Vorlesungen Mechatronik im Wintersemester

Vibrations in Nuclear Applications 2021

TECHNISCHE UNIVERSITÄT DARMSTADT

Annual Energiforsk Seminar (10.11.21)

Modelling, Monitoring and Vibration Control of High Performance Turbomachinery

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Technische Universität Darmstadt and Fraunhofer Institute LBF

MiM

Modelling, Monitoring and **Vibration Control of High Performance Turbomachinery**

- **Introduction: High Performance Turbomachinery**
- **Vibration Phenomena in High Performance Turbomachinery**
- **Modelling of High Performance Turbomachinery**
  - Modelling of the Laval Shaft
  - Modelling of High Performance Turbomachinery
- **Monitoring of High Performance Turbomachinery**
  - Conventional Monitoring
  - Future Model Based Monitoring & Diagnosis
- **Vibration Control of High Performance Turbomachinery**
  - Passive Vibration Control
  - Active Vibration Control

Introduction: High Performance Turbomachinery
Turbogenerator: Steamturbine and Generator

Disciplines : Thermodynamics **Mechanics** Electrodynamics

Stresses
Strength of material

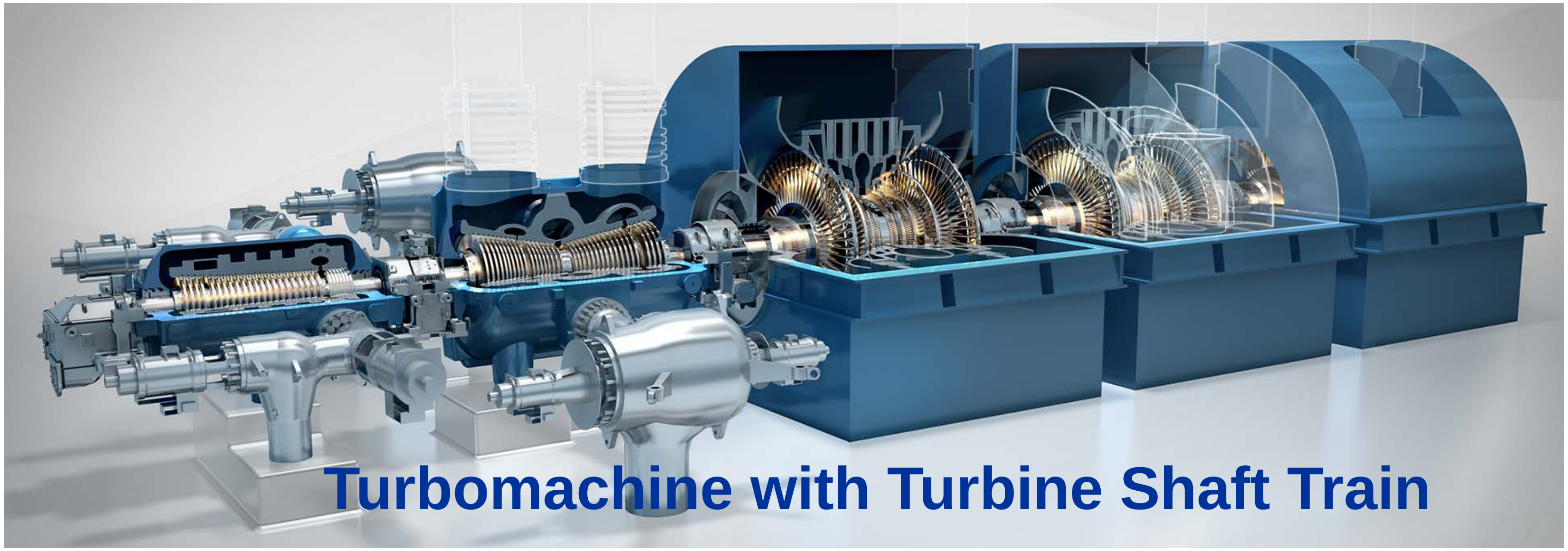
Lateral and Torsional
Rotordynamics

Critical speeds
Unbalance Response
Air Gap Torques

Stability

Introduction: High Performance Turbomachinery
Model of a Steam Turbine Shaft Train

Power: 1000 MW
Shaft Length: 55,8 m
Shaft Weight: 426 to
Bearing Diameter: 600mm



Modelling, Monitoring and **Vibration Control of High Performance Turbomachinery**

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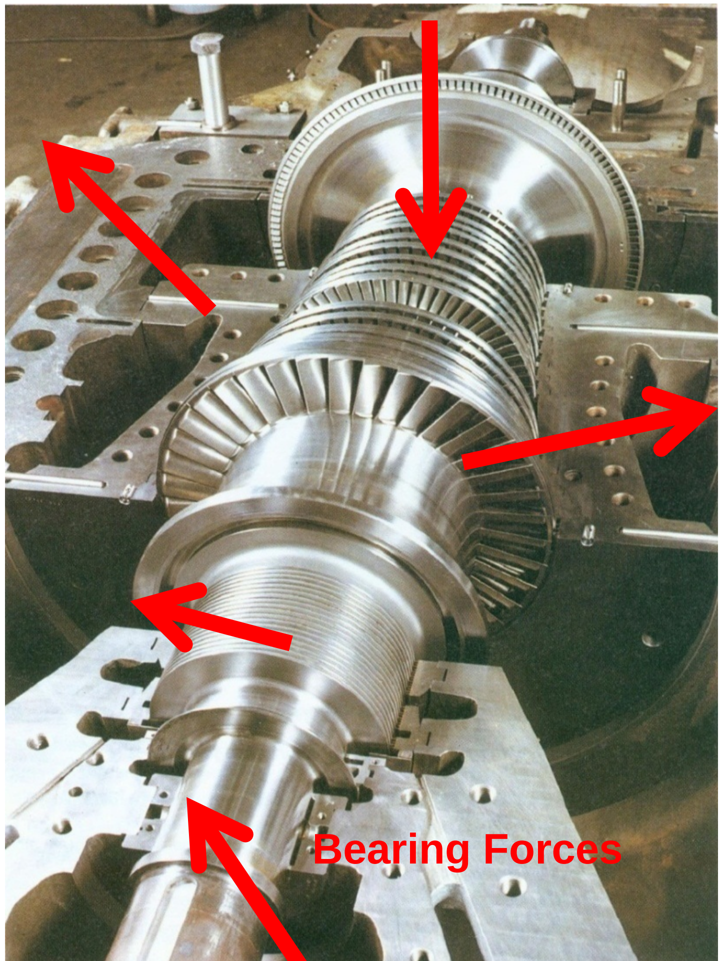
Vibration Phenomena in High Performance Turbomachinery - Static and Dynamic Forces

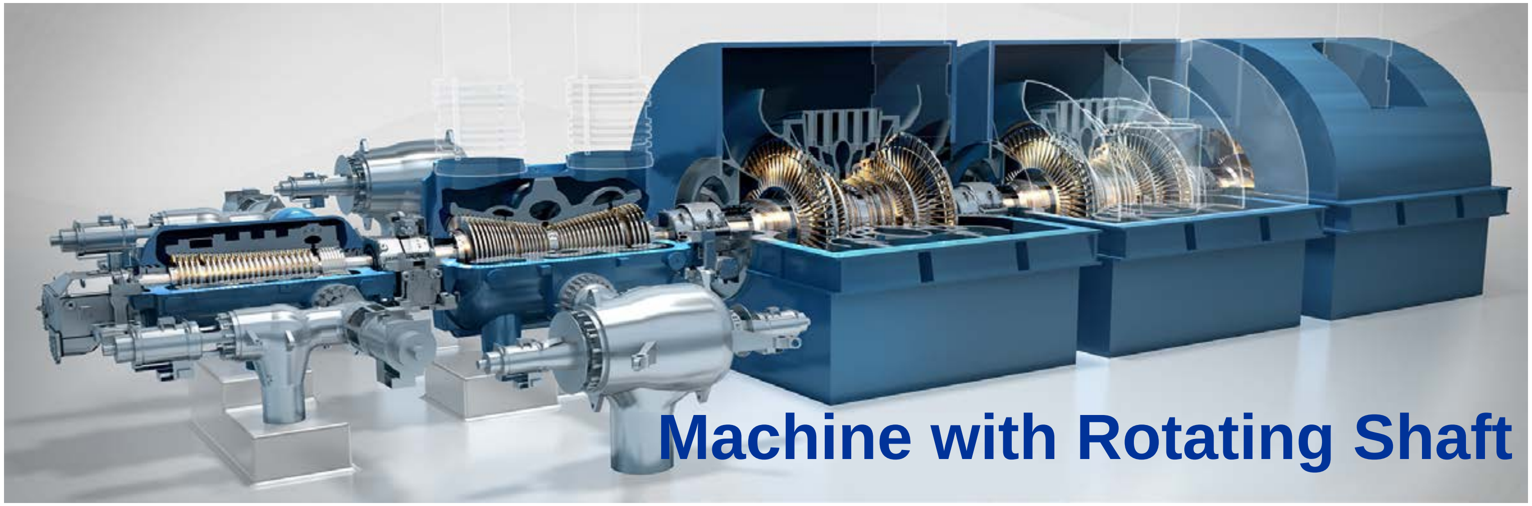


Vibration Phenomena in High Performance Turbomachinery
Why do we need Rotordynamics?

- **How is the influence of time dependent Forces and Moments on the dynamic behavior of a Machine?**
- **Which Motions of Vibration and which internal Stresses act on the rotating and on the non-rotating Machine Parts?**
- **Are Critical Conditions (Resonances, Instabilities) possible?**
- **Can Vibrations destroy Machine Parts?**
  Rubbing, Blade Loss, Shaft Cracks, Bearing Failures, large Deformations,…
- **Which Interactions have to be considered?**
  Fluid Structure Interaction, Rotor Structure Interaction, Rotor Blade Interaction,
- **Electromechanical Interaction**

Vibration Phenomena in High PerformanceTurbomachinery

Lateral and Torsional Vibrations of Shaft Trains



Coupling of Lateral and Torsional Vibrations usually negligible

Vibration Phenomena in High PerformanceTurbomachinery
Lateral and Torsional Vibrations of Shaft Trains

Which Phenomena are of Practical Relevance?

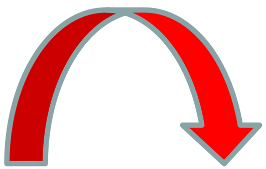
Lateral Vibrations: **Lateral Vibrations perpendicular to the Shaft axis with Bending along the Shaft line. Physical Effects: Inertia (masses), Siffness and Damping of System Components (Shaft, Bearings).**

Dynamic Characteristics: Natural Frequencies, Critical Speeds, Natural Modes, Stability, Amplitudes and Phase angles of the Vibration Response due Excitations

Excitation: Mechanical and thermal Unbalances, Bow (Unbalance) due to Coupling Errors, Excitation due to Instabilities in Fluid Bearings and Seals

Which Phenomena are of Practical Relevance?

Torsional Vibrations: Torsional Vibrations around the Shaft axis with torsional deformations along the Shaft line, Physical Effects: Moments of Inertia, Torsional Stiffness and Damping of the System components



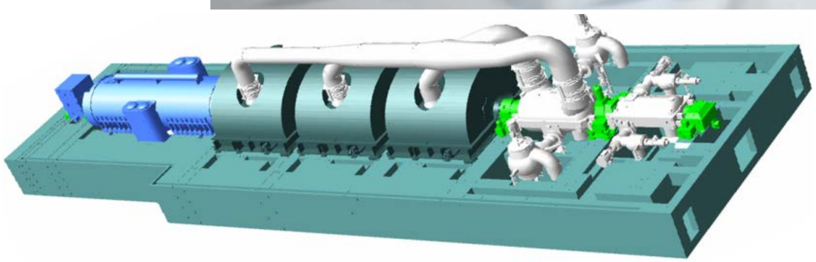
Dynamic Characteristics: Natural Frequencies, Natural Modes, Modal Damping, Amplitudes and Phase angles of the Vibration Response due to Excitations.

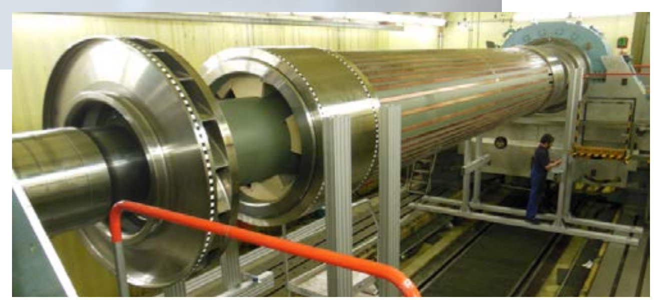
Excitation: Air Gap Torques in Electrical Machines due to Electromagnetic Coupling. Higher Harmonics occur.

Vibration Phenomena in High PerformanceTurbomachinery
Different Interactions have an Influence on Rotordynamics

Rotor-Fluid Interaction: Oil Film Bearings, Seals

Rotor-Blade Interactions





Rotor-Structure Interaction: Casing, Foundation

Elektromechanical Interaction: Generator,Grid

Modelling, Monitoring and Vibration Control

Modelling of High Performance Turbomachinery

Monitoring of High Performance Turbomachinery

Passive and Active Vibration Control of High Performance Turbomachinery

Modelling, Monitoring and **Vibration Control of High Performance Turbomachinery**

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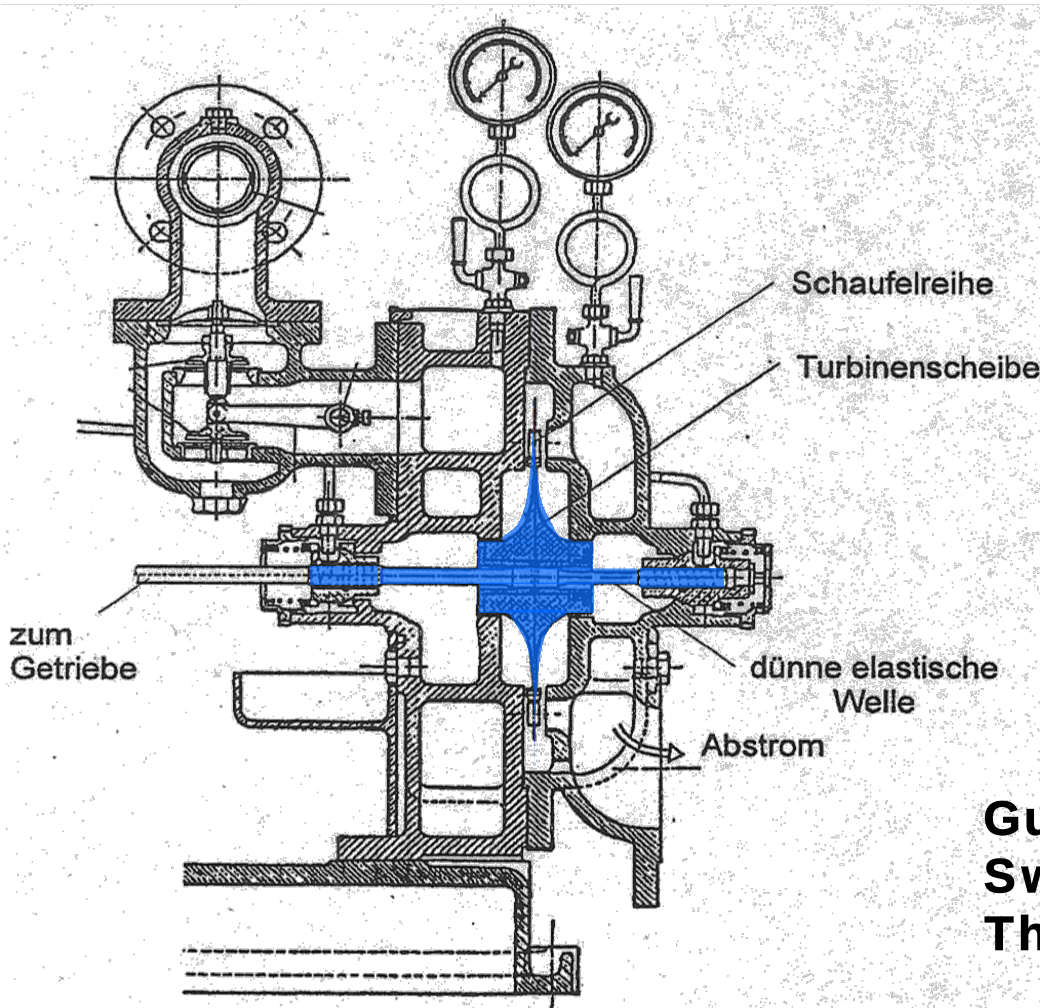
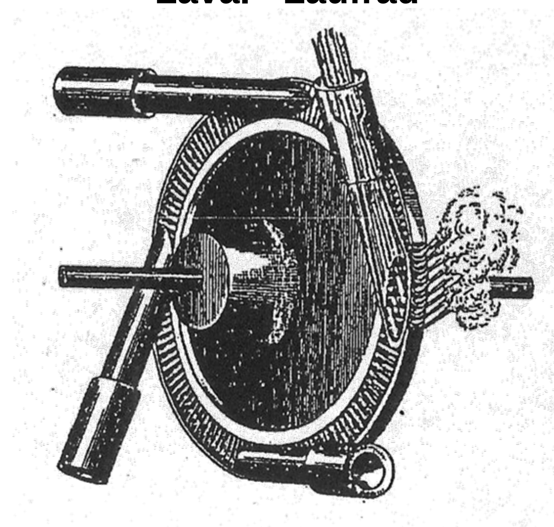
Modelling of the Laval Shaft - Jeffcott Rotor

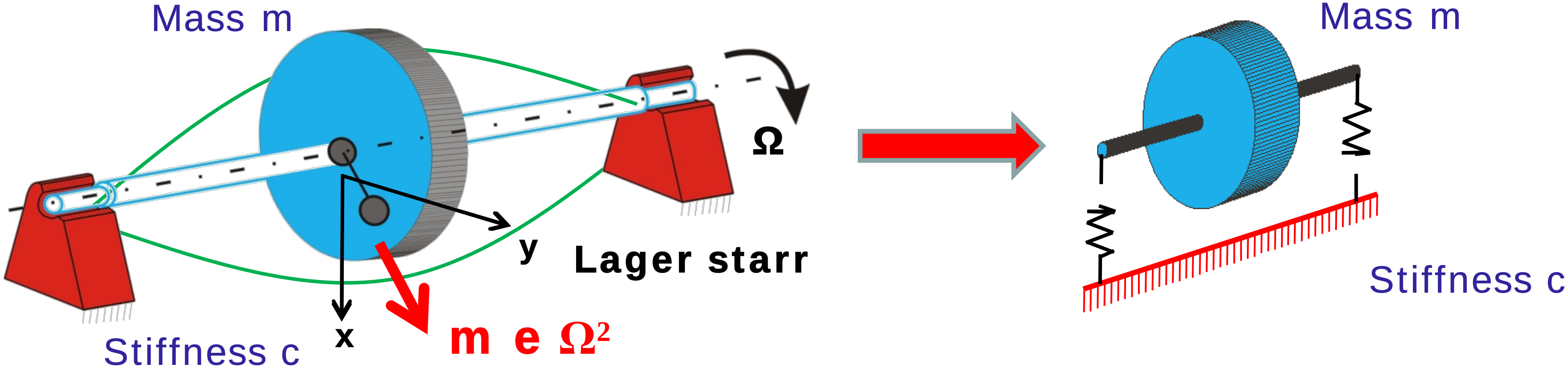


Laval - Laufrad



Gustav de Laval (1845 – 1913)
Swedish Engineer,
Theory by Föppl 1895

Modelling of the Laval Shaft
Equations of Motion for Lateral Vibrations



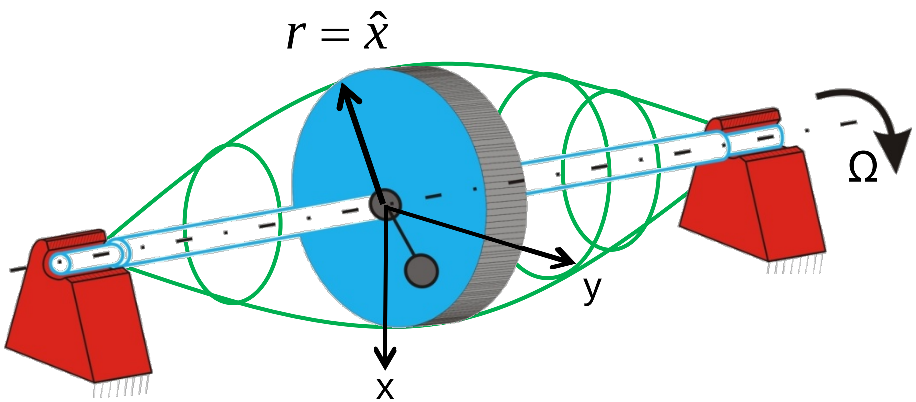
$$m\,\ddot{x} + d\,\dot{x} + c\,x = m\,e\,\Omega^2 \sin \Omega t$$

$$m\,\ddot{y} + d\,\dot{y} + c\,y = m\,e\,\Omega^2 \cos \Omega t$$

Inertia Damping Stiffness Unbalance

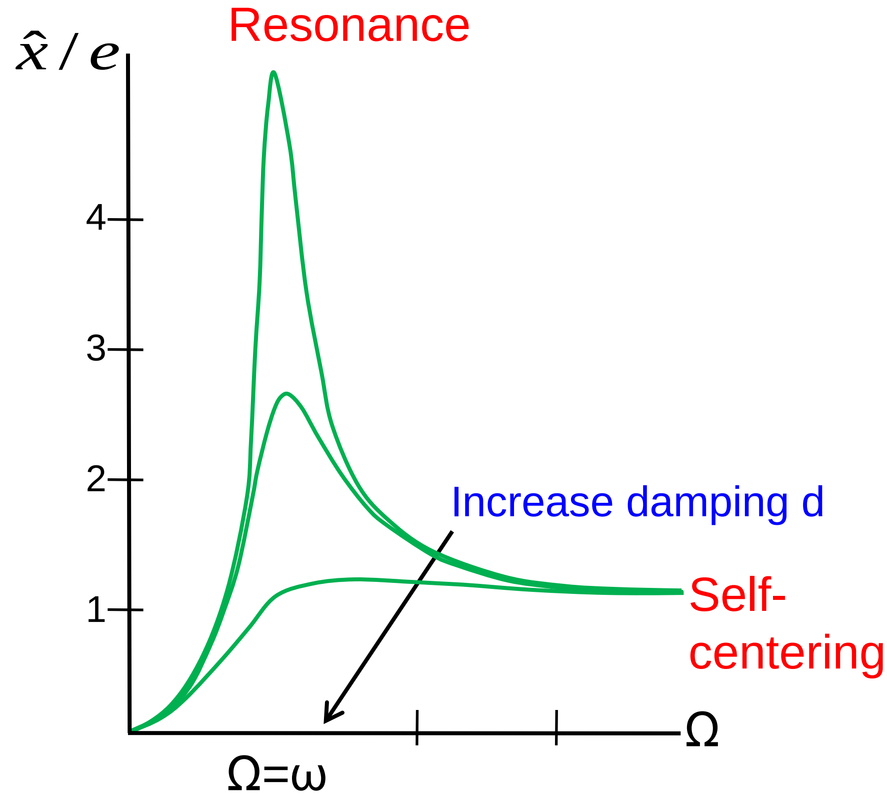
Modelling of the Laval Shaft
Forced Unbalance Vibrations

Shaft vibrations with circular orbit,

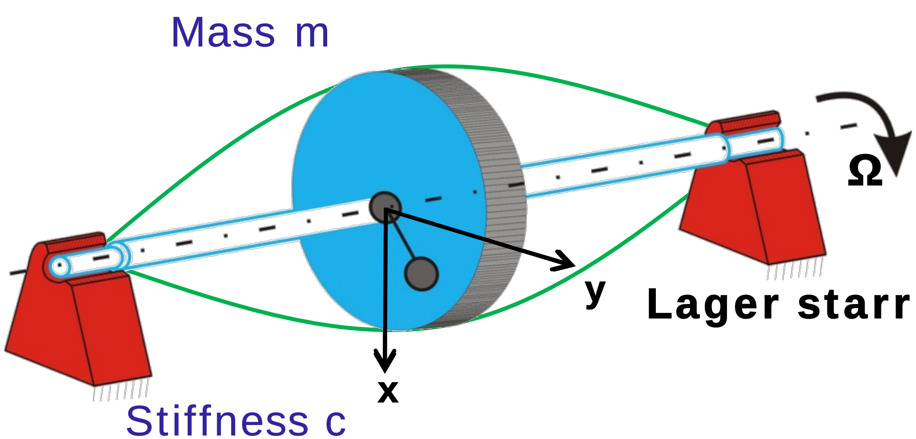


$$\left(\frac{\hat{x}}{e}\right) = \frac{m\Omega^2}{\sqrt{\left(c - m\Omega^2\right)^2 + \left(d\Omega\right)^2}}$$

Amplitude $\hat{x}$ of unbalance vibration

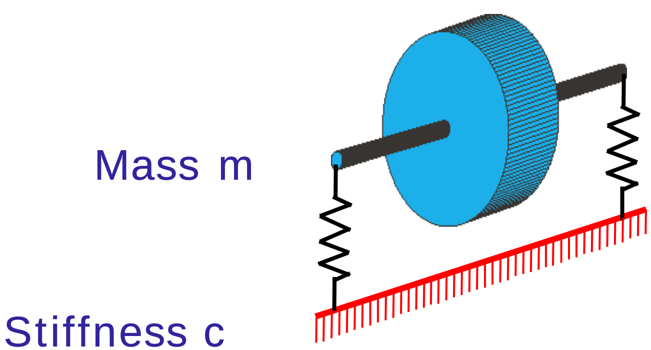


Modelling of the Laval Shaft
Natural Frequency of Laval-Shaft (Lateral)



Natural Frequency: Laval Shaft

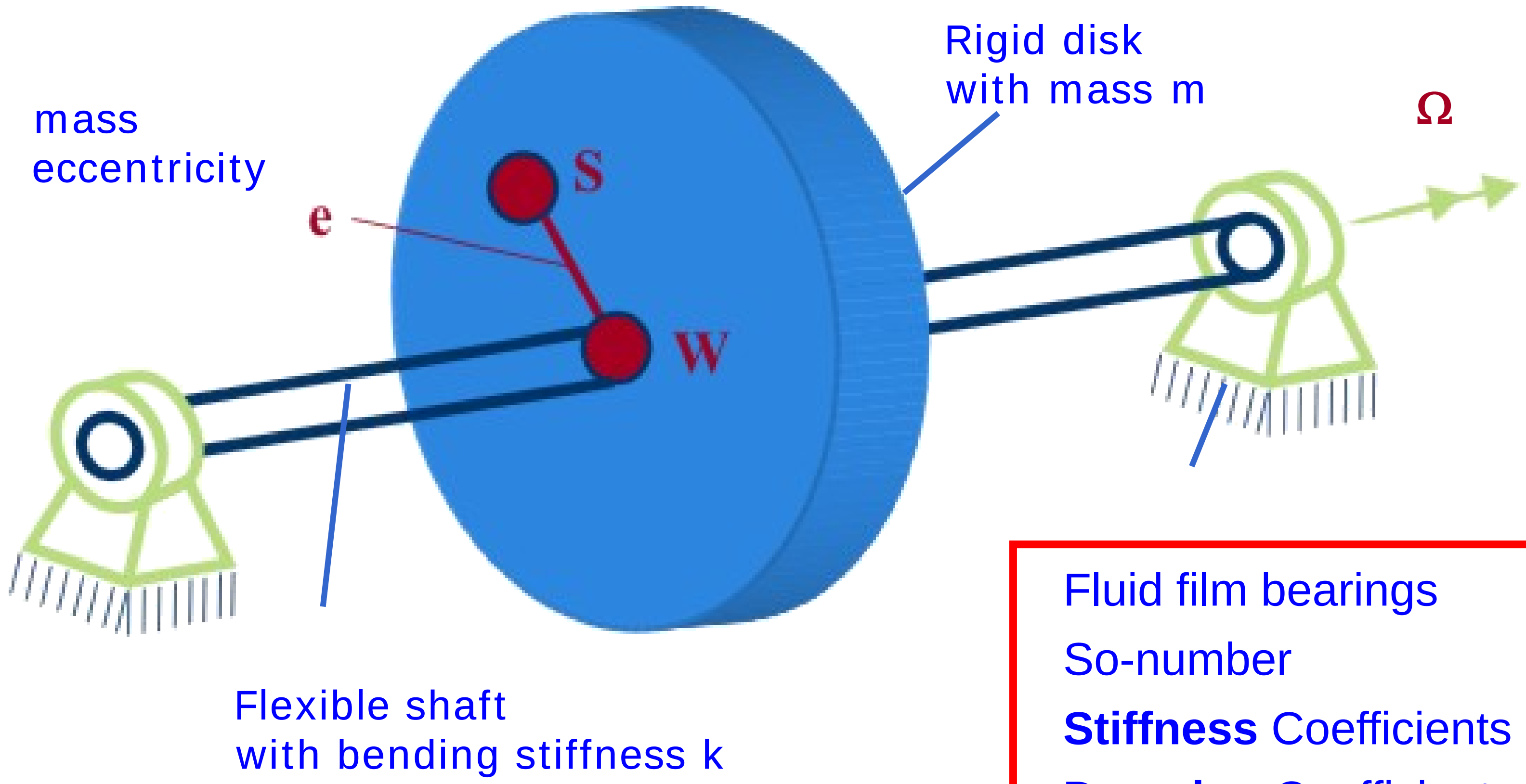
$$\boxed{\omega = \sqrt{c/m}}$$



Excitation with Frequency ω

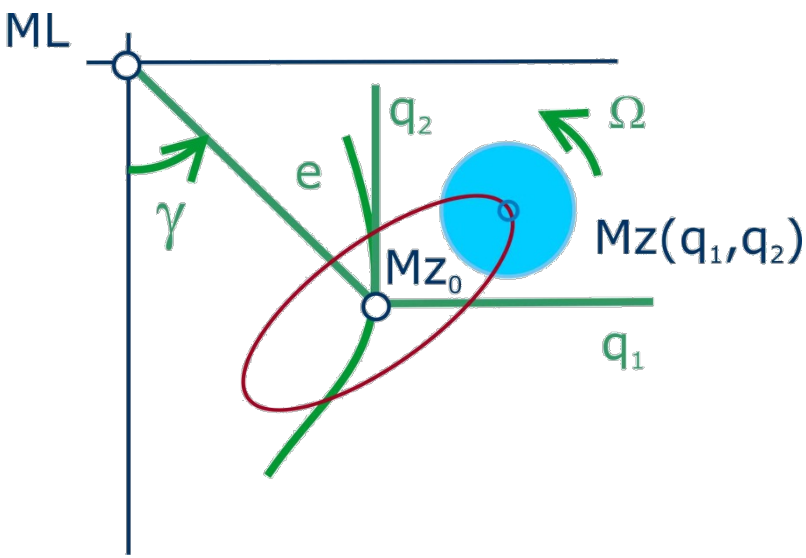
Resonance Case

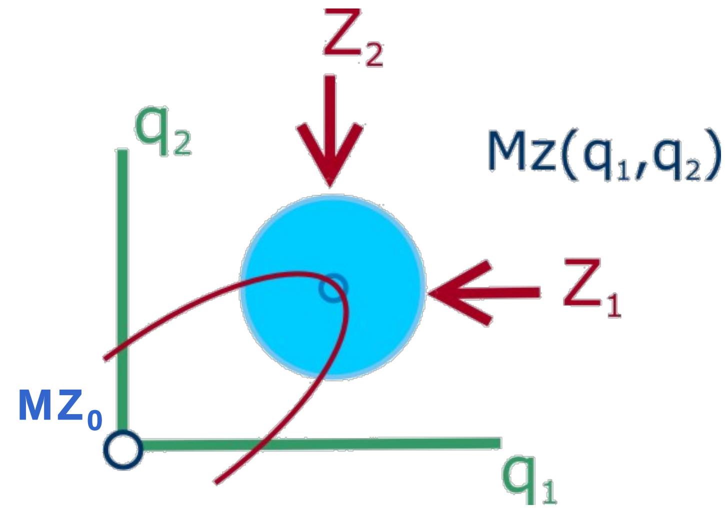
Modelling of the Laval Shaft
Influence of the Fluid Film Bearings



Fluid film bearings
So-number
Stiffness Coefficients
Damping Coefficients

Modelling of the Laval Shaft
Fluid Film Bearings (Linear Theory)





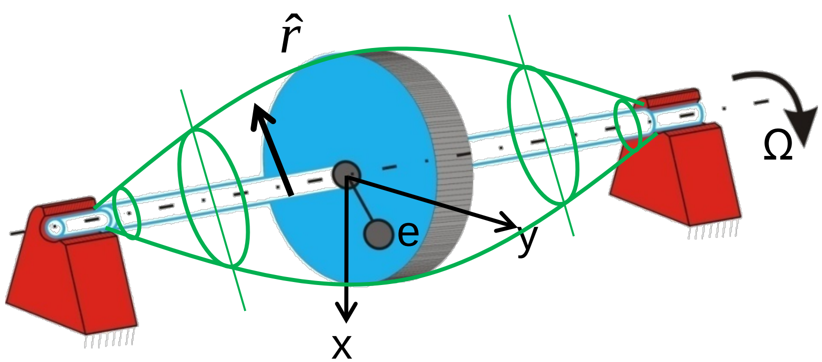
Dynamic forces Z_1, Z_2:

$$\boxed{\begin{aligned} Z_1 &= k_{11} q_1 + k_{12} q_2 + d_{11} \dot{q}_1 + d_{12} \dot{q}_2 \\ Z_2 &= k_{21} q_1 + k_{22} q_2 + d_{21} \dot{q}_1 + d_{22} \dot{q}_2 \end{aligned}}$$

k_{ik}: stiffness coefficients d_{ik}: damping coefficients

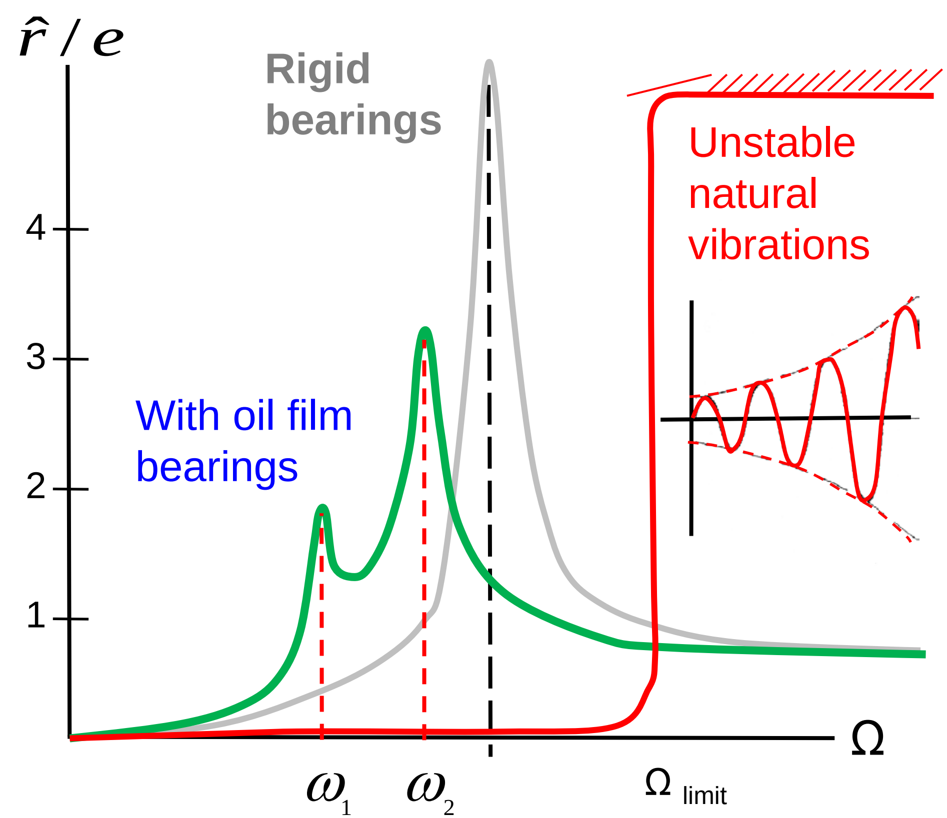
Modelling of the Laval Shaft
Influence of Oil Film Bearings in comparison to Rigid Bearings

Shaft vibration with elliptical orbits



Influence of Oil film bearings:

Stiffness, Damping, Anisotropy
Danger of self-excitation



Modelling of the Laval Shaft
Stiffness and Damping Coefficients

The stiffness and damping coefficients can be determined by numerical calculations (Reynolds-equations, CFD) or by Experiments. For a bearing with a special geometry the coefficients depend on the Sommerfeld number or on the static displacement.

Fluid film bearings have usually anisotropic behavior :

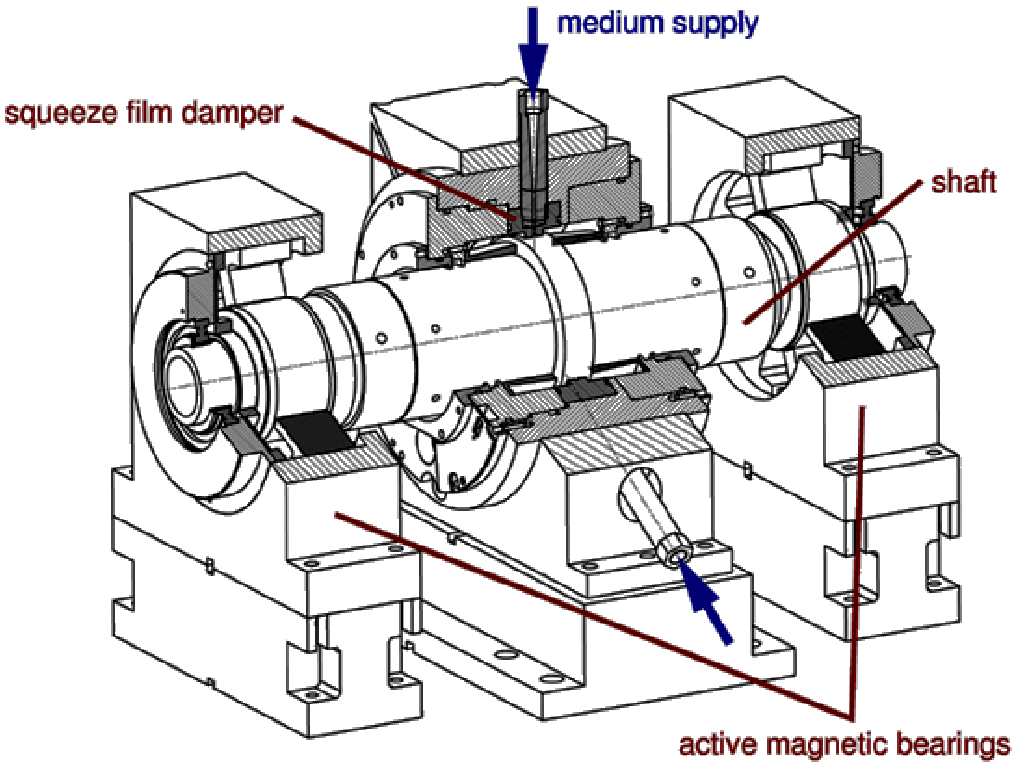
$k11 \neq k22$ and $d11 \neq d22$

The coupling coefficients differ from each other

$k12 \neq k21$ and $d12 \neq d21$

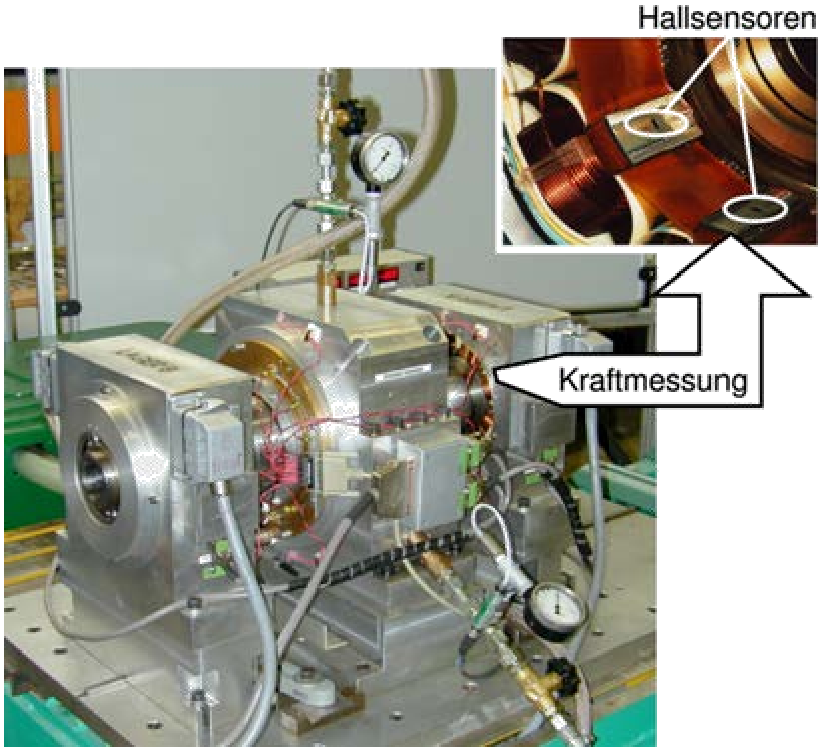
A measure for instability sensitivity is the difference: $(k12 - k21)$. Damping Coefficients d11 and d22 are good for stability. Coefficients can be found in tables or diagrams.

Modelling of the Laval Shaft
Test Rig with AMB‘s to measure Rotordynamic Coefficients



Journal Bearing Test Rig

Force Measurement



Modelling, Monitoring and **Vibration Control of High Performance Turbomachinery**

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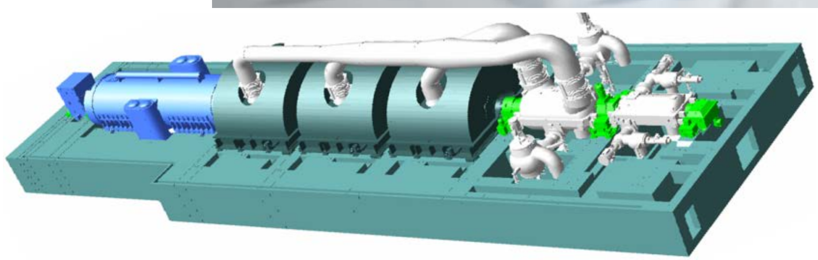
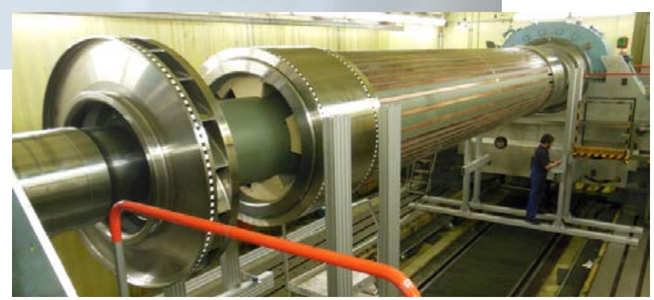
Modelling of High Performance Turbomachinery Different Interactions on Rotordynamics

Rotor-Fluid Interaction: Oil Film Bearings, Seals

Rotor-Blade Interactions

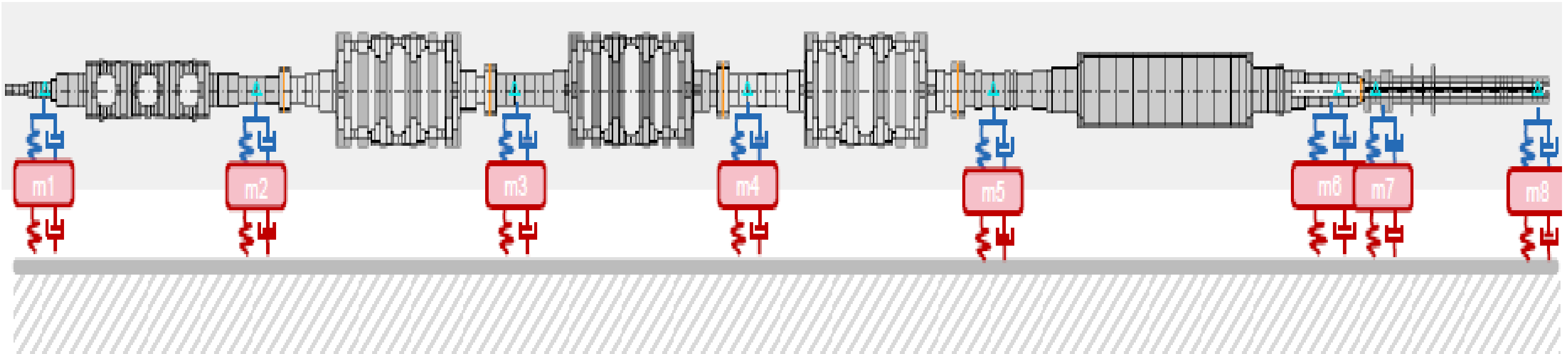




Rotor-Structure Interaction: Casing, Foundation

Elektromechanical Interaction: Generator,Grid

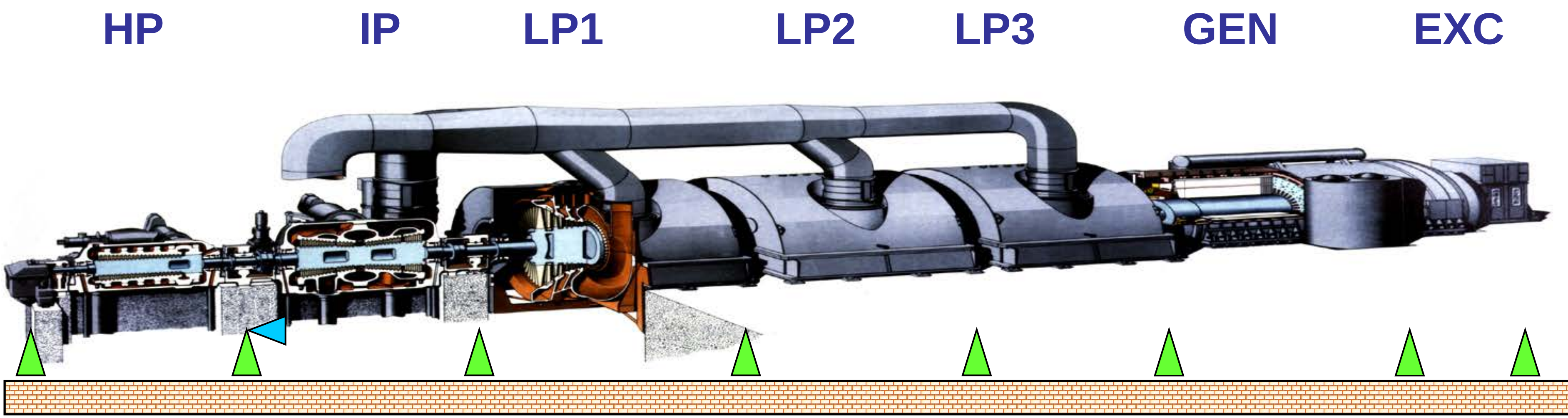
Modelling of High Performance Turbomachinery
Lateral Vibrations of Shaft Train



$$\mathbf{M}\,\ddot{x}(t)+(\mathbf{D}(\Omega)+\mathbf{G}(\Omega))\,\dot{x}(t)+\mathbf{K}(\Omega)\,\mathbf{x}(t)=\mathbf{F}(t)$$

The **Equations of Motion** for **Lateral Vibrations** of the **Turbogenerator** contain the stiffness and damping information of the shaft train, the bearings and the supports

Modelling of High Performance Turbomachinery
Oil Film Bearing Types, used in Turbine Trains



Cylindrical Bearings

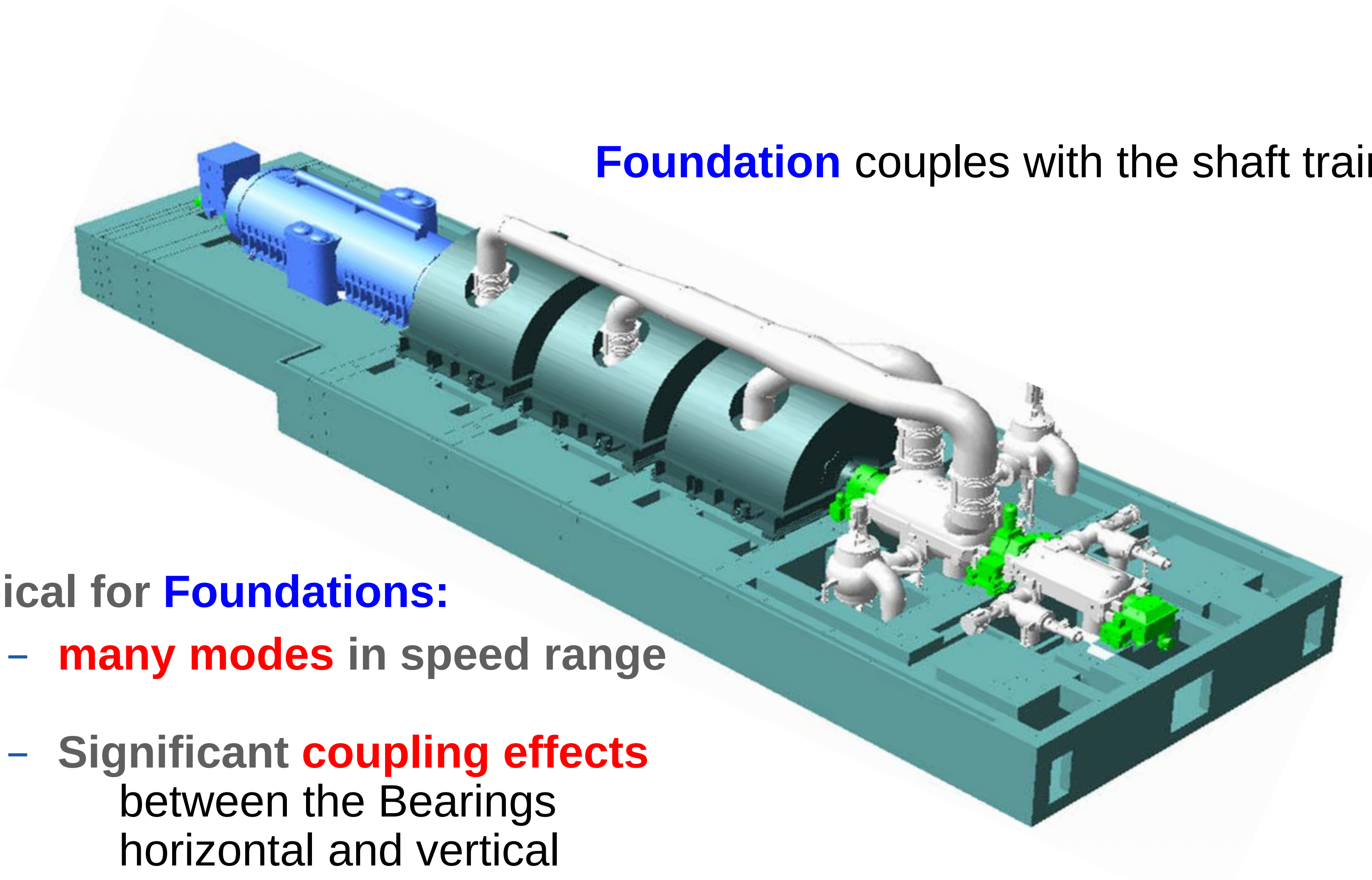
Elliptical bearings (Lemon Bore)

3-Wedge Bearings

Pocket Bearings with one or two Oil Inlets

Tilting Pad Bearings

Modelling of High Performance Turbomachinery Rotor Structure Interaction – Foundation



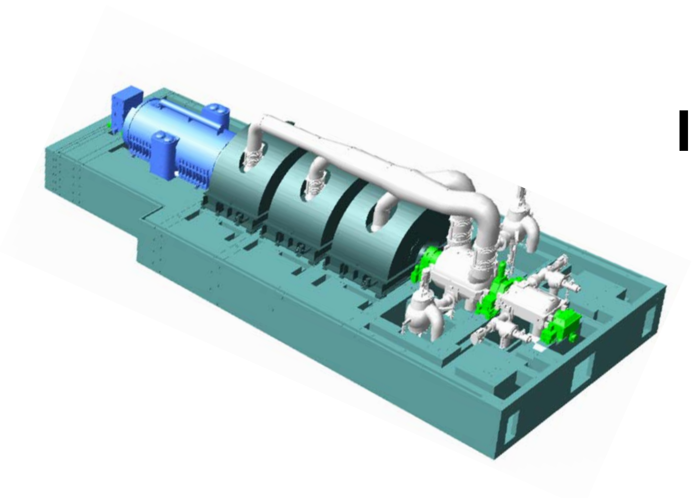
Foundation couples with the shaft train

Typical for Foundations:

- **many modes in speed range**
- **Significant coupling effects** between the Bearings horizontal and vertical

Modelling of High Performance Turbomachinery Rotor – Structure Interaction - Foundation

FE - Method
Modal Analysis
for the Foundation



Procedure

– Estimation of mode shapes and natural frequencies of the foundation without the rotor (FRF for bearing locations)

Input for Rotordynamics:

→Components of the mode shapes at the bearing locations.

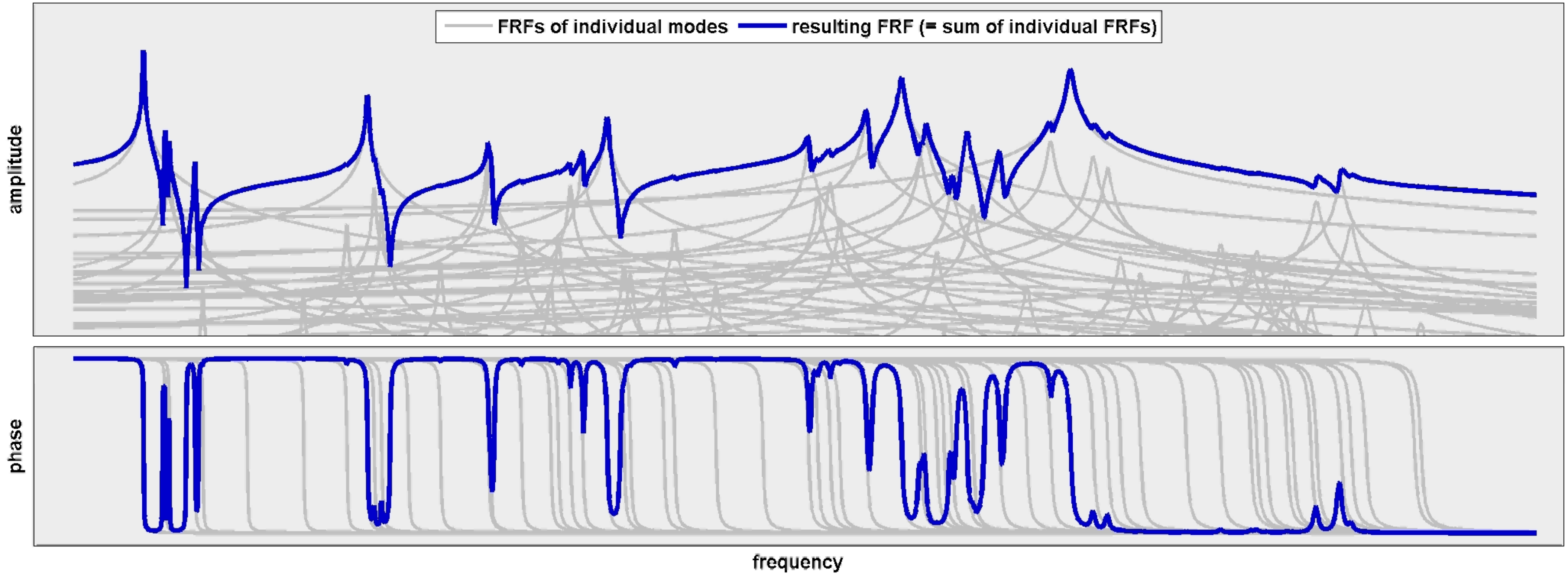
→Natural Frequencies

→Modal Damping

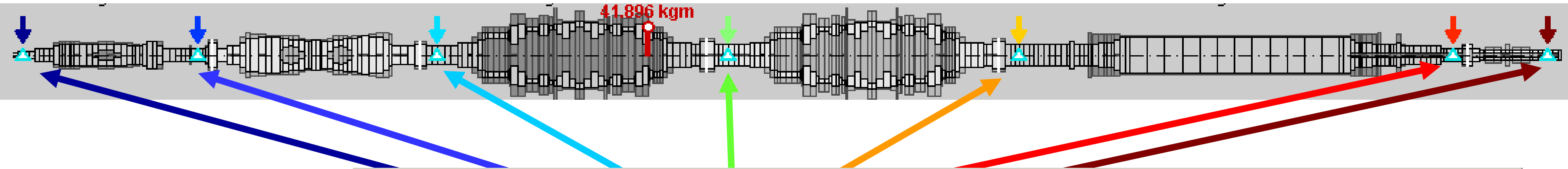
Modelling of High Performance Turbomachinery
Rotor – Structure Interaction - Foundation

FRFs from

- 3D FEM Calculations (Natural frequencies, Mode shapes)
- or Modal Analysis- Measurements



Modelling of High Performance Turbomachinery
Coupling of FRF's from Foundation with Rotor Train



	Shaft 1 Station 3		Station 39		Shaft 3 Station 4		Shaft 4 Station 5		Shaft 5 Station 5		Station 72		Shaft 6 Station 16	
	2	3	2	3	2	3	2	3	2	3	2	3	2	3
2	TFU	+	TFU	+	TFU	+	TFU	+	TFU	+	TFU	+	TFU	+
3		TFU	+	TFU	+	TFU	+	TFU	+	TFU	+	TFU	+	TFU
2			TFU	+	TFU	+	TFU	+	TFU	+	TFU	+	TFU	+
3				TFU	+	TFU	+	TFU	+	TFU	+	TFU	+	TFU
2					TFU	+	TFU	+	TFU	+	TFU	+	TFU	+
3						TFU	+	TFU	+	TFU	+	TFU	+	TFU
2							TFU	+	TFU	+	TFU	+	TFU	+
3								TFU	+	TFU	+	TFU	+	TFU
2									TFU	+	TFU	+	TFU	+
3										TFU	+	TFU	+	TFU
2											TFU	+	TFU	+
3												TFU	+	TFU
2													TFU	+
3														TFU

Legend:

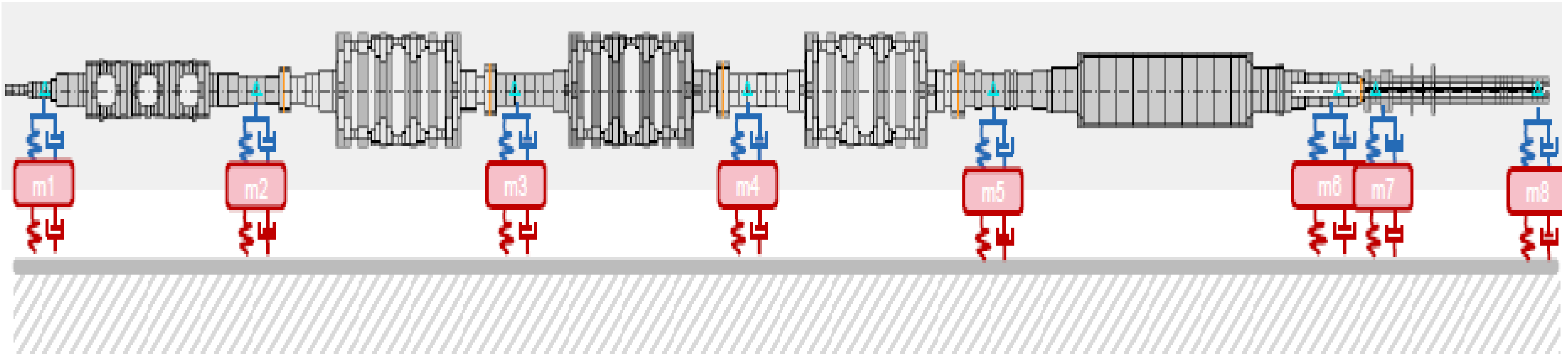
- TFU - TFU is defined
- TFU - TFU (polynomials) is defined
- \+ - TFU can be defined
- \- - TFU can't be defined, define diagonal TFU first

TFU = frequency response function (FF

Use right-mouse-button context menu to see TFU plots and other options.
See tool tip for file name of TFU.

Print Exit

Modelling of High Performance Turbomachinery
Some Eigenvalues of a large Turbogenerator



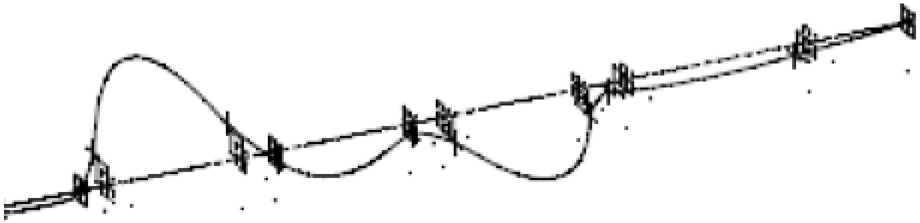
Eigenvalue Problem for Turbogenerator

$$(\lambda^2 \mathbf{M} + \lambda(\mathbf{D}(\Omega) + \mathbf{G}(\Omega)) + \mathbf{K}(\Omega)) \cdot \mathbf{x} = \mathbf{0}$$

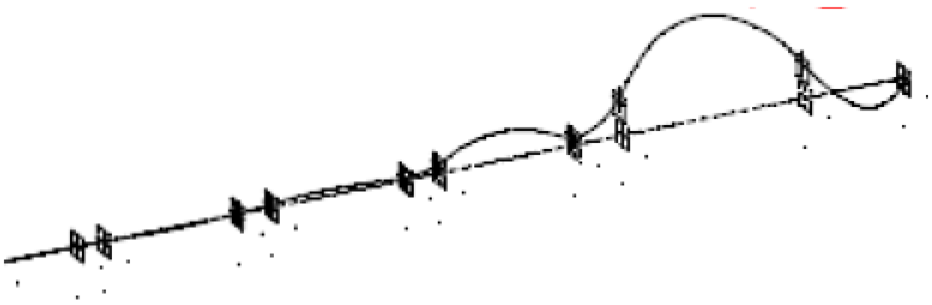
Complex Eigenvalues : $\lambda = \alpha + j\,\omega$

Information for damping (stability) and natural frequencies

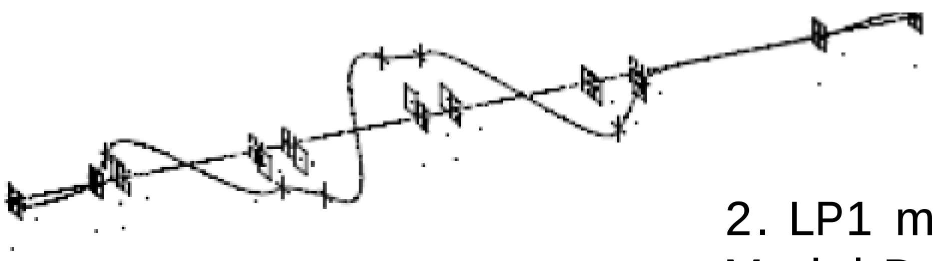
Modelling of High Performance Turbomachinery
Selected Eigenvalues and Mode Shapes



1. LP1 mode at 10.7 Hz
Modal Damping 1,1 %

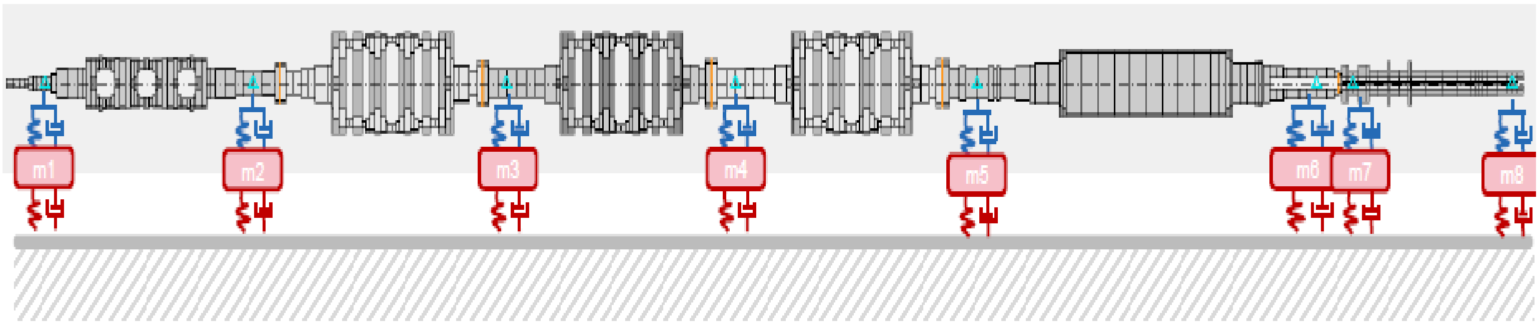


1. Generator mode at 11.0 Hz
Modal Damping 0,9 %



2. LP1 mode at 27.6 Hz
Modal Damping 9 %

Modelling of High Performance Turbomachinery
Unbalance Response of a large Turbogenerator



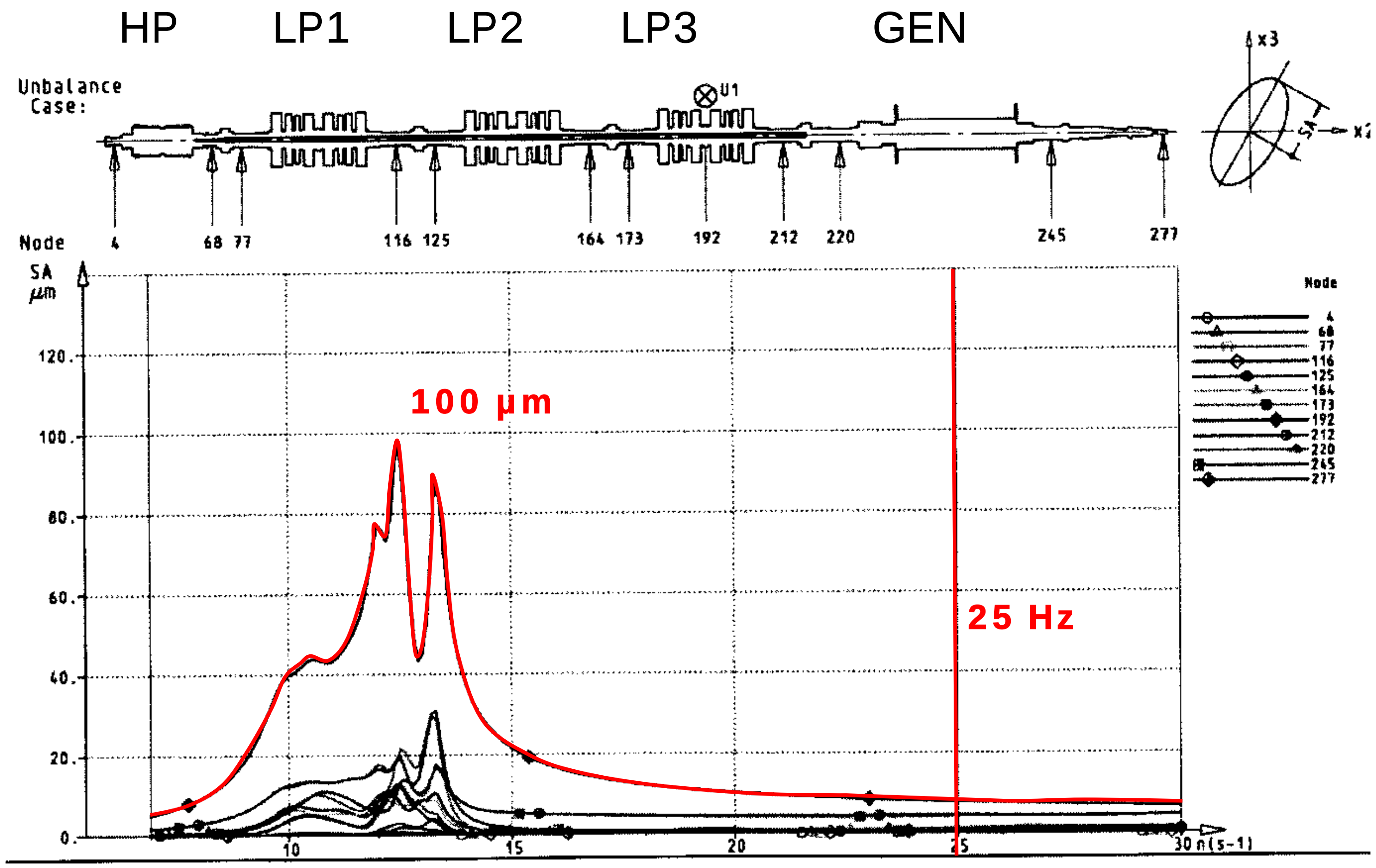
Complex Equations for **Unbalance Response**

$$((K(\Omega) - \Omega^2 M) + j \cdot \Omega (D(\Omega) + G(\Omega))) \cdot \tilde{x} = U \cdot \Omega^2 \cdot \tilde{F}$$

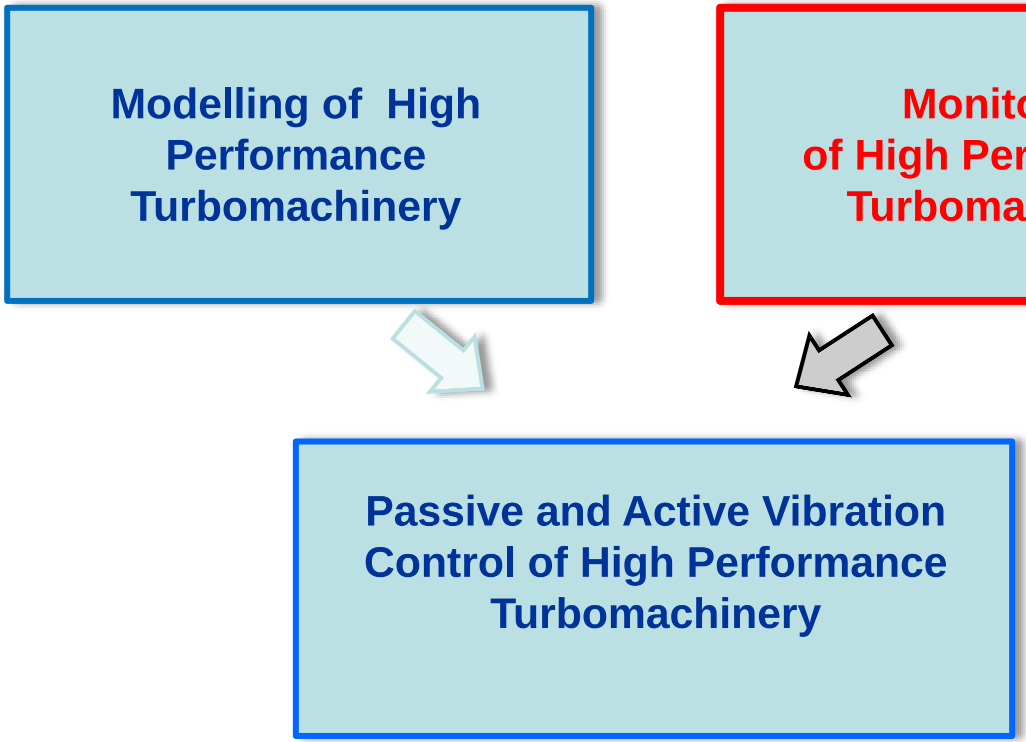
$$\tilde{x}_i = \tilde{x}_{i\,Re} + j \cdot \tilde{x}_{iIm}$$

Complex System Response contains Amplitude and Phase

Modelling of High Performance Turbomachinery
Unbalance Vibration Response of Shaft Train



Modelling, Monitoring and Vibration Control



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Conventional Monitoring: High Performance Turbomachinery

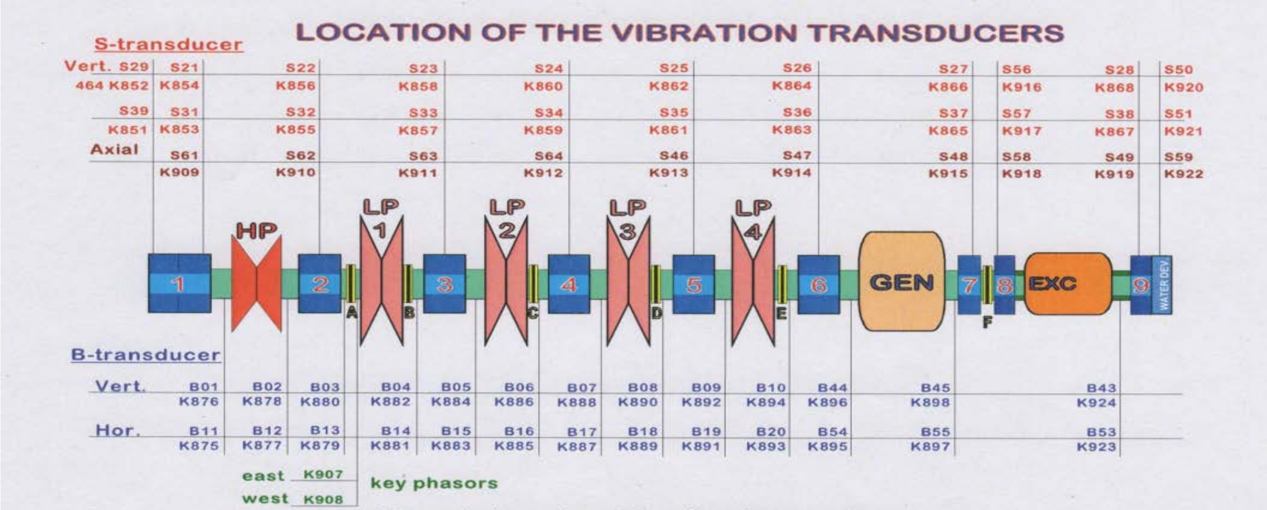
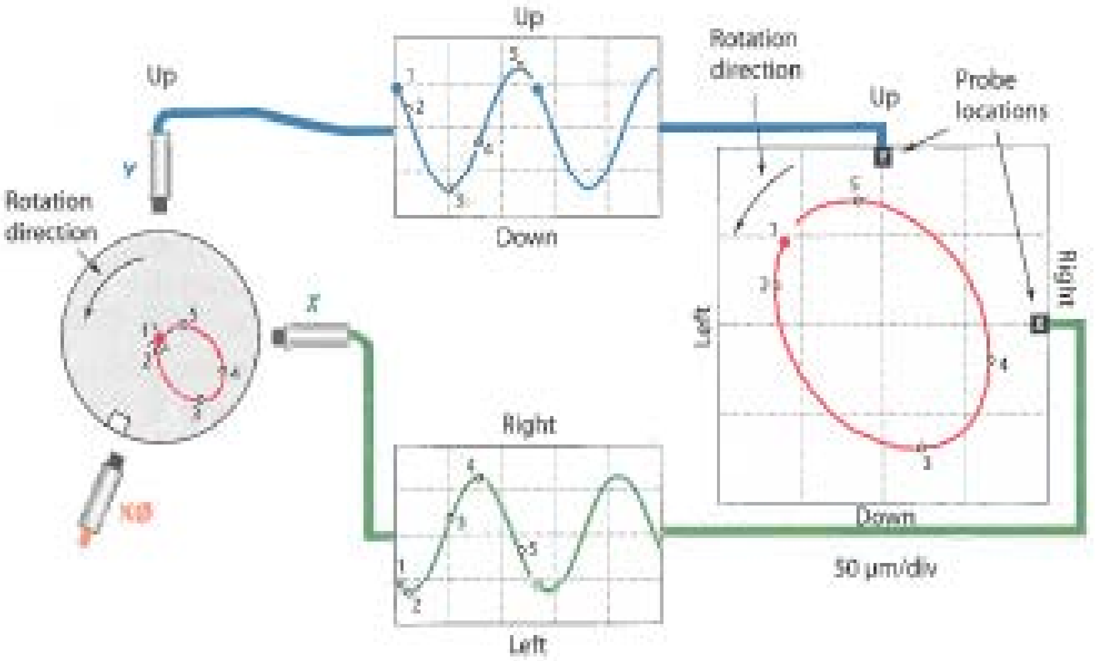


Figure 1. Location of the vibration transducers.

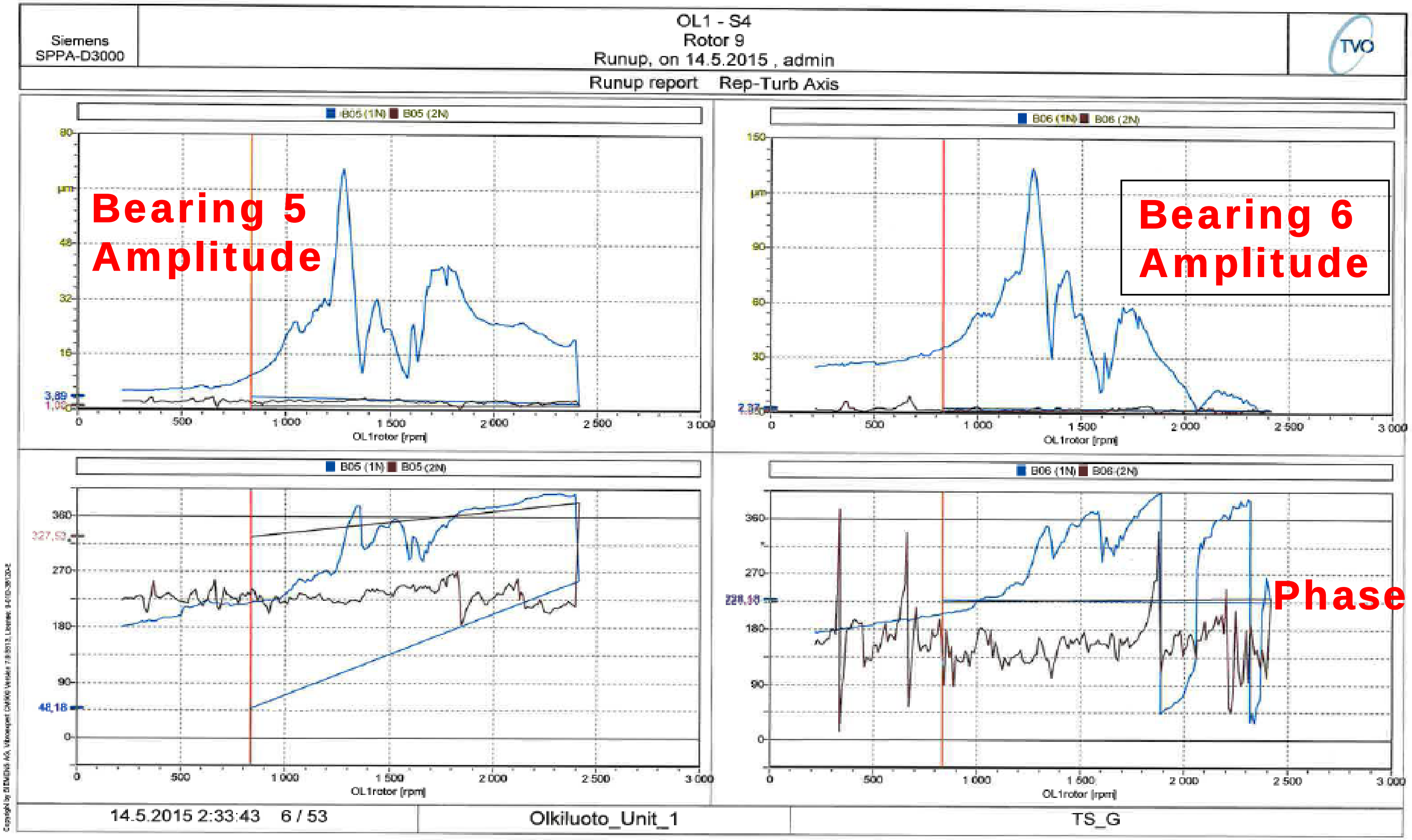
For Monitoring of High Performance Turomachinery absolute vibration velocities in mm/sec and/or relative shaft vibrations in µm at defined locations (Bearings) are usually measured. Evaluation by ISO Standards

Conventional Monitoring: Relative Vibrations of the Shaft

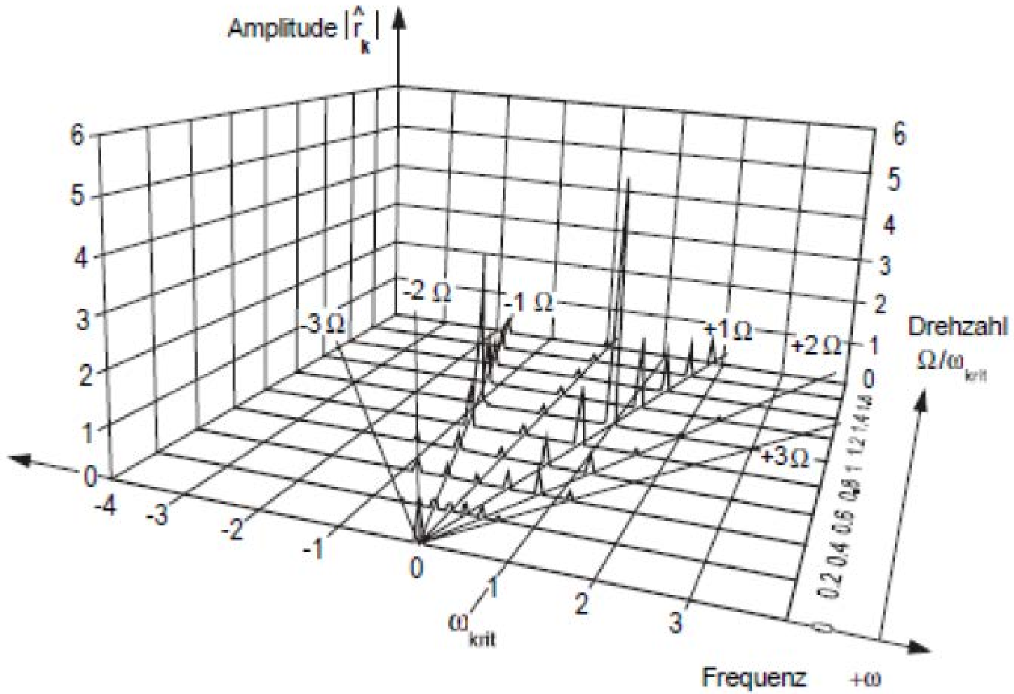
Example: Relative Vibrations of the Shaft
in horizontal and vertical direction. By Superposition
of the two signals Orbits can be determined. These
are the shaft motions in the measurement plane.



Conventional Monitoring :
Amplitudes and Phase versus speed (Run up and down)



Conventional Monitoring:
Frequency Analysis of forward and backward frequencies.



Frequency Spectra with Forward- and Backward frequencies. Both are very helpful to analyze Vibrations and to diagnose Failures.

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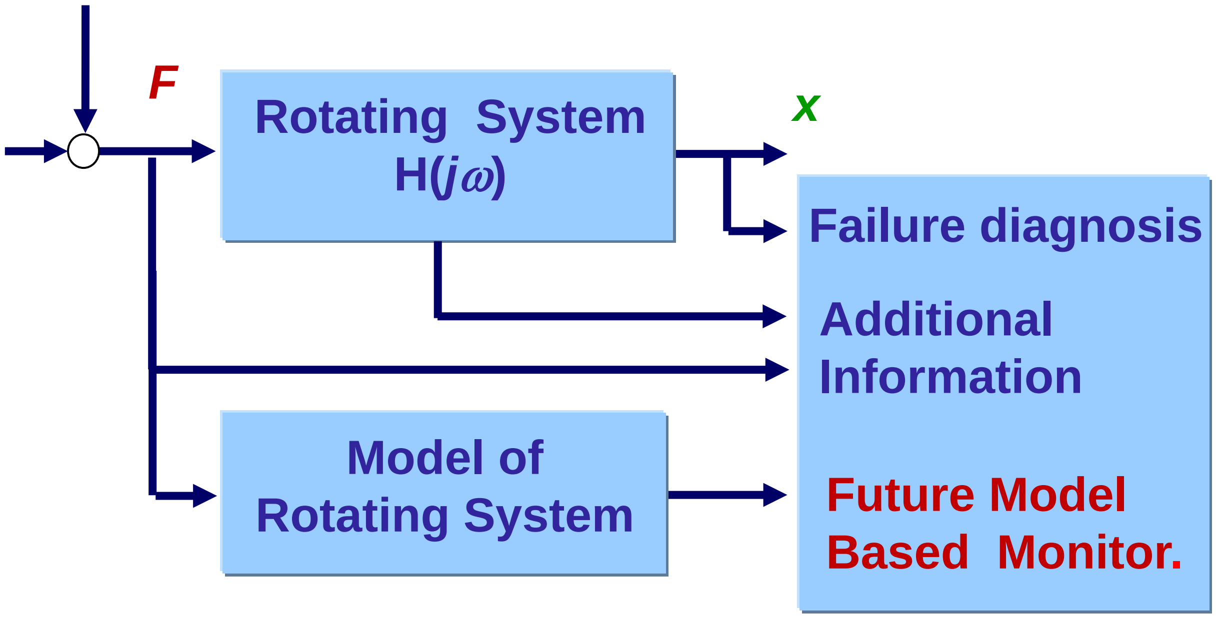
  - Conventional Monitoring
  - Future Model Based Monitoring & Diagnosis
- **Vibration Control of High Performance Turbomachinery**
  - Passive Vibration Control
  - Active Vibration Control

Future Model Based Monitoring & Diagnosis
Failure Diagnosis by means of FRF H(jω)



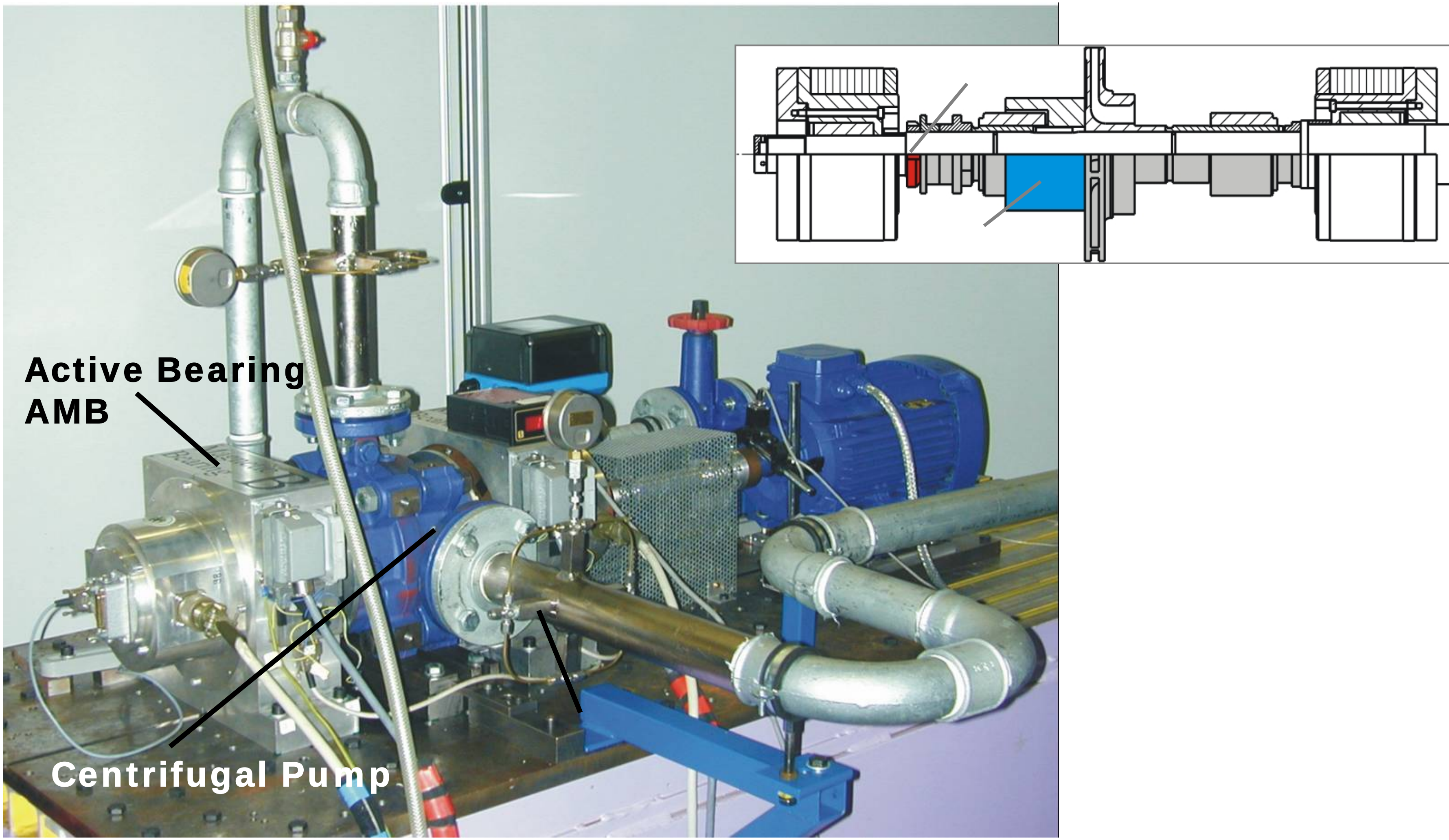
$$H(j\omega)\, F\,(j\omega) = x\,(j\omega)$$

Parameter-Estimation

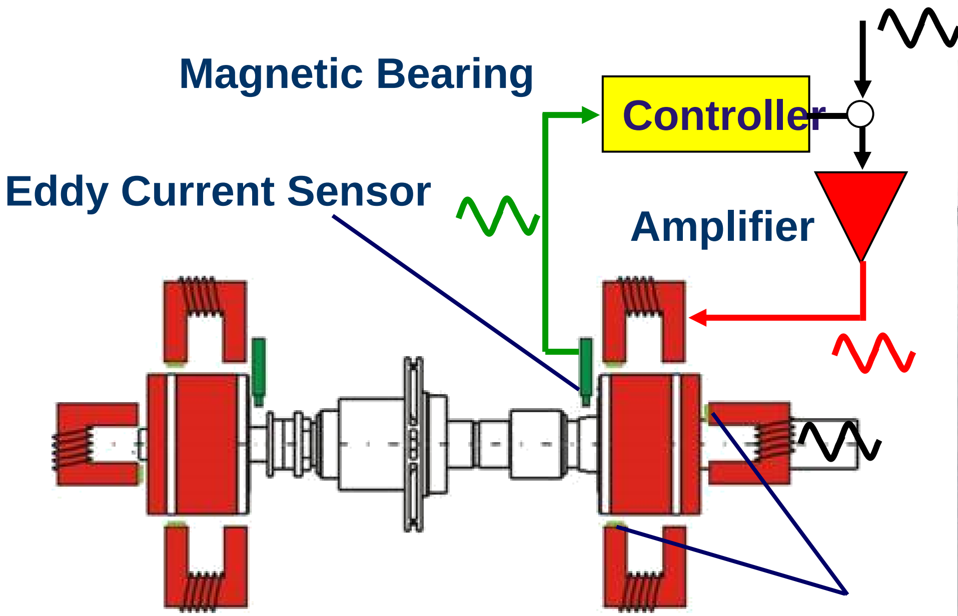
Question:

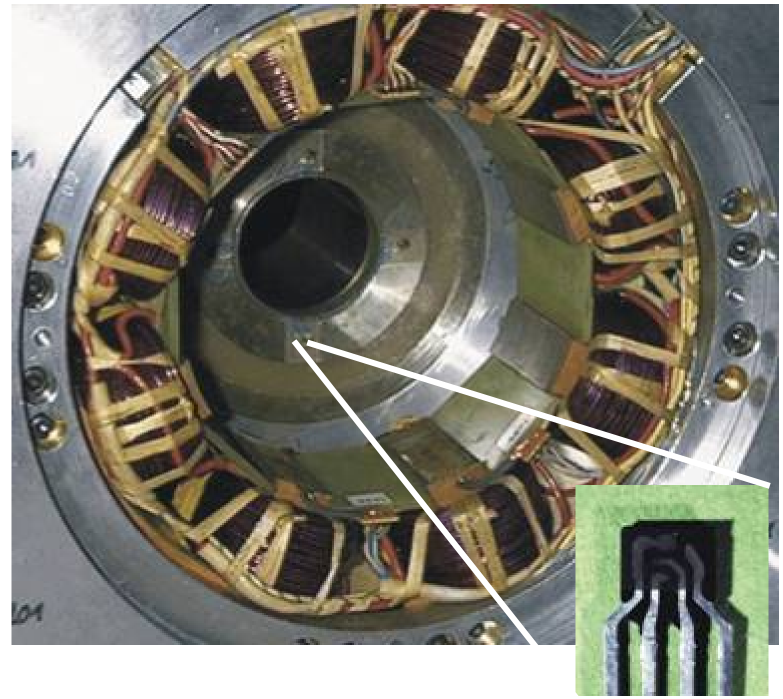
How strong is the influence of failures on the FRF H(*jω*)?

Future Model Based Monitoring & Diagnosis
Centrifugal Pump with Active Magnetic Bearings AMB's



Future Model Based Monitoring & Diagnosis
Rotating Pump Shaft with Active Magnetic Bearings





Hall Probe

Sensor Characteristic

Displacement x: Eddy Current Sensor

Force F: Hall Probe $F \sim B^2$

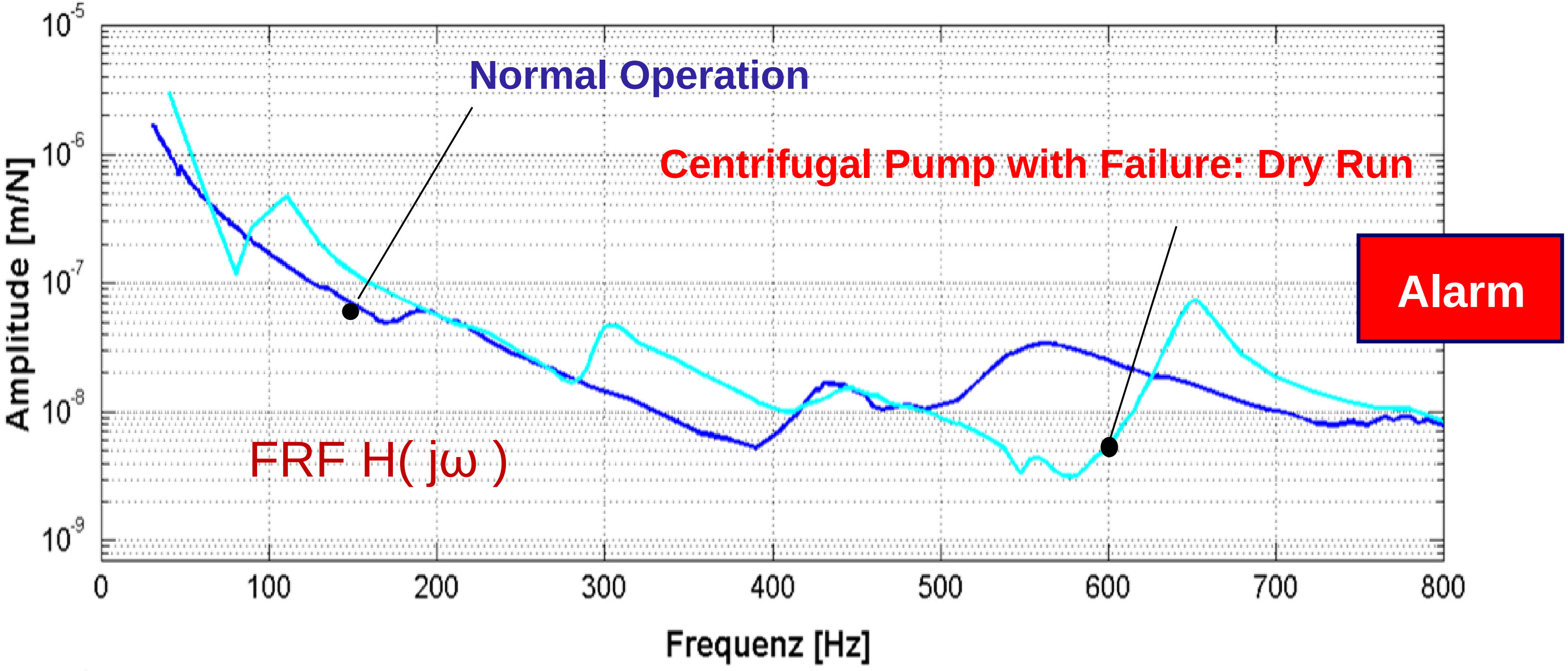
Actuator Characteristic

Excitation up to 1 kHz

Measuring of the system behaviour is possible during regular operation (rotation)
Frequency response functions $H(j\omega)$

Future Model Based Monitoring & Diagnosis.
Centrifugal Pump Shaft with Active Magnetic Bearings

Failure Diagnosis in a Centrifugal Pump with FRF H(jω)



Modelling, Monitoring and Vibration Control

Modelling of High Performance Turbomachinery

Monitoring of High Performance Turbomachinery

Passive and Active Vibration Control of High Performance Turbomachinery

Modelling, Monitoring and **Vibration Control of High Performance Turbomachinery**

- **Introduction: High Performance Turbomachinery**
- **Vibration Phenomena in High Performance Turbomachinery**
- **Modelling of High Performance Turbomachinery**
  - Modelling of the Laval Shaft
  - Modelling of Large High Performance Turbomachinery
- **Monitoring of High Performance Turbomachinery**
  - Conventional Monitoring
  - Future Model Based Monitoring
- **Vibration Control of High Performance Turbomachinery**
  - Passive Vibration Control
  - Active Vibration Control

Vibration Control of High Performance Turbomachinery
Passive and Active Vibration Control

Vibration Control Measures	Without Energy Conversion	With Energy Conversion		
	Passive		semi-Active	Active
Reduction of Excitation				
System Tuning				
Damping				
Vibration Absorber				
Isolation of disturbance				
Isolation to protect the receiver				

Conventional Solution

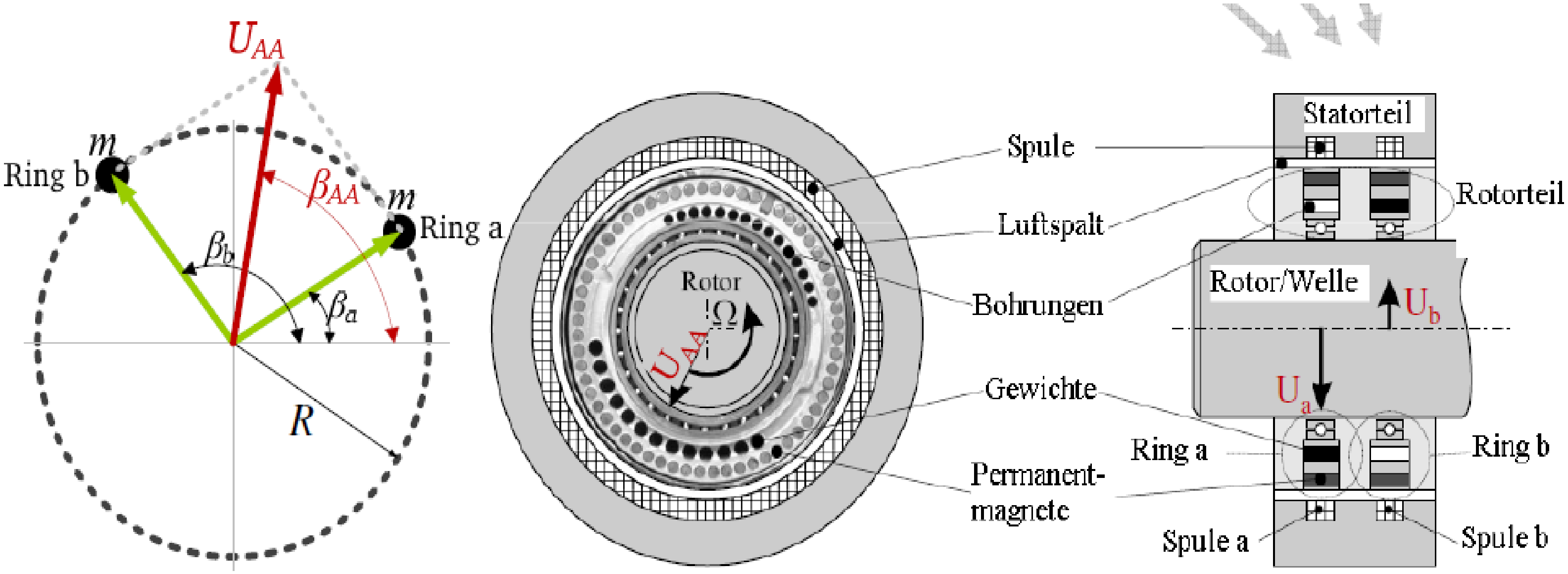
Extended Solutions

Increase of:
Effectiveness, Complexity,
more Solution Variants

Vibration Control of High Performance Turbomachinery
Passive Vibration Control: Reduction of Excitation

Balancing of LPT - Rotor

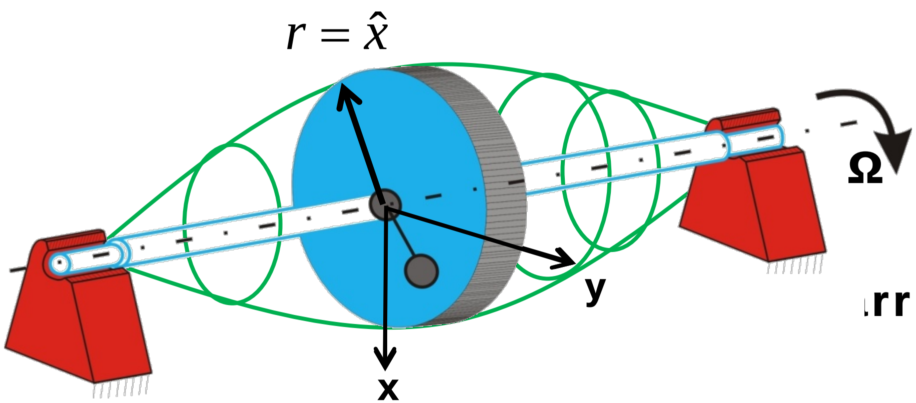
Vibration Control of High Performance Machinery
Active Vibration Control: Reduction of Excitation



Active Balancing Device

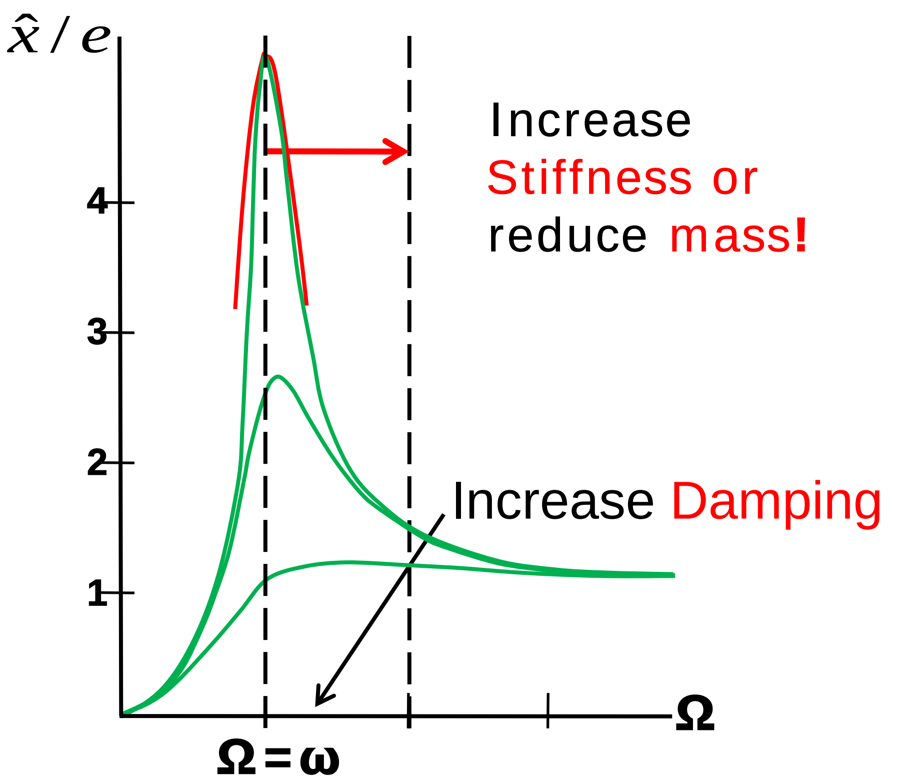
Vibration Control of High Performance Machinery
Passive Vibration Control: Tuning and Damping

Shaft Vibrations

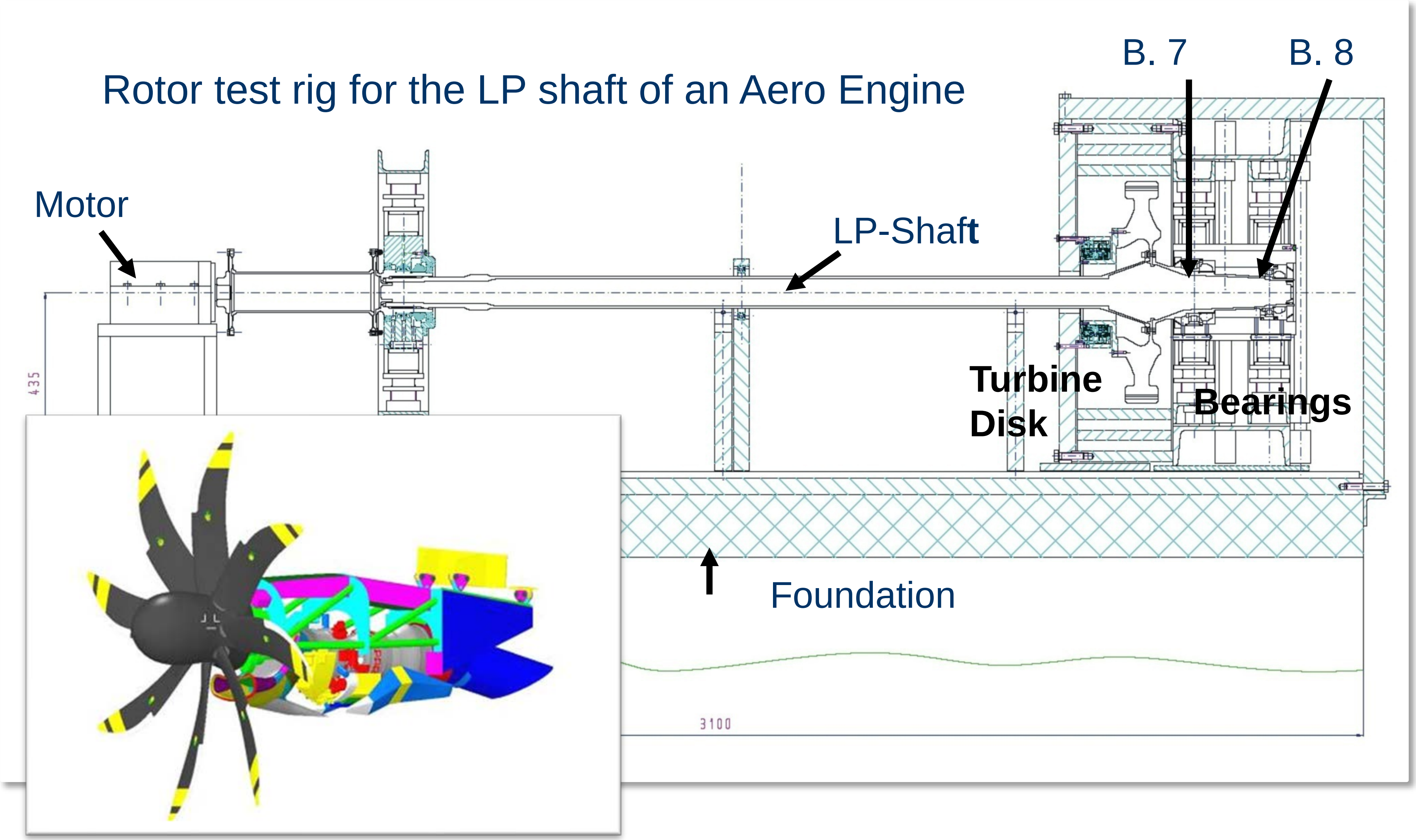


$$\left(\frac{\hat{x}}{e}\right) = \frac{m\Omega^2}{\sqrt{\left(c - m\Omega^2\right)^2 + \left(d\Omega\right)^2}}$$

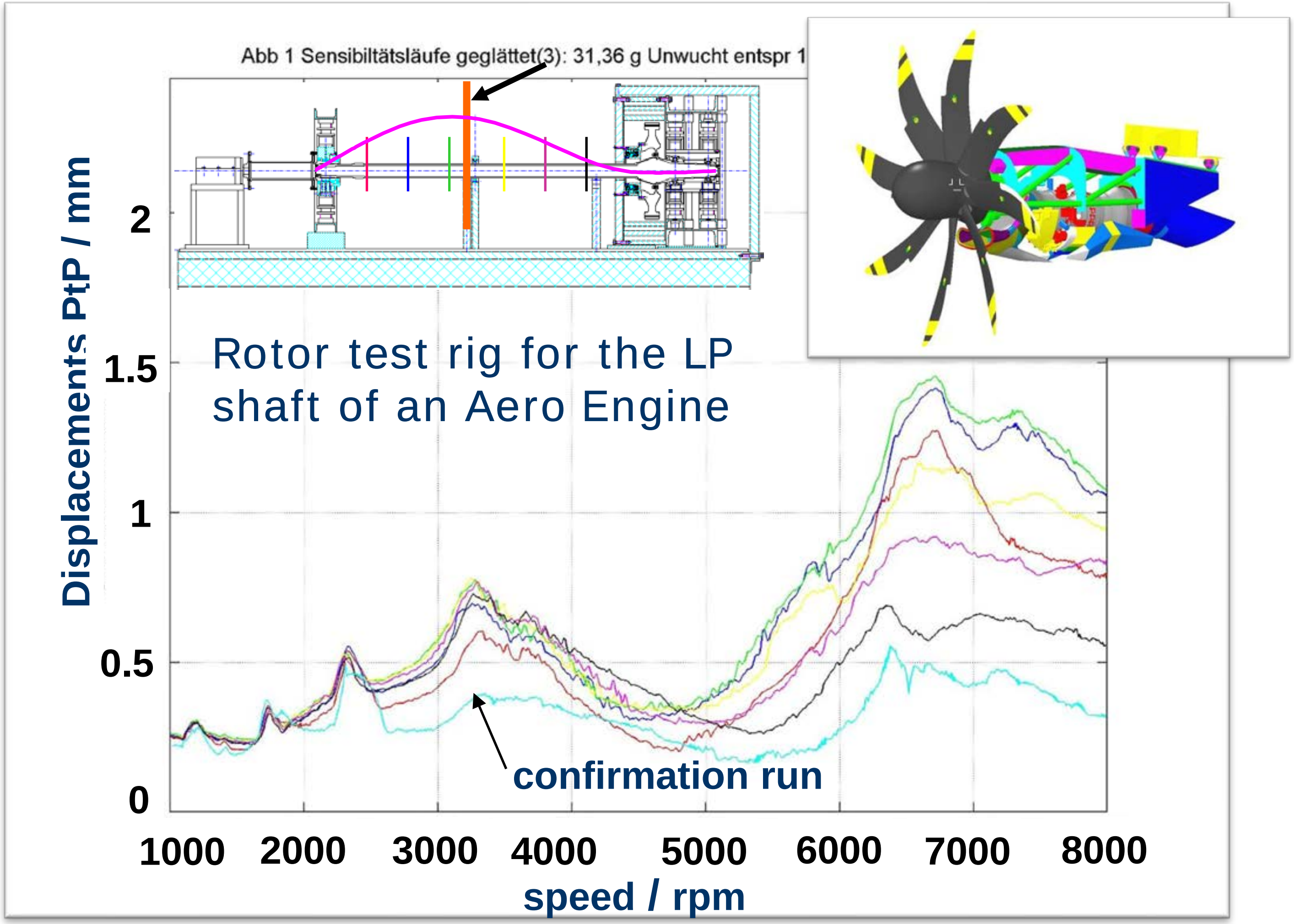
Amplitude



Vibration Control of High Performance Machinery
Passive Vibration Control: Squeeze Film Damping (SFD)



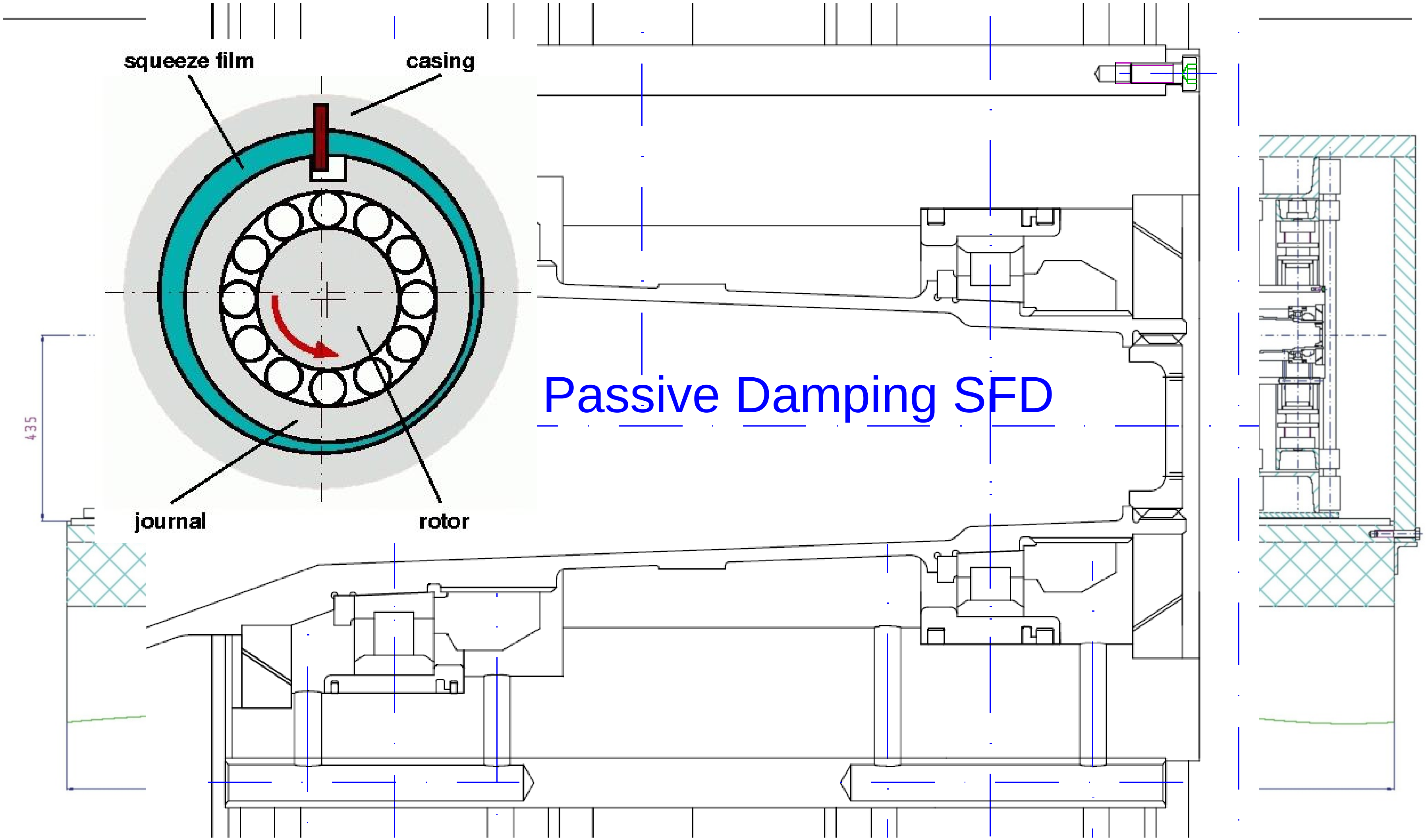
Sensitivity Run up curves of LP Turbine due to Unbalance



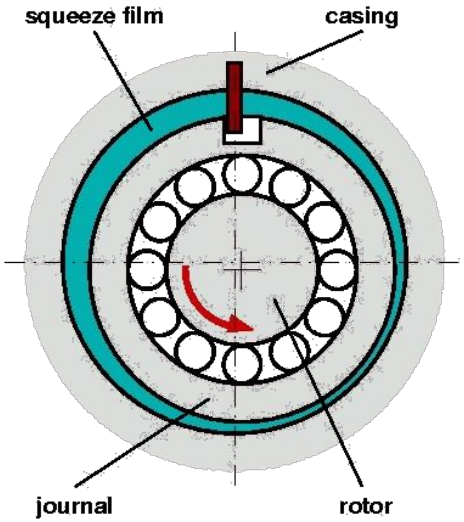
Sensitivity runs with unit unbalance at different shaft locations measured in the middle of the LP-shaft

Vibration Control of High Performance Machinery

Passive Vibration Control: Squeeze Film Damping (SFD)



Vibration Control of High Performance Machinery
Passive Vibration Control: Squeeze Film Damping (SFD)

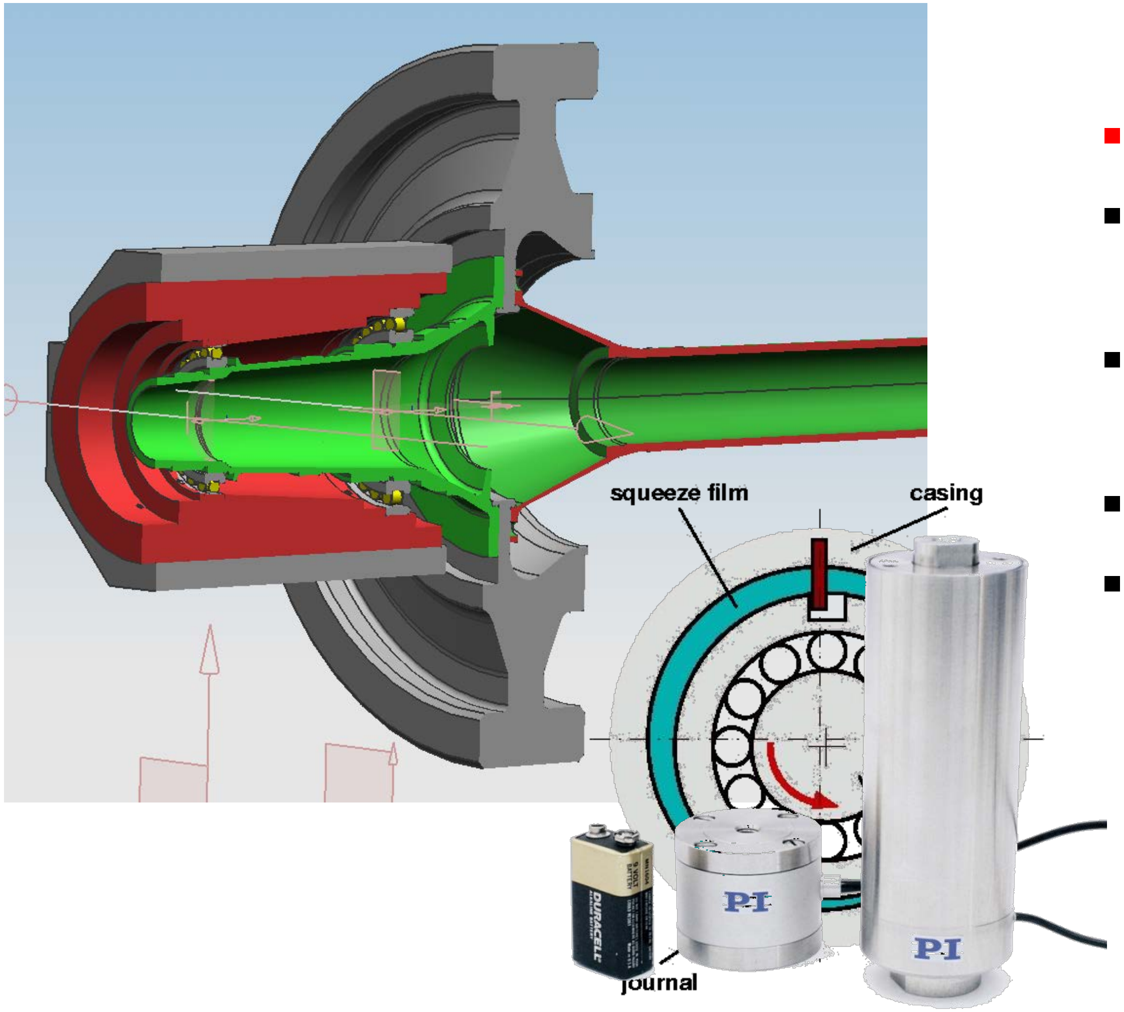


Vibration Control of High Performance Machinery

Active Vibration Control: Damping with Piezo-Actuator

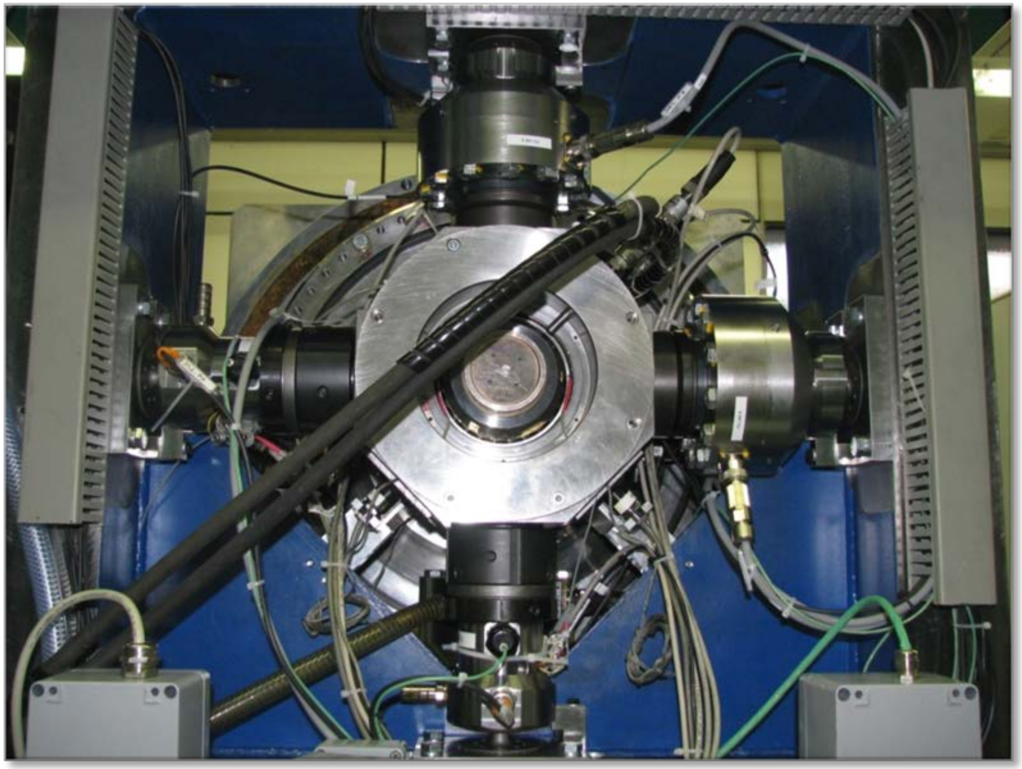
Active Damping
with Piezo Actuator

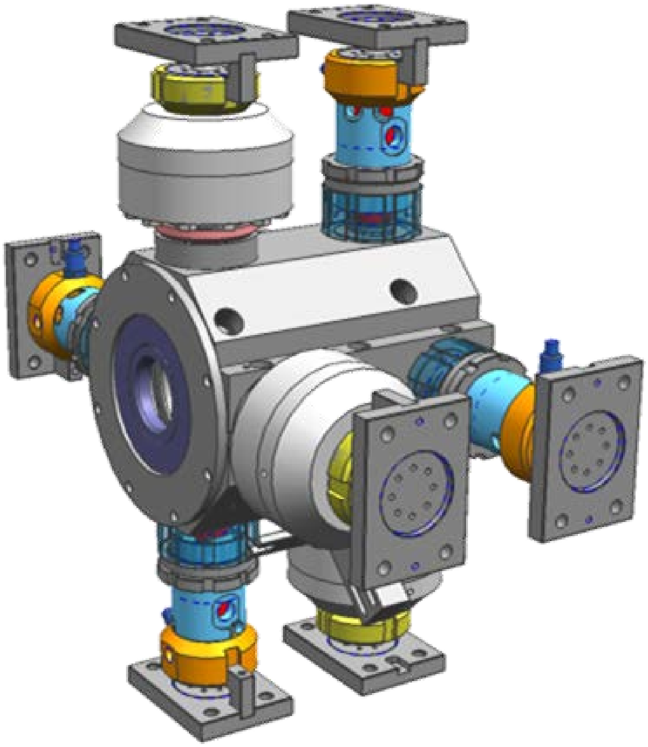
Vibration Control of High Performance Turbomachinery
Active Vibration Control: Damping with Piezo Actuator



- Active damping
- optimzation of the entire operation range
- diagnosis / failure identification
- no oil supply
- 1. step to More Electric Engine (MEE)

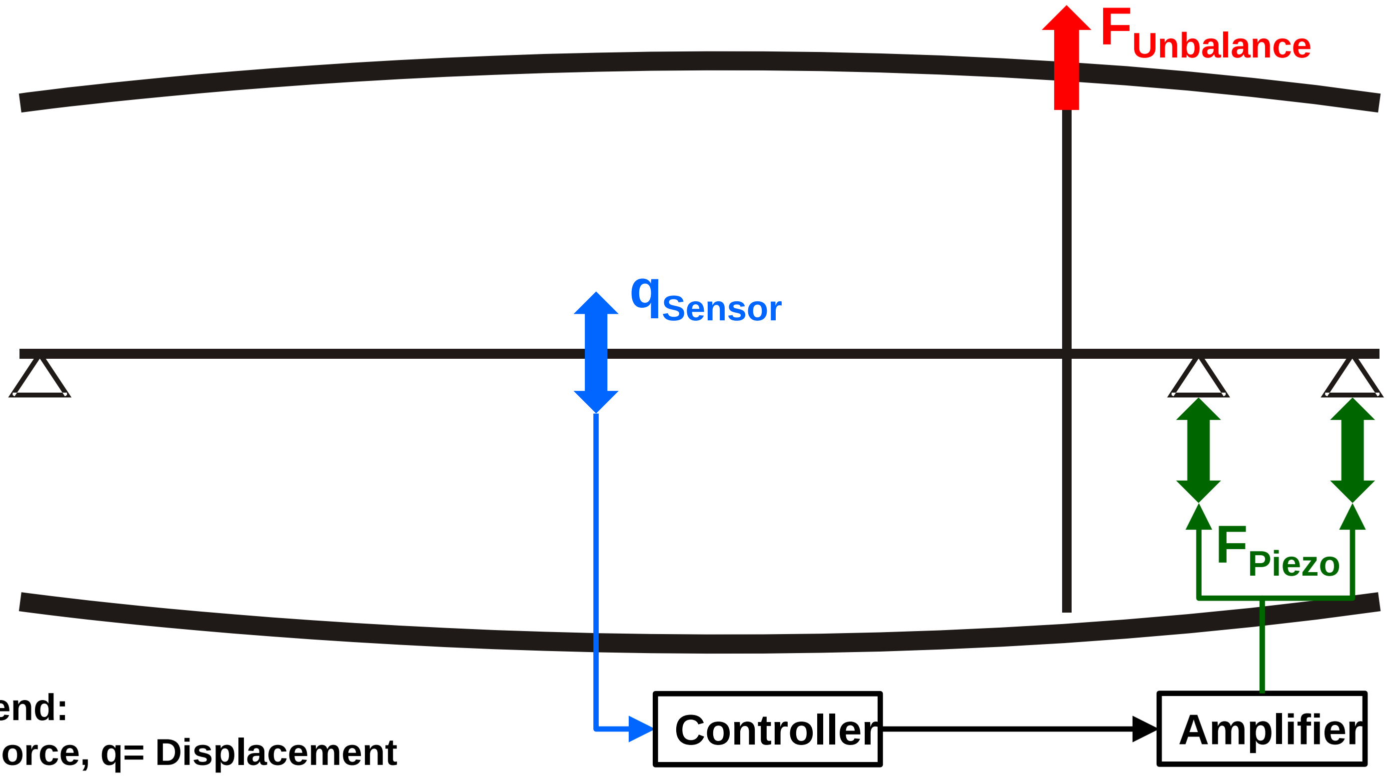
Vibration Control of High Performance Turbomachinery
Active Vibration Control: Damping with Piezo Actuator





Vibration Control of High Performance Turbomachinery

Active Vibration Control: Damping with Piezo Actuator



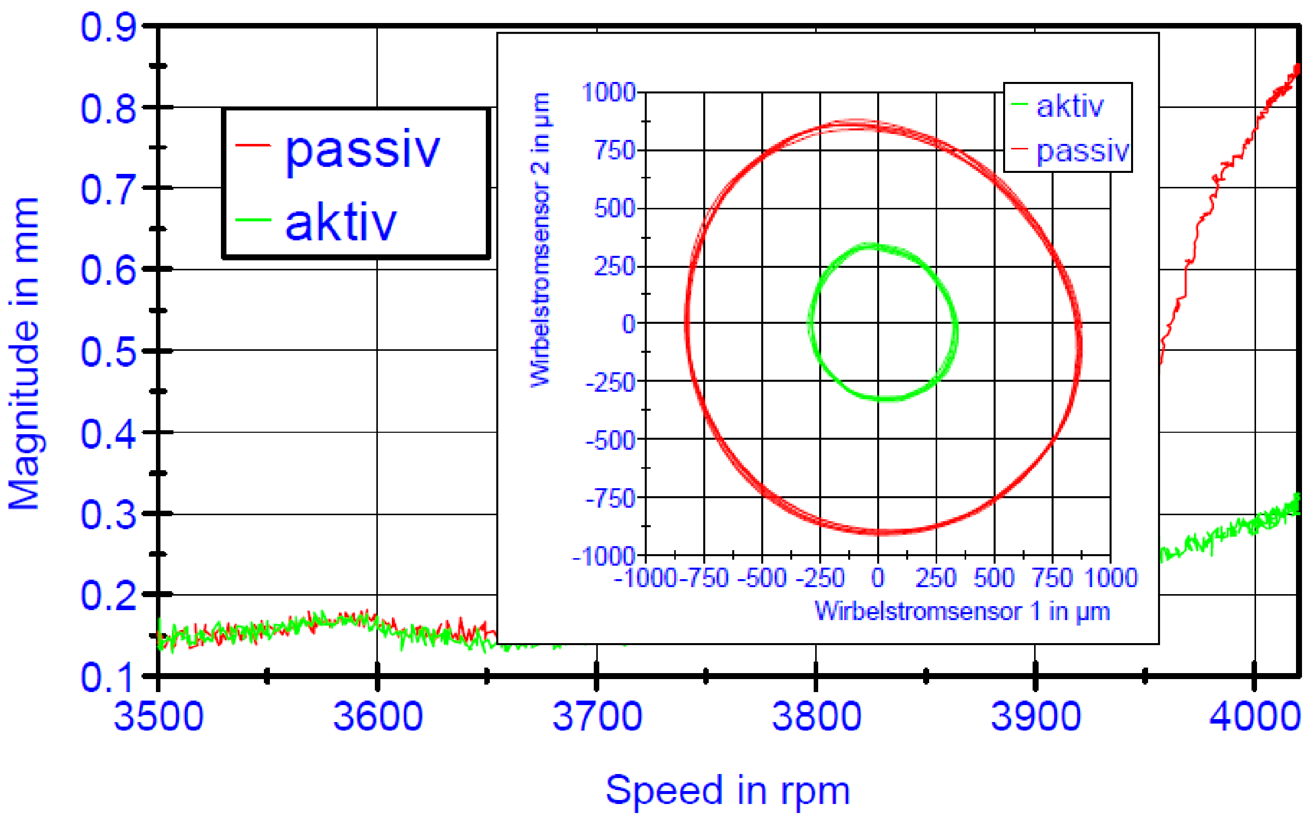
Legend:
F= Force, q= Displacement

Pros:
- **vibration damping in whole speed range**
- **fault detection**

Cons:
- **temperature-sensitive**
- **expenditure**

Vibration Control of High Performance Turbomachinery
Active Vibration Control: Damping with Piezo Actuator

Sensors: **Eddy current Sensors, Rotormitte**
Controller: **PD – Controller**



Result:

Reduction 60 %

Vibrations in Nuclear Applications 2021

TECHNISCHE UNIVERSITÄT DARMSTADT

Annual Energiforsk Seminar (10.11.21)

Modelling, Monitoring and Vibration Control for High Performance Turbomachinery

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MiM