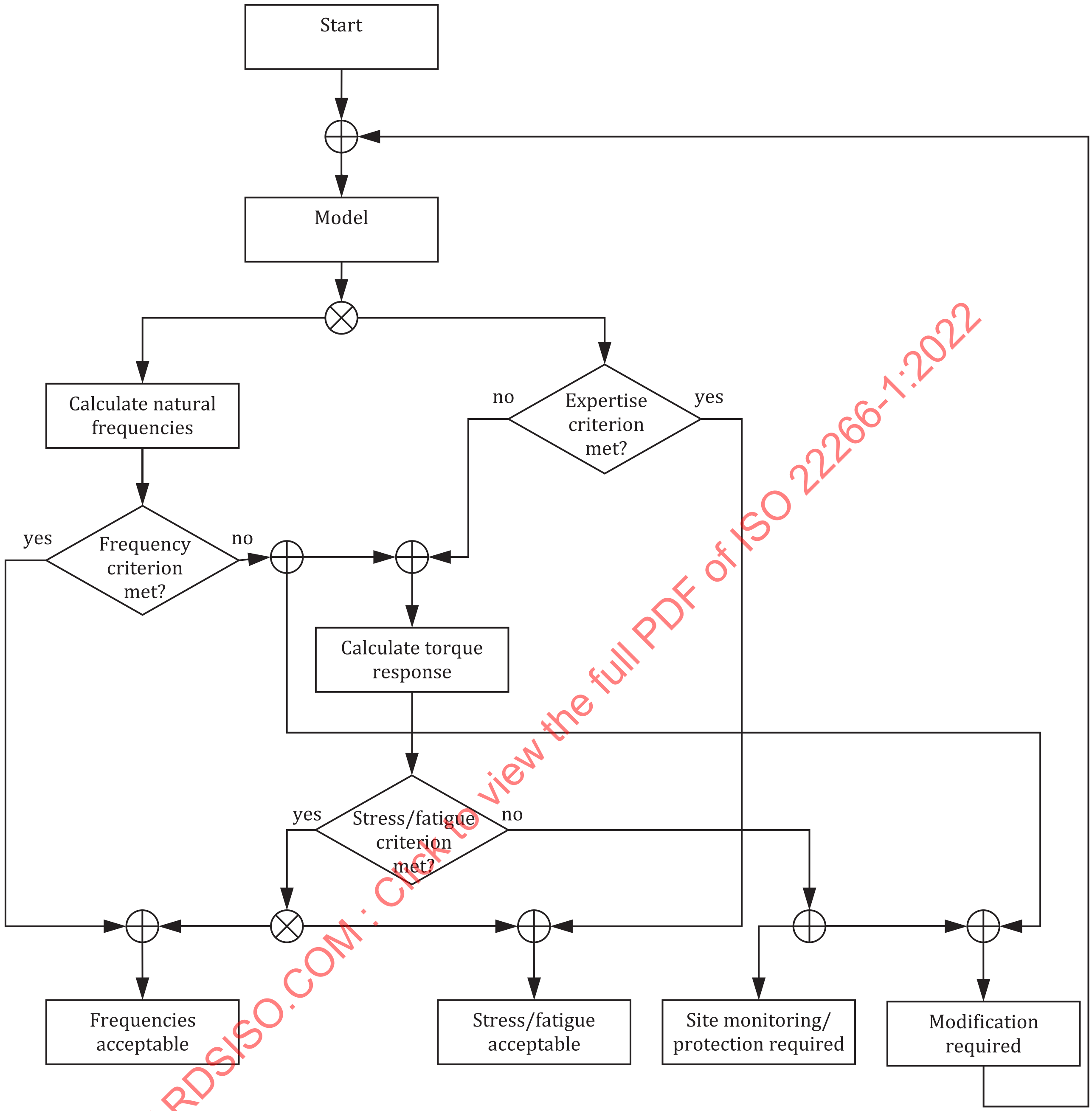
Starting with a model of the shaft train, the torsional natural frequencies shall be calculated and assessed against the criteria specified in 6.2. If required, to meet the natural frequency criteria in this clause, measurement results taken from individual rotors or shaft trains can be used to reduce the calculation uncertainty.

To ensure sufficient shaft train strength under electrical unbalance and fault conditions, experience from similar shaft trains can be used. With positive experience, there is no need to further investigate the shaft train stresses and fatigue damage caused by electrical excitation.

If the torsional natural frequency criteria cannot be met or there is no experience with a similar shaft train available, a transient fault analysis shall be performed. The resulting stress/fatigue damage shall be assessed against admissible values per fault. It should be noted that torsional natural frequencies of the shaft train corresponding to vibration modes that are insensitive to induced torsional forces are permitted within the frequency exclusion zone (see 6.3). Calculations by the equipment supplier shall indicate whether a mode is responsive or non-responsive to grid excitation.

Guidance regarding how to determine whether the results of the shaft train frequency, stress and fatigue analyses are acceptable, is given in 6.2 and 6.3. If, as a result of those analyses, any of the specified criteria cannot be met, the design of the shaft train or of individual rotors in it shall be modified so that the criteria can be met. However, in some cases, where modifications required to meet specification values cannot be justified economically, monitoring of shaft vibration or electrical quantities (e.g. negative sequence current) during operation shall be used to allow operators to take countermeasures in case excessive shaft train vibration occurs.

Further information regarding the calculation of torsional vibration is given in Clause 7.



Key

⊕ or

⊗ and

Figure 4 — Flow chart for the assessment of rotating machinery torsional vibration

6.2 Natural frequency assessment

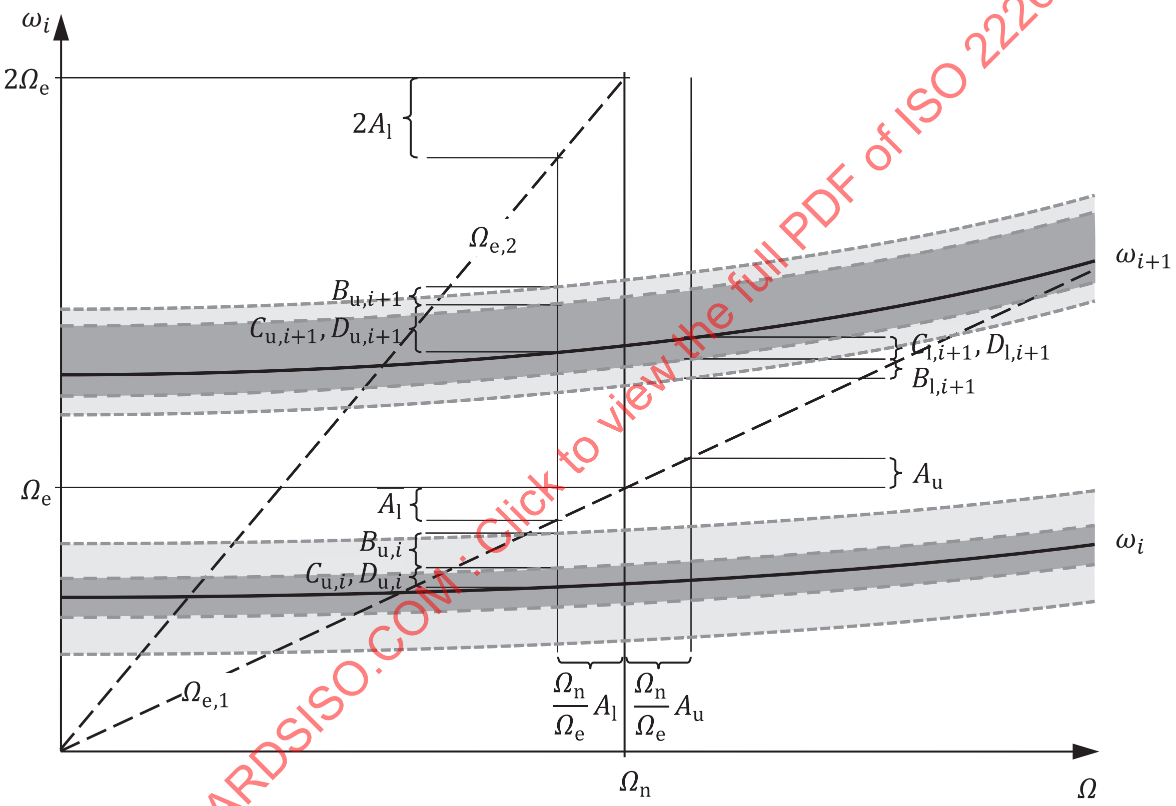
6.2.1 General

Shaft train torsional natural frequencies being 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) shall be considered for evaluating the frequency exclusion zones.

Typically, individual shaft train rotors have different modelling uncertainties and depending on their contribution to the shaft train modes, the calculation uncertainty of the torsional natural frequencies is mode dependent.

The contribution of the different vibration modes to the torsional vibration of the shaft train depends on their susceptibility to excitation by the generator. Modes having marginal susceptibility to excitation by the generator are less sensitive compared to modes having significant susceptibility to excitation by the generator. Therefore, the required margin between the natural frequency and excitation frequency of the shaft train is dependent on the vibration mode.

Therefore, it is feasible to consider different calculation uncertainties and separation margins for each vibration mode. However, applying identical separation margins $B_{u/l}$ and calculation uncertainties $C_{u/l}$ or $D_{u/l}$ to all torsional natural frequencies is also acceptable. It is not necessary to distinguish between upper and lower values but to consider them to be identical. Torsional frequency margins are shown in Figure 5, summarized in Table 1 and described in more detail in 6.2 a) to d).



Key

Ω	rotating frequency	A_u	upper grid frequency variation
Ω_n	nominal rotating frequency	$B_{l,i+1}$	lower separation margin of mode $i+1$
ω_i, ω_{i+1}	natural frequency of mode i, $i+1$	$B_{u,i}$, $B_{u,i+1}$	upper separation margin of mode i, $i+1$
Ω_e	nominal grid frequency	$C_{l,i+1}$	lower calculation uncertainty of mode $i+1$
$\Omega_{e,1}$	grid frequency	$C_{u,i}$, $C_{u,i+1}$	upper calculation uncertainty of mode i, $i+1$
$\Omega_{e,2}$	twice grid frequency	$D_{l,i+1}$	lower reduced calculation uncertainty of mode $i+1$
A_l	lower grid frequency variation	$D_{u,i}$, $D_{u,i+1}$	upper reduced calculation uncertainty of mode i, $i+1$

NOTE 1 See Table 1 for the definition of frequency margins A to D.

Figure 5 — Torsional frequency exclusion zones and margins

6.2.2 Torsional frequency margins

The grid frequency deviation limits (electrical grid frequency oscillations leading to grid and twice grid frequency excursions) are identified as A_u and A_l. These values, together with the additional margins given in Table 1, enables users to evaluate the required frequency margins specific to their grids. The grid frequency deviation limits vary for different regions throughout the world and shall be agreed between the customer and the rotor/shaft train supplier. In many cases, different upper and lower off-frequency variations A_u and A_l are specified.

Margin B is required between the natural frequency of the shaft train and the maximum/minimum permitted grid frequency to avoid any significant dynamic amplification near resonance. As the modal damping of torsional natural frequencies is typically small, the values of B_u and B_l are in most cases identical.

Margins C_u and C_l account for possible calculation inaccuracies in cases where the torsional natural frequencies of a coupled shaft train are assessed by calculation. The margin is omitted where an assessment is based on results obtained from a field test performed on the fully installed shaft train.

Confidence in the calculated frequency values increases if they are supported by experimental validation which allows the uncertainty margins C_u and C_l to be reduced. The extent to which the frequency margin can be reduced will depend on the level of testing performed and the test configuration used. For example, a field test on the fully installed shaft train will give a greater level of confidence than that provided by various levels of shop testing performed on individual rotors of the shaft train.

Validation should focus on rotors which significantly affect the torsional natural frequency of concern. Validation of rotors not participating in the associated mode does not provide a significant benefit. Margins D_u and D_l are the reduced margins for C_u and C_l respectively depending on the validation measures used.

Table 1 — Margins at grid and twice grid frequencies

Margin	Description	Frequency margin
A	Upper grid frequency deviation	A_u
	Lower grid frequency deviation	A_l
B	Upper margin between natural frequency and grid frequency	B_u
	Lower margin between natural frequency and grid frequency	B_l
C	Mode dependent upper calculation uncertainty (leading to higher natural frequencies) for nominal model	C_u
	Mode dependent lower calculation uncertainty (leading to lower natural frequencies) for nominal model	C_l
D	Mode dependent reduced upper calculation uncertainty for revised model based on validation (e.g. shop test measurement(s), measurements at similar shaft trains)	D_u
	Mode dependent reduced lower calculation uncertainty for revised model based on validation (e.g. shop test measurement(s), measurements at similar shaft trains)	D_l

Frequency margins A to D can depend on a number of other factors (e.g. the location of the power station, the integrity of the electrical network, accuracy of assessment and the operating history of the supplied hardware) and so the specification of numerical values for them is beyond the scope of this document. Examples of typical values together with the corresponding frequency margins are given for information in Annex B. However, it is emphasized that these may vary for different applications. The actual values to be used are subject to agreement between the customer and the supplier of the rotor/shaft train.

6.2.3 Natural frequency criteria

6.2.3.1 General

Calculation of torsional natural frequencies shall be performed at the stable baseload temperature condition of the turbine generator set. The torsional natural frequencies are acceptable if one of the criteria NF 1 (see 6.2.3.2), NF 2 (see 6.2.3.3) or NF 3 (see 6.2.3.4) are satisfied (see Figure 5). To apply the criteria, the torsional natural frequencies shall be calculated at the speed corresponding to the upper and lower grid frequency deviation. However, as long as only stiffening effects are included in the model (typically elastic rotor blades), checking the criterion for torsional natural frequencies calculated at nominal frequency is conservative and hence admissible.

NOTE Performing the analyses at the stable temperature state of the turbine generator set at baseload can result in torsional natural frequencies not fulfilling the criteria for a limited amount of time during warming up. The temperature primarily affects the stiffness properties (Young's modulus) of hot turbine sections. These sections typically have larger diameters and deformation is limited for the lower modes. For lower modes typically the stiffness of thin and cooler cross sections (e.g. bearing locations) has a much higher influence on the natural frequencies. As the thermal effect is temporary, its influence on the stiffness (Young's modulus) of lower modes is secondary and combined with a rather low probability to operate directly at the limits of A, a small consumption of margins B can be accepted for a limited amount of time, while the turbine generator set fully heats through from its original temperature state to the stable temperature state at load.

6.2.3.2 Natural frequency criterion 1 (NF 1)

The calculated torsional natural frequencies of the shaft train using a non-verified model shall meet the criteria expressed in Formulae (1) and (2)

$$x \cdot (\Omega_\mathrm{e} + A_\mathrm{u}) \leq \omega_i \left(\Omega_\mathrm{n} + \frac{\Omega_\mathrm{n}}{\Omega_\mathrm{e}} A_\mathrm{u} \right) - C_{\mathrm{l},i} - B_{\mathrm{l},i} \quad (1)$$

and

$$x \cdot (\Omega_\mathrm{e} - A_\mathrm{l}) \geq \omega_i \left(\Omega_\mathrm{n} - \frac{\Omega_\mathrm{n}}{\Omega_\mathrm{e}} A_\mathrm{l} \right) + C_{\mathrm{u},i} + B_{\mathrm{u},i} \quad (2)$$

where

$\omega_i \left(\Omega_\mathrm{n} + \frac{\Omega_\mathrm{n}}{\Omega_\mathrm{e}} A_\mathrm{u} \right)$ is the i-th calculated natural frequency at speed $\Omega_\mathrm{n} + \frac{\Omega_\mathrm{n}}{\Omega_\mathrm{e}} A_\mathrm{u}$;

$\omega_i \left(\Omega_\mathrm{n} - \frac{\Omega_\mathrm{n}}{\Omega_\mathrm{e}} A_\mathrm{l} \right)$ is the i-th calculated natural frequency at speed $\Omega_\mathrm{n} - \frac{\Omega_\mathrm{n}}{\Omega_\mathrm{e}} A_\mathrm{l}$;

$x = 1$ for torsional natural frequencies at grid frequency;

$x = 2$ for torsional natural frequencies at twice grid frequency.

If the criteria described in Formulae (1) and (2) are met, no further measures or calculations are required to meet the requirements for the torsional natural frequencies and the criteria described in 6.2.3.3 and 6.2.3.4 can be ignored. If the criteria are not met, either,

a) the calculation uncertainty shall be reduced, or

b) a full speed field test on the fully installed shaft train at thermally stable baseload conditions shall be performed, or

c) a more detailed stress analysis shall be performed so as to confirm that the dynamic stresses do not exceed the specified values (see 6.3), or

d) the shaft train design shall be changed to the shift torsional natural frequencies, or

e) the shaft torsional vibrations shall be monitored.

6.2.3.3 Natural frequency criterion 2 (NF 2)

If the calculated torsional natural frequencies are validated by means of measurements taken on similar rotor(s) or similar shaft trains, the calculation uncertainty can be reduced, and Formulae (3) and (4) apply:

$$x \cdot (\Omega_e + A_u) \leq \omega_i \left(\Omega_n + \frac{\Omega_n}{\Omega_e} A_u \right) - D_{l,i} - B_{l,i} \tag{3}$$

and

$$x \cdot (\Omega_e - A_l) \geq \omega_i \left(\Omega_n - \frac{\Omega_n}{\Omega_e} A_l \right) + D_{u,i} + B_{u,i} \tag{4}$$

If the criteria described in Formulae (3) and (4) are met, no further measures or calculations are required to meet the requirements for the torsional natural frequency criterion and the criteria described in 6.2.3.4 can be ignored. If the criteria are not met, either,

a) a full speed field test on the fully installed shaft train at thermally stable baseload conditions shall be performed, or

b) a more detailed stress analysis shall be performed so as to confirm that the dynamic stresses do not exceed the specified values (see 6.3), or

c) the shaft train design shall be changed to the shift torsional natural frequencies, or

d) the shaft torsional vibrations shall be monitored.

IMPORTANT — Caution should be exercised when interpreting results from tests performed on bladed rotors under standstill conditions (see A.4).

6.2.3.4 Natural frequency criterion 3 (NF 3)

If a full speed field test is carried out on the fully installed shaft train at thermally stable baseload conditions, the torsional natural frequencies shall meet the criteria of Formulae (5) and (6):

$$x \cdot (\Omega_e + A_u) \leq \tilde{\omega}_i \left(\Omega_n + \frac{\Omega_n}{\Omega_e} A_u \right) - B_{l,i} \tag{5}$$

and

$$x \cdot (\Omega_e - A_l) \geq \tilde{\omega}_i \left(\Omega_n - \frac{\Omega_n}{\Omega_e} A_l \right) + B_{u,i} \tag{6}$$

where

$\tilde{\omega}_i \left(\Omega_n + \frac{\Omega_n}{\Omega_e} A_u \right)$ is the i-th measured natural frequency at speed $\Omega_n + \frac{\Omega_n}{\Omega_e} A_u$;

$\tilde{\omega}_i \left(\Omega_n - \frac{\Omega_n}{\Omega_e} A_l \right)$ is the i-th measured natural frequency at speed $\Omega_n - \frac{\Omega_n}{\Omega_e} A_l$.

As the natural frequencies $\tilde{\omega}_i$ of the shaft train at baseload and at the speeds corresponding to the upper $\left(\Omega_\mathrm{n}+\frac{\Omega_\mathrm{n}}{\Omega_\mathrm{e}}A_\mathrm{u}\right)$ and lower $\left(\Omega_\mathrm{n}-\frac{\Omega_\mathrm{n}}{\Omega_\mathrm{e}}A_\mathrm{l}\right)$ grid frequency deviations can typically not be measured, the approximations given by Formulae (7) and (8) can be used:

$$\tilde{\omega}_i\left(\Omega_\mathrm{n}+\frac{\Omega_\mathrm{n}}{\Omega_\mathrm{e}}A_\mathrm{u}\right)\approx\tilde{\omega}_i\left(\Omega_\mathrm{n}\right)+\omega_i\left(\Omega_\mathrm{n}+\frac{\Omega_\mathrm{n}}{\Omega_\mathrm{e}}A_\mathrm{u}\right)-\omega_i\left(\Omega_\mathrm{n}\right) \tag{7}$$

and

$$\tilde{\omega}_i\left(\Omega_\mathrm{n}-\frac{\Omega_\mathrm{n}}{\Omega_\mathrm{e}}A_\mathrm{l}\right)\approx\tilde{\omega}_i\left(\Omega_\mathrm{n}\right)+\omega_i\left(\Omega_\mathrm{n}-\frac{\Omega_\mathrm{n}}{\Omega_\mathrm{e}}A_\mathrm{l}\right)-\omega_i\left(\Omega_\mathrm{n}\right) \tag{8}$$

where

$\omega_i\left(\Omega_\mathrm{n}\right)$	is the i-th calculated natural frequency at nominal frequency Ω_n;
$\tilde{\omega}_i\left(\Omega_\mathrm{n}\right)$	is the i-th measured natural frequency at nominal frequency Ω_n.

Calculations can be performed either using the nominal model or a validated model based on measurements taken on similar rotor(s) or shaft trains. If the criteria described in Formulae (7) and (8) are met, no further measures or calculations are required.

If the criteria of NF 3 are not met:

a) either more detailed stress analysis shall be performed so as to confirm that the dynamic stresses do not exceed the specified values (see 6.3), or

b) the shaft train design shall be changed to the shift torsional natural frequencies, or

c) the shaft torsional vibrations shall be monitored.

Examples of typical frequency margin values together with the corresponding frequency margins are given for information in Annex B.

6.3 Stress assessments

6.3.1 General

An assessment of the torsional rotor/shaft train stresses shall be carried out

a) in cases where no experience with similar applications is available, and

b) to confirm the acceptability of either calculated or measured torsional natural frequencies which do not satisfy the criteria given in 6.2.3.2, 6.2.3.3 or 6.2.3.4. As a consequence, the shaft train will be acceptable if the vibration modes of concern are insensitive to the excitation frequency and therefore do not pose any problem to system integrity.

It is the responsibility of the rotor/shaft train supplier to demonstrate, by calculation, that the dynamic stresses do not exceed the specified values or to demonstrate that the same or similarly designed machines are operating successfully with comparable grid conditions. Attention shall be paid to areas of potential high stress (e.g. coupling bolts, blade roots and those regions of the shaft with the smallest diameters).

The dynamic stress assessment shall take into account faults that excite grid and twice grid frequencies including steady state line unbalance excitation. Most load cases include a step change which excites all frequencies regardless of their separation margin, requiring a minimum component strength. The impact of electrical fault excitation on the shaft train can vary significantly, depending on the fault type and location. The stress assessment shall take into account that there is a static torsional stress due to the steady state torque/power transmission upon which the dynamic torsional stress is superimposed.

6.3.2 Expertise criterion

This criterion is fulfilled if experience with a similar shaft train is available. Whether a shaft train can be considered to be similar to a previous unit shall be evaluated by

a) the electrical system (e.g. excitation torques) to which the turbine generator set is to be connected to,

b) the torsional natural frequencies and mode shapes (e.g. separation of torsional natural frequencies from excitation frequencies, mode shapes and modal excitation factor[6]), and

c) shaft diameters and rotors inertia.

If these criteria are met, no dynamic torque analysis is required to evaluate the shaft torsional stress and fatigue. If these parameters cannot be met, a dynamic torque analysis shall be performed in order to establish whether the turbo set is suitable for the application it is intended for.

6.3.3 Stress/fatigue criterion

The induced stress and high-cycle fatigue in the shaft train occurring as a result of torsional vibration resulting from variations in the electromechanical torque shall not exceed limit values. Applicable limit values per incident shall be determined taking into account how often electrical faults occur, the maximum possible fatigue damage per electrical fault and the probability that the maximum possible fatigue damage occurs.

The calculated peak stress resulting from excitation of the shaft train by electromechanical torque shall be below limit values. If peak stress values exceed the limit values the fatigue caused by the electromechanical torque shall be assessed.

The modelling techniques, calculation method and acceptance criteria used are subject to agreement between the customer and the supplier (see 7.1).

A comprehensive description of fatigue assessment can be found in References [7] and [8].

7 Calculation of shaft train torsional vibration

7.1 General

Provided that the details of the individual shaft train are known, it is possible to calculate the (undamped) torsional natural frequencies and mode shapes at various speeds (e.g. nominal frequency and at the upper/lower grid frequency deviation), including blade-disc-shaft coupled effects (free vibration). If the frequency margins in 6.2 are not satisfied, the response of the system to forced excitation mechanisms (see Table C.1) shall be performed (forced vibration) for a system including damping, to ensure that the induced stress and shaft fatigue levels do not exceed the specified values and are in accordance with the rotor/shaft train supplier's experience.

7.2 Calculation data

The data to be taken into account when calculating shaft train torsional vibration are

a) the polar mass moment of inertia and torsional stiffness characteristics of each rotor in the shaft train,

b) the rotor blades elasticity (if applicable),

c) the specific operating parameters, and

d) if it is necessary to carry out a forced vibration calculation, the knowledge of the torsional vibration damping and the relevant excitation forces.

7.3 Calculation results

The calculation method used shall be capable of providing the following results for the shaft train:

a) natural frequencies and the corresponding mode shapes;

b) dynamic torsional stresses or torques;

c) shaft fatigue loading.

7.4 Calculation report

If the contract requires shaft train torsional vibration to be calculated, a suitable report shall be provided by the supplier. The contents of the report shall be agreed between the customer and supplier and shall at least contain:

a) shaft train leading particulars (e.g. inertia of rotors, length of rotors, bearing diameters);

b) configuration of the shaft train (including a summary of which blade rows have been modelled elastically);

c) calculation results;

d) if the supplier has subcontracted the calculation then, it shall be clearly stated in the calculation report.

8 Measurement of shaft train torsional vibration

8.1 General

If the initial calculation shows that there are torsional natural frequencies within the range defined in criterion NF 1, it is necessary to take further action. This involves either modifying the major rotating components or performing tests to validate the calculation results and confirm that the application of the reduced frequency margins defined in 6.2 are permissible. Depending on the particular circumstances, such measurements may be carried out on individual rotors in the factory or on the fully installed shaft train on site. The requirement for, and extent of, any such testing shall be agreed between the supplier and customer.

NOTE The requirement for testing can be waived if the supplier can demonstrate, to the satisfaction of the customer, that the accuracy of the prediction method is such that a smaller frequency exclusion range is satisfactory.

8.2 Method of measurement

A variety of different measurement techniques have been successfully employed in the past to measure shaft train torsional vibration characteristics and Annex A provides further background information. However, it is emphasized that the methods described in Annex A are not the only ones available and others may be equally applicable. The methods described in Annex A are subject to continuous improvement and so the one that is most appropriate for a specific application will depend on a number of factors.

Normally, the method adopted will be the one commonly used by the supplier. However, it should be recognized that testing of the fully coupled shaft train on site may be an expensive and time-consuming process that should only be considered under exceptional circumstances. It is for that reason, that the preferred approach is for the supplier to ensure that the frequency margins specified in 6.2 are met at the design/manufacturing stage, thereby, avoiding the necessity for site testing.

8.3 Measurement report

If torsional vibration tests are performed, a measurement report shall be provided by the rotor/shaft train supplier. The contents of the report shall be agreed between the customer and the supplier and shall contain as a minimum:

a) shaft train leading particulars (e.g. inertia of rotors, length of rotors, bearing diameters);

b) shaft train configuration;

c) the measurement parameters used;

d) the operating conditions (e.g. shaft temperature, power output) at the test site to allow for calibration of the shaft train model;

e) the type, accuracy and calibration method used for the measuring equipment and the measurement sensor positions;

f) if the supplier has subcontracted the torsional vibration measurements, then it shall be clearly stated in the measurement report.

9 General requirements

9.1 Supplier and customer responsibilities

a) The supplier of the turbine generator set shall be responsible for ensuring that the torsional vibration characteristics of the shaft train are within specification. In those cases where different manufacturers supply the turbine and generator, the turbine supplier will normally be responsible for the torsional vibration assessment.

b) Where torsional vibration calculations of the complete shaft train are requested, the supplier of the rotor/shaft train shall be responsible for the calculations, even if the calculations are subcontracted.

c) Where additional verification of the shaft train torsional vibration is required, the supplier shall be responsible for the measurements carried out even if that work is subcontracted. In particular, the rotor/shaft train supplier shall select, in agreement with the customer or the inspection organization representing him, the methods of measurement and sensor measuring locations to be used.

d) If there is a range of operating conditions where the vibration could cause damage, the turbine generator set supplier shall, with the agreement of other parties, take any necessary steps to eliminate critical vibrations or to ensure that procedures are in place to avoid operating the turbine generator set under damaging conditions.

e) Any necessary corrective actions to modify the turbine generator set are the responsibility of the supplier and shall be agreed upon by the relevant component manufacturer(s). It is the customer's responsibility to ensure that all information about the turbine generator set or other relevant details (e.g. transformers, grid) is provided to the supplier of the rotor/turbine generator set to enable the torsional vibration calculation to be performed correctly. When retrofitting some rotors of a turbine generator set, it can be reasonable to measure the torsional natural frequencies of the initial shaft train to validate the model used for non-OEM rotors. Nevertheless, and in all cases, clear responsibility for carrying out the torsional vibration analysis shall be agreed between the rotor/turbine generator set supplier and the customer.

f) The rotor/turbine generator set supplier shall not be liable for any assumptions being made in the calculations due to missing information from the customer.

g) In case of partial supply/retrofit of rotors/components by a non-OEM supplier, there shall be clear agreement between the new supplier(s) and the customer regarding the torsional vibration characteristics of the shaft train. This can also require additional verification of the torsional vibration of the complete shaft train prior to retrofit. In this case the customer of the turbine generator set shall allow for verification/measurement of the torsional characteristics.

9.2 Acceptance criterion

Any acceptance criterion to assess whether the turbine generator set will operate satisfactorily with regard to torsional vibration, are subject to agreement between the customer and the supplier.

Annex A
(informative)

Torsional vibration measurement techniques

A.1 General

A variety of different measurement techniques have been successfully employed to measure the torsional vibration characteristics of coupled shaft trains. These methods are subject to continuous improvement and, therefore, that which is most appropriate for a specific application will be dependent on a number of factors. This annex provides further background to some of these measurement techniques. It is emphasized that these are not the only available methods that can be used and other equally applicable methods developed by different OEMs are similarly useful but not discussed here.

A.2 Torsional vibration transducers and measurement systems

These devices are typically used to measure torsional vibration:

a) incremental measurement techniques (e.g. eddy current probes, inductive probes, lasers, optical decoders (non-contacting transducers used for incremental pulse timing));

b) strain gauges;

c) accelerometers positioned circumferentially around the shaft at specified angles (preferably 0° and 180°);

d) magnetostrictive working sensors.

It is emphasized that the methods listed in A.2 a) to d) are not the only available methods that can be used and others may be equally effective.

Table A.1 gives an overview of the advantages and disadvantages of typical torsional vibration transducers.

Table A.1 — Comparison of typical torsional vibration transducers and measurement systems

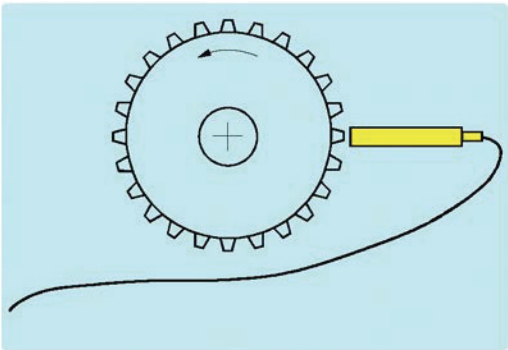
Principle	Description	Advantages	Disadvantages
Incremental method	Contactless probes generate a series of pulses from features on the shaft surface (e.g. gear teeth). The system measures the instantaneous time between two pulses. The measurand is directly proportional to the instantaneous velocity.	— No rotating sensors and no telemetry required — Non-intrusive measurement and no risks regarding environment health and safety — Very little turbine generator set downtime needed for installation. Reduced lead time and on-site effort — Suitable for long term monitoring and short term testing	— High sample frequency required

Table A.1 *(continued)*

Principle	Description	Advantages	Disadvantages
Strain gauges	Strain gauges applied to the rotating shaft. The measurand is directly proportional to the mechanical strain.	— Standard parts can be used — Proven technology	— Considerable turbine generator set downtime needed for installation — Power supply and telemetry required with risk for machine integrity, environment health and safety — Rotating test equipment can generate noise — Limitations for long term operation — Telemetry collars need to be adapted to fit the part to which they are being fitted
Rotating accelerometers	Two accelerometers mounted 180° apart from each other. Measurand is directly proportional to the acceleration in tangential direction.	— Standard parts can be used	— Considerable turbine generator set downtime needed for installation — Power supply and telemetry required with risk for machine integrity, environment health and safety — Rotating test equipment can generate noise — Limitations for long term operation — Telemetry collars need to be adapted to fit the part to which they are being fitted
Magnetostrictive sensors	Contactless probes measure the change in permeability of the shaft surface due to external loading.	— No rotating sensors and no telemetry required — Non-intrusive measurement method and no risks regarding environment health and safety — Very little turbine generator set downtime needed for installation. Reduced lead time and on-site effort — Suitable for long term monitoring and short term testing	— Sensors require calibration for accurate amplitude measurement — Amplitude measurement sensitive to shaft train axial expansion/contraction

A.2.1 Incremental measurements

For the incremental measurement of torsional vibration, any regular pattern on a shaft can be used. Either gearwheels need to be mounted on the shaft train or existing gearwheels can be used. Typical applications use a speed wheel (see Figure A.1) but any other gear on the shaft is suitable. For short term measurements zebra tapes (where the black and white pattern is detected by a laser sensor) can be used.



a) Schematic representation

b) Photo of measurement setup

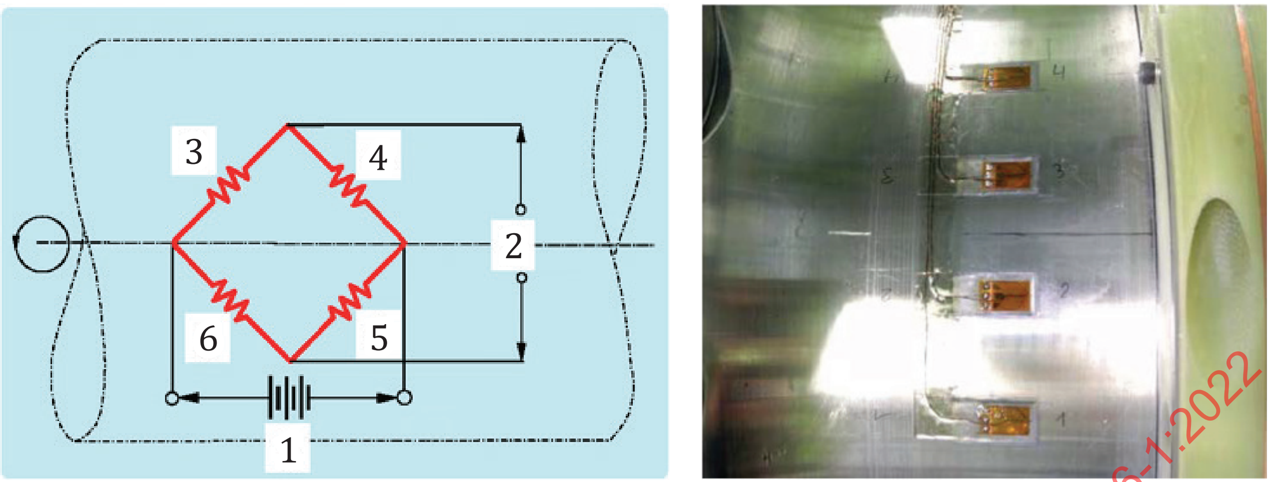
Figure A.1 — Incremental measurement technique

The measurement principle of this method is based on a taking accurate time measurement of the increments. Typical clocking rates are in the range of 30 MHz. Torsional vibrations superimposed on the constant rotational velocity lead to a non-constant time between pulses. As the angular increment is given by the number of pulses measured per revolution, the instantaneous time between pulses is proportional to the vibration velocity.

As lateral vibration affects the measuring accuracy, installing sensors in pairs (with 180° offset between them) compensates for shaft bending vibration affecting the output.

A.2.2 Strain gauge measurements

Installing strain gauges (see Figure A.2) on a shaft/component typically requires a significant amount of preparation work as coils and telemetry collars have to be adjusted to the specific shaft diameter and the mechanical integrity of all rotating components has to be ensured. Accurate attachment of the strain gauges to the shaft during shutdown is necessary taking the hardening times of the adhesive used into account. The adhesive used should be resistant to heat, thermal expansion, chemical influence and dirt. The energy supply for the strain gauges is typically provided via slip rings, coils or battery packs mounted on the shaft. The signals themselves can be transmitted wirelessly or via slip rings.



a) Schematic representation **b) Photo measurement setup**

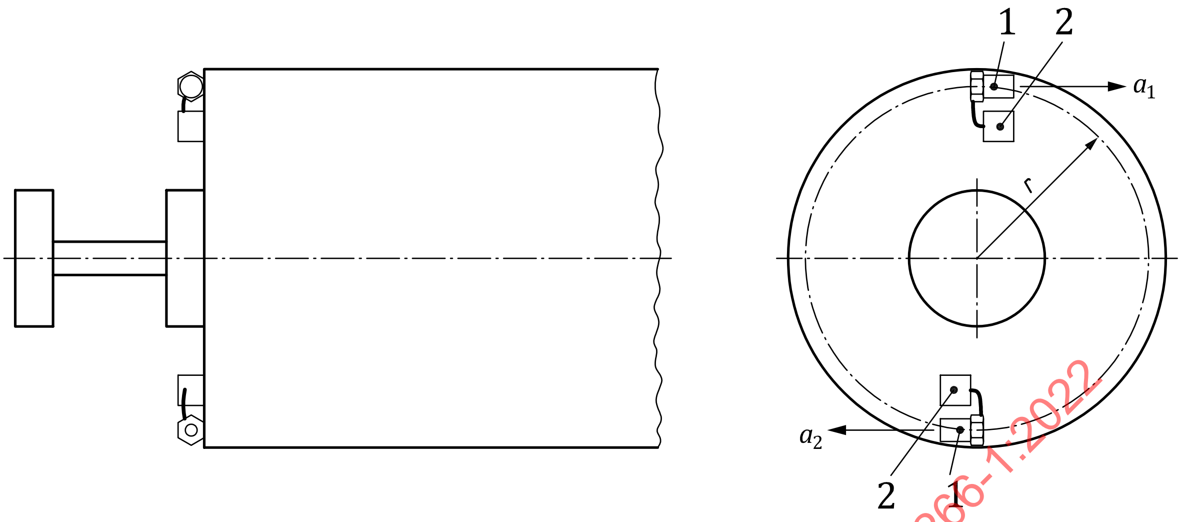
Key

1 supply voltage
2 signal voltage
3 strain gauge 1
4 strain gauge 2
5 strain gauge 3
6 strain gauge 4

Figure A.2 — Measurement with strain gauges (here with redundancy)

A.2.3 Acceleration measurements

As for any method using sensors mounted on the shaft train, the accelerometers need to be fixed accurately. In addition, the accelerometers should be installed in pairs to compensate for vibration induced by shaft bending, and mounted 180° apart from each other (see Figure A.3). Care should be taken to prevent unbalance caused by the measuring equipment mounted on the rotor, otherwise incorrect measurements caused by lateral acceleration are possible. The energy supply to the sensors is typically provided by battery packs mounted on the shaft. If battery packs are used to supply the sensors, the arrangement is limited to short term testing (e.g. for spin pit validation). The signals from the sensors can be transmitted wirelessly or via slip rings.



Key

1 accelerometer

2 radio transmitter

a_1, a_2 tangential accelerations

r radius

NOTE If two accelerometers are positioned exactly opposite at the same radius, *r*, and pointing in the same tangential direction, measuring the tangential accelerations a_1 and a_2, the torsional acceleration ϕ at that radial position is $\phi = (a_1 + a_2)/(2r)$. This arrangement avoids any influence due to lateral bending of the rotating shaft train.

Figure A.3 — Circumferential accelerometer mounting positions

A.2.4 Magnetostrictive measurements

Here, the transducer excites a constant high-frequency magnetic field via its exciter coil. Typically, two or four measuring coils are used (see Figure A.4) to record the magnetic field between the shaft surface and coil. The sensor measures the flux changes in the magnetic field. The signal quality depends on the shaft properties (e.g. material type, material processing) as well as conformity with the electrical and mechanical setting parameters.

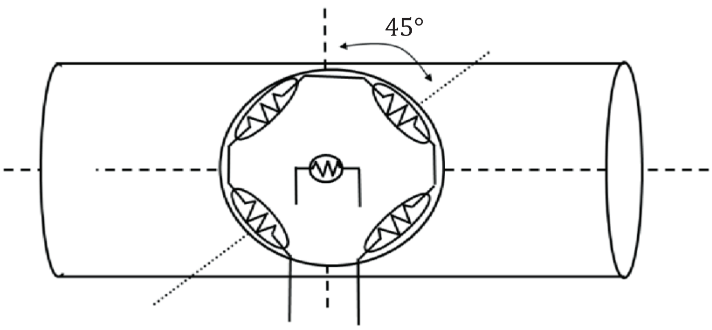


Figure A.4 — Measurement with magnetostrictive sensor

A.3 Measurement report

Depending upon the method of measurement used, it is recommended that these topics are included in the measurement report:

a) rotational speed of the turbine generator set;

b) turbine generator set power output;

c) torsional vibration magnitudes;

d) torsional strains;

e) test site ambient temperature;

f) torsional natural frequencies including modes dominated by blade zero nodal diameter vibrations;

g) speed range over which measurements are carried out;

h) date, name and signature of creator and inspector.

Depending on the measuring setup, not all the information listed in A.3 a) to h) can be provided. Other parameters may be included on agreement between the customer and the turbine generator set supplier.

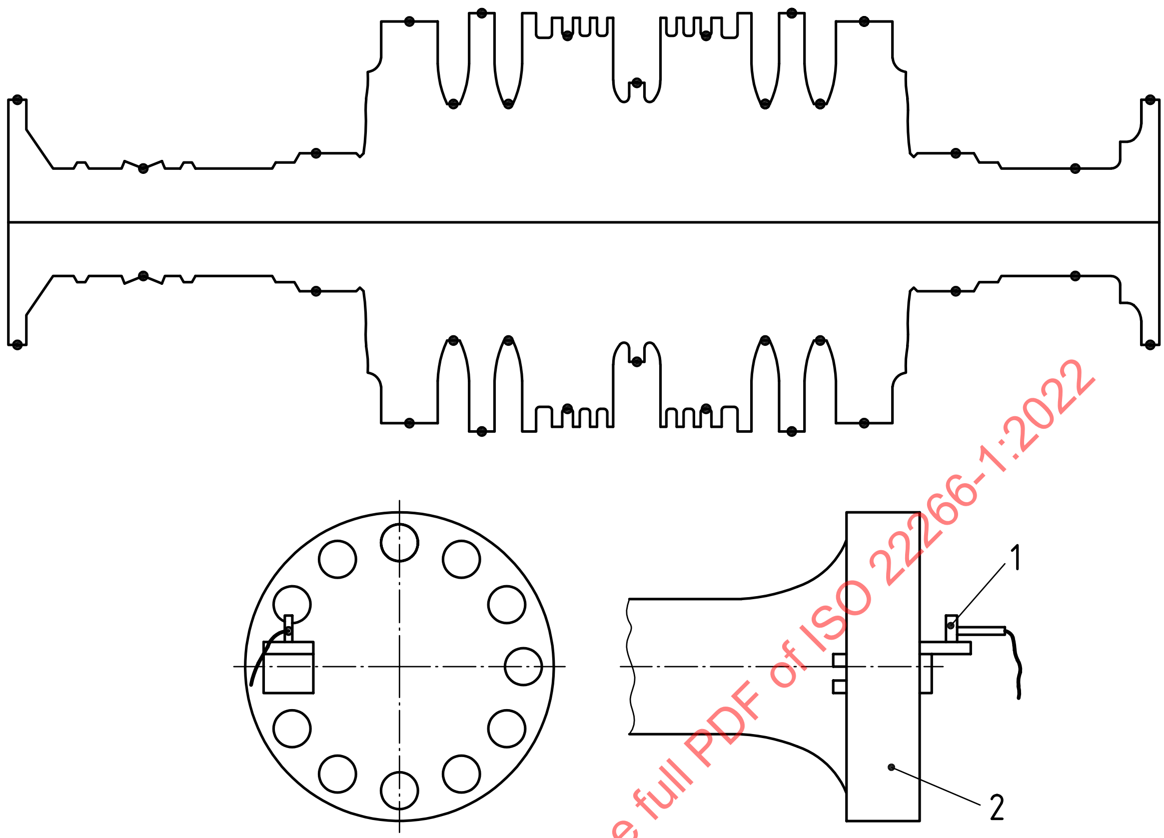
A.4 Factory tests on rotors under standstill conditions

Experimental modal analysis may be performed on rotors in their non-rotating configuration which provides verification of their predicted behaviour and, therefore, helps to calibrate rotor models for fundamental modes under stationary conditions. In recognition of the fact that boundary conditions influence the final outcome of the test, it is important either to carry out a free-free test or to support test rotors on bearing journals with hard rubber or similar supports. These supports provide little resistance to the impact energy path at the contact areas so that the relevant torsional modes of the rotor under test are properly captured in the frequency spectrum.

Non-rotating tests are especially useful for the rotor body alone. When blades are mounted on shafts or discs, the torsional vibration results may be questionable and caution needs to be taken, as a perfect contact between blade roots and the disc/shaft body may not always be present (e.g. due to the design of root employed or manufacturing tolerances, rattling can occur). As a result, the impact energy imparted by a test hammer to the structure may be disrupted at the blade-disc or disc-shaft contact area, making it difficult to capture system frequencies for those modes in which the blades have a significant influence. Even if such modes are captured in the test, they are less useful because these system frequencies continuously change with speed. In other words, the blades dynamically couple with the disc/shaft and this will continue until rated speed is reached. Similar difficulties exist when carrying out non-rotating tests on generator rotors due to the influence of the copper windings and associated wedges.

Although experimental modal analyses made under standstill conditions are helpful to calibrate rotor models, they do not generally provide an accurate assessment of the torsional natural frequencies of either the blade-disc coupled system or the generator body modes that vary with speed. Full speed (dynamic) factory or site tests are necessary to assess these effects. However, tests under standstill conditions are helpful as one step to produce a validated rotor model if a correlation between standstill and full speed tests has been established for similar rotors. In this case, based on the measurements and calculations expected, the full speed torsional natural frequencies of the shaft train can be determined.

Figure A.5 shows the arrangement for a typical factory rotor test under standstill condition. Typically, two to three measuring locations along the shaft axis of the rotor are sufficient to identify its natural frequencies and the mode shapes of interest. Nevertheless, if the primary purpose is to validate a numerical model, additional sensor planes are helpful in order to align the calculated mode shapes to the measured ones.



Key

1 impact hammer

2 end coupling flange

● possible measurement locations

Figure A.5 — Factory test under standstill condition

A.5 Full speed (dynamic) factory tests

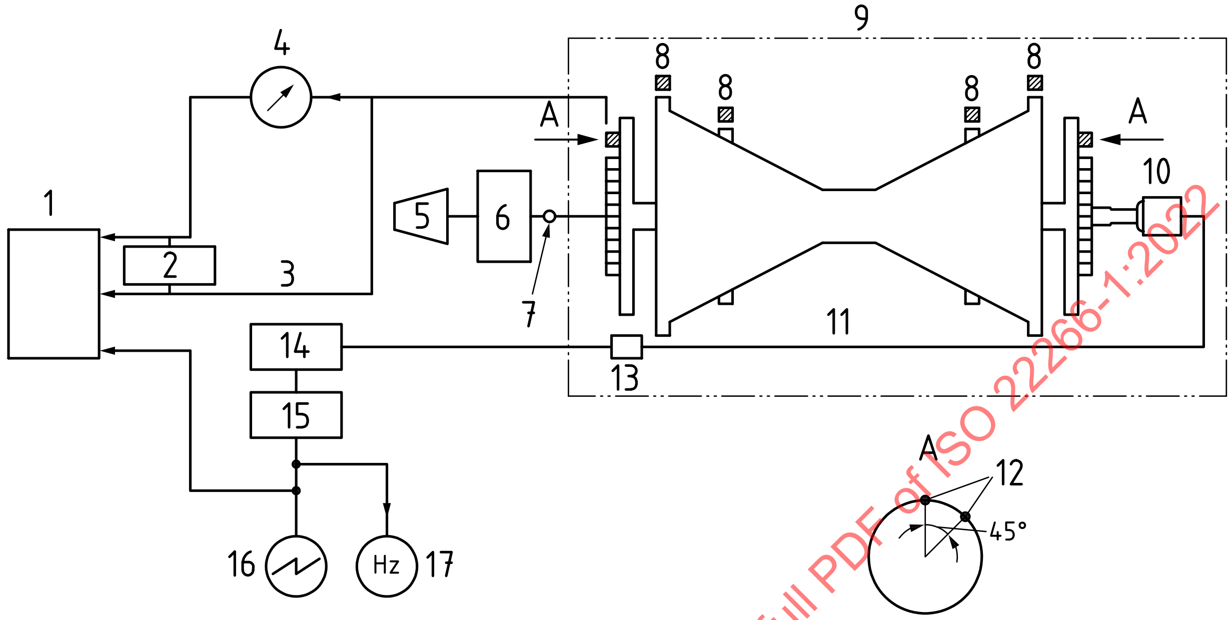
When blade-disc coupled frequencies are calculated in the proximity of twice grid frequencies, the only reliable way to verify them in the factory is to perform full speed dynamic tests. These tests provide verification of predicted behaviour and, therefore, help to calibrate models under rotating conditions. At one extreme, it provides a final opportunity to modify the design before it leaves the factory. Since the dynamic tests are conducted in the manufacturers facility (e.g. spin pit), the total cost is significantly lower and less time consuming than those performed on site. Full speed dynamic factory tests allow the tested rotor model to be improved. If several non-tested components are part of the shaft train mode of interest, care shall be taken to determine a reduced calculation uncertainty, D, from the full speed dynamic factory tests. The test can be done either with or without a defined external excitation applied.

One approach uses the system inherent background noise (e.g. produced by the drive train) as an excitation to stimulate the shaft-blade modes. With this excitation, response spectra can be derived, which are usually sufficient to identify the rotor natural frequencies.

Another method uses a defined excitation (e.g. a continuously rotating torsional exciter system (such exciters are commercially available)). In general, the excitation system is attached to one end of the shaft coupling. For a selection of torsional vibration transducers to measure torsional natural frequencies and vibration magnitudes see A.2. The selected transducers should be capable of detecting the modes of concern and need to be attached at suitable locations. Torsional natural frequencies are measured/

confirmed by alternately applying and removing excitation at the frequencies in which the torsional response reaches a peak when the rotor is rotating in its rated speed range.

Figure A.6 shows a schematic view of an example of the test setup for the dynamic test of a rotor in the factory.



Key

1 data recorder
2 real-time analyser
3 raw signal
4 torsion meter
5 drive train
6 gear
7 speed transducer
8 blade number transducer
9 vacuum spin pit
10 torsion exciter
11 hydraulic line
12 transducer
13 dual pressure manifold
14 hydraulic actuator
15 master controller
16 oscilloscope
17 frequency counter

Figure A.6 — Schematic view of a factory dynamic rotor test set up

A.6 On-site torsion tests

The various factory tests described in this Annex are an extremely helpful way of accurately calibrating numerical rotor models. Nevertheless, when rotors are connected through couplings, some amount of uncertainty remains and hence, as described in 6.2, this is taken into account when specifying the allowable frequency margins. The degree of modelling inaccuracy tends to increase for higher modes (e.g. modes with frequencies around twice grid frequency). It is also a challenge to account accurately for the influence of system damping in the analysis of modes lying within the frequency ranges defined in 6.2. In those cases where the calculated frequencies do not satisfy the criteria defined in 6.2 and the corresponding modes are responsive, the uncertainties discussed above can justify carrying out a field torsion test.

In addition, on-site torsional tests are often performed where rotors have been retrofitted, where the original torsional natural frequencies of the shaft train are used to benchmark the torsional natural frequencies of the retrofitted shaft train.

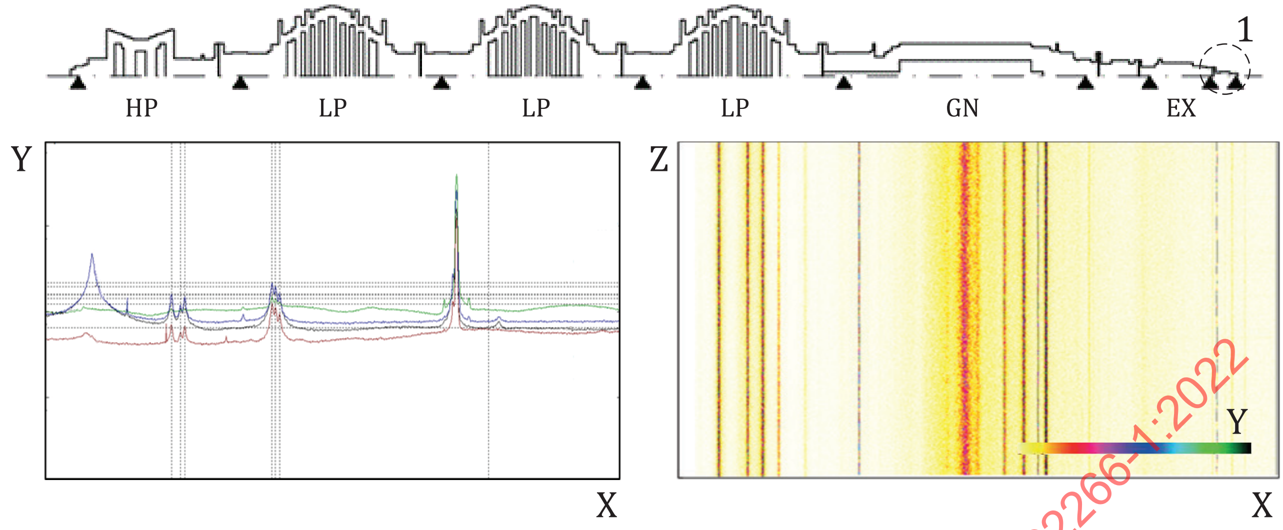
Depending on the type of mode that is critical, a field test could involve measurements taken at a few different locations. The choice of measurement positions is normally determined by examination of the predicted mode shapes, by the sensor principle and accessibility to the shaft. Strain gauges require a location of sufficient modal strain (e.g. a vibratory node) whereas incremental sensors require a sufficient angular vibration deflection or velocity (e.g. at the free end of the generator). In most cases the measurement of torsional vibration magnitudes at two locations is sufficient to capture the important turbine and generator coupled shaft and blade modes. However, if more detailed mode shapes are required, it can be necessary to measure torsional response at other locations on the shaft or at the blade tips. In any case, a pre-selection of the measurement locations based on the rotor dynamic model is extremely helpful.

For a selection of torsional vibration transducers to measure torsional natural frequencies and vibration magnitudes see A.2. The torsional signals are usually displayed on a spectrum analyser to determine their various frequency components. Although it requires more effort, utilizing multiple measuring locations is an advantage because they can potentially identify more torsional modes and because the relative magnitude at each natural frequency enables the measurement of mode shapes to be identified. This reduces the risk of missing important modes.

To measure torsional natural frequencies at a power plant site, passive excitation measurements have become state of the art. With the increasing sophistication of signal analysis techniques, it is now possible to detect all frequencies of interest by measuring the effects of small transient disturbances that occur under normal operation or even in off-grid conditions. These disturbances, which are a consequence of the minor random disturbances that are inevitably present on all electrical networks, cause excitation of those natural modes that can be excited by the generator. In some circumstances excitation can also be caused by the flow field in the turbine modules themselves. Also synchronization, load rejection or line switching events can excite the shaft train modes to measurable levels. The advantage of this technique for the customer is that, other than the time required to install the measurement equipment, there is no impact on normal operation of the power plant.

If torsional natural frequencies cannot be measured using passive excitation methods, exciting the generator in a controlled manner using an oscillatory torque developed from unbalanced currents flowing in the generator stator can be used. These unbalanced currents are obtained by the application of a line-to-neutral short circuit test connection on the high-voltage side of the generator main step-up transformer (or, alternatively, a line-to-line connection at the generator terminals) while the turbine generator set is not connected to the grid.

On-site torsional tests for the evaluation of natural frequency in accordance with criterion NF 3 (see 6.2) should be carried out at the temperature of the thermally stable set at baseload. Figure A.7 shows an example of a steam turbine shaft train, where the incremental (encoder) method has been used to validate the torsional natural frequencies. The measurement position was chosen to be at the exciter end. This location was chosen based on the rotor dynamic model, as all modes show an angular deflection at the free end of the generator. It can be seen that the individual shaft-blade modes of the three LP modules have been detected. Although the blade and the steam path of the modules are identical, three slightly separated natural frequencies can be identified.



a) Frequency spectrum

b) Spectrogram

Key

X frequency
Y angular velocity at measuring location
Z time
1 measuring location
HP high-pressure turbine
LP low-pressure turbine
GN generator
EX exciter

Figure A.7 — Example of an on-site torsional test set up

A.7 Torsional vibration monitoring

Based on the different possible input signals for a torsional vibration monitoring system and the different measurement possibilities (see A.2) the concepts of torsional vibration monitoring can be very different. Measuring the voltage and current at the generator terminals provides an alternative mean of measuring the torque going into the generator. Analysis of these subtle changes in the relationship between current and voltage can provide an indication of the shaft train torsional vibration.

The purpose of torsional vibration monitoring systems is to evaluate the vibration deflection, stress and shaft fatigue at the measurement location and/or other locations of the shaft train where no direct measurement is possible. Furthermore, the torsional natural frequencies and resonance frequencies as well as the damping of the coupled electro-mechanical system can be supervised.

The measurement system would be capable of measuring broad-band vibration over a frequency range from 5 Hz to at least three times grid frequency. Measurement and storage of the raw signal should be able to allow offline evaluation of torsional vibration, which is of special interest after a torsional excitation event (see Table C.1) has occurred. An accumulation of the fatigue (e.g. shaft, blade couplings) would be stored automatically by the monitoring system. Some torsional vibration measurement systems can predict the vibration deflection, stress and fatigue at other shaft locations, where no direct measurement is possible, based on the available measurement results.

For the identification and analysis of vibration events, it is not sufficient to only monitor the torsional vibration measurement signals, and measurement of relevant operational parameters from the digital control system becomes necessary. These parameters are for example

a) the three phases voltages of the generator (at the low and high voltage side of the generator transformer, if possible),

b) the tacho signal with one pulse/revolution,