ANALYSIS OF THE INFLUENCE COEFFICIENT MATRIX FOR ON-SITE BALANCING OF FLEXIBLE ROTORS

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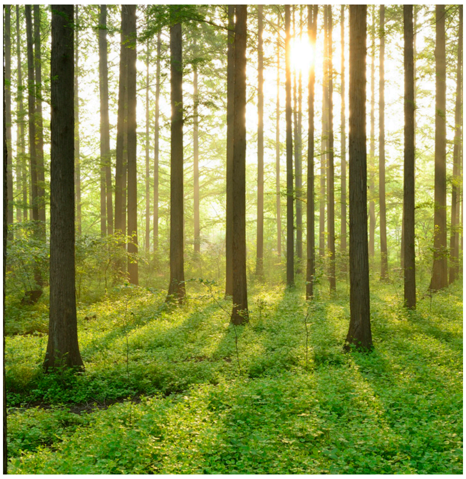

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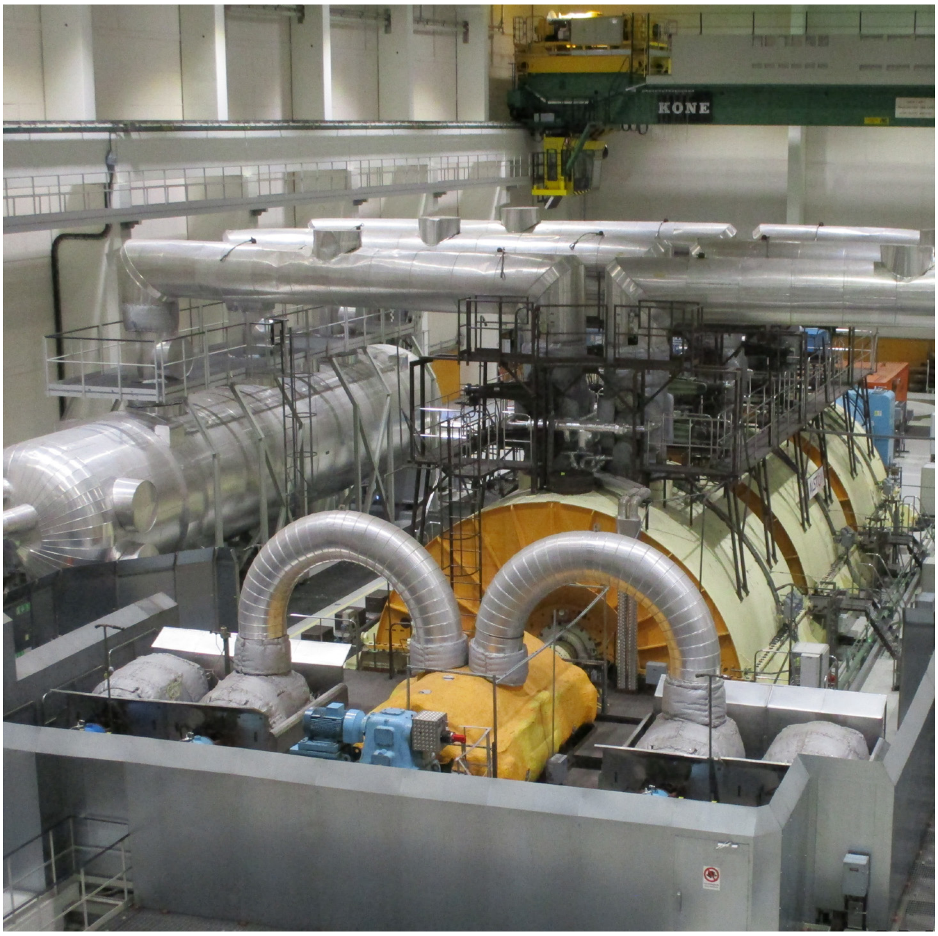

Energiforsk

Analysis of the Influence Coefficient Matrix for On-site Balancing of Flexible Rotors

Model Based Balancing

RAINER NORDMANN, ERIC KNOPF, THOMAS KRUEGER AND BASTIEN ABRATE

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Foreword

Balancing a nuclear power plant rotor by analyzing the influence of adding weights is time consuming and costly, since each plane is done separately, and it requires running the nuclear power plant up and down. If a digital twin could be created and the influence coefficients calculated, the rotor sting could instead be modeled, and compensation weights calculated.

In this project, Professor Rainer Nordmann from Technical University Darmstadt and senior researchers Eric Knopf, Thomas Krueger and Bastien Abrate from GE Power have created a digital twin of the Ringhals turbine train unit TG 32. The calculations are compared to measured results from hammer tests on the Ringhals 3 shaft performed by Åsa Collet and Jessica Fromell at Efterklang.

The project was performed within the Energiforsk Vibrations in nuclear applications program. The stakeholders of the program are Vattenfall, Uniper, Fortum, TVO, Skellefteå Kraft and Karlstads Energi.

Sammanfattning

Balansering av flexibla rotorer genom influenskoefficienter är en välkänd metod inom rotordynamikteori. Som med alla balanseringsprocedurer är den viktigaste förutsättningen det linjära förhållandet mellan obalanserade krafter och vibrationsrespons, det vill säga förskjutningar eller hastigheter och förmågan att reproducera det dynamiska beteendet under balanseringsprocessen.

Metoden kan användas för enskilda rotorer i en spin-pit, men också för godtyckliga rotoraxelsystem på plats utan antaganden rörande antal lager samt lagrens och stödens egenskaper. Därför kan rotoraxelsystem med ång- eller gasturbiner med oljefilmslager och deras särskilda styvhet och dämpningsegenskaper vara välbalanserade så länge de uppfyller de linjära kraftrörelseförhållandena. För att metoden ska fungera korrekt krävs influenskoefficienter. Procedurens första steg är att definiera koefficienterna. De definieras som vibrationssvaret, relativa axelvibrationer eller absoluta vibrationshastigheter, vid en mätpunkt som genereras av en enskild obalans i ett av balanseringsplanen. En faktisk balanseringsprocess för en flexibel rotor inleds med en mätning av vibrationssvaret på definierade mätställen. Utifrån informationen från den här mätningen och den kända matrisen av influenskoefficienter kan kompensationsvikter med amplituder och fasvinklar beräknas. De obalanserade vibrationerna minskar när de fixeras på rätt plats i balanseringsplanen. De nödvändiga influenskoefficienterna för balanseringsprocessen som beskrivs bestäms vanligen av mätningar i så kallade influenskörningar med definierade testvikter. För en vald konstant rotorhastighet måste en enskild testvikt appliceras i ett balanseringsplan, och vibrationssvaren måste fastställas för de fördefinierade mätgraderna av frihet. En influenskoefficient definieras sedan som svaret vid en grad av frihet dividerat med den tillämpade enskilda obalansen. Det experimentella förfarandet för att fastställa influenskoefficienterna kan vara mycket tidskrävande och kostsamt och kräva flera testkörningar med stopp, omstarter och nedkylningsprocesser. Influenskoefficienter kan dock också fastställas genom numerisk analys. Det här kräver givetvis en mycket god modell för rotoraxelsystemet, inklusive samtliga viktiga dynamiska rotoreffekter.

I det här projektet genomförs en prövning i syfte att definiera influenskoefficienter genom modellering och numerisk simulering. Den fullständiga proceduren för modellbaserad balansering (Model Based Balancing, MBB) består sedan av följande steg:

- Insamling av alla viktiga turbinaxeldata för olika axelkomponenter, lager, piedestaler och stöd.
- Modellering av det fullständiga turbinaxelsystemet med alla styvhets-, dämpnings- och tröghetseffekter.
- Definition av balanseringsplan och mätplan.
- Numerisk analys av systemsvaren för alla mätplan på grund av generering av testvikter för valda rotationshastigheter.

- Beräkning av influenskoefficienternas matris från systemsvaren och genererade testvikter.
- Balanseringsprocess: Mätning av vibrationssvaret vid definierade mätplatser för en vald rotationshastighet. Utifrån informationen från den här mätningen och den kända matrisen av influenskoefficienter som motsvarar den valda rotationshastigheten, kan kompensationsvikter med amplituder och fasvinklar beräknas.
- Tillämpning av beräknade kompensationsvikter på rätt plats i balanseringsplanen, för att minska vibrationer på grund av obalans.
- Med den beskrivna proceduren är det möjligt att åstadkomma en kraftig minskning av antalet balanseringskörningar. I bästa fall bör det räcka med en balanseringskörning för att balansera turbinaxelsystemet.

Summary

Balancing of Flexible Rotors by means of Influence Coefficients is a well-known method in the theory of Rotor Dynamics. As for any balancing procedure, the most important prerequisite is the linear relationship between unbalance forces and vibration responses, e.g. displacements or velocities and the ability to reproduce the dynamic behaviour during the balancing process. The method is applicable for single rotors in a spin pit but also for arbitrary rotor trains on-site with no assumptions regarding the number of bearings and the bearing- and support-characteristics. Therefore steam turbine or gas turbine rotor trains with oil film bearings and their special stiffness and damping behaviour can be well balanced, as long as they fulfil the linear force motion-relationships. For a successful application of the method Influence Coefficients are needed and have to be determined in a first step of the procedure. They are defined as the vibration response, relative shaft vibrations or absolute vibration velocities at bearing pedestals, at one measurement point excited by a single unbalance in one of the balancing planes. An actual balancing process for a flexible rotor starts with the measurement of the vibration response at defined measurement locations. With the information from this measurements and the known matrix of Influence Coefficients, compensation weights with amplitudes and phase angles can be calculated. When they are fixed at the right location of the balancing planes, the unbalance vibrations will be reduced. The required Influence Coefficients for the described balancing process are usually determined by measurements in so called influence runs with defined test weights. For a selected constant rotor speed a single test weight has to be attached in a balancing plane and the vibration responses have to be determined for the predefined measurement degrees of freedom. An Influence Coefficient is than defined as the response at one degree of freedom divided by the applied single unbalance. The experimental procedure to determine the Influence Coefficients can be very time consuming and expensive, where several test runs with stops, restarts and cooling down processes will be needed. However, Influence Coefficients can also be determined by means of a Numerical Analysis. This needs of course a very good model for the rotor train, the oil film bearings, the pedestals and the foundation, including all important dynamic effects. In this project a trial is made to determine Influence Coefficients by means of modelling and numerical simulation. The complete procedure of Model Based Balancing (MBB) then includes the following steps:

- Collection of all important turbine train data for the different part shafts, the oil film bearings, the bearing pedestals and the foundation.
- Modelling of the complete turbine train with all stiffness-, damping- and inertia effects of the components: shaft train, oil film bearings, pedestals and foundation.
- Definition of the balancing (compensation) planes and the measurement planes.
- Numerical Analysis of the system responses for all measurement planes due to each test weight excitation for selected rotational speeds, e.g. the operational speed.

- Calculation of the matrix of Influence Coefficients from the system responses and the unbalances of the test weights.
- Balancing process: Measurement of the vibration response at defined measurement locations for a selected rotational speed. With the information from this measurement and the known matrix of Influence Coefficients - corresponding the selected rotational speed- compensation weights with amplitudes and phase angles can be calculated for the compensation planes.
- The calculated compensation weights have to be attached at the right location of the balancing planes, in order to reduce the vibrations due to unbalance.
- With the described new procedure, a strong reduction of the number of balancing runs will be achieved. In the ideal case, one single balancing run should be sufficient to balance a turbine train.

Expressions

Circular Orbit	Orbit of lateral vibrations with a circular shape
Digital Twin	A Rotor Dynamic model close to reality
Gümbel curve	Static equilibrium curve for a journal in the bearing
Key phasor	Electric pulse or trigger, which is derived from a location on a rotating shaft. It serves as a reference for zero phase
On-Site Balancing	Balancing of the turbine train in the power plant
Polar plot diagram	Presentation of amplitude and phase in a polar plot
Proximity probe	Sensor type to measure relative shaft vibrations
Reference Run	Run of turbine train with the residual unbalance
Spin Pit	Balancing tunnel to balance part rotors
Sommerfeld Number	Dimensionless oil film bearing force (see (3), page 38)
Test Run	Run of turbine train with unbalance from test weight
Youngs Modulus	Material number, characterizing the material stiffness

Abbreviations

IC	Influence Coefficient
MBB	Model Based Balancing
FRF	Frequency Response Function
1xN	Expression for Vibration with rotational frequency Ω
HPT	High Pressure Turbine
LPT	Low Pressure Turbine
GEN	Generator
NPP	Nuclear Power Plant
MW	Power in Mega Watt
FEM	Finite Element Method

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1 Introduction – Project Description

1.1 OBJECTIVE OF THE PROJECT

The main objective of this project is to analyze the matrix of Influence Coefficients (IC) for turbine trains in Nuclear Power Plants (NPP) by means of FE-modelling and numerical simulation. The numerically determined Influence Coefficients will be evaluated and compared with measured data. The idea is, that the calculated IC's can be used for flexible rotor on-site balancing for the corresponding turbine train instead of determining them in several, time consuming test runs.
The project results are described in this report, which can be used for future on-site balancing processes in turbine trains, especially when changes of turbine train configurations are planned. The project results can also be used for the knowledge transfer to new personnel.

1.2 SCOPE OF THE TASK

The scope of the complete task can be described by the following subtasks:

Introduction of the method of flexible rotor balancing by means of Influence Coefficients

Determination and Interpretation of Influence Coefficients
Numerical calculation of Influence Coefficients with a rotor dynamic finite element model. Estimation of Influence Coefficients by simulated test weight settings. Interpretation of Influence Coefficients.

Selection of a NPP and a turbine train unit
A NPP and a unit shall be selected with the system components (turbines, generator, bearings, etc.) from one manufacturer only. Otherwise the modelling of the dynamic system becomes difficult.

Collection of the turbine train data for the numerical analysis
Geometrical data of the rotor train, stiffness and damping of the oil film bearings, mass and stiffness of the bearing pedestals and the supporting system, e.g. the foundation. Material data and temperature distribution. Location of the vibration sensors and definition of the balancing planes for the investigation.

Modelling of the turbine train with all components
Modelling of all important inertia, damping and stiffness parameters for the different components of the turbine train. Definition of the measurement planes (sensors) and balancing planes (test and compensation weight locations). Collection of site information regarding probe location and angular position, keyphasor, etc.

Analysis of the system responses in all measurement planes
Calculation of the vibration responses for each test weight in the balancing planes for relevant speeds (e.g. the operational speed). Selection of the balancing speeds should be made, based on the calculation results and available run up/run down data.

Calculation of the matrix of Influence Coefficients
IC's can be calculated from the system responses and the test weight excitations for selected speeds, preferably for the operational speed.

Evaluation of the analyzed Influence Coefficients
Comparison of the calculated Influence Coefficients with the measured Influence Coefficients. Decision on way forward (assess the differences, update of model). Presentation of final results for the Influence Coefficients.

2 Balancing of Flexible Rotors by means of Influence Coefficients

2.1 THEORETICAL BACKGROUND

2.1.1 Definition of Influence Coefficients as output/input relations

Influence Coefficients are output/input ratios as described in Figure 1. Input is an unbalance acting in one of the balancing planes and the output can be a relative or absolute vibration displacement or an absolute vibration velocity at one response degree of freedom, usually at the bearing locations.

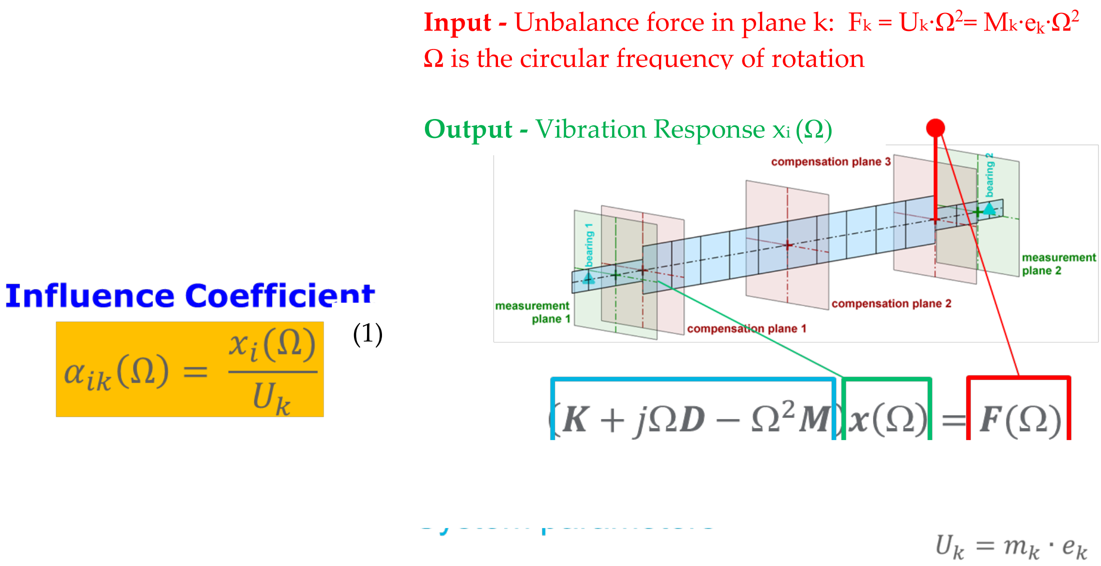


Figure 1 Influence Coefficient as output/input relation for a simplified rotor system

If 1xN-lateral vibrations in a shaft train due to unbalance are too high, related to balancing qualities or vibration amplitudes, defined in standards, e.g. ISO 20816-2, measures have to be undertaken to reduce these vibrations. Vibrations can be influenced either by system parameters (masses and stiffness of the shaft along the turbine train and stiffness and damping parameters of the other components) or by excitation parameters, e.g. by balancing weights to compensate the original high unbalance vibrations.

The most often used mitigation of a vibration problem in case of 1xN-lateral vibrations in shaft trains is balancing, where balancing with Influence Coefficients is a well-known and approved method. This method is also used in the NPP's in Sweden and Finland. In order to briefly describe the procedure, we use the simplified rotor system (Figure 2), which can be considered as a part rotor, running in two bearings (for example a turbine rotor or a generator rotor).

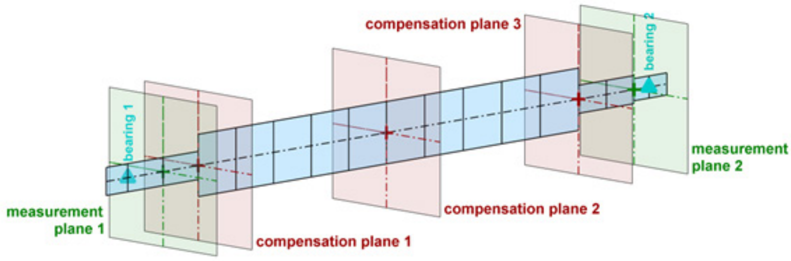


Figure 2 Balancing of a simplified rotor in two bearings by means of Influence Coefficients (three compensation planes, two measurement planes).

In the example three compensation planes are selected, where compensation weights can be attached. The unbalance vibrations can be measured in two measurement planes. Usually these planes are at the location of the bearings.

Vibration responses of the presented rotor system will be the base to obtain Influence Coefficients. They can be determined experimentally by applying test weights (unbalances) at the degrees of freedom k and by measuring the displacements or velocities (amplitude and phase) at the degrees of freedom i:

$$\alpha_{ik} = \alpha_{ik\ Re} + j \cdot \alpha_{ik\ Im} = \frac{\hat{x}i(\Omega)}{Uk} \exp\left(j \cdot \Delta_{ik}\right) \qquad (1)$$

The complex Influence Coefficients with amplitude and phase (equ. (1)) depend on the parameters mass, damping and stiffness of the shaft, the fluid film parameters of the bearings, the mass, damping and stiffness of the supporting system with pedestals and foundation and on the rotational frequency Ω. In the numerical procedure the Influence Coefficients can be calculated from the vibration responses due to unbalance (test weights). The dimension of the matrix of Influence Coefficients is dependent on the number of vibration response measurements (number of rows) and on the number of compensation planes (number of columns). In the presented example we have introduced 3 compensation planes and 2 planes for the displacement response measurements. Therefore the matrix in Figure 3 contains 6 Influence Coefficients, if only one measurement (vertical or horizontal) is considered for one measurement plane. As shown in Figure 3 the matrix of Influence Coefficients describes the relation between the 3 unbalances in the planes k = 1,2,3 and the 2 measured response displacements in planes i = 1,2. This matrix with the Influence Coefficients is the base for the balancing process.

$$\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} \alpha_{11} & \alpha_{12} & \alpha_{13} \\ \alpha_{21} & \alpha_{22} & \alpha_{23} \end{bmatrix} \begin{bmatrix} U_1 \\ U_2 \\ U_3 \end{bmatrix} \quad (2)$$

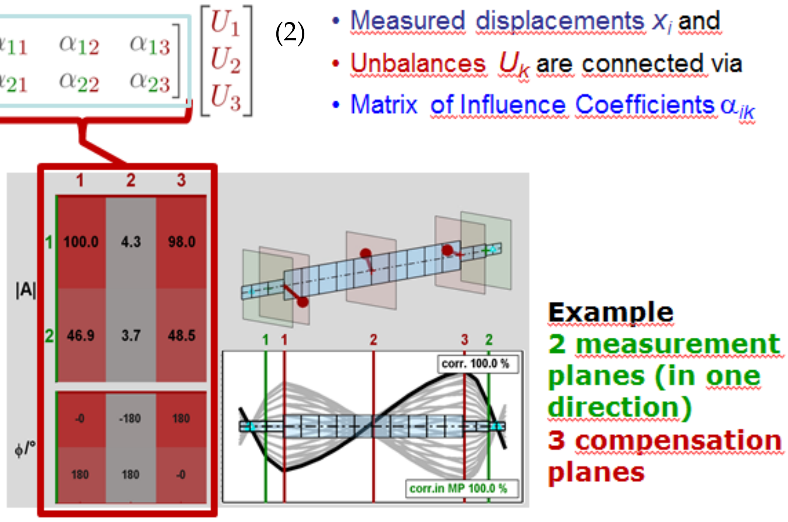


Figure 3 Matrix of Influence coefficients for a simplified rotor in two bearings with three compensation planes and two measurement planes

The determination of the Influence Coefficients of a shaft train in a power plant by measurements can be very time consuming. However, due to the fact, that modern numerical FE-models are well developed and consider all important effects (e.g. shaft-, oil film bearing-, pedestal- and foundation dynamics) numerical procedures can also be used to determine the required set of Influence Coefficients relatively fast and without impact on the availability of the unit. This will be shown in the following chapters.

2.2 PRACTICAL APPLICATION

2.2.1 Example: Experimental Determination of Influence Coefficients for a Generator Rotor

The following example of a Generator rotor demonstrates, how the Influence Coefficients usually are determined experimentally in different test runs. In a first reference run "1" with a defined rotational speed the vibrations are measured in the two measurement planes for the case of the residual unbalance (without additional weights) in the Generator rotor. The vibrations are presented in two polar plots with amplitude and phase (green arrows) for the two measurement planes (Figure 4), they are representing the amplitudes that should be minimized during balancing.

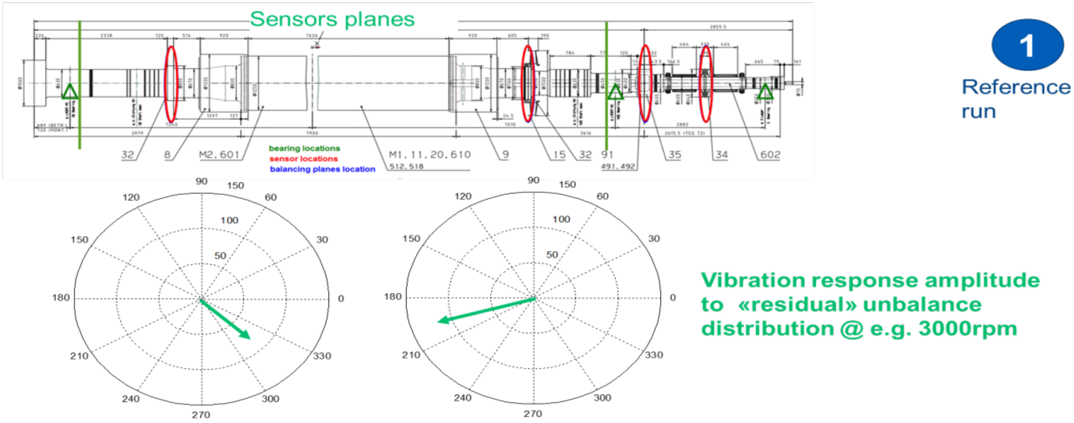


Figure 4 Vibration Response in Polar Plots for two measurement planes for the reference run with the unknown residual unbalances in the Generator rotor.

Figure 5 shows the vibration response in the polar diagrams for the two measurement planes, resulting from the unknown residual unbalance plus the unbalance from a test weight, which is attached in the shown red balancing plane. The red arrows in the two polar diagrams show the vibrations in terms of amplitudes and phase. The rotational speed in this second test run 2 has of course to be the same as in the reference run 1.

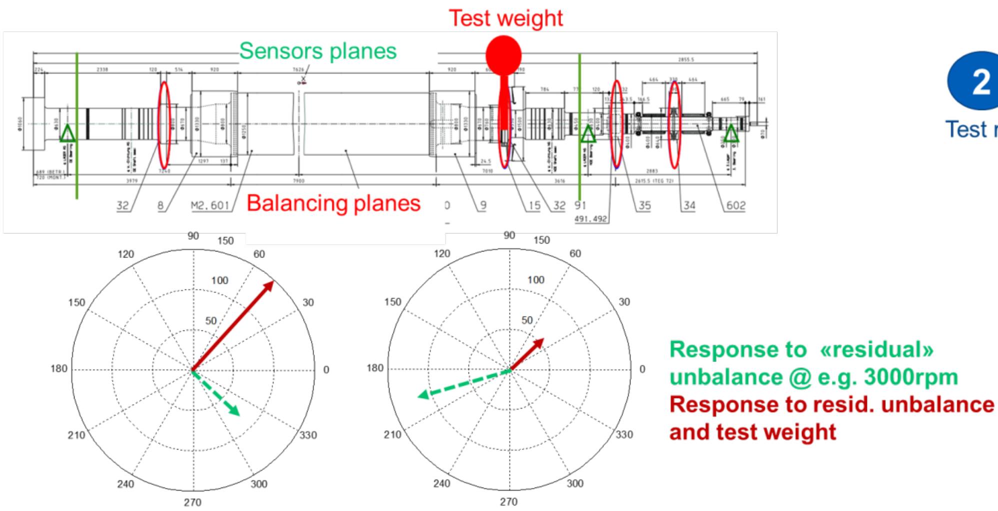


Figure 5 Vibration response in Polar Plots for the two measurement planes from the residual unbalance and the test weight in the red balancing plane (test run 2).

From the vector difference of the two vectors (green and red) the influence of the test weight on the vibration response can now be determined (light blue arrows in Figures 6 and 7).

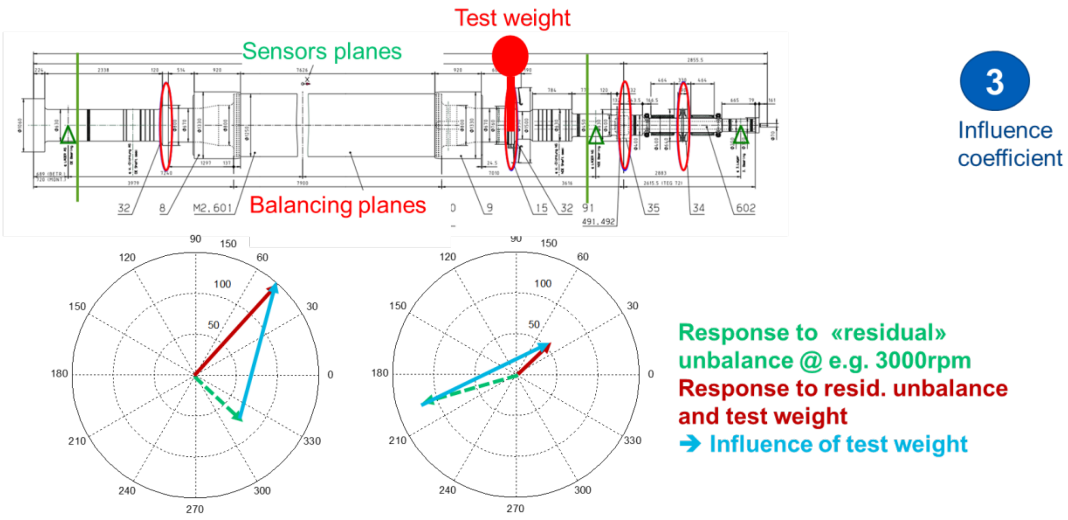


Figure 6 Vibration response in Polar Plots for the two measurement planes from the residual unbalance and the test weight in the red balancing plane. The light blue arrow shows the influence of the test weight on the vibration response.

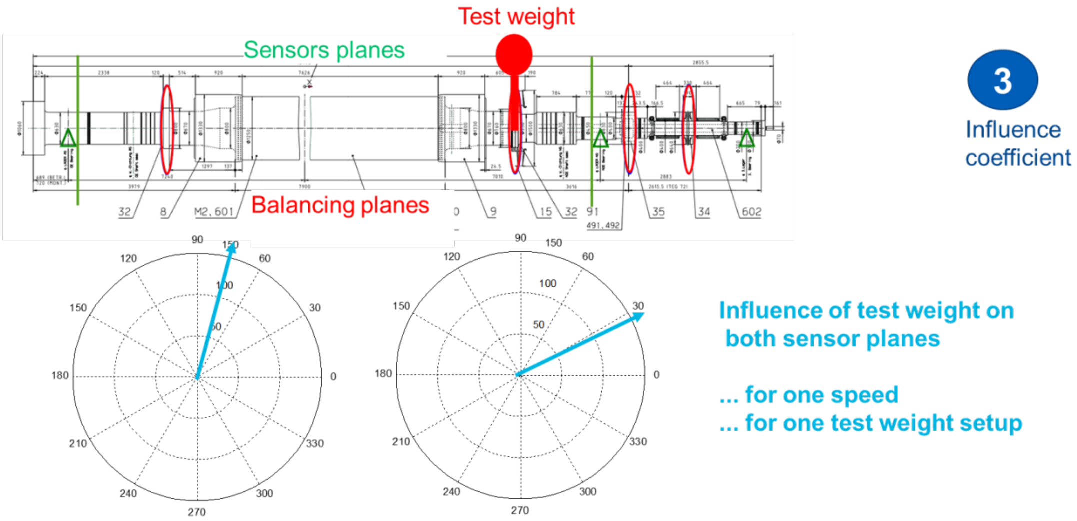


Figure 7 Vibration response in Polar Plots for the two measurement planes. The light blue arrows show the influence of the test weight on the vibration response.

By this example it has been demonstrated, how two of the Influence Coefficients (IC's) can be determined experimentally for the case of an excitation of a balancing weight in one of the compensation planes. In a similar way all other IC's can be found for the system.

2.2.2 Example: Balancing of a Flexible LPT Rotor in a Spin Pit

If the matrix of Influence Coefficients (amplitudes and phase angles) is known – experimentally determined in test runs or by numerical calculations - the Influence Coefficients can be used to determine compensation weights, based on measured vibration responses at the running machine with speed Ω. As shown in the simple example of Figure 3, the set of compensation weights can be determined out of the shown matrix relation between the measured vibration responses and the unbalances. In order to balance the rotor, the determined compensation weights have to be attached to the shaft at the compensation locations. The described procedure is applicable to balance part rotors in a spin pit, but can also be used for on-site balancing in power plants, when the corresponding Influence Coefficients of the shaft train are available. Figure 8 shows as an example a separated LPT steam turbine rotor of a shaft train (operational speed 1500 rpm) in preparation for balancing in a spin pit. On the right side of figure 8 the location for balancing weights in a circumferential groove close to the last stage blades can be seen. As a result of the balancing of the LPT steam turbine rotor the vibration amplitudes at the bearing in µm (0-peak) are presented as a run up curve (0–1500 rpm) in Figure 9. The amplitudes of the relative shaft vibrations at the bearings are less than 7 µm at the critical speed of 600 rpm. At operational speed (1500 rpm) the relative amplitudes are about 2 µm.

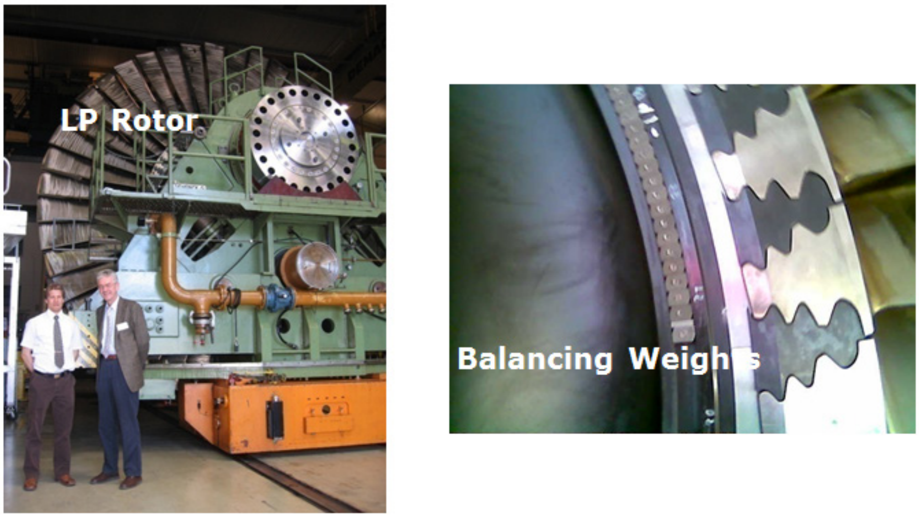


Figure 8 Preparation of a LPT Steam turbine rotor for balancing in a spin pit

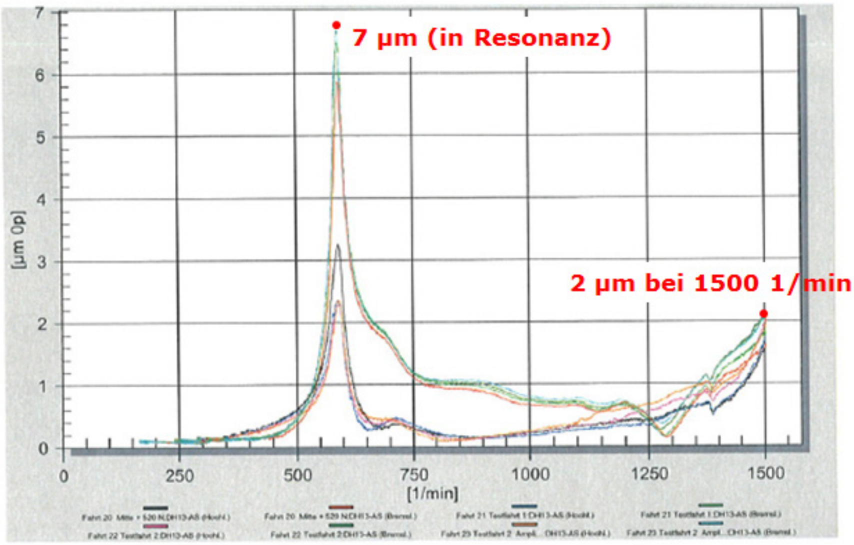


Figure 9 Vibration amplitudes (0-peak) of a LPT steam turbine at Run-up after balancing in the Spin Pit

3 Two Strategies: Balancing with measured or calculated Influence Coefficients

3.1 TWO STRATEGIES FOR BALANCING WITH INFLUENCE COEFFICIENTS

In the life time of a turbine train rotor balancing usually cannot be avoided to meet vibration criteria which ensure long term integrity of the different components including shafts, bearings, pedestals and foundation. As mentioned before typical on-site balancing is usually performed by using the Influence Coefficient (IC) method. However, the two main drawbacks of the method are: On the one hand the method is a very time consuming activity and on the other hand the balancing has often to be performed under huge time pressure. If the Influence Coefficients are not already known before the balancing process, usually different test runs (one test run per balancing plane) are needed where the unit needs to be stopped, cooled down and be restarted again. Especially, when balancing planes of the HP turbine or inside the Generator are needed, these activities may range from time consuming to very time consuming. And balancing is performed under time pressure, because it is usually the last step of an outage before the unit goes back to commercial operation. This may lead to compromises, e.g. that not the best suited balancing planes, but the best accessible planes are selected or that not all required test runs are performed. An advantage of the balancing procedure with measured Influence Coefficients is of course, that the real dynamic behavior of the unit, including the shafts, the bearings, the pedestals and the foundation with the whole environment is included in the measured Influence Coefficients.

An alternative way to determine the required Influence Coefficients for the balancing process is to build up a Rotor Dynamic model, including all important dynamic effects, and to calculate the Influence Coefficients by Numerical Analysis. This can be achieved by applying successively unbalance forces in the balancing planes, corresponding to the test weight settings with respect to amplitude and phase, and to calculate the system responses, e.g. shaft relative displacements in µm and/or absolute velocities in mm/sec at the bearing locations. The calculations can be performed based on output/input relations in the time or frequency domain, where Frequency Response Functions (FRF) can be used for example. The numerical determination of the Influence Coefficients with a Rotor Dynamic model has the clear advantage of time and cost reduction. On the one side the calculations can be performed cheaper than measurements, but more important is the fact, that a unit cannot be operated with power when Influence Coefficients are determined by measurements. A Numerical Analysis with a model also allows much more freedom to investigate different influences on the dynamic behavior. Influence Coefficients can be determined for arbitrary balancing planes and also for arbitrary measurement degrees of freedom. Furthermore it is possible to optimize the balancing by selecting the best suited combination of input and output quantities. In the following both procedures of balancing with measured and calculated Influence Coefficients are briefly described.

3.2 BALANCING WITH MEASURED INFLUENCE COEFFICIENTS

At first we consider the classical way of balancing with measured Influence Coefficients (Figure 10). If balancing is needed because of excessive vibrations, compensation weights for the balancing planes have to be calculated based on the measured actual high vibrations at defined measurement locations and on the corresponding Influence Coefficients. If last ones are not available from previous investigations, they have to be determined in test runs by applying test weights in the balancing planes and measuring the resulting vibration responses (amplitudes and phase angles) at defined measurement locations for the rotational speed of interest. Information from unit run ups and coast downs can help to determine the required measured Influence Coefficients. After calculation of the compensation weights bolts can be selected with respect to weight and position for the final balancing. After completion of the balancing process, it has to be checked whether the vibrations are acceptable or not.

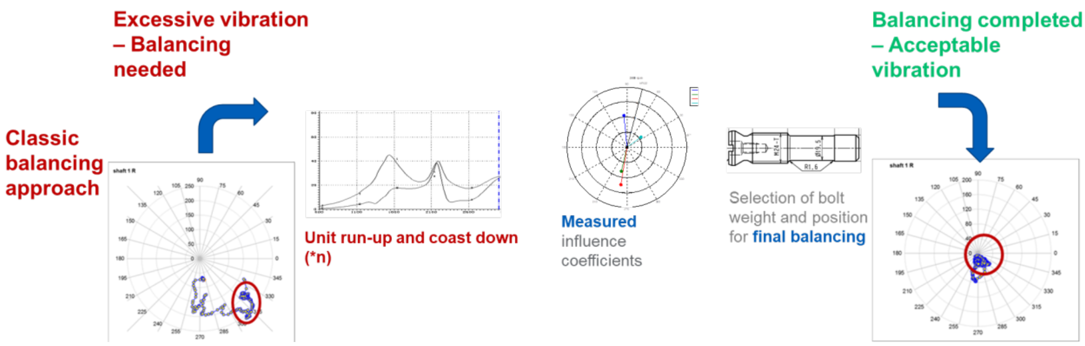


Figure 10 Balancing with measured Influence Coefficients

3.3 BALANCING WITH CALCULATED INFLUENCE COEFFICIENTS

The alternative way to the classical balancing approach with measured Influence Coefficients is balancing by means of calculated Influence Coefficients, which can also be defined as Model Based Balancing (MBB, see lower path in Figure 11). In this case physical runs are no longer necessary and they are replaced by numerical calculations with a digital twin, that means with a good Rotor Dynamic model, describing all important dynamic effects of the rotor train, that means the dynamics of all components: shafts, oil film bearings, bearing pedestals and the foundation. In other words the digital twin should be able to describe the real dynamic behavior as good as possible.

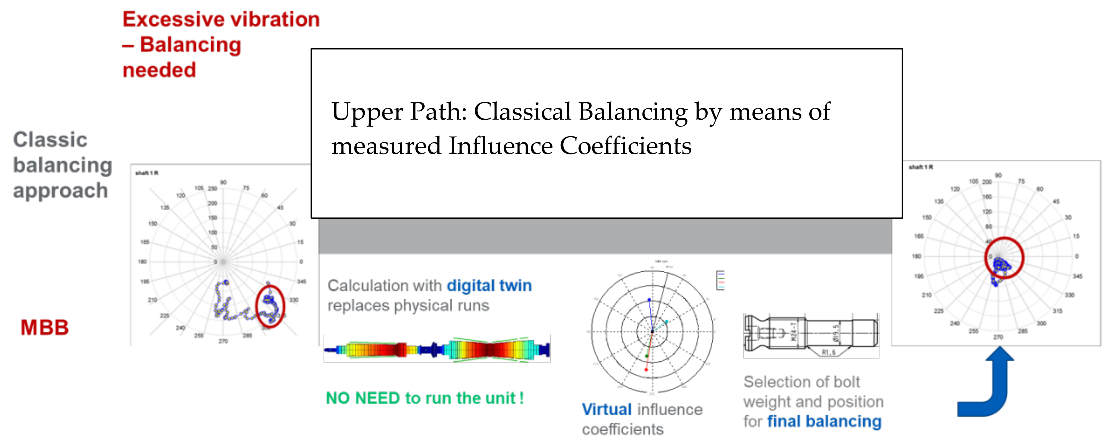


Figure 11 Balancing with calculated Influence Coefficients (Lower Path: MBB)

If this is possible, the (virtual) Influence Coefficients can be calculated by means of the Rotor Dynamic model, which is usually a Finite Element (FE) model. The remaining parts of the balancing procedure are the same as in the classical approach with measured Influence Coefficients (Figure 11).

Figure 12 summarizes in 3 simplified steps, how the Influence Coefficients have to be determined on base of a Rotor dynamic model. When the model has been created in all details and with all components, unbalance vibrations can be calculated due to an excitation of a test weight in one of the balancing planes (see blue unbalance in LP1, left side). The calculated unbalance responses: amplitude and phase angles versus rotational speed are shown in the diagrams below the rotor train model for the different response locations. From these diagrams the Influence Coefficients can be extracted at relevant rotational speeds, e.g. at the operational speed Ω.

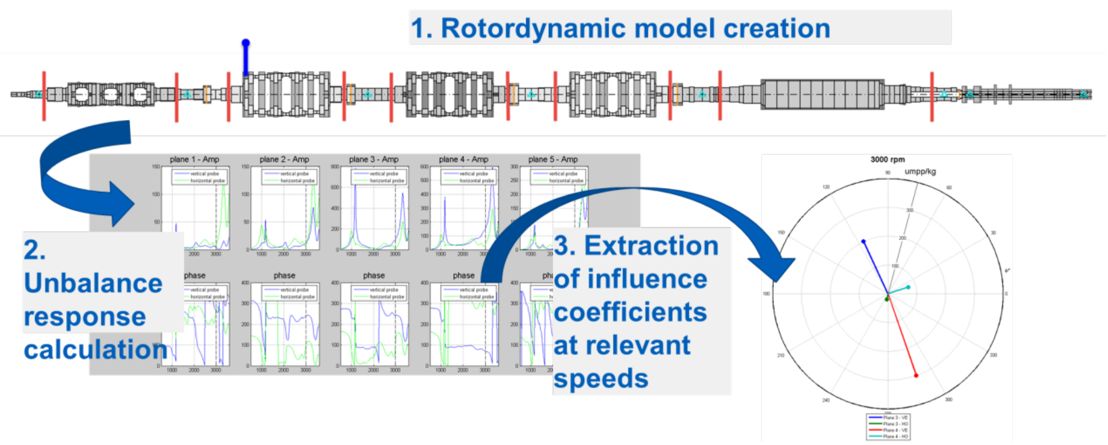


Figure 12 Extraction of Influence Coefficients by means of numerical calculation

4 Selection of a NPP Turbine Train Test Case

4.1 INTRODUCTION

After the decision of the Steering group to evaluate the balancing approach with calculated Influence Coefficients (Model Based Balancing) with a test case, a turbine train unit had to be selected from one of the Nuclear Power Plants in Sweden or Finland. To support the selection of such a test case, it was decided to start with a collection of information for the different turbine train units in the NPP's of Oskarshamn, Forsmark, Ringhals and Olkiluoto. Based on this information the best suited test case for the new balancing procedure should finally be chosen. An information Matrix was created, in order to get general information of the units (power, operating speed, manufacturer, unit operating since…..), design details (number of part rotors, number and type of bearings, bearing loads and geometry, pedestals and foundation), information about the dynamic behavior of the turbine trains (critical speeds, natural frequencies and damping, foundation dynamics), information about previous balancing trials for the unit (balancing planes, measurement locations, measured signals, balancing speeds) and finally information about available Influence Coefficients for the unit.

4.2 INFORMATION MATRIX FOR THE DIFFERENT PLANTS AND UNITS

The following four pages 1 to 4 describe in more detail the required information, which should have been given for the turbine train units of the different Plants. This Information Matrix was sent out to the Nuclear Power Plants in Sweden and Finland in order to select suited steam turbine units for the project. The Information Matrix has been filled out and delivered for special units by the three Nuclear Power Plants in Sweden: **Ringhals (Unit TG 32), Forsmark (Unit TG 3) and Oskarshamn (Unit 3).**

The following criteria have been defined for the selection of one of the units:

1. The rotor train with all turbines and the generator should come from one manufacturer only. In this case all data for the model of the shaft train are available at the manufacturers design office and can directly be used.
2. Furthermore it would be very helpful, when measured Influence Coefficients for the units are available from previous balancing processes. This values can be used, to benchmark the calculated Influence Coefficients.

Turbines and Generator from Unit 3 in **Oskarshamn** are all delivered from one manufacturer (GE/Alstom) and would fulfil the first criteria, but no measured Influence Coefficients are available. Therefore this case has been placed back. Turbines and Generator from unit TG 3 in **Forsmark** are from different manufacturers and do not fulfil the 1st criteria. Only unit TG 32 in **Ringhals** fulfils both criteria and has therefore been selected for the first test case in the project (see chapter 4.3).

Page 1

Information Matrix for Influence Coefficients
Nuclear Power Plant :
Unit number :
Power MW :
Operating Speed RPM :
Unit in operation from xx.xx.xxxx until xx.xx.xxxx

Number of Part Rotors : HP: IP: LP: GEN: EXC:

Manufacturer of Rotor : A= ALSTOM, S=SIEMENS

Who is the Manufacturer of Rotor?:

HP-Rotor	A or S
IP-Rotor	A or S
LP-Rotor	A or S
GEN-Rotor	A or S
EXC-Rotor	A or S

Number of Oil Film Bearings :

Type of Bearing : C=Circular,P=Pocket,L=Lemon,TP= Tilt Pad

If helpful, please add a drawing to show the Bearing Locations

Which Type of Bearings used for ?

HP-Rotor
IP-Rotor
LP-Rotor
GEN-Rotor
EXC-Rotor

Page 2

Information Matrix for Influence Coefficients
Nuclear Power Plant :
Unit number :
Power MW :
Operating Speed RPM :

The following **information is available** and can be delivered:

Static Bearing Loads due to the Shaft Weight:	Yes	No
Static Shaft Locations (Equilibrium) in the Bearings:	Yes	No
Bearing Geometry- Diameter, Width, Clearance:	Yes	No
Bearing Oil Film Temperatures and Viscosities:	Yes	No

Dynamic Behavior of the Shaft Train:

Critical Speeds in rpm and Mode Shapes :	Yes	No
Amplitudes and Phase of Run Up/Down curves :	Yes	No
Amplitudes and Phase at Operating Speed :	Yes	No

Dynamic Behavior of the Foundation:

Frequency Response Curves at all Bearing Locations:	Yes	No
or **Natural Frequencies, Damping and Mode shapes**	Yes	No

Page 3

Information Matrix for Influence Coefficients
Nuclear Power Plant :
Unit number :
Power MW :
Operating Speed RPM :

Balancing Planes designated for the **Balancing Process?**

HP-Rotor	Free Shaft Side		LP-Side
IP-Rotor	HP Side		LP-Side
LP-Rotor	HP-Side	Mid of LP-Shaft	GEN-Side
Gen -Rotor	LP-Side	Mid of GEN Shaft	EXC-Side
Gen-Rotor	Coupling		

Which **Vibration-Measurements** can be taken for **Balancing**?

Relative **Shaft Vibrations** at all **Bearing** Locations (Hor./Vert. or 45°)?

Relative **Shaft Vibrations** at Shaft Ends (Horizontal./Vertical or 45°)?

Absolute **Bearing Vibrations** at Bearing Locations (Hor./Vert. or 45°)?

Which **Balancing Speeds** have been selected for the **Balancing process**?

Balancing Speeds in the neighborhood of the **Critical Speeds** in the Operating Range.

Page 4

Information Matrix for Influence Coefficients
Nuclear Power Plant :
Unit number :
Power MW :
Operating Speed RPM :

Available Influence Coefficients for the Unit of this Power Plant

Are any **Influence Coefficients available** for this Unit? Yes No

Have these **Influence Coefficients** been **measured** ? Yes No

If Yes, when were the **Measurements** taken? Date?

Has the **configuration of the Unit** been changed since then? Yes No

If Yes, **what has been changed?** Describe

Which **Balancing planes** have been selected?
- Please describe according to Page 3!

Which **Vibration Measurements** have been taken?
- Please describe according to Page 3!

Which **Balancing speeds** have been selected?
- Please describe according to Page 3!

4.3 TEST CASE: TURBINE TRAIN UNIT TG 32 OF NPP RINGHALS

The following Information Matrix with pages 1 to 4 has been delivered for Unit **TG32** of the Nuclear Power Plant **Ringhals.** This information together with the design data from the manufacturer GE/Alstom was used to build up the Rotor Dynamic model for this unit, in order to calculate the Influence Coefficients for the balancing process (Comments in red are from Ringhals, comments in blue are from GE/Alstom).

Information Matrix for Influence Coefficients (Page 1)

Nuclear Power Plant : *Ringhals*

Unit number : *TG32*

Power MW : *574 MW*

Operating Speed RPM : *3000 rpm*

Unit in operation from: *1981, LP-retrofit in 07.2006, HP-retrofit in 06.2012, Generator retrofit in 2007.*

Number of Part Rotors : HP: 1 IP: - LP: 3 GEN: 1 EXC: 1

Manufacturer of Rotor : A A= ALSTOM, S=SIEMENS

Who is the Manufacturer of Rotor?: HP-Rotor A

IP-Rotor -

LP-Rotor A

GEN-Rotor A

EXC-Rotor A

Number of Oil Film Bearings : 8

Type of Bearing : C=Circular,P=Pocket,L=Lemon,TP= Tilt Pad

If helpful, please add a drawing to show the Bearing Locations

We need to check the bearing types, e.g. whether these are R1/2T or lemon/elliptical type

Which Type of Bearings used for?

HP-Rotor: Bearing 1, (believe it is called) Circular or Plain, consists of 3 wedges → *R3T?*

LP-Rotor 1: Bearing 2, Combined journal and thrust bearing. The Journal bearing is of Circular or Plain design and consists of 2 wedges

LP-Rotor 2: Bearing 3, Plain bearing (Circular design) consisting of 2 wedges

LP-Rotor 3: Bearing 4, TP, with 3 pads

GEN-Rotor: Bearing 5 and 6, TP, with 3 pads

EXC-Rotor : Bearing 7 and 8, Circular or Plain design

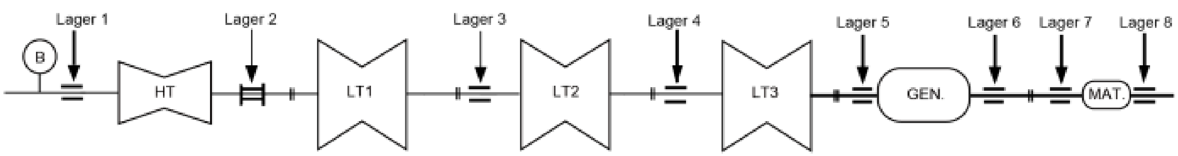


Figure 13 Rotor train of NPP Ringhals unit TG 32

Information Matrix for Influence Coefficients (Page 2)

Nuclear Power Plant : *Ringhals*

Unit number : *TG32*

Power MW : *574 MW*

Operating Speed RPM : *3000 rpm*

The following **information is available** and can be delivered:

Static Bearing Loads due to the Shaft Weight: Yes

Static Shaft Locations (Equilibrium) in the Bearings: No

Bearing Geometries- Diameter, Width, Clearance: Yes

Bearing Oil Film Temperatures and Viscosities: No (Bearing metal temperature is available)

- ➔ *Metal temperature is fine.*
  Oil inlet temperature would be helpful. In addition we need to know the oil grade (most likely ISO VG46)

Dynamic Behavior of the Shaft Train

Critical Speeds in rpm and Mode Shapes : Yes

Amplitudes and Phase of Run Up/Down curves : Yes

Amplitudes and Phase at Operating Speed : Yes

Dynamic Behavior of the Foundation

Frequency Response Curves at all Bearing Locations: No

or **Natural Frequencies, Damping and Mode shapes** No

Information Matrix for Influence Coefficients (Page 3)

Nuclear Power Plant **:** *Ringhals*

Unit number : *TG32*

Power MW **:** *574MW*

Operating Speed RPM **:** *3000 rpm*

Balancing Planes designated for the **Balancing Process?**

HP-Rotor Free Shaft Side Yes LP-Side Yes

LP-Rotor 1-3 HP-Side, Yes Mid of LP-Shaft, No GEN-Side Yes

Gen -Rotor LP-Side, Yes Mid of GEN Shaft, No EXC-Side Yes

Which **Vibration-Measurements** can be taken for the **Balancing**?

Relative **Shaft Vibrations** at all **Bearing** Locations (Horiz./Vert. or 45°)?
Vert and horiz. next to the bearings HP, vert and horiz. at LP seals, vert and horiz. next to the bearings GEN.

Relative **Shaft Vibrations** at Shaft Ends (Horiz./Vert. or 45°)?

Absolute **Bearing Vibrations** at Bearing Locations (Horiz./Vert. or 45°)?
Yes, both vertical and horizontal at all bearings.

Which **Balancing Speeds** will be selected for the **Balancing Process**?
3000 rpm

Balancing Speeds in the neighborhood of the **Critical Speeds** in the Operating Range. *No*

Information Matrix for Influence Coefficients (Page 4)

Nuclear Power Plant **:** *Ringhals*

Unit number : *TG32*

Power MW **:** *574MW*

Operating Speed RPM **:** *3000 rpm*

Available Influence Coefficients for the Unit of this Power Plant

Are any **Influence Coefficients available** for this Unit? Yes No

Have these **Influence Coefficients** been **measured** ? Yes No

If Yes, when were the **Measurements** taken? Date *06.2006*

Has the **configuration of the Unit** been changed since then? Yes No

If Yes, **what has been changed?** *The HP-rotor was changed in 2012, and the generator rotor in 2007, both are very similar in design.*

Which **Balancing planes** have been selected?

LP1 – HP-side and GEN-side

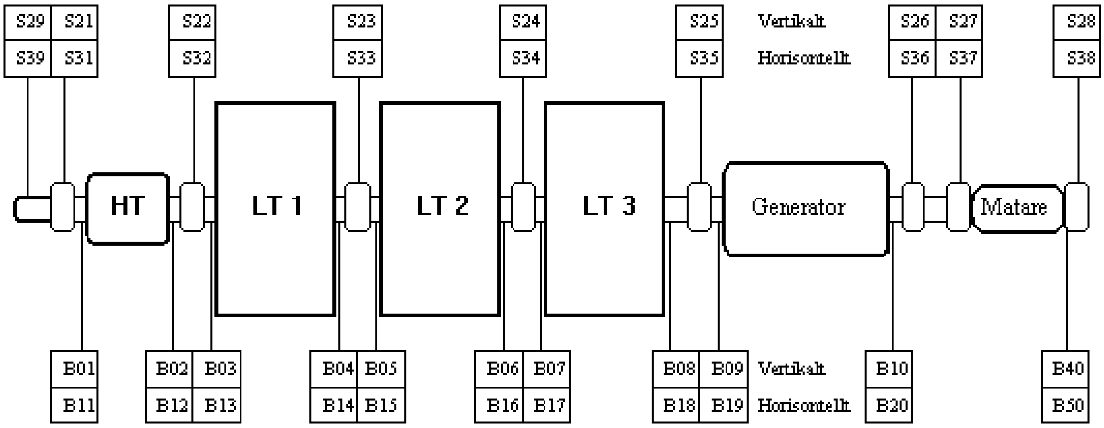
LP2 – HP-side and GEN-side

LP3 – HP-side and GEN-side

Which **Vibration Measurements** have been taken?

All relative shaft vibration and absolute bearing vibration sensors have been measured.

Which **Balancing speeds** have been selected? *3000 rpm*



B-sensor (proximity probes, relative shaft vibration)

B01	HT	TS Vertikalt	B11	HT	TS Horisontellt
B02	HT	GS Vertikalt	B12	HT	GS Horisontellt
B03	LT1	TS Vertikalt	B13	LT1	TS Horisontellt
B04	LT1	GS Vertikalt	B14	LT1	GS Horisontellt
B05	LT2	TS Vertikalt	B15	LT2	TS Horisontellt
B06	LT2	GS Vertikalt	B16	LT2	GS Horisontellt
B07	LT3	TS Vertikalt	B17	LT3	TS Horisontellt
B08	LT3	GS Vertikalt	B18	LT3	GS Horisontellt
B09	GEN	TS Vertikalt	B19	GEN	TS Horisontellt
B10	GEN	GS Vertikalt	B20	GEN	GS Horisontellt
B40	Matare	Vertikalt	B50	Matare	Horisontellt

S-sensor (seismic sensor, absolute bearing vibration)

S21	Lager 1	Vertikalt	S31	Lager 1	Horisontellt
S22	Lager 2	Vertikalt	S32	Lager 2	Horisontellt
S23	Lager 3	Vertikalt	S33	Lager 3	Horisontellt
S24	Lager 4	Vertikalt	S34	Lager 4	Horisontellt
S25	Lager 5	Vertikalt	S35	Lager 5	Horisontellt
S26	Lager 6	Vertikalt	S36	Lager 6	Horisontellt
S27	Lager 7	Vertikalt	S37	Lager 7	Horisontellt
S28	Laget 8	Vertikalt	S38	Lager 8	Horisontellt
S29	Reglerkåpa	Vertikalt	S39	Reglerkåpa	Horisontellt
S46	Lager 6	Axiellt			

Figure 14 Bearing and Vibration Sensor positions of NPP Ringhals unit TG 32

5 Components of the Ringhals Unit TG32 for a Complete Rotor Dynamic Model

5.1 INTRODUCTION

To build up the matrices of the Finite Element model for the complete turbine train system of the Ringhals Unit TG 32, including the different turbine moduls of the HP-, LPT1-, LPT2-, LPT3-turbines, the generator and exciter, the eight oil film bearings, the according bearing pedestals and the foundation, the corresponding mass-, damping- and stiffness parameter for all components are needed and have to be determined. In the following chapters the required input data for the different components and their contribution to the complete Rotor Dynamic model are discussed.

5.2 RINGHALS UNIT TG 32 SHAFT TRAIN: MASS AND STIFFNESS DATA

Figure 15 shows the complete rotor train of the Ringhals unit TG 32 with the different shafts together with the drawings of the four turbines and the generator. The individual shafts are coupled together to the complete shaft train via couplings, which are considered to be rigid.

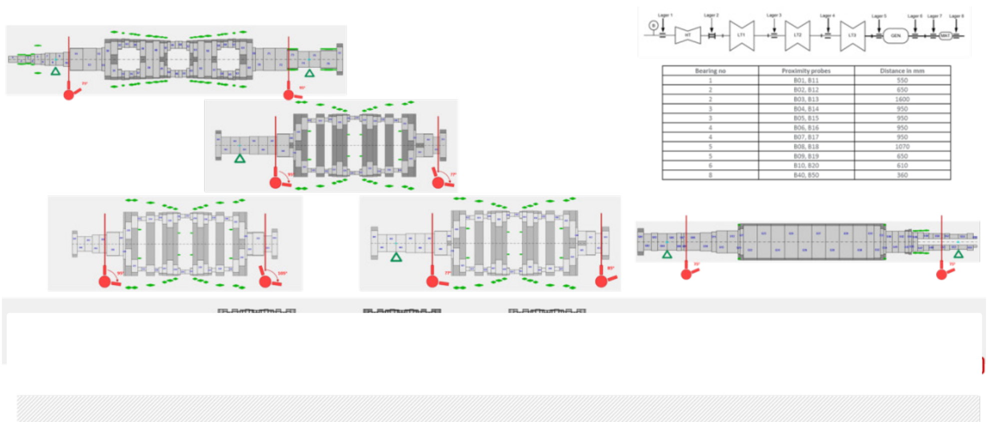


Figure 15 Rotor train of Ringhals unit TG 32 and design drawings of the turbine and generator shafts. The B-sensor locations and their orientations are also shown.

The Rotor Dynamic model follows a standard finite element approach and is built up of beam elements. The geometrical data (length L of each single beam element, inner diameter D_i and outer diameter D_a) and the shaft material data like the temperature dependent Youngs-Modul E, the mass density ρ and the poisons ratio ν are the base for the determination of the stiffness and mass matrices of all different single beam finite elements. By superposition of the matrices of the individual beam elements the mass and stiffness matrices of the part shafts (turbines and generator) and finally the mass matrix $\mathbf{M}_{ST}$ and stiffness matrix $\mathbf{K}_{ST}$ of the complete shaft train (**S**haft **T**rain = **ST**) can be determined. For this purpose all mass diameters and all stiffness diameters are defined by the manufacturer for each single beam element of a part shaft, as shown in Figures 16 and 17. All rotational effects, like rotational inertia and gyroscopic effects are included in the model. The gyroscopic effects are considered in a velocity dependent matrix $\mathbf{G}_{ST}$. Besides the contributions from the beam elements itself the mass properties of the blades are also part of the mass matrix $\mathbf{M}_{ST}$ and of the gyroscopic matrix $\mathbf{G}_{ST}$.

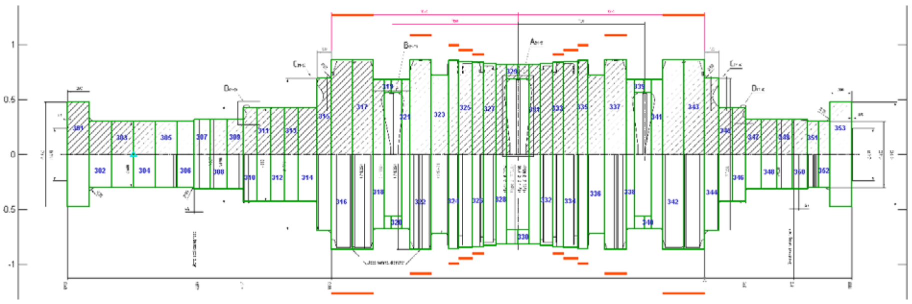


Figure 16 Mass diameter of the single beam elements within the part shaft of the LPT 2-Turbine

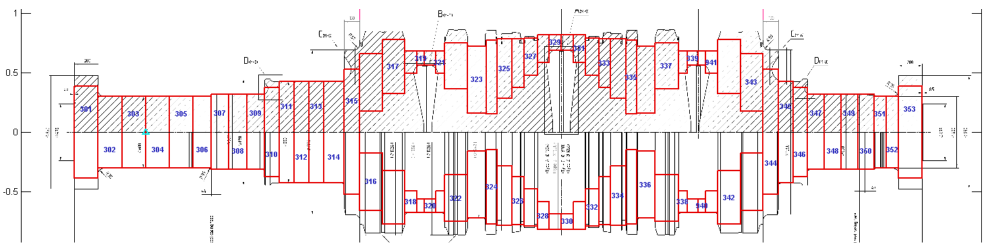


Figure 17 Stiffness Diameter of the single beam elements within the part shaft of the LPT 2-Turbine

Material damping in the beam elements of the shaft train is low and therefore neglected. Damping is mainly generated by oil film effects in the bearings and will be described in the next chapter.

5.3 RINGHALS UNIT TG 32 OIL FILM BEARINGS: ROTOR DYNAMIC STIFFNESS AND DAMPING COEFFICIENTS

The shaft train of the Ringhals unit TG 32 is running in eight oil film bearings of different design (see chapter 4.3, page 1). With respect to the Rotor Dynamic behavior the stiffness- and damping characteristics of each single bearing have to be known. Depending on the bearing type, the bearing geometry (diameter, width and clearance) and the operating parameters (rotational speed, bearing load, fluid viscosity) the static and the dynamic behavior of the oil films are defined (Fig. 18).

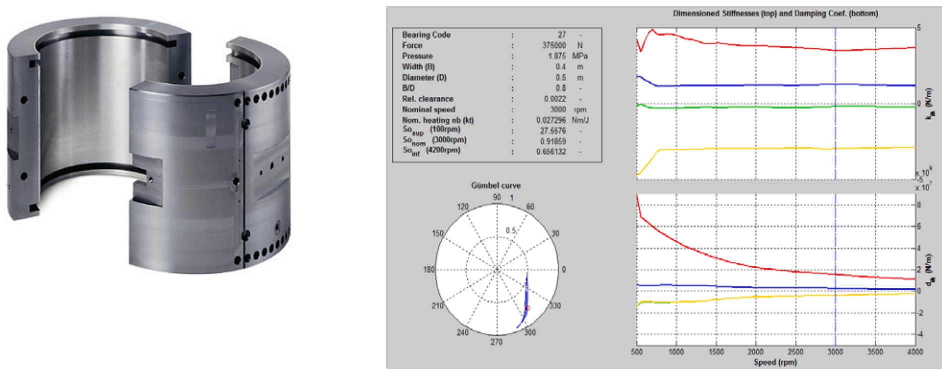
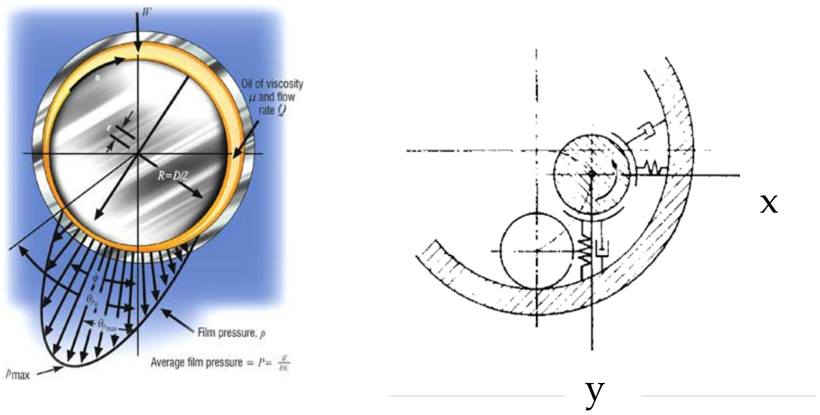


Figure 18 Oil Film Journal Bearing data with static and dynamic characteristics

For each bearing in the shaft train the static shaft position can be determined by means of the so called Gümbel-curve, which is a function of the Sommerfeld number So. The So number is calculated according to the equation

$$So = p_m \; \psi^2 / (\eta \, \Omega) \qquad (3)$$

The So number depends on the static bearing pressure p_m due to the static load (rotor weight), on the relative bearing clearance ψ, on the viscosity of the lubricant η and on the angular velocity Ω of the rotor. For each So-number or the corresponding static shaft position within the bearing clearance the dynamic characteristic of the oil film is well defined. For each journal location or for each So number the dynamic behavior of the oil film can be expressed by four stiffness and four damping coefficients (see Figures 18 and 19). The bearing forces depend on these coefficients, as it is shown in equations (4) and (5) in Figure 19. It is important to note, that the forces depend also on the displacements and velocities between the shaft and the bearing pedestals (see Figure 19).



$$F_x = k_{xx}\,x + k_{xy}\,y + d_{xx}\,\dot{x} + d_{xy}\,\dot{y} \qquad (4)$$

$$F_y = k_{yx}\,x + k_{yy}\,y + d_{yx}\,\dot{x} + d_{yy}\,\dot{y} \qquad (5)$$

Figure 19 Stiffness- and damping coefficients of the oil film bearings

The stiffness- and damping coefficients can be determined by calculations (Reynolds-equations) or by experiments (identification) and the coefficients can be found in tables or diagrams in dependence of the So number or the relative static shaft position (eccentricity). Oil film bearings have anisotropic behavior and the coupling coefficients differ from each other (non-symmetric stiffness and damping matrices). The cross coupled stiffness coefficients of the fluid film are destabilizing parameters, while the main damping coefficients are important for the damping of the vibrations.

As example the stiffness coefficients k_{xx}, k_{xy}, k_{yx} and k_{yy} for the eight bearings of the Ringhals unit TG 32 are presented versus the rotational frequency (Hz) in Figure 20. These coefficients have been determined by the manufacturer GE (Baden). All stiffness and damping coefficients have a strong influence on the dynamic behavior of the complete Rotor train and therefore also on the Influence Coefficients α_{ik}. In general the dynamic coefficients of the bearings are sensitive and sometimes there are uncertainties to get the right values for the Rotor Dynamic model of the complete rotor train.

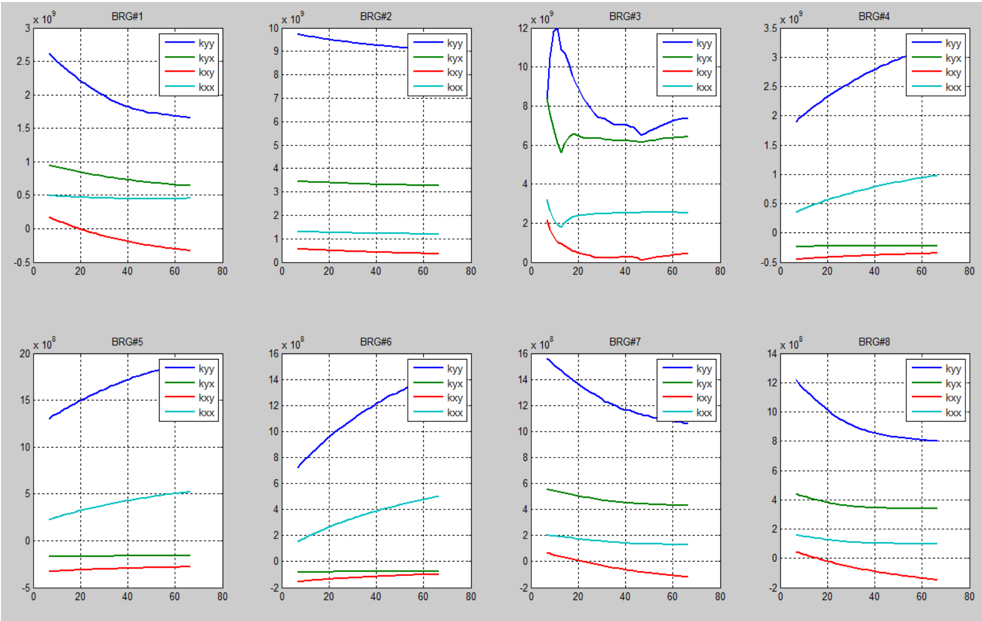


Figure 20 Stiffness coefficients k_{xx}, k_{xy}, k_{yx} and k_{yy} versus rotational frequency in Hz for the eight bearings of the Ringhals unit TG 32

It was agreed with NPP Ringhals, that the Influence Coefficients should be calculated for the operational speed of 3000 rpm. For this case the stiffness- and damping coefficients in the bearings have to be taken for 50 Hz (see Figure 20). The numerical values for this frequency can be found in the table of Figure 21.

	x = horizontal				y = vertical			
Oil film data bearing #	**kxx**	**kxy**	**kyx**	**kyy**	**cxx**	**cxy**	**cyx**	**cyy**
1	4.46E+08	2.52E+08	-6.90E+08	1.73E+08	1.84E+06	-6.20E+05	4.00E+05	4.10E+06
2	1.21E+09	-3.95E+08	-3.30E+09	9.17E+09	2.50E+06	-4.53E+06	1.59E+06	1.12E+07
3	2.56E+09	-1.80E+08	-6.21E+09	6.69E+09	5.64E+06	-8.31E+06	-8.56E+06	3.17E+07
4	8.69E+08	3.67E+08	2.24E+08	2.96E+09	1.03E+06	-1.47E+05	1.47E+05	8.84E+05
5	4.68E+08	2.88E+08	1.63E+08	1.80E+09	4.88E+05	-6.96E+04	6.96E+04	4.18E+05
6	4.35E+08	1.08E+08	7.62E+07	1.31E+09	5.74E+05	-8.20E+04	8.20E+04	4.92E+05
7	1.35E+08	8.66E+07	-4.41E+08	1.12E+09	4.89E+05	-4.68E+05	3.51E+05	2.23E+06
8	1.00E+08	1.14E+08	-3.42E+08	8.26E+08	4.34E+05	-2.84E+05	3.47E+05	1.95E+06

Figure 21 Stiffness in N/m and damping coefficients in Ns/m at rotational frequency 3000 rpm (50 Hz) for the eight bearings of the Ringhals unit TG 32

5.4 RINGHALS UNIT TG 32 BEARING PEDESTAL AND FOUNDATION: MASS, DAMPING AND STIFFNESS DATA, FREQUENCY RESPONSE FUNCTIONS

Up to now we have considered for the Rotor Dynamic model of the Ringhals unit TG 32 the mass-, damping- and stiffness information for the complete shaft with the turbines and the generator and the eight oil film bearings along the shaft train. But it is also very important, to include the dynamics of the bearing pedestals and the foundation in the complete model of the system. In other words the Rotor Structure Interaction in the model is needed to obtain good results.

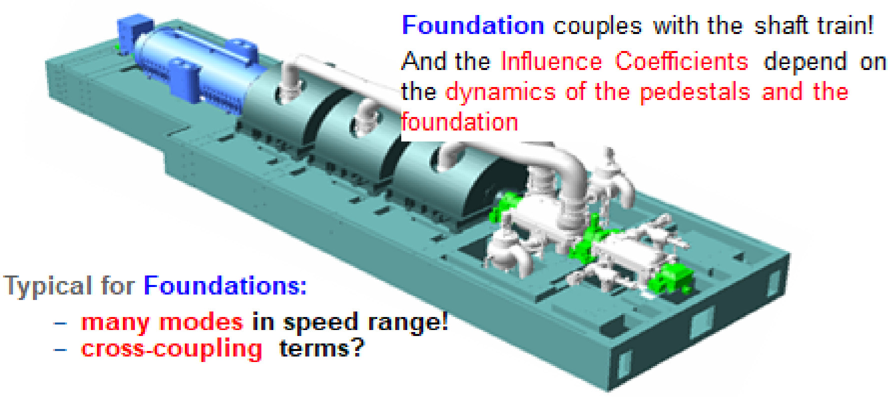


Figure 22 Modelling of the complete turbine train with bearing pedestals and the foundation

The foundation and the bearing pedestals are dynamically coupled with the shaft train. In the model the point of coupling is considered to be the surface of the bearings (Bearing shell). The shaft train and the oil film act on the one side of each connecting point by dynamic forces and the bearing pedestal and the foundation act on the other side. It is typical for this kind of Rotor-Structure-Interaction, that many mode shapes exist in the speed range of the turbine train. Furthermore coupling effects can be observed between the different bearings and between the horizontal and the vertical direction as well.

To introduce the dynamic behavior of the pedestals and the foundation into a complete Rotor Dynamic model a 3D Finite Element model for the two components with mass and stiffness matrices $\mathbf{M}_{PF}$ and $\mathbf{K}_{PF}$ (PF Pedestal & Foundation) can be developed and coupled to the shaft train and the oil film bearing components. For this coupling a GE procedure can be used, that was developed at ALSTOM some years ago. Instead of the matrices $\mathbf{M}_{PF}$ and $\mathbf{K}_{PF}$ the Frequency Response Functions of the Bearing Pedestal & Foundation can also be used to couple the two part systems.

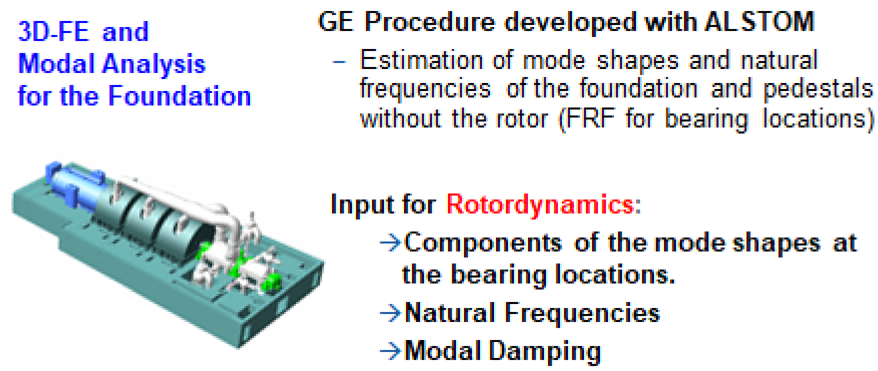


Figure 23 Modelling of the complete turbine train with pedestals and foundation

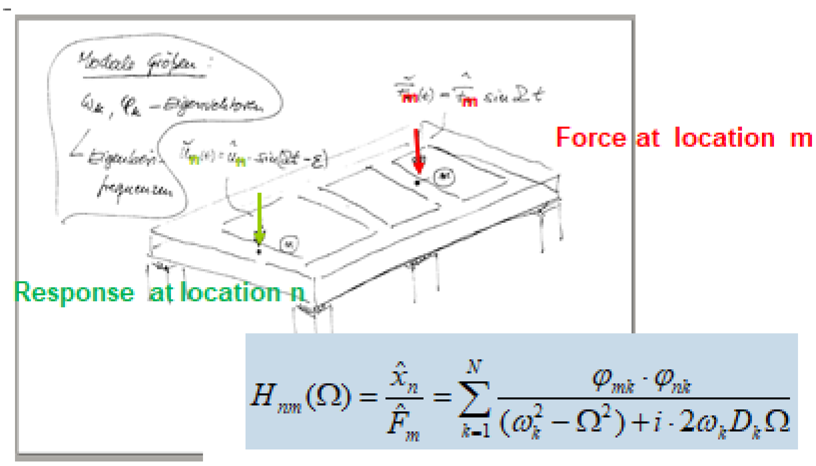


Figure 24 Modelling of the pedestals and the foundation by means of Frequency Response Functions (FRF) at the connecting points

At all connecting points (Bearing shells). The Frequency Response Functions H_{nm} can be built up by means of Modal Analysis using the natural frequencies and the mode shape components at the coupling points of the FE model for the pedestal and the foundation. Some more details are shown in Figures 23 and 24.

Figure 25 shows amplitudes and phase angles of calculated Frequency Response Functions (FRF's) of a Bearing Pedestal & Foundation system at the coupling points (Bearing shells) for the different bearings (7 bearings in the shown example). The FRF's present the driving point functions only, these are the response functions, where the response is taken at the same location at which the system was excited (main diagonal elements of the coupled FRF matrix). FRF's in the upper part are for the vertical direction those in the lower part are for the horizontal direction. In the frequency range of interest (0 to 60 Hz) all functions show a strong dynamic behavior with changing amplitudes and phase angles and several resonances. These results show, that a classical **S**ingle **D**egree **O**f **F**reedom (SDOF) model for the dynamics of the complete Pedestal & Foundation system would deliver a wrong response behavior. The red functions in the diagram with nearly constant amplitudes and phase angles present such a SDOF system with a resonance above 100 Hz. This dynamic behavior can be observed, when pedestals are considered without any foundation influence.

If we consider the complete dynamic behavior of a pedestal & foundation system at the connecting points (Bearing shell) to the shaft train, the matrix of the FRF's is coupled. Such coupling effects can be observed between the different bearings and also between the horizontal and the vertical direction. Figure 26 shows the matrix of the coupled Frequency Response Functions of a Pedestal Foundation system, which is connected with the shaft train at the coupling points. As mentioned before the matrix does not only have main diagonal FRF's, as presented in Figure 25, but there is a more or less completely coupled dynamic behavior.

The best way to consider the complete dynamic behavior of the part system Bearing Pedestal & Foundation in an overall Rotor Dynamic model would therefore be, to connect the full matrix of the FRF's, as shown in Figure 26, to the turbine train & oil film model.

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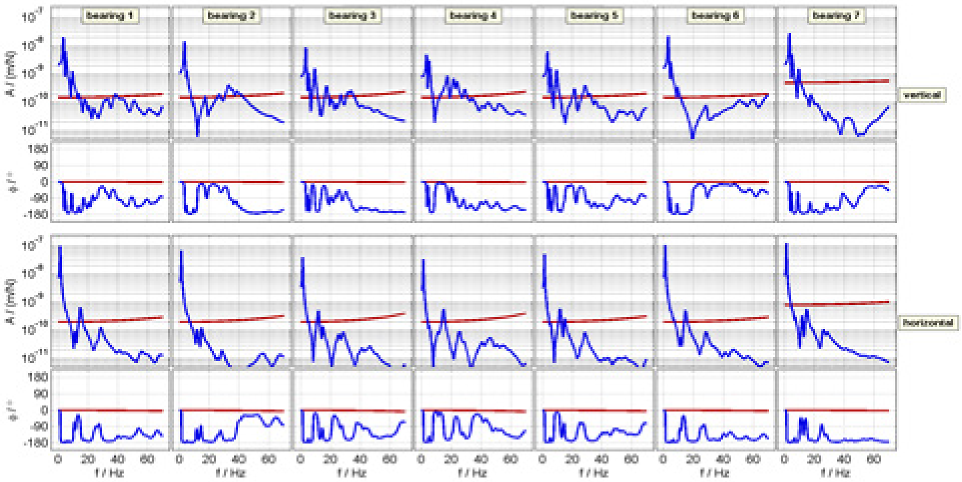


Figure 25 Amplitude and phase angles of the vertical and horizontal Frequency Response Functions of a Bearing Pedestal & Foundation system at the coupling points (Bearing shells). Red curve: SDOF dynamic behavior

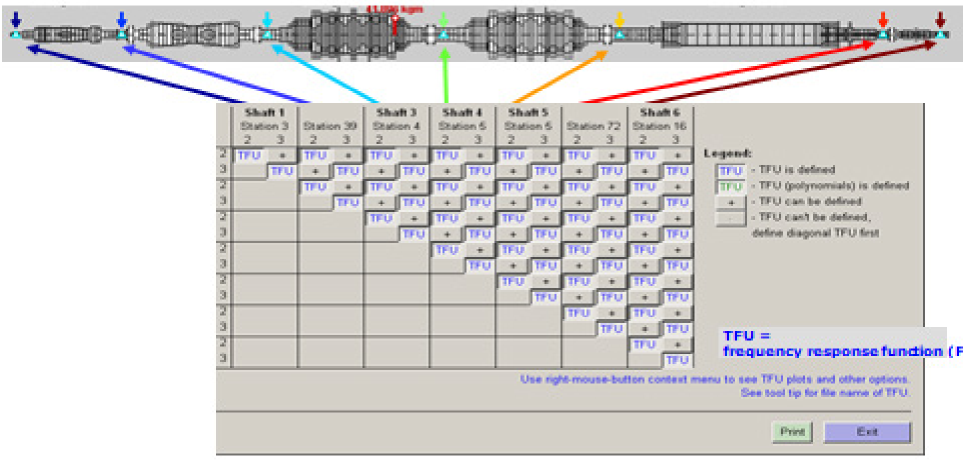


Figure 26 Matrix of coupled Frequency Response Functions of a Pedestal Foundation system, which is connected with the shaft train at coupling points

5.5 RINGHALS UNIT TG 32 BEARING PEDESTAL WITHOUT FOUNDATION: MASS, DAMPING AND STIFFNESS DATA-BASELINE MODEL

No foundation data (Mass, damping and stiffness or Frequency Response Functions) were available for the Ringhals unit TG 32. Therefore the described modelling procedure for the complete Bearing Pedestal plus Foundation system with a matrix of the coupled Frequency Response Functions (see chapter 5.4, Figure 26) was not possible. Instead of this a trial was made to model the dynamic behavior of the supporting system by means of the Bearing Pedestals only without any influence from the foundation. Figure 27 shows in a simplified form this supporting system between the rotating shaft and the foundation, the last one is considered as a rigid boundary (grey). Forces are transferred through the oil film (blue stiffness and damping) to the mass of the pedestal and then via the pedestal stiffness and damping (red mass, stiffness and damping) to the rigidly assumed foundation. Due to the fact, that the foundation data are not included in this simplified Baseline model the supporting system has of course some uncertainties and this may lead to deviations for the calculated Influence Coefficients.

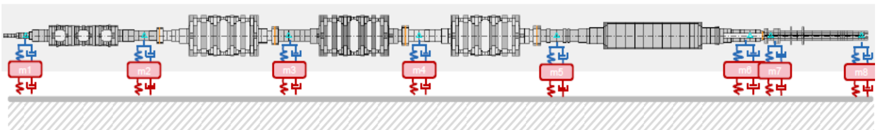


Figure 27 Turbine train with oil film bearings (blue) and bearing pedestals (red)

Figure 28 shows an example of a bearing pedestal in a CAD presentation and a simple corresponding model with the mass of the pedestal m_P and the connection to the rigid foundation with stiffness k_P and damping d_P. In Figure 28 this is only shown for the vertical direction. But the pedestal can also move in horizontal direction, therefore stiffness and damping have also to be defined for the horizontal direction.

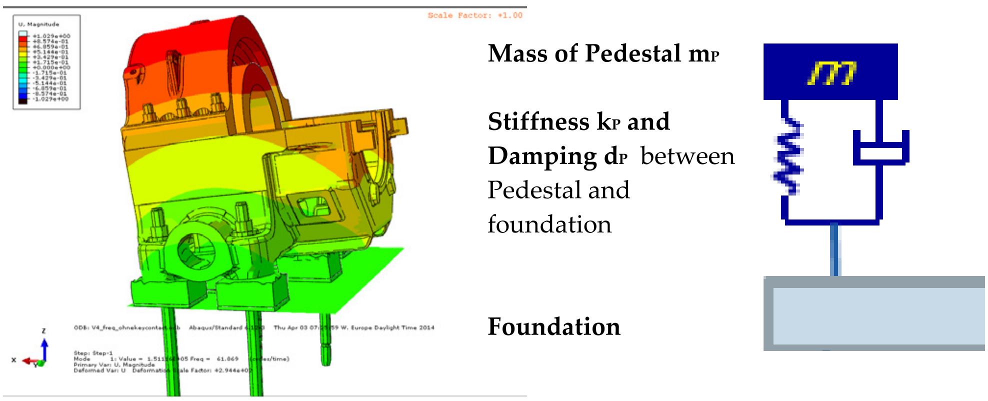


Figure 28 Example of a bearing pedestal and a simplified model for the vertical suspension system of the pedestal

In the real case the movement of the pedestal is a combination of horizontal translation and tilting (two degrees of freedom). This can however be reduced to only one degree of freedom movement in horizontal direction by a reduction method (e.g. Modal or Guyan). The table in Figure 29 presents mass, stiffness and damping values for the eight pedestals of unit TG 32 for the two degrees: vertical and horizontal movement. These data are based on calculations and on experience of GE Power.

Pedestal		#1	#2	#3	#4	#5	#6	#7	#8
Mass (kg)		11000	7000	6500	6500	12500	4500	5500	4500
Stiffness (N/m)	VE	6.00E+09	5.00E+09	5.00E+09	5.00E+09	6.00E+09	6.00E+09	5.00E+09	5.00E+09
	HO	3.33E+09	3.33E+09	3.33E+09	3.33E+09	3.33E+09	3.33E+09	3.33E+09	3.33E+09
Damping (%)	VE	3	3	3	3	3	3	3	3
	HO	5	5	5	5	5	5	5	5

Figure 29 Mass, stiffness and damping values for the eight pedestals of Ringhals unit TG 32 for the horizontal (HO) and the vertical (VE) direction.

Due to the fact, that masses were added to the pedestals 5 to 8 (see Figure 30), correction of the masses were also considered in the model.

Figure 30 Added mass on one of the pedestals of the Ringhals unit TG 32

The Rotor Dynamic Model, presented in figure 27, with the shaft train, the oil film bearings and the bearing pedestals with constant SDOF mass, damping and stiffness values for each connecting point was selected as the Baseline model for the calculation of the Influence Coefficients. Results will be shown in chapter 8.

5.6 RINGHALS UNIT TG 32 WITH PEDESTAL AND FOUNDATION: MASS, DAMPING AND STIFFNESS DATA, IDENTIFIED FROM MEASUREMENTS. UPDATED MODEL

As will be shown later in chapter 8, the results for the Influence Coefficients - calculated with the Baseline Rotor Dynamic model (chapter 5.5) - did not satisfy the expectations. The deviations relative to the measured Influence Coefficients were partly too high. Due to this reason it was decided to find better values for the SDOF mass, damping and stiffness parameters of the supporting structure (bearing pedestal plus foundation) based on measurements at the connecting points. Frequency Response Functions (FRF's) of the supporting structure were therefore measured at these connecting points (horizontal and vertical) at the most important bearing pedestals B2, B3, B4 and B5 (Bearings of the Low Pressure Turbines LPT1, LPT2, LPT3). These measured complex FRF'S (amplitude and phase) describe the frequency dependent dynamic behavior of the complete supporting system, including the bearing pedestals plus the foundation. Figure 31 shows amplitude and phase of measured compliance functions (displacement in mm/Force in N) as a function of the frequency in the range from 0 to 100 Hz.

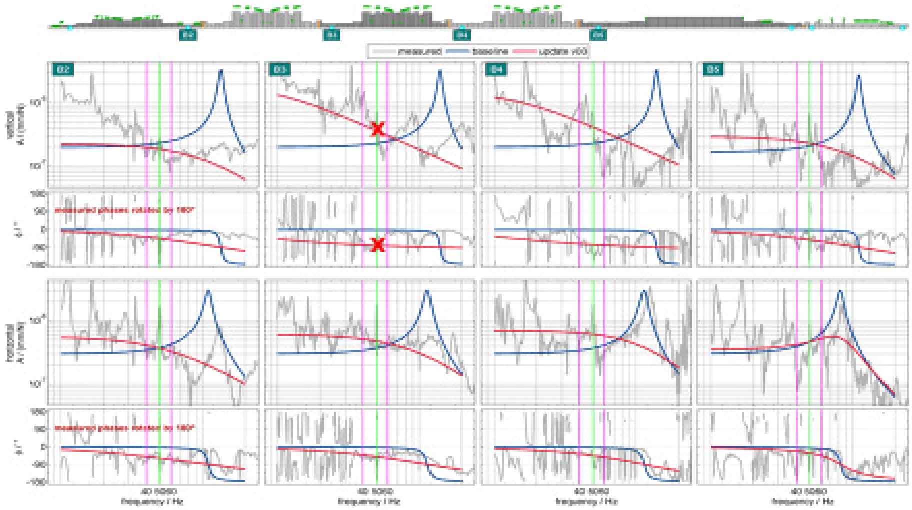


Figure 31 Amplitude and phase of the Compliance Functions in mm/N in dependence of the frequency for the supporting structure (bearing pedestal plus foundation) in vertical and horizontal direction. Bearing pedestals B2, B3, B4, B5

Measured Compliance function (grey curves) in mm/N

Updated Compliance function (curve fit at 50 Hz, x) in mm/N

Baseline compliance function for SDOF system in mm/N

Besides the measured compliances (grey curves) the Baseline compliances (blue curves) of the SDOF bearing pedestal systems with constant mass, damping and stiffness parameters are also presented. The diagrams show very clear differences in the amplitudes and particularly in the phase angles.

It is important to note, that only the main diagonal FRF's were measured. This is a simplification, because all coupling effects between the measurement points have been neglected (see Figure 26 with all couplings). It has also to be mentioned, that the weight of the shaft train is part of the system (jacking oil in the bearings) and may have an influence on the results.

The inverse main diagonal functions are dynamic stiffness functions. They contain the frequency dependent SDOF mass, damping and stiffness values of the supporting structure. By curve fitting (red curve) the parameters can be determined for each frequency. In our case the mass, damping and stiffness values for 50 Hz are of interest in order to determine the Influence Coefficients for the operating speed of 3000 rpm (vertical green line in the diagrams of Figure 31).

The Rotor Dynamic Model (figure 27) with the shaft train, the oil film bearings and the supporting structure can now be used for both the Baseline model as well as for the Updated model. In the Baseline model the SDOF parameters for the bearing pedestals are constant mass, damping and stiffness values for each connecting point. In the Updated model for a selected frequency the SDOF parameters are also constant values. However, they are different compared to the parameters of the Baseline model because they are representing the dynamic behavior of the bearing pedestals plus the foundation.

6 Equations of Motion of the Rotor Dynamic Model for the Ringhals unit TG 32

6.1 INTRODUCTION: EQUATIONS OF MOTION FOR THE DETERMINATION OF THE INFLUENCE COEFFICIENTS

As mentioned before the best suited model to calculate the most realistic Influence Coefficients of the Ringhals unit TG 32 would of course be a model that considers all the components of the system: shaft train, oil film bearings, bearing pedestals and the foundation. Such a model was demonstrated in chapter 5.4, where the two part systems shaft train plus oil film bearings and bearing pedestals plus foundation were coupled at the connecting points. The figures 22 to 26 have shown how this complete Rotor Dynamic Model with all components should be built up.

However, due to fact that foundation data were not available for the Ringhals unit TG 32, an alternative way of modelling had to be selected by introducing SDOF mass, damping and stiffness values for each connecting point between the rotor-bearing shaft line and the supporting structure. This was demonstrated in chapter 5.5. by introduction of a Baseline Model (Figure 27) with constant SDOF parameters. The same model can also be used for the Updated system (chapter 5.6). This means that the model structure itself does not change. However, the parameters change due to the additional dynamic influence of the foundation.

6.2 EQUATIONS OF MOTION WITH MASS-, DAMPING-, GYROSCOPIC - AND STIFFNESS MATRICES FOR THE BASELINE AND UPDATED MODEL

The following equations of motion (6) describe the dynamic behavior of the complete Ringhals turbine TG 32 with the shaft train, the oil film bearings and the supporting structure (Bearing Pedestal with or without foundation influence).

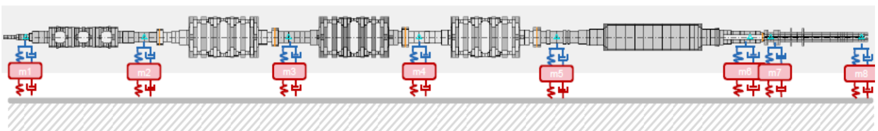


Figure 31 Rotor Dynamic Model for the Ringhals unit TG 32

They represent the equilibrium of inertia forces, damping- and gyroscopic forces, the stiffness forces and the external forces.

$$\mathbf{M}\,\ddot{\mathbf{x}}(t)+(\mathbf{D}(\Omega)+\mathbf{G}(\Omega))\,\dot{\mathbf{x}}(t)+\mathbf{K}(\Omega)\,\mathbf{x}(t)=\mathbf{F}(t) \qquad (6)$$

Inertia forces **Gyroscopic forces** **External forces**

Damping forces **Stiffness forces**

M is the global mass matrix and consists of the translatory and rotatory mass terms of the shaft beam elements, the masses and äquatorial moments of inertia of the turbine blades and the pedestal masses

G is the global gyroscopic matrix and consists of the polar moments of inertia of the shaft beam elements and the turbine blades

D is the global damping matrix and consists mainly of the four damping terms d_{xx}, d_{xy}, d_{yx}, d_{yy} for each of the eight Oil Film Journal Bearings along the shaft train. The matrix contains also terms from the pedestal.

K is the global stiffness matrix and consists of the stiffness terms of the shaft beam elements. The matrix consists also of the four stiffness terms k_{xx}, k_{xy}, k_{yx}, k_{yy} for each of the eight Oil Film Journal Bearings along the shaft train. The matrix contains also stiffness terms from the pedestal.

$\mathbf{x}(t)$ is the global vector of the displacements and rotations of the beam elements along the shaft train and the pedestal displacements.

$\mathbf{F}(t)$ is the global vector of external forces along the shaft train. In this report the forces are particularly due to test weight unbalances.

The global matrices of the equations of motion can be built up based on the data for the components of the complete system: turbine and generator shafts, oil film bearings and bearing pedestals with or without foundation influence, described in chapter 5. It has to be considered, that some of the matrices consist of parameters, which are dependent on the angular velocity Ω of the turbine train. This is for example the case for the stiffness and damping coefficients of the oil film bearings and for the gyroscopic effects in the rotating shafts. All global matrices are assembled automatically by the Finite Element Method (FEM) routine, where the local matrices are inserted on the right place corresponding to the order of the single displacements and rotations within the global displacement vector $\mathbf{x}(t)$. The right order of the single displacements and rotations in the global vector is important for a good structure (e.g. band structured matrices) of the matrices, regarding an economic calculation process. Besides the global displacements $\mathbf{x}(t)$, also the global velocities $\dot{\mathbf{x}}(t)$ and the global accelerations $\ddot{\mathbf{x}}(t)$ are part in the equations of motion.

6.3 SYSTEM INPUT: BALANCING PLANES FOR THE TEST WEIGHTS

As explained earlier Influence Coefficients are defined as output/input relations, where the input is an unbalance due to a test weight, acting in one of the balancing planes and the output can be a relative or absolute displacement or a velocity at one of the response degrees of freedom, usually a relative displacement between the rotor and a stator part or an absolute velocity of the bearing housing. The task in this project is to determine Influence Coefficients by a Numerical Analysis. For this purpose the model should allow to set a test weight at an arbitrary location along the shaft. However, due to practical and design reasons the Balancing planes are selected on both sides of each part shaft (turbines and generator). This is also the case in the Ringhals unit TG 32, which is shown in Figure 32 by the red points.

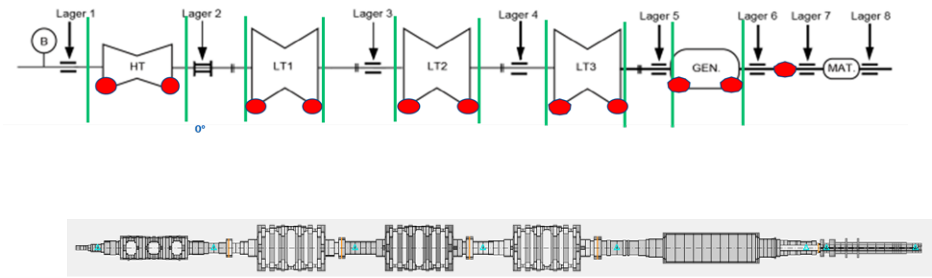


Figure 32 Possible Locations for Balancing Planes (red points = Test Weight setting) and sensor planes (green lines) for relative vibrations of the Ringhals unit TG 32.

If a Balancing plane has been selected a Test Weight can be defined and attached to the rotor. The corresponding unbalance force is inserted in the global external force vector **F**(t) (see chapter 7).

6.4 SYSTEM OUTPUT: MEASUREMENT PLANES AND SENSOR LOCATIONS

With the model for the complete turbine train it is possible to determine any displacement or rotation of the global vector **x**(t) due to an unbalance excitation. However, in the real turbine train we can only measure the system response at locations, where sensors are installed. Due to this fact, the calculated response values will also be determined for these sensor locations.

When calculated and measured relative shaft vibrations are compared, it has to be considered, that the sensors in the real machine are not always oriented in horizontal and vertical direction as it is usually assumed in case of the calculation. Figure 33 shows the sensor orientation in the 10 measurement planes of the Ringhals unit TG 32. When the calculations have been made for the 10 planes, usually in vertical and horizontal direction, the determined values have to be adjusted with respect to the sensor orientations.

It has also to be considered, that in the Ringhals unit TG 32 the relative shaft vibrations are measured between the seal segments of the turbines and the shaft, as it is shown in Figure 34. This arrangement could not directly be realized in the Rotor Dynamic model, where the relative shaft vibrations are usually considered between the bearing housing and the shaft. However it was observed, that the vibrations of the turbine casing are small and that the beam of the sensor clamping is very stiff. Due to this fact the calculated absolute shaft vibrations at the sensor locations were in a quite good correlation to the corresponding measured relative shaft vibration.

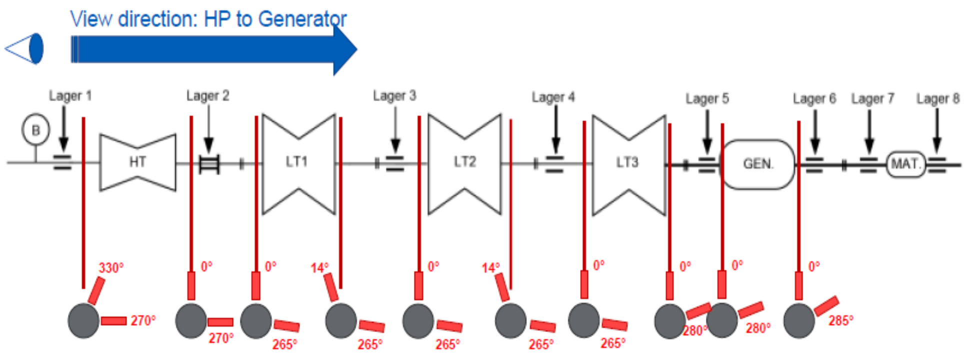


Figure 33 Sensor locations and orientation of the sensors to measure relative shaft vibrations in 10 measurement planes of Ringhals unit TG 32

Figure 34 Measurements of the relative shaft vibrations between the turbine casing and the shaft. The sensor is fixed in a sensor clamping.

Besides the relative shaft vibrations also absolute vibration velocities at the bearing housings are measured at Ringhals unit TG 32. They have also been determined by calculations with the Rotor Dynamic Model.

7 Calculation of System Responses due to a Test Weight in a Balancing Plane

7.1 UNBALANCE FORCES DUE TO A TEST WEIGHT IN A BALANCING PLANE

The procedure to determine Influence Coefficients - by measurements or by calculations – starts with setting of a Test Weight in one of the Balancing (Compensation-) planes. Such a Test Weight in a Balancing plane is characterized by its unbalance $U = M \cdot e$ (M mass of the Test Weight, e eccentricity) and by its phase angle β with respect to a reference line on the rotating shaft. This means, that the reference line serves as a zero phase reference for determining where an unbalance is on a rotor. In Figure 35 the reference line –characterized with a notch – is the horizontal line (x_1-axis) for the nonrotating shaft (left side). The Test Weight with the unbalance $U = M \cdot e$ and the phase angle β are also shown in this figure. When the shaft rotates with angular velocity Ω the following forces F_1 and F_2 act on the shaft, which are caused by the unbalance (see Figure 35 on the right side).

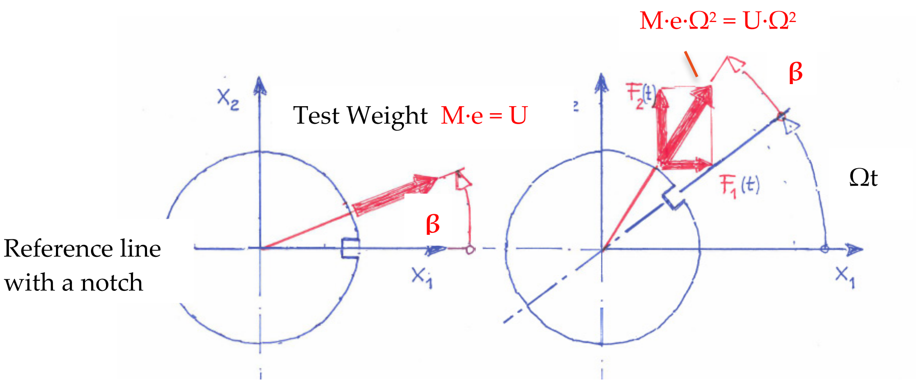


Figure 35 Size and phase angle of a test weight M·e = U in a Balancing plane (left) Unbalance Forces F_1 and F_2 due to rotating shaft with angular velocity Ω (right)

$$F_1(t) = M \cdot e \cdot \Omega^2 \cdot \cos(\Omega t + \beta) = M \cdot e \cdot \Omega^2 \; \mathrm{Im}\,(j \cdot \exp(j \cdot (\Omega t + \beta))) \qquad (7)$$

$$F_2(t) = M \cdot e \cdot \Omega^2 \cdot \sin(\Omega t + \beta) = M \cdot e \cdot \Omega^2 \; \mathrm{Im}\,(1 \cdot \exp(j \cdot (\Omega t + \beta))), \text{ where} \qquad (8)$$

$$\exp(j \cdot (\Omega t + \beta)) = \cos(\Omega t + \beta) + j \sin(\Omega t + \beta) \quad \text{and} \quad M \cdot e = U$$

If we want to determine the system responses of the complete rotor train due to the single Test Weight only we have to solve the equations of motion from chapter 6.2 with a force vector **F**(t), that only includes the two force components F_1 and F_2, as shown in Figure 35.

7.2 EQUATIONS OF MOTION WITH EXCITATION DUE TO A TEST WEIGHT

The equations of motion as derived before are:

$$\mathbf{M}\,\ddot{\mathbf{x}}(t)+(\mathbf{D}(\Omega)+\mathbf{G}(\Omega))\,\dot{\mathbf{x}}(t)+\mathbf{K}(\Omega)\,\mathbf{x}(t)=\mathbf{F}(t) \qquad (6)$$

And the force vector **F** (t) due to a Test Weight with two components in the coordinate directions x_1 and x_2 can be expressed as follows, where the force components have to be inserted at the right position in the vector, depending on the selected balancing plane:

$$\mathbf{F}(t)=\begin{pmatrix}0\\0\\0\\0\\F1\,(t)\\F2\,(t)\\0\\0\\0\end{pmatrix}=M\cdot e\cdot\Omega^{2\cdot}\,\mathrm{Im}\Bigg(\underbrace{\begin{pmatrix}0\\0\\0\\0\\j\exp\,(j\cdot ß)\\1\exp\,(j\cdot ß)\\0\\0\\0\end{pmatrix}}_{\tilde{\mathbf{F}}}\cdot\exp\,(j\cdot\Omega t)\Bigg)=M\cdot e\cdot\Omega^{2\cdot}\,\mathrm{Im}\,(\,\tilde{\mathbf{F}}\exp\,(j\cdot\Omega t)) \qquad (9)$$

We now introduce this force vector in the equations of motion. For the solution a mathematical simplification is used. Instead of solving the equations of motion for the correct real right hand side, as shown above

$$\mathbf{F}(t)=\ M\cdot e\cdot\Omega^{2\cdot}\,\mathrm{Im}\,(\,\tilde{\mathbf{F}}\ \exp\,(j\cdot\Omega t))=\ U\cdot\Omega^{2\cdot}\,\mathrm{Im}\,(\,\tilde{\mathbf{F}}\ \exp\,(j\cdot\Omega t)) \qquad (10)$$

we solve them for a complex force vector, without the term Im ( ).

$$\mathbf{F^*}(t)=\ M\cdot e\cdot\Omega^{2\cdot}\,\tilde{\mathbf{F}}\exp\,(j\cdot\Omega t)=\ U\cdot\Omega^{2\cdot}\,\tilde{\mathbf{F}}\ \exp\,(j\cdot\Omega t) \qquad (11)$$

The equations of motion with the complex force vector can now be represented as

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

$$=\ \mathbf{F^*}(t)=U\cdot\Omega^{2\cdot}\,\tilde{\mathbf{F}}\exp\,(j\cdot\Omega t) \qquad (12)$$

7.3 SOLUTION: EQUATIONS OF MOTION WITH TEST WEIGHT EXCITATION

For the solution of the global response vector $\mathbf{x}(t)$ with displacements and rotations we assume the following form, corresponding to the right hand side of the equations of motion:

$$\mathbf{x}(t) = \tilde{\mathbf{x}} \exp(j \cdot \Omega t) \qquad (13)$$

$$\dot{\mathbf{x}}(t) = j \cdot \Omega \cdot \tilde{\mathbf{x}} \exp(j \cdot \Omega t) \qquad (14)$$

$$\ddot{\mathbf{x}}(t) = -\Omega^2 \cdot \tilde{\mathbf{x}} \exp(j \cdot \Omega t) \qquad (15)$$

This finally leads to the complex algebraic equation system in the frequency domain

$$((\mathbf{K}(\Omega) - \Omega^2 \mathbf{M}) + j \cdot \Omega (\mathbf{D}(\Omega) + \mathbf{G}(\Omega))) \cdot \tilde{\mathbf{x}} = \mathbf{U} \cdot \Omega^2 \cdot \tilde{\mathbf{F}}_i \qquad (16)$$

The solution is the complex global vector $\tilde{\mathbf{x}}$ for all degrees of freedom of the system, consisting of absolute displacements and rotations.

Each component $\tilde{X}_i$ of the complex vector consists of an amplitude $\hat{X}_i$ and a phase angle λ_i, as shown for the i-th degree of freedom, which is for example an absolute displacement of a bearing pedestal or an absolute displacement of the shaft:

$$\tilde{X}_i = \hat{X}_i \exp(j \cdot \lambda_i) = \tilde{X}_{i\,Re} + j \cdot \tilde{X}_{i\,Im} \qquad (17)$$

7.4 DETERMINATION OF THE COMPLEX INFLUENCE COEFFICIENTS

If we assume that the Test Weight was set at the Balancing plane k with an unbalance expressed by

$$(M \cdot e)_k \cdot \exp(j \cdot \beta_k) = U_k \cdot \exp(j \cdot \beta_k) \qquad (18)$$

we can now finally express the complex Influence Coefficient α_{ik}, as it was defined earlier in this report in a more simplified way:

$$\alpha_{ik} = \alpha_{ik\,Re} + j \cdot \alpha_{ik\,Im} = \frac{\hat{x}_i \cdot \exp(j \cdot \lambda_i)}{U_k \cdot \exp(j \cdot \beta_k)} = \frac{\hat{x}_i}{U_k} \exp(j \cdot (\lambda_i - \beta_k)) = \frac{\hat{x}_i}{U_k} \exp(j \cdot \Delta_{ik}) \qquad (19)$$

In the same way all other Influence Coefficients can be determined, especially Influence Coefficients which are defined with relative shaft displacements or with absolute vibration velocities as responses. For a Balancing process the required number of Influence Coefficients can then be collected and assembled in a Influence Coefficient matrix, as was shown in Figure 3 of chapter 2.

8 Calculation of System Responses and Influence Coefficients due to a Test Weight

8.1 PRELIMINARY REMARKS FOR THE CALCULATIONS

The system responses or the Influence Coefficients for the Ringhals Power Plant turbine train TG 32 were calculated for the operational speed of 3000 rpm. The finite element model for the complete turbine train was built up with available design data from GE for the system components and from on-site measurement results (compliance functions), which were determined at the bearing pedestals in horizontal and vertical direction. The structure of the used Rotor Dynamic model was already presented in Figure 31. With this model numerical calculations were performed for two different parameter sets regarding the supporting structure: Bearing pedestals without foundation dynamics (**Baseline**) and bearing pedestals with foundation dynamics from the additional measurements (**Updated**).

Baseline Case: The supporting structure was modelled with constant SDOF parameters mass, damping and stiffness for the bearing pedestal only, as described in chapter 5.5. These parameters are based on calculations and experience from GE. Influences from the foundation could not be considered in this case because no data were available for the foundation.

Updated Case: In this case the supporting structure was also modelled with SDOF parameters mass, damping and stiffness, which were identified from the measured compliance functions for the supporting structure including the dynamics of the bearing pedestal and the foundation. This way has been described in chapter 5.6. The SDOF parameters are now frequency dependent. However, for the assumed constant frequency of 50 Hz they are also constant, but their values differ from the Baseline case. In both cases the components of the shaft train and the oil film bearings were modelled in the same way, as described in chapters 5.2 and 5.3.

Excitation: Six balancing planes were selected at both ends of the three Low Pressure Turbines LT1, LT2 and LT3 (see Figure 32). Corresponding to this planes six load cases (k = 1, 2,. 6) can be defined and the calculations can then be performed for each of these load cases, one after another. As example, for load case "k" a single test weight with an unbalance $(M\cdot e)_k$ and a phase angle $ß_k$ is assumed in the balancing plane k. The resulting unbalance force due to this single test weight is inserted in the force vector **F**(t) of the Rotor Dynamic model, as it was explained in equations (7) to (12) in chapter 7.1 and 7.2.

System Responses: For each of the "k" test weight settings the system responses at defined measurement planes can then be calculated with the complex algebraic equation system (16) in the frequency domain. As an example, the result of a complex absolute displacement response at a measurement degree i will be:

$$\tilde{X}_i = \hat{X}_i \exp(j \cdot \lambda_i) = \tilde{X}_{i\,Re} + j \cdot \tilde{X}_{i\,Im} \qquad (20)$$

as it was already shown before in equation (17).

Calculated Influence Coefficients: With the system response (20) and the assumed test weight in plane "k" (see equation 18) the complex Influence Coefficients can finally be determined in the following way out of equation (21) (same as equation (19) before):

$$\alpha_{ik} = \alpha_{ik\,Re} + j \cdot \alpha_{ik\,Im} = \frac{\hat{x}i}{Uk} \exp\left(j \cdot (\lambda_i - ß_k\right) = \frac{\hat{x}i}{Uk} \exp\left(j \cdot \Delta_{ik}\right) \qquad (21)$$

The calculated results (20) and (21) are especially defined for absolute displacements. In the same way system responses and Influence Coefficients can also be calculated for relative shaft displacements and for absolute bearing vibration velocities.

Comparison with measured Influence Coefficients: In order to evaluate the calculated results, they can be compared with measured results, which were determined in former years at unit TG 32. This comparison can be made either directly for the system responses, corresponding to equation (20) or for the Influence Coefficients corresponding to equation (21). The Influence Coefficients in the following figures 36 to 41 are weighted IC's. They all show the influence of a 1 kg bolt at 0°, multiplied with the radius where the weight is attached at the rotor. The response measurements were taken in horizontal and vertical directions either as relative shaft displacements in μm (0-p) between a shaft location and a corresponding stator part, e.g. the stator seal segments of the turbines (see Figure 34), or as absolute vibration velocities in mm/sec (0-p) at the bearing pedestals B1 to B6 along the shaft train (see figures 32 and 33).

With the developed finite element model all kind of system responses can be determined along the shaft line for one load case. This includes absolute displacements in μm along the shaft line and absolute displacements at the stator parts, e.g. at the bearing pedestals. Based on these results also relative shaft displacements in μm at the bearing locations and absolute vibration velocities in mm/sec at the bearing housings can be derived. Measurement results for comparison with the calculations are of course only available at the existing sensor locations, as described in chapter 6.4. Measured and calculated Influence Coefficients will be considered in the next chapter 8.2.

8.2 COMPARISON OF CALCULATED AND MEASURED INFLUENCE COEFFICIENTS FOR THE RINGHALS UNIT TG 32

In the following figures 36 to 41 calculated and measured Influence Coefficients for the operating speed of 3000 rpm are compared in polar diagrams (amplitudes and phase angles) for the different test weight cases k = 1,2..6, (see chapter 6.3). In each of the figures 36 to 41 the Baseline Influence Coefficients are compared with the Updated Influence Coefficients. To explain the content of the presented results, we consider as example figure 36, which belongs to the load case k =1: Test Weight attached at LP1/ Turbine side. The upper part of figure 36 shows the Influence

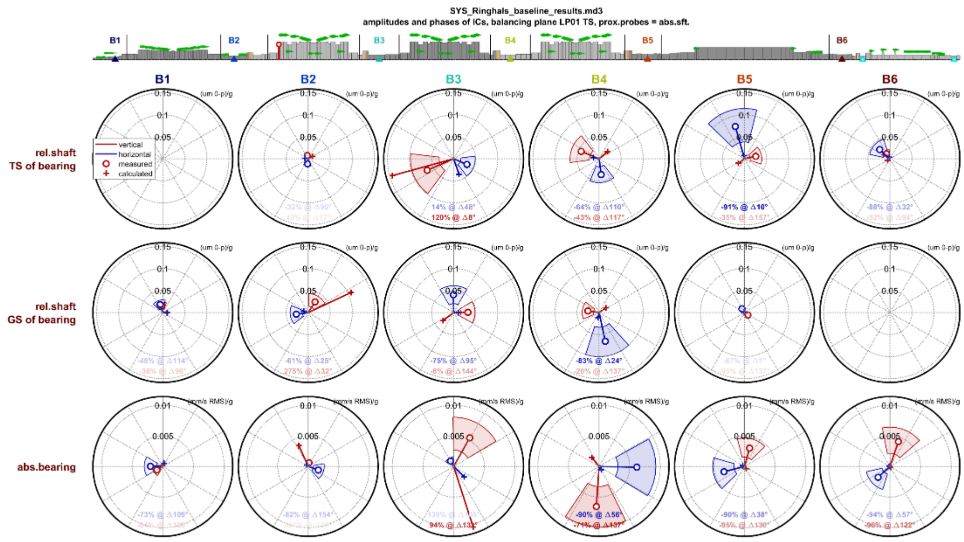
Coefficients for the Baseline model, the lower part for the Updated model. In both cases the shaft train is shown with 6 bearing locations. The red beam marks the location of the test weight, in this case at LP1/Turbine side. The weighted Influence Coefficients are presented in three rows of polar diagrams for different sensor locations. In the first row relative shaft vibrations in µm at the bearing locations B2 to B6 on turbine side (TS of bearing) are shown, in the second row the results are presented in the same way at the bearing locations B1 to B5 on the generator side (GS of bearing). The third row contains the absolute vibration velocities in mm/sec for the bearing housings B1 to B6. In each polar diagram the vectors (amplitudes and phase angels) of the calculated and the measured Influence Coefficients are compared. All red vectors belong to the vertical vibration direction, all blue vectors to the horizontal vibration direction. Crosses (+) stand for calculated results, circles (o) for measured results.

In each polar diagram there are also two segments shown, one for the vertical direction (red) and one for the horizontal direction (blue). These segments represent tolerance areas for the Influence Coefficients.

When the calculated and the measured Influence Coefficients are compared for the different test weight settings and the selected measurement locations, we find out that the Baseline results have partly quite large deviations in amplitudes as well as in phase angles. On the other hand the results show a very clear improvement, when the Influence Coefficients are determined with the Updated model based on the measured compliances. The calculated amplitudes and particular the phase angles of the Updated model fit much better to the measured values when the additional influence from the foundation dynamics is considered. This improvement of the calculated Influence Coefficients with the Updated model is confirmed for all load cases with different test weight settings in figures 36 to 41. With such improved calculated Influence Coefficients it will be possible to perform on-site balancing of a shaft train successfully without additional test runs to determine measured Influence Coefficients.

Although the calculated Influence Coefficients with the Updated model fit much better to the measured values, there are still some deviations existing. In the following chapter 8.3 possible reasons for such deviations are discussed.

BASELINE IC for TEST WEIGHT at LP1/TURBINE SIDE



UPDATE IC for TEST WEIGHT at LP1/TURBINE SIDE

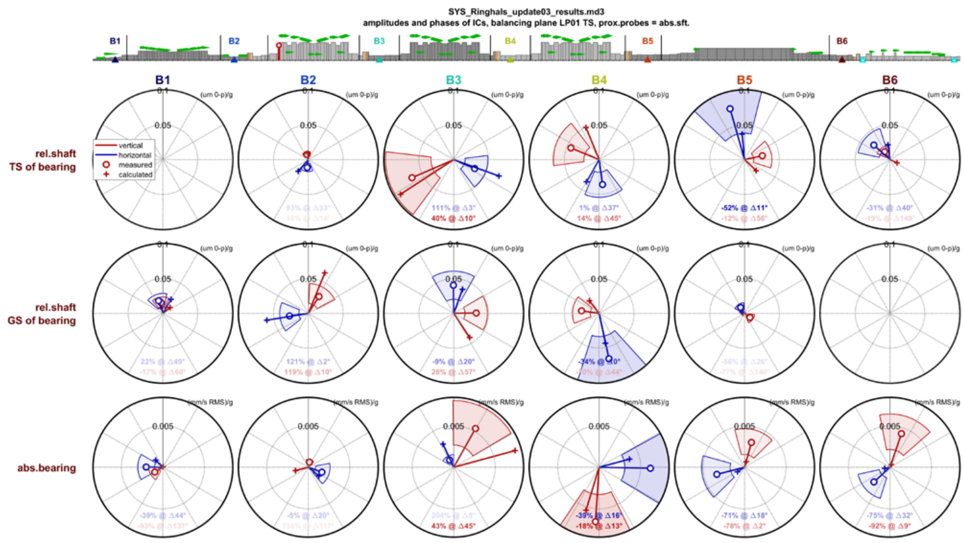
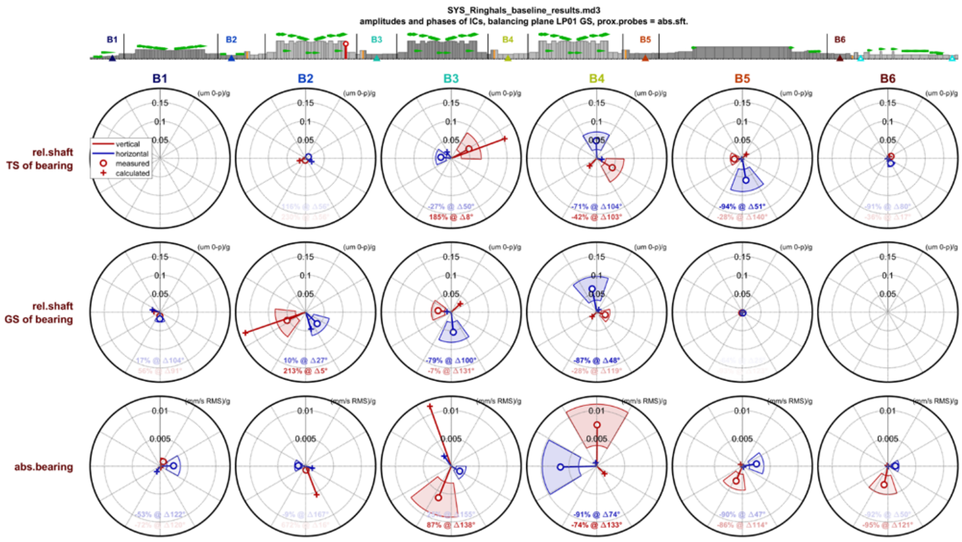


Figure 36 Comparison of calculated and measured Influence Coefficients for the rotational speed of 3000 rpm, Amplitudes and phase angles in Polar Diagrams, Test Weight attached at LP1/TS, Measurement planes on both sides of the Bearings B1 to B6 (rel. vibrations) and on Bearing Pedestals B1 to B6 (abs. vibrations), Rel. shaft vibrations in µm 0-p/g, Abs. Bearing vibrations in mm/sec 0-p/g

BASELINE IC for TEST WEIGHT at LP1/GENERATOR SIDE



UPDATE IC for TEST WEIGHT at LP1/GENERATOR SIDE

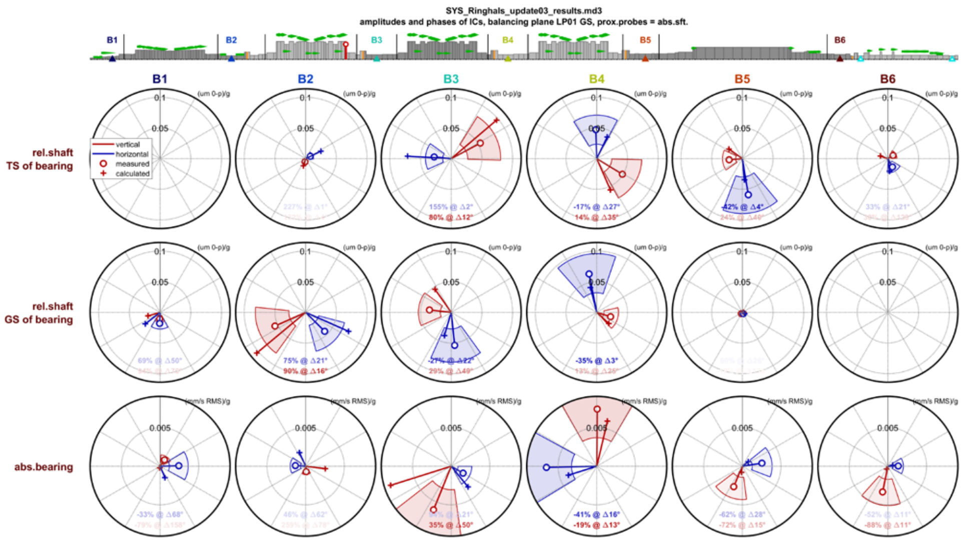
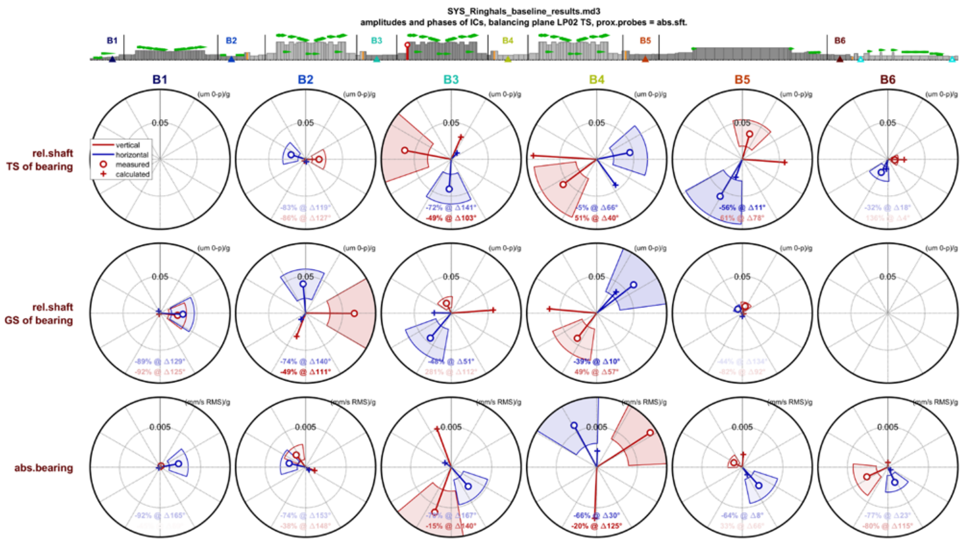


Figure 37 Comparison of calculated and measured Influence Coefficients for the rotational speed of 3000 rpm, Amplitudes and phase angles in Polar Diagrams, Test Weight attached at LP1/GS, Measurement planes on both sides of the Bearings B1 to B6 (rel. vibrations) and on Bearing Pedestals B1 to B6 (abs. vibrations), Rel. shaft vibrations in µm 0-p/g, Abs. Bearing vibrations in mm/sec 0-p/g

BASELINE IC for TEST WEIGHT at LP2/TURBINE SIDE



UPDATE IC for TEST WEIGHT at LP2/TURBINE SIDE

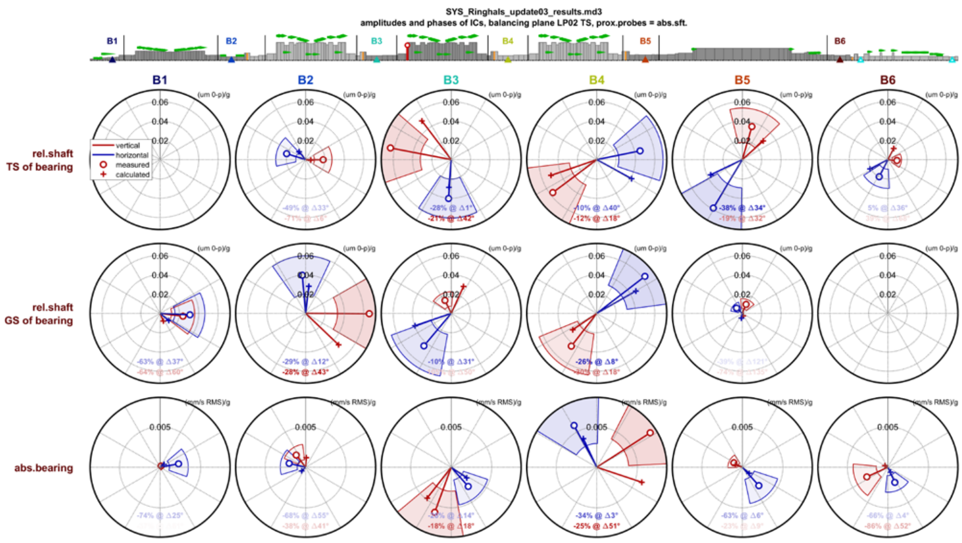
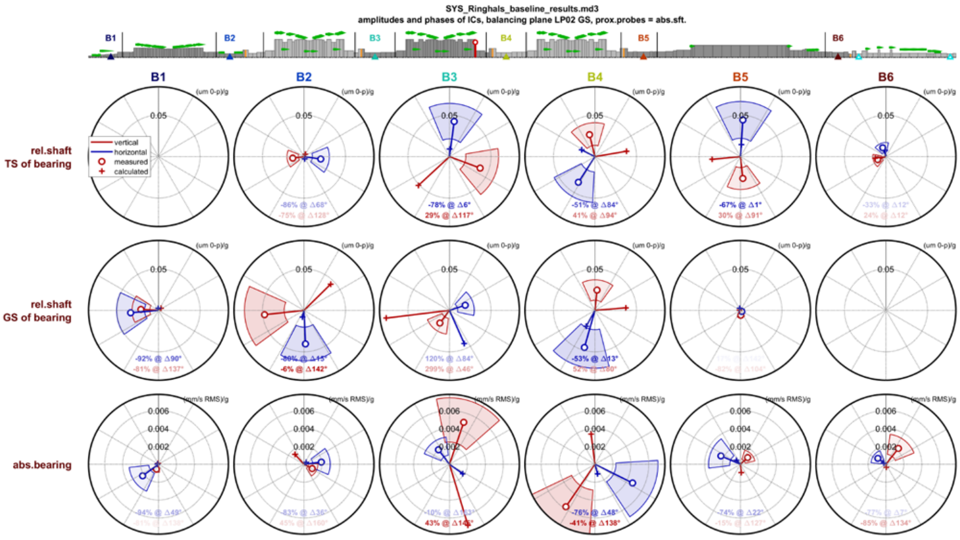


Figure 38 Comparison of calculated and measured Influence Coefficients for the rotational speed of 3000 rpm, Amplitudes and phase angles in Polar Diagrams, Test Weight attached at LP2/TS, Measurement planes on both sides of the Bearings B1 to B6 (rel. vibrations) and on Bearing Pedestals B1 to B6 (abs. vibrations), Rel. shaft vibrations in μm 0-p/g, Abs. Bearing vibrations in mm/sec 0-p/g

BASELINE IC for TEST WEIGHT at LP2/GENERATOR SIDE



UPDATE IC for TEST WEIGHT at LP2/GENERATOR SIDE

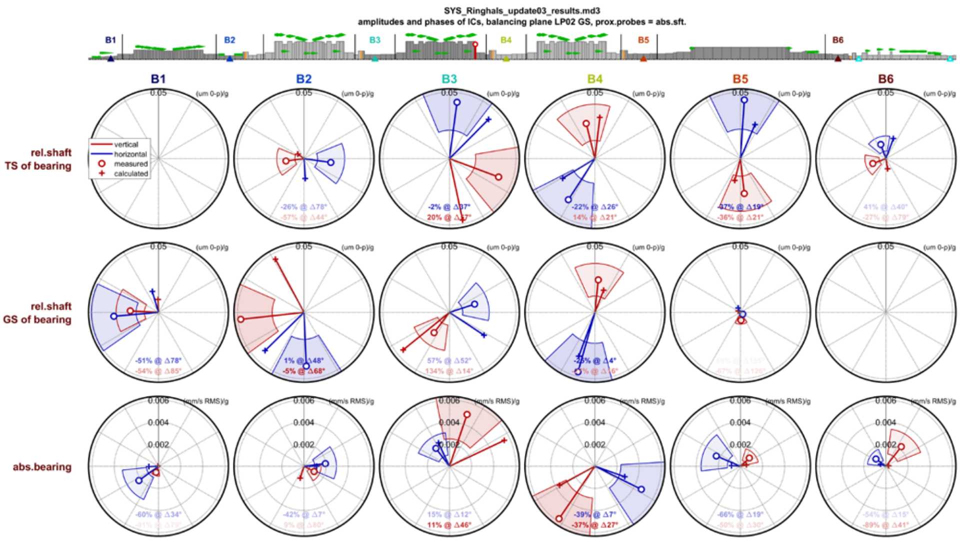
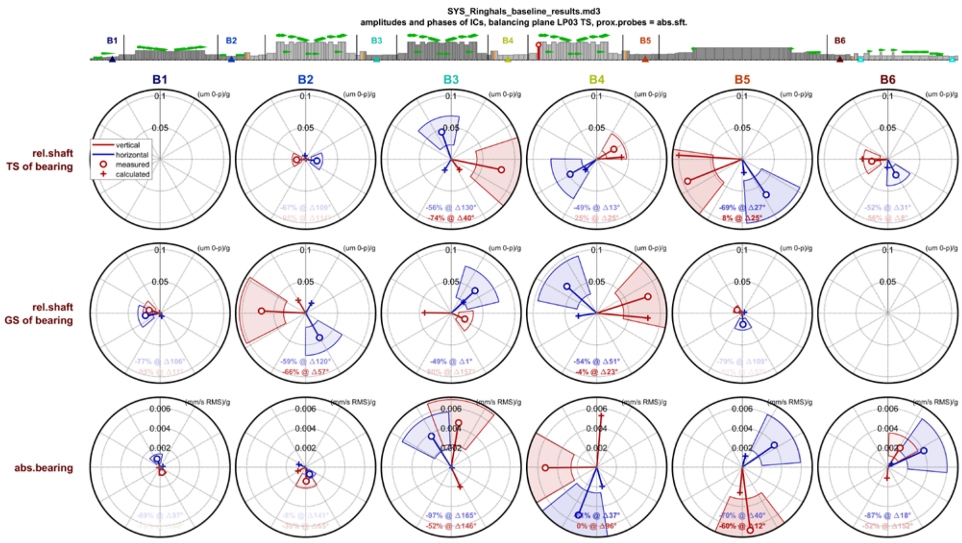


Figure 39 Comparison of calculated and measured Influence Coefficients for the rotational speed of 3000 rpm, Amplitudes and phase angles in Polar Diagrams, Test Weight attached at LP2/GS, Measurement planes on both sides of the Bearings B1 to B6 (rel. vibrations) and on Bearing Pedestals B1 to B6 (abs. vibrations), Rel. shaft vibrations in µm 0-p/g, Abs. Bearing vibrations in mm/sec 0-p/g

BASELINE IC for TEST WEIGHT at LP3/TURBINE SIDE



UPDATE IC for TEST WEIGHT at LP3/TURBINE SIDE

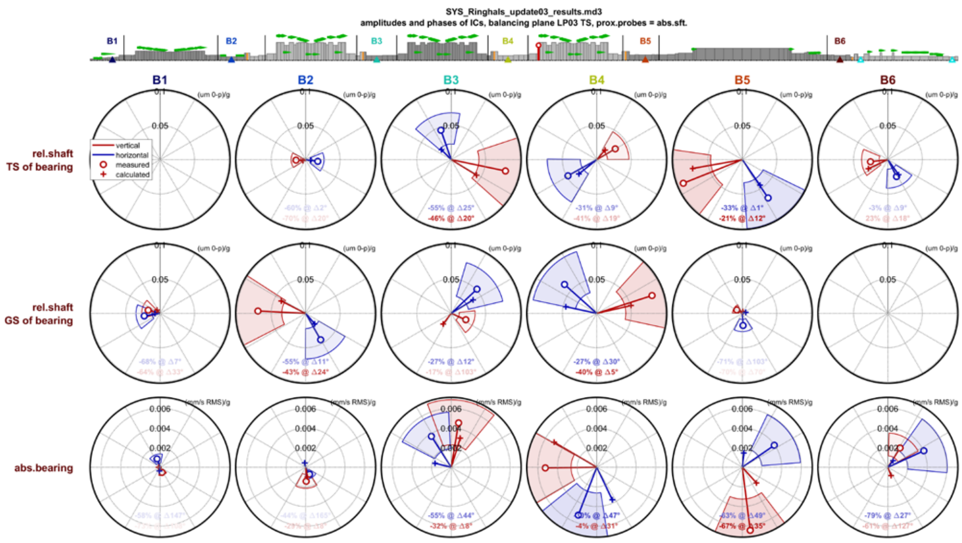
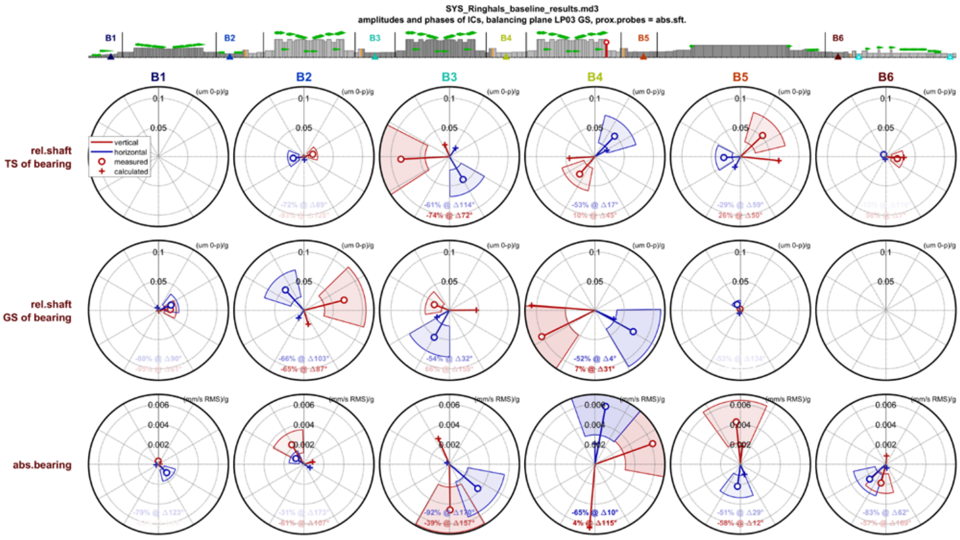


Figure 40 Comparison of calculated and measured Influence Coefficients for the rotational speed of 3000 rpm, Amplitudes and phase angles in Polar Diagrams, Test Weight attached at LP3/TS, Measurement planes on both sides of the Bearings B1 to B6 (rel. vibrations) and on Bearing Pedestals B1 to B6 (abs. vibrations), Rel. shaft vibrations in µm 0-p/g, Abs. Bearing vibrations in mm/sec 0-p/g

BASELINE IC for TEST WEIGHT at LP3/GENERATOR SIDE



UPDATE IC for TEST WEIGHT at LP3/GENERATOR SIDE

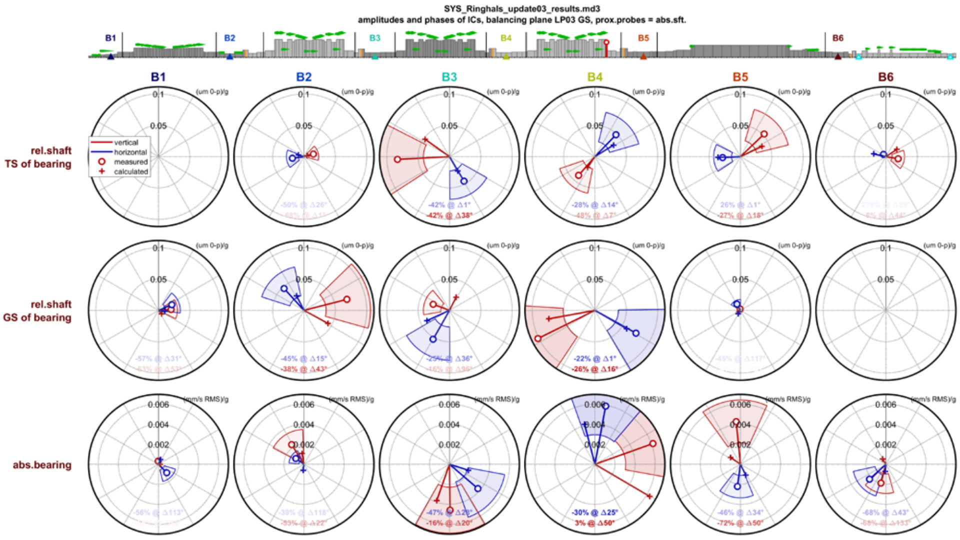


Figure 41 Comparison of calculated and measured Influence Coefficients for the rotational speed of 3000 rpm, Amplitudes and phase angles in Polar Diagrams, Test Weight attached at LP3/GS, Measurement planes on both sides of the Bearings B1 to B6 (rel. vibrations) and on Bearing Pedestals B1 to B6 (abs. vibrations), Rel. shaft vibrations in µm 0-p/g, Abs. Bearing vibrations in mm/sec 0-p/g

8.3 DISCUSSIONS OF POSSIBLE REASONS FOR THE DEVIATION BETWEEN CALCULATED AND MEASURED INFLUENCE COEFFICIENTS

The search for possible reasons of still existing deviations between the calculated and the measured Influence Coefficients has to consider different aspects. One question is, whether the System- and Testing conditions during the measurements (year 2006) still coincide with the assumptions made for the Rotor Dynamic model. It has also to be approved, whether the input/output relations in the tests are the same as in the model. Finally and very important, the quality of the Rotor Dynamic model has to be checked with respect to its structure and the system parameters.

8.3.1 Do the Model Assumptions coincide with the TG 32-System-Conditions from the year 2006?

The measured Influence Coefficients of Ringhals unit TG 32 were taken several years ago in 2006. One reason for the deviations between calculated and measured system responses could be, that the system hardware configuration from today is a different one compared to the year 2006 (e.g. change of the system behavior due to retrofits for HP-Turbine and Generator).

8.3.2 Input- and Output quantities for Measurements and Calculations

Several checks have been made by GE in order to clarify, whether the input and output quantities in the model are the same in the tests in 2006. With respect to the input the unbalance size and the phase angle of the test weights have to be the same for calculation and measurement. It has also to be considered, that the test weight has the correct axial position in the model, regarding the 6 defined balancing planes in the LP turbines. The relative vibration displacements - or the absolute vibration velocities - as output quantities in the calculation process have to be determined for the sensor locations during the measurements. The correct sensor orientations, as they are shown for example for the relative displacements, have also to be considered.

It has also to be considered, that in the Ringhals unit TG 32 the relative shaft vibrations are measured between the seal segments of the turbines and the shaft, as it was shown in Figure 34. This arrangement could not directly be realized in the Rotor Dynamic model, where the relative shaft vibrations are usually considered between the bearing housing and the shaft. However it was observed, that the vibrations of the turbine casing are small and that the beam of the sensor clamping is very stiff. Due to this fact the calculated absolute shaft vibrations at the sensor locations were in a quite good correlation to the corresponding measured relative shaft vibration.

Several checks have been made very careful by Sebastian Abrate GE-Baden in cooperation with Lena Skoglund from Ringhals and Jonas Backlund from GE Sweden in order to obtain the same input and output quantities for the calculations and the measurements.

8.3.3 Influence of the Model Parameters on the Calculated Influence Coefficients

As it was shown in equation (16) the system responses and therefore also the Influence Coefficients depend on the Excitation Forces **F**(t), the operational speed Ω and the dynamic behavior of the turbine train system , expressed by the model matrices **M**, **D**(Ω), **G**(Ω) and **K**(Ω). This means, that the model with its matrices has a strong influence on the quality of the calculated Influence Coefficients.

In general a model is characterized by its structure and its parameters. The structure of the model is determined by the physical laws and the equations of motion, which describe the dynamic behavior. The equations of motion or the structure of the model can be linear or nonlinear. In our application of turbine train systems the equations of motion are usually assumed to be linear, which is for a normal operation in good correlation with the practical experience. And this is - as mentioned before - a very important assumption for the application of Influence Coefficients in balancing processes. With the assumption of linearity the structure of the model is defined and fixed. In this case the quality of the model is only dependent on the quality of the parameters in the above described matrices. The model matrices **M**, **D**(Ω), **G**(Ω) and **K**(Ω) contain mainly mass and stiffness parameters of the shafts of the turbines and the generator, stiffness and damping parameters of the oil film bearings and the mass, damping and stiffness parameters of the pedestals and the foundation.

Shafts of the Turbines and the Generator: It is well known, that the turbine train shafts can be modeled very accurate and that calculated and measured results (e.g. natural frequencies checked by experimental Modal Analysis) are in a very good correlation. Therefore the mass and stiffness characteristics of the shaft components can be considered as very well modeled in the overall turbine system.

Oil Film Bearings: There are some uncertainties to get the right parameter values for the eight stiffness and damping coefficients for each of the 8 oil film bearings. They have a strong influence on the dynamic behavior of the complete rotor train system and therefore on the Influence Coefficients. The rotational speed dependent stiffness and damping coefficients for the nominal case of unit TG 32 have been determined by the project partner and manufacturer GE of the turbine train. As example the stiffness coefficients for the 8 bearings have been shown before in dependence of the rotational frequency in Figure 20 (chapter 5.3). In this chapter it was also explained, that the stiffness and damping coefficients depend on the static shaft position in the bearing, which is a function of the Sommerfeld-number

$$So = p_m \; \psi^2 / (\eta \, \Omega) \qquad (22)$$

Equation (22) points out, that the So-number is a function of the static bearing pressure p_m (static bearing load), of the relative bearing clearance ψ, the viscosity of the lubricant η and of the angular velocity Ω of the rotor. Some of these factors, e.g. the bearing pressure p_m, the relative bearing clearance ψ or the viscosity of the lubricant η may change during operation, e.g. due to temperature changes. As a consequence the static shaft position and the dynamic bearing coefficients will change as well, leading to other system responses and to other Influence

Coefficients. The sensitivity of the dynamic bearing coefficients leads to some uncertainties to obtain the right calculated Influence Coefficients.

Bearing Pedestals and Foundation: It was discussed before that the best Rotor Dynamic model for the complete unit should include the bearing pedestal and foundation dynamics (chapter 5.4). Due to the fact that no foundation data were available a simplified Baseline model was used by GE in a first step. In this Baseline model only the bearing pedestals were considered, each as a SDOF approach with mass, stiffness and damping values for the vertical and horizontal direction (see figures 27 -30 in chapter 5.4). These simplified models are based on investigations by GE for the different types of pedestals. However, the mass-, damping- and stiffness characteristics of the foundation could not be considered in this Baseline system. A remarkable improvement for the calculated Influence Coefficients could be achieved with the Updated model by introduction of the foundation dynamics via the additional measured compliances at the bearing pedestals B2, B3, B4 and B5. It is important to note that only the main diagonal compliances were measured. This was a simplification because all couplings between the measurement points were neglected. Another simplification was that the compliances have not been measured at all pedestals, compliances missing at B1, B6, B7 and B8. It has also to be mentioned, that the shaft weight on jacking oil influences probably the dynamic behavior and the compliances of the bearing pedestal plus foundation system. In summary it can be concluded, that a complete bearing pedestal plus foundation model would further improve calculated system responses and the Influence Coefficients.

9 Conclusions and Recommendations for Future Work

Balancing of Flexible Rotors by means of Influence Coefficients is a well-known method in Rotor Dynamics. The method is applicable for single rotors in a spin pit but also for arbitrary rotor trains on-site, e.g. steam turbine shaft trains in power plants. For a successful application of the method Influence Coefficients are needed and have to be determined in a first step of the procedure. They are defined as the vibration response, relative shaft vibrations or absolute vibration velocities at bearing pedestals, at one measurement point excited by a single unbalance in one of the balancing planes. The required Influence Coefficients for the described balancing process are usually determined by measurements in so called influence runs with defined test weights. This experimental procedure can be very time consuming and expensive. However, Influence Coefficients can also be determined by means of a Numerical Analysis. This needs a very good model for the rotor train, the oil film bearings, the pedestals and the foundation, describing all important dynamic effects. **The main objective of this Energiforsk research project was to determine Influence Coefficients by means of modelling and numerical simulation.**

The procedure was tested for the Ringhals power plant unit TG 32 for the operational speed of 3000 rpm. The finite element model for the complete turbine train was built up with available design data from GE for the system components. Due to the fact that no foundation data were available a simplified **Baseline model** was used at the beginning of the project. In this Baseline model only the bearing pedestals were considered for the supporting structure, each as a SDOF approach with mass, stiffness and damping values for the vertical and horizontal direction. However, the mass-, damping- and stiffness characteristics of the foundation could not be considered. By comparison with measured Influence Coefficients the calculated Baseline results had partly quite large deviations in amplitudes as well as in phase angles. A remarkable improvement for the calculated Influence Coefficients could be achieved with an **Updated model** by introduction of the foundation dynamics via measured compliances at the bearing pedestals B2, B3, B4. This improvement of the calculated Influence Coefficients with the Updated model was confirmed for all load cases with different test weight settings in figures 36 to 41. **With the Updated model it was approved, that a good quality of Influence Coefficients can be achieved by means of calculations with a good Rotor Dynamic model.**

However, although the calculated Influence Coefficients with the Updated model fitted much better to the measured values, there are still some deviations existing. The reasons for these deviations were discussed in chapter 8.3. **In summary it can be concluded that a complete bearing pedestal plus foundation model would further improve calculated system responses and the Influence Coefficients. This should be investigated and confirmed for another power plant unit, where all system data are available.**

ANALYSIS OF THE INFLUENCE COEFFICIENT MATRIX FOR ON-SITE BALANCING OF FLEXIBLE ROTORS

Balancing a nuclear power plant rotor by analyzing the influence of adding weights is time consuming and costly, since each plane is done separately, and it requires running the rotor up and down.

If a digital twin could be created and the influence coefficients calculated, the rotor string could instead be modeled, and compensation weights calculated.

Here Professor Rainer Nordmann from Technical University Darmstadt and senior researchers Eric Knopf, Thomas Krueger and Bastien Abrate from GE Power have created a digital twin of the Ringhals turbine train unit TG 32. The modeled influence coefficients have been compared to previously measured influence coefficients.

The project shows that it is not an easy task to calculate the influence coefficients solely based on knowledge on the different components in the rotor string. Information of the stiffness and structural behavior of the entire structure with pedestals, floor etc. is also required.