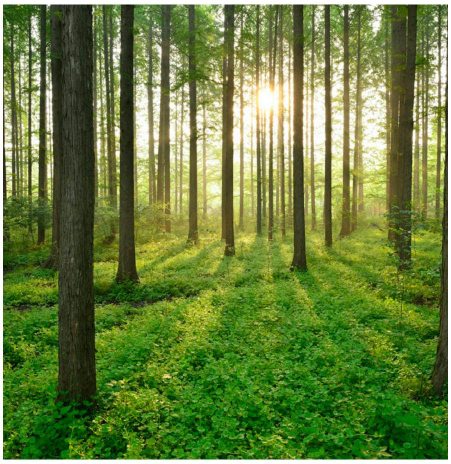
BLADE VIBRATIONS IN TURBINES

REPORT 2025:1137



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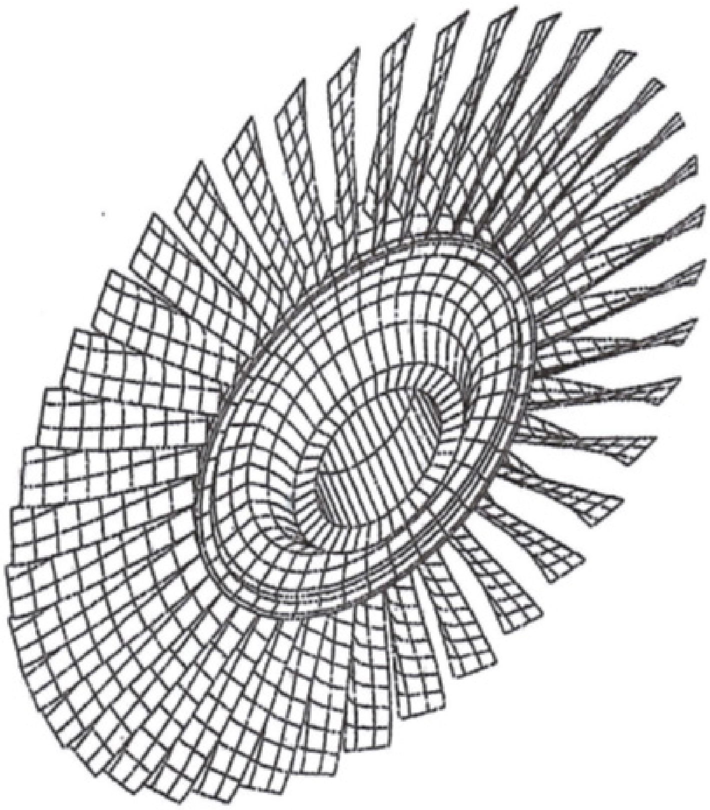

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Blade Vibrations in Turbines

Steam Turbines in Nuclear Power Plants

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Foreword

This report forms the results of a project performed within the Energiforsk Vibrations in Nuclear Applications Program.

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 aim of this study was to analyze the causes of steam turbine blade vibrations and propose effective mitigation strategies. Vibrations can lead to mechanical fatigue and fracture, so understanding the causes of vibrations and finding mitigation measures can both improve turbine performance and lessen the risk of extensive maintenance or even unplanned outages.

The results indicate that specific design modifications and maintenance practices can significantly reduce blade vibrations. Implementing these recommendations will improve the overall efficiency and safety of the nuclear power plant.

The study was carried out by Dr. Rainer Nordmann, Technical University Darmstadt. The study was performed within the Energiforsk Vibrations in Nuclear Applications Program, which is financed by Vattenfall, Uniper, Fortum, TVO, Skellefteå Kraft and Karlstads Energi.

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 a generator are very important components in Nuclear Power Plants. The main task of these units 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 power to mechanical power is performed in the blades of the turbines via steam forces. The effectiveness of this conversion process should be as high as possible in order to achieve good efficiency of the turbines. Therefore the aerodynamic and thermodynamic design of the blades is a very important task in the design process of turbines. Although steam turbine blades may have excellent aerodynamic performance, they also experience problems due to mechanical vibrations, which may lead to material fatigue or even to fracture with a possible stop of operation.

Besides the requirement of a good aerodynamic performance it is therefore also necessary to assure a smooth operation of the turbines with an admissible vibration behavior of the blading system. Blade vibrations is the main topic of this Energiforsk research project.

There are mainly two input sources of blade vibrations in large steam turbines: The excitation of blade vibrations due to time varying steam forces acting on the rotating turbine blades and the excitation of blade vibrations due to the coupling effect between the torsional turbine vibrations and the blade vibrations. As a result of these excitations blade vibrations will occur as a system response. They depend on the above two mentioned excitations, but also on the dynamic characteristics of the blading system itself (natural frequencies, modes, damping values). Besides the two types of excitation blade vibrations can also be caused by an aeroelastic instability. This Fluid Structure Interaction (FSI) is also known as Flutter.

The main purpose of the research project Blade Vibrations is to present a clear physical description, how blade vibrations are generated and which parameters are of importance. It will be shown, which facts the manufacturer has to consider in the design process of a turbomachine blade system from the vibrations point of view. This includes correct assumptions of the blade excitation mechanisms, the selection of well suited blade models, the prediction of the natural frequencies, mode shapes and damping of the blades as well as the vibration responses (displacements and stresses) of the blades due to the excitations. Of importance for the operator of steam turbines is the information, how blade vibrations can be measured during operation (Monitoring) and how they can be evaluated by Vibration Standards. The mitigation of vibrations of blades is also an important factor.

Keywords

Blade Vibrations Modelling, Aerodynamic Excitation, Rotor Blade Interaction, Aeroelastic Instability (Flutter), Campbell Diagram, Monitoring

List of content

1 Introduction

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. For an optimal conversion the aerodynamic and thermodynamic design of the blades is an important task in the design process of the turbines. Although steam turbines may have an excellent aerodynamic performance, they also experience problems due to mechanical vibrations, which may lead to fatigue or even to fracture with a possible stop of the operation and to financial losses. Besides the requirement of a good aerodynamic performance another important design task is therefore to assure a smooth operation of the turbines with an admissible vibration behaviour. Blade vibrations is the main topic of this Energiforsk research project.

1.1 SCOPE OF THE REPORT

Turbine blades are most important elements of steam turbines, and the reliability of the steam turbine depends on the reliability of the blades. Because most of the blade failures are related to blade vibrations, such as resonance induced overstress, high cycle fatigue, corrosion fatigue, etc., it is necessary to exactly predict the vibration characteristics of blades at the design stage and to increase the mechanical strength of the blades. Therefore, it is indispensable for blade designers (manufacturers) and operators as well to understand the various vibration phenomena of blades that occur on steam turbines and to learn the analysis and measurement methods of blade vibration.

The main purpose of this Energiforsk project Blade Vibrations in Turbines is to present a clear physical description, how blade vibrations are generated and by which system parameters they are influenced. In the report it will be shown, which facts are of importance for the turbine-manufacturer, that have to be considered in the design process from the blade vibrations point of view. This includes the following tasks:

- the modelling of the turbine blade system for the simulation of the blade vibrations
- the determination of natural frequencies, mode shapes and modal damping values of the turbine blades
- the determination of amplitudes and frequencies of the aerodynamic forces acting on the blades
- the determination of amplitudes and frequencies of the torsional vibrations, which can excite the vibrations of the blades by Rotor-Blade Interaction effects
- the evaluation of blade vibrations by means of the Campbell diagram

Of importance for the operator of steam turbines is a smooth operation of the steam turbines with a permissible state of blade vibrations. To control the actual vibration state, blade vibrations can directly be measured (Monitoring) during operation. By such measurements indications can be obtained about possible failure mechanisms. Vibration Standards help for an evaluation of the state of blade vibrations. A very important task for a safe operation is also, by which measures blade vibrations can be mitigated.

As a consequence of the above comments regarding blade vibrations this Energiforsk project report will include the following topics:

- Blade Modelling
- Free Blade Vibrations
- Sources of Excitation
- Forced Blade Vibrations due to Steam Flow
- Forced Blade Vibrations due to Rotor-Blade Interaction
- Aeroelastic Instabilities (Flutter)
- Vibration Measurement Techniques
- RPM Restrictions and Avoidance Zones
- Vibration Mitigation Methods
- Failure Mechanisms

1.2 SIGNIFICANCE OF BLADE VIBRATIONS IN THE LAST STAGES OF LPT'S

Figure 1 shows one of the three Low Pressure Turbines (LPT) of the Nuclear Power Plant unit OL3 (Olkiluoto). This LPT is one of the largest LPT's in the world. It has a length of about 12 m, a blade tip diameter of about 6 m and a weight of 300 tons. The rotational frequency of the half-speed OL3-turbogenerator is 1500 rpm (25Hz).

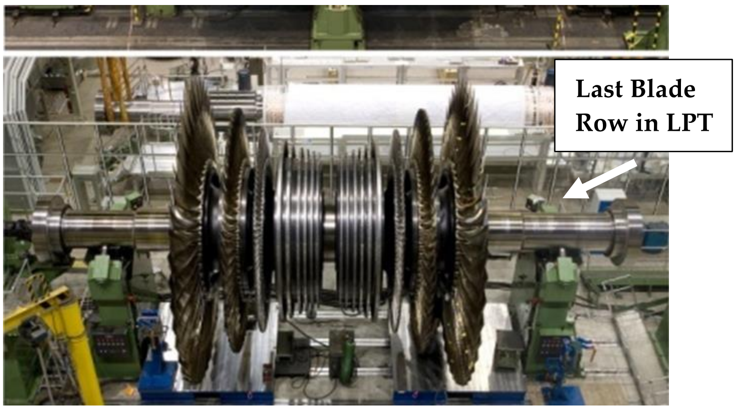


Figure 1 The OL3 Low Pressure Turbine of the Power Plant Olkiluoto

The largest blades of the LPT are in the last blade row. Each of the blades has a length of about 1,5 m and a weight of ca 200 kg.

In general the turbine blades of Low Pressure Steam Turbines of Nuclear Power Plants - particularly the long and heavy last stage blades – must be securely mounted to the rotating shaft to withstand centrifugal forces, steam flow loads and vibrations over long periods. Different mounting techniques have been developed over the years to achieve a reliable mechanical connection between the blade root and the rotating disk or drum of the rotor. The most common attachment methods, the Fir Tree Root (Grantopp) and the Dovetail Root are shown in Figure 2. Besides the shown ones other fixations have been developed, which are not presented in this report.

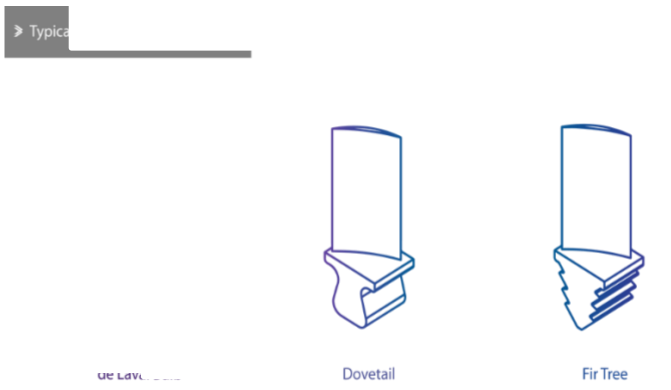


Figure 2 Most common Mounting Methods of Steam Turbine Blades to the Rotor (Source Nordmann)

The blades can be considered as stand-alone beam like models. The large length makes them more susceptible to bending vibrations, torsional vibrations, coupled natural frequencies and resonances. In the models the stiffening effect of the centrifugal forces has to be considered. It means, that the blade natural frequencies are dependent on the rotational frequency. The presence of wet steam contributes to droplet erosion and unsteady flow conditions, which can further increase the vibration risks.

2 Blade Modelling

In large steam turbines used in nuclear power plants, the accurate modelling of turbine blades is essential to predict the vibrational phenomena, that may occur during operation. Figure 3 shows the input output relation for the blade system in a linear form. The response (output) of the blade system in terms of vibrations and/or stresses depends on the excitation (input) acting on the blade system and on the dynamic characteristic of the blade system itself. For an accurate prediction of the vibration behaviour we need both: the excitation of the blade system and the blade model, which represents the dynamic characteristics of the blade system. This chapter describes the methods and steps used for the modelling of turbine blades with concentration of the long blades in the large Low Pressure Turbines.

As shown in figure 3 the model of a blade system can be described by a mass matrix **M**, a damping matrix **D** and a stiffness matrix **K**. The three matrices can be built up, when the design parameters and the boundary conditions of the blade system are known. It has to be considered, that the stiffness matrix is influenced by the stiffening effect due to centrifugal forces acting on the blades.

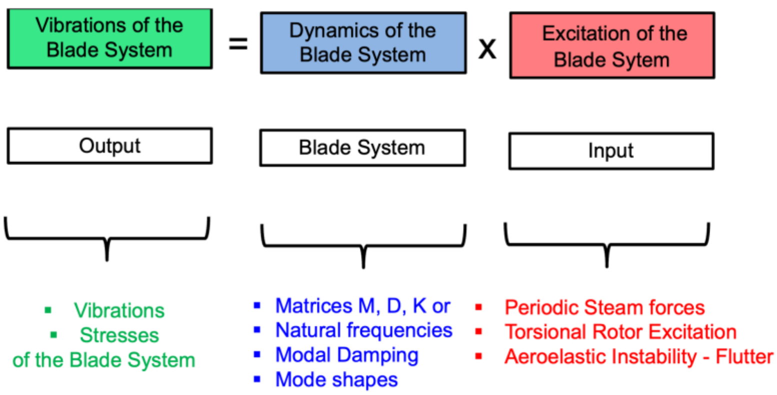


Figure 3 Input–Output Relations: Linear model of a Blade system (Source Nordmann)

If we assume linear relations for the blade system the equations of motion (2.1) can be built up with the matrices **M**, **D** and **K**(Ω). **q**(t) is the vector of the vibration displacements and **F(t)** represents the excitations expressed by forces (see chapters 5, 6, 7)

$$\mathbf{M}\,\ddot{\mathbf{q}}\,(t) + \mathbf{D}\,\dot{\mathbf{q}}\,(t) + \mathbf{K}\,(\boldsymbol{\Omega})\,\mathbf{q}\,(t) = \mathbf{F}\,(t) \qquad (2.1)$$

2.1 GEOMETRICAL AND MATERIAL MODELLING

When modelling the blade system an important step is the accurate definition of the blade's geometry and the material properties. The blade profile typically changes from the root to the tip of the blade in order to optimize the aerodynamic performance and to minimize the stress concentrations. Due to the varying steam flow conditions along the blade length the geometry may be very complex with twisted cross sections. An example is shown in figure 4.

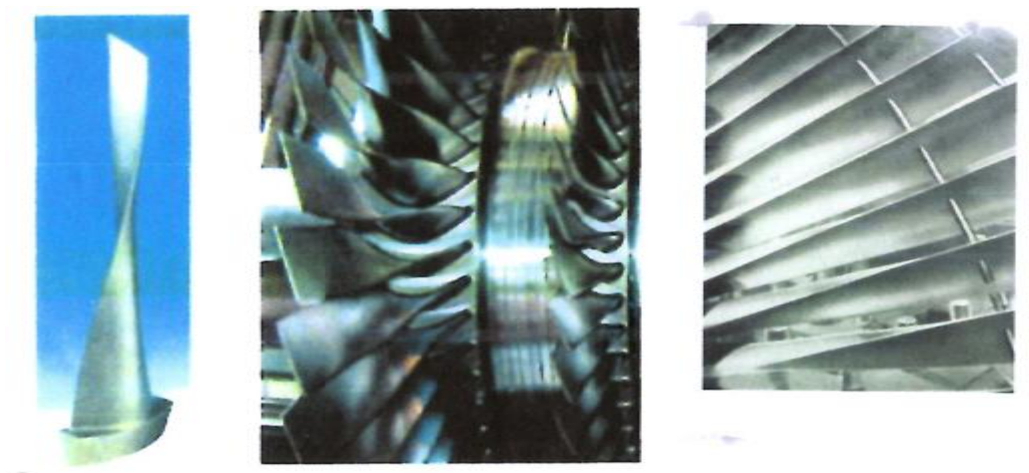


Figure 4 Blade Profile with twisted Cross Sections along the Blade and Lacing Wires

For the blade modelling the 3D-CAD geometry with shapes along several sections is used. If available, features such as shrouds, snubbers or damping wires can be included.

The material properties have to be defined, using Young's modulus, the Poisson's ratio and the material density. Due to the high operating temperatures and pressures temperature dependent material data have to be selected.

2.2 BOUNDARY CONDITIONS

The dynamic behaviour of the blade system is significantly affected by the support conditions. Examples for support conditions are:

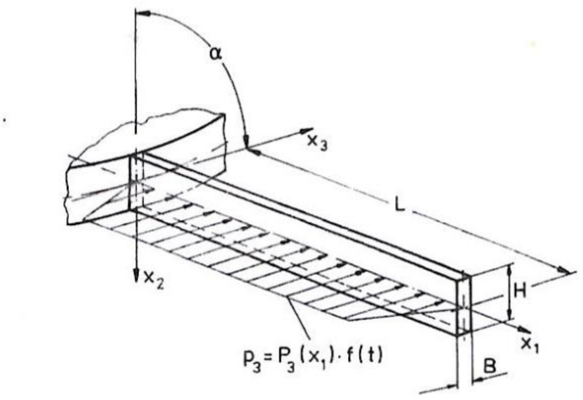
- **Root Fixation**: Depending on the type of blade root (fir-tree, dovetail, etc.), different degrees of freedom are constrained (see figure 2). Contact stiffness and friction can introduce nonlinear behavior.
- **Shrouds**: Circumferential shrouds at the blade tips provide structural coupling between adjacent blades, increasing overall stiffness and introducing tuned mass-damper behavior. In the last stages of the LPT's the blades are usually free without shrouds at the tips.
- **Lacing Wires**: Lacing wires (figure 4) connect free-standing blades, especially in long last-stage blades, adding coupling and damping effects.

- **Centrifugal stiffening**: The operating conditions produce centrifugal forces influencing the natural frequencies.

2.3 SIMPLE BLADE MODELS – CONTINUOUS BEAM OR LUMPED MODELS

For preliminary analyses simplified continuous beam structures (Figure 5, [2]) or lumped parameter models can be effective. The last one represent the blade as a mass-spring-damper system:

- Simplified Continuous Beam Structure (see figures 5 and 8)
- Single degree-of-freedom (SDOF) models approximate bending in one direction. Multi-DOF models can include torsional coupling and/or more bending masses.
- Such models are useful as a first step for estimating resonance frequencies, conducting parametric studies, or developing reduced-order models for system simulations.
- The models use effective mass, stiffness, and damping values derived from detailed structural analyses or from experiments.



Blade length L [m]
Blade cross section A [m^2]
Geometrical Moment of Inertia I [m^4]

Young's Modulus E [N/m^2]
Poisson's Ratio ν [-]
Material Density ϱ [kg/m^3]

Figure 5 Simplified Blade Model as Continuous Beam with Tangential Load (Source [2])

2.4 MODELS BASED ON FINITE ELEMENTS

Nowadays Finite Element (FE) Analysis is the industry standard for detailed blade modeling and Analysis. A Finite Element Procedure usually consists of the following parts:

It starts with the **Mesh Generation**: 3D solid or shell elements are used depending on complexity and computational cost. By means of **Modal Analysis** natural frequencies and mode shapes, including bending, torsion, and coupled modes can be determined, followed by a **Harmonic and/or transient analysis**: Simulates forced response due to steam excitation, unbalance, or shaft vibration.

Nonlinear Contact Modeling captures the friction effects at root, shroud, or lacing wire interfaces.

Modern FEA tools enable validation against experimental data through modal testing and can include temperature-dependent properties and centrifugal loading. For the calculation of the natural frequencies of one stage the FE model of figure 6 can be used. With this simplified model the geometric similarity of each blade with dovetail, disk and rotor segment can be used to perform the calculation via the assumption of a cyclic symmetry.

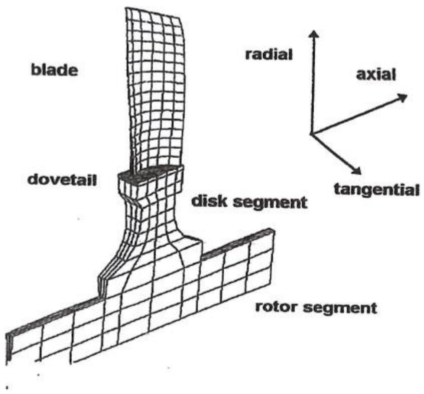


Figure 6 FE Model of a Rotor Blade System with cyclic symmetry (Source [3])

2.5 MODELS WITH ROTOR-BLADE-INTERACTION RBI

When the blade system is excited via the torsional vibrations of the shaft train by a base excitation at the root of the blades we have to consider the Rotor Blade Interaction RBI [3]. A model for this case is much more complicated, because it should include the torsional dynamics of the shaft train coupled with the dynamics of the blades system. Figure 7 demonstrates the relations for the Rotor Blade Interaction. The shaft train of the turbogenerator can be excited to torsional vibrations by the Air Gap Torque due to the Electro-Mechanical Interaction (EMI) in the generator. The torsional vibrations of the shaft train can then excite the blade system via the base excitation at the blade root. The severity of the resulting blade vibrations depends on the dynamics of the two systems of the shaft train and the blades (see chapter 6 for more details).

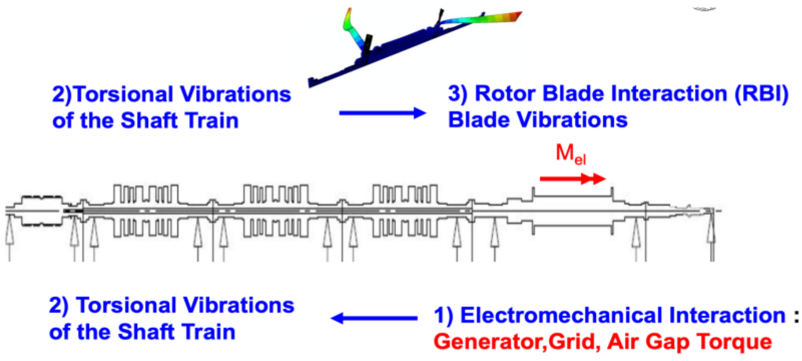


Figure 7 Blade Vibrations due to Rotor Blade Interaction RBI (Source Nordmann)

2.6 MODELS WITH FLUID STRUCTURE INTERACTION FSI

In this modeling stage, the interaction between steam flow and structural dynamics is captured through **Fluid-Structure Interaction (FSI)** models:

- **One-way FSI**: Aerodynamic forces from steady or unsteady CFD are applied as loads to the structural model.
- **Two-way FSI**: Coupled simulations allow for feedback between structural deformation and fluid pressure fields.
- **Modal aerodynamic damping** and **unsteady pressure distributions** are computed to assess flutter stability.

FSI analysis is crucial for the design of long last-stage blades where aeroelastic instabilities can lead to catastrophic failures. Accurate modeling of the steam path, pressure fluctuations, and flow separation is required.

3 Free Blade Vibrations – Modal Parameters

To get more information about the dynamic characteristics of a turbine blade system the first step is usually to determine the natural frequencies (imaginary parts of the complex eigenvalues) and the corresponding mode shapes (eigenvectors). If we again assume, that the blade vibrations can be described by linear equations of motion (eq. 2.1), we can derive an eigenvalue problem from the homogenous equations. The solutions of this eigenvalue problem are the natural frequencies (eigenvalues) of the blade system and the corresponding vibration mode shapes (eigenvectors). These quantities together with the modal damping are called the modal parameters.

3.1 EIGENVALUE-ANALYSIS: NATURAL FREQUENCIES AND MODE SHAPES

In chapter 2 we have introduced the general linear equations of motion for blade systems. We assume, that the type of this linear equations is applicable to all of the presented different models, described in the previous chapters 2.3 to 2.6. If the excitation forces **F**(t) in the equation system (2.1) are set to zero, we obtain the homogenous equations of motion (3.1) for the natural vibrations of the blade system. Due to the fact, that the blade 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.

$$\mathbf{M}\,\ddot{\mathbf{q}}\,(t) + \mathbf{D}\,\dot{\mathbf{q}}\,(t) + \mathbf{K}(\Omega)\,\mathbf{q}\,(t) = \mathbf{0} \qquad (3.1)$$

$$\mathbf{M}\,\ddot{\mathbf{q}}\,(t) + \mathbf{K}(\Omega)\,\mathbf{q}\,(t) = \mathbf{0} \qquad (3.2)$$

With the mathematical approach (3.3) for the natural or free vibrations (motion of the blade system) and the derivative (3.4) for the acceleration we obtain the Eigenvalue problem (3.5).

$$\mathbf{q}(t) = \hat{\mathbf{q}} \sin \omega\, t \qquad (3.3)$$

$$\ddot{\mathbf{q}}(\mathbf{t}) = -\,\omega^2\, \hat{\mathbf{q}} \sin \omega\, t \qquad (3.4)$$

$$(\mathbf{K}(\Omega) - \omega^2 \cdot \mathbf{M}) \cdot \hat{\mathbf{q}} = \mathbf{0} \qquad (3.5)$$

The solution of the Eigenvalue problem leads finally to the natural frequencies ω_j and the corresponding eigenvectors or mode shapes $\hat{\mathbf{q}}_j$. The natural frequencies, expressed in Hz are $f_j = \omega_j / 2\,\pi$.

3.2 DEPENDENCY OF THE MODAL PARAMETERS ON DIFFERENT FACTORS

3.2.1 Dependency on Geometrical and Material Data

As can be seen from the eigenvalue problem (3.5) we need the mass matrix **M** and the stiffness matrix **K**(Ω) in order to determine the modal parameters. As already described in chapter 2.1 the two matrices depend on the blade's geometry and on the material properties. If we consider for example in more detail a simple beam like blade structure (figure 5) , we would need to know

the geometrical data:

- blade length L [m]
- blade cross section A [m^2]
- geometrical moment of inertia I [m^4]

and the temperature dependent material data:

- the Young's modulus E [N/m^2]
- the Poisson's ratio ν [-]
- the material density ϱ [kg/m^3]

In more extended models (e.g. coupling between bending and torsion, or 3D FE Models) additional geometrical and material data will be needed.

3.2.2 Dependency on Boundary Conditions

Besides the geometrical and material data of the blade structure itself, the influence of the boundary conditions on the modal parameters has to be considered as an important factor. In chapter 2.2 different boundary conditions have been described. The Last Stage Blades of the LP Turbines usually have a Fixation (fir-tree dovetail) at the Root, while the blade tip is free. In the modelling process it has to be decided, whether the Root Fixation can be considered as a rigid or a flexible boundary with contact stiffness and friction. .

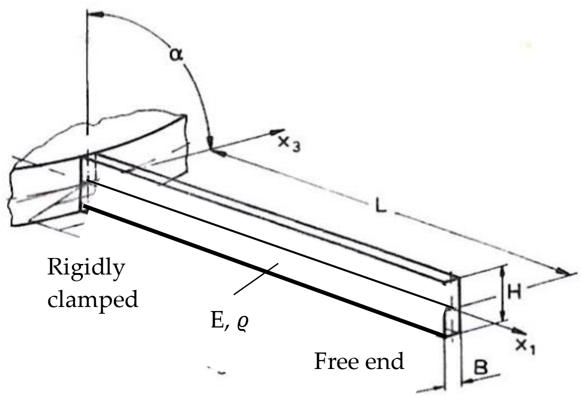
3.2.3 Dependency on Centrifugal Forces

When a blade rotates, centrifugal forces act along its length. In a deflected state of a beam element the centrifugal force acting on this element has a restoring effect. It supports the restoring stiffness force of the beam element itself. This phenomenon is known as centrifugal stiffening**.** As a result, the natural frequencies of the blade increase with the rotational speed. This must be considered when analyzing blade dynamics in turbines, as it impacts both the natural frequencies and the vibration response of the blades.

3.3 MODAL PARAMETERS OF BEAM LIKE BLADE STRUCTURES

3.3.1 Modal Parameters for a simple Blade Structure without Rotation

To become more familiar with the modal parameters: we determine the first natural frequencies f_j [Hz] and the corresponding mode shapes (eigenvectors) $\hat{\mathbf{q}}_j$ of the simple beam like structure (figure 8). For the untwisted blade we investigate the lateral (bending) vibrations in the tangential direction x_3 only. The material of the blade model is steel. The beam has the length L, the cross section A and the geometrical moment of inertia I (Parameter values, see figure 8).



Geometrical Data **Material Data**

Blade length	L = 1 m	Young's Modulus	$E = 2{,}1e{+}11\ N/m^2$
Blade width	B = 0,028 m	Poisson's Ratio	$\nu = 0{,}3$
Blade height	H = 0,300 m	Material Density	$\varrho = 7850\ kg/m^3$
Blade cross section	$A = 0{,}0084\ m^2$		
Geom. Moment of Inertia	$I = 0{,}55e\text{-}6\ m^4$		

Figure 8 Untwisted Beam like Blade Model with constant cross section (Source [2])

At the blade root rigid clamping is assumed, the blade tip is considered to be free. We follow the theory of continuum mechanics for the considered beam structure with continuous mass and stiffness distribution. In this special case the mass and stiffness distribution is constant. Without going in more detail the exact solution for the natural frequencies f_j is as follows:

$$f_j = \omega_j / 2\pi = k_j\,(1/L^2)\,\mathrm{sqrt}\,(E\,I/\varrho\,A) = k_j\,c\,i / L^2 \quad j = 1, 2, 3, \ldots\ldots\infty \quad (3.6)$$

k_j natural frequency factor, determined from continuums mechanics

$c = \mathrm{sqrt}\,(E/\varrho)$ speed of sound for steel: 5172 m/s (3.7)

$i = \mathrm{sqrt}\,(I/A)$ inertia radius of the cross section: 0,0081 m (3.8)

The continuum mechanics also yields the results for the mode shapes (eigenvectors or eigenfunctions) based on the approach with sin, cos, sinh and cosh functions.

The results for the three first natural frequencies are shown in Table I

Number j	1	2	3
Nat. Frequency Factor k_j	$k_1 = 0{,}56$	$k_2 = 3{,}51$	$k_3 = 9{,}82$
Nat. Frequency f_j [Hz]	$f_1 = 23{,}4$ Hz	$f_2 = 146{,}6$ Hz	$f_3 = 410{,}8$ Hz

Table I Natural frequencies of the simple Beam like Structure of Figure 8

The corresponding mode shapes (eigenvectors) are presented in figure 9. These mode shapes are the bending deformations, when the shaft is vibrating in one of its natural frequencies.

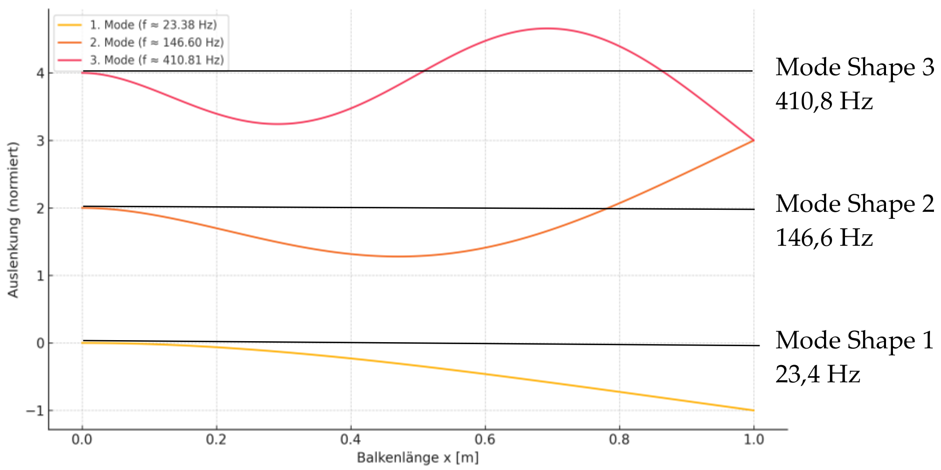
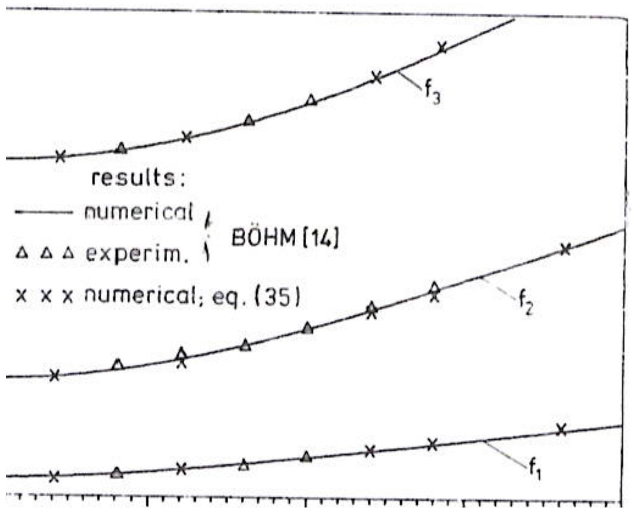


Figure 9 Mode Shapes of the Simple Beam like Blade Structure (Source [2])

3.3.2 Natural Frequencies for a Beam like Test Rig Blade with Rotation

To show the effect of Centrifugal Stiffening due to rotation Irretier [2] has investigated the lateral natural frequencies for a similar beam like test blade, as treated in chapter 3.3.1, in dependence of the rotational frequency. Figure 10 presents qualitatively the three lowest measured natural frequencies versus the rotational frequencies and compares with corresponding calculated natural frequencies.

Lateral Natural Frequencies ot the Blade



Rotational Frequency

Figure 10 Natural Frequencies of a Test Rig Blade versus Rotational Frequency, Source [2]

3.4 MODAL PARAMETERS OF FE MODELS WITH ROTOR BLADE INTERACTION - RBI

When the blade system is excited via the torsional vibrations of the shaft train by a base excitation at the root of the blades we have to consider the effect of Rotor Blade Interaction - RBI. A model for this case including the shaft, the disk and the blades is much more complicated, because it has to include the torsional dynamics of the shaft train coupled with the dynamics of the blade system (see chapter 2.5). A very complex calculation model is not useful. Figure 11 demonstrates for example a complete FE-model for one LPT blade stage only, including the disk and all blades. The number of degrees of freedom is very high. A more simplified model to determine the RBI effect will be described in 3.4.1.

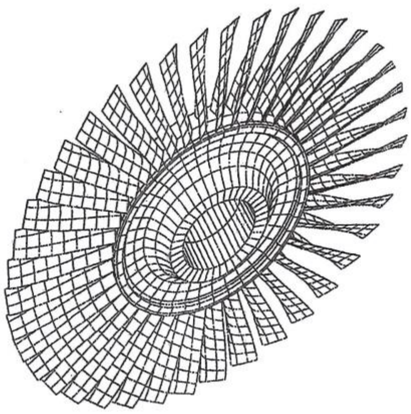


Figure 11 Complex FE Model for one LPT Blade Stage with Disk and Blades (Source [3])

3.4.1 Modal Parameters of a FE Model for Cyclic Symmetric Calculations

The simplified model shown in figure 12 exploits the geometric similarity of each blade with the dovetail and the disk and rotor segments in order to investigate the rotor blade interaction via the cyclic symmetry. The geometry of the disk-blade system is rotationally symmetrical (cyclic symmetric). Regarding the zero crossings the mode shapes of a full LPT stage (figure 11) can systematically be subdivided into modes with node diameters N and node circles M. Without discussing this in detail, the node diameter N = 0 corresponds to mode shapes in which all blades vibrate synchronously in the tangential (circumferential) direction and partly in axial direction (umbrella mode) . These mode shapes correlate with the mode shapes of the shaft torsion, resulting in a common natural frequency.

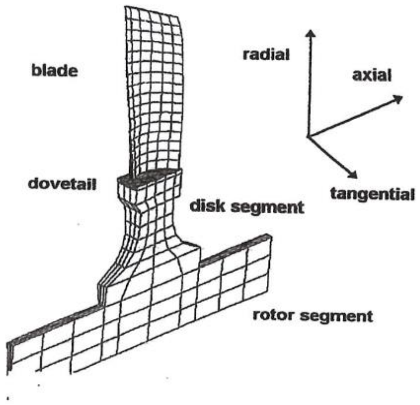


Figure 12 FE-Model Blade-Rotor-System for a Cyclic Symmetric Calculation (Source [3])

The mode shapes for the complete blade system of one blade stage (figure 11) have their main vibration directions in tangential and axial direction. For the coupling with the torsional motion of the shaft only those blade mode shapes have to be considered, which have a synchronous motion in tangential direction (N = 0). The vibration behaviour of all blades of one stage can then be concentrated to one single degree of freedom system. In figure 13 four of the mode shapes with the node diameter N = 0 are presented. The percentage number in each mode presents the contribution of this mode with respect to the tangential motion.

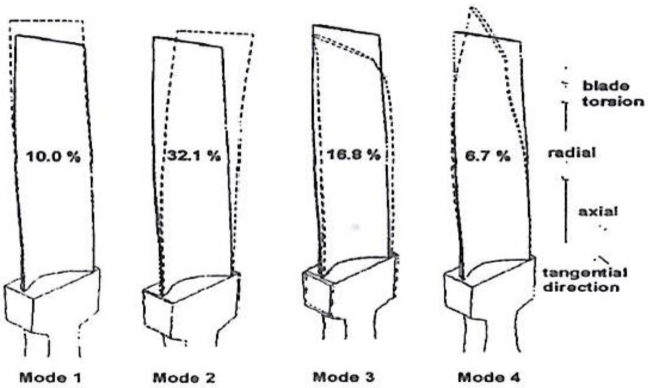


Figure 13 Mode Shapes of Blade-Disk-Shaft System with Node Diam. N = 0 (Source [3)

3.4.2 Modal Parameters of the OL3 Turbogenerator FE Model

As another example of Modal Parameters, resulting from Rotor Blade Interaction, we select the Turbogenerator OL3 of the Finish Nuclear Power Plant Olkiluoto. As was already shown in figure 1 especially the six Last Blade Rows in the three Low Pressure Turbines LPT are susceptible to RBI vibrations. The FE model used for the determination of the modal parameters of the OL3 shaft line with one HPT, three LPT's and the generator is shown in figure 14. Besides the complete shaft line the six Last Blade Rows were modelled and coupled to the shaft line in a similar way as was presented in figure 12. This is not shown in detail in figure 14.

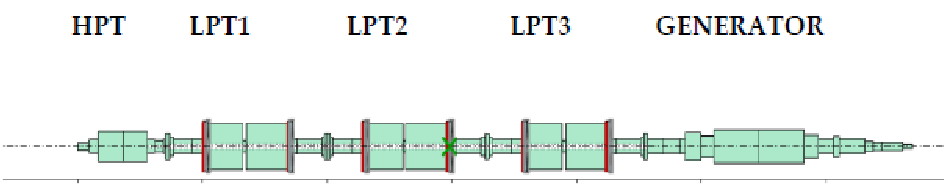


Figure 14 FE-Model of the OL3-Turbogenerator with Rotor-Blade-Interaction (Source 4])

The calculated frequency spectrum with natural frequencies up to 100 Hz is presented in figure 15. The measured frequency spectrum during operation is also shown for comparison. The last one has been determined by measurements of the shaft torsional vibrations with installed sensors at specified locations between the three LPT's.

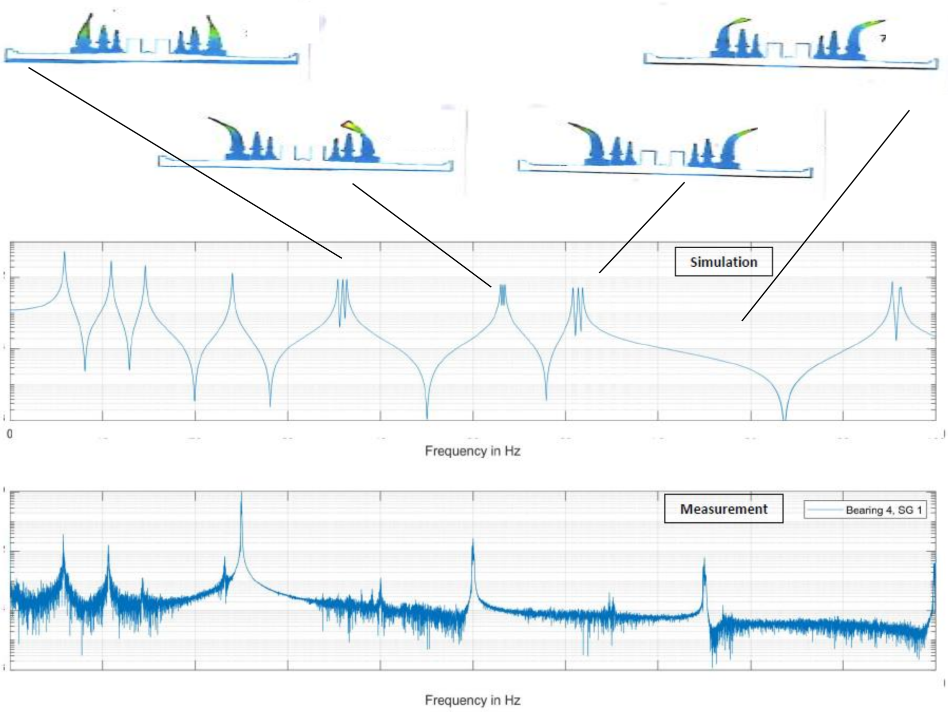


Figure 15 Frequency Spectrum Nat. Frequencies of the OL3 Turbogenerator (Source [4])

There is a quite good correlation between calculations and measurements. This is in particular true for the lower natural frequencies up to 30 Hz. In the higher frequency range the measured results are much influenced by measurement noise. However, the natural frequency peaks can still be detected and correlated to the calculations.

Calculations are very helpful to find out those natural frequencies, where Rotor Blade Interactions (RBI) appear. Figure 15 gives a clear picture about the type of blade vibrations with different mode shapes at different natural frequencies. Closely adjacent natural frequencies appear in groupings of three. This can be explained by the fact, that there are small differences in the dynamic behaviour of the three LPT's, which are not completely similar in the design [4].

3.5 THE CAMPBELL DIAGRAMM

In a Campbell diagram two types of frequencies (y-axis) are plotted

- the system's natural frequencies (modal parameters) and the
- the excitation frequencies of the system

versus the rotational frequency (x-axis) of rotating machinery, like turbines, pumps, compressors etc. The Campbell diagram is a very helpful tool in Rotor Dynamics and general Vibration Analysis to study the interaction between the rotational speed dependent natural frequency curves and the excitation frequencies, which can often be presented as straight line harmonics of the rotational speed. Intersections between these curves or lines indicate potential resonance frequencies, where the rotating system may experience high vibrations.

The Campbell diagram helps engineers to identify critical speeds (resonances) and to avoid dangerous operating conditions that could lead to strong vibrations and mechanical failure. It is a valuable tool during the design and testing of rotating systems to ensure a safe and reliable operation. A typical Campbell diagram with four speed dependent natural frequencies and excitation lines up to the 10th order is shown in figure 16 [5].

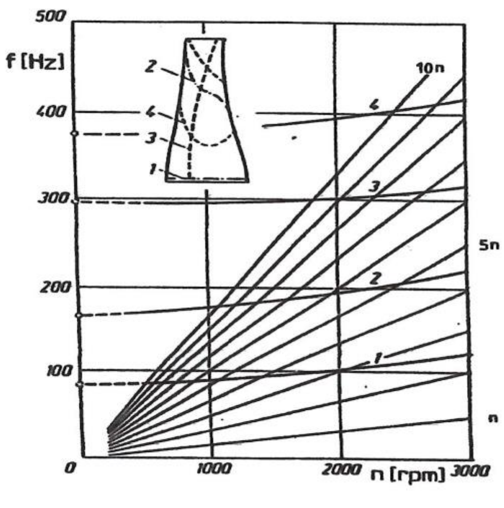


Figure 16 Campbell Diagram of a Shell like Blade of a Final LPT Stage (Source [5])

4 Static and Dynamic Forces acting on Blades

There are Static Forces as well as Dynamic Forces acting on the Blade system. They lead to static as well as dynamic stresses. The manufacturer has to investigate these stresses at critical locations in order to estimate the blades life time.

4.1 SUPERPOSITION OF STATIC AND DYNAMIC FORCES AND STRESSES

Figure 17 presents an overview of steady and non-steady forces, acting on a blade system, which lead to static (steady) as well as dynamic stresses. By knowledge of these stresses the blades life time can be estimated.

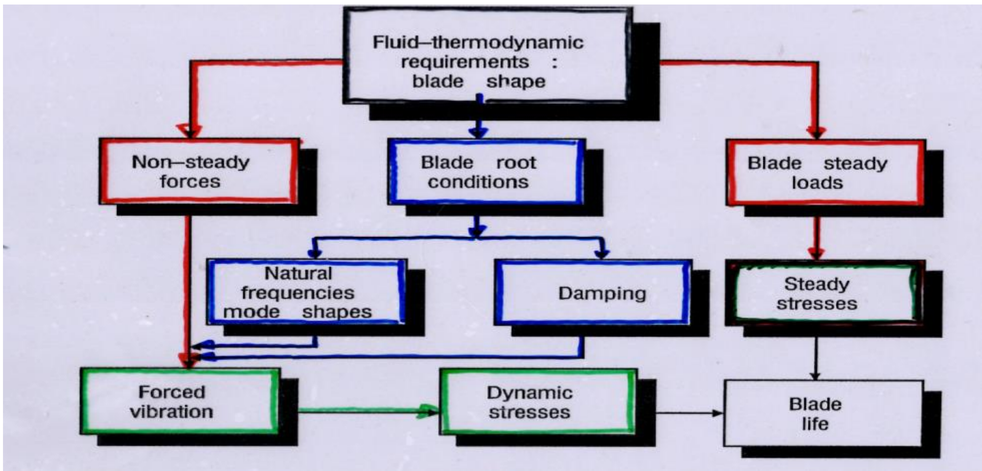


Figure 17 Static and Dynamic Forces acting on a Blade System (Source Nordmann)

When calculating the dynamic stresses it has to be considered, that they are on the one side influenced on the time dependent (dynamic) forces but also on the dynamic characteristics (figure 3), and on the boundary conditions (e.g. blade root) as well.

4.2 THE STATIC CENTRIFUGAL FORCES AND STRESSES

Due to the fact, that the static loads and the static stresses due to centrifugal forces determine the main part of the overall stresses, this load part influences the size and the change of the cross sections of the blades in radial direction. As demonstrated in figure 18 the highest static reaction forces due to the centrifugal forces are in the root of the blade system. The changing radial loads along the radial direction have therefore an influence on the size of the cross sections, which is also influenced by the fluid thermodynamic requirements.

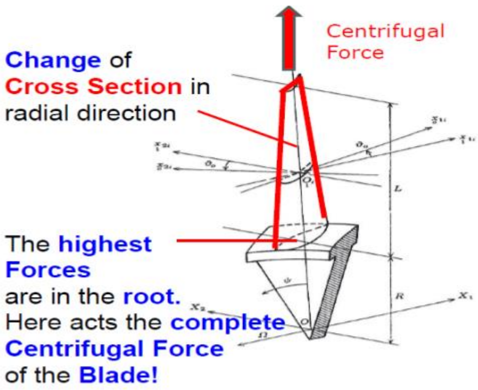


Figure 18 Static Centrifugal Forces acting on a Blade System (Source [2])

4.3 THE DYNAMIC FORCES AND STRESSES OF THE BLADE SYSTEM

Besides the static loads (Centrifugal Forces) there are mainly three dynamic sources for vibrations of the turbine blades. These are two time dependent excitations due to forces or base motions and a Fluid-Blade-Interaction FBI, which is an aerodynamic instability (Flutter):

- 1) the time varying periodic aerodynamic steam forces act directly on the blades, the Nozzle Excitation and the Excitation due to Partial Admission are typical examples for this type (chapter 5). This is also an interaction between the fluid (steam) and the blade, but in this case the excitation forces are time dependent and belong to the force term on the right hand side of the equations of motion 2.1 in chapter 2.
- 2) the Excitation due to the interaction between the torsional vibrations of the shaft line and the blade vibrations: Rotor-Blade-Interaction RBI (chapter 6). The torsional vibrations are the direct excitations of the blade system. But they have their origin mainly from excitations in the generator. Examples for Air Gap Torques due to electro-mechanical interactions are Short circuit Torques, Air Gap Torques due to unsymmetric loaded grid and Air Gap Torques due to SSR). They are all time dependent right hand terms in the overall equations of motion.
- 3) the Aerodynamic Instability is a Fluid-Blade-Interaction (FBI), in which the blade is excited by steam forces, which depend on the vibration state of the blade (displacements and velocities) and on the aerodynamic state of the surrounding fluid (steam velocity and pressure).

In Figure 19 the input-output-relations are shown again with the three different sources of excitation. It is important to note again, that the dynamic characteristics of the blade system (modal parameters, chapter 3) have also an important influence on the vibrations of the blade system (output).

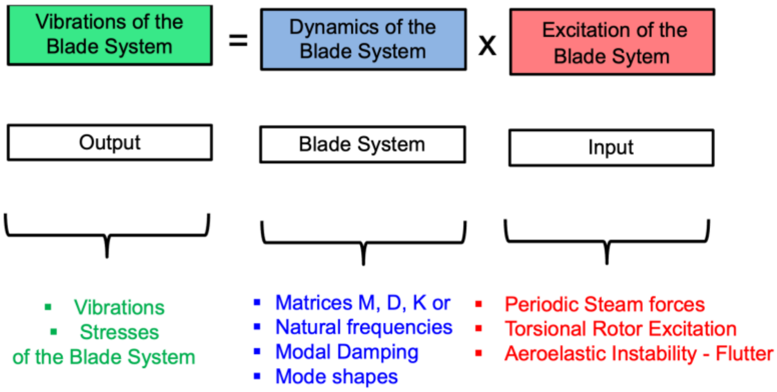


Figure 19 Input-Output Relations of the Blade System with Terms of Excitation (Source Nordmann)

4.4 OPERATIONAL INFLUENCES (LOAD CHANGES, TRANSIENTS)

At off-design conditions, flow separation can occur in the blade passages, leading to special effects, that will not be treated in this report.

5 Excitation due to Periodic Steam Forces

In large steam turbines used in nuclear power plants, forced vibrations are a primary concern due to their ability to excite blade resonances and to cause fatigue. These vibrations originate from deterministic, periodic aerodynamic steam forces acting on the blades, which arise from known flow phenomena such as nozzle passing, partial admission, non-uniform inlet conditions and others. The dynamic steam forces can be expressed by time dependent periodic functions, where the time period is usually related to the angular rotational frequency Ω of the turbine shaft. In this chapter we consider the two cases of the Nozzle Passing Frequency Excitation and the Excitation due to Partial Admission.

5.1 NOZZLE PASSING FREQUENCY EXCITATION

One of the most common sources of vibrations in LP turbine blades is the steam excitation with the Nozzle Passing Frequency (NPF). This frequency corresponds to the periodic interaction between the rotating blades and the stationary vanes or nozzles. The dynamic steam load on a blade is not constant with respect to the rotational angle, but varies during one rotation of a blade. The reason for this is,

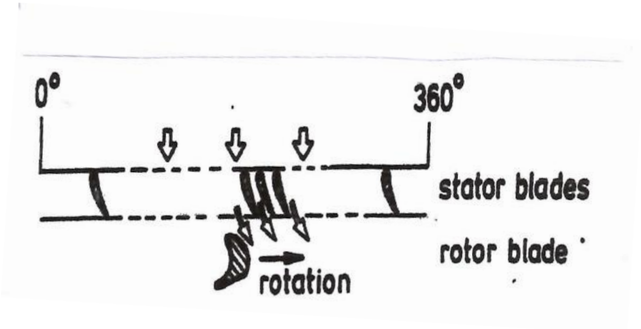


Figure 20 Steam Excitation of a Rotating Blade with the Nozzle Passing Frequency (Source [5])

that the rotor blades run behind a row of still standing stator blades, which are fixed in the casing of the machine (see figure 20). Thus, the rotor blade is loaded by fluid forces in tangential and axial direction. Due to the permanent repetition of the rotation the load on the blades is periodic and can be described by a Fourier Series. In the expression of the Fourier series the Angular Excitation Frequencies or Nozzle Passing Frequencies are given by

$$\Omega_k = k\, z\, \Omega\, ; \;\; k = 1, 2,3, \ldots ,$$

where z is the number of the stator blades, Ω is the angular speed of the turbine shaft and k are integer numbers.

In a Campbell diagram the Nozzle-Passing Frequencies Ω_k (k =1, 2, 3…) can be shown as dotted lines for the different integer numbers k (ordinate). They are linear dependent on the angular rotational frequency Ω (abscissa) of the turbine shaft. The blade natural frequencies ω_1, ω_2, ω_3 .. are shown as solid curves. Due to the centrifugal forces they are also a function on the angular rotational frequency Ω of the turbine shaft.

At each intersection of a natural frequency curve ω_j with an Excitation line Ω_k a resonance is defined (see red star as an example in figure 21). The Campbell diagram is well suited to find such resonances, however it is not possible, to determine the amplitude of the blade vibrations in this diagram. For this task a numerical solution would be needed, based on the equations of motion 2.1 (see chapter 2). As input data for such a simulation the amplitudes of the steam forces have to be known, besides the system parameters of the blade system.

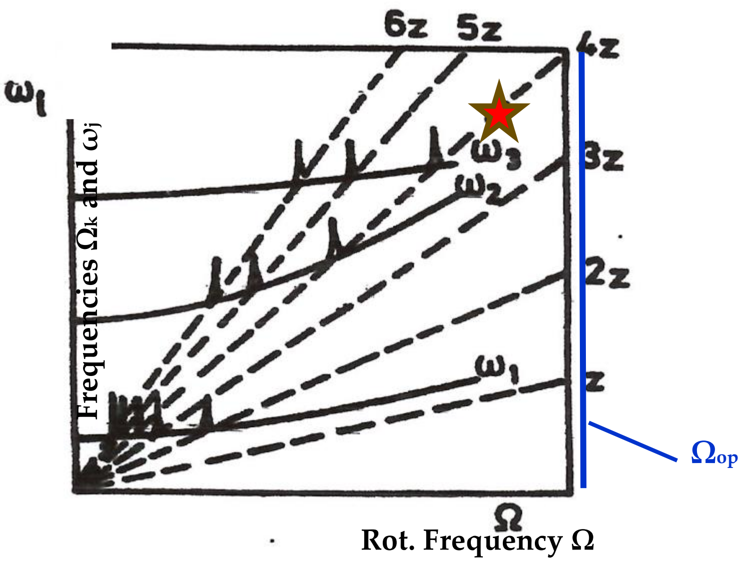


Figure 21 Campbell Diagram with Blade Natural Frequencies and Excitation Lines (Source [5])

As can be seen in the example of figure 21 several intersection or resonances are possible. One design task for the turbine manufacturer should be to avoid such intersections, especially at the operating speed $\mathbf{\Omega_{op}}$ (see blue line in figure 21). Blades are typically tuned to avoid resonance within the operational speed range. Campbell diagrams are used during the design phase to identify and mitigate such crossings. However, manufacturing tolerances and flow variations can still lead to unintended resonances, especially at part-load or during transient conditions. In general for long LPT blades—especially for those in the last stages—it is a critical design consideration due to their slender, flexible geometry and high susceptibility to vibratory excitation.

Figure 22 shows the natural frequencies and the Excitation lines for an example of a turbine blade, where the coupling between the bending and torsional vibrations is taken into account. For the first three coupled modes various resonance points in the rotor speed range between 400 and 2000 rpm are visible. The figure also shows the cross sections at different distances from the blade root [5] .

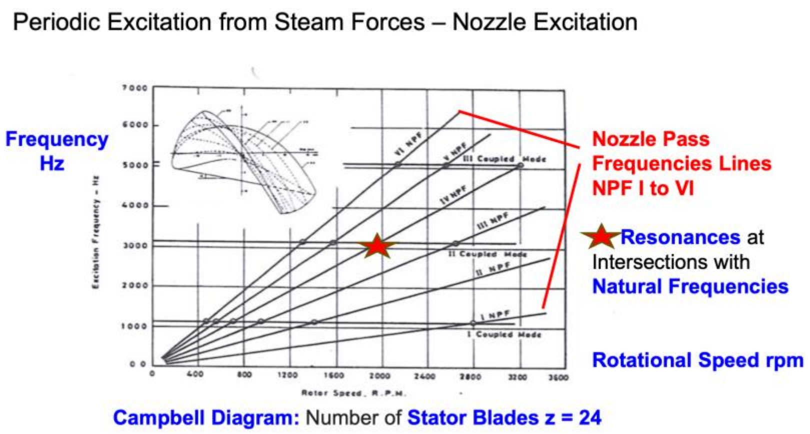


Figure 22 Natural Frequencies and Excitation Lines of a Turbine Blade (Source [5])

5.2 EXCITATION DUE TO PARTIAL ADMISSION

In nuclear turbines, part-load operation can be achieved by partial admission, i.e., by opening only a subset of steam control valves. This leads to asymmetric steam admission into the turbine inlet, introducing different fluid flow effects, which also influence the vibration behavior of the blades. We concentrate here on the effect, that the character of the excitation lines in the Campbell diagram will change in case of a partial admission. Figure 23 shows a nonuniform load during one revolution due to partial admission. The force acting on the blade consists of the partial admission part and the high frequency nozzle wake part. In one or more sectors around the turbine the rotor blades are loaded by fluid forces, while the other sectors remain unloaded. In the example the part of partial admission blade force is approximated by a rectangular shape.

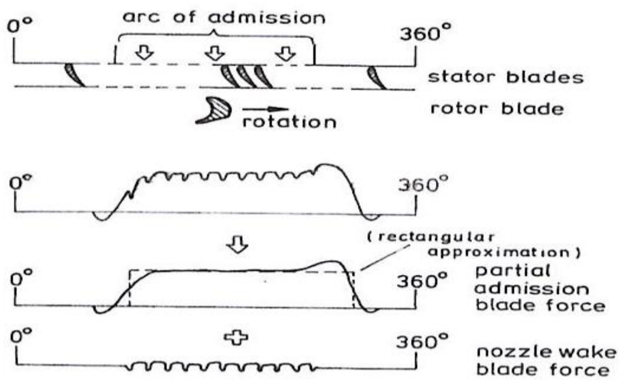


Figure 23 General Character of Blade Load due to Partial Admission (Source[2,5])

A special partial admission case is considered in figure 24, where in two sectors of 90° a blade is subjected to the load, while the blade is unloaded when passing the other two sectors. During one blade revolution the load appears two times for the blade. The base excitation frequency is therefore two times of the angular frequency Ω of the turbine shaft. And due to the rectangular shape of the load the higher Harmonics are 3x, 5x, 7x…of the base frequency. The corresponding excitation lines are shown in the Campbell diagram together with the blade natural frequencies

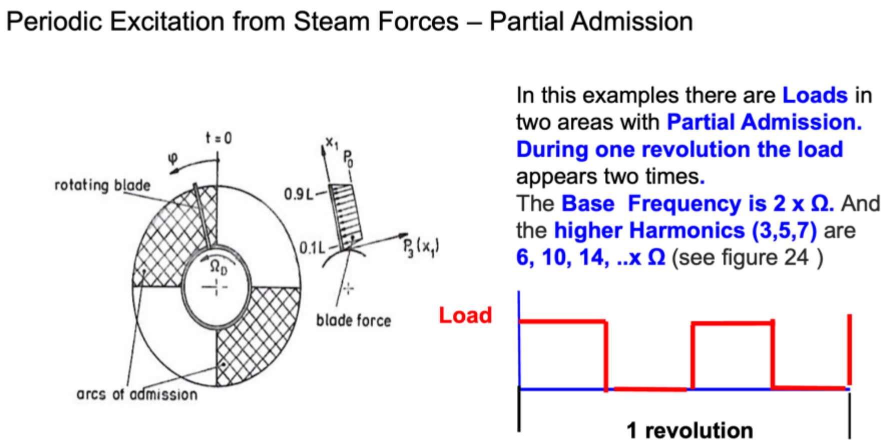


Figure 24 Rotating Blade and Load due to two Areas of Partial Admission (Source [2,5])

The corresponding excitation lines are shown in the Campbell diagram together with the blade natural frequencies. Compared to the excitation lines of the nozzle wake part the partial admission excitation frequencies are much lower.

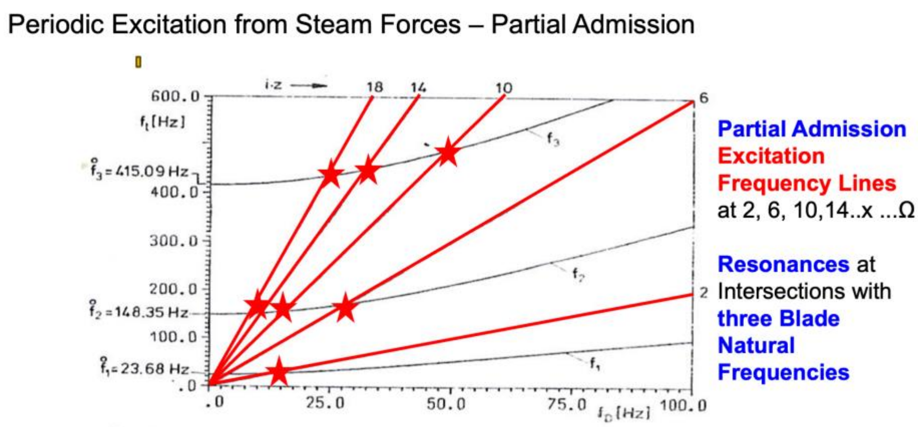


Figure 25 Campbell Diagram: Nat. Frequencies and Excitation Lines for Partial Admission (Source [2, 5])

6 Excitation due to Rotor Blade Interaction

As mentioned before, in large Low Pressure Turbines (LPT) utilized in nuclear power plants, the phenomenon of blade vibrations is a critical factor, influencing the mechanical integrity and operational reliability of the turbine. Besides the Excitation due to Periodic Steam Forces (chapter 5) the forced vibrations due to different dynamic interactions within the turbine structure are of great importance. Among those, forced vibrations due to Rotor Blade Interactions are particularly relevant in the final stages of the LP turbines, where long, slender blades operate under complex loading and boundary conditions.

6.1 ROTOR BLADE INTERACTIONS – GENERAL REMARKS

Rotor blade Interactions refer to the coupling between the blades and the rotor system, where dynamic forces and motions —originating from shaft torsion or blade-root interface dynamics induce vibrations of the blades. Unlike direct aerodynamic excitation (chapter 5), these interactions involve the transfer of mechanical energy through structural paths, often accompanied by nonlinear effects such as friction, contact forces, and relative motion. These vibrations can originally be caused by transient operating conditions (such as start-up or load rejection) and by electro-mechanical interactions in the generator including grid-induced torsional oscillations. Internal resonances between blade and rotor modes usually amplify these vibrations. The complexity of Rotor Blade Interactions in the LPT's of nuclear power plants is further influenced by additional factors:

- Longer blades, which are more flexible and susceptible to dynamic interactions.
- Relative low modal damping of the blade system
- Large spans and mass, leading to lower blade natural frequencies that may overlap with torsional natural frequencies of the rotor or with harmonics of the electrical grid.

The comprehensive understanding of forced vibrations due to Rotor Blade Interaction is important not only for the design optimization of the blade system but also for condition monitoring and diagnostics. Excessive blade vibration can lead to fatigue cracking, fretting at root interfaces, or in extreme cases to catastrophic blade failures. As a requirement, robust modelling strategies combining finite element analysis (FEA), substructure and reduction methods and multi-body dynamics should be employed to assess the interaction mechanisms, to identify resonance risks and to develop mitigation strategies.

6.2 ROTOR TORSIONAL VIBRATIONS AND BLADE VIBRATIONS - RBI

One of the most critical Rotor Blade Interactions (RBI) in Turbogenerators appears when torsional vibrations of the shaft train are coupled with blade vibrations. In Figure 26 the Rotor Blade Interaction is demonstrated for the case, when torsional vibrations of the shaft train are caused by Air Gap Torques in the generator due to electro-mechanical interactions.

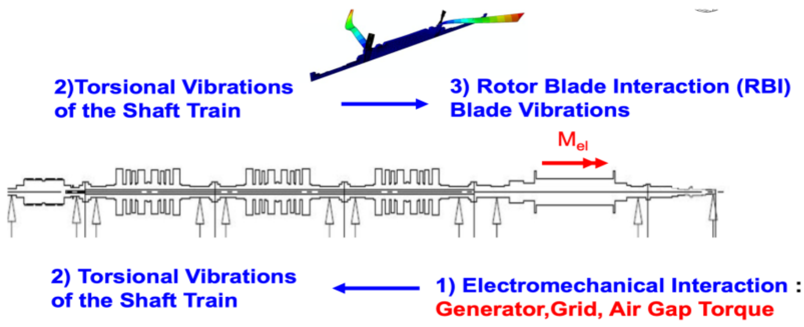


Figure 26 Blade Vibrations due to Rotor Blade Interaction RBI (Source Nordmann)

The first part in the excitation sequence is, that torsional vibrations along the shaft train of the turbogenerator are excited by the time dependent Air Gap Torques. If the torsional vibrations also appear for example at the locations of the last stage blades - we assume no node at this location - the blade roots will be excited by the torsional or tangential motion of the shaft. This base excitation is equal for all blades in this stage, which means that all blades are excited synchronously and will have the same vibration. If a full model with the shaft train and the coupled blade system is available, the calculation of the blade vibrations can be performed consistently in one step from the Air Gap Torque to the blades. This is the procedure, that Turbine manufacturer will apply during the design, because they know all design data, including the shaft train and the blade system and can built up a complete model.

In an early Siemens paper [3] from 1997 (SIRM conference) the problem has been described as follows:

The Excitability of blade vibrations due to torsional turbine vibrations has world-wide caused several blade damages. In these cases the excitation appears in a system resonance, in which due to the low damping large vibration amplitudes are excited. The original excitation is coming from the electrical system via the Air Gap Torque. Typical electro-mechanical excitation cases are:

- Unbalanced electrical loads
- Short circuits
- Sub Synchronous Resonances (SSR)

In the design process of large turbogenerators, it is common practice to express the air gap torques due to electrical disturbances by fixed formulas instead of using the more complicated formula with a coupling of electrical and mechanical effects. This is a conservative approach for an estimation of the vibration behavior. But experience shows that it is a useful approximation, which has often been confirmed by experimental results. Usually, the following electrical fault cases are considered during the design, which have the following excitation frequencies:

2-phase short circuit 1x and 2x grid frequency 3-
phase short circuit 1x grid frequency Faulty
synchronization 1x grid frequency Unbalanced
electrical load 2x grid frequency

In the following sub-chapter 6.3 we consider the simplified formulas of the air gap torques for the two cases: 2-phase short circuit and an unbalanced electrical load.

6.3 AIR GAP TORQUE FOR TWO ELECTRICAL EXCITATION CASES

For the determination of the blade vibrations due to Rotor-Blade-Interaction the first step is to define the excitation, that means the Air Gap Torques. For the two examples of the 2-phase short circuit and the unbalanced electrical load we present the simplified formulas as mentioned before. Figure 27 shows the electrical air gap torque for a not decaying 2-phase short circuit. Excitation frequencies are the single grid frequency Ω and the double grid frequency 2 Ω. The formula contains also some generator characteristics. In reality the air gap torque is decaying, which improves the vibration response [6].

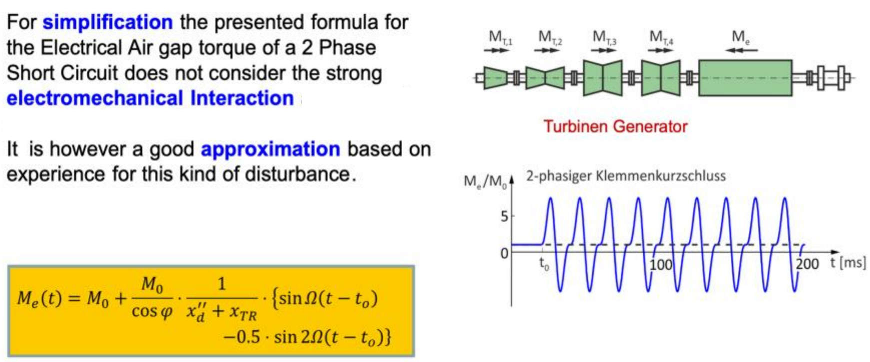


Figure 27 Assessment of Torsional Vibrations : 2-Phase Short Circuit (Source [4])

Due to unbalanced electrical loads in the electrical grid, unsymmetry will occur in the 3-phase electrical system. By electrical derivatives it can be shown, that due to this fact, the nominal air gap torque is superimposed by a pulsating torque with double grid frequency. In a 50 Hz grid system the air gap torque excites the shaft train with a frequency of 100 Hz, which is a harmonic excitation (Figure 28).

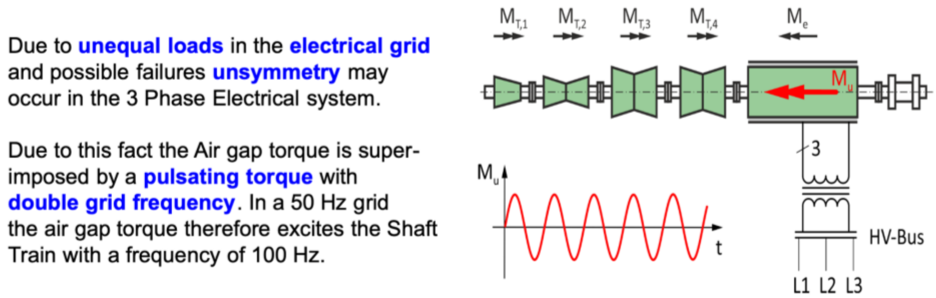


Figure 28 Assessment of Torsional Vibrations : Unbalanced Electrical Load (Source [4])

6.4 TORSIONAL VIBRATION RESPONSE AND BLADE VIBRATIONS

The two presented Air Gap Torques in the previous chapter can be considered as the time dependent excitations F(t) of equation 6.1 (see also chapter 2).

$$\mathbf{M}\,\ddot{\mathbf{q}}\,(\mathrm{t}) + \mathbf{D}\,\dot{\mathbf{q}}\,(\mathrm{t}) + \mathbf{K}\,(\boldsymbol{\Omega})\,\mathbf{q}\,(\mathrm{t}) = \mathbf{F}\,(\mathrm{t}) \qquad (6.1)$$

The equation system (6.1) (similar to 2.1) itself can either present a complete FE-model with many degrees of freedom for the shaft train (Torsional Vibrations) as well as for the blade system (Torsional and bending) in an accurate modelling with many degrees of freedom . The alternative model is also a complete FE-model for the shaft train, but with a more reduced model structure for the blade system. Such a model has for example been derived in the Energiforsk report: “Development of a Digital Twin for Torsional Vibrations of Turbogenerators.” [4] This model can be used with less effort, to obtain even good results for the torsional vibrations of the shaft train, particularly at the roots of the last stage blades.

6.5 TORSIONAL VIBRATION RESPONSE AT THE ROOT OF A LS BLADE

To determine the torsional vibration response at the root of a Last Stage Blade system the turbogenerator of the OL3 unit (Olkiluoto) was used again. The calculation was performed with the alternative model consisting of the complete shaft train model and the more reduced blade system model (see chapter 6.4). As a special type of Excitation F(t) the Air Gap Torque was assumed as harmonic Unit Amplitude Torque in the frequency range of operation. The amplitudes of the resulting torsional response at the last stage blade of the LPT3 are shown in figure 29. This solution was found by means of the Modal Analysis technique, which concentrates on the part solutions for the different natural frequencies and mode shapes. From the more general result for the whole frequency range specific excitation cases can be selected. For example the result for the frequency at the red star is shown in figure 29.

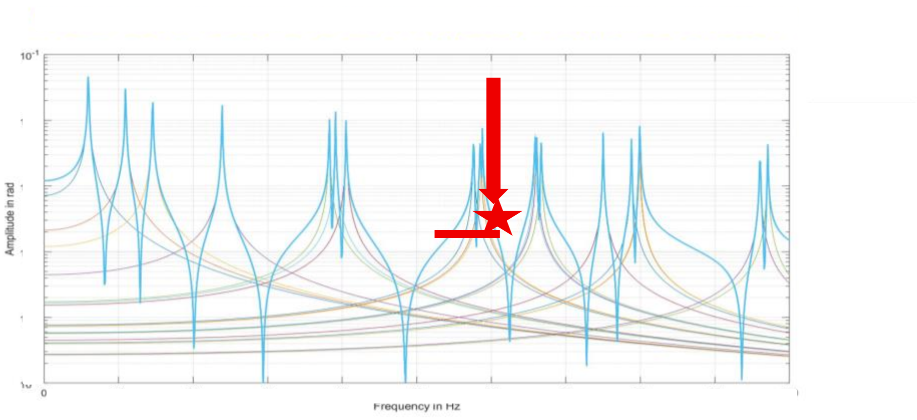


Figure 29 Response Amplitudes due to Harmonic Unit Air Gap Torque (Source [4])

The result of figure 29 can be used as base excitations for the blade roots in a more detailed blade model in order to obtain more accurate results (displacements and stresses) within the blade structure.

6.6 ISO-STANDARDS FOR AN ASSESSMENT OF TORSIONAL VIBRATIONS

Several international standards provide guidelines and methodologies pertinent to the assessment and management of vibrations in steam turbines, particularly also for the case of torsional shaft vibrations and blade vibrations. Examples for Standards are:

- **ISO 22266-1:2022**: This standard offers guidelines for evaluating torsional natural frequencies and component strength under normal operating conditions for coupled shaft trains, including long elastic rotor blades in steam turbine generator sets. It specifically addresses torsional responses at grid and twice grid frequencies due to electrical excitation .

The standards serve as essential references for engineers and designers to ensure that turbine systems operate within an acceptable vibration range, thereby enhancing safety and performance.

Example: The diagram in Figure 30 shows two torsional natural frequencies of a shaft train as a function of the rotational frequency of the shaft (Centrifugal Stiffening). To avoid a strong unbalanced electrical load excitation of the shaft train with the double grid frequency of 100 Hz the torsional natural frequencies at operational speed should have a safety margin to the excitation frequency (2 x grid frequency). More details can be found in ISO 22266.

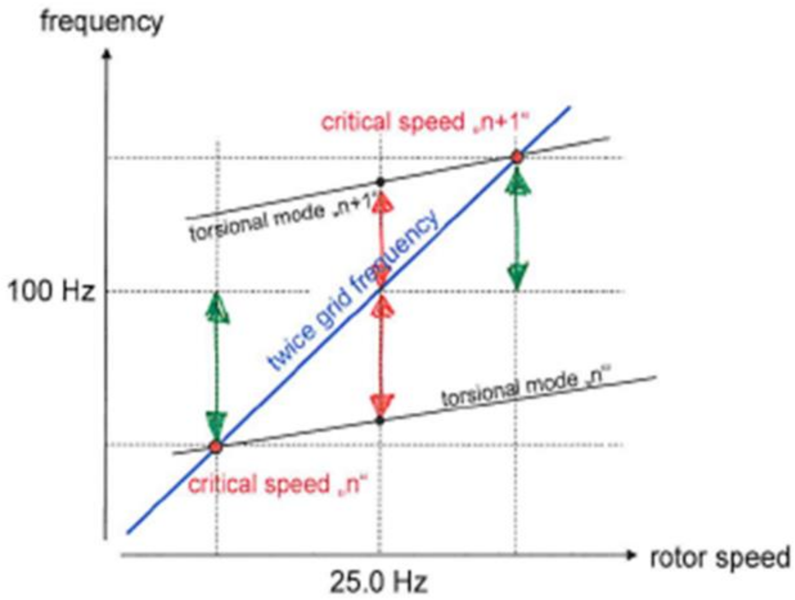


Figure 30 Safety margin or avoidance zone for Torsional Vibrations ISO 22266-1:2022: (Source Nordmann)

7 The Aeroelastic Instability (Flutter)

The Aeroelastic Instability, also called Flutter, of a Blade System is a self-excited dynamic instability resulting from the interaction between unsteady aerodynamic fluid forces and the dynamics of the blade structure. Unlike forced vibrations, flutter can occur without any external periodic forcing. When the energy fed into the blade by aerodynamic forces exceeds the structural damping energy, the vibration amplitude can grow rapidly. This type of instability can lead to catastrophic failures if not properly predicted and controlled. The key distinction lies in the Energy Source: In forced vibrations, Energy comes from External Flow Features. In Flutter, the system extracts energy from the steady flow via feedback between aerodynamic forces and blade motion

7.1 ONSET CRITERIA, STABILITY MARGINS AND DESIGN STRATEGIES

Flutter onset is governed by the coupling of aerodynamic, inertial, and elastic structural blade forces. Several parameters influence whether flutter will occur, including flow velocity, blade mode shapes and damping characteristics. A widely used criterion for flutter onset is the work-per-cycle approach: if the net aerodynamic work done on a vibrating blade over one oscillation cycle is positive, the system is unstable and flutter occurs. To quantify this, engineers evaluate the aerodynamic damping of specific vibrational modes. This damping can be positive (system is stable) or negative (system is unstable). The stability margin is often defined as the percentage of critical damping required to prevent flutter under worst-case conditions.

Stability assessments typically consider

- Campbell diagrams to identify resonances between blade modes and aerodynamic excitations and
- Operating envelope evaluations, where steam conditions (pressure, temperature, mass flow) are varied to find the most critical flutter-prone scenarios.

The design strategies to maintain sufficient margins include:

- Intentional frequency detuning
- Use of shrouds or snubbers to alter mode shapes and stiffness
- Adding damping via materials or mechanical features

7.2 CFD AND FEA COUPLED SIMULATIONS

Fluid Structure Interaction (FSI) analysis is crucial for the design of long last-stage blades where aeroelastic instabilities can lead to catastrophic failures. Accurate modeling of the steam path, the pressure fluctuations and a flow separation is required. To accurately predict and mitigate flutter, engineers increasingly rely on coupled simulations combining Computational Fluid Dynamics (CFD) for the fluid and Finite Element Analysis (FEA) for the blade. This Fluid-Structure Interaction (FSI) approach allows for:

- resolving unsteady pressure fields around the vibrating blades.
- calculating aerodynamic damping as a function of the mode shapes and the frequencies.
- capturing nonlinear effects in both the fluid and structural domains.

To investigate the Flutter problem the typical workflow for a turbine manufacturer includes:

- Modal Analysis (FEA): Extract the natural frequencies and mode shapes of the blade system
- Unsteady CFD simulations: Apply those mode shapes as imposed motions in the fluid solver and compute resulting unsteady aerodynamic forces

These unsteady aerodynamic forces are then mapped back onto the structural model, enabling the calculation of the resulting dynamic response of the blades. By iterating between the CFD and FEA solvers, engineers can capture the coupled behavior of the system and identify critical conditions where flutter may occur. This integrated workflow enhances the reliability of flutter predictions and supports design optimizations aimed at improving blade stability and lifespan.

- Energy Extraction: Integrate the aerodynamic work done over one oscillation cycle to assess damping
- Stability Analysis: Determine whether the system is stable or flutter prone under specific conditions

7.3 A STEAM TURBINE FLUTTER TEST CASE FROM KTH

In a Steam Turbine Flutter Test Case from KTH Stockholm the influence of the tip clearance on the aeroelastic stability of a last-stage turbine blade was investigated. The study found that the global aerodynamic damping increased with the tip gap height up to a certain point, after which it decreased, indicating a complex relationship between tip clearance and flutter stability.

7.4 LP-TURBINE LSB VIBRATION MEASUREMENTS

During inspections of a retrofitted LP Turbine rubbing marks were discovered on the tips of some of the last stage blades (figure 31). An investigation was carried out, which concluded that the rubbing could be attributed to high vibration amplitude of the blades. Therefore it was recommended to perform in service Blade Vibration Measurements (BVM) in order to identify the operating conditions at which the suspected high vibration amplitudes occur (see also chapter 8).

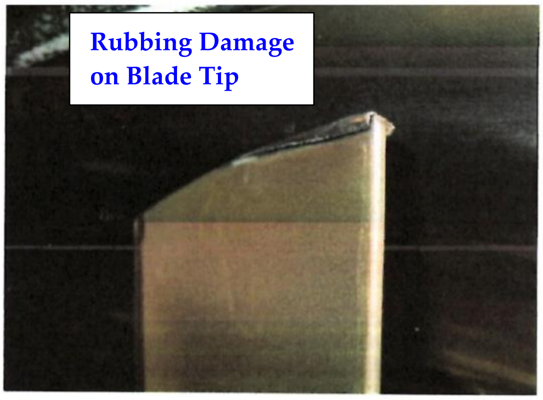


Figure 31 Rubbing Damage on Last Stage Turbine Blades of a LP Turbine

For this task blade tip vibration sensors were installed in the housing of the Turbine (see figure 32).

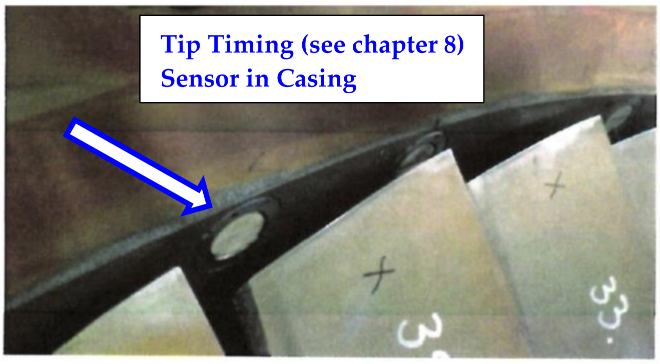


Figure 32 Eddy Current Tip Timing Sensors for Vibration Measurements (see chapter 8)

The blade vibration measurements at operation with maximum load have shown that high vibrations occurred during a run down with partial vacuum breaking. Partial vacuum breaking refers to the intentional introduction of a small amount of air into the condenser and turbine exhaust to reduce the vacuum. This is typically done to prevent excessively low pressure, which can lead to problems like blade cavitation or reduced turbine efficiency.

In this example partial vacuum breaking had the negative effect of high blade vibrations. Figure 33 shows the increase of vibrations during the run down together with the changing (increasing) vacuum pressure. Due to the partial vacuum breaking the fluid flow conditions change, which may have an influence on the blade vibrations. It can therefore be assumed, that a Fluid Structure Interaction causes the strong increase of vibrations in the range of 41 to 42 Hz, which may be an Aeroelastic Instability. However in the report it was not directly defined as an Aeroelastic Instability.

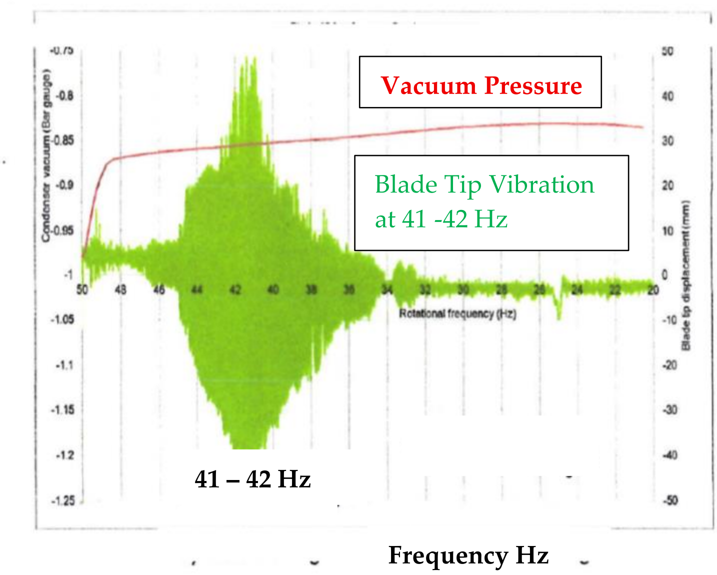


Figure 33 Variation of Blade Vibrations & Vacuum Pressure at Run Down

As a result of the investigation, it was recommended that partial vacuum breaking during run down should be de-activated on the relevant Forsmark turbines.

It is possible, that an Aeroelastic Instability occurs, when the partial vacuum breaking is activated during run down.

8 Blade Vibration Measurement Techniques

Effective monitoring and analysis of blade vibrations are critical for ensuring the reliability and safety of low-pressure (LP) steam turbines, particularly in nuclear power plants. A combination of advanced measurement techniques and signal processing methods is employed to detect, characterize, and diagnose blade vibrational phenomena, including potentially dangerous aeroelastic instabilities, like flutter.

8.1 EXPERIMENTAL ANALYSIS: A TOOL TO SOLVE VIBRATION PROBLEMS

Figure 34 shows 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 behavior 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, using sensors and devices for signal processing. By different combinations of the three tools the following tasks can be performed: Simulation, Validation and Identification (figure 34). By the continuous measurement the Monitoring of blade vibrations can be performed. And by the combination of Experimental Analysis and Modelling possible blade failures can be found by Diagnosis.

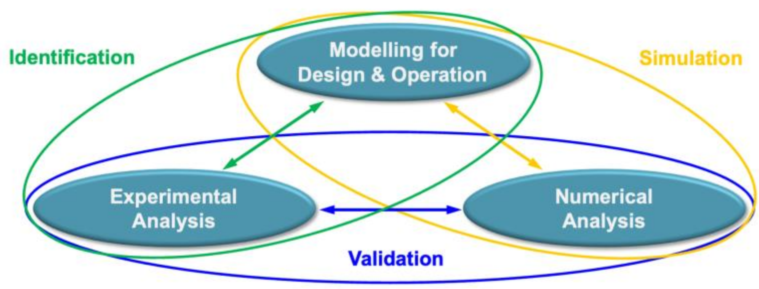


Figure 34 : Tools and Tasks to solve Vibration Problems in Mechanical Engineering, (Source [7])

In the next subchapters we will concentrate on the tool of Experimental Analysis, particularly for blade vibrations, with a description of the applied methods, sensors and signal processing.

8.2 BLADE TIP TIMING SYSTEMS (BTT)

Blade Tip Timing (BTT) is a non-intrusive technique widely used for monitoring blade vibrations in rotating machinery. It uses casing-mounted sensors—typically optical or eddy current sensors, to detect the passing time of blade tips. By comparing the actual arrival times with expected times based on the rotational speed, one can infer blade deflections and derive vibration characteristics. The advantages of the method are:

- non-contact and suitable for high-temperature environments.
- the method can monitor multiple blades simultaneously.
- Effective in detecting asynchronous and mistuned blade behavior.

However, the method has some limitations:

- it requires precise calibration.
- the accuracy can be affected by rotational jitter and noise

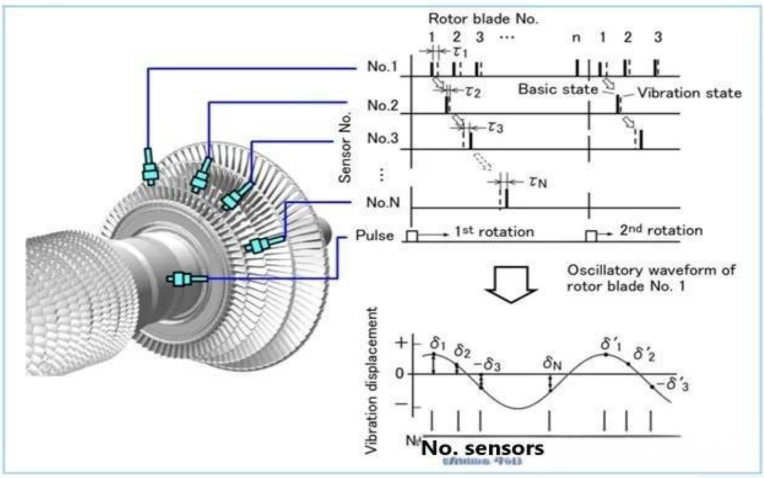


Figure 35 Blade Tip Timing System for Monitoring of Blade Vibrations (Source [6])

The principle of the tip timing measurement to detect blade vibrations is shown in figure 35. The pulse signal given is the blade tip timing signal output by the sensor. The rising edge of the timing signal records the moment when the blade reaches the sensor.

The angle reference synchronization sensor is placed at the root of the blade. When the root of each blade is swept, a pulse signal is output. Since the root of the blade does not vibrate, it can be used as a synchronization signal for the blade tip vibration signal to minimize the measurement error caused by the uneven speed of the rotating equipment.

In figure 36 the principle of the blade tip timing is presented in another sketch with the instrumentation of five sensors. In the right picture the real flexible rotor measurement is compared with the theoretical rigid rotor estimation for one revolution [6].

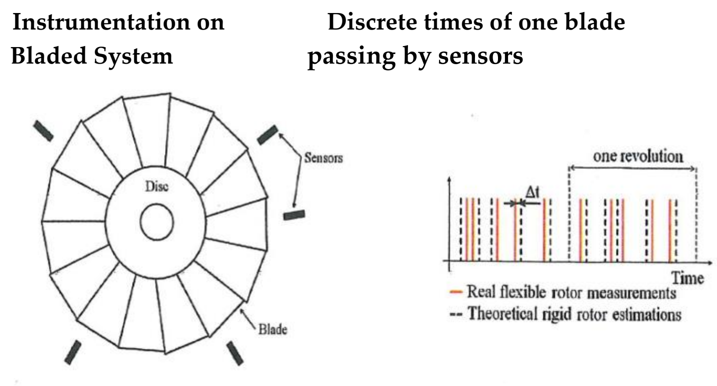


Figure 36 Tip Timing Instrumentation on a Bladed Disc with five Sensors (Source [6])

8.3 SENSOR TYPES: EDDY CURRENT, STRAIN GAUGES, OPTICAL

Eddy Currents are noncontact sensors, that detect the displacement of metallic turbine blades by measuring changes in magnetic fields (see figure 32).

Advantages:

- ➢ Suitable for high temperature and high pressure environment
- ➢ Non-intrusive and durable
- ➢ Capable of high frequency response

Disadvantages:

- ➢ Sensitive to electromagnetic interference.
- ➢ Requires precise alignment and calibration
- ➢ Measures only relative displacement near sensor location

Strain gauges bonded to blade surfaces provide direct measurements of dynamic stress and strain during operation. When combined with telemetry systems or slip rings, they can transmit real-time data from rotating blades (see figure 37)

Advantages:

- ➢ High accuracy for local stress measurements.
- ➢ Useful for validation of numerical simulations (mode shapes, stresses)

Disadvantages:

- ➢ Intrusive; requires physical installation on blades.
- ➢ Limited lifespan in high-temperature or corrosive environments.

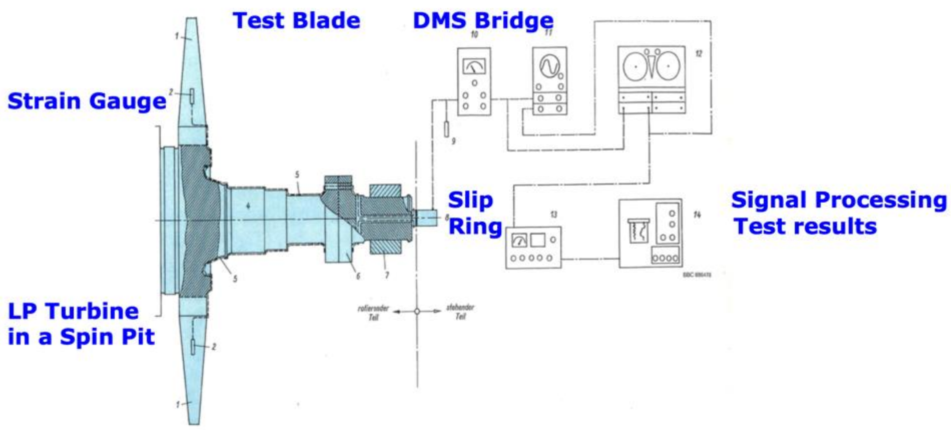


Figure 37 Test Equipment for LP Turbine Blade Measurements with Strain Gauges (Source [1])

Advanced optical methods, such as laser Doppler vibrometry (LDV) and high-speed imaging, enable full-field and non-contact vibration measurement. These techniques are often used in experimental or laboratory environments due to their complexity.

Laser Doppler Vibrometer (LDV):Measures surface velocity and displacement with high spatial and temporal resolution.

High-Speed Cameras: Capture the visual oscillation of blades, allowing for modal shape reconstruction.

Advantages:

- Non-contact and high resolution.
- Useful for full-field vibration analysis.

Disadvantages:

- Sensitive to environmental conditions (dust, steam).
- High cost and complexity limit application in operational turbines

8.4 SIGNAL PROCESSING TECHNIQUES

By todays available Signal Processing Techniques the raw signals in the time domain (voltage or strain) are treated in various processing steps. Some examples for these processes are:

- Filtering to remove noise and Amplification
- Fast Fourier Transformation to identify dominant frequencies

Additionally, advanced signal processing algorithms such as Fast Fourier Transform (FFT) and wavelet analysis are often employed to extract frequency components and transient features from the measured signals. These techniques help in isolating important vibration characteristics and support more accurate diagnostics of blade dynamics.

- Peak detection
- Blade Passage time extraction
- Compare actual versus expected arrival times
- Derive Blade Tip Deflection from timing deviations

The following list presents some of the quantities from the different signal processing steps, which are needed for an analysis and evaluation of the blade vibration problem:

- Blade deflections versus time
- Amplitudes of displacement or strain at the blade root and at the blade tip
- Dominant vibration frequencies: natural and excitation frequencies
- Identification of mode shapes as deformation across the blade span
- Energy dissipation characteristics for the damping ratio
- Phase lag between blades or with the excitation
- Strain cycle account for a fatigue and damage estimation

9 RPM Restrictions and Avoidance Zones

Understanding and managing blade vibrations in large low-pressure steam turbines is crucial for ensuring the integrity and safe operation of nuclear power plants. One of the primary strategies to mitigate vibratory risks is the identification of critical speeds and the establishment of RPM avoidance zones. This section outlines the principles and procedures surrounding this aspect of turbine operation.

9.1 CRITICAL SPEEDS AND RESONANCE CROSSINGS

Critical speeds refer to specific rotational speeds at which the excitation frequency of the turbine coincides with one of its natural frequencies, leading to a resonance, either for torsional shaft vibrations or for blade vibrations During resonance, vibratory amplitudes can increase significantly, leading to a severe risk to blade integrity and overall turbine reliability.

In large low-pressure steam turbines, the blade rows exhibit multiple natural frequencies due to their geometry and boundary conditions. These natural frequencies are influenced by blade length, material properties and operational conditions such as pressure and temperature. The critical speeds are typically identified through Campbell diagram analysis, which plots natural frequencies against rotor speed, and highlights intersection points with excitation harmonics. The excitation frequencies can be synchronous speed harmonics or grid harmonics.

9.2 EXPLANATION OF AVOIDANCE ZONES

Avoidance zones are defined ranges of rotational speeds around identified critical speeds where operation should be minimized or completely avoided. These zones are established with safety margins to account for uncertainties in frequency estimation and operational variances. Operating within these zones can lead to high vibration amplitudes, fatigue damages and potential blade failures.

The width of an avoidance zone depends on the damping ratio of the blade system, the accuracy of the frequency predictions and the nature of the excitation. In nuclear turbines, where reliability is paramount, conservative avoidance margins are typically applied.

An example of an avoidance zone has been shown in figure 30 of chapter 6.6. More details can be found in the ISO- Standard 22266-1:2022

9.3 OPERATING RANGE PLANNING

To ensure safe turbine operation, engineers define permissible operating ranges that lie outside the avoidance zones. The goal is to operate continuously at RPMs that are far from critical speed or resonance condition. This requires a detailed analysis of all blade and rotor torsional natural frequencies and of potential excitation sources, including harmonics from Rotor Blade-Interactions (RBI, chapter 6.2), Nozzle Passing Frequencies (NPF, chapter 5.1) and Partial Admission Frequencies (chapter 5.2).

In practice, operating ranges are often constrained by other turbine or plant system requirements (e.g., power output, condenser pressure), therefore multidisciplinary coordination is essential. Simulation tools integrating FEA and CFD are used to validate that the selected operating points remain safely distanced from vibration-prone zones.

9.4 SAFE RAMP UP AND COAST DOWN PROCEDURES

Passing through critical speeds is sometimes unavoidable during startup or shutdown. To mitigate risks, specific ramp-up and coast-down strategies are implemented. These include:

- Fast traversal of critical speed zones to minimize dwell time
- Monitoring vibration levels in real-time to detect abnormal resonance behavior.
- Use of tuned mass dampers or blade dampers to suppress amplitudes during crossing.

Operators are trained to follow strict procedures and turbine control systems are configured with safeguards to ensure transitions through avoidance zones are executed safely.

10 Vibration Mitigation Methods

Blade vibrations, especially in the last stages of low-pressure steam turbines can lead to severe mechanical failures and reduced operational life. Due to this fact, effective vibration mitigation strategies are crucial for the safe and efficient operation of the turbines in nuclear power plants. This section discusses key methods for mitigating vibrations in turbine blades, including reduction of excitation, tuning, damping, steam flow optimization, and other innovative approaches.

10.1 MITIGATION METHODS – GENERAL OVERVIEW

The table in figure 38 presents a general overview about vibration mitigation methods. Conventional (passive) as well as extended (active) solutions have been developed in the theory of vibrations. In the following chapters we describe some vibration control measures, which are applicable for blades in large Low Pressure Turbines.

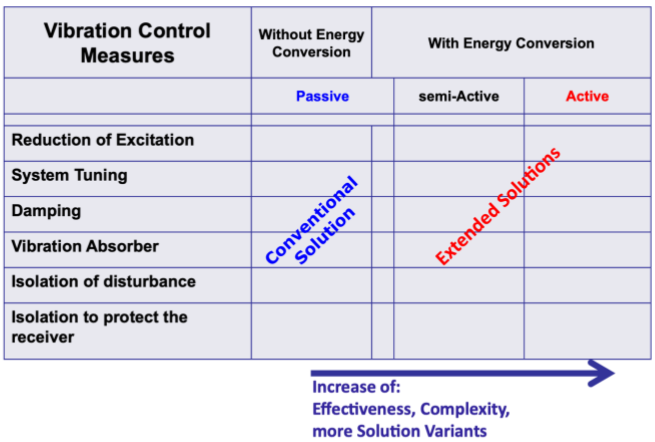


Figure 38 Passive and Active Vibration Control Measures (Source Normann)

10.2 VIBRATION CONTROL BY BLADE TUNING

Blade tuning involves modifying the blade geometry or stiffness to shift its natural frequencies away from known excitation frequencies (e.g., harmonic components of the rotor speed or flow-induced forces). Proper tuning helps to avoid resonance conditions that can amplify vibration amplitudes.

10.3 VIBRATION CONTROL BY DAMPING

Damping in the blade system is helpful to reduce blade vibrations, especially in resonance conditions. Unfortunately the damping for the free standing blades in the last stages of the LP turbines is relatively low. It results from the blade material damping, from friction damping in the roots and from the surrounding steam damping. The following additional damping mechanisms can help to increase the damping. However, they are usually not installed in the Low Pressure Turbines.

Additional Friction dampers, often installed at root or shroud interfaces, utilize sliding friction to dissipate vibrational energy. Their effectiveness increases with vibration amplitude, making them particularly valuable under transient or high-load conditions.

Shrouds are circumferential connections at the blade tips that increase structural stiffness and provide additional damping by frictional energy dissipation at contact surfaces. They reduce relative blade motion and suppress vibratory modes, especially in grouped blade configurations.

Snubbers act as restraints that limit excessive blade displacement. They are particularly useful in nuclear turbines where safety margins are critical. By providing localized constraints, snubbers effectively alter the dynamic behavior of blade rows.

10.4 VIBRATION CONTROL BY REDUCTION OF EXCITATION

Unsteady aerodynamic forces from steam flow are a major source of excitation. Optimizing steam path design by CFD helps minimize these forces. Methods include:

- Optimizing nozzle and blade profiles to reduce flow excitations
- Staggering blade rows to minimize wake interactions
- Controlling inlet conditions and moisture content to stabilize flow patterns

10.5 VIBRATION CONTROL BY OTHER MITIGATION METHODS

Operating turbines within safe speed ranges and avoiding known resonance regions can significantly reduce vibratory risk. Automated control systems can adjust operating parameters in real-time.

Material Selection: Using materials with high fatigue resistance and good damping characteristics (e.g., titanium alloys, martensitic steels) enhances blade durability.

Blade Monitoring and Feedback: Modern turbines employ blade health monitoring systems (e.g., tip-timing, strain gauges, accelerometers) to detect and respond to vibratory anomalies early.

11 Failure Mechanisms due to Blade Vibrations

Vibration-induced damage to turbine blades is a major risk to the operational safety and service life of steam turbines – especially in the last stages of large low-pressure turbines in nuclear power plants, where long blades and high steam velocities are present. The most important vibration-related failure mechanisms are explained in the following short subchapters. Some experience from turbine manufacturers confirm the vibration related failure mechanisms.

11.1 FATIGUE CRACKS AT THE BLADE ROOT

The blade root is subjected to particularly high mechanical stresses, as this is where the blade is clamped into the rotor and where bending loads from vibrations are maximal. Repeated loads near the natural frequency (e.g., from periodic excitation by the steam flow by Rotor Blade Interactions RBI) lead to vibration fatigue:

- **Mechanism**: Crack initiation due to cyclic stresses, often at notches, fits, or surface defects.
- **Progression**: Microcracks → crack growth over many load cycles → sudden fracture.
- **Contributing factors**: Resonant excitation, poor surface quality, geometric notches, harsh operating conditions (e.g., high temperature, humidity).

11.2 BLADE FAILURE DUE TO RESONANCE OVERLOAD

If an excitation frequency approaches a natural frequency, resonance can occur. The resulting amplitudes may become so large that the allowable stresses in the blade material are exceeded, especially when the damping is low:

Mechanism: Supercritical vibrations → plastic deformation → structural failure.
Typical scenarios: Poor tuning of natural and excitation frequencies (e.g., harmonic excitation from rotor (RBI), nozzle passing frequency , or disturbances in the steam flow).
Outcome: Complete blade break-off, possibly causing secondary damage from flying parts.

11.3 EROSION INDUCED WEAKENING WITH SUBSEQUENT FRACTURE

In the last stages of low-pressure turbines, water droplet erosion often occurs, especially in condensing wet steam. This causes localized material loss on the blade surface:

- **Mechanism**: Material removal due to erosion → local cross-section weakening → increased stress → fracture under existing vibration loads.

- **Especially affected areas**: Blade edges, leading edge, regions with high droplet concentration.
- **Combined effect**: Pre-damage due to erosion significantly reduces fatigue strength, making even small vibrations capable of causing failure.

To determine the occurrence of possible water droplet erosion the Wilson line is helpful. The Wilson Line is a thermodynamic boundary in the phase diagram of water/steam, which marks the onset of condensation in a rapidly expanding steam flow. In turbines, it separates superheated or dry saturated steam from wet steam.

11.4 EXPERIENCE FROM TURBINE MANUFACTURERS

Here are some comments from turbine manufacturers regarding failure mechanisms due to blade vibrations in turbines of Nuclear Power Plants:

- No problems due to blade vibrations in the HP Turbines
- Cracks and occasional fractures in the L-2 blade row. Material: X20Cr13. Cracks started at the blade root in the upper carrying shoulder
- No corrosion and no fretting wear detected at the break starting points
- All cracks were caused by High Cycle Fatigue (HCF) showing rest lines, which means that the vibrations leading to crack growth, occurred only at special load conditions
- Cracks also occurred in the L-0 blades, where the initial crack was in the upper carrying shoulder.
- No fracture in the L-0 blade rows (material X20Cr13)

12 Literature

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13 List of Abbreviations

AGT Air Gap Torque

BC Boundary Condition

BTT Blade Tip Timing

CFD Computational Fluid Dynamics

EMI Electro-Mechanical Interaction

FEM Finite Element Method

FEA Finite Element Analysis

FBI Fluid Blade Interaction

FSI Fluid Structure Interaction

GEN Generator

HPT High Pressure Turbine

LPT Low Pressure Turbine

MA Modal Analysis

NPF Nozzle Passing Frequency

RBI Rotor Blade Interaction

RPM Revolutions Per Minute

SDOF Single Degree of Freedom System

SSR Sub Synchronous Resonance

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BLADE VIBRATIONS IN TURBINES

The main purpose of this Energiforsk project Blade Vibrations in Turbines is to present a clear physical description, how blade vibrations are generated and by which system parameters they are influenced. In the report it will be shown, which facts have to be considered for the turbine-manufacturer in the design process regarding blade vibrations. Of importance for the operator is a smooth operation of the steam turbines with a permissible state of blade vibrations. To control the actual state, blade vibrations can be measured during operation (Monitoring). By such measurements indications can be obtained about possible failure mechanisms. Vibration Standards help for an evaluation of the blade vibrations.