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Mechanical vibration — Torsional vibration of rotating machinery —

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

Vibrations mécaniques — Vibration de torsion des machines tournantes —

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

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

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

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

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

Introduction

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

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

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

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

Mechanical vibration — Torsional vibration of rotating machinery —

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

1 Scope

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

This part of ISO 22266 is applicable to

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

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

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

2 Normative references

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

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

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

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

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

3 Terms and definitions

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

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

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

NOTE 1 Figure 1 shows an example.

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

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

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

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

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

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

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

where η is the damping ratio.

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

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

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

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

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

NOTE Figure 2 shows examples.

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

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

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

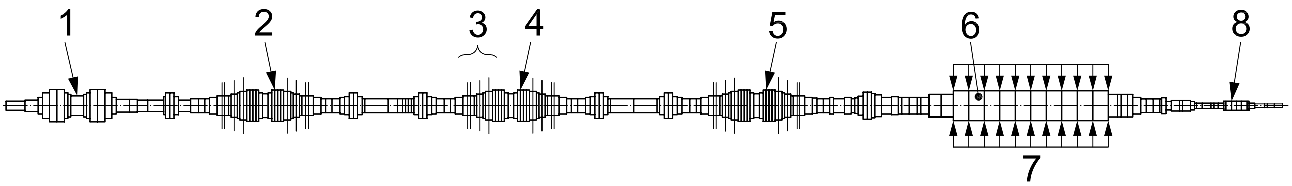
NOTE When the rotor disc and the blades couple under dynamic conditions, the combined system produces several new “all-in-phase” frequencies that are different from the individual disc and blade frequencies (see Figure 3). These modes are often referred to as *zero-nodal diameter* or “umbrella” modes.

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

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

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

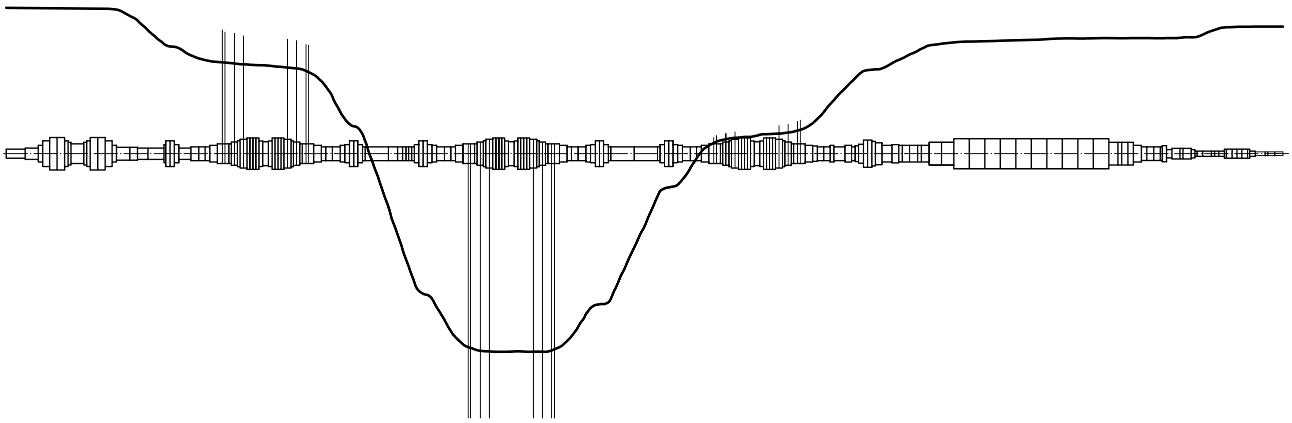
3.14
additional torsional stress
stress due to the torsional vibrations of a given excitation harmonic superimposed on the torsional stress corresponding to the mean torque transmitted in the given section of the shaft system being considered



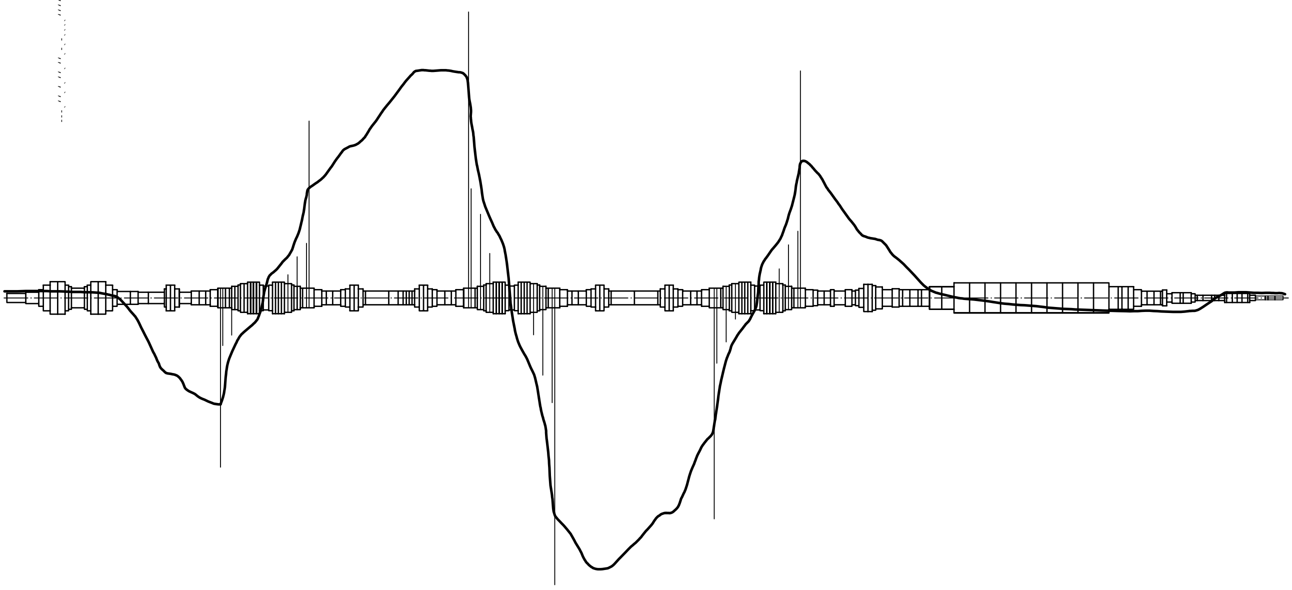
Key

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

Figure 1 — Six-rotor steam turbine generator system



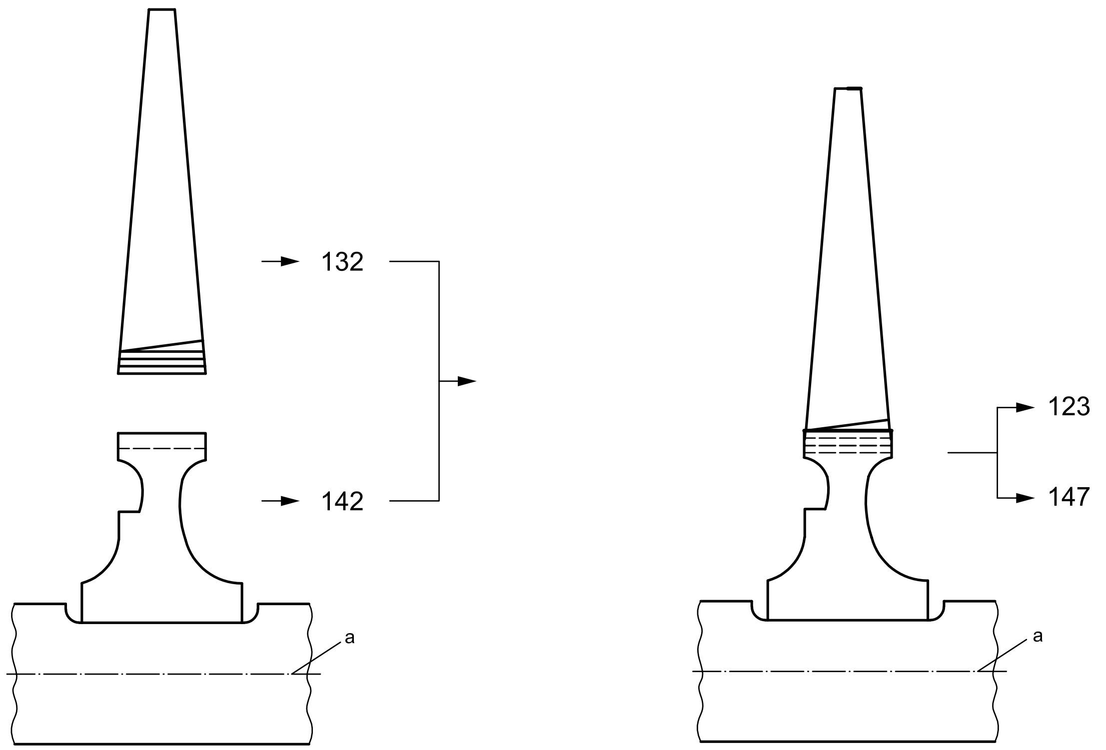
a) Second mode of vibration or two-node mode of vibration



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

Figure 2 — Typical torsional mode shapes of the shaft system

Frequencies in hertz



a) Uncoupled frequencies of separated blade and disc

b) Coupled frequencies of blade-disc assembly

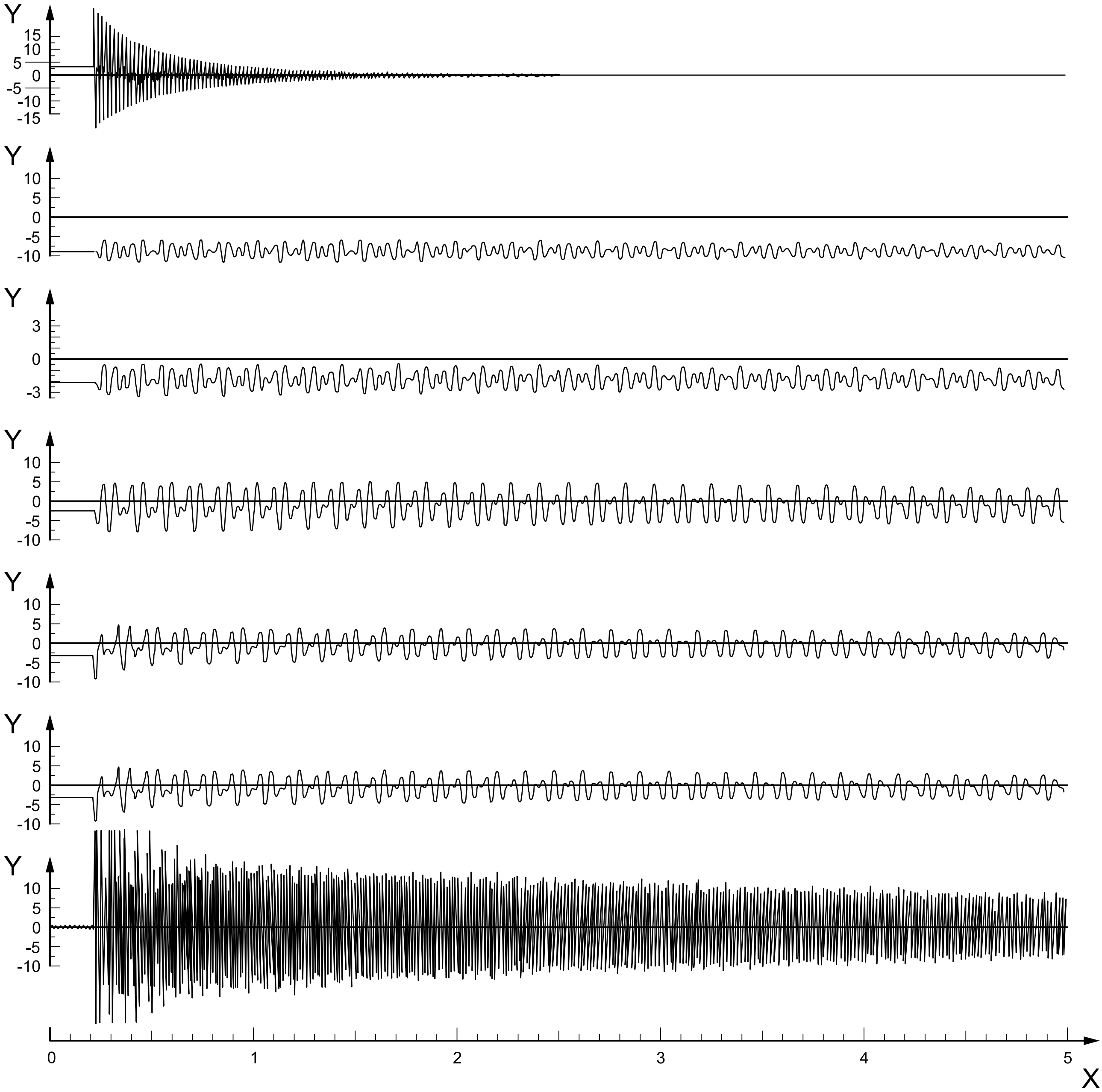
[a] Rotor central axis.

Figure 3 — Schematic illustration of blade-disc dynamic coupling

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

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

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



Key

X time, s

Y torsional stress

Figure 4 — Typical dynamic torsional stress

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

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

4 Fundamentals of torsional vibration

4.1 General

Torsional vibrations in turbine generator shaft systems are most commonly excited by variations in electromechanical torque induced at the air gap of the generator. When 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 vibrations. 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 torque.

Table 1 summarizes the typical components of air gap torque variations for various types of system disturbances. The magnitudes of these components depend upon the nature and severity of each disturbance. These disturbances can be categorized as *transient* and *steady-state*. In general, transient disturbances are cleared after a short time, but steady-state disturbances can persist for extended periods. Further details of various electrical faults that could occur in power plants are provided in Annex C.

Table 1 — Types of disturbances

Types of disturbances	Step change	Excite at line frequency	Excite at twice line frequency	Excite at (between 0,1 and 0,9) of line frequency
Transient:				
Three phase fault	×	×		
Unbalanced fault[a]	×	×	×	
Synchronization out-of-phase	×	×		
Open transmission line (three phases)	×			
Close transmission line (three phases)	×	×		
Single pole switching	×		×	
Transient sub-synchronous resonance (SSR)				×
Disturbances in the grid due to thyristor controlled loads (e.g. variable speed electric motors)		×	×	
Steady-state:				
Line unbalance[b]			×	
Load unbalance[c]			×	
Steady-state sub-synchronous resonance (SSR)				×

[a] Unbalanced fault can be either line-to-line, line-to-ground or twice line to ground short circuits. Such faults can be seen either on the transmission system or more severely at the generator terminals.

[b] Line unbalance: Unbalance in transmission line or system, for example, untransposed transmission lines.

[c] Load unbalance: Unbalance of the electrical load of the system.

In summary, torsional excitation of turbine generator shaft systems is induced at the generator terminals due to the following reasons:

a) unbalanced short circuits that produce unidirectional, line and twice line frequency transient torques;

b) out-of-phase synchronization of the unit to the grid, which could produce very high levels of unidirectional and line frequency transient torques;

c) excitations from other sources, including

- three-phase short circuits,
- transmission line switching, and
- load variations induced and transmitted by heavy-duty operating equipment (such as electric arc furnaces) in the vicinity;

d) sub-synchronous resonance, which can occur if the generator is connected to long transmission lines and could excite the sub-synchronous torsional modes. Simple lump mass-spring systems are used in grid system stability studies to model these sub-synchronous frequencies and their mode shapes;

e) line or load unbalance resulting in negative sequence currents that produce torques at twice the line frequency.

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 line and twice line system frequencies. For those modes that can be excited by torsional oscillations of the generator and are evaluated to be critical to the integrity of the unit, there shall be sufficient margin from both the line and twice line system frequencies. This is the primary consideration for avoiding any torsional vibration issues on large turbine generators. The following steps are usually taken into account when defining the margin:

- calculation uncertainty due to inaccuracies of the mathematical models,
- experimental validation of the torsional natural frequencies,
- desired margin between shaft system natural frequencies and the excitation frequency,
- any specified/experienced grid frequency excursions, and
- operating temperature effects.

Mechanical parts that are connected to the main rotor body could participate in torsional vibration if not adequately designed for strength or tuned away from grid frequencies. These parts include shrunk-on couplings, coupling bolts and long steam turbine blades. Among them, blade dynamic behaviour in torsional vibration is complex and is discussed in more detail below.

4.2 Influence of blades

The mode shapes of zero-nodal diameter natural frequencies of blade rows are such that all blades in a row vibrate in phase with one another. The tangential component of such modes can therefore be excited by torsional oscillations of the shaft system. In addition, modal interaction takes place between the blades, discs and shaft system such that the resulting natural frequencies of the combined blade-disc-shaft system are different from those of the uncoupled 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 oscillations of the shaft system.

For short- and medium-height blade rows (e.g. high-pressure and medium-pressure turbines, and the first several rows of low-pressure turbines), 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 coupled shaft system, such blades can be considered as rigid and only their torsional inertias need be taken into account when calculating the shaft system torsional natural frequencies.

For longer blades (such as the last and penultimate rows of the LP turbine or the first compressor stage), the frequencies of the zero-nodal diameter modes can be within the range of, or sufficiently close to, the line and/or the twice line 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 shaft system in such a way that additional coupled shaft system modes are introduced with various combinations of blade vibration in phase and anti-phase with the shaft system. Under adverse conditions, such modes could amplify rotor/blade stresses due to external torques arising from grid disturbances. Consequently, when calculating the natural frequencies of the coupled shaft system and blades, it is necessary to model the long blades as branched systems that fully replicate the zero-nodal diameter (all-in phase) modes of these blades.

The criterion for assessing whether the blades can be represented by their torsional inertia only, or as branched systems, is as follows. If the lowest zero-nodal diameter mode of the blade row and disc (or rotor section for drum type rotors) is less than 2,5 times the nominal line frequency of the electrical grid system (i.e. 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 should be given to modelling the blade row as a branched system. Otherwise, it is only necessary to lump the total inertia of a blade row at the appropriate point in the shaft system model. In general, it could be required that the last stage LP blades (and in some cases, penultimate stage LP blades) be modelled as branched systems.

4.3 Influence of generator rotor windings

Special knowledge of the generator rotor structural design is needed for modelling the stiffness effects of the rotor body section with its copper windings and wedges.

5 Evaluation

5.1 General

This part of ISO 22266 provides two methods for the evaluation of the torsional vibration characteristics of coupled shaft systems including the blades:

a) the maximum frequency margin between the calculated natural frequencies and the relevant electrical grid system frequencies (see 5.2);

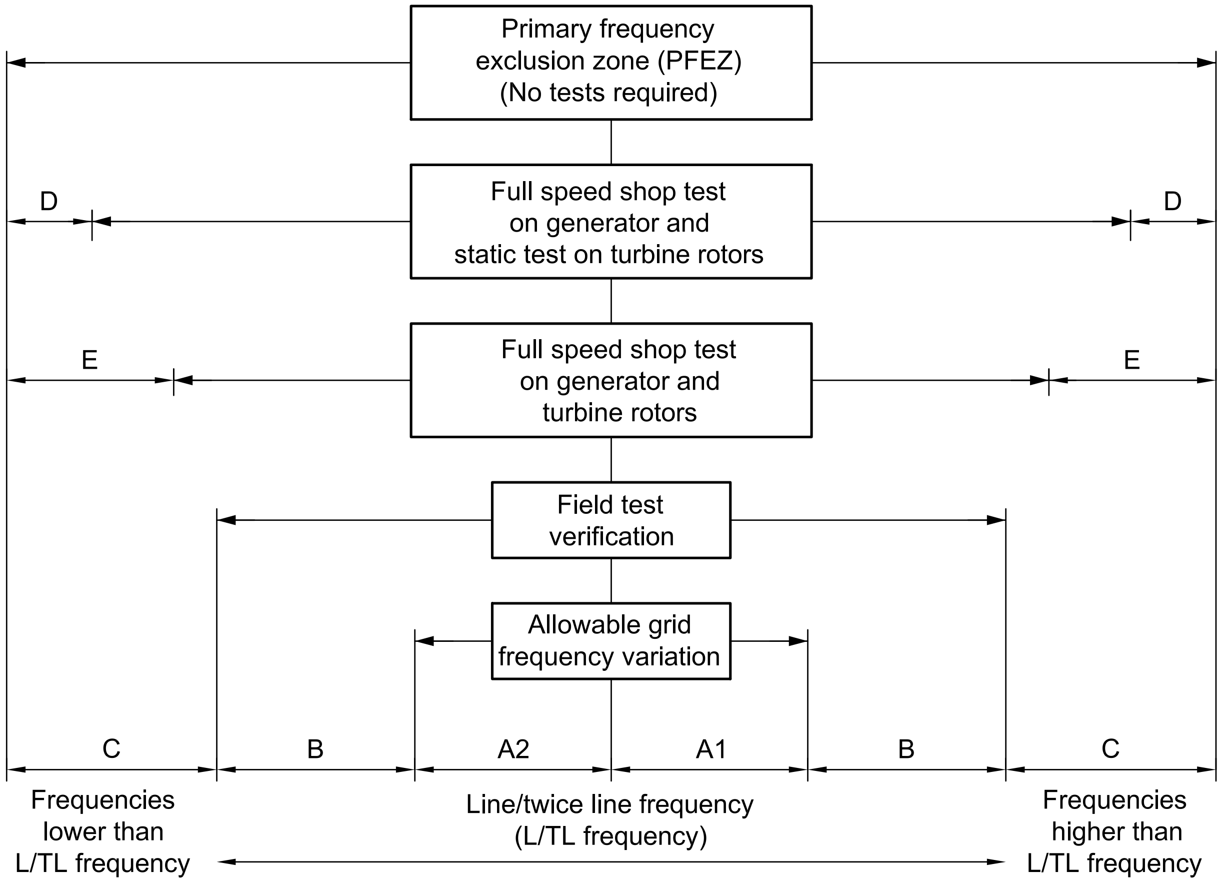
b) dynamic stress analysis to ensure that the peak stresses induced by the transient fault conditions listed in Table 1 are satisfactory (see 5.3).

Stress analysis for steady-state fault conditions is only required if the frequency margins defined in 5.2 are not achieved.

Further information regarding the calculation of torsional vibration is given in Clause 6.

5.2 Frequency margins

The objective is to provide criteria that ensure that there are no shaft system modes that can be excited by torsional oscillations of the shaft within close proximity of the line and twice line excitation frequencies of the electrical grid system. It should be noted that the shaft system frequencies and associated modes, which are insensitive to induced torsional forces, are permitted within the frequency exclusion zone (see 5.3). Calculations by the equipment supplier would indicate whether a mode is responsive or non-responsive to grid excitations. The allowable torsional frequency margins are shown in Figure 5 and given in Table 2, and are described in a) to g) below.



If tests/calculations are carried out at room temperature, the relevant upper and lower frequency limits should be increased by the temperature correction factor F.

NOTE See Table 2 for the definition of A to E.

Figure 5 — Definition of torsional frequency exclusion zone

Table 2 — Margins at line and twice line frequencies for both 50 Hz and 60 Hz machines

	Description	Frequency margin
A	Allowable upper grid frequency deviation	A1
	Allowable lower grid frequency deviation	A2
B	Margin between maximum/minimum allowable grid frequency and resonance peak	B
C	Calculation uncertainty	C
D	Reduction in calculation uncertainty if a full-speed (dynamic) shop test carried out on generator rotor and static shop test (e.g. modal testing) carried out on LP rotor	D
E	Reduction in calculation uncertainty if full-speed shop tests carried out on the rotor(s) of concern, e.g. the generator, exciter, LP rotor, or if successful operating experience is available for a similar shaft system	E
F	Temperature effects This compensates for the change in shaft stiffness in those cases where calculations or tests are carried out at room temperature instead of the normal operating temperature. F is zero if temperature effects have been taken into account.	F

a) The allowable grid frequency deviation (electrical grid frequency oscillations leading to line and twice line frequency excursions) limits, which apply for continuous full-load operation of the particular application, is identified as A. This value, together with the additional margins given in Table 2, enables users to evaluate the required frequency margins specific to their grids. This limit varies for different regions throughout the world and should be agreed between the customer and the supplier. In many cases, different upper and lower off-frequency variations (A1 and A2) are specified.

b) Margin B is required between the shaft system natural frequency and the maximum/minimum permitted grid frequency to avoid any significant dynamic amplification near resonance.

c) The confidence in the accuracy of the assessment of the torsional natural frequency of the coupled shaft system. For example, if the assessment is based on calculations alone, then a frequency margin identified by C will apply to allow for possible calculation inaccuracies. This is the case if the calculations are not validated by testing.

d) Confidence of the assessed frequency values increases if they are supported by experimental validation permitting the calculation uncertainty margin C to be reduced. The extent to which the frequency margins can be reduced will depend on the level of testing performed and the test configurations. For example, a field test on the fully installed shaft system will give a greater level of confidence than that provided by various levels of shop testing performed on individual components of the shaft system.

e) D is the reduction in margin C if a full-speed shop test on the generator and a static test on the LP rotor is carried out. E is a larger reduction margin which applies if full-speed tests are carried out on generator, LP or exciter rotors, or if successful operating experience is available for a similar shaft system.

f) Temperature influences the dynamic stiffness of rotors. Therefore, the actual operating temperature should be included in the analysis. If the calculations are carried out at room temperature, compensation F for temperature effect is required when the frequency margin is evaluated.

g) Different values of C, D and E can be applied for line and twice line frequencies, depending on customer needs and special requirements of units.

The above frequency margin types can be dependent on a number of other factors, such as the location of the power station, the integrity of the electrical network, accuracy of assessment and the operating history of the supplied hardware. The specification of numerical values for factors A to F is therefore beyond the scope of this part of ISO 22266. Examples of typical values together with the corresponding frequency margins are given for information only 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 specific application.

The torsional vibration frequencies should be acceptable if one of the following criteria are satisfied (see Figure 5).

Criterion 1

The calculated torsional natural frequencies of the coupled shaft and blade system without test verification should be outside the range specified as $+(A1 + B + C)$ and $-(A2 + B + C)$ of the nominal line and twice line system frequencies. This is the *primary frequency exclusion zone* (PFEZ), as shown in Figure 5. If this criterion is met, no test whatsoever is needed.

If the calculation method has been confirmed by means of factory tests on individual rotors, the modelling uncertainty is reduced. Hence, the required frequency margin can be reduced and the following alternative criteria can be applied:

Criterion 2

If the calculation is validated by means of a full-speed shop dynamic test on the generator rotor and a static test on an adjacent LP turbine rotor, the restricted frequency range would be $+(A1 + B + C - D)$ and $-(A2 + B + C - D)$ of the nominal line and twice line system frequency. If this criterion is satisfied, there is no requirement to carry out any further measurements to validate the calculations.

IMPORTANT — Caution should be exercised when interpreting results of static tests on bladed rotors (see Clause A.4)**.**

Criterion 3

If the calculation is validated by means of full speed shop dynamic tests on the generator rotor and an associated LP turbine, the effect of blade coupling at full speed will be fully established. In this case, the restricted frequency range could be further reduced to +(A1 + B + C − E) and −(A2 + B + C − E) of the nominal line and twice line system frequency. If this criterion is satisfied, there is no requirement to carry out any further measurements at the site.

Criterion 4

If a full-speed field test is carried out on the fully installed shaft system, the measured torsional natural frequencies should primarily lie outside the range +(A1 + B) and −(A2 + B) of the nominal line and twice line system frequency if the torsional natural frequencies are sensitive to grid frequencies. If the field test indicates that the torsional natural frequencies are insensitive to grid system frequencies, the frequency exclusion zone provided for by this criterion can be waived.

If the calculations or tests are carried out at room temperature conditions, the frequencies will be marginally higher than those under service conditions due to the influence of temperature on the modulus of elasticity. The effect of temperature on the modulus of elasticity for different materials is well established and the appropriate correction factor can be readily calculated. In such cases, the value of F will be zero. Alternatively, if such information is not available, a value provided by the equipment supplier should be applied for F, and the calculated or measured frequencies at room temperature conditions should be reduced by this factor before applying the frequency margins defined for Criteria 1 to 4.

If the above frequency margin criteria are not satisfied, action should be taken either to perform a more detailed stress analysis to confirm that the dynamic stresses are satisfactory (see 5.3) or to modify the design of critical components.

5.3 Dynamic stress assessments

Dynamic stress assessments shall be carried out to confirm the following:

a) the peak stresses induced by the transient fault conditions listed in Table 1 are satisfactory;

b) that the frequency margins specified in 5.2 are not achieved, in which case the shaft system could be acceptable if the modes of concern are insensitive to the excitation and therefore do not pose any problem to the system integrity.

In both of these cases, it is the responsibility of the supplier to demonstrate by calculation that the dynamic stresses do not exceed acceptable values or to demonstrate that the same or similarly designed machines are operating successfully in other units with comparable grid conditions. In particular, careful attention should be paid to areas of potential high stress such as coupling bolts, blade roots and those regions of the shafts with the smallest diameters.

The modelling techniques, calculation method and acceptance criteria are subject to agreement between the customer and the supplier (see 6.1).

6 Calculation of torsional vibration

6.1 General

Provided that the details of the individual shaft system components are known, it is possible to calculate the undamped, torsional natural frequencies and mode shapes of the shaft system, including blade-disc-shaft coupled effects (free vibration). Then, if the frequency margins in 5.2 are not satisfied, the response of the system to forced excitation mechanisms per Table 1 should be performed (forced vibration) and stress levels should be in accordance with the supplier's experience.

The supplier of the shaft system should be responsible for the calculation of torsional vibration using a conventional method including, where appropriate, the excitation cases to be considered and any allowable calculation simplification. The method should be agreed upon by the parties concerned.

6.2 Calculation data

The data to be taken into account for the torsional vibration calculation of the shaft system are the polar mass moment of inertia and torsional stiffness characteristics of each constituent part of the complete shaft system, its coupled blade-disc branched systems and the specific operating parameters. In addition, if it is necessary to carry out a forced vibration calculation, knowledge of the torsional vibration damping and the relevant excitation forces is required.

In some cases, the supplier might not be the original equipment manufacturer (OEM) of some of the shaft system components (e.g. the turbine and generator may be manufactured by different suppliers).

6.3 Calculation results

The results obtained using the calculation methods described can determine

a) the natural frequencies and the corresponding mode shapes, and

b) the torsional stress margins or torque in the shaft system.

6.4 Calculation report

If the contract requires a torsional vibration calculation to be carried out, a suitable report should be provided by the set supplier. The contents of the report should be decided between the customer and the supplier. In general, the report should contain leading particulars of the unit, configuration of the shaft system (including a summary of which blade rows have been modelled as branched systems) and calculation results. If the set supplier has subcontracted the calculation then, it should be clearly stated in the report.

7 Measurement of torsional vibration

7.1 General

If the initial calculation shows that there are torsional natural frequencies within the PFEZ, it is necessary to take further action. This involves either modifying the shaft system components or performing tests to validate the calculation results and confirm that the application of the reduced frequency margins defined in 5.2 is permissible. Depending on the particular circumstances, such measurements may be carried out on individual components in the factory or on the fully installed unit on site. The requirement for, and extent of, any such testing should be agreed between the set 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 PFEZ margin is satisfactory.

7.2 Method of measurement

A variety of different measurement techniques have been successfully employed in the past to measure the torsional vibration characteristics. Annex A provides further background information. However, it is emphasized that these are not the only available methods and others may be equally applicable. These methods are subject to continuous improvement. Therefore, the one that is most appropriate for a specific application will be dependent on a number of factors. Normally, the method adopted will be that which is commonly used by the set supplier. Any variation from this is subject to agreement between the set supplier and the customer. However, it should be recognized that testing of the fully coupled unit 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 set supplier to ensure that the frequency margins specified in 5.2 are met at the design/manufacturing stage, thereby avoiding the necessity of site testing.

7.3 Measurement test report

If tests are carried out, a torsional vibration measurement test report should normally be provided by the set supplier. The contents of the report should be decided between the customer and the supplier. It should contain leading particulars of the unit, configuration of the shaft system, the measurement parameters and the conditions at the test site. In addition, the type, accuracy and calibration method of the measuring equipment and the positions of the measurement sensors should be recorded. If the set supplier has subcontracted the torsional vibration measurements, then it should be clearly stated in the test report.

When the measurement conditions differ from the normal operating conditions, agreed correction factors should be used to compensate for the effect of the different conditions.

8 General requirements

8.1 Set supplier responsibilities

The supplier of the turbine generator set shall be responsible for ensuring that the torsional vibration characteristics of the shaft line are satisfactory. In those cases where different manufacturers supply the turbine and generator, the turbine manufacturer will normally be responsible for the torsional vibration assessment. It is the customer's responsibility to ensure that all generator information required to enable the torsional vibration calculation to be performed correctly is provided to the turbine manufacturer. Nevertheless, in all cases, a clear responsibility agreement should be established between the respective suppliers and customer.

8.2 Guarantees

Any guarantees that the set will operate satisfactorily with regard to torsional vibration are subject to agreement between the customer and the set supplier.

8.3 Responsibilities

Where torsional vibration calculations of the complete shaft system are requested, the supplier of the set shall be responsible for the calculations, even if he subcontracts them.

Where additional verification of the torsional vibration of the complete shaft system is required, the supplier of the set shall be responsible for the measurements carried out even when he subcontracts them. In particular, the supplier should select, in agreement with the customer or the inspection organization representing him, the methods of measurement to be used.

If there is a range of operating conditions for the set where the vibrations could cause damage, the supplier of the set should, with the agreement of the other parties, take the necessary steps to eliminate the critical vibrations or ensure that appropriate procedures are in place to avoid operating under these conditions.

Any necessary corrective actions to modify the shaft system are the responsibility of the set supplier and shall be agreed upon by the relevant component manufacturer or manufacturers, if different.

In case of partial retrofit of rotor components by a non-OEM supplier, there should be clear agreement between the new supplier and the customer with regard to the torsional vibration characteristics of the retrofitted shaft line.

Annex A
(informative)

Torsional vibration measurement techniques

A.1 General

A variety of different measurement techniques have been successfully employed in the past to measure the torsional vibration characteristics of coupled shaft systems. 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 and that other equally applicable methods developed by the different OEMs are not discussed here.

A.2 Torsional vibration transducers

The following devices may be used as torsional vibration transducers:

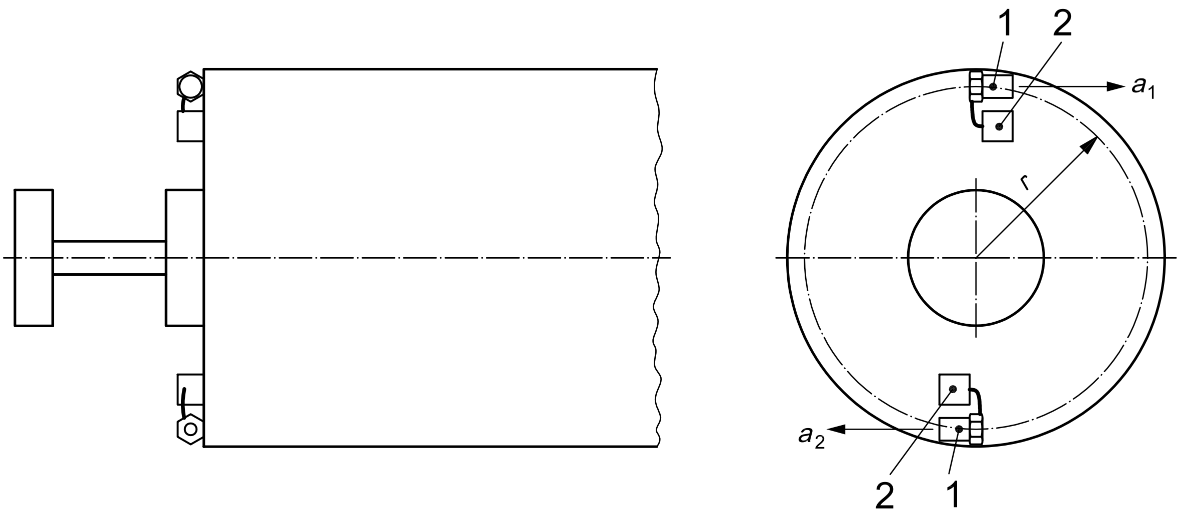
a) eddy current probes, inductive probes, lasers, etc. (non-contacting transducers);

b) strain gauges;

c) optical decoders;

d) accelerometers positioned circumferentially at specified angles (preferably at 0° and 180°, see Figure A.1 and, for example, Reference [2], Figures 8 and 9).

Other methods may be used by agreement between the customer and set supplier.



Key

1 accelerometer

2 radio transmitter

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

Figure A.1 — Circumferential accelerometer positions

A.3 Measurement parameters

Depending upon the method of measurement, it is recommended that the following be included in the test report:

a) rotational velocity of the shaft system;

b) turbine generator set power output;

c) torsional vibration magnitude;

d) torsional strain;

e) ambient temperature of test site;

f) torsional natural frequency;

g) speed range over which measurements are carried out.

An additional parameter that can influence the torsional vibration is

h) blade-disc coupled nodal diameter frequency.

Other parameters can be measured on agreement between the customer and set supplier.

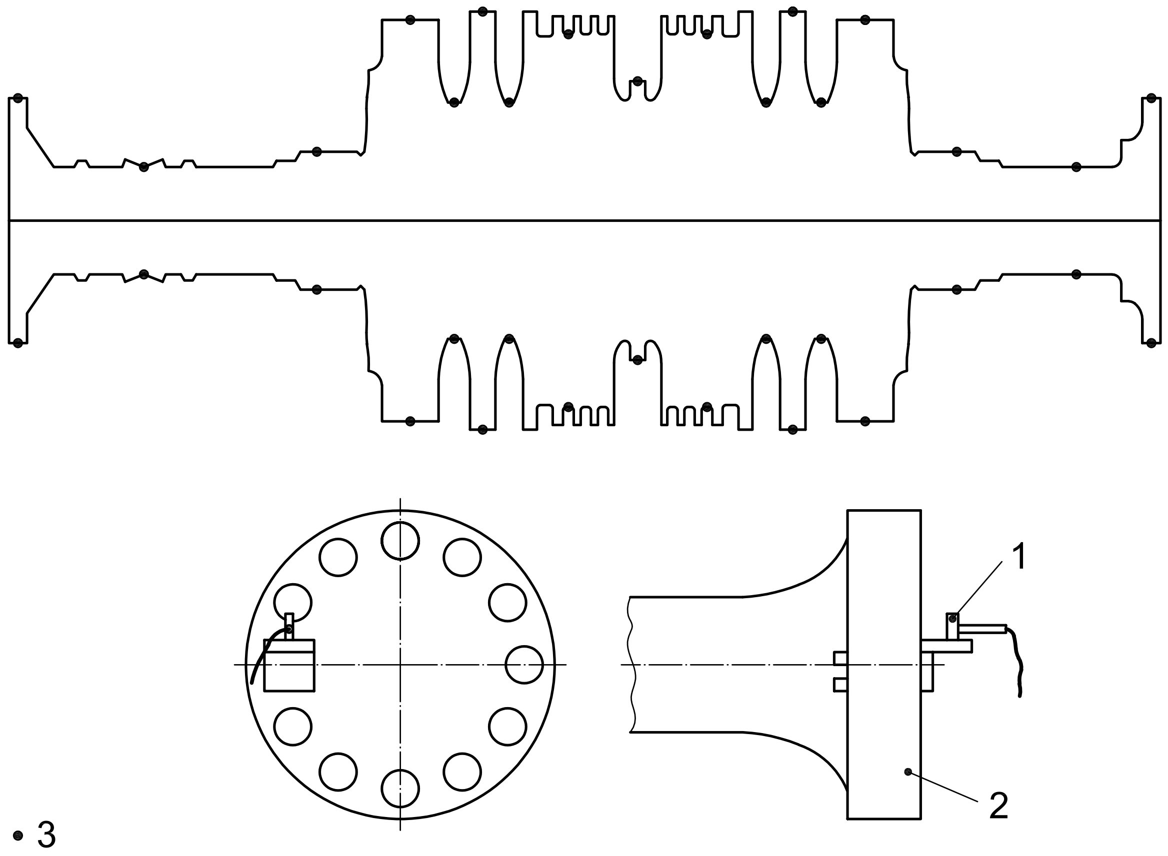
A.4 Factory static tests on stationary rotors

"Modal tests" may be performed on rotors in their static (i.e. non-rotating) configuration in the factory. This is one way of measuring mode shapes and natural frequencies under static conditions. They provide verification of predicted behaviour and, therefore, help to calibrate shaft system models for fundamental rotor body and overhang 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.

When blades are mounted on rotors, a perfect contact between blade roots and the main rotor body may not always be achievable when the rotor is stationary, due to the design of root employed or manufacturing tolerances. As a result, the impact energy imparted by a test hammer to the structure may be disrupted at the blade-rotor body contact areas, making it difficult to capture blade-disc and rotor coupled frequencies and their associated modes. Even if such modes are captured in the test, they are less useful because the blade-disc frequencies will continuously change with speed. In other words, the blades dynamically couple with the rotor and this will continue until rated speed is reached. This coupling effect, along with stress stiffening due to speed, creates new sets of torsional frequencies that are different from individual blade and disc alone frequencies under stationary conditions. Similar difficulties exist when carrying out static tests on generator rotors due to the influence of the copper windings and associated wedges.

Therefore, although static "modal tests" are helpful to calibrate rotor body models, they do not generally provide an accurate assessment of the 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.

Figure A.2 shows the arrangement for a typical factory static rotor test.



Key

1 impact hammer

2 end coupling flange

3 measurement locations

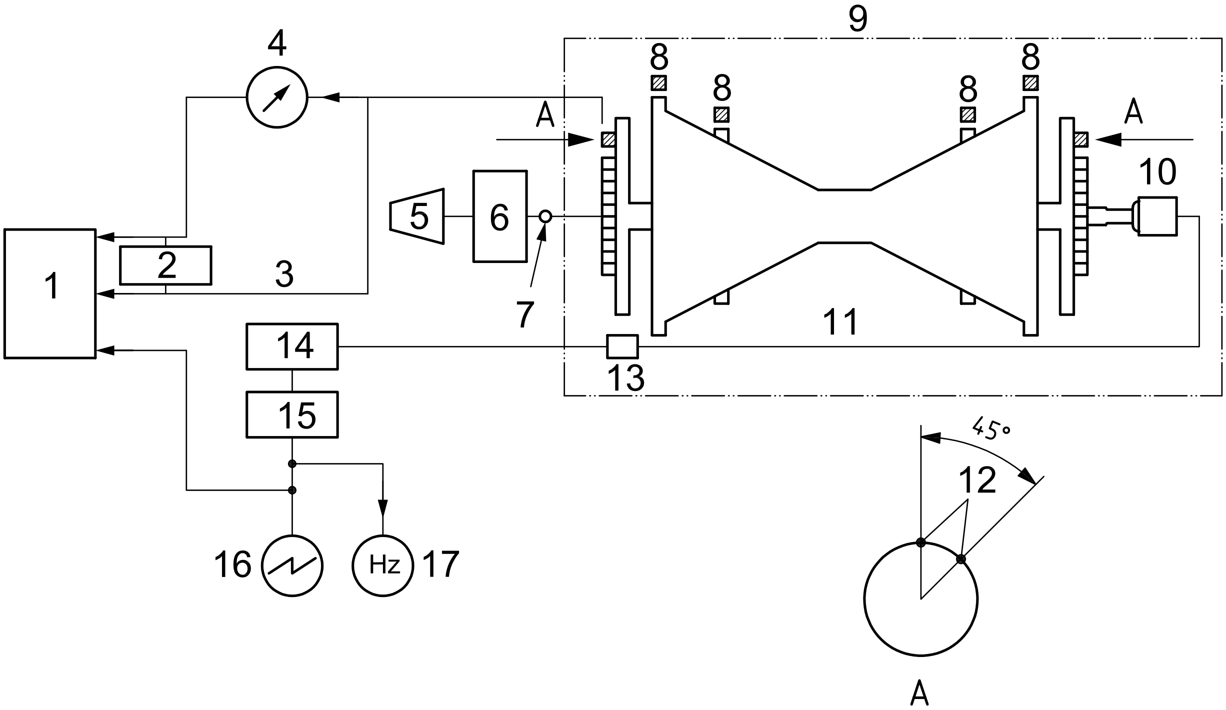
Figure A.2 — Factory static test

A.5 Full-speed (dynamic) factory tests

When blade-disc coupled frequencies are calculated in the proximity of twice line frequencies, the only reliable way to verify them is to perform full-speed dynamic tests in the factory. OEMs routinely perform such tests to verify new designs and confirm the accuracy of analytical models. At one extreme, it provides the OEM with a final opportunity to modify the design before it leaves the factory. Since the dynamic tests are conducted at the OEM's facility, the total cost is significantly lower and less time-consuming than that for the on-site test.

A continuously rotating torsional exciter system, similar dynamic exciter systems or other appropriate excitation methods can be used to perform full-speed tests in the factory (such exciters are commercially available). In general, the excitation system is attached to one end of the rotor coupling. The torsional vibration magnitudes are measured using non-contacting proximity transducers at gear tooth wheels that are mounted along the rotor as required or at any targets that have equal spacing, such as blade tip areas or alternate black and white strips on a shaft or disc surface. In some cases, strain gauges are used on the shaft and/or blade surfaces. Torsional 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 the rated speed range. The dynamic tests performed for the LP and generator rotors in particular are very helpful for improving model accuracy and in the use of the information for calibration of similar rotor configurations.

Figure A.3 shows a schematic view of the test set up 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 box
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.3 — Factory dynamic test of a rotor

A.6 On-site torsion tests

The various factory tests described above are an extremely helpful way of accurately calibrating the analytical rotor models. Nevertheless, when rotors are connected through couplings, some amount of uncertainty remains, and hence — as described in 5.2 — this is taken into account when specifying the allowable frequency margins. The degree of modelling inaccuracy tends to increase for higher modes, for example those whose frequencies are in the proximity of twice line 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 5.2. In those cases where the calculated frequencies do not satisfy the criteria defined in 5.2 and the corresponding modes are responsive, the uncertainties discussed above can justify a field torsion test.

NOTE Tests show that system damping ratios vary from mode to mode and typically range anywhere between 0,1 % and 0,4 %.

Depending on the type of mode that is critical, a field test could involve measurements at a few locations or be performed on a more elaborate scale. The choice of measurement positions is normally determined by examination of the predicted mode shapes, but in most cases the measuring of torsional vibration magnitudes at two locations, such as the turbine to generator shaft region and the permanent magnet generator, is sufficient to capture the important turbine and generator coupled rotor 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 order to measure torsional natural frequencies at a power plant site, it is necessary to provide a means of detecting torsional vibration at one or more locations along the shaft. This can be done by various means: toothed wheels and magnetic or proximity transducers, painted or taped stripes and optical transducers, permanent magnet generator (PMG) voltage signals, strain gauges, etc. The demodulated torsional signals are then usually displayed on a spectrum analyzer 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 their relative magnitudes at each natural frequency enable the measurement of mode shapes.

When a turbine generator is assembled at a power plant site, it becomes a challenge to measure torsional natural frequencies, particularly those in the vicinity of twice line frequency. Sub-synchronous (below-line frequency) torsional natural frequencies can typically be easily measured with the unit on-line, as they are usually excited to measurable levels by random power system fluctuations. If necessary, synchronization or line switching tests can be done to excite these sub-synchronous torsional modes. Such techniques, however, may not be successful in exciting and measuring super-synchronous (above-line frequency) torsional natural frequencies with much confidence. This is why in the mid-1970s the off-line torsional frequency response test was developed.

An off-line torsional frequency response test involves exciting the generator in a controlled manner using oscillatory torque developed from unbalanced currents flowing in the generator stator. 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 unit is shut down and not connected to the grid. With the turbine generator running at various speeds, a small amount of field excitation is applied to the generator, which induces unbalanced or negative sequence current in the generator. The field excitation is applied in a controlled amount small enough not to exceed the negative sequence current heating limits of the generator but large enough such that the induced negative sequence currents can excite shaft system natural frequencies to measurable levels at resonance.

The generator air gap torque induced by these negative sequence currents occurs at a frequency equal to twice the electrical frequency of the generator. By changing the speed of the generator during this test (possible because the generator is off-line), the electrical frequency and thus the frequency of the air gap torque also changes. Thus, by slowly ramping the speed, a slow sweep of the air gap torque frequency is obtained. In this manner, torsional natural frequencies can be detected by the occurrence of resonant peaks in the torsional response as the speed is ramped. Later in the test, the speed can be held at each of these resonant peaks in order to measure precisely the torsional natural frequency, and also to confirm that the measured response is indeed due to torsion, by removing the field excitation (and thus the air gap torque) and observing that the response magnitude changes accordingly.

For several years, the off-line frequency test was the only reliable method of accurately identifying the torsional natural frequencies of the fully installed unit at site. However, 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. These disturbances, which are a consequence of the minor random disturbances that are inevitably present on all electrical networks, cause transient excitation of those natural modes that can be excited by the generator. 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 the normal operation of the power plant.

Annex B
(informative)

Examples of frequency margins relative to line and twice line frequencies for shaft system modes that can be excited by torsional oscillations of the shaft

A1, A2, B, C, D, E and F defined in 5.2 and Table 2 are dependent on a number of factors specific to the location of the power station, the integrity of the electrical network it supplies, the accuracy of frequency assessment and the operating history of the machines. The specification of numerical values for these factors is therefore beyond the scope of this part of ISO 22266. Example values for these factors are given in Table B.1. However, it is emphasized that these will normally vary for different applications. The values given in Table B.1 are general guidelines that will ensure in most cases the avoidance of gross deficiencies or unrealistic requirements. Alternative values can be used, provided that these can be justified by further technical assessment (see 5.3) or proven experience for similar machine types. The actual values to be used for a particular application are subject to agreement between the customer and the supplier for the specific application.

Table B.1 — Examples of frequency margin factors

	Frequency margin
Allowable grid frequency deviations [a], A1	2,5 %
Allowable grid frequency deviations [a], A2	2,5 %
Margin from resonance, B	1 %
Calculation uncertainty, C	2,5 %
Full-speed shop test on generator and LP static test, D	1 %
Full-speed shop tests on LP and generator rotors or operating experience, E	2 %
Temperature effect (zero if temperature is included in the analysis), F	1 %

[a] Frequency deviations depend on the particular grid system (50 Hz or 60 Hz).

The actual line and twice line frequency margins corresponding to the above factors and for each of the criteria defined in 5.2 are listed in Table B.2 for both 50 Hz and 60 Hz grid systems.

Table B.2 — Frequency margins for turbine generator sets operating on 50 Hz and 60 Hz grid systems

Criterion	Frequency margins	Required frequency margins Hz			
		50 Hz	100 Hz	60 Hz	120 Hz
1	+(A1 + B + C)	+3	+6	+3,6	+7,2
1	−(A2 + B + C)	−3	−6	−3,6	−7,2
2	+(A1 + B + C − D)	+2,5	+5	+3	+6
2	−(A2 + B + C − D)	−2,5	−5	−3	−6
3	+(A1 + B + C − E)	+2	+4	+2,4	+4,8
3	−(A2 + B + C − E)	−2	−4	−2,4	−4,8
4	+(A1 + B)	+1,75	+3,5	+2,1	+4,2
4	−(A2 + B)	−1,75	−3,5	−2,1	−4,2

Applying the above frequency margins for Criteria 1 to 4 as defined in 5.2, the frequency avoidance range can be summarized as follows:

a) **50 Hz machines**

— **Criterion 1**

Between 47,0 Hz and 53,0 Hz for line and between 94,0 Hz and 106,0 Hz for twice line frequencies.

— **Criterion 2**

Between 47,5 Hz and 52,5 Hz for line and between 95,0 Hz and 105,0 Hz for twice line frequencies.

— **Criterion 3**

Between 48,0 Hz and 52,0 Hz for line and between 96,0 Hz and 104,0 Hz for twice line frequencies.

— **Criterion 4**

Between 48,25 Hz and 51,75 Hz for line and between 96,5 Hz and 103,5 Hz for twice line frequencies.

b) **60 Hz machines**

— **Criterion 1**

Between 56,4 Hz and 63,6 Hz for line and between 112,8 Hz and 127,2 Hz for twice line frequencies.

— **Criterion 2**

Between 57 Hz and 63,0 Hz for line and between 114 Hz and 126,0 Hz for twice line frequencies.

— **Criterion 3**

Between 57,6 Hz and 62,4 Hz for line and between 115,2 Hz and 124,8 Hz for twice line frequencies.

— **Criterion 4**

Between 57,9 Hz and 62,1 Hz for line and between 115,8 Hz and 124,2 Hz for twice line frequencies.

Annex C
(informative)

Commonly experienced electrical faults

As seen in Table 1, all transient disturbances generally introduce a step change in torque. In each case, such a change is followed by a low frequency (typically around 1 Hz to 2 Hz) oscillation that decays in magnitude as the power system returns to equilibrium or until another disturbance occurs. These step changes in torque tend to excite the subsynchronous torsional natural frequencies of the shaft system, typically causing the highest stresses in connecting shafts and coupling bolts. Oscillatory air gap torques at one or two times line frequency can also be induced, as shown in Table 1.

Power system faults involve both a step change in air gap torque and a temporary oscillatory torque with a frequency equal to line frequency (50 Hz or 60 Hz). For all the transient disturbances that indicate that they have a component of line frequency torque in Table 1, this torque component is the result of a transient dc current that is induced in the generator stator as a result of the disturbance. This decaying dc current interacts with the rotation of the generator rotor to create transient torques at line frequency.

Unbalanced system conditions, both transient and steady-state, induce ac generator phase currents that are unbalanced among the three phases. These unbalanced phase currents, which can be expressed in terms of negative sequence currents, induce generator air gap torque oscillations at twice line frequency.

Referring to Table 1, a three-phase fault is a short circuit that occurs simultaneously on all three phases and induces both a step change in torque and an oscillatory torque at the line frequency. An unbalanced fault is a short circuit that involves only one or two phases and induces an additional oscillatory torque at twice line frequency, due to the resulting negative sequence currents.

The severity of these faults, and thus the magnitude of the resulting torques, depends upon the electrical impedance between the generator and the fault. A fault occurring electrically near a generator will thus tend to induce higher magnitude torques than for a more remote fault. The highest severity fault, which includes torques at both line and twice line frequency, is an unbalanced phase-to-phase fault at the generator terminals.

Another aspect to consider when evaluating faults is the duration of the fault. Most faults occur on transmission lines in which the effect of the fault on the grid can be removed quickly (in a few cycles) by circuit breaker action. In such cases, the oscillatory torques applied to a generator can occur for a very short time. Another consideration is that the act of clearing the fault can introduce a second step change in torque, which has the possibility of reinforcing torsional oscillations that occurred as a result of the fault itself.

Transmission line switching (opening and closing) also produces step changes in air gap torque in generators. The magnitude of these step changes depends upon the power distribution or load flow within the grid and the specific line that is being switched. The step change in torque due to line closing is typically greater in magnitude than that due to line opening. Also, line closing introduces an additional air gap torque component at line frequency.

As discussed above, the opening of a transmission line to clear a fault introduces a step change in generator air gap torque. This typically involves opening all three phases of the transmission line. When an unbalanced transmission line fault involves only one phase, sometimes an attempt will be made to isolate and clear only that phase. Single-pole switching is the tripping and re-closing of one pole of a multi-pole circuit breaker without opening the other two phases. As a result of the unbalanced conditions that occur when opening only one phase of a transmission line, single-pole switching introduces an oscillatory component of air gap torque at twice line frequency, which persists until the pole is re-closed.

Synchronizing a generator to the grid out-of-phase (with the angle across the synchronizing breaker not exactly 0°) will induce an air gap torque with both step change and an oscillatory component at line frequency, as shown in Table 1. For normal synchronizations that are only slightly out-of-phase, these torque magnitudes will be small and of no consequence. However, for severely out-of-phase synchronizations, these torques can be extremely large.

Line and load unbalance occurs in varying degrees on all transmission systems, resulting in continuous non-zero levels of negative sequence currents in almost all generators and, consequently, a continuous oscillatory air gap torque at twice line frequency. Line unbalance is due to the impedances among the three phases of transmission lines not being perfectly equal. Load unbalance is due to the loads on a transmission system not being perfectly equal among the three phases. Severe or abnormal levels of line or load unbalance will increase the magnitude of this oscillatory torque at twice line frequency. Off-frequency events will shift the frequency of this torque accordingly.

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