ASSESSING TORSIONAL VIBRATIONS IN TURBOGENERATORS BY MEANS OF MODERN SENSOR- AND MONITORING-TECHNOLOGIES

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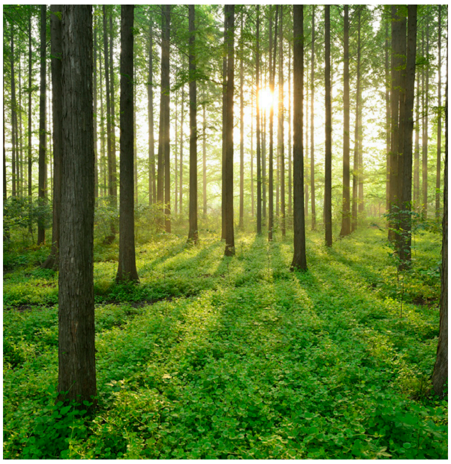

VIBRATIONS IN
NUCLEAR APPLICATIONS

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Assessing Torsional Vibrations in Turbogenerators by means of modern Sensor- and Monitoring-Technologies

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Foreword

Torsional vibrations in turbogenerator shaft systems are a critical issue for nuclear power plants, as excessive oscillations can lead to fatigue damage, reduced component lifetime, and potential operational disturbances affecting reliability and safety.

The purpose of this study is to assess the characteristics, causes, and potential impacts of torsional vibrations in turbogenerators, and to improve understanding of how operating conditions and system interactions influence vibration behaviour in nuclear power applications.

The results show that torsional responses are influenced by system design, electrical interactions, and transient events. The study concludes that improved modelling, monitoring, and consideration of critical operating scenarios are essential to manage risks and ensure reliable long-term operation.

This report forms the results of a project performed within the Energiforsk Vibrations in Nuclear Applications Program, which is financed by Vattenfall, Uniper, Fortum, TVO, Skellefteå Kraft and Karlstads Energi. The Vibrations program aims to increase the knowledge of causes, monitoring and mitigation of vibrations, thereby contributing to the safety, maintenance and development of a diverse range of machinery in the Nordic nuclear power plants.

The study was carried out by Dr. Rainer Nordmann, Technical University Darmstadt.

These are the results and conclusions of a project, which is part of a research Program run by Energiforsk. The author/authors are responsible for the content.

Summary

Turbogenerators, consisting of several steam turbines and the generator are important components in Nuclear Power Plants. Their main task is to convert thermal power of the steam into mechanical power of the shaft train, followed by the conversion into electrical power in the generator. The conversion process from thermal to mechanical power is performed in the turbine blades via steam forces leading to the rotational mechanical energy of the Turbogenerator shaft train. The conversion process from this mechanical power to electrical power takes place in the air gap of the Generator. Although the complete conversion process usually works very well the shaft train of the Turbogenerator may also experience problems due to mechanical vibrations, for example torsional vibrations which are the main subject of this project. Besides the requirement of a good energy conversion process it is therefore also necessary to assure a smooth operation of the Turbogenerator with an admissible vibration behavior of the shaft train and the connected blades.

In case of disturbances in the electrical system (Short circuits, unbalanced phases, Sub Synchronous Resonances) time dependent air gap torques can excite the shaft train to strong torsional vibrations. Due to the high air gap torque amplitudes and the low damping in the Turbogenerator the Torsional Vibrations may become very large. This may lead to material fatigue, to fracture (cracks) with a possible stop of the operation or in the worst case to catastrophic damages with high financial losses. Considering the possible torsional vibrations for the sensitive components shaft train and blades, the plant operator should carefully observe the mechanical stress state at critical locations of these components during the operation. For this task sensors are used to measure the torsional vibrations in terms of torsional angle, torsional velocity and/or mechanical stresses. By monitoring methods with an automatic analysis of the measured data the torsional vibrations and the mechanical stresses can be assessed.

The title of this research project is “Assessment of Torsional Vibrations in Turbogenerators by means of modern Sensor- and Monitoring-Technologies”. It has to be mentioned, that the assessment is not only based on experimental methods, but uses also the tools of Theoretical Modeling and Numerical Analysis for a comprehensive analysis of the vibration problem. Therefore this report contains besides Existing and New Sensor- and Monitoring-Technologies also the parts of Theoretical Modeling and Numerical Analysis for Torsional Vibrations of Turbogenerators. Already existing Measurement and Monitoring Tools of Turbogenerator manufacturers are presented and discussed.

Keywords

Turbogenerators; Electromechanical Interactions; Air Gap Torques; Torsional Vibrations; Measurements and Monitoring; Assessment of Stresses;

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1 Torsional Vibrations in Turbogenerators

Turbogenerators, consisting of several steam turbines and the generator are important components in Nuclear Power Plants. Their main task is to convert thermal power of the steam into mechanical power of the shaft train, followed by the conversion into electrical power in the generator. The conversion process from thermal to mechanical power is performed in the turbine blades via steam forces leading to the rotation of the Turbogenerator shaft train with high mechanical energy. The conversion process from this mechanical power to electrical power takes place in the air gap of the Generator. Although the complete conversion process usually works very well the shaft train of the Turbogenerator may also experience problems due to mechanical vibrations, e.g. Torsional Vibrations which are the main subject of this project [1], [2], [4].

1.1 SOURCES OF TORSIONAL VIBRATIONS IN TURBOGENERATORS

In case of disturbances in the electrical system (Short circuits, unbalanced phases, Sub Synchronous Resonances SSR) time dependent air gap torques can excite the shaft train to strong torsional vibrations. The connected blades are vibrating as well. While the main sources of the Torsional Vibrations are the electromechanical interactions in the air gap, the blade vibrations are mainly caused by the base excitation from the shaft.

1.2 EFFECTS OF TORSIONAL VIBRATIONS IN TURBOGENERATORS

Due to high electromechanical air gap torques and the low system damping in the Turbogenerator the Torsional Vibrations may become very high for the turbine shaft. The connected turbine blades may experience heavy vibrations as well. This can cause material fatigue, fractures (cracks) with a possible stop of the operation or in the worst case catastrophic damages with high financial losses.

1.3 PURPOSE OF THE PROJECT: ASSESSING TORSIONAL VIBRATIONS

Considering the possible torsional vibrations for the sensitive components shaft train and blades, the plant operator should carefully observe the mechanical stress states at critical locations of these components during the operation. For this task sensors are used to measure the torsional vibrations in terms of torsional angle, torsional velocity and/or mechanical stresses. By monitoring methods with an automatic analysis of the measured data (signal processing, frequency analysis, etc.) the torsional vibrations can be assessed. Due to the high risk of interactions between the electrical grid and the Turbogenerator in a power plant a robust, reliable and accurate Monitoring System for torsional vibrations is needed. For an optimal realization of such a system besides the required Experimental methods also the other tools of machine dynamics should be used, which are: Theoretical Modelling and Numerical Analysis (see chapter 2). The main purpose of the planned research project is to present existing and new Sensor- and Monitoring-Technologies, which can be used for the assessment of Torsional Vibrations in Turbogenerators. Sensor- and Monitoring-Technologies are particularly focused on the tool of Experimental Analysis. However, as mentioned before for a good assessment of the measured results the tools of Theoretical Modelling and Numerical Analysis are of importance and will be included in this project as well.

1.4 SCOPE OF THE PROJECT

The Energiforsk research project will include the following topics, subdivided in scientific basics and practical applications:

- Torsional Vibrations of Turbogenerators
- Numerical and Experimental Methods to Assess Torsional Vibrations
- Modelling of Torsional Vibrations
- Torsional Natural Frequencies, Mode Shapes, Damping
- Torsional Vibrations due to Electromechancal Interactions
- Presentation and discussion of Existing and New Sensor Techniques to measure Torsional Vibrations of Turbogenerators
- Presentation and discussion of Excisting and New Monitoring Technologies for the Assessment of Torsional Vibrations
- Presentation and discussion in more detail the developments from Siemens to measure and to monitor Torsional Vibrations in Turbogenerators. This will include the application of the developments in the Unit OL3 of the NPP Olkiluoto (Finland).
- Presentation and discussion in more detail the developments from GE (General Electric) to measure and to monitor Torsional Vibrations in Turbogenerators. This will also include the application of the developments in Power Plants.

2 Numerical and Experimental Methods for Assessing Torsional Vibrations

Vibrations in Rotating Machinery have been investigated in both Academia –Universities and Research Institutes - as well as in modern Industry for a long time. Such investigations are based on different disciplines of Physics, mainly Mechanics and Vibrations, Fluid Dynamics and nowadays more and more by Mechatronics, Electromechanics, Control and Digital Techniques. Over the years Methods and Technologies have been further improved. They help the design engineers, the test engineers and later the operator to guarantee accepted vibrations of Rotaing Machinery over the machines life time. This is possible by using well developed tools for machine dynamics and Rotordynamics. The three most important tools are presented in this chapters and it will be shown how these tools are used for Assessing Torsional Vibrations in Turbogenerators during the design and testing phase and later during the longterm operation in the Nuclear Power Plant.

2.1 THE THREE TOOLS: THEORETICAL MODELING, NUMERICAL ANALYSIS AND EXPERIMENTAL ANALYSIS

With the three tools presented in Figure 1: Theoretical Modeling, Numerical Analysis and Experimental Analysis the three tasks in Rotordynamics: Simulation, Validation and Identification can be solved in the design and testing process and during the longterm operation of Rotating Machinery.

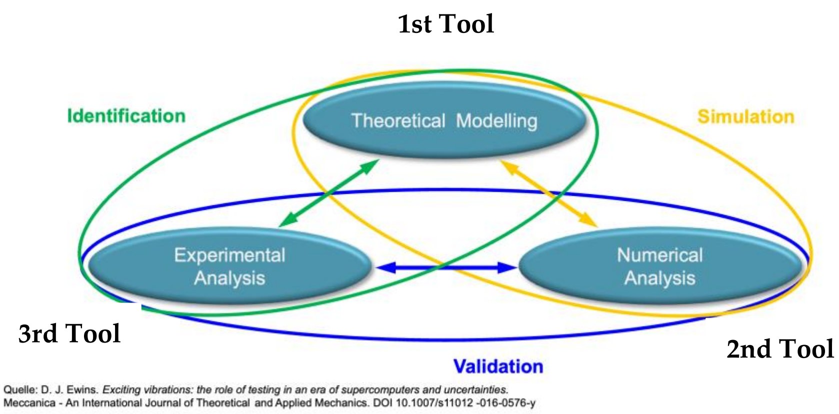


Figure 1 Tools and Tasks for the Solution of Vibration Problems in Rotating Machinery

Based on the physical laws of Mechanics (Newton's Law) theoretical models can be built up, which describe the dynamic behavior of a rotating machine, like in our application the Torsional Vibrations of a Turbogenerator. By using the tool of Theoretical Modeling the consideration of strong interactions of the mechanical system with other disciplines, like Fluid Dynamics, Electromechanics, Control, etc.. is in many cases necessary. In the following two subchapters we describe the application of the three tools for Turbogenerators during the phase of design and testing and during the following phase of long term operation with monitoring of the vibrations.

2.2 THEORETICAL MODELING, NUMERICAL ANALYSIS AND EXPERIMENTAL ANALYSIS DURING THE DESIGN AND TESTING PHASE

In the design and testing (commissioning) phase of a Turbogenerator the design engineer uses at first the tool of Theoretical Modeling (e.g. Finite Element Method) together with the second tool of Numerical Analysis to predict the torsional (and also lateral) vibration behavior by means of the first task of **Simulation**. Later, when the Turbogenerator has been built up, measurements of the Torsional Vibrations will be taken during the testing and commissioning phase. In the second task: **Validation** of the Torsional Vibration behavior (see Fig. 1) the third tool of Experimental Analysis (Testing) is needed. This means, that the Torsional Vibrations are measured by sensors and the measured results are compared with the calculated results from the Numerical Analysis (second tool). If there is a good correlation the Theoretical Model describes the real vibrations very well and can be considered as a by means of measurements validated model, which can be used for further Simulations. In case of deviations between the calculations and the measurements incorrect physical model parameters (Inertia, Damping, Stiffness) in the Theoretical Model might be the reason. In this case an adjustment of the parameters can be achieved by **Identification** (see Fig. 1) which works with the combination of the tools Theoretical Modelling and Experimental Analysis. With the validated model baseline values can be determined for different operating conditions, which will later be used as reference values for the Monitoring in the operation phase.

In the design and commissioning phase the Standard ISO 22266-1 is used. This standard is primarily a design and evaluation standard for steam turbine-generator sets under electrical excitation. Its scope is the assessment of torsional natural frequences and component strength of the coupled shaft train under normal operating conditions, especially with respect to grid-frequency and twice-frequency excitation. The ISO 22266-1 design standard is not written for the long term operation as a day-to-day monitoring limit standard similar to other known standards like ISO 20816, e.g. for lateral vibrations. However, by means of ISO 22266-1 baseline values at defined operating conditions can be prepared for the long term Monitoring process.

2.3 THEORETICAL MODELING, NUMERICAL ANALYSIS AND EXPERIMENTAL ANALYSIS DURING THE OPERATION

The title of this research project is „Assessing Torsional Vibrations of Turbogenerators by means of modern Sensor- and Monitoring-Technologies. With respect to the above presented tools and tasks it seems, that this project is mainly concentrating on the tool of Experimental Analysis, in which torsional vibrations are measured by sensors at specified locations of the Turbogenerator and the measured data are analyzed with a following interpretation and assessment by Monitoring. It has to be mentioned, that the here described Monitoring for Torsional Vibrations is not comparable with the Monitorig for Lateral Vibrations, where measured lateral vibrations at defined shaft/bearing locations are opposed to fixed sandard values, which are subdivided in zones A/B/C/D (see ISO 20816, ISO 7919).

Torsional Monitoring in the operation phase -as described in ISO 17359 „Condition Monitoring and Diagnostics of Machines – General Guidelines" – is based on baseline values established at known operation conditions, on trending of the measured torsional quantities over time and on comparison with machine-specific allowable stresses (Fatigue usage). If for example during operation shaft torsional stresses are measured in the time domain the monitoring value can be used in the following ways:

Peak Surveillance: Looking for unusually high stress peaks during grid disturbances, load rejection, negative sequence conditions or network faults

Spectral Monitoring: Tracking of critical torsional frequencies. Changes of the torsional natural frequencies (modes) and changes of their amplitudes (damping, stability). Change of response amplitudes of excitation frequencies, e.g. grid frequency and twice grid frequency. Logging of transient vibration events and use of data for fatigue accumulation

Fatigue/ Lifetime Consumption Monitoring: Conversion of measured torsional response into local shaft/blade stresses and then into a cumulative fatigue usage. The lifetime consumption of a unit can then be tracked over time.

For Torsional Vibrations the damage mechanism is fatigue, which is influenced by a variety of factors: the most stressed location can differ from mode to mode, the allowable level depends on local geometry, stress concentration, material, mean stress and even the measurement location. If we search for comparison values in relation to the measured data, this means such comparison values are machine specific and they have to be developed considering:

- the OEM Torsional Model with torsional natural frequencies &mode shapes
- the shaft/blade geometry and local stress concentration factors
- the material fatigue properties
- the measurement location and the
- commissioning baseline measurements

In other words: The operator of the Turbogenerator should not only compare measured shaft stresses with raw material numbers. Instead he should include torsional mode shapes, local notch effects, mean stress and the fact that the selected sensor location may not be the worst stress location. By means of the three presented tools it is possible to provide a conversion from the measured sensor signal to local stresses at critical shaft sections or blades and from there to allowable event stress and /or cumulative fatigue damage.

Following this idea the combination of the three tools Theoretical Modelling, Numerical Analysis and Experimental Analysis results finally in the idea of a Digital Twin (Fig. 2). With measurements of shaft torsional vibrations and the machine specific Theoretical Model the stress levels at every point of the shaft or at the blades can be inferred and critical inspection locations can be identified.

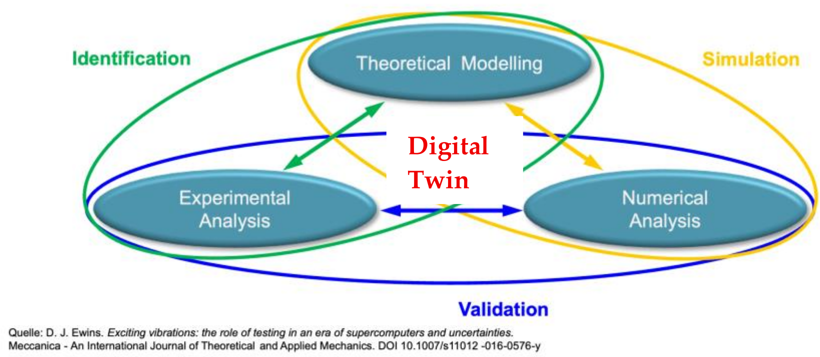


Figure 2 A Digital Twin for Assessing Torsional Vibrations in Turbogenerators

A more detailed Monitoring concept for the operation of Turbogenerators will be presented in chapter 7 of this report

3 Finite Element Modelling (FEM) of Torsional Vibrations of Turbogenerators

Torsional Vibrations of Turbogenerators, consisting of several steam turbines, the generator and the exciter can be modeled by means of the Finite Element Method (FEM). By application of this method the equations of motion of the Turbogenerator can be built up, expressing the dynamic equilibrium of the torsional inertia-, damping- and stiffness characteristics and the excitation torques. The linear equations of motion can be presented in global matrix form, which describe the relations between the vibration response q(t) of the torsional angles and the time dependent Air Gap torques $\mathbf{M}_{el}$ (t). The Theoretical Modeling of the Turbogenerator can be very helpful for the Assessment of the Torsional Vibrations during the Operation.

3.1 FE-MODEL FOR TORSIONAL VIBRATIONS OF TURBOGENERATORS

Torsional vibrations of Turbogenerators can be modelled by means of Finite Elements leading to global mass- damping- and stiffness-matrices **M**, **D**, **K** and an Air Gap torque vector $\mathbf{M}_{el}$ for the excitation due to electromechanical interactions with the electrical system (grid).

The mechanical elements to build up the equations of motion are mainly the cylindrical elements of the shaft train and the attached turbine blades The cylindrical shaft train elements, as shown in fig. 3 for a general turbogenerator influence the dynamic behaviour with their moments of inertia in the mass matrix **M** and with their torsional stiffness in the stiffness matrix **K.**

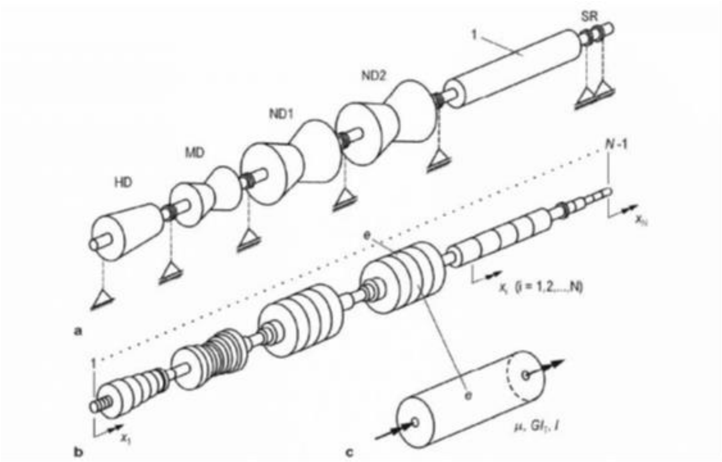


Figure 3 Finite Element Model for Torsional Vibrations of a Turbogenerator

The turbine blades with large moments of inertia contribute mainly to the polar moment of inertia terms in the mass matrix **M**. However, especially the last stage blades in the Low Pressure Turbines (LPT) have also to be considered in the stiffness matrix **K**. The flexible beam elements usually influence the torsional natural frequencies between 30 to 120 Hz. In this frequency range the blades form a dynamic rotor-blade interaction system with the shaft. Furthermore the dynamic behaviour of the blades depend on the rotational speed of the shaft train due to the stiffening effect by centrifugal forces.

3.2 THE EQUATIONS OF MOTION OF THE FE MODEL FOR TORSIONAL VIBRATIONS OF A TURBOGENERATOR

A typical shaft train of a large Turbogenerator in a Nuclear Power Plant with steam turbines (HPT, 3 LPT's), generator (GEN) and exciter (EXC) is shown in Fig. 4.

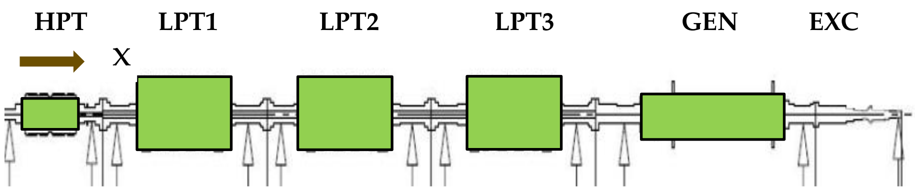


Figure 4 Turbogenerator of a Nuclear Power Plant with Steam Turbines and Generator

The linear equations of motion for the torsional vibrations of such a Turbogenerator are described by a model with global mass- damping- and stiffness-matrices **M**, **D**, **K** and an excitation vector $\mathbf{M}_{el}(t)$ for the air gap torques (Figure 4 and formula 3.1). The order of the matrices is NxN with N equal the number of torsional degrees of freedom.

$$\mathbf{M}\,\ddot{\mathbf{q}}(t) + \mathbf{D}\,\dot{\mathbf{q}}(t) + \mathbf{K}\,\mathbf{q}(t) = \mathbf{M}_{el}(t) \qquad (3.1)$$

The content of the matrices **M** and **K** has already been explained in chapter 3.1. The torsional damping of turbogenerators is in general very small. The sources of the torsional damping in matrix **D** are in the mechanical as well as in the electrical system. However, the most significant damping comes from the electrical parts of the generator and the grid, while the contributions from the mechanical turbine shaft (material damping, friction, steam,..) are very low. As will be explained later, the torsional damping is usually considered as modal damping, which means that each modal damping is related to one of the torsional natural frequencies (see chapter 4.1).

The linear equations 3.1 can be used for the calculation of the torsional natural frequencies and the corresponding torsional mode shapes. In this case the homogenous equations ($\mathbf{M}_{el}(t) = 0$) have to be solved, leading to an eigenvalue problem (chapter 4). For the vibration response due to electromechanical Air Gap excitations $\mathbf{M}_{el}(t)$ the complete equations (3.1) have to be solved (chapter 5).

4 Torsional Natural Frequencies, Mode Shapes and Modal Damping of Turbogenerators

During the design process of a Turbogenerator the calculation of the modal parameters: Torsional Natural Frequencies and the corresponding Mode Shapes is a very important standard task. The Theoretical Modeling by means of the FE-Method leads to very accurate results for the Natural Frequencies and the Mode Shapes. Such results are best suited to use them as reference values (baseline) for the Monitoring during the operation of a Turbogenerator.

4.1 HOMOGENEOUS EQUATIONS AND THE EIGENVALUE PROBLEM

In chapter 3 we have introduced the general linear equations of motion (3.1) for the torsional vibrations of a Turbogenerator. If the excitation torques $\mathbf{M}_{el}$ (t) in the equation system are set to zero, we obtain the homogenous equations of motion for the natural vibrations of the Turbogenerator. Due to the fact, that the damping is usually small, we can neglect the damping term, which has not much influence on the natural frequencies and the mode shapes. This leads finally to the reduced homogenous equations (3.2), in which only the inertia matrix **M** and the speed dependent stiffness matrix **K**(Ω) determine the natural frequencies and the mode shapes.

$$\mathbf{M}\,\ddot{\mathbf{q}}\,(t) + \mathbf{D}\,\dot{\mathbf{q}}\,(t) + \mathbf{K}(\Omega)\,\mathbf{q}\,(t) \;=\; \mathbf{0} \qquad (4.1)$$

$$\mathbf{M}\,\ddot{\mathbf{q}}\,(t) \;+\; \mathbf{K}(\Omega)\,\mathbf{q}\,(t) \;=\; \mathbf{0} \qquad (4.2)$$

With the mathematical approach (3.3) for the natural or free vibrations and the derivative (4.4) for the acceleration we obtain the Eigenvalue problem (4.5).

$$\mathbf{q}(t) \;=\; \boldsymbol{\varphi}\,\sin\omega\,t \qquad (4.3)$$

$$\ddot{\mathbf{q}}\mathbf{(t)} \;=\; -\,\omega^2\,\boldsymbol{\varphi}\,\sin\omega\,t \qquad (4.4)$$

$$(\mathbf{K}(\Omega) - \omega^2\cdot\mathbf{M}\,)\cdot\boldsymbol{\varphi} = \mathbf{0} \qquad (4.5)$$

The solution of this Eigenvalue problem leads finally to the torsional natural frequencies ω_n and the corresponding eigenvectors or mode shapes $\boldsymbol{\varphi}_n$ (n = 1,2,…N). The natural freqencies, expressed in Hz are $f_n = \omega_n / 2\,\pi$.

4.2 TORSIONAL NATURAL FREQUENCIES OF TURBOGENERATORS

Experience shows, that for a typical large Turbogenerator the lowest 20 torsional natural frequencies in the frequency range up to 120 Hz can be determined with high accuracy, expressed by a relative error of less than 2 %. Such accurate values are best suited, to use them as **Baseline values for the Monitoring** during the operation of the Turbogenerators. The following examples show the advantages to use the calculated torsional natural frequencies as **Baseline** during the operation:

Avoidance of Resonances: The designer determines the values of the natural frequencies in a way to avoid a match with the excitation frequencies of the Air Gap Torques. These are mainly the grid frequency and twice of the grid frequency. In a 50 Hz grid system the torsional natural frequencies should therefore not be in the frequency range between 45 to 55 Hz and 90 to 110 Hz (10 % safety margin). During operation the operator observes the respective natural frequencies in the frequency spectrum during operation , which are in the neighborhood of 50 Hz and 100 Hz.

Avoidance of the Sub Synchronous Resonance (SSR): In this specific case a resonance may occur in the frequency range below 50 Hz. The excitation frequency has its origin in the electrical natural frequencies of the grid system. If a Sub Synchronous Resonance (SSR) appears, one of the torsional natural frequencies may be excited in a resonance conditions with increasing torsional angle amplitudes. An increase of the torsional angle amplitude of one of the torsional natural frequencies below 50 Hz may therefore be a hint for an **SSR-event**.

Monitoring of Natural Frequency Changes: Any change of natural frequencies has to be checked by Monitoring in the Frequency Spectrum. Natural frequency changes can for example be a hint for stiffness changes in the shaft train: crack propagation, material fatigue, local plastic deformation, temperature influences (Youngs Modulus). In couplings a loosening of bolts, plastic deformations and a change of preload may happen. It is also possible, that the electromagnetic stiffness coupling with the grid changes. A trial has to be made to identify the cause of the natural frequency changes.

Low Cycle Fatigue in case of Transient Disturbances (see also chapter 5) **:** Any electromechanical transient disturbance of the grid starts with a step change for the shaft train. The step change response of the Turbogenerator consists of torsional natural vibrations with the natural frequencies of the shaft train. The amplitudes of this response decay slowly due to the low system damping. The stress amplitudes of this decaying vibration response are contributions to the low cycle fatigue (LCF) lifetime consumption.

4.3 TORSIONAL MODE SHAPES OF TURBOGENERATORS

Besides the torsional natural frequencies ω_n the Eigenvalue Analysis also leads to the corresponding mode shapes $\boldsymbol{\varphi}_n$. As example figure 5 shows the mode shapes of the three lowest natural frequencies of a turbogenerator of a large NPP.

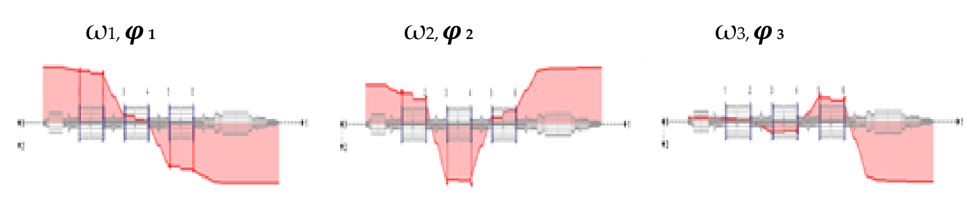


Figure 5 The Mode Shapes of the three lowest torsional Natural Frequencies

In these mode shapes the torsional deformations are mainly in the shaft train, while the blades follow the shafts angular motion as rigid bodies. At higher natural frequencies, e.g. from 40 to 120 Hz the blades form a dynamic rotor-blade interaction system with the shaft. In this frequency range the blades are included in the mode shapes with their own deformations relative to the shaft. Due to the fact, that the torsional vibration response of a Turbogenerator can always be expressed in terms of the mode shapes (see chapter 5), the major significance of the mode shapes becomes clear for the **Monitoring** besides the torsional natural frequencies. The following examples show the advantages to use the mode shapes for the Monitoring.

Which mode shapes can be excited: We assume, that the torsional vibrations are mainly excited by electromechanical interactions via the Air Gap torque $\mathbf{M}_{el}$ in the generator. In this case vibration energy in a vibration mode shape $\boldsymbol{\varphi}_n$ can only be transferred, when the modal participation factor $\boldsymbol{\varphi}_n^T \mathbf{M}_{el}$ (scalar vector product) is unequal zero for this mode. All three mode shapes in fig. 5 have nearly constant torsional angles in the generator part of the mode and can therefore be well excited by an air gap torque. If the air gap torque due to electromechanical interaction has an excitation frequency equal to one of the torsional natural frequencies below 50 Hz, the critical **Sub Synchronous Resonance (SSR)** case appears.

Mode Shapes show the deformation (torsional angles) along the shaft line in Resonances: In the resonance case for torsional vibrations in a Turbogenerator, the torsional deformation of the shaft train with the connected blades corresponds mainly to the mode shape of the resonance frequency $\Omega = \omega_n$. If in the monitoring process sensors for torsional angles are used, optimal sensor locations can be selected based on the known mode shapes.

Mode Shapes with the Stress distribution along the shaft line in Resonances : Besides the mode shapes for torsional angles also stress mode shapes can be derived from the theoretical model. An example for both types of mode shapes is shown in fig. 6. In the resonance case stresses of the shaft train with the connected blades correspond mainly to the stress mode shape of the resonance frequency $\Omega = \omega_n$. If in the monitoring process sensors for stresses are used, optimal sensor locations can also be selected based on the known stress mode shapes.

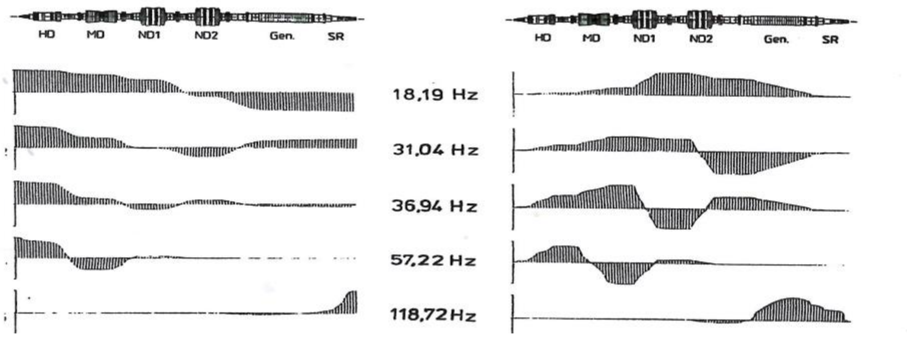


Figure 6 Torsional Angle- and Stress-Mode Shapes of a Turbogenerator

4.4 TORSIONAL MODAL DAMPING OF TURBOGENERATORS

The damping of the combined mechanical-electrical vibration system consists of two parts: the **mechanical damping** of the turbogenerator shaft train and the **electrical damping** from the generator and the grid. It is well known, that the mechanical damping (material, steam, blade joints) of the shaft train is small compared to the electrical damping. It has also to be considered, that the mechanical damping as well as the electrical damping depend on the load of the turbogenerator.

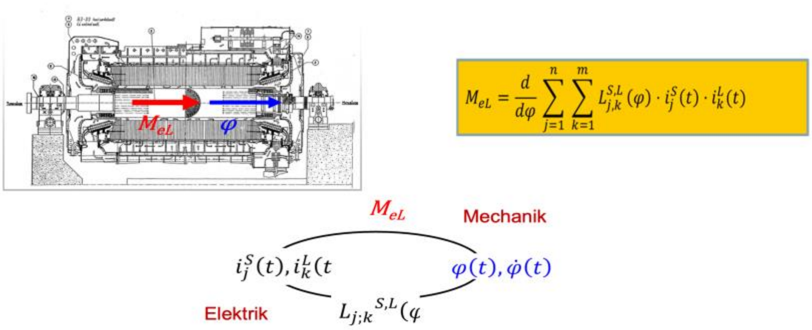
The torsional natural frequencies and the corresponding mode shapes can be calculated very accurate, if only the mass matrix **M** and the stiffness matrix **K** are considered The damping is not needed for this calculation, because the damping of turbogenerators is very small and has nearly no influence on the natural frequencies. However, damping becomes important when forced torsional vibrations of a turbogenerator due to air gap torque excitations have to be investigated. In case of resonances, particularly in the case of the **S**ub **S**ynchronous **R**esonance (**SSR**), positive damping of any size helps to stabilize the vibrations. Damping should therefore in any case be considered, when forced torsional vibrations are investigated. Due to this fact the torsional damping of a turbogenerator is usually determined for different operation conditions by testing procedures during the commissioning phase, e.g. by the Half Power Bandwidth method or the Logarithmic decrement method. It is a recognized practise to consider the damping as a modal damping D, by assigning the damping values to the different mode shapes. Unfortunately the torsional modal damping values of Turbogenerators are very small with the consequence, that free vibrations decay only slightly and that the resonance peaks of forced vibrations may become very high and sharp (see also chapter 5). From measurements at large Turbogenerators modal damping values are known to be in the range of D ~ 0,001 – 0,002. That means that very high amplifications 1/ 2D of 250 to 500 may occur in resonances [10].

5 Torsional Vibrations due to Electro-Mechanical Interaction

The most dominant excitation of Torsional Vibrations in Turbogenerators is caused by electromechanical interactions between the electrical system – Generator electrical part plus Grid - and the mechanical part of the Turbogenerator. Also present but not as significant is the excitation due to steam forces. In this chapter we are only considering the electromechanical excitation torques in the Air Gap of the Generator.

5.1 EXCITATION OF TORSIONAL VIBRATIONS DUE TO ELECTROMECHANICAL INTERACTIONS

In case of disturbances in the electrical generator-grid system besides the nominal Air Gap Torque additional torques due to electromechanical interactions will appear. They depend on electrical quantities (currents in the rotor- and stator-windings, coupling inductances) and on mechanical quantities (see in Figure 7 and in formular 5.1: the torsional angles and velocities and the electrical quantities).



$$M_{eL} = \frac{d}{d\varphi} \sum_{j=1}^{n} \sum_{k=1}^{m} L_{j,k}^{S,L}(\varphi) \cdot i_j^S(t) \cdot i_k^L(t) \qquad (5.1)$$

Figure 7 Air Gap Torque due to Electromechanical Interactions

The disturbance related air gap torques can be subdivided into the cases of transient disturbances and steady state disturbances. Examples for transient disturbances are 2-phase and 3-phase short circuits, synchronization out of phase, opening and closing of transmission lines, single-pole switching and the transient Sub Synchronous Resonance (SSR). Steady state disturbances are line and load unbalances (negative sequence current) and a possible steady state Sub Synchronous Resonance. In the table of figure 8 the different cases of transient and steady state disturbances are presented with the associated excitation frequencies, the table was taken from the standard ISO 22266 [7] . The transient disturbances always start with a very short but strong excitation (step change) with the specified excitation frequencies, followed by free torsional vibrations with the torsional natural frequencies only. Due to the low damping these free vibrations are weakly decaying.

Type of disturbance	Step change	Excite at grid frequency	Excite at twice grid frequency	Excite between 0,1 and 0,9 of grid frequency
Transient				
Three-phase fault	×	×	—	—
Unbalanced fault[a]	×	×	×	—
Synchronization out-of-phase	×	×	—	—
Opening of transmission line (three phases)	×	—	—	—
Closing of transmission line (three phases)	×	×	—	—
Single-pole switching	×	—	×	—
Transient 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 SSR	—	—	—	×

Figure 8 Table of Transient and Steady State Disturbances with Excitation Frequencies

With respect to the stress assessment this transient torsional vibrations belong to the group of Low Cycle Fatigue (LCF) vibrations with a relative low number of cycles, high stress and strain levels and possible plastic deformations. Torsional vibrations caused by steady state disturbances are characterized by steady forced vibrations with an excitation frequency, e.g. twice the grid frequency. They have a high number of long lasting cycles, lower stress levels and a more elastic behaviour in the shaft train and belong to the group of High Cycle Fatigue (HCF). The free vibrations are not considered in the steady disturbance case. For a safe and reliable operation of a Turbogenerator the transient as well as the steady state electromechanical interaction processes have to be considered. The resulting torsional vibrations of the shaft train should therefore be well investigated during the design process and later during the operation continuously be measured and monitored for a comprehensive assessment.

5.2 CONSIDERED DISTURBANCES IN THE DESIGN PROCESS

In the design process of large turbogenerators it is common practise to express the Air Gap Torques due to electromechanical disturbances by fixed formulas with specific frequencies instead of using the more complicated formula (5.1). This is a conservative approach compared to the real excitation cases. But experience shows that it is a useful approximation, which is often also confirmed by experimental results. Usually the electrical fault cases described in chapter 5.1 have the following excitation frequencies (see also the table in figure 8).

Transient disturbances:

2-phase short circuit	1x grid and 2x grid frequency
3-phase short circuit	1x grid frequency
Faulty synchronization	1x grid frequency
Closing of transmission lines	1x grid frequency
Single pole switching	2x grid frequency
Sub Synchronous Resonance	Sub synchronous frequency < 50 Hz

Steady state disturbances:

Line and/or Load Unbalance	2x grid frequency
Sub Synchronous Resonance	Sub synchronous frequency < 50 Hz

5.2.1 Transient Disturbance: A 2-phase Short Circuit

As an example for a transient electromechanical disturbance we consider the Air Gap Torque for the case of a 2-phase short circuit. The simplified formula for this case in figure 9 shows the electrical Air Gap Torque without a decay, which is a conservative assumption. The duration of this disturbance is very short, in the order of 50 to 150 msec. In reality the torque decays versus time. Excitation frequencies during this short time are the single grid frequency Ω and the double grid frequency 2 Ω. The formula contains also some generator characteristics.

For **simplification** the presented formula for the Electrical Air gap torque of a 2 Phase Short Circuit does not consider the strong **electromechanical Interaction** as previously shown.
It is however a good **approximation** based on experience for this kind of disturbance.

$$M_e(t) = M_0 + \frac{M_0}{\cos\varphi} \cdot \frac{1}{x''_d + x_{TR}} \cdot \{\sin \Omega(t - t_o) - 0.5 \cdot \sin 2\Omega(t - t_o)\} \quad (5.2)$$

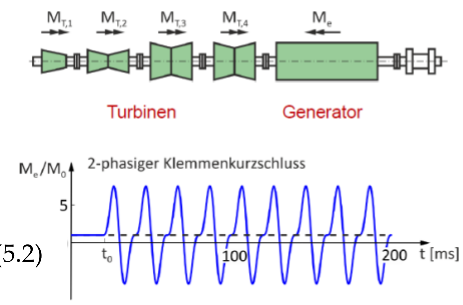


Figure 9 Simplified Air Gap Torque for a 2-Phase Short Circuit

5.2.2 Steady State Disturbance: Load Unbalance (Negative Sequence Current)

Due to unequal loads in the electrical grid an unsymmetry will occur in the 3-phase system. By electrical derivatives it can be shown, that due to this the nominal Air Gap Torque is superimposed by a pulsating torque with twice grid frequency. In a 50 Hz system the air gap torque therefore excites the shaft train continously with a frequency of 100 Hz, which is a steady state excitation (Figure 10).

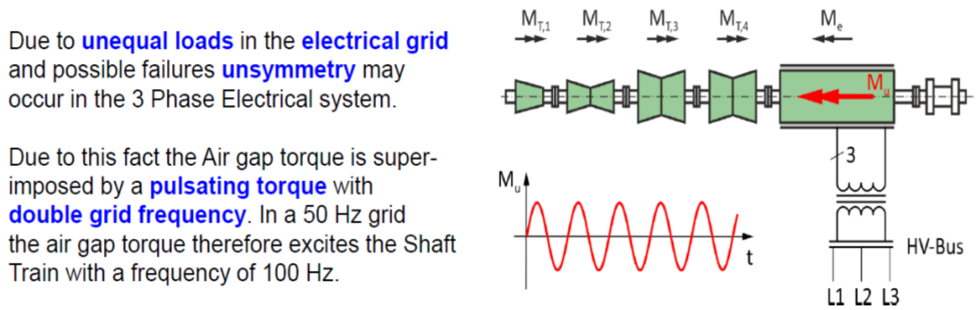


Figure 10 Steady State Air Gap Torque due to Load Unbalance

5.3 MODAL ANALYSIS FOR TORSIONAL VIBRATIONS

5.3.1 Introduction

Modal Analysis is a very efficient and powerful method to calculate free and forced vibrations of mechanical systems. In this subchapter we will demonstrate, how Modal Analysis can be used to determine Torsional Vibrations of large turbogenerators by calculations in case of electromechanical disturbances.
According to ISO 22266, we have classified these disturbances into transient and steady-state events, which differ fundamentally in their excitation characteristics and the resulting torsional vibration response.
Transient disturbances introduce a non-stationary excitation torque of short-duration. During these disturbances, the shaft system is first subjected for a short time to a forced response, however, once the excitation diminishes, the system predominantly exhibits free torsional vibrations at its natural frequencies. These oscillations are characterized by the superposition of the different torsional mode shapes, which decay over the time, depending on the modal damping values of the system.
In contrast, steady-state disturbances are characterized by continuous harmonic or periodic excitation torques. The torsional response is therefore mainly governed by forced vibrations at the excitation frequencies and their harmonics. If these excitation frequencies coincide with - or are close to - a torsional natural frequency, resonance effects may occur, leading to very high vibration amplitudes and stresses (see also chapter 4.4).
In practical systems, both mechanisms may coexist, and the total torsional response can be a superposition of forced and free vibration components. However, it is more likely, that the free vibrations mainly occur in case of transient disturbances after a short but strong initial disturbance, while the forced vibrations have mainly to be considered at steady state disturbances. In any case Modal Analysis is a well suited solution procedure for both cases, considering the vibration response in terms of modal expansion.

5.3.2 Decoupling of Equations of Motion by means of Modal Analysis

In this subchapter it will now be shown, how torsional vibrations can be calculated by means of Modal Analysis. We start with the coupled equations of motion (5.3) for the turbogenerator, which have already been described in chapter 3.2.

$$\mathbf{M}\,\ddot{\mathbf{q}}\,(t) + \mathbf{D}\,\dot{\mathbf{q}}\,(t) + \mathbf{K}\,\mathbf{q}\,(t) = \mathbf{M}_{el}\,(t) \tag{5.3}$$

The Modal Analysis procedure decouples the **M**ulti **D**egree **O**f **F**reedom (MDOF) equation system (5.3) of order N (N number of degrees of freedom) into N **S**ingle **D**egree **O**f **F**reedom (SDOF) systems, each one consisting of a modal mass, a modal stiffness, a modal damping and a modal excitation force. This decoupling of the equations of motion can be achieved by a development of the torsional displacement vector **q**(t) in terms of the eigenvectors $\boldsymbol{\varphi}_n$ of the system without damping (see chapter 4.1 and equation (5.4)). In equation (5.4) each eigenvector is multiplied with the so called time dependent modal coordinate p_n (t). This modal coordinate expresses how strong an eigenvector $\boldsymbol{\varphi}_n$ contributes to the overall solution **q**(t)

$$\mathbf{q}(t) = \sum_{n=1}^{N} p_n(t)\,\boldsymbol{\varphi}_n \tag{5.4}$$

We introduce equation (5.4) into equation (5.3), multiply from the left side with an transposed eigenvector $\boldsymbol{\varphi}_k^T$ and obtain the following equation (5.5)

$$\sum_{n=1}^{N}\left\{\boldsymbol{\varphi}_k^T\mathbf{M}\boldsymbol{\varphi}_n\,\ddot{p}_n(t) + \boldsymbol{\varphi}_k^T\mathbf{D}\boldsymbol{\varphi}_n\dot{p}_n(t) + \boldsymbol{\varphi}_k^T\mathbf{K}\boldsymbol{\varphi}_n\,p_n(t)\right\} = \boldsymbol{\varphi}_k^T\,\mathbf{M}_{\mathbf{el}}(t) \tag{5.5}$$

In the theory of vibrations it is shown, that the following eigenvector products fulfill the orthogonality characteristics

$$\boldsymbol{\varphi}_k^T\mathbf{M}\boldsymbol{\varphi}_n = m_k \text{ for } k = n \quad \text{and} \quad \boldsymbol{\varphi}_k^T\mathbf{M}\boldsymbol{\varphi}_n = 0 \text{ for } k \# n \tag{5.6}$$

$$\boldsymbol{\varphi}_k^T\mathbf{K}\boldsymbol{\varphi}_n = k_k \text{ for } k = n \quad \text{and} \quad \boldsymbol{\varphi}_k^T\mathbf{K}\boldsymbol{\varphi}_n = 0 \text{ for } k \# n \tag{5.7}$$

where m_k is the so called modal mass and k_k is the modal stiffness, both of them belong to the natural torsional eigenfrequency f_k or ω_k (subscript k). In practise the assumption is often made, that the damping matrix **D** is proportional to **K** and **M** (Rayleigh-Damping)**.** In this case the orthogonality relation is also valid for the damping

$$\boldsymbol{\varphi}_k^T \mathbf{D} \boldsymbol{\varphi}_n = d_k \text{ for } k = n \quad \text{and} \quad \boldsymbol{\varphi}_k^T \mathbf{D} \boldsymbol{\varphi}_n = 0 \text{ for } k \neq n \qquad (5.8)$$

With the introduced quantities m_k , d_k , k_k we now obtain N decoupled equations of motion (5.9), which belong to the different eigenvectors $\boldsymbol{\varphi}_k$ with its corresponding natural frequencies ω_k.

Each of the N equations describes the dynamic equilibrium for one of the **S**ingle **D**egree **o**f **F**reedom (SDOF) systems with the modal coordinate p_k (t), the modal mass m_k , the modal stiffness k_k and the damping d_k

$$m_k \ddot{p}_k(t) + d_k \dot{p}_k(t) + k_k p_k(t) = \boldsymbol{\varphi}_k^T \mathbf{M}_{el}(t) \quad (k = 1, 2, 3,\ldots.N) \quad (5.9)$$

On the right hand side of each equation the scalar vector product $\boldsymbol{\varphi}_k^T \mathbf{M}_{el}(t)$ represents the time dependent modal excitation . By means of this vector product it can be found out, how strong the torsional vibrations can be excited in the mode shape $\boldsymbol{\varphi}_k$. This is especially important for the case of a resonance, when an excitation frequency Ω is equal to the torsional natural frequency ω_k . In a resonance condition the other important factor is the damping, for which we can introduce the modal damping D_k. The relation between the modal damping D_k and the damping d_k from equation 5.8 is

$$D_k = d_k / (2\, m_k\, \omega_k) = d_k / 2\, (\sqrt{k_k\, m_k}) \qquad (5.10)$$

When the SDOF equations (5.9) have been solved, the results p_k (t) can be introduced in equation 5.4 and the solution for the overall vector $\mathbf{q}$(t) with the torsional displacements can be calculated by superposition of all eigenvectors $\boldsymbol{\varphi}_k$.

5.3.3 Torsional Vibrations of the SDOF System

The general torsional vibration response $\mathbf{q}$(t) of the Turbogenerator consists of two solution parts:

General Torsional Vibrations = Homogeneous Solution + Particular Solution

$$\mathbf{q}(t) = \sum_{k=1}^{N} p_k(t)\, \boldsymbol{\varphi}_k = \mathbf{q}_{hom}(t) + \mathbf{q}_{part}(t) \qquad (5.11)$$

The homogeneous solution represents the free vibrations and the particular solution the forced vibrations due to the air gap torques $\mathbf{M}_{el}(t)$. We can built up the general solution (5.11) step by step, starting with the simple SDOF-equation (5.9), for which the general solution $p_k(t)$ is well known from basic vibration theory. The general solution of 5.9 is presented by formula (5.12) (derivation is not shown here).

$$p_k(t) = p_{k\,hom}(t) + p_{k\,part}(t) = e^{-D_k \omega_k t} \{A_c \cos \omega_k t + A_s \sin \omega_k t\} + p_{k\,part}(t) \quad (5.12)$$

The general solution $p_k(t)$ of the SDOF system also consists of the two parts: the homogeneous solution (free vibrations) and the particular solution (forced vibrations). Both solutions are presented in the two following subchapters.

5.3.4 Homogeneous Solution of the SDOF System

The homogeneous solution of the SDOF System with natural frequency ω_k consists of a decaying exponential function , which is multiplied with harmonic functions with the natural frequency ω_k (equ. 5.13). Strictly speaking there is an influence of the damping on the natural frequency ω_k , but this can be neglected here because of the small damping values.

$$p_{k\,hom}(t) = e^{-D_k \omega_k t} \{A_c \cos \omega_k t + A_s \sin \omega_k t\} \qquad (5.13)$$

An example of the time function of the homogeneous solution (free vibrations) is shown in figure 11. The torsional natural frequency ω_k is known from the eigenvalue problem and the modal damping D_k should be known from measurements in the commissioning phase. Furthermore the parameters A_c and A_s have to be determined from the initial conditions for $\mathbf{q}$ (t = 0) = $\mathbf{q}_0$ and $\dot{\mathbf{q}}$ (t = 0) = $\dot{\mathbf{q}}_0$ **.** This determination will be not further considered here but has to be completed in practical applications.

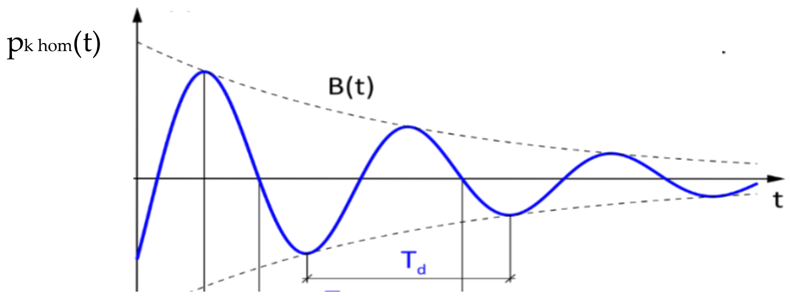


Figure 11 Free Vibrations of the SDOF System with Natural Frequency ω_k

Due to the fact that the modal damping values D_k are very small, the free vibrations p_k decay only slightly. That means that the free torsional vibrations of the Turbogenerator have a relative long decay time after a transient disturbance (see table in figure 8).

5.3.5 Particular Solution of the SDOF System

The particular solution $p_{k\,part}$ (t) can be solved analytically or for arbitrary scalar vector products $\boldsymbol{\varphi}_k^T$ $\mathbf{M}_{el}$ (t) (modal excitations) numerically with suitable algorithms. From the table in Figure 8 (ISO 22266-1) we know, that the disturbances from the electrical grid can be considered as harmonic or periodic Air Gap torques. For this cases the excitation frequencies are the grid frequency, twice the grid frequency or sub synchronous frequencies from the grid. Nearly all disturbance cases have only one excitation frequency. The basis for the particular solution $p_{k\,part}(t)$ of most of the cases is therefore the solution for a harmonic Air Gap excitation with one exciter frequency Ω, which can be described by the excitation (5.13)

$$M_{el}(t) = \hat{M}_{el} \sin(\Omega t) \qquad (5.13)$$

If we introduce this excitation into the equation (5.9) of the SDOF-System, we obtain the particular solution in the following form (Derivation not shown here) with the definition of $\eta_k = \Omega / \omega_k$ and ψ_k as phase angle between the response and the excitation:

$$p_{k\,part}(t) = \underbrace{\{ \boldsymbol{\varphi}_k^T \hat{\mathbf{M}}_{el} / m_k \omega_k^2 \}}_{\text{Modal Participation Factor}} \underbrace{\{ 1/\sqrt{(1-\eta_k{}^2)^2 + (2 D_k \eta_k)^2} \}}_{\text{Transfer Function } V(\eta_k, D_k)} \underbrace{\sin(\Omega t - \psi_k)}_{\text{Time Function } f(t)} \qquad (5.14)$$

The particular solution consists of three factors:

- the **Modal Participation Factor** contains the vector scalar product $\boldsymbol{\varphi}_k^T \hat{\mathbf{M}}_{el}$. The factor determines how excitable the mode shape $\boldsymbol{\varphi}_k$ is. If we assume, that $\hat{\mathbf{M}}_{el}$ is constant along the generator air gap, then all mode shapes $\boldsymbol{\varphi}_k$ with also constant amplitudes in the generator part can well be excited. Considering the three mode shapes in Figure 5 they are all well excitable.
- The **Transfer Function** depends on the frequency ratio $\eta_k = \Omega / \omega_k$ and on the modal damping D_k. The function is shown in Figure 12. Depending on the relation of the excitation frequency Ω and the torsional natural frequency ω_k the amplification V and the phase angle ψ can be determined.

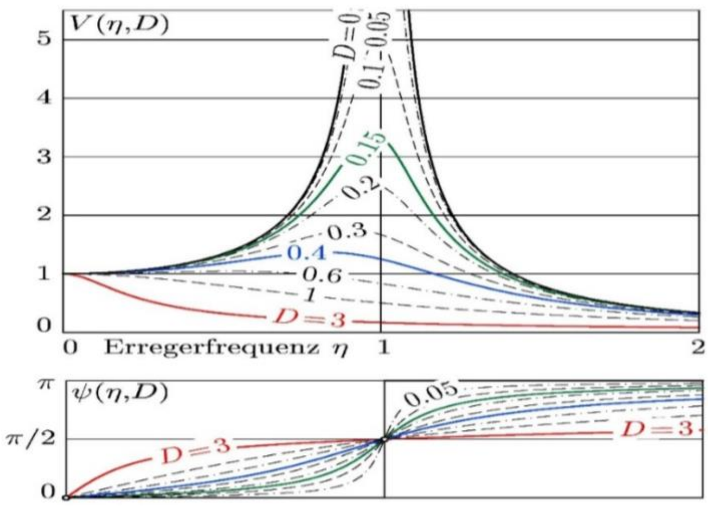


Figure 12 Transfer Function $V(\eta_k, D_k)$ dependent on Frequency Ratio and Modal Damping

- The most critical case appears in case of a resonance for $\Omega = \omega_k$ or $\eta_k = 1$, then the amplitudes of V reach the maximum. As can be seen from (5.14) the maximum in the resonance case is $V_{max} = 1/\ 2\ D_k$. Realistic modal damping values for Turbogenerators are in the range of $D_k \sim 0{,}001 – 0{,}002$ (0,1% – 0,2 %), which means that the amplification in a resonance may become very high: $V_{max} \sim 250 – 500$.
- The **Time Function** only shows, that the torsional vibrations appear as a harmonic function with frequency Ω and have a phase difference of ψ in relation to the excitation.

The most important statement for the forced torsional vibrations in Turbogenerators should be: **Avoid Resonance Cases: $\Omega = \omega_k$** because of the very low damping.

5.3.6 General Torsional Vibrations by Superposition of all Modes

Based on the results $p_k(t)$ we can now come back to equation (5.11) and can express the general torsional vibrations by superposition of all modes $\boldsymbol{\varphi}_k$. This operation includes the multiplication of each component $p_k(t)$ with the eigenvector $\boldsymbol{\varphi}_k$ and the superposition of all products $p_k(t)\ \boldsymbol{\varphi}_k$.

$$\mathbf{q}(t) = \sum_{k=1}^{N} p_k(t)\ \boldsymbol{\varphi}_k = \mathbf{q}_{hom}(t) + \mathbf{q}_{part}(t) = \sum_{k=1}^{N} (p_{k\,hom}(t)\ \boldsymbol{\varphi}_k + p_{k\,part}(t)\boldsymbol{\varphi}_k) \quad (5.15)$$

with

$$\mathbf{q}_{hom}(t) = \boldsymbol{\Sigma}_k \left[e^{-D_k \omega_k t} \{A_c \cos \omega_k t + A_s \sin \omega_k t\} \right] \boldsymbol{\varphi}_k \qquad (5.15a)$$

and (5.15b)

$$\mathbf{q}_{part}(t) = \boldsymbol{\Sigma}_k \left[\{ \boldsymbol{\varphi}_k^T \hat{\mathbf{M}}_{el} / m_k \omega_k^2 \} \{ 1/ \sqrt{(1-\eta_k{}^2)^2 + (2 D_k \eta_k)^2} \} \sin(\Omega t - \psi_k) \right] \boldsymbol{\varphi}_k$$

With this step we have now introduced additional information about the distribution of the torsional vibrations along the shaft train axis due to the eigenvectors $\boldsymbol{\varphi}_k$. If one wants to know especially how the vibrations are at predefined sensor locations this information can be obtained by selecting only the corresponding component $q_m(t)$ out of eigenvector $\mathbf{q}(t)$ (e.g. m = sensor location).

Depending on the disturbance case the equations (5.15), (5.15a) and (5.15b) can be used as a combined solution, including the homogeneous and the particular solution together or in a simplified version with only one solution part. For example in the case of a two phase short circuit the disturbance starts with a very short but strong Air Gap torque with the two frequency components of grid frequency and twice grid frequency. During that short time especially the particular solution is important. After the short time disturbance the Air Gap torque disappears. However, the free vibrations with a slow decay rate are still active and have to be considered by the homogeneous solution alone. On the other hand in case of a steady state disturbance with a harmonic excitation function (Line or load unbalance) the particular solution can be used alone. An advantage of the Modal Analysis solution is, that in practical cases usually not all of the eigenvectors are needed. This helps to perform the numerical investigation more efficient.

5.3.7 Torsional Moments and Shear Stresses in the Shaft Train

In chapter 6 it will be described, how torsional vibrations can be recorded during operation. Depending on the sensor type either angular displacements q(t) or torsional shear stresses τ(t) at the boundary of a circular shaft cross section can be measured. The following relationship holds between the torsional moment M_T (x, t) in a cylindrical cross section of the shaft and the angular displacement q(x, t), where x is the axial coordinate of the shaft and q′ (x,t) – the torsional strain -is the derivation of q(x,t) with respect to the axial coordinate:

$$M_T = G\, I_T\, dq/dx = G\, I_T\, q'(x,t) \qquad (5.16)$$

G is the shear modulus of the material (G = 80 GPa for steel) and $I_T = \pi D^4 / 32$ is the torsional moment of inertia with the diameter of the cross section D.

Due to the torsional moment M_T the maximum torsional shear stresses in the cylindrical cross sector occurs at the outer boundary. They can be determined by the formular:

$$\tau_{max} = \alpha_k\, (M_T / W_T) = \alpha_k\, (G\, I_T\, q'(x,t) / W_T) = \alpha_k\, D/2\, (G\, q'(x,t)) \qquad (5.17)$$

with $W_T = \pi D^3 / 16$ and the notch factor α_k.

By means of equation (5.17) we recognize the relationship between the maximum shear stress τ_{max} and the derivation of the angular displacements q(x,t) with respect to the axial coordinate.

5.4 TORSIONAL VIBRATIONS DUE TO A TRANSIENT DISTURBANCE

5.4.1 Excitation due to a 2-Phase Short Circuit

For the torsional vibration response in case of a transient disturbance we show as example a 2-phase short circuit in a 600 MW Turbogenerator from [12]. It has already been mentioned in 5.3.6 that in this case the disturbance starts with a very short but strong Air Gap torque with the two frequency components of grid frequency and twice grid frequency. During that short time especially the particular solution is of importance.

After the short time disturbance the Air Gap torque disappears. However, the free vibrations with a slow decay rate are still active and have to be considered by the homogeneous solution alone. However, in this example we will only consider the strong short time disturbance (duration ~100 – 200 msec) with the superposition of the two harmonic functions with grid frequency and twice grid frequency (see figure 13)

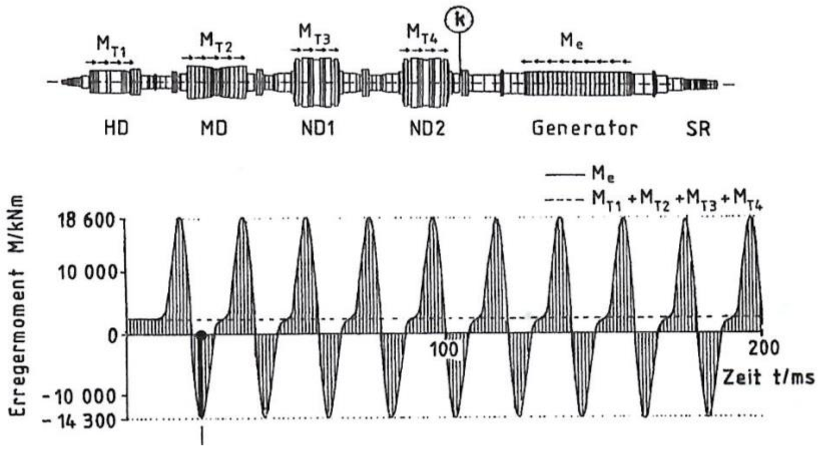
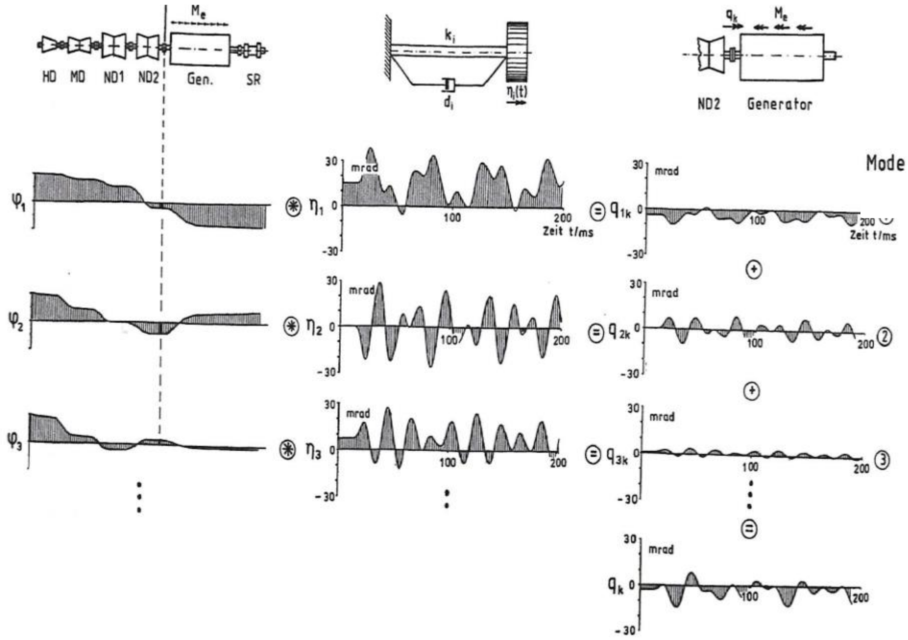


Figure 13 Turbogenerator Excitation by an Air Gap Torque of a 2-Phase Short Circuit

5.4.2 Solution of Torsional Vibrations by means of Modal Analysis

Figure 14 shows the contributions of the first modes of the Turbogenerator to the angular displacements at the location of the coupling between the Low Pressure Turbine ND2 and the Generator. This is the most loaded location. Figure 14 represents exactly the determination of the particular torsional vibration related to equation (5.15b). It shows the mode shapes on the left side, the SDOF system in the center of Figure 14 and the result of the particular solution of the SDOF system for the location of the mentioned coupling due to the 2 phase short circuit excitation with the two frequencies. From the shown three modes only the two lower ones contribute a considerable amount to the angular displacements. One reason for this is that the third mode shape has only small angular displacements in the generator region. The contribution of the third mode to the torsional vibration in the coupling is therefore insignificant.



Eigenvectors φ_k Torsional Mode Shapes | Modal coordinates $p_k(t)$ for SDOF system | Torsional Angles at Generator Coupling

Figure 14 Modal Analysis: Expansion of torsional angles in terms of eigenvectors $\boldsymbol{\varphi}_k$

5.4.3 Moments at the Coupling Locations

Figure 15 shows the moment at the coupling between the Low Pressure Turbine and the Generator (location k). Its highest amplitude is four times of the static moment. It is also the maximum moment in the complete Turbogenerator train. The contributions of the mode shapes to the maximum coupling moments are shown in Figure 16. On the left side we see besides the 2-phase excitation torque how the individual modes superimpose to the ND2-Generator coupling torque. On the right hand side the contributions of the different mode shapes to the different couplings of the Turbogenerator system can be seen. The maximum moments can again be identified at the ND2-Generator coupling. The lowest three mode shapes have a determining influence on the maximum values of the couplings. The example shows that Modal Analysis can help to better understand the torsional vibration behavior of the Turbogenerator

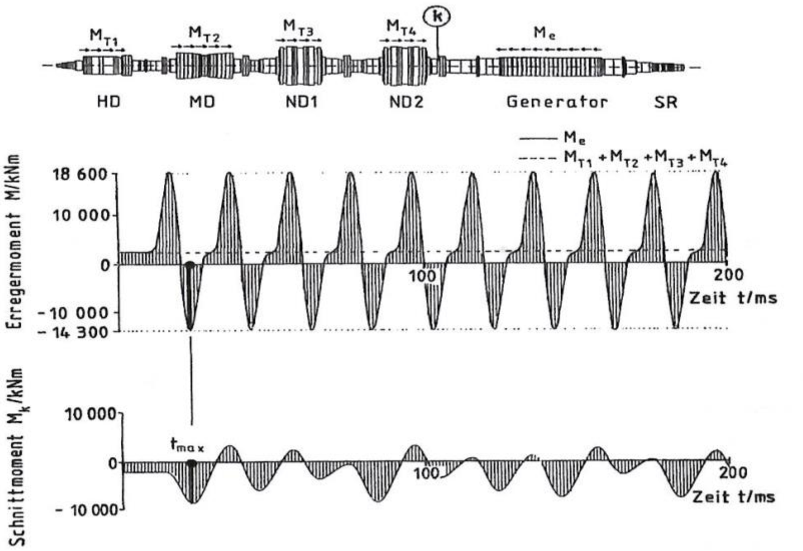


Figure 15 Excitation Air Gap Torque and Coupling Torque at Generator Coupling

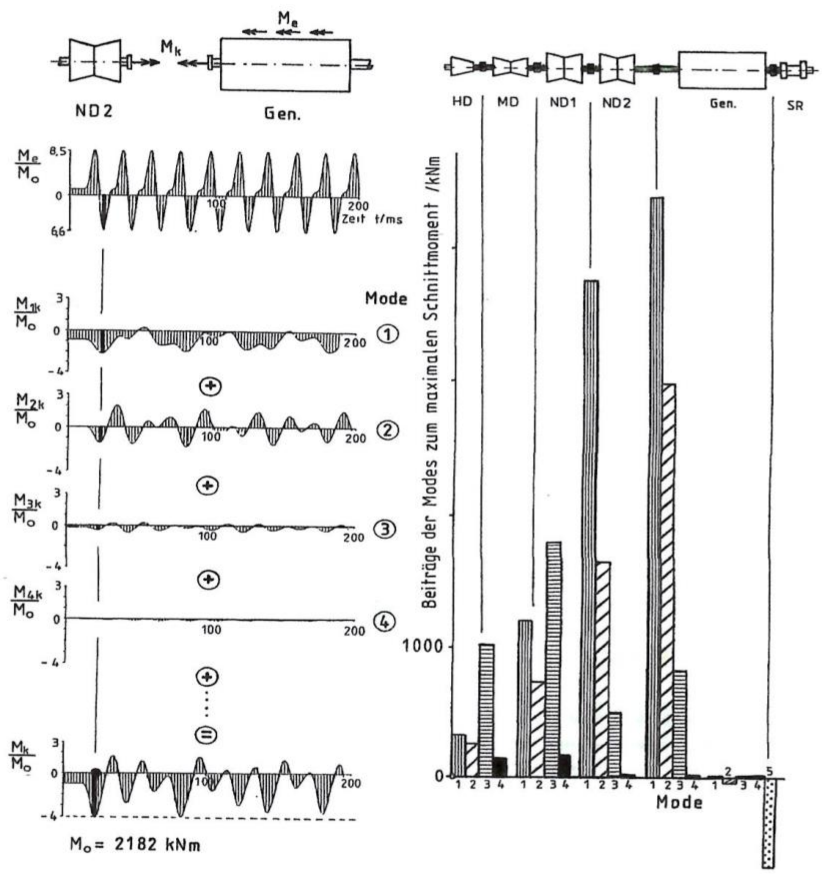


Figure 16 Contribution of the Mode Shapes φ_k to the maximum Coupling Moments

5.5 TORSIONAL VIBRATIONS DUE TO A STEADY STATE DISTURBANCE

5.5.1 Excitation with 100 Hz due to a Steady State Load Unbalance

As an example for a steady state disturbance we consider the case of a load unbalance in the electrical grid, which leads to an unsymmetry in the 3-phase system. It was already shown in chapter 5.2.2, that this disturbance leads to a harmonic air gap torque with an excitation frequency of twice grid frequency.

$$\mathbf{M}_{el}(t) = \hat{M}_{el} \sin(\Omega t) \quad \text{with } \Omega = 2\pi f \text{ and } f = 100 \text{ Hz} \tag{5.18}$$

Figure 17 shows once more this harmonic excitation type with the frequency of 100 Hz.

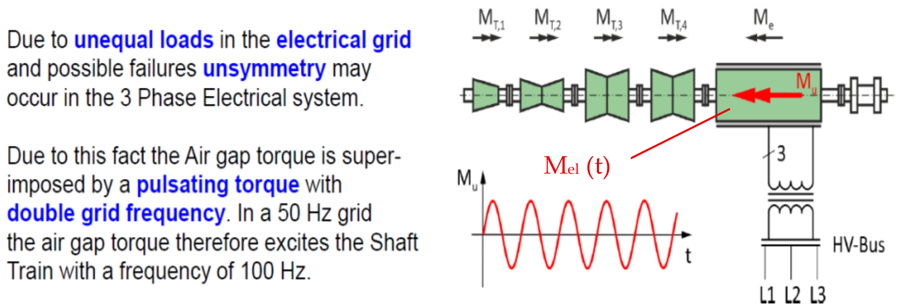


Figure 17 Steady State Air Gap Torque with 100 Hz due to Load Unbalance

5.5.2 Solution of Torsional Vibrations by means of Modal Analysis

For the determination of the torsional vibrations we again use the Modal Analysis procedure. Due to the fact that the excitation is a steady state type we can use the particular solution only, which has been derived in chapter 5.

$$\mathbf{q}_{part}(t) = \Sigma_k \left[\{ \boldsymbol{\varphi}_k^T \hat{\mathbf{M}}_{el} / m_k \omega_k^2 \} \{ 1/\sqrt{(1-\eta_k^2)^2 + (2 D_k \eta_k)^2} \} \sin(\Omega t - \psi_k) \right] \boldsymbol{\varphi}_k \tag{5.19}$$

The solution represents the superposition of the different mode shapes across the entire frequency range. The different mode shapes contribute more or less to the overall solution, which is determined by the transfer functions $V(\eta_k, D_k)$ and by the modal participation factors $\boldsymbol{\varphi}_k^T \hat{\mathbf{M}}_{el}$. The transfer functions $V(\eta_k, D_k)$ contribute with high values to $\mathbf{q}_{part}$, when the torsional natural frequencies ω_k are close to the excitation frequency of 100 Hz**.** This case with values of η_k close to 1 has to be avoided (see also figure 12). The second influencing factor, the modal participation factor $\boldsymbol{\varphi}_k^T \hat{\mathbf{M}}_{el}$ determines, how strong the excitation is influenced from the mode shapes $\boldsymbol{\varphi}_k$.

5.5.3 Safety Margin: Avoidance of Resonance Conditions

To avoid, that torsional natural frequencies ω_k come to close to the 100 Hz excitation frequency (Resonance condition), the machine designer has to take into account a safety margin of about 10 % around the 100 Hz excitation (see ISO 22266). That means the frequency range between about 90 Hz and 110 Hz should be free of torsional natural frequencies. In this higher frequency range the mode shapes of the natural frequencies are usually characterized by a coupling of torsional shaft vibrations and blade vibrations. It has also to be considered, that the natural frequencies can depend on the rotational speed of the Turbogenerator. Figure 18 shows an example for the safety margin in case of a Turbogenerator, operated with a rotational speed of 1500 rpm (25 Hz). At the operational frequency 25 Hz the safety margins between the twice grid frequency 100 Hz and the torsional natural frequencies f_n and f_{n+1} are shown by the red arrows.

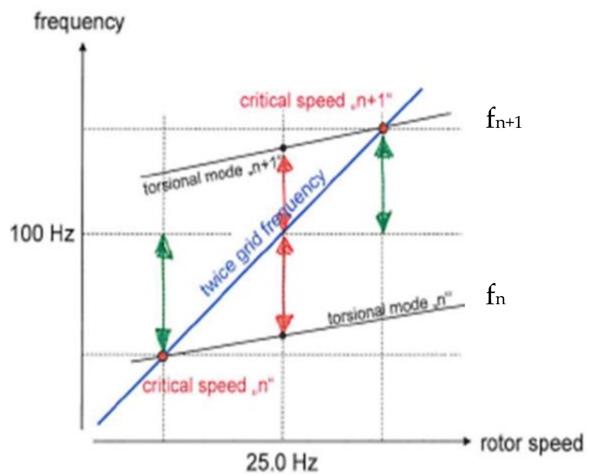


Figure 18 Safety margin or avoidance zone for Torsional Vibrations ISO 22266-1

5.5.4 A Steady State Sub Synchronous Resonance

In the table of Figure 8 the Sub Synchronous Resonance is also considered as one of a possible Steady State disturbance. This situation can occur, when the difference between the synchronous grid frequency Ω_S and one of the electrical grid natural frequencies ω_{el} is equal to one of the torsional natural frequencies ω_k of the mechanical Turbogenerator

$$\text{Excitation Frequency:} \qquad \Omega = (\Omega_S - \omega_{el}) = \omega_k \qquad (5.20)$$

If this resonance condition with $\eta_k = 1$ ($\Omega = \omega_k$) occurs, the particular solution for the torsional response of the Turbogenerator has the simplified form

$$\mathbf{q}_{part}(t) = \{ \boldsymbol{\varphi}_k^T \hat{\mathbf{M}}_{el} / m_k \omega_k^2 \} \{ (1/2 D_k) \} \sin(\Omega t - \psi_k)] \ \boldsymbol{\varphi}_k \qquad (5.21)$$

Considering for example the three mode shapes of Figure 5, all of them have torsional natural frequencies below the synchronous frequency, that means they are sub synchronous. Furthermore due to their mode shapes all of them are very well excitable $\boldsymbol{\varphi}_k^T \hat{\mathbf{M}}_{el}$. And due to the low damping value D_k the angular amplitudes in the resonances may become very large (see also Figure 12 for small damping D_k). From formular (5.21) it can also be seen, that the vibration mode shape $\mathbf{q}_{part}(t)$ in the sub synchronous resonance follows the mode shape $\boldsymbol{\varphi}_k$. If the combined electrical and mechanical damping should become zero the mathematical solution for $\mathbf{q}_{part}(t)$ will become a more and more increasing time function , as it is shown in Figure 19.

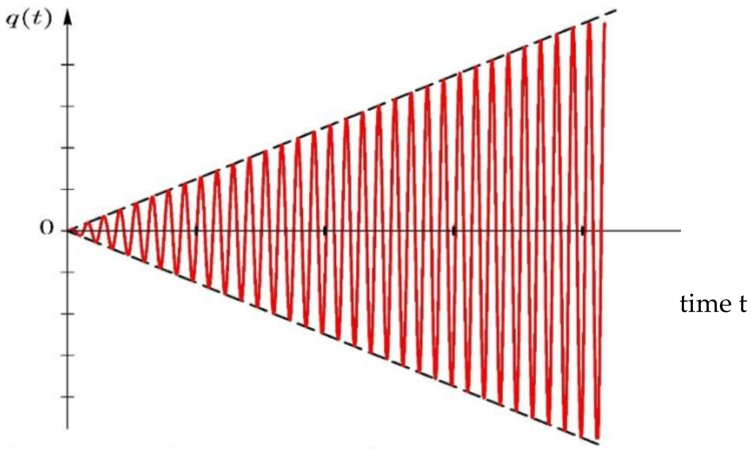


Figure 19 Increasing Angular Displacements in case of damping D_k = 0

Such a vibration behavior has been observed in a large Turbogenerator during an SSR event in 2024.

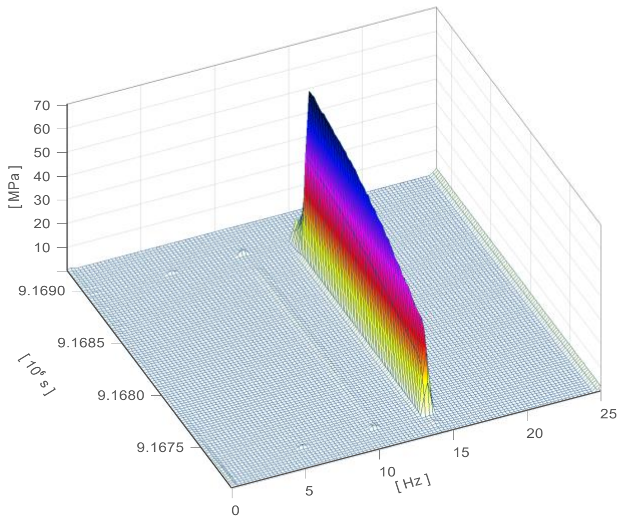


Figure 20 Increasing Shear Stresses versus time in an SSR Event

6 Existing and New Sensor-Technologies for Torsional Vibrations

In the previous chapters it was shown, that disturbances in the electrical grid system with resulting electromechanical interactions in the air gap of the generator can lead to strong torsional vibrations of the shaft train. To control problems of this kind the task of manufacturers is to design turbogenerators with an optimal torsional vibration behavior. On the other hand the plant operator should carefully observe the mechanical stresses at critical locations of the shaft train and the connected blades during the operation. For this task sensors are used to measure and the torsional vibrations. This chapter describes the different methods to measure torsional vibrations. Existing and new Sensor-Technologies will be presented. Advantages and disadvantages of the different Sensors will be discussed.

6.1 DIRECT MEASUREMNT METHODS

Direct Measurement methods work with sensors, which are directly mounted on the rotating shaft. We consider the following cases of accelerometers, strain gauges and interesting combinations of both. Furthermore we study the function of Dual-Beam Laser Laser Interferometer.

6.1.1 Direct Measurement Method with Linear Accelerometers

Function: Two linear accelerometers are fixed in parallel on the rotating shaft. They can measure the tangential accelerations. They move in the opposite direction in the fixed system of the rotation axis. Any translational acceleration of the shaft is cancelled by taking the average of both accelerometers (Figure 21)

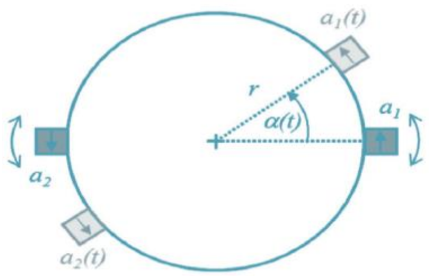


Figure 21 Two linear Accelerometers to measure tangential accelerations

- **Advantages:**
  High Dynamic Range, Low Sensitivity to Translational Vibrations

- **Disadvantages:**
  Expensive Telemetry System or Slip Rings needed, High Centrifugal Forces; No Absolute angular position available

6.1.2 Direct Measurement Method with Strain Gauges

Function: The Strain Gauges are glued on the Shaft to measure the torsional elongation or shear stress. Torsional Vibrations are mostly measured in Half Bridge configuration with two Strain Gauges positioned at a 45 degree angle on the Shaft. With a Full Bridge configuration a compensation of Bending Stress is possible.

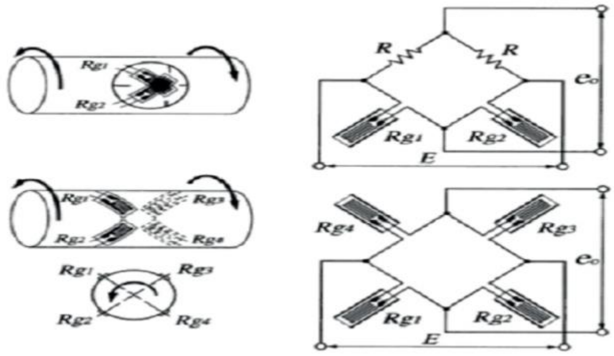


Figure 22 Strain Gauges to measure Torsional Elongation or Shear Stress

- **Advantages:**
  Direct Measurement of Shear Stresses, Low Sensitivity to other Deformations or Vibrations

- **Disadvantages:**
  Expensive Telemetry System or sensitive Slip Rings needed, High Centrifugal Forces may lead to loss of sensors; Exact Angular Speed and position not known.

6.1.3 Direct Measurement Method: Combined Angular Accelerometer and Strain Gauge (EPRI-Subrock)

The EPRI-Subrock system is one of the modern concepts for the continuous torsional vibrations measurement and monitoring of large Turbogenerators [3]. In Figure 23 the overall concept including Measurement and Monitoring is explained. It consists of the three parts: the rotating sensor unit with two different sensors, the wireless transmission of data by means of the Radio Frequency (RF) telemetry and the stationary receiver station as the Monitoring and Analysis part. In this chapter we mainly consider the rotating Sensor unit and the wireless transmission part.

Function: The EPRI-Subrock Sensor system works with two different sensor types at the same time (see Figure 23, the rotating sensor part):

- Semiconductor Strain Gauges to measure the torsional strain q' (x,t) (see also equ. (5.16) in chapter 5.3.7.
- MEMS Acceleration Sensors to measure the tangential acceleration, which is proportional to the angular acceleration q̈ (x,t).
-

The reason for the synchronous capturing of signals from the two different sensor types has been justified by a better identification of modes and an improvement of diagnosis tasks. For comparison classical systems use one sensor type only.

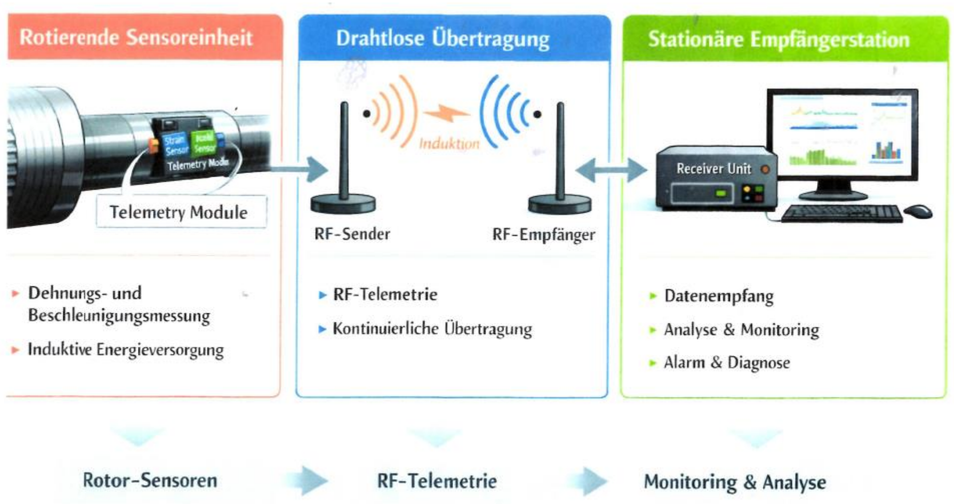


Figure 23 The EPRI-Subrock Measurement and Monitoring System

The small sensor types are glued directly on the shaft surface and fixed by means of Epoxid+Kevlar/Karbon-Fabric (see Figure 24). Due to the small sensor masses mounting of the sensors are even possible in narrow regions. Furthermore the distances between sensors can be very small, which can be an advantage for the assessment.

Figure 24 Mounting and Fixing of the Sensors on the Shaft Surface

The Energy supply is wireless (no battery), with an inductive supply for the rotating sensors. The power requirement is very low. A continuous operation is possible.

The measured data are continuously transferred via Radio Frequency from the rotating sensor telemetry module to the stationary receiver unit. As Figure 23 shows the data transfer occurs between the rotating transceiver and the stationary receiver, where the incoming data are processed in a central computer.

- **Advantages:**
  Two different small sensor types measure at the same time torsional strain and angular accelerations. Improved Mode Identification possible, Continuous longtime operation possible, Data Transmission by means of Radio Frequency (RF) telemetry, Wireless energy supply
- **Disadvantages:**
  Elaborate installation of sensors, Very high costs compared other solutions, Long-term reliability educed in harsh environment and at high centrifugal loads, Signal transmission may be disturbed due to electromagnetic disturbances, Maintenance may become expensive, it needs specialist

6.1.4 Touchless Dual-Beam Laser Interferometer

A laser vibrometer is a measuring device for quantifying mechanical vibrations. It can be used to measure vibration frequency and vibration amplitude. Vibrometers contain a laser that is focused on the surface to be measured. Due to the Doppler effect, when the surface to be measured moves, the frequency of the backscattered laser light shifts. This frequency shift is evaluated in the vibrometer using an interferometer and output as a voltage signal or digital data stream. A scanning vibrometer allows a spatial measurement of vibrations.

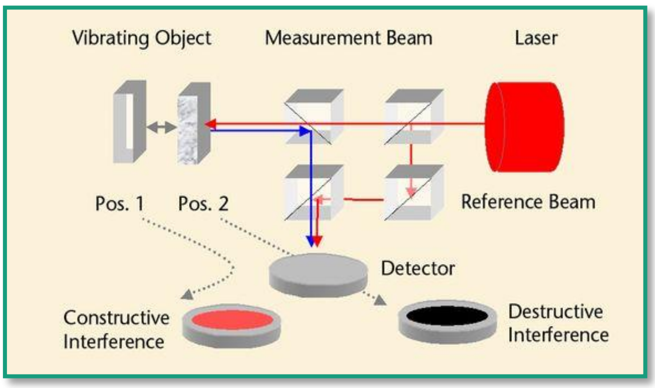


Figure 25 The Laser Doppler Vibrometer for the Measurement of Lateral Vibrations

The basic idea of the presented Laser Doppler Vibrometer for lateral vibrations can be extended for the case of Torsional Vibrations. In the case of lateral vibrations the velocity is measured along the laser beam direction. In the case of torsional vibrations the rotational motion has to be converted into measurable linear velocity. The core principal of a rotational Laser Vibrometer is to measure two linear velocities at two different circumferential points. For this a dual beam concept is realized. The basic idea is: The two laser beams hit the shaft at two points. The two points have different tangential velocity components. From these different velocities the angular velocity can be determined. In todays most modern systems the two beams come from one optical head (see Figure 26).

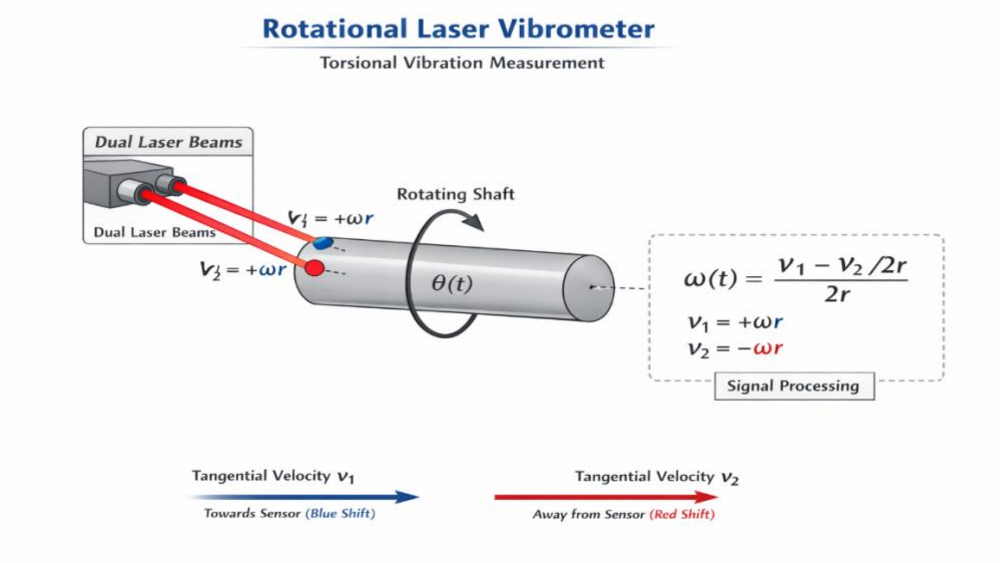


Figure 26 Configuration and Physical relations of a Dual Beam Laser Vibrometer

They have slightly different angles and two nearby spots on the shaft. A typical commercial solution is offered by Polytec RLV-5500.

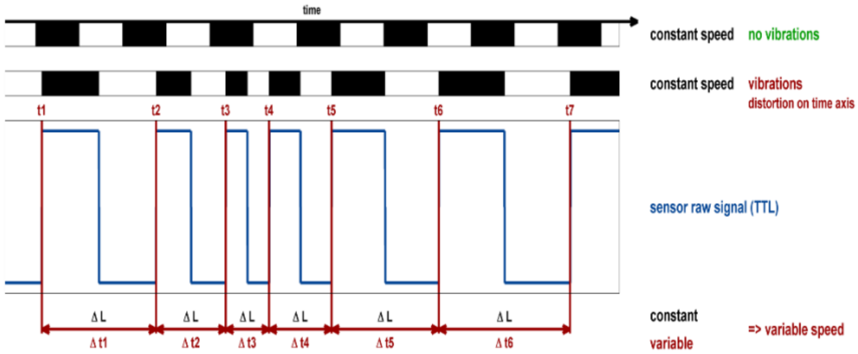
At Turbogenerators measurement locations can be at the shaft surface, at couplings and at the generator shaft. A reflective tape or a rough surface is sufficient. Markers are not required. Measurement quantities can be angular velocities, angular displacements (integration)

- **Advantages:**
  Non contact measurement, no sensors on the shaft, no telemetry and no slip rings. High measurement bandwidth and accuracy. Dual Beam Laser Vibrometers suitable for Steady State vibrations, electrical grid related torsional excitations, sub synchronous and higher order torsional modes. Minimal influence from other motions. Fast setup for measurements. No impact on Rotor Dynamics.

- **Disadvantages:**
  A clear optical path to the shaft is necessary. Sensitivity to harsh environments (steam, oil, dust). Dependence on Surface conditions. Long term stability. For permanent monitoring mechanical stability of the sensor mount is critical. High initial investment five to six figure range.

6.2 CODER BASED MEASUREMENT METHODS

Coder Based Measurement methods are using equidistantly-spaced markers on the shaft. Every time a marker passes in front of a sensor the sensor is measuring, following predefined time stamps. The time difference between two markers is used to estimate the angular velocity (see Figure 27). The general advantage of coder based techniques is, that they can deliver RPM and discrete angle positions. The data resolution is determined by the number of markers.



$$\dot{\varphi} = \frac{1}{R} \cdot \frac{\Delta L}{\Delta t} [rad / s]$$

Figure 27 Estimation of the Angular Veloity by Coder based Measurements

Different types of setups are used to provide markers (coder), e.g. stripes on the shaft (zebra tape), teeth of a gear. Also different sensors can be used for the marker detection: Electro-magnetic pickups, eddy current sensors or optical sensors. The output signals of the sensors are processed in order to determine angular displacements, angular velocities and or angular positions. We consider two cases of coder based methods, which work with equidistantly- spaced markers: 1. Gear on the rotor with teeth and 2. Coders with sufficient visible contrast mounted on the shaft (zebra tapes, zebra discs

6.2.1 Magnetic Pickup with gear teeth

Function: Gear on the Rotor with Teeth. The magnetic pickups detect changes in the magnetic field or magnetic flux. Sensor types can be magnetic resistive, magneto inductive or Hall sensors. Eddy Current sensors are possible as well.

Figure 28 Coder Based Measurement by means of Gear Teeth and Magnetic Pick Up

Advantages: Simplicity of instrumentation, Sensor fixed on nonrotating part;

Metallic Teeth used as Coder, Low Sensitivity to dust;

Disadvantages: Number of gear teeth limits pulses per revolution (ppr).

Machining accuracy limits accuracy of measurement

Fictive Torsional Vibrations due to relative movements.

6.2.2 Optical Sensor with zebra tape

Function: Coders with sufficient visible contrast mounted on the shaft (Zebra Tapes or dics). The optical sensor generates an electrical signal depending on the light intensity.

With the configuration with two zebra tapes at locations A and B it is possible to measure the Shear Stresses and the Torque between A and B. In a first step the angular velocity and the angular displacement can be determined for both locations as shown before. The Torque between A and B is proportional to the difference of the angular displacements.

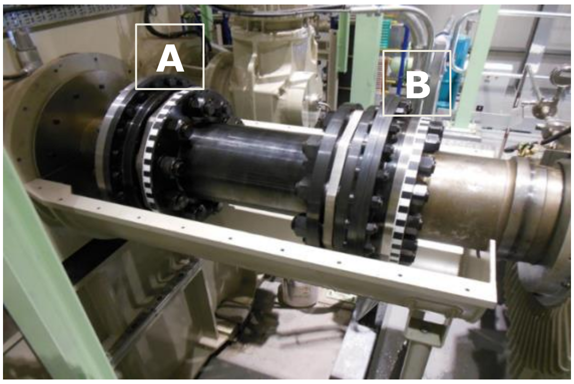


Figure 29 Coder Based Measurement by means of Zebra Tape and Optical Sensor

Advantages: Simplicity of instrumentation, Sensor fixed on nonrotating part;

Very high pulse rate possible.

Disadvantages: High sensitivity to ambient light; Visual contact between shaft and sensor required

6.3 TOUCHLESS MAGNETOSTRICTIVE TORQUE SENSOR

The touchless Magnetostrictive Torque Sensor is based on the following physical principal [9]. Figure 30 shows the 45-degree oriented tensile and compressive stresses in the shaft caused by a torque. The so called Villari-Effect states that the magnetic permeability μ of the shaft material dependents on these stresses. Due to this fact the magnetic flux B will change in a magnetic circuit with constant field strength H when the permeability μ changes. This effect is used in the magnetostrictive sensor.

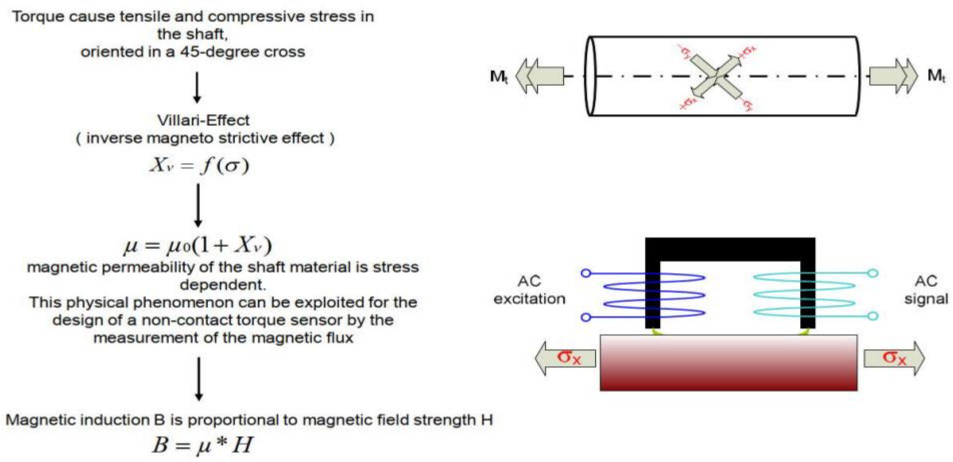


Figure 30 Physical Principal of the Touchless Magnetostrictive Torque Sensor

6.3.1 Touchless Siemens SSR Sensor

Function: The contactless Siemens Torque Sensor measures the static and dynamic torques at the rotating Turbogenerator shaft train. As described before it is based on the influence of mechanical stresses on the magnetic permeability of ferromagnetic materials. The Siemens SSR sensor can measure stresses up to 1000 MPa and works in a frequency range up to 200 Hz.

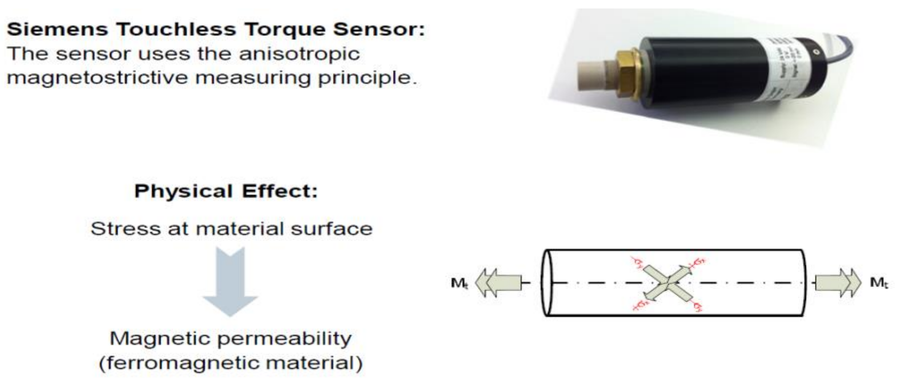


Figure 31 Practical Realisation: Contactless Siemens SSR Torque Sensor

Advantages: Contactless Sensor; Compact Equipment;

Disadvantages: EMV Sensitivity in Generator Area can be a problem

Mechanical and Electromagnetic Installation can be difficult.

Specialists needed.

The Siemens SSR sensor is used in the Turbogenerator of the OL3 unit in Olkiluoto. Figure 32 shows the measurement locations of the sensors

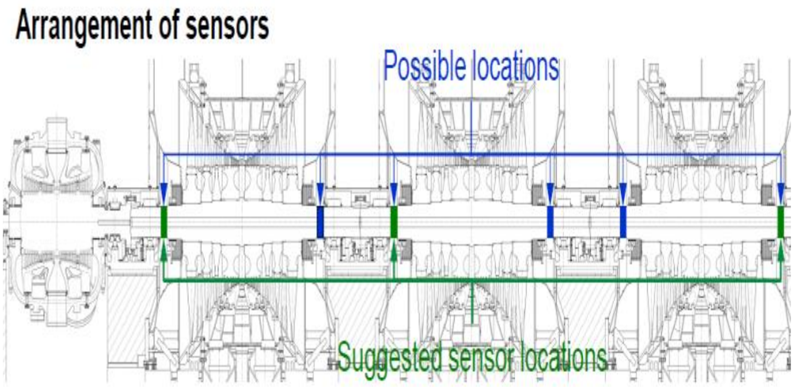


Figure 32 Measurement Locations of SSR Sensors in OL3

6.4 SENSOR SPECIFICATIONS FOR TORSIONAL VIBRATIONS

For the selection of the right sensor for a specific task Sensor Specifications have been defined, the most important ones are presented in the following chapter 6.4.1:

6.4.1 Key Sensor Specifications

- **Sensitivity:** The ratio of the sensor's output signal change to the input physical change. For example, in proximity sensors, it is often expressed as millivolts per micron
- **Frequency Range / Bandwidth:** The range of vibration frequencies (in Hertz), over which the sensor operates accurately without phase distortion or attenuation.
- **Resolution:** The smallest incremental change in the measured quantity that the sensor can reliably detect. For encoders, this is often the number of Pulses Per Revolution (PPR).
- **Measurement Range:** The minimum and maximum limits of the physical quantity (e.g., gap distance or angular displacement) the sensor can measure before losing accuracy.

- **Accuracy / Linearity:** The maximum deviation of the sensor's calibration curve from a straight theoretical line across its full measurement range.
- **Sample Rate / Max Frequency:** The frequency at which the sensor samples data. It must be high enough to capture high-order torsional harmonics without aliasing (typically >10 kHz for turbogenerators).

6.4.2 Comparison Matrix for Torsional Vibration Sensors

The table below synthesizes the most important performance specifications for the selected sensors from chapters 6.1 to 6.3 which can be applied for monitoring of the torsional vibrations of Turbogenerators.

Sensor Type	Parameter	Sensitivity	Frequency	Resolution	Telemetry/ Wireless
1.Piezo Acceleromete	Tangent. Displ.	10 to 100 mV/g	1 Hz to 10 kHz	Very High < 0,001 g	Yes Slip ring / Telemetr.
2.Strain Gauge (Metal Foil)	Torsional Shear Strain	Gauge Factor approx 2.0	0 Hz (DC) to >50 kHz	approx 1 mu epsilon	Yes Slip ring / Telemetry
3.Semicond. Strain Gauge	Torsional Shear Strain	Gauge Factor approx 100 to 150	0 Hz (DC) to >10 kHz	Extremely High <0.01 mu epsilon	Yes Slip ring / Telemetry
4.MEMS Acceleration Sensor	Tangent. Displ.	100 to 1000 mV/g	0 Hz (DC)} to 3 kHz	Medium to High <0.1 mg	Yes Slip ring / Telemetry
5.Dual-Beam Laser Vibrometer	Angular Velocity	10 to 100 (mV/s)/ rpm	0.5 Hz to 20 kHz	< 0.001 rpm	No Touchless
6.Gear Teeth + Magnetic Pick-up	Pulse / Time Interval	Pulse amplitude varies with speed	10 Hz to 5 kHz	Limit by tooth density	No Touchless
7.ZebraTape + Optical Sensor	Pulse / Time Interval	Digital pulse edges	1 Hz to 5 kHz	Very High (via edge timing)	No Touchless
8.Magneto-strictive Torque Sensor	Magnetic Permeability	1 to 10 V/Nm	0 Hz (DC) to 1 kHz	High	No Touchless proximity

Table 1 Specifications for selected Sensor types to measure Torsional Vibrations

6.4.3 Additional Comments for Torsional Vibration Sensors

Some additional comments are added to the sensor specifications in table 1:

- **1. Piezo Accelerometer:** Two accelerometers are mounted radially opposite each other on the shaft, oriented tangentially. Subtracting their signals cancels out translational lateral vibrations, leaving pure angular (torsional) acceleration. An on shaft wireless telemetry transmitter or a slip ringassembly is required in order to send power and transmit signals across the rotating boundary. High centrifugal forces may be a problem.
- **2. Strain Gauge (Metal Foil):** By using a full Wheatstone bridge pure torsional shear strain can be isolated, while ignoring bending and axial forces. The Sensitivity $V_{out} / V_{in} = GF\,\gamma$ (γ strain) is governed by the Gauge Factor GF, which is usually around 2.0. Sensors are highly accurate, but installing adhesive foil gauges on massive Turbogenerator shafts is labor intensive. Rotating Telemetry system is needed.
- **3. Semiconductor Strain Gauge:** Piezoresistive Silicon crystals can be used instead of metal wire or foil, but with the same bridge configuration. Sensitivity: The strain Gauge Factor GF = 100 bis 150 is very high compared to metal foil gauges (GF = 2). The resolution of this sensor is very high. During the commissioning phase of the Turbogenerator OL3 the turbine rotor has been instrumented by the manufacturer with this strain gauge type together with the permanent touchless SSR-Sensors (see 8. Magnetostrictive Torque Sensor). The main purpose of the temporary installed strain gauges was the comparison with the results from the SSR Sensors: Verification of the natural frequencies and the mode shapes. This was possible at low vibration levels, while at higher vibration levels the strain gauge signals are saturated due to exceeding the measurement range. The bearing location between LPT3 and Generaeor was best suited for a comparison. Both sensors could be placed very close to each other at this location (for comparison of the results see 8. Magnetostrictive Torque Sensor). In the application of the OL3 Turbogenerator the Semiconductor Strain Gauge can measure accurately a dynamic range of about 490 kNm. However, it cannot measure the whole (static) torque range (about 10190 kNm at 1600 MW).
- **4. MEMS Acceleration Sensor:** Compared to the Piezo accelerometers the MEMS sensors perform better for very low frequencies. This can be an advantage for half speed turbogenerators (1500 rpm), where the SSR modes may occur at very low frequencies. MEMS are compact and highly power efficient, allowing them to run on small batteries or inductive power loops combined with low power wireless chips (see also chapter 6.1.3 EPRI Subrock Monitoring system)
- **5. Dual Beam Laser Vibrometer:** This touchless sensor type works with two parallel laser beams aimed at the rotating shaft and determines by means

of the optical Doppler frequency shift the angular velocity of the shaft with a very high velocity resolution. The Dual Beam Laser Vibrometer works completely non-contact and touchless. It removes the need for telemetry, battery maintenance and shaft balance modifications. But the system is highly sensitive to dust and requires a clear optical path line of sight.

- **6. Geer Teeth + Magnetic Pick up:** This magnetic stationary sensor (variable reluctance or Hall effect) tracks the passage of individual teeth on an existing gear wheel (e.g, turning gear). The sensitivity relies on the number of pulses per revolution and the resolution is limited by geometric machining tolerances of the gear teeth. The sensor system is robust and highly reliable. It uses existing machine components, eliminating the need to add any components to the rotating assembly

- **7. Zebra Tape + Optical Sensor:** The stationary optical or fiber-optic sensor projects light onto a high-contrast black-and-white striped adhesive tape (Zebra tape) wrapped securely around the shaft circumference. A high speed timer processes the reflected pulse train. The sensitivity is proportional to the line density of the tape (number of lines per revolution). The resolution is guaranteed by a high accuracy due to sharp, crisp digital transitions between the black and white lines. The pulse density can be higher compared to standard gear wheels. However the tape can degrade over time due to heat, centrifugal forces, which limits its use primarily to short term diagnostics rather than permanent monitoring.

- **8. Magnetostrictive Torque Sensor:** As already described in chapter 6.3 this torque sensor measures changes in the magnetic properties of the shaft material itself under mechanical stress. Torsional twist creates mechanical stress, altering the shaft's magnetic permeability. The stationary excitation and the sensing coils sit adjacent to the shaft. As mentioned before the Turbogenerator of OL3 has been instrumented by the manufacturer with this type of touchless torque sensor as a permanent, so called SSR-Sensor. Their measurement range covers both static and dynamic operational strain ranges. Some technical data of this SSR-sensor are described by Siemens, e.g. the measurement range 1..1000 MPa, the resolution 0,1 N/mm² and the bandwith of 0..200 Hz
  The sensitivity depends on several influences, where the material is an important one. Therefore a onetime individual setting of the sensitivity is necessary for each rotor. This can for example be done in the power plants by comparing the static portion of the sensor signal with the current mechanical power. By comparison of the measurement results of the SSR-Sensor with the Strain Gauge Sensor a difference of about 10 % was found. In the test phase it was also found, that the noise level of the SSR-Sensors increased in the frequency range above around 50 Hz. One reason for the poor signal to noise ratio above 50 Hz can be, that the dynamic torque response of the Turbogenerator can be much smaller in the higher frequency range. However, for the touchless magnetostrictive torque sensor, poorer results above about 50 Hz can also come from other effects:

So can the practical bandwidth in a large turbogenerator installation be much lower because the whole system consists of shaft material, magnetized zone, pickup coils/Hall sensors, air gap, electronics, filtering,etc. shielding, and calibration. Therefore to achieve a higher torsional vibration range, e.g. up to 150 Hz, a **dynamic calibration** should be performed. One conclusion is, that the SSR Sensor as installed in the OL3 unit seems to be capable of measuring strains in the low frequency range, e.g. below 50 Hz. On the other hand the signals are less reliable at higher frequencies above 50 Hz.

6.5 INFLUENCE OF THE RADIATION ON SENSOR MEASUREMENTS

During the discussions in the progress meetings with the members of the reference group the question came up, how sensors can handle radiation. It was confirmed, that for some turbogenerator units radiation is no problem, but others may have radiation levels in the range of 3 mSv/h.

6.5.1 Gamma Radiation in the Machine Building of NP Plants

In the machine-building area of a nuclear power plant, the relevant radiation type for possible sensor degradation is primarily gamma radiation. Alpha radiation has very low penetration and is normally stopped by surface layers, housings or contamination barriers. Beta radiation may be relevant only in the case of local surface contamination or exposed sensitive materials.

Gamma radiation, however, is penetrating and can reach sensor internals, cable insulation and local electronics. Its principal damage mechanism is ionization: gamma photons generate secondary electrons and electron-hole pairs. In semiconductors this leads to total-ionizing-dose effects such as trapped charge, threshold shifts, leakage currents, offset drift and increased noise. In polymers, adhesives and cable insulation, gamma radiation causes bond breaking, radical formation, oxidation, chain scission or cross-linking, leading to embrittlement, insulation degradation and loss of mechanical or electrical stability. Therefore, the radiation risk for torsional vibration sensors is mainly a long-term ageing and calibration-stability problem, not a direct mechanical influence on the shaft torsional vibration itself.

6.5.2 Influence of Radiation Fields on the Measurement Equipments

Although the turbogenerator is located in the machine building and not in the reactor containment, local radiation fields may still be relevant for torsional vibration monitoring equipment. At dose rates of the order of several mSv/h, the torsional motion of the shaft itself is not affected, but sensors, cables, connectors and particularly local electronic signal conditioning units may suffer from cumulative radiation ageing. Gamma radiation mainly causes ionization in semiconductors and polymers, leading to leakage currents, threshold shifts, increased noise, signal drift, embrittlement of insulation materials and possible loss of calibration. Therefore, the measurement chain should be designed with radiation-tolerant passive sensors in the local area, qualified cables and connectors, and remote placement of active electronics and signal-processing units in a low-radiation environment.

The radiation sensitivity of torsional-vibration sensors depends strongly on the transducer principle. Metallic targets, gear teeth, magnetic cores and passive coils are comparatively robust against gamma radiation. The critical weak points are usually organic materials, such as cable insulation, potting compounds, adhesives and protective coatings, as well as semiconductor-based sensor elements and local electronics. Metallic foil strain gauges are mainly affected through adhesive, backing and insulation ageing, whereas semiconductor strain gauges and MEMS accelerometers may suffer directly from total-ionizing-dose effects, leading to resistance drift, offset drift, scale-factor changes, leakage currents and increased noise. Optical systems may experience radiation-induced attenuation, detector dark-current increase and loss of optical contrast. Therefore, for permanent torsional monitoring in a radiation field, passive inductive or magnetic pickups, remote optical systems, or coil-based magnetostrictive sensors are generally more favourable than shaft-mounted MEMS or semiconductor-based sensors.

7 Existing and New Monitoring-Technologies for Torsional Vibrations

The topic Torsional Vibrations of large Turbogenerators in Nuclear Power Plants is a critical element for Condition Monitoring. Unlike lateral vibrations, torsional oscillations are not directly assessable by measured vibration data only, but require dedicated measurement and evaluation techniques. According to the principles outlined in ISO 17359 „Condition Monitoring and Diagnostics of Machines“ an effective monitoring strategy must integrate appropriate sensor technologies, signal processing methods and diagnostic criteria to ensure a reliable machine operation. This chapter starts with the general idea of Monitoring Torsional Vibrations and presents besides the Existing Technology also a New Technology based on a Digital Twin for Torsional Vibrations.

7.1 THE GENERAL IDEA OF MONITORING TORSIONAL VIBRATIONS

Torsional Vibrations in Turbogenerators are primarily excited by transient electrical disturbances, such as grid faults and switching events, or by steady-state disturbances of electromechanical interactions , such as Load unbalances. These oscillations can lead to significant cyclic stresses, potentially causing fatigue damage in shafts, turbine blades, couplings and generator components (e.g. retaining rings). Therefore, continuous or periodic monitoring is essential to detect abnormal conditions at an early stage. A comprehensive Monitoring System typically includes the measurement of torsional strain, angular velocity fluctuations or indirect indicators derived from electrical signals. The acquired data are processed in both time and frequency domains to identify characteristic excitation frequencies and resonance conditions. Trending of the measured data , the comparison with baseline conditions and alarm thresholds form the basis for an assessment of the Condition of a Turbogenerator during operation. The monitoring process involves data acquisition, signal validation, feature extraction, condition evaluation and decision support. The ultimate objective is to enhance operational reliability, to prevent unexpected failures and to support maintenance strategies such as condition-based or predictive maintenance. Chapter 7 introduces the fundamental concepts of Torsional Vibration Monitoring, including measurement techniques (already presented in chapter 6), data evaluation procedures, and practical implementation in large power plant Turbogenerators. In today's applications Torsional Monitoring [5] is usually based on:

- Baseline values established at known operating conditions (Commissioning testing data from the manufacturer)
- Trending of the measured torsional quantities over time (angular displacements q(t), torsional stresses τ)

- Comparison with machine specific allowable stresses, fatigue values. The general Condition Monitoring standard is ISO 17359. It describes, that the Monitoring should be performed at predetermined operation conditions. Those predetermined conditions are used to establish Baselines (OEM). Later measurements during operation are compared with the Baselines and trended to detect changes

If for example shaft torsional stresses are measured in the time domain during operation the measurements can be used in the following ways, as it was already described in chapter 2.3.

- Peak event surveillance: looking for unusually high stress peaks during disturbances
- Spectral monitoring: Track amplitudes at critical torsional natural frequencies. Watch for changes in natural mode frequencies or in excitation frequencies: Grid frequency, twice grid frequency.
- Fatigue / Life consumption monitoring, important is the knowledge about fatigue damage. Practical Torsional Monitoring systems should therefore convert measured torsional response data into local shaft/blade stress values and then into cumulative fatigue usage. The main interest in torsional monitoring should be the lifetime consumption of a unit. And the fatigue life time consumption should also be tracked over the time.

For the assessment the machine specific comparison values should be used, developed from:

- the OEM Torsional Model with torsional natural frequencies and mode shapes (e.g. Finite Element Model of Turbogenerator, see chapter 3)
- the shaft/blade geometry and local stress concentration factors
- the material fatigue properties
- the measurement location
- the commissioning Baseline measurements

7.2 THE BASICS OF EXISTING TORSIONAL MONITORING TECHNOLOGIES

This subchapter 7.2 describes in more detail the important basics of existing Monitoring Technologies, which are used in order to achieve an optimal Monitoring procedure for Torsional Vibrations. These basics are described in the following subchapters 7.2.1 – 7.2.15.

7.2.1 Monitoring Objectives and Scope

The first step is to define the objective of the torsional vibration monitoring system. For large turbogenerators, the main objective is not only to measure torsional motion, but to assess the mechanical stress condition of the shaft train during normal operation and during electrical or mechanical disturbances. The monitoring system should support:

- early detection of abnormal torsional vibration,
- identification of excitation mechanisms,
- assessment of fatigue-relevant shaft stresses,
- comparison with reference or baseline conditions,
- support of condition-based maintenance decisions.

The monitored components typically include the turbine-generator shaft train, couplings, turbine blades, the generator rotor, the exciter shaft and other torsionally sensitive elements.

7.2.2 Definition of Relevant Torsional Excitation

Before selecting sensors and evaluation methods, the relevant excitation mechanisms must be identified. In large turbogenerators, torsional vibrations may be caused by:

- grid faults,
- short circuits,
- switching events,
- **s**ub **s**ynchronous **r**esonance (SSR)
- converter or grid interactions,
- steady-state electromagnetic torque pulsations (e.g. Load unbalance),
- mechanical coupling effects with blades.

A distinction should be made between transient disturbances and steady-state disturbances (see Table in Figure 8). Transient disturbances usually excite free torsional vibrations of the shaft train following a short but strong forced excitation (e.g. 2 phase short circuit), whereas steady-state disturbances may cause forced torsional vibration at specific excitation frequencies (e.g. 2x grid frequency in case of load unbalance) over a long time period

7.2.3 Identification of Torsional Natural Frequencies

A key element of the monitoring concept is the knowledge of the torsional natural frequencies of the complete shaft train. These frequencies are normally determined during the design and commissioning phase by analytical models, finite element calculations, and measurements (see chapter 4.2)

For monitoring purposes, the following information is required:

- torsional natural frequencies,
- torsional mode shapes,
- critical shaft sections,
- modal stress distributions,

- damping values,

This information forms the basis for defining frequency bands, alarm limits, and diagnostic criteria.

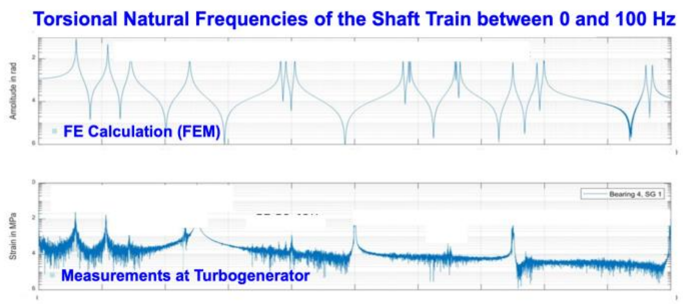


Figure 33 Calculated and Measured Torsional Natural Frequencies of a Turbogenerator

Figure 33 shows calculated and measured torsional natural frequencies of a large Turbogenerator in the frequency range between 0 and 100 Hz.

7.2.4 Selection of Measurement Quantities

Torsional vibrations can be monitored using different physical quantities. Typical measurement quantities are:

- shaft stress or strain,
- shaft torque,
- angular displacement,
- angular velocity variation,
- angular acceleration,
- electromagnetic air-gap torque,
- electrical power or current signals as indirect indicators.

Direct strain or torque measurement gives the closest relation to fatigue damage. Speed-based measurements are often easier to install but require conversion into torsional deformation or stress using a shaft train model (see chapter 5.3.7)

7.2.5 Sensor Selection and Installation

The sensor concept depends on accessibility, operating conditions, and required accuracy. Possible sensor systems include (see chapter 6):

- strain gauges on the shaft,

- accelerometer on the shaft
- telemetry systems for the data transmission,
- magnetic or optical encoders,
- toothed-wheel probes,
- rotational laser vibrometers,
- touchless magnetostrictive stress sensor
- generator electrical signal analysis,

The sensor locations should be selected according to the torsional mode shapes and the expected high-stress regions. In practice, sensors are often installed near couplings, shaft ends, generator rotor sections, or other accessible shaft locations.

7.2.6 Data Acquisition and Signal Conditioning

The data acquisition system must be suitable for both slow trends and fast transient events. Important requirements are:

- sufficiently high sampling rate,
- synchronized acquisition of mechanical and electrical signals,
- anti-aliasing filters,
- trigger functions for transient events,
- storage of pre-trigger and post-trigger data.

For power plant applications, it is useful to record not only torsional vibration signals but also electrical quantities such as voltage, current, active power, reactive power.

7.2.7 Signal Processing

The purpose of signal processing is to transform raw measurement signals into physically meaningful quantities such as torsional stress, modal response, and fatigue-relevant parameters. This step is essential because torsional vibrations are often not measured directly but inferred from indirect quantities (e.g., speed fluctuations). The measured signals are processed in the time and frequency domains. Important evaluation methods are:

- **Time-domain analysis:** Time domain evaluation is the primary tool for transients events such as short circuits, grid faults, load rejection. Typical analysis includes peak to peak torsional oscillations, maximum stress amplitudes and decay behavior after excitation. The decay of oscillation is particularly important for estimating modal damping, which is a key parameter for system stability. In case of transient analysis special attention must be given to high sampling rates and event triggered recording
- **Frequency-domain analysis**: The transformation into the frequency domain is usually performed using the Fast Fourier Transform (FFT). Key objectives are the identification of torsional natural frequencies, the detection of excitation

frequencies (e.g. the grid frequency and the 2xgrid frequency), the separation of forced and natural vibrations.

- **Order analysis**: Order tracking is essential when rotational speed varies (startup or shutdown)
- **Campbell diagrams:** Tracking of resonance crossings
- **Modal decomposition:** If multiple sensors are available, Modal Analysis can be performed. The Modal Decomposition makes it possible, to separate different torsional modes, to identify mode shapes and to localize critical shaft sections. This is particularly useful for Diagnostics.
- **Stress Conversion:** Measured signals (e.g. angular velocity differences) can be converted into torque and stress using a shaft model (see chapter 5.3.7). This step connects measured data with mechanical integrity.
- **Rainflow Counting**: Rainflow Counting is used for fatigue assessment (see 7.2.13)

7.2.8 Conversion to Mechanical Stress

For the condition assessment of a Turbogenerator, torsional vibration amplitudes should be converted into mechanical stress. This requires a mechanical model of the shaft train.

The basic relations have already been derived in chapter 5.3.7

$$\tau_{max} = \alpha_k \ (M_T / W_T) = \alpha_k \ (G \, I_T \, q' \ (x,t) / W_T) = \alpha_k D / 2 \ (G \, q' \ (x,t)) \qquad (7.1)$$

with $W_T = \pi \, D^3 / 16$ and the notch factor α_k and the torque

$$M_T = G \, I_T \, dq / dx = G \, I_T \ q' \ (x,t) \qquad (7.2)$$

The stress-based evaluation is important because the relevant damage mechanism is usually fatigue and for a fatigue evaluation stresses are needed.

7.2.9 Definition of Reference Values and Baselines

According to the general ISO 17359 philosophy, monitoring should be based on reference conditions. For torsional vibration monitoring, such reference values may include:

- commissioning measurements from the manufacturer,
- calculated design values,
- manufacturer limit values,
- normal-operation baselines,
- permissible stress ranges,
- fatigue-based limit values (Rainflow, Wöhler Curves, Life time)

Because universal torsional vibration limits are often not available, the assessment is usually machine-specific and needs data from the manufacturer.

7.2.10 Alarm Levels and Diagnostic Criteria

Alarm levels should be defined in a structured way. A practical approach is:

- normal condition,
- increased observation,
- warning level,
- alarm level,
- trip or protection level

The criteria may be based on:

- torsional stress amplitude,
- frequency components near natural frequencies,
- accumulated fatigue damage,
- deviations from baseline,
- repeated occurrence of critical events.

The alarm philosophy should distinguish between short transient events and continuous steady-state excitation.

7.2.11 Event Detection and Classification

The monitoring system should automatically detect and classify relevant events. Typical event classes are (see also the table in figure 8):

- grid fault,
- short circuit,
- synchronization event,
- load rejection,
- sudden load change,
- subsynchronous resonance event,
- steady-state torque pulsation (Load or line unbalance)

For each event, the system should store the time histories, spectra, maximum stress amplitudes, affected modes, and estimated fatigue contribution (life time)

7.2.12 Trending and Long-Term Condition Assessment

In addition to the event-based monitoring, long-term trending is required. Important trend parameters are:

- maximum torsional stress per event,
- dominant frequency components,
- damping behaviour,

- accumulated fatigue damage,
- changes in natural frequencies,
- changes in response amplitude.

Trending allows detection of slow changes in the shaft train or operating environment.

7.2.13 Fatique Assessment

The ultimate goal of Torsional Vibration Monitoring for large Turbogenerators is not only to detect torsional vibrations but to assess Fatique Damage in the shaft train. The monitoring system should therefore estimate the fatigue relevance of measured torsional vibration. For this estimation the relevant quantity is torsional stress. A typical procedure for the assessment can be subdivided as follows:

- **Conversion from measured torsional response into stress:** The measured signals are converted into shear stress, following the formulas shown in chapters 5.3.7 or 7.2.8. This results in a stress time history, which is the input for the Fatique Analysis.
- **Identification of Stress Cycles:** The real torsional signals are irregular and contain transient oscillations, multi-frequency components and varying amplitudes. Due to this fact the stress cycles must be extracted using the Rainflow Counting method.
- **The Rainflow Counting method:** The principle of Rainflow Counting is, that the stress time signal is decomposed into individual cycles, where each cycle is characterized by a stress range $\Delta\tau$, by a mean stress τ_m and the number of occurences n_i of these cycles. This method ensures a physically meaningful representation of fatigue-relevant load cycles.
- **Evaluation of Stress Cycles in Wöhler Curves (S-N Curves):** Each stress cycle is evaluated using material fatique data (see figure 34). In the Wöhler Curves or S-N Curves (S represents the Stress amplitude and N the number of cycles to failure). The Wöhler Curves are provided by manufacturer data, material standards or experimental testing.

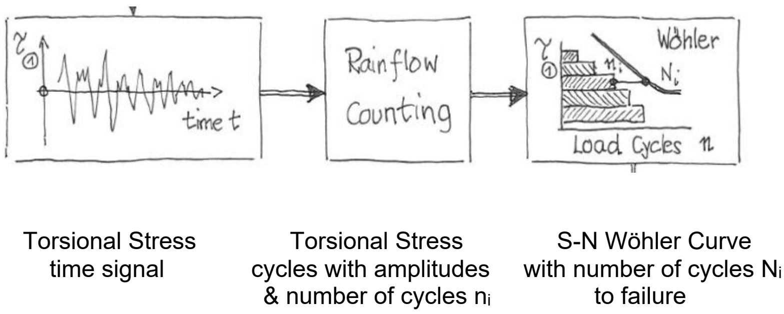


Figure 34 Life Time Assessment by means of Rainflow Counting

- **Damage Calculation by means of the Miner's Rule:** Fatigue damage is accumulated using the Miner's Rule (see figure 34)

$$D = \Sigma \; n_i / N_i \qquad (7.3)$$

where n_i is the number of observed cycles and N_i ist he number of cycles to failure at that stress level.
If D < 1, the condition is safe. D = 1 is the limit where a failure is expected (see figure 34).
In Turbogenerators fatigue is often dominated by rare events like short circuits, grid disturbances and synchronization events. Fot these events damage increments have also to be calculated and stored. For a long term Fatigue Monitoring this has to be included in the total Fatigue damage value D_{total} over time

$$D_{total} = D_{events} + D_{steady\ state} \qquad (7.4)$$

With D_{total} the remaining lifetime can be estimated and maintenance can be planned.
The Fatique Assessment can be integrated in the monitoring system by an automatic rainflow counting and a continuous update of fatigue usage.

7.2.14 Diagnostics and Root Cause Analysis

If an abnormal torsional vibration is detected, the next step is diagnosis. The monitoring system should help to answer:

- Which torsional mode was excited?
- Which excitation frequency was responsible?
- Was the event caused by the grid, the generator, or the shaft train?
- Was the response transient or steady-state?
- Was a resonance condition involved?
- Is the event fatigue-relevant?

This diagnostic step is essential for the decision, whether operation can continue or whether further inspection is necessary.

7.2.15 Reporting and Maintanance Decision

The final step is reporting and the decision support. The monitoring system should provide clear results for operators, maintenance engineers, and specialists:

- description of the event,
- measured time histories,
- frequency spectra,
- maximum torsional amplitudes,
- estimated stresses,
- comparison with limits,
- fatigue damage contribution,
- recommendations for further actions.

The result may be continued operation, increased monitoring, detailed inspection, model update, or operational restriction.

7.3 THE ISO/EPRI DIAGRAM FOR MONITORING TORSIONAL VIBRATIONS

A complete Block Diagram for Monitorig of Torsional Vibrations recommended by ISO/EPRI is shown in figure 35. The Monitoring chain consists of the following components.

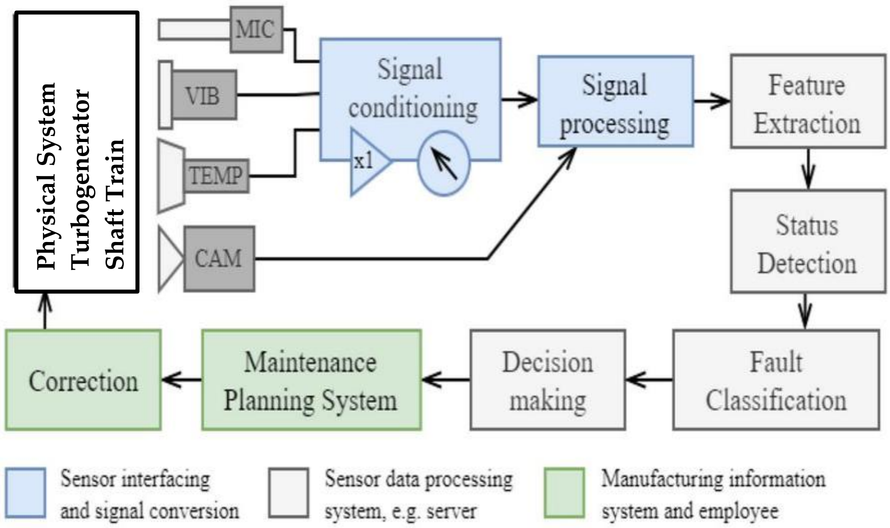


Figure 35 The ISO/EPRI Block Diagram for Monitoring of Torsional Vibrations

- In our case the **Physical System** is the Turbogenerator with the shaft train, consisting of the turbines, the generator and the exciter, the couplings, the turbine blades and other connected parts.

- In the primary measurement layer the **Sensors** to measure torsional vibrations (VIB) can be strain gauges, encoders (e.g. zebra tapes), a touchless torsional laser vibrometer or a touchless magnetostrictive sensor. The mechanical torsional motion or the mechanical stresses are converted into electrical signals.

- In some of the cases the sensor data have to be transfered from the rotating to the stationary frame by telemetry (radio transmission), slip rings, by inductive or optical transmissions. The following part of **Signal Conditioning** includes all operations applied to the raw sensor signals before a deeper analysis is performed. This includes for example amplification, filtering and finally the Analog to Digital Conversion (ADC) with typical sampling rates, leading to a digital time series.

- The **Signal Processing** is one of the most important blocks. The purpose of this block is to transform the signals into physically meaningful quantities and diagnostic features. Typical analysis tasks are: time domain analysis (time histories e.g. stress or torque vs time), frequency domain analysis (FFT) to identify natural frequencies or excitation frequencies and signal transformations (Integration, differentiation).

- The **Feature Extraction** reduces the data to characteristic values, e.g. natural frequencies, grid frequency, peak to peak values of torsional stresses, modal parameters, damping estimation. The idea is to have a more concentrated description of the state of torsional vibrations.

- The **Status Detection** determines whether the Turbogenerator condition deviates from a defined baseline. It is a comparison with reference values (baselines, trend history, alarm thresholds). The main question is: Is the Turbogenerator behaving normal or not?

- Once an abnormal condition is detected, the block **Fault Classification** identifies the type and the source of the fault. The input is coming from the Status Detection and the output of this block presents the identified fault type. Typical fault categories in Turbogenerator systems are: Electrical network excitation with a strong 50 Hz component, SSR (Sub Synchronous Resonance) with a torsional natural frequency of the shaft train, Mechanical degradation (e.g. crack) with an increasing amplitude over time, a Short Circuit with high transient peaks.

- The block **Decision Making** converts diagnosis into actions or recommendations. In the processing part of this block an evaluation of risks is considered. This can for example be based on the Fatique Assessment (Rainflow Counting, Wöhler Curves, Miner's Rule, Lifetime estimation, see 7.2.13) or can be based on short time but strong stresses (Short circuit). Typical decisions can be: Preventive by scheduling inspections or adjusting operating conditions or Protective by load reduction, controlled shut down or trip signal.

- Based on the Decision making the **Maintanance** and possible **Corrections** can be planned

7.4 NEW TORSIONAL MONITORING-TECHNOLOGY WITH A DIGITAL TWIN

7.4.1 Introduction – Monitoring with a Digital of the Turbogenerator

The previous chapters have shown, that Monitoring Technologies for Torsional Vibrations are based on several machine specific vibration data: Torsional natural frequencies, excitation frequencies, mode shapes, geometrical design data and material data. These data are provided by the manufacturer, who determines these data by means of simulations with a Theoretical model and by Experimental Analysis (testing) in the phase of commissioning. Considering the fact, that the

Monitoring uses the data of a Theoretical Model via a numerical Analysis for the Torsional Vibrations , the question arises why the Monitoring could also be performed directly by the combination of the three tools: Theoretical Modelling, Numerical Analysis and Experimental Analysis. This leads to the idea of a Digital Twin [8], [10], [11], which would be an integrated part of the Monitoring system, presenting the real system as a digital copy (see also figure 37). At the beginning of this chapter the basic idea of a Digital Twin will be presented. After a more general description the application of the Digital Twin for the New Torsional Monitoring Technology for Turbogenerators will be presented.

7.4.2 The Digital Twin – A Digital Copy of the Real System

A Digital Twin is a digital copy of a real system. The basic idea of such a Digital Twin is shown in figure 36 for an arbitrary system at operation. By means of sensors at some specified locations of the real system the dynamic behavior (e.g.the torsional vibrations of the Turbogenrator) can be measured and observed continously. On the other hand with a model of the real system inside the Digital Twin predictions of the dynamic behavior can also be performed at the sensor locations. The deviations between sensor observations at the real system and predictions from the model inside of the Digital Twin are the base for an identification and update of the system parameters and the operational conditions. The identified system changes can for example be an indication for internal system failures or external disturbances. The deviations can also be used to generate signals for an active system control. In addition, recommendations for experts can be derived from the deviations for a possible change of the operation conditions.

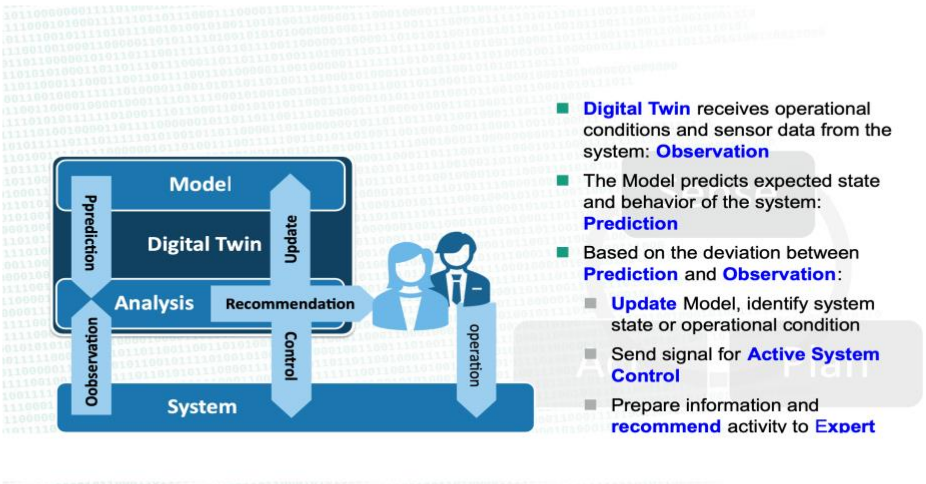


Figure 36 A Digital Twin for a General System at Operation

From figure36 it can be recognized again , that for the function of the Digital Twin a relevant model and analysis procedures are needed. With respect to the development of a Digital Twin for the torsional vibrations of a turbogenerator the three engineering tools: Theoretical Modelling, Numerical Analysis and

Experimental analysis are necessary. These tools will be discussed in the next chapter together with the tasks for the function of the Digital Twin.

7.4.3 The Tools for the Development of the Digital Twin

Figure 37 shows again the three tools, which are usually applied to solve vibration problems in mechanical engineering. These tools are besides theoretical modelling the numerical and the experimental analysis. The theoretical modelling is based on physical laws, particularly from mechanics. They lead to equations of motion, which express the dynamic behaviour of mechanical systems. By means of numerical analysis equations of motion can be solved for natural as well as for forced vibrations. And vibrations can be measured by experimental analysis with sensors, actuators and devices for signal processing. By different combinations of the three tools the following tasks of Simulation, Validation and Identification can be performed, which is shown in figure 37.

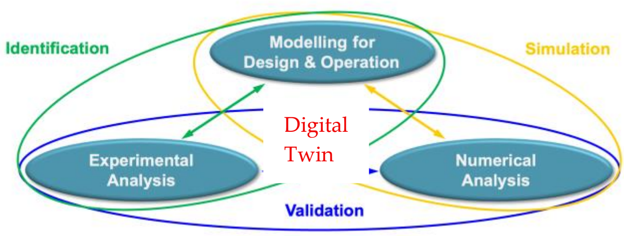


Figure 37 Tools and Tasks to solve Vibration Problems in Mechanical Engineering

7.4.4 Digital Twin Concept for the New Torsional Monitoring Technology

The concept of the Digital Twin for the Monitoring of Torsional Vibrations of Turbogenerators is presented in figure 38. The real system of the turbogenerator (black frame) is excited by air gap torques leading to real torsional vibrations along the shaft train. Some of these vibrations are measured by sensors at defined locations. These sensor signals are input data for the Digital Twin of the turbogenerator (blue frame). They are transferred to the internal difference location of the Digital Twin: **2.**

An important part of the Digital Twin is a Finite Element Model: **1**. With this FE Model natural vibrations as well as forced torsional vibrations due to the excitation can be calculated at each location along the shaft line. The part of the calculated torsional vibrations corresponding to the sensor positions is also transferred to the Difference location: **2**.

The differences between the measured and the calculated vibrations are introduced via a feedback into the calibration component of the Digital Twin. The feedback and the calibration are subdivided into two parts, one for calibrating the system parameters: **3** of the model and one for the excitation: **4**.

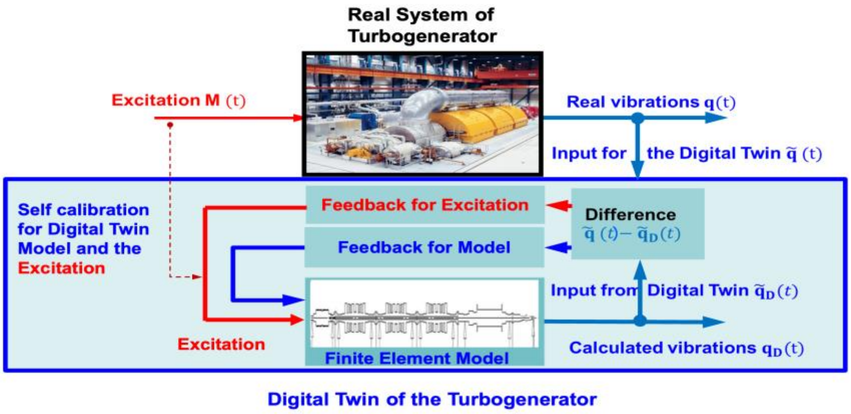


Figure 38 **Concept of the Digital Twin for Torsional Vibrations of Turbogenerators**

This is necessary due to the fact that the torsional vibrations as system output can be influenced either by the excitation (system input) or by possible changes of the system parameters. If the Digital Twin works without problems in the way of the described concept the state of the torsional vibrations of the turbogenerator can be determined at any time of operation and at any location of the FE model. Besides the torsional vibrations other derived quantities like shear stresses, torques etc can be determined as well. In figure 38 the four components of the Digital Twin are marked by their numbers: **1**. Finite Element Model of the turbogenerator, **2**. Comparison point of the Digital Twin (Difference: measured an calculated vibrations), **3**. Calibration of the mass and stiffness matrices and calibration of the modal damping and **4**. Calibration of the air gap torque.

The Digital Twin is well suited for the further development of a New Monitoring Technology for Torsional Vibrations of Turbogenerators. It can run permanently parallel to the real turbogenerator, working

- as a **Monitoring system**, which determines continuously the torsional vibration state of the Turbogenerator at each location of the shaft train in the time domain as well as in the frequency domain (Virtual Sensor
- as a **Detection system** for the identification and diagnosis in case of system failures, disturbances (Air Gap Torques) and parameter changes (mass, stiffness, damping)
- as a **Fatique Assessment system** using the Rainflow Counting method together with Wöhler curves and the Miner's Rule to estimate the Life time of components of the Turbogenerator: Shaft lines of Turbine and Generator, Couplings, Turbine blades, Retainer Rings.

The successful operation of the Digital Twin could be demonstrated by some simulation cases. For the operation in a Nuclear Power Plant a Data Logging system is needed to prepare the measured sensor signals. First trials to operate the Digital Twin in a plant could be started at the OL3 Turbogenerator (Olkiluoto), because OL3 is already operating with SSR-sensors and has a Data Logging system.

8 Sensor- and Monitoring Technology developed by Siemens

8.1 SIEMENS SENSOR TECHNOLOGY

8.1.1 Selection of the Measurement quantity

In order to monitor the shaft torsional behavior during an electromechanical excitation event (especially SSR, but also others like short circuits, load unbalance, etc.) Siemens has chosen to measure the torsional stress directly at the rotor rather than measuring shaft twist angles or electrical quantities. The focus is therefore less on torsional displacements, but more on torsional shear stresses, dynamic torque oscillations and ultimately on fatigue assessment of the turbine-generator shaft train with its components couplings, turbine blades, retainer rings and others.

8.1.2 The Touchless Magnetostrictive Sensor (SSR Sensor)

To measure the torsional stress in a Turbogenerator shaft, the touchless torque sensor can be used. The physical principle of this sensor type has already been presented in detail in chapter 6.3. As shown before, the sensor is based on the variation of the magnetic permeability under torsional stress materials. The Siemens SSR sensor can measure stresses up to 1000 MPa and works in a frequency range up to 200 Hz. (evtl. noch Sensordaten)

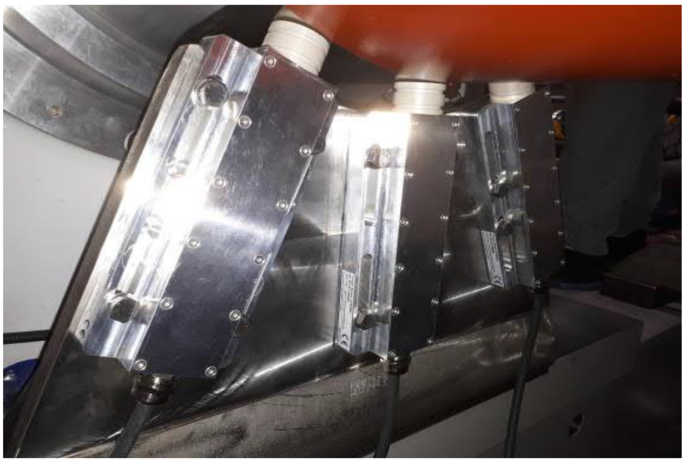


Figure 39 Touchless Magnetostrictive Sensor – SSR Sensor

A major advantages of the SSR-sensor is, that it works contactless. Furthermore there are no slip rings and no telemetry needed. A permanent installation (see figure 39) on very large Turbogenerators is possible. A disadvantage can be the EMV sensitivity in the generator area. The mechanical and electromagnetic installation can be difficult, where specialists are needed.

8.1.3 Preparation of the SSR-Sensor - Calibration

The sensitivity of the SSR-Sensor depends, among other things, on the material. Therefore, a one-time individual setting of the sensor sensitivity is necessary for each rotor. This can be done in power plants by comparing the static portion of the sensor signal with the static torque at the current mechanical power.

8.1.4 Measurement Locations for the Siemens Monitoring Technology

The development and design of the Siemens SSR-Sensor and Monitoring System is based on the assumption, that during a Sub Synchronous Resonance (SSR) event mainly one torsional mode shape will be excited and that the shaft train responds with the corresponding torsional eigenvector.

In order to detect an SSR event by means of a stress measuring sensor - like the SSR sensor - the sensor should be located at a shaft position, where high stresses occur. With a high stress level a good signal to noise ratio can be expected. The information of high shaft stresses for the sensitive sub-synchronous natural frequencies below 50 Hz can be obtained from the stress mode shapes (stress eigenvectors). Usually only the torsional deflection mode shapes are calculated in a Rotordynamic Analysis. However, there is a simple relation between the torsional deflection mode shapes and the torsional stress mode shapes. This has already been shown in figure 6 of chapter 4.3. High slopes in the deflection mode shapes correspond with maximum stresses in the stress mode shapes. This relation is presented in Figure 4o for the four lowest torsional natural frequencies of a large Turbogenerator. The highest stresses in the first four modes will occur in the areas of the red lines. Basically all bearing locations are suited for sensor locations. However, due to the analysis of the deflection and stress mode shapes the red Bearings B3, B5, B8 have been recommended by Siemens as measuring locations in this application, because of the highest slopes dq/dx in these areas.

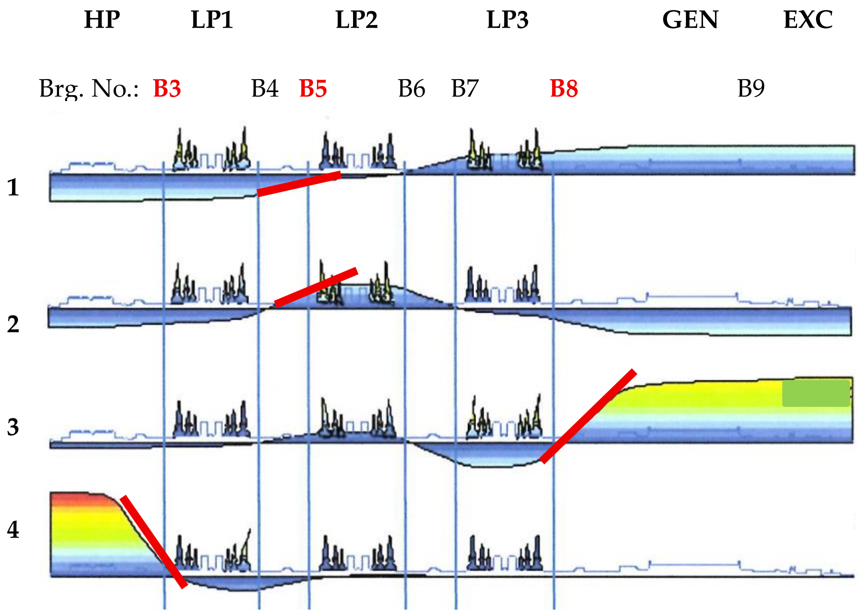


Figure 40 The four lowest Torsional Deflection Mode Shapes q(x) with Slopes dq/ dx

It is well known, that the deflection as well as the stress mode shapes only present relative values between the different shaft locations. However, the stress eigenvectors can be scaled to reach limit values of the material at the life leading location (also called life limiting location). These are the locations in the mode shapes, where the stress reaches its maximum. Due to the fact, that this location is usually not the same location as the measuring location an adjustment of the stress limit value has to be made. Siemens has used the limit value of 1/ $\sqrt{3}$ $R_{p\text{-}0,2}$ for the rotor material based on the yield strength $R_{p\text{-}0,2}$ (Steel 500- 800 N/mm^2 for the turbine rotor and 700 – 1100 N/mm^2 for the coupling bolts – alloy steel).

In Table 2 the limit values of the mode depending torsional dynamic stresses for the different bearing locations B1 – B11 are shown, based on the scaled eigenvectors. These dynamic stress levels are used as a reference for preliminary notification levels for the operator, which are at 50 % and 75 % of the limit values in table 2. The selected measuring locations and the corresponding limit stress values are highlighted in bold letters (Table 2 from a internal TVO meeting, not published)

Location	Bearing No.	Stresses [N/mm^2 0-p]							Max/Min [-]	Max/Min Monitoring Modes [-]
		Mode 1	Mode 2	Mode 3	Mode 4	Mode 5	Mode 6	Mode 7		
		6.0 Hz	10.8	14.4	23.4	36.7	38.0	39.0		
S	B1	0.1	0.2	0.0	0.7	0.7	0.7	0.7	20.5	
	B2	55.6	76.7	13.8	273.1	273.1	273.1	273.1	19.8	
LP 1	**B3**	33.6	46.2	8.3	**159.6**	**150.8**	**149.8**	**148.9**	19.3	1.1
	B4	321.0	372.4	48.4	25.3	106.6	48.3	190.5	14.7	
LP 2	**B5**	**322.4**	**370.6**	47.7	26.5	76.0	63.5	189.0	14.0	1.1
	B6	351.8	291.3	126.2	4.2	13.0	142.9	147.8	83.6	
LP 3	B7	350.6	293.6	125.0	4.4	49.6	134.1	151.7	79.8	
	B8	98.4	178.0	**260.0**	0.8	123.8	82.2	44.1	324.8	1.0
GN	B9	95.3	174.1	257.1	0.8	142.1	95.6	51.9	311.5	
	B10	4.4	8.2	12.5	0.0	10.4	7.3	4.1	277.6	
Exc.	B11	0.0	0.1	0.1	0.0	0.1	0.1	0.0	269.5	

Table 2 Mode dependent Dynamic Torsional Stress Values at Bearing Locations

It has to be mentioned , that the defined Monitoring concept with limit stress values has been particularly developed for SSR events. Other electrical faults have an impulse excitation and/or an excitation at grid or double grid frequency (see table in Figure 8). In such cases the vibration response of the sub synchronous modes will be much smaller, than the overall response. Siemens concludes, that the vibration amplitudes at other electrical faults do not exceed the above mentioned notification levels.

8.2 SIEMENS MONITORING TECHNOLOGY

8.2.1 The Signal Conditioning

As described in chapter 7.3 the Signal Conditioning is applied to the raw sensor signal before a further analysis is performed. This includes amplification, filtering, offset corrections and finally the Analog to Digital conversion (ADC) with sampling and quantization. Signal conditioning prepares the signal that it is noise reduced and in digital form for the following signal processing.

8.2.2 The Signal Processing

In the Siemens Monitoring technology the purpose of Signal Processing is also to transform the signals into physically meaningful quantities and features for the Diagnosis. Typical signal processing tasks are to determine time histories for the torsional stresses and the shaft torques at specified locations, the transformation of the time signals into the frequency domain by Fast Fourier Transformation (FFT) and to present frequency spectra with machine specific discrete frequencies.

8.2.3 The Feature Extraction

The Feature Extraction reduces the data to characteristic values: e.g.the torsional natural frequencies of the Turbogenerator, the excitation frequecies (grid and double grid frequency), the peak values (maximum) of the torsional stresses at the bearing locations B3, B5, B8, the damping estimation of the decaying oscillations. The idea of the feature extraction is to have a more concentrated description of the state of the torsional vibrations of the Turbogenerator.

8.2.4 The Status Detection

The Status Detection has to find out, whether the condition of the dynamic behavior of the Turbogenerator has changed. Which deviations from predefined Baselines have been observed? Have the torsional stress values at the defined bearing locations B3, B5, B8 changed to higher values? How is the distance to the torsional limit stress values at these locations? Have the torsional natural frequencies changed? If the damping characteristic is not the same as before, the danger of an instability (SSR-event) exists?

8.2.5 The Fault Classification

By **Fault Classification** the type and the source of the fault will be identified. The input for the Fault Classification is coming from the Status Detection and the output presents the identified fault types. The process to find out the fault type includes a pattern recognition (frequency signatures, time domain patterns and if available mode shapes) and a comparison with known fault signatures (Data base, model based approaches, AI machine learning results). Typical fault categories in Turbogenerator systems can be: Grid faults and short circuits with high transient peaks, change of load unbalance with an increasing amplitude of the twice grid frequency, Sub Synchronous Resonance (SSR) with increasig vibrations with a torsional natural frequency of the shaft train below the grid frequency (50 Hz/60 Hz). In case of changing torsional natural frequencies with changing vibration amplitudes the sources can be changes of the mechanical system parameters like stiffness changes in the shaft train: crack propagation, material fatigue, local plastic deformation, temperature influences (Youngs Modulus), loosening of bolts and plastic deformation. The more general output of the fault classification can be: Electrical excitation, transient fault events, increasing load unbalance, possible shaft damages (cracks) and resonance crossings. Depending on the fault type the fatigue damage may be more or less influenced.

8.2.6 The Decision Making

The **Decision Making** block converts Diagnosis into actions or recommendations. The input information for the Decision Making is the status of the system (normal, warning, alarm), the Fault classification and the operating conditions (load, speed, grid state). In the processing part of this block an evaluation of risks is considered. This can for example be done by a Fatique Assessment (Rainflow Counting, Wöhler Curves, Miner's Rule, Lifetime estimation, see 7.2.13) or can be based on short time but strong stresses (Short circuit). Output decisions can be: Preventive decisions by scheduling inspections or adjusting operating conditions and Protective decisions (critical cases) by load reduction, controlled shut down or changes for the trip signal. Based on the **Decision Making** the **Maintanance** and possible **Corrections** can be planned (see also Figure 35 in chapter 7.3)

9 Sensor- and Monitoring Technology developed by General Electric (GE)

9.1 GENERAL ELECTRIC (GE) SENSOR TECHNOLOGY

9.1.1 Selection of the Measurement quantity

In order to monitor the shaft torsional vibrations of Turbogenerators in case of electromechanical excitations General Electric (GE) has chosen a different way of measurements compared to Siemens [6], [8]. GE prefers to measure torsional vibration velocities of the rotor by means of encoder based methods, using equidistantly-spaced markers on the shaft. Every time a marker passes in front of a sensor the sensor is measuring, following predefined time stamps. The time difference between two markers is used to estimate the angular velocity. This has already been shown in chapter 6.2 (Fig.27). The two methods using a magnetic pickup with a gear teeth and an optical sensor with a zebra tape were demonstrated in chapters 6.2.1 and 6.2.2. The data resolution is determined by the number of markers. The general advantage of coder based techniques is, that they can deliver RPM and discrete angle positions. Torsional stresses can be derived by measurements with two separated sensors in axial direction or via a model based procedure. Another advantage of the coder based method is, that the sensors are located on the nonrotating part and therefore telemetry or slip rings are not needed. Furthermore the method is suitable for short and longtime monitoring at large Turbogenerators and has a proven robustness.

9.1.2 The Coder Based GE-Sensor with zebra tape

To measure the torsional angular velocity in a Turbogenerator shaft, the coder based GE-Sensor with a zebra tape can for example be used (Figure 41). The physical principle of this sensor type has been described in chapter 6.3. It is important, that the coders have sufficient visible contrast mounted on the shaft. The optical sensor generates an electrical signal depending on the light intensity.

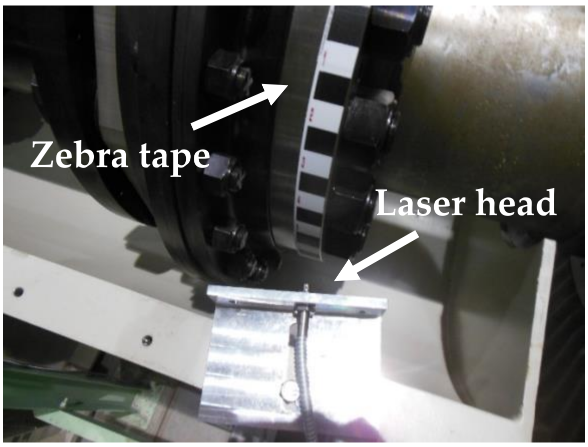


Figure 41 Torsional Vibration Measurements by means of a optical Laser & Zebra Tape

With a large number of markers a very high pulse rate can be achieved and the data resolution can be determined. An advantage of the optical sensor with the zebra tape is the simple instrumentation. and that it works contactless. Furthermore there are no slip rings and no telemetry needed. A permanent installation on very large Turbogenerators is possible. A disadvantage can be the high sensitivity to ambient light. A visual contact between shaft and sensor is required.

9.1.3 Preparation of the GE-Sensor – Corrective Measures

The measurement of torsional vibrations with an optical sensor and a zebra tape may be inaccurate, if lateral vibrations disturb. However, this disturbance can be corrected by using two sensors, which work in opposite directions, one at 0 degree and the other one at 180 degree. It can be shown, that with this sensor arrangement the lateral vibrations are cancelled out.

9.1.4 Measurement Location for the GE Monitoring Technology

The strategy of the GE- Monitoring System consists also of the two parts:

- Measuring of the torsional vibrations by means of an optimal Sensor-Technology and
- Assessing of the torsional vibrations by means of an appropriate Monitoring-Technology

The measurement part with a selection of the sensors has been described in chapters 9.1.1 and 9.1.2.. The question how many sensors should be used and where they should be located has to be discussed by considering both technologies. General Electric combined the two Technologies in the following strategy.

The Turbogenerator manufacturer (e.g. General Electric) has extensive knowledge about the torsional dynamic behavior of the shaft train from the design process. On the one hand this is based on the manufacturers experience and on the other hand on simulation results with a good Theoretical Model – e.g. a detailed Finite Element Model - of the Turbogenerators shaft train including all important components like couplings, turbine blades, retainer rings at the generator, etc. This knowledge from the manufacturer can be very helpful also for the long term operation of the Turbogenerator an the idea came up to use the Theoretical Model as a Digital Twin during the whole life time, continuously running parallel to the real machine. Besides other developments [6], also General Electric developed this idea as part of a **Model Based Monitoring.** As described in 7.4.4 some torsional measurements (stresses and /or angular velocities) are needed from the installed sensors for the Monitoring procedure with a Digital Twin. Contrary to other Digital Twin developments [10], GE uses only one sensor location for the Monitoring process at the end of the shaft line (see Figure 42).

A **critical remark** on this procedure: Based on Modal Analysis, if all torsional natural modes are included in the only sensor signal at the shaft end, the procedure works well. This is usually the case in the lower – subsynchronous - frequency range. However, if a mode shape has a node at the sensor location the procedure will probably not work for this case.

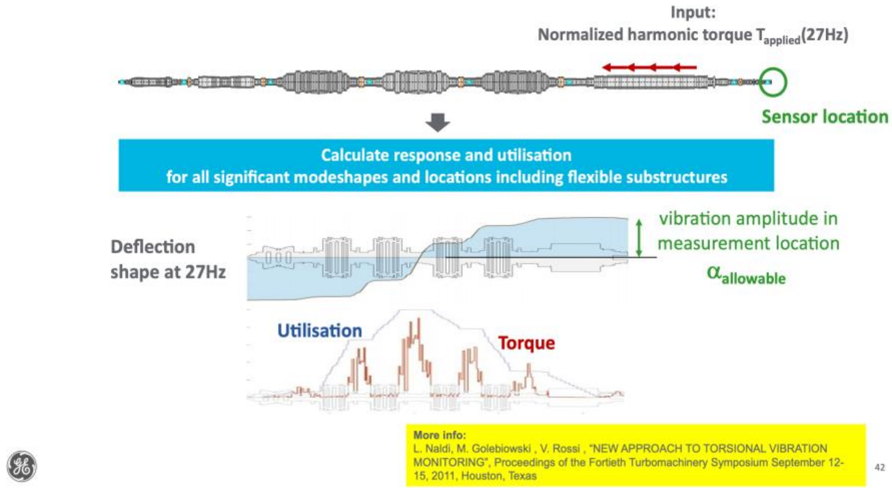


Figure 42 Monitoring of Torsional Vibrations with only one Sensor Location

General Electric (GE) has tested the Monitoring procedure at different Turbogenerator systems. As an example the following application is described in the publication [6].

This example from GE demonstrates the use of model-based measurement approach that has been deployed in a 1300MW nuclear plant. The torsional vibration was measured only in a single plane, utilizing existing speed wheel behind the generator's exciter (see Figure 43). Combination of accurate measurement of angular vibration (high signal to noise ratio in 0-500Hz range; 0.01mrad/sec resolution) with a detailed rotordynamic model allows insight into every critical location along the shaft-line. The spectrum of measured torsional vibration proves the accuracy and validity of the model. The modes of the low-pressure steam turbine blades (coupled with rotor modes) where successfully measured despite the distance between vibrating blades and sensor location (more than 25m).

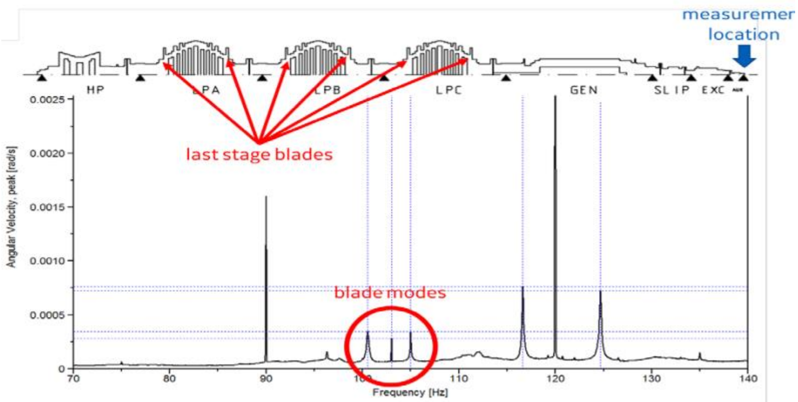


Figure 43 Measured Torsional Vibration Spectrum in the range of 2 x Grid Frequency

9.2 GENERAL ELECTRIC MONITORING TECHNOLOGY

As already described in chapter 9.1.4 the General Electric (GE) Monitoring Technology can be summarized in the Figure 44. It consists of the hardware layer with the electrical and the mechanical system and the Monitoring part with the Digital Twin and the Visual and analytical part (Protocol of the Vibration state)

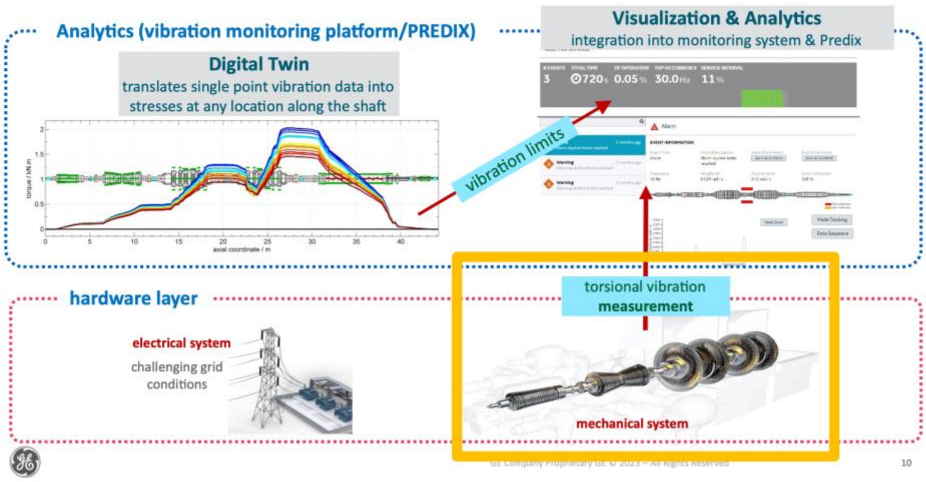


Figure 44 GE Monitoring - Model Based Monitoring with a Digital Twin

The more detailed components of the General Electric Monitoring system are the same as described in chapter 8 for the Siemens Monitoring system. The next chapters 9.2.1 – 9.2.6 are therefore a repetition only.

9.2.1 The Signal Conditioning

As described in chapter 7.3 the Signal Conditioning is applied to the raw sensor signal before a further analysis is performed. This includes amplification, filtering, offset corrections and finally the Analog to Digital conversion (ADC) with sampling and quantization. Signal conditioning prepares the signal that it is noise reduced and in digital form for the following signal processing.

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ASSESSING TORSIONAL VIBRATIONS IN TURBOGENERATORS BY MEANS OF MODERN SENSOR- AND MONITORING-TECHNOLOGIES

The purpose of this Energiforsk project is to present a clear description, how Torsional Vibrations of Turbogenerators can be assessed by means of modern Sensor- and Monitoring Technologies. An important result of the project is, that reliable assessment statements are only possible by combining the three Vibration Analysis tools of Theoretical Modeling, Numerical Analysis and Experimental Analysis. An important part for the Assessment is the Torsinal Vibration Monitoring system. The Monitoring system starts with the data acquisition, the signal conditioning and processing, followed by a feature extraction. The extracted features are then used for fault detection, diagnosis, and prognosis, enabling a reliable assessment of the mechanical integrity and remaining lifetime of the turbogenerator shaft system. Besides existing Monitoring systems new developments based on Digital Twins are also treated in this report.

A new step in energy research

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