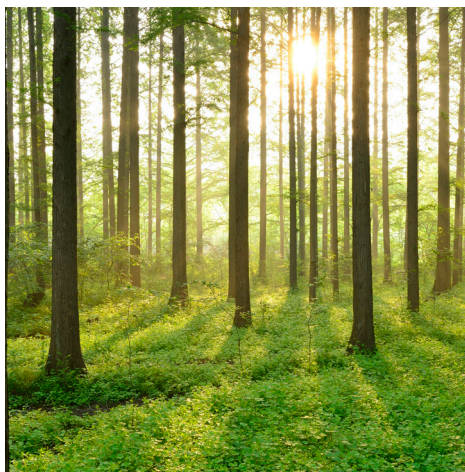


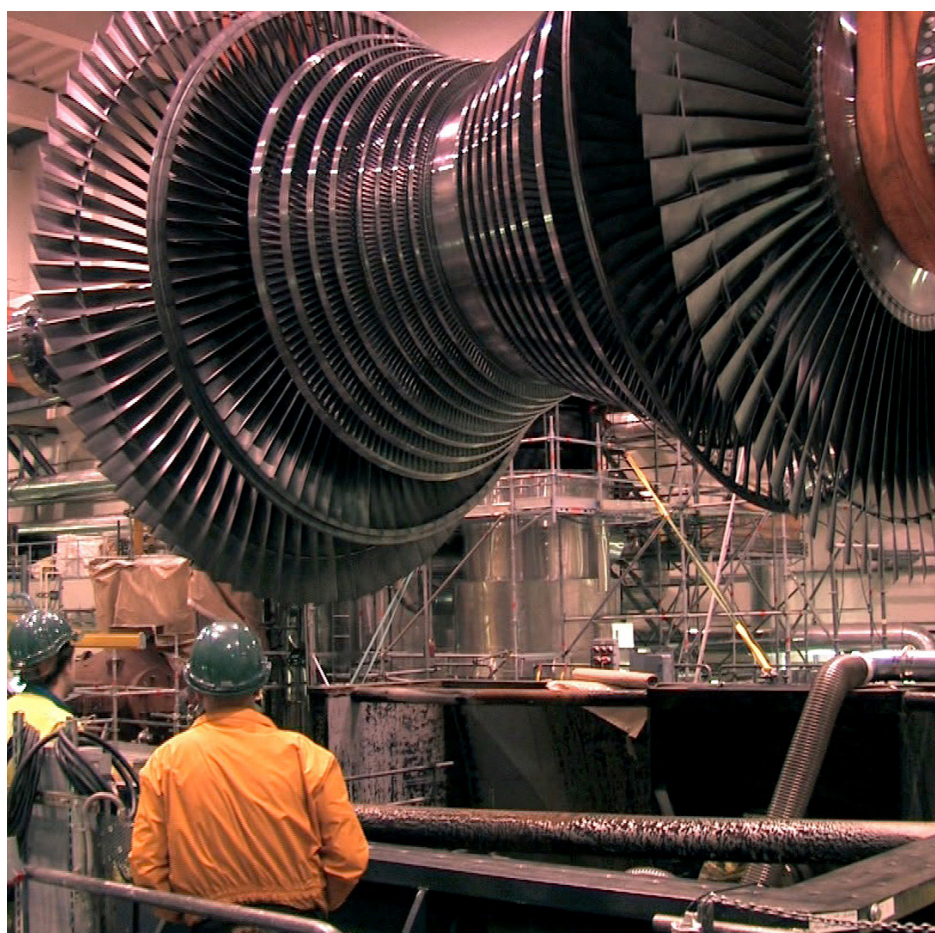
DIAM, A MATRIX TOOL FOR TURBINE AND GENERATOR VIBRATIONS

REPORT 2017:440



NUCLEAR

VIBRATIONS IN
NUCLEAR APPLICATIONS



Energiforsk

DIAM – A Matrix Tool for Turbine and Generator Vibrations

Detection, Investigation, Analysis and Mitigation

RAINER NORDMANN

Foreword

A number of different nuclear power plant turbine- and generator vibration phenomena in Nordic nuclear power plants were mapped in previous projects within R&D program Vibrations. During this work, the idea of developing a tool for structured detection, investigation, analysis and mitigation of vibration phenomena was raised by senior vibration specialist Paul Smeekes at Teollisuuden Voima Oyj.

In this project, the DIAM (Detection Investigation Analysis Mitigation) matrix tool is developed by professor Rainer Nordmann and Paul Smeekes. Professor Nordmann is working at TU Darmstadt and the Fraunhofer Institute. In a parallel project focused on pipe vibrations Mikko Merikoski at FS Dynamics developed Excel templates for the DIAM matrices, these templates are also used in this project. Apart from the matrix development, information from vibration problems in other countries and from other types of power plants was added to the previous information obtained from the Nordic nuclear power plants.

This project has been carried out within the Energiforsk Vibrations in nuclear applications research program. The stakeholders of the Vibrations program are Vattenfall, Uniper, Fortum, TVO, Skellefteå Kraft and Karlstads Energi.

Reported here are the results and conclusions from a project in a research program run by Energiforsk. The author / authors are responsible for the content and publication which does not mean that Energiforsk has taken a position.

Sammanfattning

Huvudsyftet med detta fortsatta projekt "Turbin- och generatorvibrationer - Analys och begränsning" är att få ytterligare kunskap om turbin- och generatorvibrationer i laterala vibrationsfall i andra länders kärnkraftverk och i andra typer av kraftverk. Precis som för rapporterna 2016:294 och 2016:295 om laterala- respektive torsions/statorvibrationer var syftet i detta projekt att skaffa sig djupare kunskap om hur vibrationsproblemen kan identifieras och sedan begränsas. Med resultaten från det föregående projektet grupperades problemen för olika laterala vibrationer, inklusive obalansvibrationer, cykliska vibrationer, friktionsinducerade vibrationer, havsvattentemperaturberoende vibrationer, vibrationer på grund av ojämna tröghetsmoment och instabila vibrationer.

I detta fortsatta projekt har fyra nya laterala vibrationsproblem lagts till, orsakade av axelsprickor, feluppriktning av rotorsträng, fel i de styva kopplingarna i turbin- och generatorrotorerna samt styvhet och dämpande förändringar i stödsystemen. Alla dessa problem med lateral vibration samlades ihop och definierades som totalt 18 vibrationsfenomen. För att lösa problem relaterade till sådana vibrationsfenomen måste problemen detekteras, undersökas och analyseras samt därefter begränsas. För att hitta en optimal lösning för dessa delmoment har ett matrisverktyg tillsammans med ett flödesschema utvecklats, vilket var ett annat viktigt syfte för detta projekt. I de 3 matriserna M1, M2 och M3 definierades de 18 olika vibrationsfenomenen i matrisernas rader som i sin tur jämfördes med 15 olika detekteringsmetoder i matris M1, 18 olika undersökningsmetoder i matris M2 och 10 olika analysmetoder i matris M3, som var och en presenterades i matrisernas kolumner.

Vibrationsidentifieringar börjar med detekteringar och fortsätter med undersöknings- och analysmetoderna. Om en eller flera vibrationsfenomen har identifierats fortsätter proceduren med vibrationsbegränsning. Ett flödesschema har utvecklats för att kontrollera informationsflödet mellan matriserna M1 till M4. Sannolikhetstal används i dessa matriser för att identifiera vibrationsfenomen i de beskrivna identifieringsstegen som följs av valet av en lämplig begränsningsstrategi.

Proceduren som används för matriser och flödesdiagram för detta projekt fungerar med linjära sannolikhetsprocedurer, organiserade i Excel-tabeller. Denna procedur för turbin- och generatorvibrationer har utvecklats enligt en procedur från ett annat Energiforsk-projekt gällande rörvibrationer i kraftverk. Matrisverktyget har, med goda resultat, testats för allvarliga laterala vibrationsfenomen. En förbättring med mer sofistikerade sannolikhetsprocedurer baserade på Bayesian Networks verkar emellertid vara möjligt och bör undersökas under ytterligare studier på kraftverken.

Resultaten från detta projekt kan vara till stor hjälp för att lösa vibrationsproblem med laterala turbiner och generatorer i kraftverk på ett mer effektivt sätt än tidigare. Proceduren bör nu testas och förbättras genom att välja sannolikhetstal baserat på den individuella erfarenhet som specialisten för vibration och övervakning har på respektive kraftverk. Resultaten från projektet är också mycket viktiga för att kunna överföra kunskap till ny personal i kraftverken.

Summary

The first objective of this continuation project “Turbine and Generator Vibrations – Analysis and Mitigation” is to assemble further knowledge in the area of turbine and generator lateral vibration cases in Nuclear Power Plants of other countries and in other kind of Plants. As in the project before (report 2016:294 and 2016:295) a further objective is to study in some depth how vibration problems can be identified and then mitigated. With the results of the previous project a grouping into different lateral vibration problems was achieved, including unbalance vibrations, cyclic vibrations, friction induced vibrations, sea water temperature dependent vibrations, vibrations due to unequal moments of inertia and unstable vibrations.

In this continuation project four new lateral vibration problems have been added, which are caused by shaft cracks, by misalignment in shaft trains, by errors in the rigid couplings of the turbine and generator rotors and by stiffness and damping changes in support systems. All these lateral vibration problems were assembled together and defined as a total of 18 Vibration Phenomena. In order to solve the problems related to such Vibration Phenomena the two tasks of Identification - with the subtasks Detection, Investigation and Analysis - and Mitigation have to be managed. For an optimal solution of these tasks a matrix tool together with a flow chart has been developed, which was another important objective of this project. In 3 matrices M1, M2, and M3 the 18 different Vibration Phenomena were defined in the rows of the matrices versus 15 different detections in matrix M1, 18 different investigations in matrix M2 and 10 different analysis methods in matrix M3, each presented in the columns of the matrices.

The Vibration Identification starts with Detections and continues with Investigations and Analysis Methods. If one or even more Vibration Phenomena have been identified, the procedure continues with the Vibration Mitigation. A flow chart has been developed to control the flow of information between the matrices M1 to M4. Probability numbers are used in these matrices in order to identify the Vibration Phenomena in the described Identification steps followed by the selection of a suited Mitigation strategy.

The used matrix and flow chart procedure of this project works with linear probability procedures, organized in Excel tables. This procedure for turbine and generator vibrations has been developed in accordance with a procedure for Power Plant pipe vibrations in another Energiforsk project. The matrix tool has been tested for several lateral Vibration Phenomena with good results. However, an improvement with more sophisticated probability procedures based on Bayesian Networks seems to be possible and should be investigated in further studies in the Plants.

The findings of this project can be very helpful, to solve lateral turbine and generator vibration problems in Power Plants in a more efficient way as before. The procedure should now be tested and improved by selecting probability numbers based on the individual experience of the vibration and monitoring specialists in the Power Plants. The results of the project can be very useful as well to transfer knowledge to new personnel in the Power Plants.

Expressions

Banana Orbit	Orbit of lateral vibration with a banana shape
Circular Orbit	Orbit of lateral vibration with a circular shape
Elliptical Orbit	Orbit of lateral vibration with an elliptical shape
Forward frequency	Orbit rotation in direction of rotor rotation
Backward Frequency	Orbit rotation in opposite direction of rotor rotation
Single spectrum	Frequency spectrum with forward frequencies only
Full spectrum	Frequency spectrum with forward & backward fr.
Cyclic vibration	Vibration with continuously rotating vector
Counter Balancing	Balancing a rotor against a bow
Laval rotor	Rotor model, named after a Swedish engineer Laval
Operat. Modal Analysis	Determination of vibration mode at operation
Polar plot	Presentation of amplitude and phase in polar diagram
Proximity probe	Sensor type to measure relative shaft vibrations
Spin Pit	Balancing tunnel to balance rotors

Abbreviations

M1	Matrix relates Vibration Phenomena and Detections
M2	Matrix relates Vibration Phenomena and Investigations
M3	Matrix relates Vibration Phenomena and Analysis Methods
M4	Matrix relates Vibration Phenomena and Mitigations
CoV	Causes of Vibration
DoV	Detections of Vibrations
IoV	Investigation of Vibrations
AoV	Analysis of Vibrations
MoV	Mitigation of Vibrations
1xN	Vibration with rotational frequency
2xN	Vibration with two times rotational frequency
3xN	Vibration with three times rotational frequency
0.5xN	Vibration with half rotational frequency
HPT	High Pressure Turbine
LPT	Low Pressure Turbine
GEN	Generator Rotor
NPP	Nuclear Power Plant
MW	Power in Mega Watt
FEM	Finite Element Method
OMA	Operational Modal Analysis

List of content

1	Introduction – Project Description	10
1.1	Objective of the Project	10
1.2	Scope of the Task	10
2	Vibration Problems observed in other Plants	12
2.1	Introduction	12
2.2	Sources of Information	12
2.3	Lateral Vibrations in Shaft Trains due to Cracks	12
2.4	Lateral Vibrations in Shaft Trains due to Misalignment	16
2.5	Lateral Vibrations in Shaft Trains due to Coupling Errors	20
2.6	Lateral Vibrations in Shaft Trains due to Support Stiffness Changes	24
3	Extended Vibration Problem Areas	32
3.1	New Grouping into Lateral Vibration Problem Areas	32
3.2	Identification of the Lateral Vibration Problems	32
3.3	Mitigation of the Lateral Vibration Problem	37
4	Information for the Matrix Development	39
4.1	Introduction	39
4.2	List of Vibration Phenomena (COV-Cause of Vibration)	39
4.3	List of Detections of Vibrations (DoV)	40
4.4	List of Investigations of Vibrations (IoV)	41
4.5	List of Analysis Methods of Vibrations (AoV)	43
4.6	List of Mitigation of Vibrations (MoV)	44
5	Matrix and Flow Chart Development	45
5.1	Introduction – Concept of Matrix Development & Flow Chart	45
5.2	Development of Matrix M1 – Phenomena versus Detection	46
5.2.1	Structure of Matrix M1, Input Data	46
5.2.2	Data Processing in M1 Excel Table and Output Data	47
5.3	Development of Matrix M2 – Phenomena versus Investigation	49
5.3.1	Structure of Matrix M2, Input Data	49
5.3.2	Data Processing in M2 Excel Table and Output Data	49
5.4	Development of Matrix M3 – Phenomena versus Analysis	51
5.4.1	Structure of Matrix M3, Input Data	51
5.4.2	Data Processing in M3 Excel Table and Output Data	51
5.5	Development of Matrix M4 – Phenomena versus Mitigation	53
5.5.1	Structure of Matrix M4, Input Data	53
5.5.2	Data Processing in M4 Excel Table and Output Data	53
5.6	Further Use of the Developed Matrices and Flow Chart	53
6	Appendix – Physical Description of new Lateral Vibration Problems	55
6.1	Lateral Vibrations in Shaft Trains due to Cracks	55
6.1.1	Introduction	55

6.1.2	Physical Description of Lateral Vibrations in case of a Cracked Rotor	55
6.1.3	Examples of Methods to evaluate Rotor Cracks from Vibration Measurements	63
6.2	Lateral Vibrations in Shaft Trains due to Misalignment	64
6.2.1	Introduction	64
6.2.2	Physical Description of Lateral Vibrations in case of Angular Misalignment	64
6.2.3	An Example for Vibrations in a Shaft Train due to Misalignment	68
6.3	Lateral Vibrations in Shaft Trains due to Coupling errors	69
6.3.1	Introduction	69
6.3.2	Physical Description of Lateral Vibrations due to Coupling Errors	69
6.4	Lateral Vibrations in Shaft Trains due to Support Stiffness Changes	72
6.4.1	Introduction	72
6.4.2	Physical Description of Lateral Vibrations due to Support Stiffness Changes	73

1 Introduction – Project Description

1.1 OBJECTIVE OF THE PROJECT

The objective of this project is to further assemble knowledge and experience in the area of turbine and generator vibrations and to study in some depth how they were investigated and mitigated. The continuation of the previous project with the same title shall now include also vibration problems that have occurred in NPPs in other countries and in other kind of plants (relevant problems that could also occur in NPPs). In this continuation project the already started Matrix tool shall further be developed. In 4 matrix presentations the different vibration problems will always be related to different detection, investigation and mitigation methodologies:

- Matrix 1 – monitoring methodologies
- Matrix 2 – deeper investigation methodologies
- Matrix 3 - analytical or numerical methodologies
- Matrix 4 - mitigation methodologies

A flow chart will also be presented as a guideline how to use these matrices for the solution of existing vibration problems.

The collected new information of vibration problems and the development of the Matrix-Tool with the flow chart will be described in this report. It can be used for knowledge transfer to new personnel as well as for the case of planned changes in turbine trains or when problems occur in the turbine trains.

1.2 SCOPE OF THE TASK

The scope of the overall task of this project can be subdivided into the following subtasks:

1. To assemble further information about Vibration phenomena (reports, publications, conference proceedings) from Nuclear Power Plants (NPPs) in other countries and other power plants with large turbine trains.
2. The new information and documentation will be grouped into problem areas as in the previous project. This means, that the structure of the work will again not be plant specific but will be focused on the different vibration phenomena encountered.
3. The development of the already started Matrix tool from the previous project will be continued. This includes the development of the 4 matrices M1 to M4. In these matrices the different vibration phenomena will be related to selected detection-, investigation-, analysis- and mitigation- methodologies. For the detection of a vibration phenomenon Matrix 1 shows the relation between an irregularity detected by some existing monitoring system, inspection or other normal plant monitoring procedure and vibration problems.

For a further Investigation of the vibration problem Matrix 2 shows relations between the vibration problems and the different ways to perform a more in depth investigation to ensure the source of the problem.

If the vibration problem needs to be further studied by analytical, numerical or experimental analysis methodologies, Matrix 3 shows which methodologies are best to further investigate the vibration problem.

After Identification of the vibration problem by means of the Matrices 1, 2, 3 Matrix 4 presents different possible methodologies for the Mitigation of the identified Vibration problem.

Finally a Flow chart has been developed as a guideline on how to use the 4 Matrices M1 to M4 for the complete solution of an existing vibration problem.

This Energiforsk Research project was suggested by members of the Vibrations Steering group of the four power plants Olkiluoto (Finland), Oskarshamn, Forsmark and Ringhals (Sweden). The geographical location of this plants are shown in Figure 1.



Figure 1. Geographical location of the power plants in Sweden and Finland.

2 Vibration Problems observed in other Plants

2.1 INTRODUCTION

Subject of the investigations in the previous project have been vibration problems in the turbine generator shaft trains of the three power plants Oskarshamn, Olkiluoto and Forsmark. One task of this continuation project is to find new vibration problems for large turbine trains. Information about vibration phenomena in other plants has only be obtained from literature and personnel contacts, as briefly described in 2.2 Sources of information.

2.2 SOURCES OF INFORMATION

Sources of information are Proceedings from Rotordynamic Conferences (IFTToMM, VIRM (IMechE), SIRM, ASME (since about 1980), International Journals, Vibration Problem Data collections, e.g. the v_Base Vibration Problem Data Rotating Machinery from JSME in Japan, special publications from Polytechnico Milano about Vibration Problems of Rotating Machinery in Power Plants (Italy) and the Orbit Journal and the Diagnostic Handbook from Bently Nevada (USA). Good personal contacts with colleagues in Japan (Prof. Masato Tanaka and Prof. Osami Matsushita) and in Italy (Prof. Paolo Pennachi and Prof. Nicolo Bachschmid), all of them specialists in vibration and diagnosis of Rotating Machinery helped to better understand the published turbine and generator vibration problems.

Valuable sources of information have also been received from the Engineers in the Swedish and Finish Power Plants. Two corresponding publications, written by Leif R.K. Nilsson (STAL LAVAL) and Ylva Vidhög (Forsmark) are briefly described in chapters 6.1.3 and 6.2.3.

The author of this report is also involved in a University Network in Europe, USA, Asia and Australia and Chairman of the ISO Working group Rotordynamics and Vibrations of Machines. He is also a member of the IFTToMM Technical Committee Rotordynamics and has practical experience from 3 years work (2009 until 2012) as Manager Rotordynamics at ALSTOM Power in Baden (Switzerland) and from consulting activities at TVO from 1996 until 2009.

This chapter contains a collection of Vibration Problems from the Information Sources described above.

2.3 LATERAL VIBRATIONS IN SHAFT TRAINS DUE TO CRACKS

CASE 1: PP Oak Creek, LP Steam Turbine, Unit 7, P=320 MW, n=1800 rpm

VIBRATION PHENOMENA: Lateral Vibrations, caused by a transverse shaft crack. Increase of $1xN$, $2xN$ and $3xN$ per revolution vibrations in a low pressure steam turbine during steam temperature reduction operation (boiler deslagging operation). The increased response of Rotor vibrations is due to the asymmetric stiffness changes, induced by the growing crack.

DETECTION METHOD: The crack was detected by vibration signals of the relative shaft vibrations and absolute Bearing vibrations at the Low pressure steam turbine. Signature Analysis of Vibration data recorded over a two year period prior to crack identification are correlated to fatigue crack growth, which occurred intermittently during transient temperature decreases. Unfortunately the paper only refers to the amplitudes of the 1xN, 2xN and 3xN vibration components. Nothing has been reported about the corresponding phases (Fig.3)

INVESTIGATION & ANALYSIS: Vibration and Fracture Mechanics analyses suggested the presence of a transverse shaft crack, which was identified by ultrasonic inspection. A fatigue crack propagation analysis was performed.

MITIGATION OF PROBLEM: Even with the detected crack, it was decided to continue operation of unit 7 with revised procedures. Steady state full load operation showed no trend towards vibration increase. The crack was only dangerous and propagating at special conditions. The operating restrictions continued until the unit was shut down, ultrasonically inspected and the rotor removed for repair.

SOURCE OF INFORMATION: NASA Conference Publication 2250, Rotordynamic Instability Problems in High Performance Turbomachinery

PROBLEM CASE OR RESEARCH STUDY: This is a practical problem case. It occurred in a power plant of Oak Creek, Unit 7 (USA).

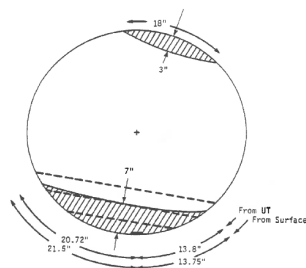


Figure 2. Oak Creek, Unit 7 LP turbine shaft cracking.

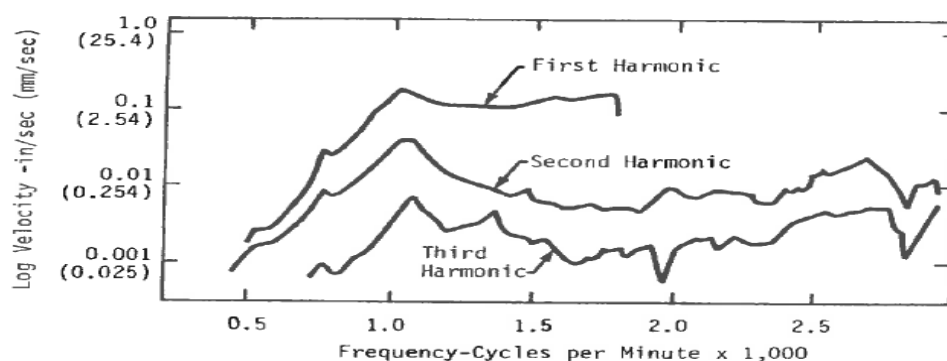


Figure 3. Amplitude vs. Frequency for 1xN, 2xN and 3xN Harmonics.

CASE 2: PP No Name: LP Steam Turbine, P=670 MW, Detection of Cross Sectional Cracks

VIBRATION PHENOMENA: Lateral Vibrations, caused by a transverse shaft crack. Increase of overall vibrations, mainly in the first low pressure steam turbine. The increased response of rotor vibrations is due to the asymmetric stiffness change, induced by the growing transverse crack.

DETECTION METHOD: Bearing and Shaft vibrations of this machine were monitored. Two sensors are used in the vicinity of the bearings at both ends of each cylinder. Shaft vibrations were measured contactless with Inductive sensors. The strongest vibration increase was observed in the first Low Pressure Steam Turbine. Only overall vibrations were measured. However, it is assumed, that vibrations mainly consist of $1xN$ and $2xN$ Harmonics.

INVESTIGATION & ANALYSIS: Machine was shut down, when overall vibration values reached $400 \mu\text{m}$.

MITIGATION OF PROBLEM: In this publication no further information was presented with respect to Mitigation.

SOURCE OF INFORMATION: IFToMM Conference Proceedings Sevilla (Spain) 1987, Session Proceedings Rotordynamics. Lecture: Vibration Monitoring..., H.J. Bohnstedt, Allianz Centre, Munich (Germany)

PROBLEM CASE OR RESEARCH STUDY: This is a practical problem case, that occurred in a power plant with a three cylinder 670 MW saturated Steam Turbine (see Figure 4)

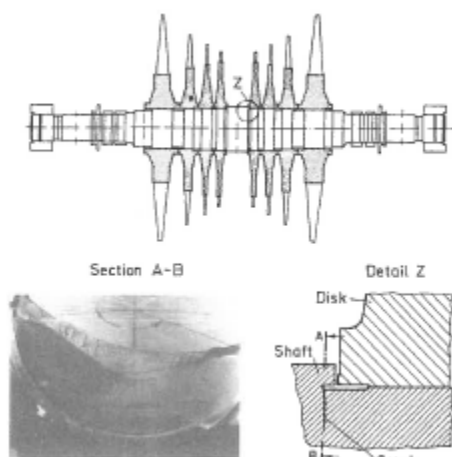


Figure 4. Fatigue Crack in the LP shaft of a 670 MW Turbine.

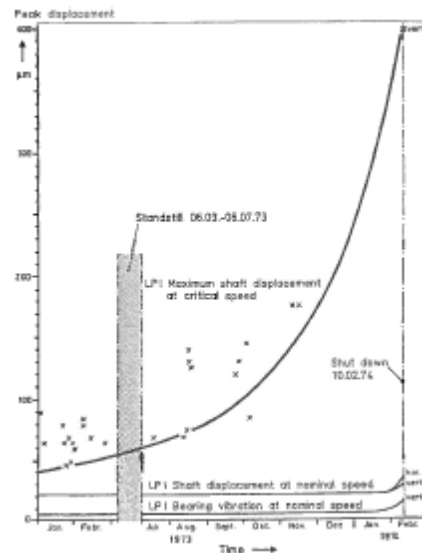


Figure 5. Increase of vibrations due to crack propagation in the LP shaft.

CASE 3: Research: Use of directional Spectra for Diagnosis of SHAFT CRACKS in Rotor Systems

VIBRATION PHENOMENA: Vibrations of rotating shafts with unequal Moments of Inertia (Shaft Asymmetry) usually show forward as well as backward $1xN$ and $2xN$ frequency components in their spectra. Shafts with a transverse crack are typical examples for Shaft Asymmetry, where additional frequency components may occur in the spectra due to nonlinearities (gap opening and closing in the cracked shaft).

DETECTION METHOD: Bearing and Shaft vibrations are measured - in the usual way by means of vibration sensors. They measure absolute bearing vibrations and relative shaft vibrations. By signal processing the two sided spectrum with forward and backward frequencies can be determined.

INVESTIGATION & ANALYSIS: A method for the diagnosis of asymmetry in Rotor Systems, using the two sided directional spectrum is presented and tested with a laboratory Rotor Bearing System Test Rig (see Fig. 6). The experimental results show, that the directional spectra can be effectively used for the diagnosis of asymmetry (e.g. shaft cracks) by investigation of $\pm 1xN$, $\pm 2xN$ and higher components (see Figure 7).

MITIGATION OF PROBLEM: In this publication Mitigation of Asymmetry is not considered.

SOURCE OF INFORMATION: IFToMM Rotordynamics Conference Proceedings Chicago (USA) 1994, Use of Directional Spectra for Diagnosis..., Chon Won Lee et al, page 97.

PROBLEM CASE OR RESEARCH STUDY: This is a Research study, using the two sided directional spectra in order to identify forward and backward frequency components in a rotating shaft with unequal moments of inertia (Asymmetry). These spectra can be very useful for the diagnosis of Shaft Cracks (see also chapter 5.1).

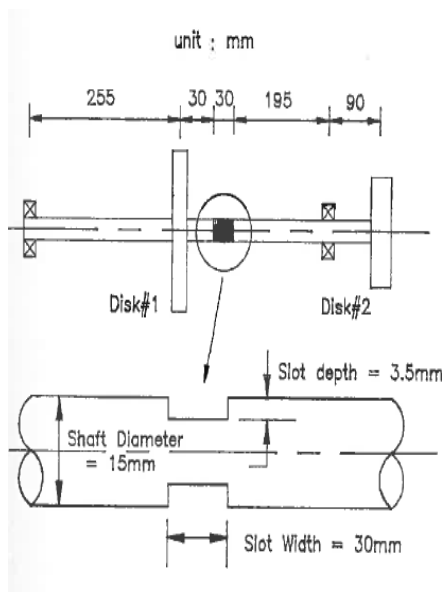


Figure 6. Location and configuration of a shaft slot.

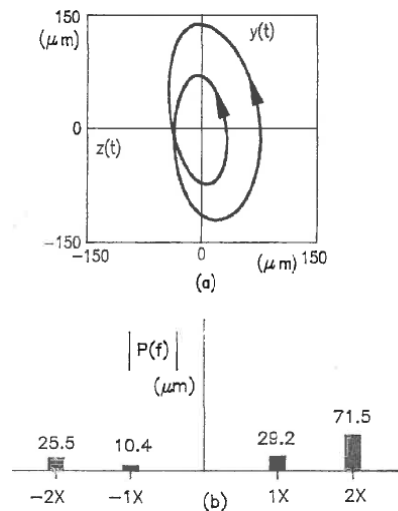


Figure 7. Response of the slotted system a) whirl orbit, b) directional spectrum.

2.4 LATERAL VIBRATIONS IN SHAFT TRAINS DUE TO MISALIGNMENT

Preliminary Remarks for chapters 2.4 and 2.5 regarding different **Bearing configurations**: The lateral vibration behavior of shaft trains due to Misalignment (chapter 2.4) and due to Coupling Errors (chapter 2.5) is influenced by the Bearing configuration. We distinguish between the two Bearing concepts:

- the **Double Bearing Concept (DBC)**. Each single rotor is supported in two bearings. In other words, two Bearings are located at two adjacent rotors. Therefore, if the number of rotors is N , the number of Bearings will be $2N$.
- the **Single Bearing Concept (SBC)**. In this case only one Bearing is located at two adjacent rotors. If the number of rotors is N , the number of Bearings will be $N+1$.

Simple Example: If a rotor system consists of 3 rotors, the Double Bearing Concept leads to 6 Bearings, the Single Bearing Concept to 4 Bearings.

Examples of Nuclear Power Plants in Sweden and Finland: (without Exciter).

Oskarshamn 3: $N = 5$ (HPT, 3xLPT, GEN). Single Bearing Concept (SBC) with $N+1 = 6$ Bearings.

Forsmark TG3: $N = 5$ (HPT, 3xLPT, GEN). Single Bearing Concept (SBC) with $N+1 = 6$ Bearings.

Ringhals TG32: $N = 5$ (HPT, 3xLPT, GEN). Single Bearing Concept (SBC) with $N+1 = 6$ Bearings.

Olkiluoto OL1&2: $N = 6$ (HPT, 4xLPT, GEN). Single Bearing Concept (SBC) with $N+1 = 7$ Bearings.

Olkiluoto OL3: N = 5 (HPT, 3xLPT, GEN). Double Bearing Concept (DBC) with 2N = 10 Bearings.

In the following chapters the cases 5, 6 and 7 (see Figures 9, 12, 15) belong to the Double Bearing Concept, case 4 belongs to the Single Bearing Concept.

In general the following aspects have to be considered, when we evaluate the today more often used Single Bearing Concept. The Single Bearing Concept:

- reduces the number of Bearings
 - increases the distances between the Bearings
 - increases the Bearing loads for the inner Bearings
 - reduces probably the natural frequencies due to larger Bearing distances.
 - influences the damping in the Bearings because of different Bearing pressure
 - makes the Alignment of the shafts at erection of the train more difficult.
- However, the sensitivity of the Bearing loads to Misalignment is lower, because of more distance between the Bearings.

CASE 4: PP No Name: 3-Cylinder (HP, IP, LP)-Turbogenerator, P=120 MW, n=3000 rpm

VIBRATION PHENOMENA: Misalignment is a static phenomenon and cannot directly be considered as a vibration problem. However due to Misalignment static forces and displacements can change in the rotor system and this may finally lead also to a change in the dynamic characteristics. If for example two shafts of a turbine train have a radial or angular misalignment, static shaft bending can occur when the two shafts are coupled rigidly together. Due to the compulsive forces (moments) in the rigid coupling the static forces in the bearings, the journal displacements and the dynamic coefficients of the oil film will change as well. Depending on the change of the static bearing forces, the bearing can be unloaded or can be higher loaded, compared to the aligned situation. With regard to lateral vibrations this may lead to instability ($1/2 \times N$), change of unbalance vibrations ($1 \times N$) or to vibrations with higher Harmonics ($2 \times N$), usually with a “Banana Orbit”. In the described problem case thermal changes in the relative heights of bearing pedestals were shown to cause bearing No. 1 (HP Turbine) to become unloaded, leading to instability and excessive vibrations.

DETECTION METHOD: Lateral vibrations were measured in terms of absolute velocities in mm/sec at the bearings and as relative shaft displacements in μm . In Bearing No. 1, the journal riding in the center of the bearing and was taking little or no load. Bearing 2 was adequately loaded. From the frequency analysis a sub-synchronous component at $1/2 \times N$ confirmed the oil film instability.

INVESTIGATION & ANALYSIS: By means of a computer program, bearing loads have been determined in dependence of the Bearing locations (Alignment) for the statically undetermined Rotor Bearing System.

MITIGATION OF PROBLEM: The load of Bearing No. 1 needed to be increased by either raising Bearing No. 1 or lowering Bearing No. 2 (Bearing Alignment). Technically the easier way was to lower Bearing No.2, by a displacement of 0,64 mm (Result of the calculation).

SOURCE OF INFORMATION: IMechE, Conference Proceedings Vibrations in Rotating Machinery, Cambridge 1976, Paper C191/76, Vibrational Problems in Modern Power Station Plant, Author: W. Davies et al.

PROBLEM CASE OR RESEARCH STUDY: This is a practical problem case, that occurred in a power plant with a three cylinder 120 MW Turbogenerator.

Brg. 1 **Brg. 2**

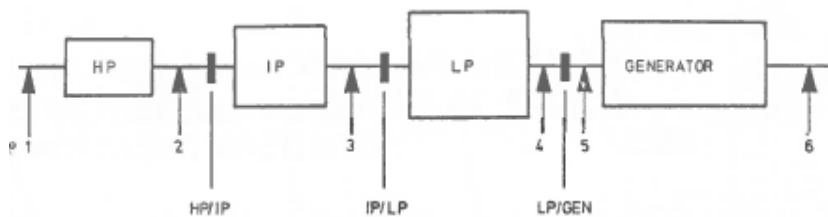


Figure 8. Configuration of a three cylinder Steam Turbine.

CASE 5: ENEL Power Plant, Steam Turbine with HP, IP and LPT, P=240 MW, n=3000 rpm

VIBRATION PHENOMENA: After several years of operation, during cold start-ups high vibrations began to occur at Bearings 2 and 3 (see Fig. 9 and 11). The vibrations were so high (200 μm p-p), that the machine had to be shut down. After a few hours on the turning gear, a new start up was possible (see Fig. 11). From the beginning attention was paid to a possible misalignment that developed during the warm up of the turbine. Measurements illustrated, that the IPT rose relative to HPT and LPT, especially on the HPT side. They also showed a twisting movement of the T-beam, supporting Bearings 2 and 3 (see Fig. 10). This rotation is the cause of the vertical misalignment, which caused rubbing of the shaft on Bearing 2, which in turn produced the high vibrations due to a thermal bow during cold start-ups. Further examinations indicated, that the problem might well be initiated by a non-completely free expansion of the IP-cylinder, because of unusually high friction between the supporting pads and the foundation plate of the T-beam (see Fig. 10).

DETECTION METHOD: Lateral vibrations were measured in terms of absolute velocities in mm/sec at the bearings and as relative shaft displacements in μm .

INVESTIGATION & ANALYSIS: 1) An instrumentation was used capable of monitoring the changes in vertical alignment of the Bearings. 2) By means of a computer program, the distribution of the bearing loads had been determined in dependence of the Bearing locations (Alignment).

MITIGATION OF PROBLEM: The difficulty was overcome by lowering Bearing 3 by 0.2 - 0.3 mm to compensate for part of the rise of the Bearing in the warm condition. This avoided the rubbing and the high vibrations which occurred in the critical phase of the start-up. Vibrations after alignment correction never exceeded 80 μm (see Figure 11).

SOURCE OF INFORMATION: IFToMM Rotordynamics Conference Proceedings, Rome (Italy) 1982, M. Cadeddu, et.al.: Vibration Problems on Large Rotating Machinery, page 469

PROBLEM CASE OR RESEARCH STUDY: This is a practical problem case, that occurred in a power plant in a Steam Turbine with HPT, IPT, LPT and Generator, 240 MW, 3000 rpm

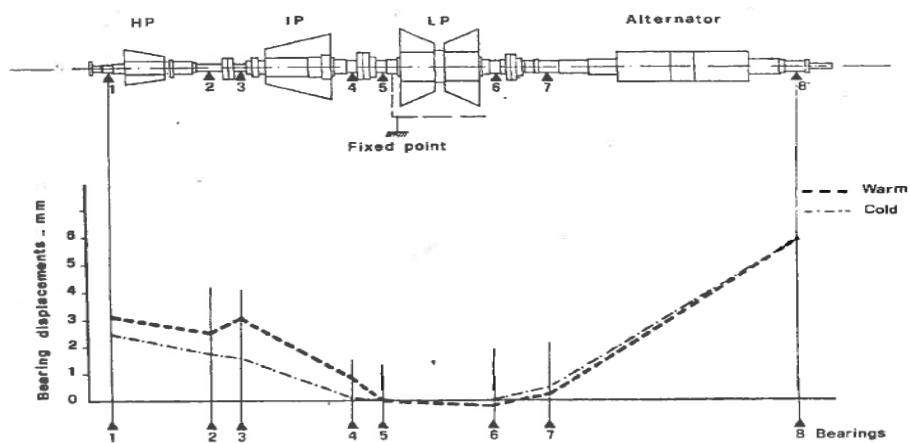


Figure 9. 240 MW Steam Turbine Bearing levels at cold and warm condition.

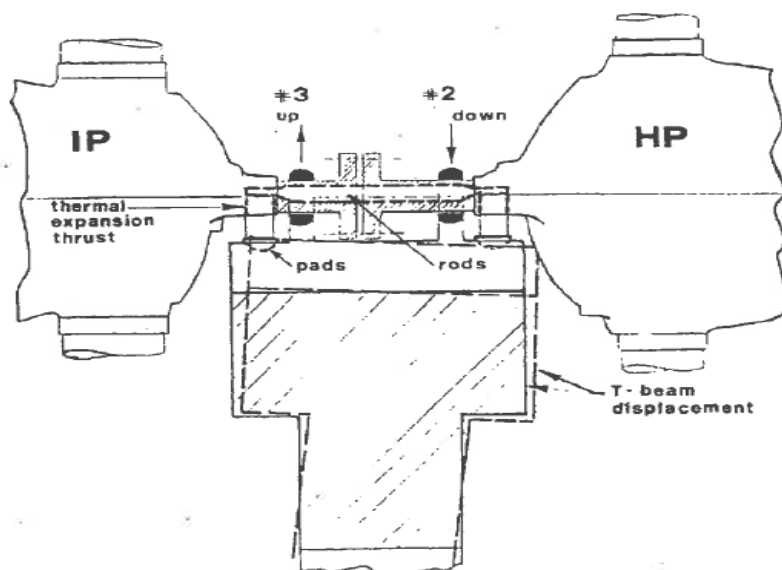


Figure 10. 240 MW Turbine – Schematic details of the turbine to illustrate how pad friction may develop twisting of the T-beam and Bearing misalignment.

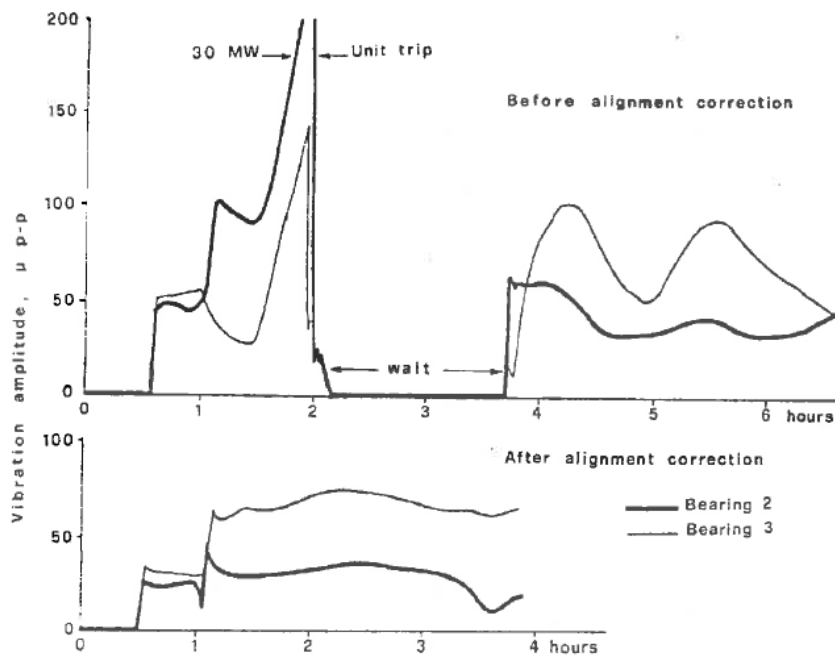


Figure 11. 240 MW Turbine – Vibrations recorded during cold start up before Alignment (upper diagram) correction and after Alignment correction (lower diagram.)

2.5 LATERAL VIBRATIONS IN SHAFT TRAINS DUE TO COUPLING ERRORS

CASE 6: PP No Name: 5-Cylinder (HPT, IPT, 3x LPT)-Turbogenerator, P=500 MW, n=3000 rpm

VIBRATION PHENOMENA: A 500 MW turbogenerator was returned to service with a replacement generator rotor, taken from an identical machine. This rotor showed high vibration levels (100 μm p-p) at full speed and also during run-down, especially at the generator inboard Bearing No. 11 (see Fig. 12). Relative high amplitudes of higher modes lead to the conclusion, that the source of vibration was at one end of the rotor. By some tests normal alignment errors were ruled out and the vibration levels were also insensitive to changes in the rotor excitation current. At this point the conclusion was, that the problem was caused either by a change in the state of balance of the generator or by a permanent bow in the shaft line. The most probable cause was a bow at the LP3/Generator coupling. Such a bow can be caused by coupling errors (please distinguish between misalignment of bearings and coupling errors in the rotor). It was more likely here, that a coupling error (swash of the coupling faces) is the source of a bow and the vibrations. It is obvious, that bolting together a normal coupling face with one containing a small degree of swash is equivalent to introducing a permanent sharp bow in the shaft line between the two adjacent Bearings. A solution to reduce the bow could be to insert a suited spacer between the two coupling parts. In this way the coupling error can be corrected.

DETECTION METHOD: Lateral vibrations were measured in terms of absolute velocities in mm/sec at the bearings and as relative shaft displacements in μm at Bearing locations.

INVESTIGATION & ANALYSIS: By means of a Rotor Dynamic computer program it was calculated, which bow at the LP3/generator coupling is needed to obtain the observed level of vibration response of $100 \mu\text{m}$ p-p. The height of this bow leads to a value of $h = 50 \mu\text{m}$ (see Fig. 14). With this value and a mathematical model the equivalent face swash $s = 50 \mu\text{m}$ 0-pk at either coupling face (total run-out = $100 \mu\text{m}$) was determined. This value was too high and the plant operators were advised to investigate the LP3/generator coupling for swash of this order. The coupling was therefore examined and dismantled. It was found a swash of $38 \mu\text{m}$ on the generator coupling face and $51 \mu\text{m}$ on the other coupling side. That means that the total swash was roughly $90 \mu\text{m}$ pk-pk at the LPT coupling face. And this caused a significant bend of about $100 \mu\text{m}$ pk-pk in the shaft line when the coupling was bolted up.

MITIGATION OF PROBLEM: The spacer was machined to compensate for the generator coupling face swash. When the machine was recommissioned, the on load vibration level at the generator had been reduced to less than 30 % of the previous value.

SOURCE OF INFORMATION: IMechE, Conference Proceedings Vibrations in Rotating Machinery, Cambridge 1976, Paper C191/76, W. Davies et.al.: Vibrational Problems in Modern Power Station Plant.

PROBLEM CASE OR RESEARCH STUDY: This is a practical problem case, that occurred in a power plant with a five cylinder 500 MW Steam Turbine (see Figure 12).

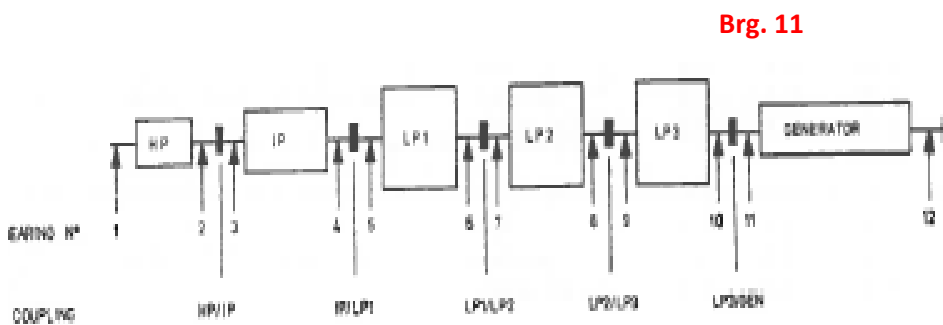


Figure 12. Configuration of a 500 MW five Cylinder Turbogenerator.

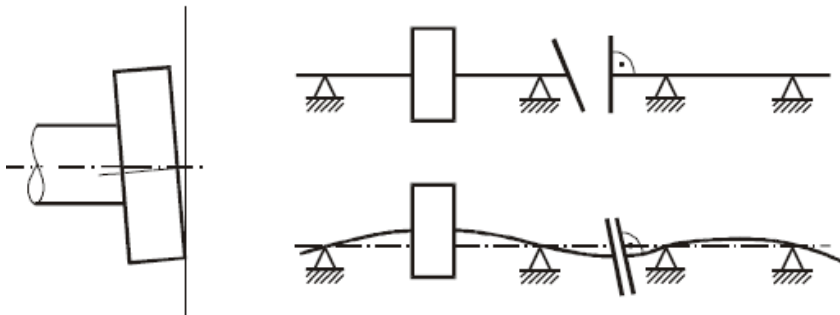


Figure 13. Bow due to a coupling error in the LP3/generator coupling.

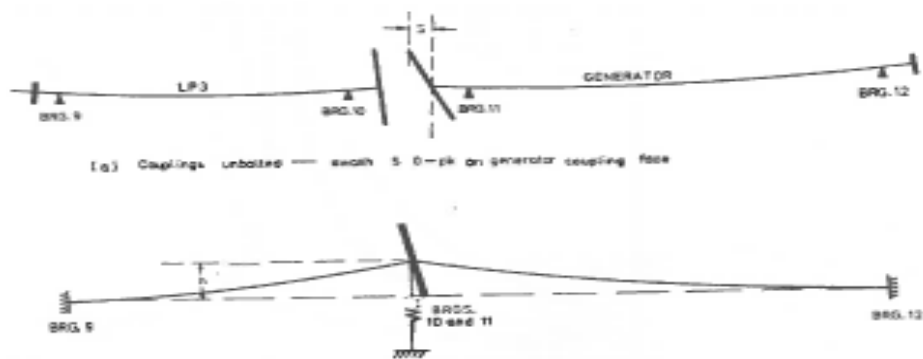


Figure 14. Model illustrating the effect of coupling face swash.

CASE 7: PP No Name: Steam Turbine-Generator Rotor System, P=320 MW, n=3000 rpm

VIBRATION PHENOMENA: This vibration phenomena was observed on a 320 MW Steam Turbine Generator System with an operating speed 3000 rpm (see partial view with proximity probes in Fig.15). The measured $1xN$ vibrations of $110 \mu\text{m}$ pk-pk at Bearing No. 2 were relatively high compared to normal limits. When Bearing No. 2 had been disassembled for clearance inspection, eccentricity measurements were carried out with the HP rotor supported only by Bearings No. 1 and 3 (see Fig. 15). The measured static shaft eccentricity $100 \mu\text{m}$ pk-pk was very high. In the past a heavy rub between the HP-Turbine rotor and stator elements had occurred during a run up. This rub caused some damage to seals and partly to Bearing No.2. No evident damage was detected on the shaft. However the detected high eccentricity lead to the assumption, that a permanent local bow of the HP Turbine rotor was caused by the rub event.

DETECTION METHOD: Lateral vibrations were measured in terms of absolute velocities in mm/sec at the bearings and as relative shaft displacements in μm at Bearing locations.

INVESTIGATION & ANALYSIS: As the HP-turbine rotor with a bow could not be replaced in a short time, it was decided to investigate the possibility of reducing

the dynamic effects of this permanent bow by introducing a taper in the existing spacer between the HP and IP Coupling flanges (see Fig. 16). The insertion of such a spacer involves a change in the shaft eccentricities, whose phase and magnitude depend on the value of the angle defined by the lateral surfaces of the spacer. The amplitude of this angle should be calculated by a mathematical model in order to obtain a rotor distortion, due to the spacer, equal to the permanent bow, which was measured near the journal bearing No. 2. The tapered spacer must be mounted with the correct angular position in order to ensure that the two bows lie in the same plane but with opposite phase. If these conditions are satisfied, the permanent distortion can be compensated by the effects caused by the spacer. The Spacer design could be supported by numerical analysis for the rotor system with oil film bearings, by determining Frequency Response Functions and the Unbalance response at Bearing No. 2. As a result of the numerical investigations a 40 μm flatness correction of the spacer was suggested.

MITIGATION OF PROBLEM: As a mitigation the described tapered spacer was inserted between the HP IP coupling flanges. A first check on the turning gear showed that an eccentricity reduction at Bearing No.2 had been obtained. Afterwards a machine run-up was carried out. A comparison of the 1xN Unbalance response at Bearing No. 2 during this run-up and before inserting the spacer is shown in Figure 17. The insertion of the tapered spacer between the HP-IP Coupling flanges partially reduced the high vibration levels at Bearing No. 2, while no significant changes in vibration levels occurred at Bearings No. 1 and 3.

SOURCE OF INFORMATION: IMechE, Conference Proceedings Vibrations in Rotating Machinery, Bath 1992, Paper C432/067, G.L. Lapini et.al.: Correction of vibration problems on an HP steam turbine..., page 233.

PROBLEM CASE OR RESEARCH STUDY: This is a practical problem case, that occurred in a power plant with a 320 MW Steam Turbine (see Figure 15).

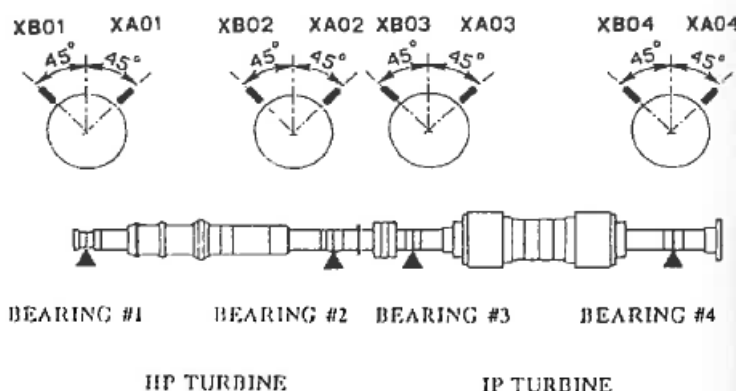


Figure 15. Partial model of the Turbine Train with HP and IP Turbines.

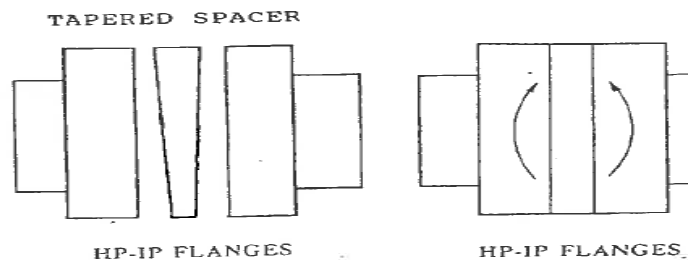


Figure 16. Tapered spacer inserted between the HP-IP Coupling. Caculation shows equivalent moment at the two nodes of the Finite Element Model.

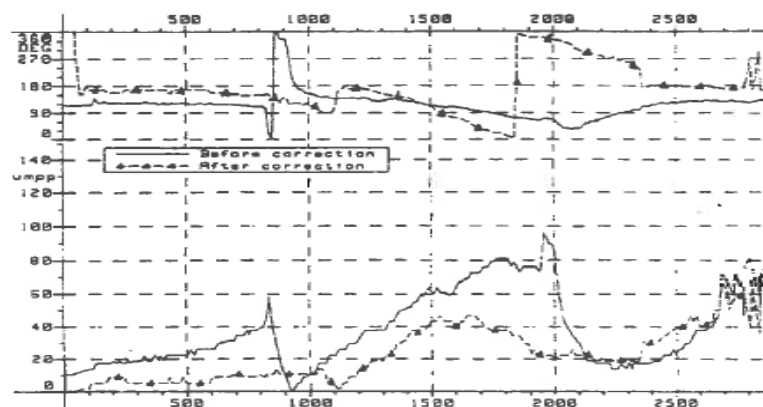


Figure 17. Relative Vibrations at Bearing No. 2 during run-ups before and after correct.

2.6 LATERAL VIBRATIONS IN SHAFT TRAINS DUE TO SUPPORT STIFFNESS CHANGES

CASE 8: Own Experience from Power Plant Turbine Trains, manufactured by ALSTOM. Influence of the supporting structures (Bearing pedestals, Casings and Foundation).

VIBRATION PHENOMENA: In order to achieve an optimal lateral vibration behavior for Steam Turbine Trains in Power Plants, the Turbine manufacturers use Shaft Train Models (E.g. Finite Element Models) and simulate the Rotordynamic behavior: Natural frequencies, Critical Speeds, Stability and Unbalance Response. The quality of the predictions regarding the lateral vibrations depends very much on the quality of the model. To achieve a good model, all important effects regarding the Rotor Dynamic behavior shall be considered. This includes the inertia, damping and stiffness characteristics of the different turbine rotor shafts, the stiffness and damping coefficients of the oil Film Bearings and Seals (Rotor-Fluid-Interaction) and the inertia, damping and stiffness contributions from the supporting structures: Bearing housing, Bearing pedestal, Turbine and Generator casings and last but not least the Foundation (Rotor-Structure-Interactions). Well suited Unbalance distributions are further assumed in order to predict realistic Unbalance Response Vibrations. It is well known, that the overall stiffness or flexibility of the supporting structure with the above mentioned components may strongly influence the dynamic behavior of the overall Rotor-Bearing-Support-System. It is therefore very

important to have a good model for all components of the supporting structure. This is possible today with the powerful Finite Element Method, but needs some effort for realization. An alternative way, to include the dynamic behavior in the overall model, is to introduce measured or (modal-) calculated Frequency Response Functions at the Bearing locations, which include the complete frequency dependent dynamic characteristic of the supporting structure. The experimental determination of such Frequency Response Functions has been successfully performed by ALSTOM (ABB, BBC) in Power Plants. For this purpose the bearing casing mounted on the foundation is excited by a hydraulic exciter, fixed in the middle of the bearing. Frequency Response Functions can then be measured at all points of interest. The procedure can also be used for Generator bracket Bearings.

DETECTION METHOD: Figure 18 shows one of the Bearing pedestals of a 1000 MW Turbine train, mounted on the foundation of this Power Plant. The hydraulic exciter is fixed in the center of the Bearing. Excitation is possible in vertical as well as in horizontal direction. Force time signals can be either of harmonic type for different frequencies or of sweep type for a selected frequency range. Response signals can be taken at all Bearing locations (horizontal and vertical), either as absolute velocities or accelerations.

INVESTIGATION & ANALYSIS: With the described method a selected set of Frequency Response Functions can finally be determined at the connecting points to the rotor, which are all Bearing locations along the shaft line. The result of this Investigation and Analysis is the measured Flexibility Frequency Response Function for the Bearing Block and for the Foundation in vertical direction, see Figure 19. In the lower frequency range the flexibility is determined mainly by the foundation (Pedestals is rigid). In case of higher frequencies increasing relative movements between Bearing pedestal and foundation can be detected until a strong natural frequency of the Bearing pedestal occurs. The flexibility in the horizontal direction is normally much greater and is determined more by the Bearing pedestal itself than by the foundation. Figure 20 shows the Frequency Response Function for the horizontal case at the pedestal. The resonance is much lower than in vertical direction.

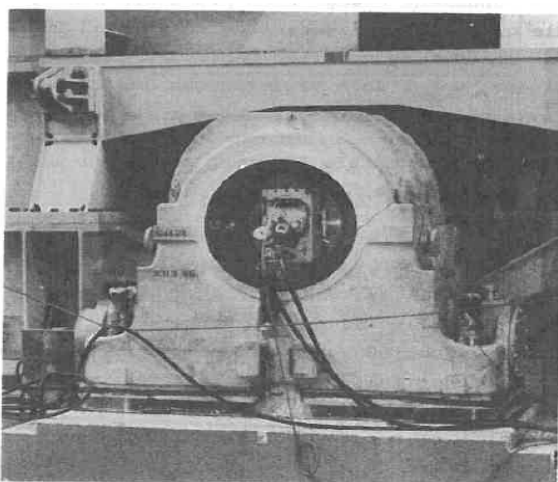


Figure 18. Experimental Set up for the measurement of Frequency Response Functions of the supporting System: Bearing Pedestal and Foundation.

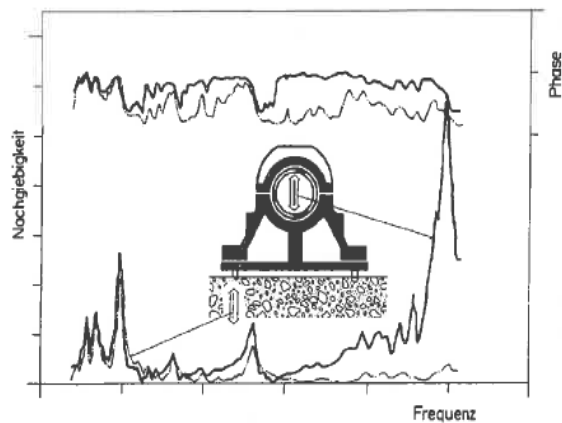


Figure 19. Flexibility Frequency Response Functions for the Bearing pedestal (bold curve) and the foundation (thin curve) in vertical direction.

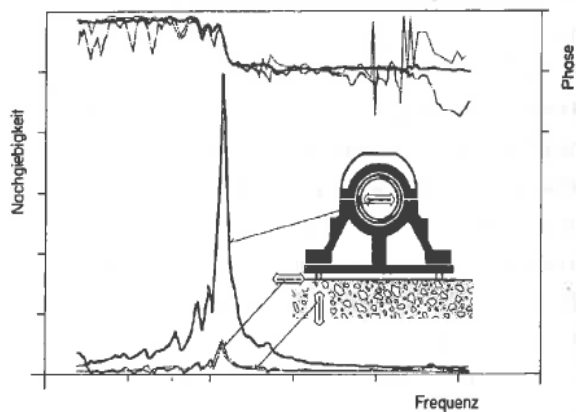


Figure 20. Flexibility Frequency Response Functions for the Bearing pedestal (bold curve) and the foundation (thin curve) in horizontal direction.

CASE 9: Possible Vibration phenomena due to Support Stiffness Changes Experience from Power Plant Turbine Trains, manufactured by ALSTOM

VIBRATION PHENOMENA: In Case 8 the influence of the dynamic characteristic of the supporting structure (Bearing housing, pedestal and foundation) on the overall Rotor-Bearing-Support-System Dynamics has been briefly described by Frequency-Response Functions. These functions include all dynamic effects in terms of stiffness-, damping- and inertia-forces with respect to the supporting structure. If the vibrations of the complete system are changing, this may be caused by a change of the excitation (e.g. unbalance forces) or by a change in the system dynamics. Change in the system dynamics may be on the Rotor-Oil-Film-Bearing side or on the supporting structure side. There may be manifold causes for a vibration change, if the supporting structure is responsible for the vibration problem. Very often increased vibrations occur, when the Rotor-Bearing-System is not well connected to the foundation. This often happens for a singular Bearing only showing high vibrations, concentrated at this Bearing location. Resonance effects may amplify the vibration amplitudes. For the identification of the vibration problem it is often necessary to have a clear view about the force transfer from the

shaft via the bearings, the bearing pedestals to the foundation. From a machine dynamics point of view it is important to know the dynamic behavior of the complete coupled system. Therefore it is also important to know the support stiffness in vertical and horizontal direction and their influence on the dynamic behavior of the complete system. The vibrations of the system may increase continuously over a long period. Due to settlement of the concrete the foundation bolts loose pre-stress. Settlement usually leads to a reduction of the horizontal and vertical stiffness values. In Rotors, running in under-critical conditions, vibrations will increase. The opposite will appear if Rotors are running over-critical.

DETECTION METHOD: During normal operation lateral vibrations are continuously measured in terms of absolute vibration velocities in mm/sec at the bearing housings and as relative shaft vibrations in μm between the shaft and the Bearing housing. If the information from detection is not sufficient to find out the cause of the vibration phenomena, further investigations and analysis work is needed.

INVESTIGATION & ANALYSIS: For a better identification of the vibration phenomenon regarding support stiffness changes the following investigations may be helpful:

- Check of the clearance between the bearing supporting ring and the bearing casing. Compare with the recommended clearance.
- Check the white metal surface of the bearing.
- Measure the vibration velocities in different heights of the Bearing casing (Points A, B, C in Figure 22) for the 4 corners (see Fig. 21 and 23).

MITIGATION OF PROBLEM: Depends on the specific problem case. Examples for Mitigation are:

- Fix the clearance between the bearing supporting ring and the bearing casing to recommended values.
- Correct the pre-stress in the foundation bolts. Information from one of the Swedish nuclear stations: **It is a standard procedure to check every year that the foundation bolts are tightened.**

SOURCE OF INFORMATION: The information is based on the experience from the author.

PROBLEM CASE OR RESEARCH STUDY: Knowledge from different practical problem cases.

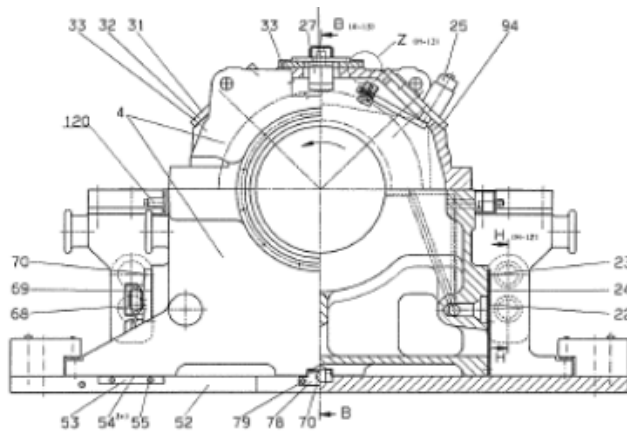


Figure 21. Bearing pedestal (Casing) with base plate.

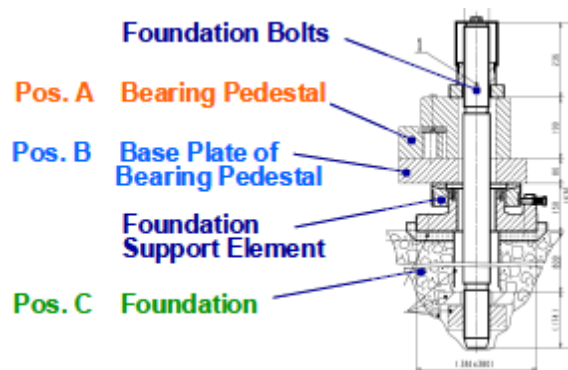


Figure 22. Connection of the Bearing pedestal with base plate to the foundation with foundation bolts.

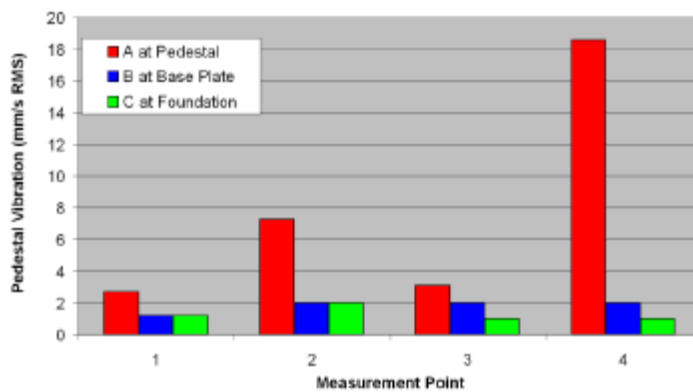


Figure 23. Vertical Vibrations in mm/sec RMS at locations A, B, C (see also Fig. 22) for the 4 corner points of the Bearing pedestal (see also Fig. 21).

CASE 10: PP No Name: Steam Turbine-Generator Rotor System, P=320 MW, n=3000 rpm

VIBRATION PHENOMENA: This vibration phenomenon was observed on a Steam Turbine Generator System (HP, IP, LP1, LP2, GEN) with a power of 320 MW, operating at speed 3000 rpm (see partial view with proximity probes in Fig.24). The Shaft train has 11 Bearings. The HP and IP turbines are equipped with Tilting Pad Bearings (Figure 25). With the proximity probes the relative shaft vibrations of the machine were measured and analyzed during all its working conditions, e.g. run-ups, run-downs and load variations. The vibration response contains mainly the 1xN frequency component due to Unbalance excitation. Measured vibration values (amplitudes and phase) at Bearings No. 1 and 3 always showed normal behavior. Contrary to that, Bearing No. 2 had much higher vibrations. Figure 26 shows the static mean shaft locations in μm and the relative shaft vibrations in μm pk-pk for Bearing No. 2 for different values of the generated Power. Very high static displacements of 885 μm and also high relative shaft vibrations of 255 μm have been detected at P = 190 MW. Since the static displacement of 885 μm was much larger than the design diametrical clearance of 450 μm for Bearing No. 2 it was evident, that something was wrong with either the Bearing clearance or fixing. To clarify this open question, further Bearing investigations became necessary.

DETECTION METHOD: The Bearings of the machine are each equipped with two proximity probes placed at +/- 45 ° from the vertical for measuring the relative vibration between shaft and Bearing housing as well as the shaft position inside the Bearing (Static displacement).

INVESTIGATION & ANALYSIS: The inspection of Bearing No. 2 showed, that the clearance was found at its design value. But an unwanted clearance of 570 μm was discovered between the ring that supports the pads and the Bearing housing (see Fig. 25, locations I and II).

MITIGATION OF PROBLEM: The unwanted clearance of 570 μm between the ring and the Bearing housing was removed by proper mechanical working and the machine was restarted. The relative shaft vibrations strongly reduced their amplitudes and also their load dependence. The vibrations were brought back to values that are normal for this kind of machine (see Fig.27). Also the variations of the mean shaft positions reduced significantly.

SOURCE OF INFORMATION: IMechE, Conference Proceedings Vibrations in Rotating Machinery, Bath 1992, Paper C432/067, G.L. Lapini et.al.: Correction of vibration problems on an HP steam turbine...., page 233.

PROBLEM CASE OR RESEARCH STUDY: This is a practical problem case, that occurred in a power plant with a 320 MW Steam Turbine (see Figure 24).

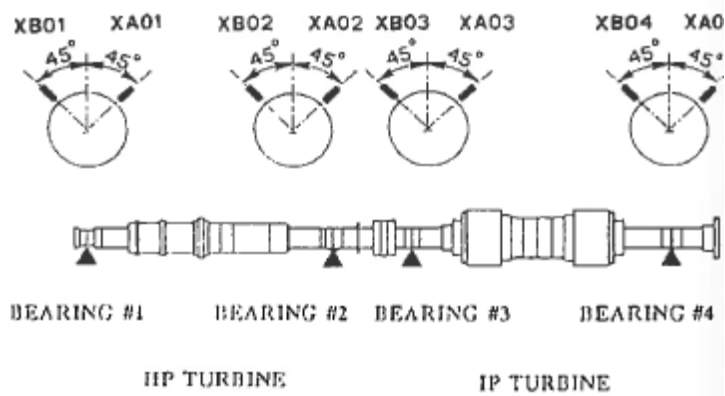


Figure 24. Partial model of the Turbine Train with HP and IP Turbines.

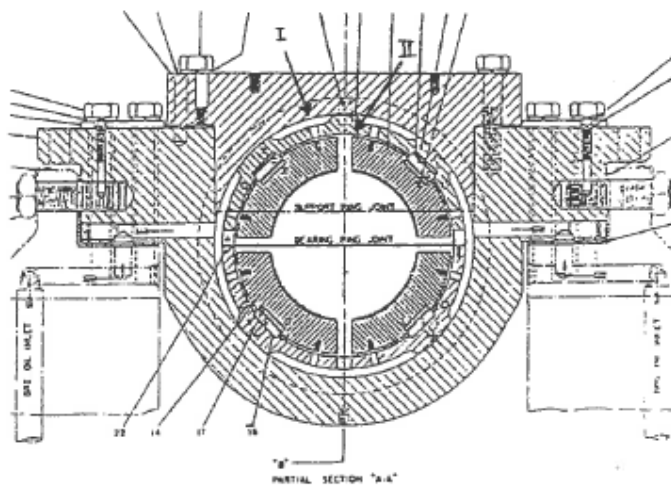


Figure 25. Critical Bearing No. 2 of the Shaft train.

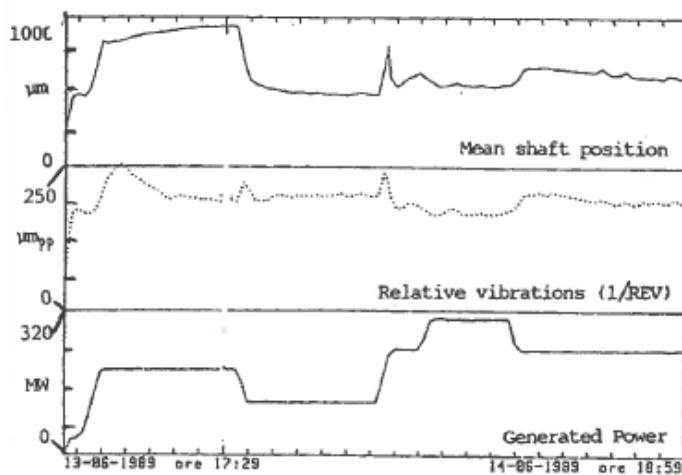


Figure 26. Mean shaft positions and relative vibrations at Brg. No.2 - before Mitigation.

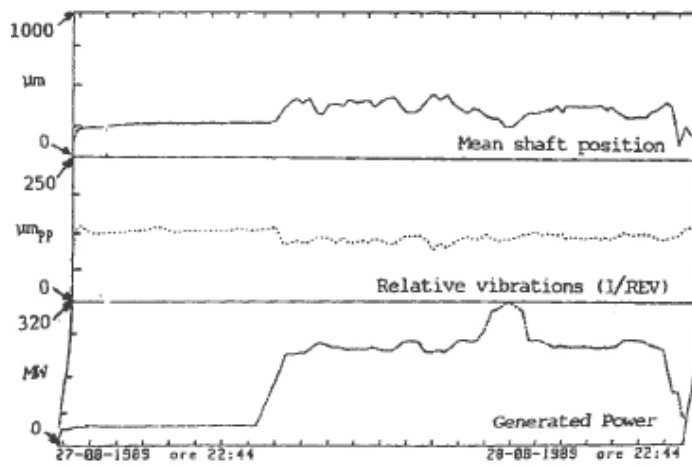


Figure 27. Mean shaft positions and relative vibrations at Brg. No.2 - after Mitigation.

3 Extended Vibration Problem Areas

3.1 NEW GROUPING INTO LATERAL VIBRATION PROBLEM AREAS

In the previous project (2015-2016) the delivered reports from the three power plants Oskarshamn, Forsmark and Olkiluoto and the visits in the power plants and interviews with the vibration specialists resulted in a detailed grouping into vibration problem areas for lateral vibrations and torsional vibrations. Especially for lateral vibrations the following groups had been identified (see former report).

Lateral Vibration Problems from the previous project:

- 1xN Lateral Vibrations in shaft trains due to Unbalance
- Cyclic or Spiral Lateral Vibrations
- Friction Induced Lateral Vibrations in Generators
- Lateral Vibrations due to changes of Seawater temperature
- Lateral Vibrations due to Unequal Moments of Inertia
- Unstable Lateral Vibrations

They are now extended by new Lateral Vibration problems:

- Lateral Vibrations in Shaft trains due to cracks
- Lateral Vibrations in Shaft trains due to Misalignment
- Lateral Vibrations in Shaft trains due to Coupling Errors
- Lateral Vibrations in Shaft trains due to Support Stiffness changes

They are described in Chapter 2 of this report. Deeper physical descriptions can be found in the appendix (chapter 6). In this Matrix Development project we will further concentrate on Lateral Vibrations. Torsional Vibrations are treated in another project. The extended Vibration Phenomena for Lateral Vibrations will be listed in the following chapter 4.2. The 18 listed Vibration Phenomena will later be used as Input Data for the Matrix-Development (Chapter 5).

3.2 IDENTIFICATION OF THE LATERAL VIBRATION PROBLEMS

In case of a Turbine and Generator vibration problem in a Power Plant the very important first task is the Identification of the Vibration Phenomenon. When the problem has been identified, a suitable Mitigation method can be applied to solve the problem. The Identification of a Vibration problem can be subdivided into Detection, Investigation and Analysis.

Detection: Detection is performed with the standard systems in the Power Plant (e.g. Monitoring System with sensors and amplifiers) and inspections that belong to the normal practice in the plants. This includes normal post processing and reporting.

Investigations: As compared to the detection investigations are performed with additional methods to investigate the problem more and assure that the problem source is better known.

Analysis: Analysis is done with the results of Detection and Investigation. An Analysis can also give additional information that is necessary for the Mitigation. It mostly involves a deeper investigation of the problem with some numerical, experimental or analytical Analysis method.

The base for the Identification of Lateral vibration problems in power plants is to measure absolute vibration velocities in mm/sec on the bearings and relative shaft vibrations in μm (displacements) close to the bearings or at shaft ends. Measurements are usually taken in two orthogonal directions (horizontal and vertical or in other directions, e.g. 45 deg.). These measurements belong to the part of Detectio (see Figure 28). As a general rule lateral vibrations should normally always be monitored by vector evaluation (amplitudes and phase) for different multiples.

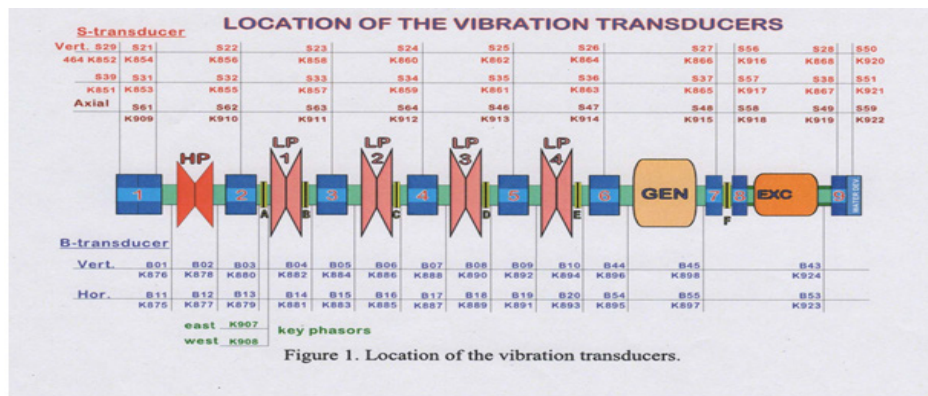


Figure 28. Detection: Locations of the Vibration transducers in a shaft train.

Besides these displacements and velocities some information with respect to temperatures is also of importance to explain a Vibration Phenomenon. As example, the oil film temperature in the Bearings or the temperature of the cooling water in the rotor windings of the Generator have an important influence on the Vibrations of the Turbine train. Also these temperature measurements belong to the Detections. A detailed description of all Detections can be found in Chapter 4.3.

By signal processing different functions in the time or in the frequency domain can further be determined. Some examples are:

- vibration-orbits for different locations of the shaft train
- Frequency spectra to analyze the frequency content, e.g. $1 \times N$, $2 \times N$, $3 \times N$, $1/2 \times N$, higher Harmonics and broadband frequencies
- Static location of the shaft in the Oil Film Bearings
- Vibration orbits at the Bearing locations

These are usually determined by an online monitoring system during normal operation of the shaft train at nominal Speed and Power. Figures 29 to 32 demonstrate this kind of Problem Identification. In this report they are defined to belong to the group of Investigations. In chapter 4.4 all 18 Investigations are

defined and listed. With respect to the frequency spectra it has to be pointed out, that the full spectrum with positive and negative frequency components has usually much more information than a one sided single spectrum.

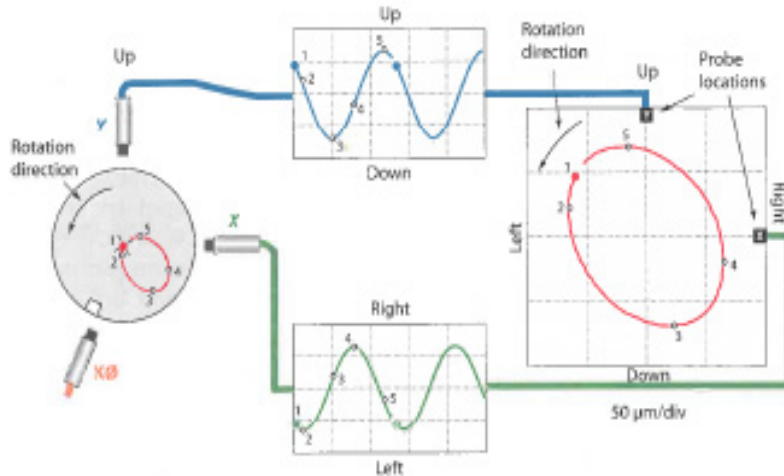
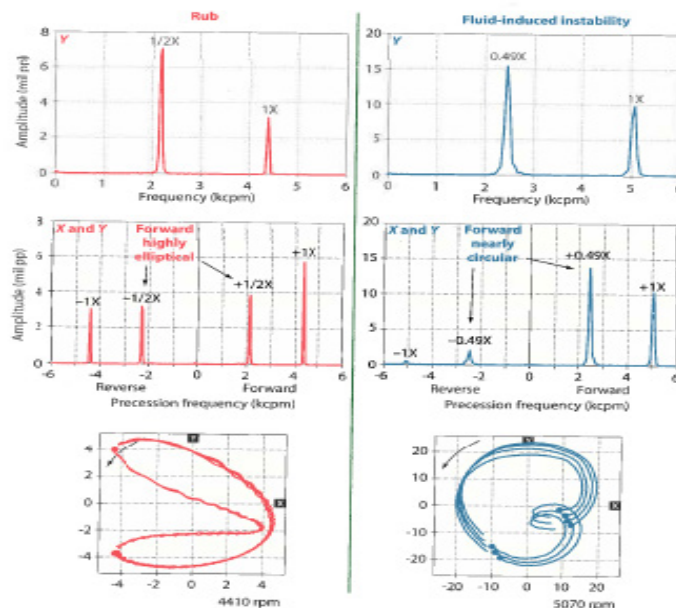


Figure 29. Vibration Orbits at different locations of the shaft train.



Top: One sided -forward frequency – spectrum
 Middle: Double sided – forward and backward – spectrum
 Bottom: Orbits

Figure 30. Frequency spectra to analyze the frequency content of signals.

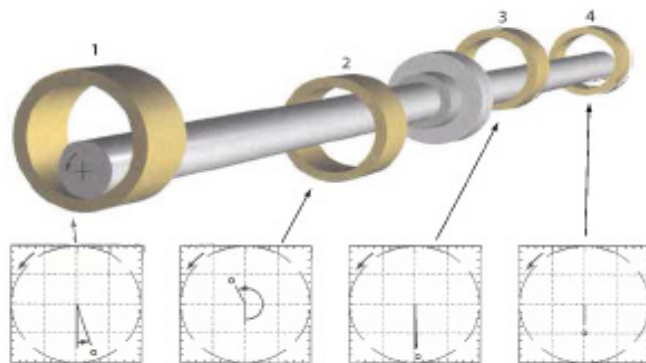


Figure 31. Static location of the shaft in the Oil Film Bearings at normal operation.

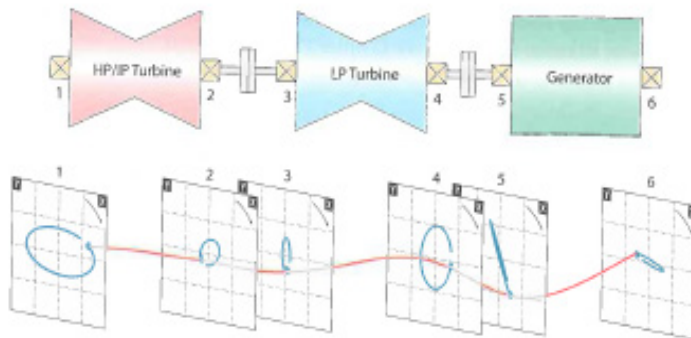


Figure 32. Elliptical Orbits along the shaft line during normal operation.

Deeper investigations of the vibration problem with Analysis methods belong to the last group of Vibration Problem Identification. They include analytical, numerical as well as experimental techniques. Typical presentations of lateral vibrations in the frequency domain are shown in following Figure 33. They result from an experimental Analysis of the shaft train.

- Run up and Run down curves (Power = 0) with $1 \times N$ amplitudes and phase versus rotational speed. $2 \times N$ and higher components can be added.
- Run up and Run down polar plots with amplitude and phase, rotational speed is the parameter in the plot. (Power = 0).
- Polar plot at low speed (e.g. 500 rpm), to check whether the vibration vectors are stable and within a defined control circle before run up.

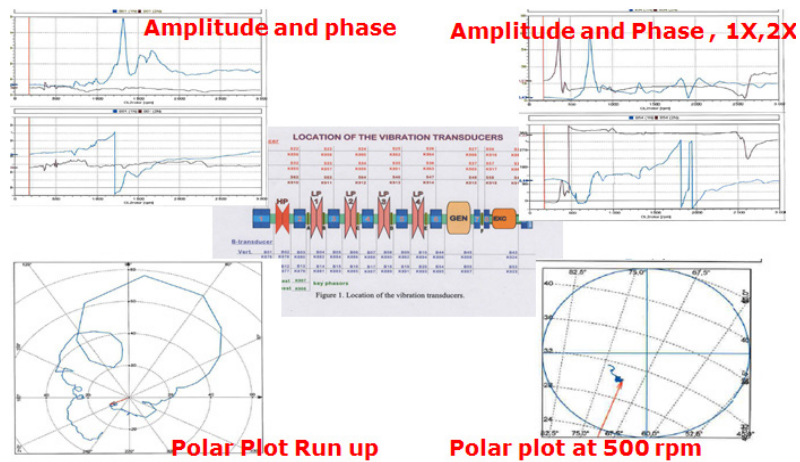


Figure 33. Deeper Investigations of Vibrations (Analysis) for shaft trains.

As a numerical Analysis method the Finite Element method is a powerful tool to study different Vibration phenomena. Figure 34 shows for example the amplitudes of the relative shaft vibrations during a run up.

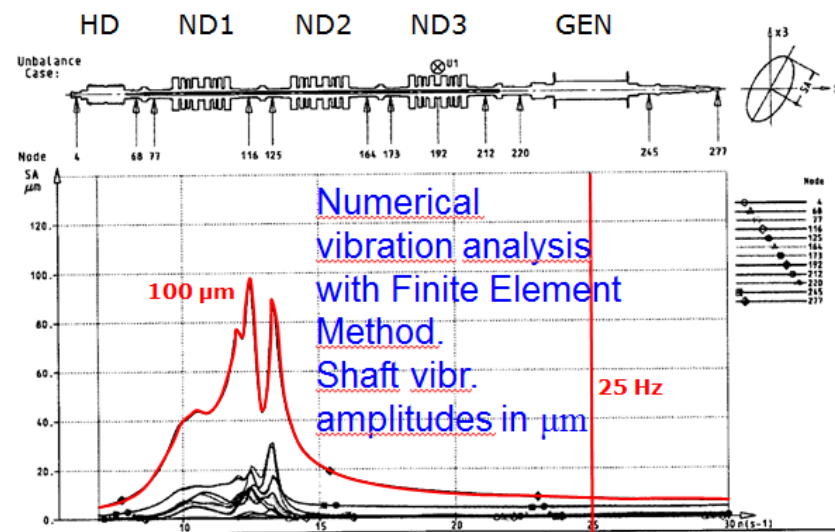


Figure 34. FE Simulation of a Run up curve of a 1600 MW shaft train.

The two presented examples belong to the group of Analysis methods. A total of 10 Analysis methods have been selected. They are defined and listed in Chapter 4.5.

3.3 MITIGATION OF THE LATERAL VIBRATION PROBLEM

After Identification of the Vibration Phenomena in a Turbine shaft train the problem has to be solved by Vibration Mitigation. Typical examples are:

- Balancing (Figure 35),
- Change of Bearing or Bearing parameters to avoid instability,
- Increase damping in Oil Film Bearings,
- Control of temperature of cooling water in Generator windings (see Fig. 36 and 37).
- Alignment of the shaft line.

The total list of Mitigation Methods is described in Chapter 4.6.

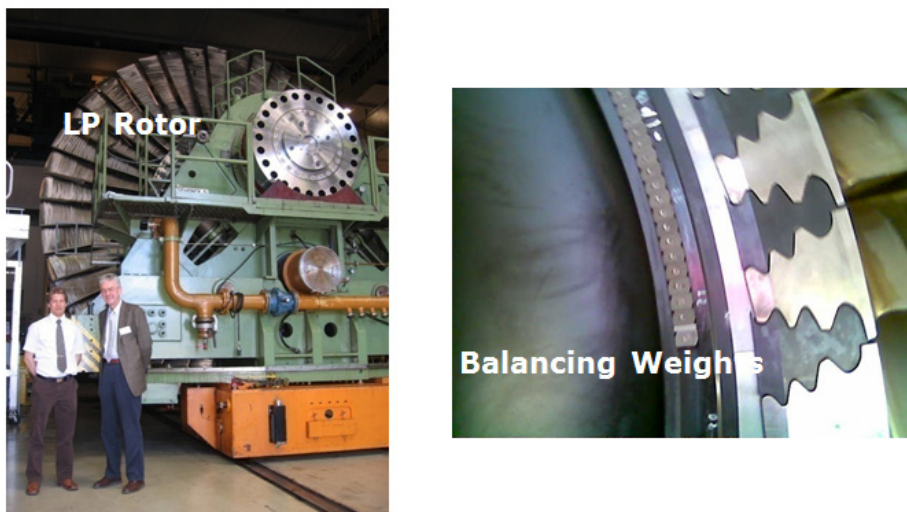


Figure 35. Preparation of a LP Steam turbine rotor for balancing in a Spin Pit.

Table 5. Runup log, VibroCam times.

TIME	RPM	DESCRIPTION
24.2.2015 01:30	20	Runup started from barring.
01:43	500	500 rpm reached.
01:43	500	Generator rotor started to move out as always
02:15	600	Up to 600 RPM, B44 and B45 vibrations increased --> decided to return to 500 RPM
02:20	500	back at 500 rpm.
02:30	500	Rotor temperature was increased +3°C, 1xN vectors moved in favorable direction, More temperature increase would not have improved the situation and as the absolute vibrations were acceptable it was decided to continue with the runup
02:45	500	Runup was continued.
03:20	3000	3000 rpm reached.
03:25	3000	Rotor temperature was increased +5°C to check whether this has influence on the vibrations, As expected and hoped for no changes in the vibration behavior occurred and it was decided that from a vibration point of view there was no obstruction to connect the generator to the grid.
03:43	3000	Synchronization to the national grid.

Figure 36. Protocol for a run up of a Turbine train to pass the first generator Critical.

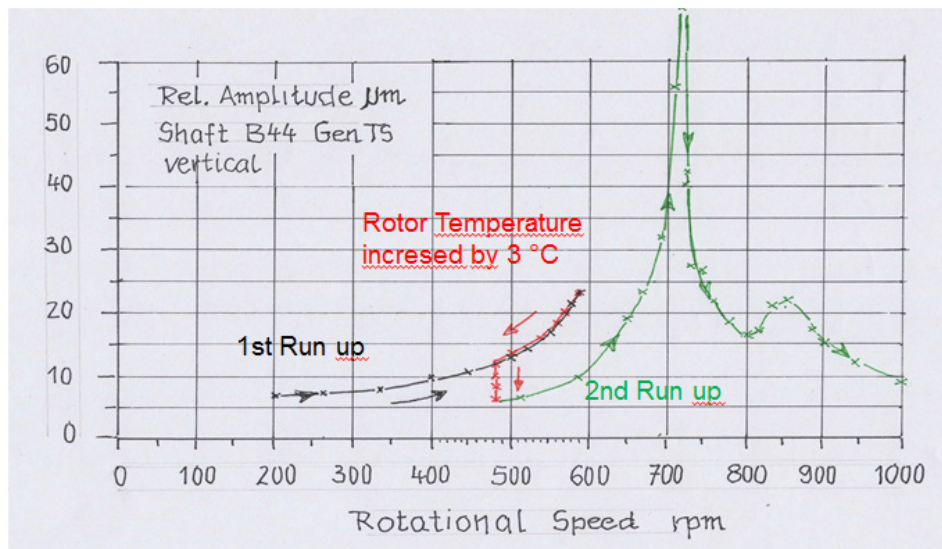


Figure 37. Run up of a shaft train with vibration control via the Winding temperature.

4 Information for the Matrix Development

4.1 INTRODUCTION

For the Matrix development the following input information is needed, which can be subdivided in the following categories:

Vibration Phenomena: CoV (Cause of Vibration)

Detection of Vibrations: DoV

Investigation of Vibrations: IoV

Analysis of Vibrations: AoV

Mitigations of Vibrations: MoV

In the following chapters 4.2 until 4.6 five lists with the mentioned topics are presented, which will be introduced later into the Matrices M1, M2, M3, M4.

4.2 LIST OF VIBRATION PHENOMENA (COV-CAUSE OF VIBRATION)

A detailed description of the Phenomena can be found in the reports:

Report A: Energiforsk Report Turbine and Generator Vibrations, Analysis and Mitigation. Part I Lateral Vibrations. 2016.

Report B: Energiforsk Report Turbine and Generator Vibrations, Matrix Development for the Identification and Mitigation of Vibration Problems. 2017 (this report).

- PHENOMENON 1:** **Slow Change of Unbalance**, e.g. due to a Thermal effects with a thermal bow or due to slowly increasing Mechanical Unbalance (Report A, 5.1).
CoV 1
- PHENOMENON 2:** **Sudden Change of a Mechanical Unbalance**, e.g. due to a Blade loss or a fast moving part in the rotating system (Report A, 5.1).
CoV 2
- PHENOMENON 3:** **Cyclic or Spiral Vibrations**. Superposition of an Original Mechanical Unbalance with a rotating Thermal Unbalance (Report A, 5.2).
CoV 3
- PHENOMENON 4:** **Change of Oil Film Bearing Coefficients**, e.g. due to oil film temperature, Static Bearing loads or due to clearance changes (Report A, 5.6).
CoV 4
- PHENOMENON 5:** **Friction induced mechanical bow in Turbogenerators**, depends on temperature, rotational speed and mounting pressure (Report A, 5.3).
CoV 5
- PHENOMENON 6:** **Change of Sea water temperature with Condenser deformation and change of Static Bearing location**. (Report A, 5.4).
CoV 6
- PHENOMENON 7:** **Change of Sea water temperature with a thermal bow of the LPT**. Due temperature and pressure change in condenser (Report A, 5.4).
CoV 7

- PHENOMENON 8:** **Oil Film Instability in Bearings of the turbine train, half frequency whirl.** Onset: ~ 2 times the first natural frequency (Report A, 5.6).
CoV 8
- PHENOMENON 9:** **Labyrinth Seal Instability in HP or LP Steam Turbines,** due to pressure forces in labyrinth. Instability at 1st natural frequency (Report A, 5.6).
CoV 9
- PHENOMENON 10:** **Instability in Steam turbines due to circumferentially unsymmetrical steam forces.** Clearance excitation (Report A, 5.6).
CoV 10
- PHENOMENON 11:** **Misalignment in Shaft train I.** Change of static bearing forces, when shafts are rigidly coupled: **Instability may occur.** (Report B, 2.4 & 6.2).
CoV 11
- PHENOMENON 12:** **Misalignment in Shaft train II.** Change of static bearing forces, when shafts are rigidly coupled: **Change 1xN Vibration** (Report B, 2.4 & 6.2).
CoV 12
- PHENOMENON 13:** **Misalignment in Shaft train III.** Change of static bearing forces, when shafts are rigidly coupled: **2xN Vibr., Banana Orbit** (Report B, 2.4 & 6.2).
CoV 13
- PHENOMENON 14:** **Generator Rotor has unequal moments of inertia.** This causes a 2xN Lateral Vibration due to weight of the Generator. (Report A, 5.5).
CoV 14
- PHENOMENON 15:** **Transverse Shaft crack in a Turbine train.** This causes changes in 1xN, 2xN vibrations and higher Harmonics (3xN), (Report B, 2.3, 6.1).
CoV 15
- PHENOMENON 16:** **Radial or angular Coupling errors lead to a Rotor bow,** when the coupling components are rigidly connected. (Report B, 2.5 & 6.3).
CoV 16
- PHENOMENON 17:** **Support Stiffness changes in turbine trains may change the Unbalance Response 1xN.** (Report B, 2.6 & 6.4).
CoV 17
- PHENOMENON 18:** **Rotor turbine train is running in a Resonance with high amplitudes.** Frequency is 1xN. (Report A, 5.1, Report B, 2.6 & 6.4).
CoV 18

4.3 LIST OF DETECTIONS OF VIBRATIONS (DOV)

- DETECTION 1:** **Slow change of the amplitude of the 1xN Lateral vibrations.** Can be caused by a change of Excitation or by a change of the system dynamic behavior.
DoV 1
- DETECTION 2:** **Slow change of the phase of the 1xN Lateral vibration.** Can be caused by a change of Excitation or by a change of the system dynamic behavior.
DoV 2
- DETECTION 3:** **Sudden change (increase) of the amplitude of the 1xN Lateral vibration.** Can be caused by a sudden change of Unbalance (e.g. by a Blade loss).
DoV 3

DETECTION 4: DoV 4	Sudden change of the phase of the 1xN Lateral vibration. Can be caused by a sudden change or by a change of Unbalance (e.g. by a Blade loss).
DETECTION 5: DoV 5	Change of amplitude and phase of the 1xN Lateral vibration. Vibration Vector in polar diagram rotates continuously (period 1 to 10 hours).
DETECTION 6: DoV 6	Change (increase) of the amplitude of the ½ xN Lateral vibration. Can be caused by a sub-synchronous instability in Oil Film Bearings or in Seals.
DETECTION 7: DoV 7	Change of the phase of the ½ xN Lateral vibration component. Can be caused by a sub-synchronous instability in Oil Film Bearings or in Seals.
DETECTION 8: DoV 8	Change (increase) of the amplitude of a Lateral Natural Frequency component at around ½ xN, e.g. caused by a Seal Instability.
DETECTION 9: DoV 9	Change of the phase of a Lateral Natural Frequency component at around ½ xN, e.g. caused by a Seal Instability.
DETECTION 10: DoV 10	Change (increase) of the amplitude of the 2xN Lateral vibration. Can be caused by Misalignment (Banana) or by unequal moments or by cracks.
DETECTION 11: DoV 11	Change of the phase of the 2xN Lateral vibration. Can be caused by Misalignment (Banana) or by unequal moments or by cracks.
DETECTION 12: DoV 12	Change (increase) of the amplitude of the 3xN Lateral vibration. This can be caused by a transverse shaft cracks (Nonlinearity of crack behavior).
DETECTION 13: DoV 13	Change of Oil Film temperature.
DETECTION 14: DoV 14	Change of cooling water in Generator Rotor Windings.
DETECTION 15: DoV 15	Slow Change of Vibrations on a group of transducers. Can be caused by a Change of Sea water temperature, due to summer/winter conditions.

4.4 LIST OF INVESTIGATIONS OF VIBRATIONS (IOV)

INVESTIGATION 1: IoV 1	Any remarkable events in the past two weeks, before the Vibration change was detected? Thunderstorms, Electrical Disturbances?
INVESTIGATION 2: IoV 2	What did people in the control room observe? Is information from different sensors compatible? Measurement Errors?
INVESTIGATION 3: IoV 3	Does Frequency in the Spectra mainly consist of the 1xN component? Vibration frequency is in this case the rotational frequency Ω .

- INVESTIGATION 4:** Do frequencies in the Spectra consist of the 1xN-component & a remarkable contribution of the 2xN-component?
loV 4
- INVESTIGATION 5:** Do frequencies in the Spectra consist of the 1xN-component & a remarkable contribution of the 1/2xN-component?
loV 5
- INVESTIGATION 6:** Do frequencies in the Spectra consist of the 1xN-component & a remarkable contribution of the 1st Natural frequency of a part rotor?
loV 6
- INVESTIGATION 7:** Do frequencies in the Spectra consist of the 1xN-component & contributions of the 2xN- and 3xN components & evtl. higher Harmonics.
loV 7
- INVESTIGATION 8:** Has the static location of the shaft in a Bearing changed to higher eccentricities? That means to higher Static Bearing forces?
loV 8
- INVESTIGATION 9:** Has the static location of the shaft in a Bearing changed to lower eccentricities? That means to lower Static Bearing forces?
loV 9
- INVESTIGATION 10:** Does the orbit at Bearing locations look like a closed elliptical orbit with mainly the 1xN rotational frequency component?
loV 10
- INVESTIGATION 11:** Does the orbit at Bearing locations look like a orbit with the two frequency components 1xN and 2xN? An example for this is Unbalance plus Misalignment.
loV 11
- INVESTIGATION 12:** Does the orbit at Bearing locations look like a orbit with the two frequency components 1xN and 1/2xN? Ex.:Oil film instability.
loV 12
- INVESTIGATION 13:** Does the orbit at Bearing locations look like a orbit with the main frequency 1xN and higher Harmonics? Example: Cracked Shaft.
loV 13
- INVESTIGATION 14:** Did the oil film temperature in the bearings increase? Increase means lower viscosity and higher Sommerfeld number. Higher eccentricity.
loV 14
- INVESTIGATION 15:** Did the oil film temperature in the bearings decrease? Decrease means higher viscosity and lower Sommerfeld number. Lower eccentricity.
loV 15
- INVESTIGATION 16:** Did only one Bearing show not normal Vibration behavior, Alarm? There is probably an anomaly at this Bearing only (clearance, bolts?).
loV 16
- INVESTIGATION 17:** Has the static location of the shaft in Seal elements change? Danger of rubbing and cyclic vibrations!
loV 17
- INVESTIGATION 18:** Did condenser pressure and temperature change, when the sea water temperature has changed?
loV 18

4.5 LIST OF ANALYSIS METHODS OF VIBRATIONS (AOV)

- ANALYSIS 1:
AoV 1** **FE Analysis for the Shaft Train: Critical Speeds, Stability, Mode shapes.** Critical speeds close to running speed? Relevant modes? Stability check?
- ANALYSIS 2:
AoV 2** **FE Analysis for the Shaft Train: Unbalance Amplitudes and Phase versus Speed.** Compare calculated & measured values! Vibration modes?
- ANALYSIS 3:
AoV 3** **FE Analysis for the Shaft Train: Simulation of Cyclic vibrations.** Model for heat input&output. Thermal bow Response. Influence of Parameters!
- ANALYSIS 4:
AoV 4** **Analyse Stiffness & Damping Coefficients for Bearings &Seals** How is their influence on the dynamic behavior (Response, Stability).
- ANALYSIS 5:
AoV 5** **Calculate Static Shaft Centerline of Shaft Train with several Bearings.** Loads are from weight and from possible additional Bearing forces.
- ANALYSIS 6:
AoV 6** **Analyse the Sensitivities of Parameters on Vibrations of the Shaft Train.** e.g. Oil temperature, Clearances, Sea water temperature.
- ANALYSIS 7:
AoV 7** **Analyse the run up and run down vibration behavior by measurements.** Compare with reference functions & calculations. Analyse differences.
- ANALYSIS 8:
AoV 8** **Analyse the influence of Power on the vibration behavior at operational Speed.** Change of frequency components? Influence on Seal stability.
- ANALYSIS 9:
AoV 9** **Analyse the run out of the shaft train at low speeds (Bow identification).** Other run out frequency components? Detection of Rotor cracks!
- ANALYSIS 10:
AoV 10** **Operational Modal Analysis during normal operation with full Power.** Identify the frequency components at operation. Stability behavior!

4.6 LIST OF MITIGATION OF VIBRATIONS (MOV)

MITIGATION 1: MoV 1	High 1xN-Lateral Vibrations due to Unbalance: Balancing of Flexible Rotor by means of Influence Coefficients. Avoid Critical Speeds close to Operating speed. In case of Blade Loss: Shut down the machine. After repair work: Balancing.
MITIGATION 2: MoV 2	High Lateral Vibrations due to Unbalance (1xN) or Instability: Improve Damping in Oil Film Bearings. Contact Manufacturer.
MITIGATION 3: MoV 3	Cyclic or Spiral Vibrations: Avoid rubbing in Seals, Bearings and Exciter Brushes. Adjust Rotor-Stator Distance. Reduce Friction Forces. Improve Damping. Control pressure in Oil Seal Systems. Contact Manufacturer for Design Changes in critical elements.
MITIGATION 4: MoV 4	Friction induced Mechanical Bow: Avoid un-symmetry in Circumference for friction, pressure and temperature. Control of winding temperature by Cooling water.
MITIGATION 5: MoV 5	Change of Sea water temperature with thermal bow: Protect rotor by means of a metal sheet against water. Reduce Active Power.
MITIGATION 6: MoV 6	Oil Film Instability in Bearings: Change Bearing Type or parameters (clearance, oil viscosity, width), Improve damping. (Manufacturer).
MITIGATION 7: MoV 7	Seal and Clearance Instability: Change of Seal parameters, e.g. Clearance, pressure difference, fluid flow. Improve damping. Reduce Power in case of HP instability. Contact Manufacturer.
MITIGATION 8: MoV 8	Misalignment in Shaft Trains: Change the vertical Bearing positions along the shaft line in order to reduce Misalignment. This will bring back the static Bearing forces to the design point.
MITIGATION 9: MoV 9	Unequal Moments of Inertia in Generator Rotors: Slots in 2 pole Generator rotors equalize the different moments of inertia. With this measure the weight resonance can be reduced.
MITIGATION 10: MoV 10	Transverse Shaft Cracks in Rotor Trains: After a Inspection of a Crack it has to be decided, whether the rotor can be used again. Crack is already very deep: No repair. Rotor destroyed! Crack is not to deep: Repair is possible. Rotor can be used again!
MITIGATION 11: MoV 11	Radial or Angular Coupling Errors: The bow in the rotor due to coupling errors leads to an Unbalance like 1xN Excitation. One Mitigation can be Counter-Balancing. In case of a angular coupling error a spacer can be inserted between the two coupling parts.
MITIGATION 12: MoV 12	Support Stiffness Changes in the Supporting Structure: Check all Clearances in the Bearing casing and the pre-stress of the bolts. Fix the clearances to recommended values and correct the pre stress in the foundation bolts.
MITIGATION 13: MoV 13	No Mitigation needed

5 Matrix and Flow Chart Development

5.1 INTRODUCTION – CONCEPT OF MATRIX DEVELOPMENT & FLOW CHART

If a Vibration problem in a Turbine train appears, the problem has first to be identified and then a suitable Mitigation has to be applied in order to solve the problem. One possible solution is to apply the following Matrix concept. Figure 38 points out the basic idea of the procedure with the matrices M1, M2, M3 and M4.

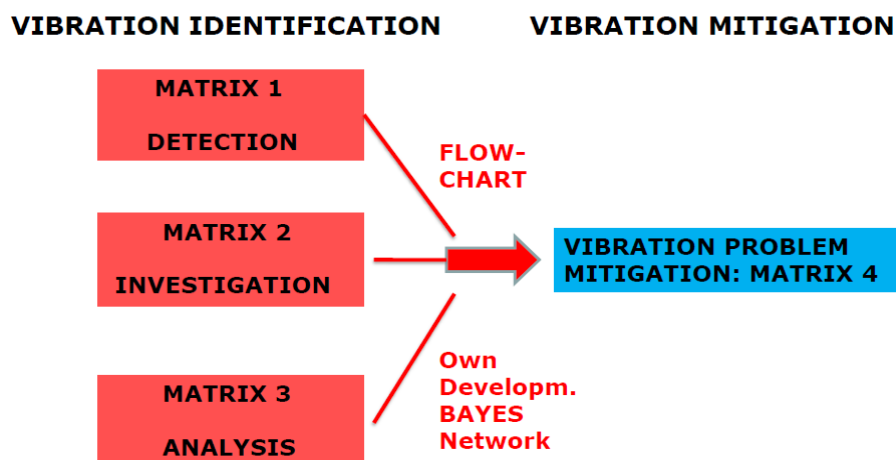


Figure 38. Matrix Concept for Vibration Problem Identification and Mitigation.

In Matrix M1 relations between the Vibration Phenomena (Chapter 4.2) and the Vibration Detections (Chapter 4.3) are presented. Accordingly M2 relates the Vibration Phenomena with Vibration Investigations (Chapter 4.4) and M3 relates the Vibration Phenomena with Vibration Analysis (Chapter 4.5). The Vibration Identification starts with the Detection and- if necessary - continues with the Investigation and Analysis. If one or even more Vibration Phenomena have been identified, the procedure continues with the Vibration Mitigation (Chapter 4.6). A Flow chart has to organize the flow of information between the Matrices. Such Flow charts may result from own developments or can be based on more sophisticated Probability methods, e.g. the Bayes Networks. A very simple Flow Chart is pointed out in Figure 39. From the Detection input in Matrix M1 possible Vibration Phenomena are identified. If the cause of Vibration (CoV) is already known from this Matrix M1, the output can be transferred to Mitigation. If this is not the case the procedure continues with Investigations in Matrix M2, where relations between the Phenomena and the Investigations may lead to an output to Mitigation with identified Vibration Phenomenon. If this is still not possible Analysis Method may finally help in a similar way to solve the problem.

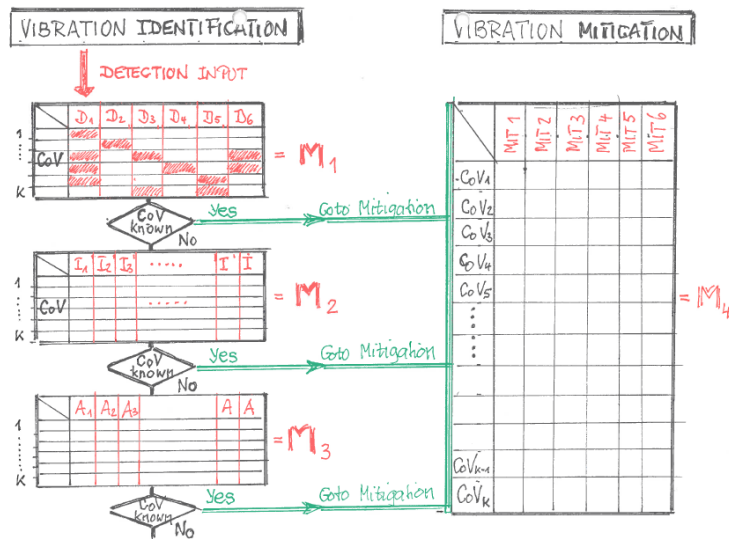


Figure 39. Flow Chart for Vibration Identification and Mitigation.

The following described Matrix Development and Flow Chart procedure works with Excel Tables. The procedure for Turbine and Generator Vibrations has been developed in accordance with a procedure for Power plant pipe vibrations, derived by Mikko Merikoski in another Energiforsk project. This was agreed with Paul Smeeke in order to have similar Matrix processes for different Vibration problems (Lateral vibrations, Torsional Vibrations, Pipe Vibrations) in Power Plants.

5.2 DEVELOPMENT OF MATRIX M1 – PHENOMENA VERSUS DETECTION

5.2.1 Structure of Matrix M1, Input Data

The structure of Matrix M1 is shown in Figure 40. In the rows of the matrix the 18 Vibration Phenomena (Chapter 4.2) are inserted. Detections (Vibration signals and temperatures) are presented in the 15 first columns of M1 (Chapter 4.3). Three further columns include the conditions of operation for the evaluation. And the two last columns consider the terms of commonness and severity of the problem. The numbers in the Matrix represent probabilities in the range between 0 to 5. A high probability 5 expresses, that there is a high relation between the considered Vibration Phenomenon and the Detection. In other words this detection is a very good proof for this Phenomenon. Number 1 means: not very probable and 0 means: not possible. If detections (changes from the original state) have been made during operation of the machine corresponding x have to be inserted in the yellow line of the Matrix and the calculation in the Excel table starts.

Vibration Phenomenon	Anomaly description	Sensor detection										Temperature	Condition		Commonness	Severity	Sum of probabilities, if no blanks	Relative probabilities	Commonness	Severity	Commonness & severity weighted probability				
		1	2	3	4	5	6	7	8	9	10		1	2											
Slow Change of Unbalance	Slow change of 1X Amplit.	5	5			2									2	3	3	3	3	4	2	0			
Sudden Change of Mechanical Unbalance	Slow change of 1X phase			5	5												3	1	1	2	5	0			
Cyclic Vibrations. Mech. & Therm. Unbalance	Sudden change of 1X ampplit.	3	3			5											3	2	2	2	2	0			
Change of Oil Film Bearing Coefficients	Sudden change of 1X phase	4	4			1	2	2	2	2			5	2	3	1	3	2	2	2	2	0			
Friction induced Mechanical Bow	Vector rotation in Polar diagram	2	2	3	3								1	5	1	3	3	2	4		0				
Change of Sea Water Temp. & Cond. Deform.	Change of 1/2X amplitude	3	3				2	2	2	2	1	1	2	5	3	1	1	3	1	0	0				
Change of Seawater Temp./ Thermal bow	Change of 1/2X phase	4	4					1	1				1	5	3	1	1	2	2	0	0				
Oil Film Instability in Bearings	Change of nat. freq. -Ampl.					5	5						3	1	2	2	3	2	5	0	0				
Labyrinth Seal Instability in HP or LP Turbines	Change of 1/2X phase					2	2	5	5			1			2	2	3	2	5	0	0				
Instability in Steam Turb. Clearance Excitation	Change of nat. freq. Phase					2	2	4	4						2	2	3	2	5	0	0				
Misalignment in Shaft trains I - Instability	Change of 2X amplitude					3	3						2		1	2	3	2	5	0	0				
Misalignment in Shaft trains II - Change 1X vibr.	Change of 3X amplitude	2	2					1	1			3			1	2	2	3	2	0	0				
Misalignment in Shaft trains III - Banana Orbit	Change of Oil Film Temper.	2	2					1	1	4	4	2	3		2	3	2	3	3	0	0				
Generator Rotor with Uneq. Moments of Inert.	Change of Gen. Water temp.								5	5	1				3	2	3	4	1	0	0				
Transverse Shaft cracks in Turbine trains	Change of Sea water temp.	3	3							4	4	3	2		3	1	1	2	5	0	0				
Radial or Angular Coupling Errors	During normal operation	2	2	1			1	1	1	1	1	2			2	3	2	2	3	0	0				
Support Stiffness changes in Turbine trains	During startup	4	4	2	2	1	1			1	1	2	1		3	1	1	3	4	0	0				
Turbine train is running in a Resonance condition	During power uprate tests	2	2	2	3									2	3	3	3	2	5	0	0				

Figure 40. Structure of Matrix M1.

5.2.2 Data Processing in M1 Excel Table and Output Data

For each row of M1 the probabilities are summed up for those columns with a yellow cross from detection. This leads to the sum of probabilities on the right hand side of the Matrix (see Figure 41). Then relative probabilities are determined and with commonness and severity a weighting of the probabilities is considered.

		Sensor detection											Temperature	Condition												
Vibration Phenomenon	Anomaly description	Slow change of 1X Amplit.	Slow change of 1X phase	Sudden change of 1X amplit.	Sudden change of 1X phase	Vector rotation in Polar diagram	Change of 1/2X amplitude	Change of 1/2X phase	Change of nat. freq. Ampl.	Change of nat. freq. Phase	Change of 2X amplitude	Change of 2X phase	Change of 3X amplitude	Change of Oil Film Temper.	Change of Gen. Water temp.	Change of Sea water temp.	During normal operation	During startup	During power uprate tests	Commonness	Severity	Sum of probabilities, if no blanks	Relative probabilities			
																							Commonness	Severity	Commonness & Severity weighted probability	
Slow Change of Unbalance		5	5			2								2	3	3	3	3	3	4	2	0				
Sudden Change of Mechanical Unbalance				5	5												3	1	1	2	5	0				
Cyclic Vibrations. Mech. & Therm. Unbalance		3	3			5											3	2	2	2	2	0				
Change of Oil Film Bearing Coefficients		4	4			1	2	2	2	2			5	2	3	3	1	3	3	2	2	9	20%	18%	11%	10%
Friction induced Mechanical Bow		2	2	3	3								1	5		1	3	3	2	4	0					
Change of Sea Water Temp. & Cond. Deform.		3	3			2	2	2	2	1	1		2	5	3	1	1	3	1	1	6	13%	18%	4%	5%	
Change of Seawater Temp./ Thermal bow		4	4					1	1				1	5	3	1	1	2	2	0						
Oil Film Instability in Bearings						5	5						3	1	2	2	3	2	5	13	29%	26%	38%	36%		
Labyrinth Seal Instability in HP or LP Turbines						2	2	5	5				1		2	2	3	2	5	5	11%	10%	15%	14%		
Instability in Steam Turb. Clearance Excitation						2	2	4	4							2	2	3	2	5	0					
Misalignment in Shaft trains I - Instability						3	3						2		1	2	3	2	5	8	18%	16%	24%	22%		
Misalignment in Shaft trains II - Change 1X vibr.		2	2					1	1				3		1	2	2	3	2	0						
Misalignment in Shaft trains III - Banana Orbit		2	2					1	1	4	4	2	3		2	3	2	3	3	0						
Generator Rotor with Uneq. Moments of Inert.									5	5	1				3	2	3	4	1	0						
Transverse Shaft cracks in Turbine trains		3	3							4	4	3	2		3	1	1	2	5	0						
Radial or Angular Coupling Errors		2	2	1				1	1	1	1	2			2	3	2	2	3	0						
Support Stiffness changes in Turbine trains		4	4	2	2	1	1			1	1	2	1		3	1	1	3	4	4	9%	12%	9%	13%		
Turbine train is running in a Resonance condition		2	2	2	3										2	3	3	3	2	5	0					

Figure 41. Output of Matrix M1 with input of Detection for Oil Film Instability.

An application is shown for the case of a 1/2 X frequency detection, which leads as a result to six possible Vibration Phenomena (green colour in rows). The corresponding relative probabilities are presented on the right hand side of M1. They clearly indicate for this case, that Oil Film Instability in Bearings is the most probable phenomenon with 36 %.

5.3 DEVELOPMENT OF MATRIX M2 – PHENOMENA VERSUS INVESTIGATION

5.3.1 Structure of Matrix M2, Input Data

The structure of Matrix M2 is shown in Figure 42. In the rows of the matrix again the 18 Vibration Phenomena are inserted. Investigations are presented in the 18 columns of M2 (Chapter 4.4). The numbers in the Matrix represent again probability in the range between 0 to 5 (0 means low probability and 5 high). The row easiness expresses the difficulty (0 to 5) or effort of an Investigation (5 = easy to realize). It helps when an Investigation has to be selected.

Investigation objective	Investigation objective																		Sum of probabilities in detection	Relative probabilities in detection	Commonness in detection	Severity in detection	Commonness & Severity weighted probability in detection
	ioV1 Remarkable Events before	ioV2 Observations from Control Room	ioV3 Have spectra mainly 1X	ioV4 Have Spectra 1X and 2X	ioV5 Have Spectra 1X and 1/2 X components	ioV6 Have Spectra 1X & nat. freq.?	ioV7 Have Spectra 1X & Higher Harmonic	ioV8 Static Brg. Loc.: higher ecc.	ioV9 Static Brg. Loc.: lower ecc.	ioV10 Bearing Orbit with 1X	ioV11 Bearing Orbit with 1X & 2X	ioV12 Bearing Orbit with 1X & 1/2X	ioV13 Bearing Orbit with 1X & Higher Harmonics	ioV14 Increase of oil temperature?	ioV15 Decrease of oil temperature?	ioV16 Only one Bearing not normal?	ioV17 Static shaft loc. of seals change?	ioV18 Did condenser pressure change?					
Easiness	4	4	4	4	4	4	4	4	4	3	3	3	3	4	4	4	4	4					
Objectives, sum	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0					
Objectives, %																							
Easiness weighted %																							
Vibration Phenomenon																							
Slow Change of Unbalance			4					2	2		4						2	2	0				
Sudden Change of Mechanical Unbalance			4								5						1	1	0				
Cyclic Vibrations. Mech. & Therm. Unbalance			5								5						3	1	0				
Change of Oil Film			2	2			3	3	2	2				4	4	1			0				
Friction induced Mechanical Bow			4							4									0				
Change of Sea Water Temp. & Cond. Deform.			3				3	3	3					2	2				0				
Change of Seawater Temp./ Thermal bow			3							3								5	0				
Oil Film Instability in Bearings					5		1	4				5				2			0				
Labyrinth Seal Instability in HP or LP Turbines						5		1							1	4			0				
Instability in Steam Turb. Clearance Excitation						5													0				
Misalignment in Shaft trains I - Instability					4	1	1	3				4		1	1				0				
Misalignment in Shaft trains II - Change 1X vibr.			4				1	2	2	4			1	1	1				0				
Misalignment in Shaft trains III - Banana Orbit			4				3	3			4		3	2					0				
Generator Rotor with Uneq. Moments of Inert.			4								4								0				
Transverse Shaft cracks in Turbine trains			4				4				4		4						0				
Radial or Angular Coupling Errors			4							4									0				
Support Stiffness changes in Turbine trains			1	1			3	2	2	1	1		3	1	1	5			0				
Turbine train is running in a Resonance condition			4	1		1	1	1	4	1	1		1	1	1	1			0				

Figure 42. Structure of Matrix M2.

5.3.2 Data Processing in M2 Excel Table and Output Data

In the Excel presentation Matrix M2 has two parts: Investigation objectives and Investigation findings. The Matrix Investigation Objectives receives the results from M1 (identified Vibration Phenomena) and calculates based on the M2-

probabilities and the easiness values the best suited Investigation methods for the identified Vibration problem. The result is shown in Figure 43 highlighted by the blue color. Frequency spectra, the static bearing location and orbits are recommended to confirm the phenomenon of Oil Film Instability.

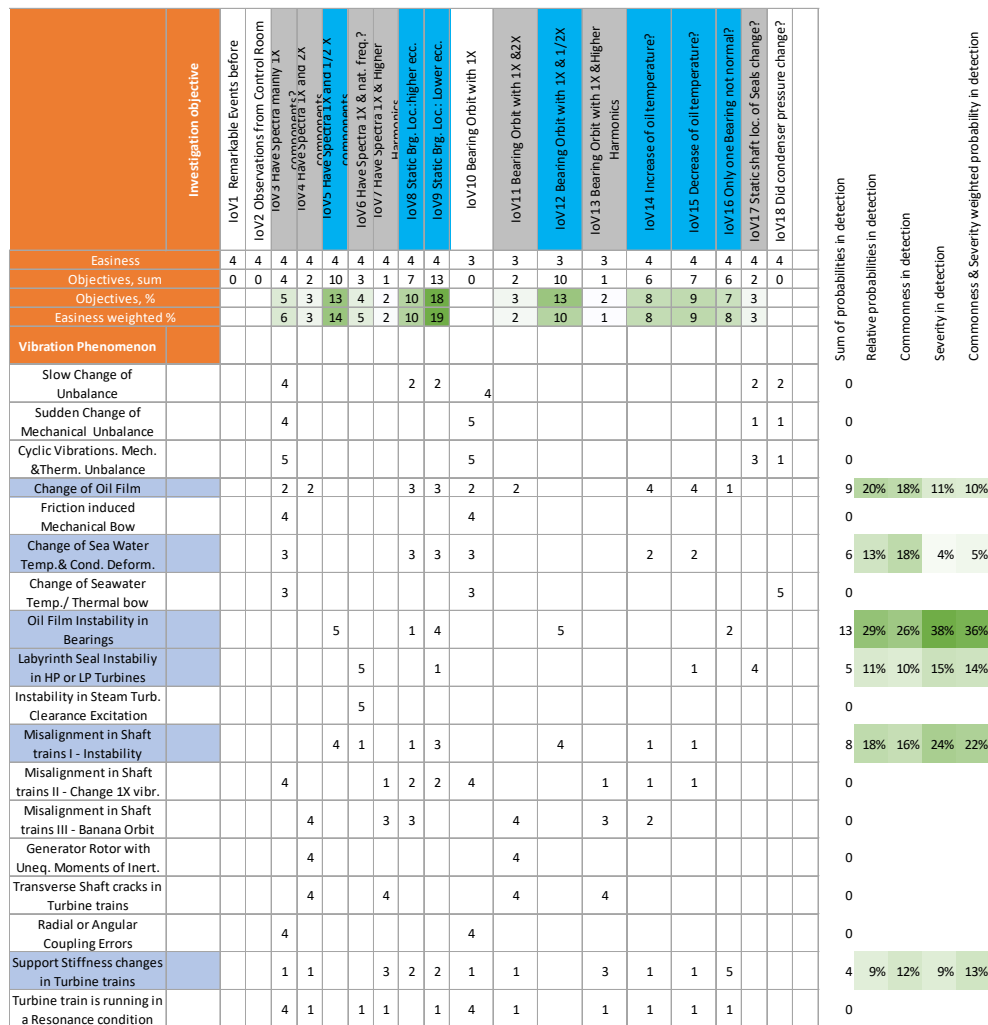


Figure 43. Recommended Investigations for the identified Vibration Phenomena.

In the second Matrix of M2 - Investigation findings - the user can select the investigation method by a x in the yellow line (see Figure 44). With the recommended Spectra and orbit Investigation the probability calculation with the probability numbers in the Matrix M2 leads to a relative probability of 56 % for the Oil Film Instability, followed by 44 % for a Misalignment in the shaft with unloaded Bearing and corresponding Instability.

		Investigation findings																					
		ioV1	ioV2	ioV3	ioV4	ioV5	ioV6	ioV7	ioV8	ioV9	ioV10	ioV11	ioV12	ioV13	ioV14	ioV15	ioV16	ioV17	ioV18	Sum of probabilities, if no blanks	Additional requirements	Relative probabilities	
		0																					
Objectives																							
Major objectives																							
Vibration Phenomenon		0																					
Slow Change of Unbalance				4					2	2		4							2	2	0	1	
Sudden Change of Mechanical Unbalance				4								5							1	1	0		
Cyclic Vibrations. Mech. &Therm. Unbalance				5								5							3	1	0		
Change of Oil Film				2	2				3	3		2	2			4	4	1			0		
Friction induced Mechanical Bow				4								4									0		
Change of Sea Water Temp. & Cond. Deform.				3					3	3		3				2	2				0		
Change of Seawater				3								3								5	0	1	
Oil Film Instability in Bearings						5			1	4			5					2			10		56%
Labyrinth Seal Instability in HP or LP Turbines							5			1							1		4		0		
Instability in Steam Turb. Clearance Excitation							5														0		
Misalignment in Shaft trains I - Instability					4	1		1	3				4		1	1					8		44%
Misalignment in Shaft trains II - Change 1X vibr.				4				1	2	2		4		1	1	1					0		
Misalignment in Shaft trains III - Banana Orbit				4				3	3			4		3	2						0		
Generator Rotor with Uneq. Moments of Inert.				4								4									0		
Transverse Shaft cracks in Turbine trains				4				4				4		4							0		
Radial or Angular Coupling Errors				4							4										0		
Support Stiffness changes in Turbine trains				1	1			3	2	2		1	1	3	1	1	1	5			0		
Turbine train is running in a Resonance condition				4	1		1	1	1	1	4	1		1	1	1	1				0		

Figure 44. Identification of the Vibration Phenomenon Oil Film Instability.

5.4 DEVELOPMENT OF MATRIX M3 – PHENOMENA VERSUS ANALYSIS

5.4.1 Structure of Matrix M3, Input Data

The structure of Matrix M3 is shown in Figure 45. In the rows of the matrix again the 18 Vibration Phenomena are inserted. Analysis Methods are presented in the 10 columns of M3 (Chapter 4.4). The numbers in the Matrix represent also probabilities in the range between 0 to 5 (0 means low probability and 5 high). And there is again a row easiness, expressing the difficulty (0 to 5) or the effort of an Analysis Method. This shall also help to select an optimal Analysis Method. M3 takes over the data from Investigation findings and uses this information for the output of M3.

5.4.2 Data Processing in M3 Excel Table and Output Data

For the Vibration Phenomena determined in M2 (Investigation findings), the probability numbers in Matrix M3 and the consideration of the easiness Analysis methods are recommended as output of Matrix M3. In this example the recommendation is to perform a Numerical Finite Element Analysis to calculate

the Stability behavior of the Shaft train. Due to the fact, that already the Investigation findings were relatively clear, it has to be decided, whether this calculation is necessary?

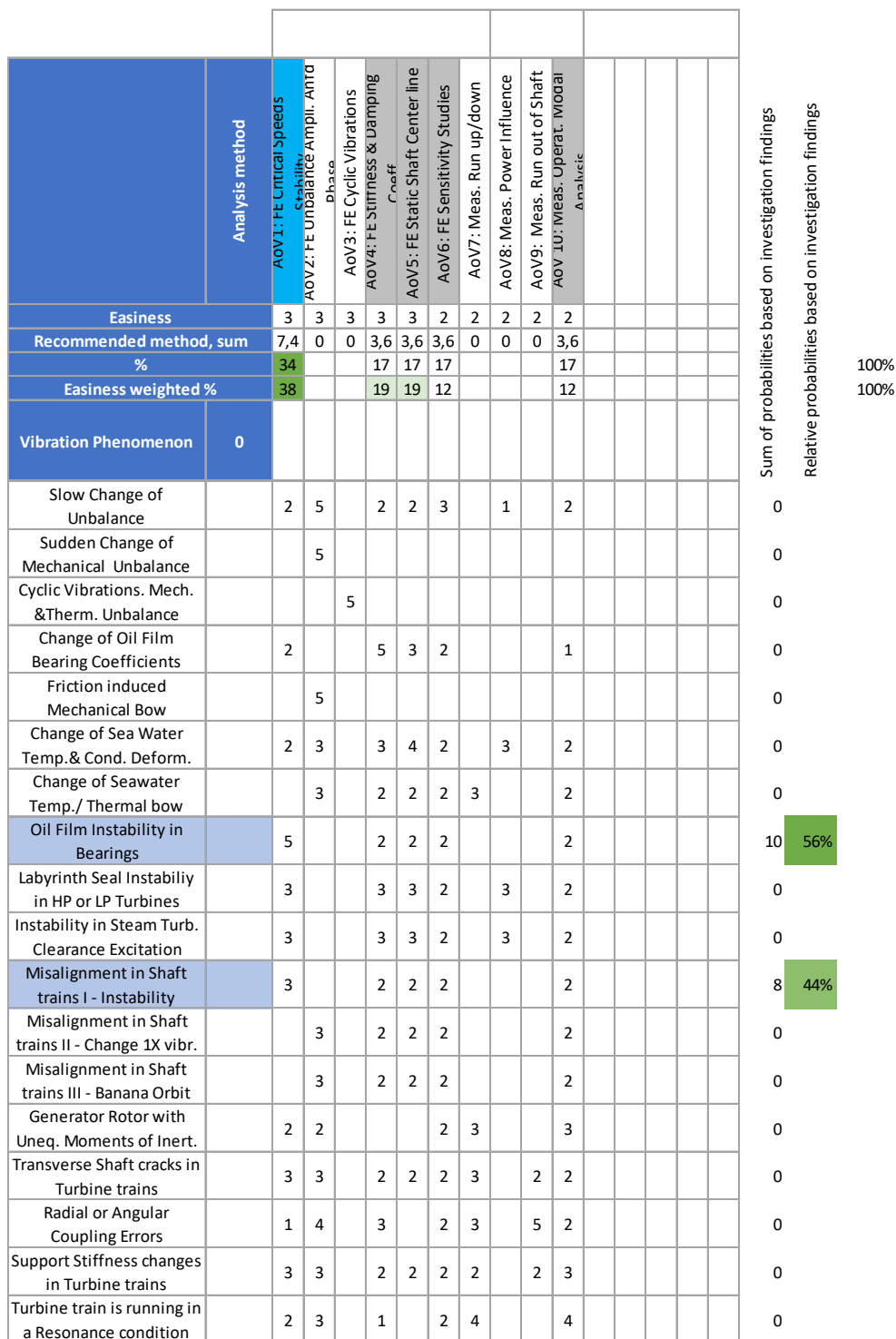


Figure 45. Structure of Matrix M3 with Input from Investigation Findings and Output.

5.5 DEVELOPMENT OF MATRIX M4 – PHENOMENA VERSUS MITIGATION

5.5.1 Structure of Matrix M4, Input Data

The structure of Matrix M4 is shown in Figure 46. In the rows of the matrix again the 18 Vibration Phenomena (Chapter 4.2) are inserted. Mitigation Methods are presented in the 13 columns of M4 (Chapter 4.5). The numbers in the Matrix represent probabilities in the range between 0 to 5, as before. The row easiness expresses the difficulty (0 to 5) or the effort of an Mitigation Method. This shall also help to select an optimal Mitigation Method. M4 takes over the data from Investigation findings (M2) or from Analysis (M3) and uses this information (x as input) for the output of M4.

5.5.2 Data Processing in M4 Excel Table and Output Data

In Matrix M4 the best suited Mitigation Method is calculated on the base of the Matrix probabilities, the identified Vibration Phenomena (marked by a cross) and the easiness for realization. In case of Oil film instability it is recommended, to improve the system damping and to change Bearing type or Bearing Parameters (see Figure 46).

5.6 FURTHER USE OF THE DEVELOPED MATRICES AND FLOW CHART

In the previous chapters the structure of the Matrices M1 to M4 and the data flow between the Matrices was explained. To demonstrate the process of Vibration Problem Identification and Mitigation a simple example of Oil film Instability was used. However, the Matrix tool has been tested for several other Lateral Vibration Problems with good results. After this introduction and application of the Matrix Concept the tool is available and can be used for cases of lateral turbine and generator vibration problems in Power Plants. The procedure should now be tested and improved based on the individual experience of the vibration and monitoring specialists in the Plants.

All Matrices can of course be extended to further Phenomena, Detections, Investigations, Analysis Methods and Mitigations. The success of the procedure is dependent on the probability numbers in the different Matrices. At this state the numbers are recommendations from the author of this report. When using the procedure in practice, each Power Plant should define their best probability numbers, based on experience.

The used matrix and flow chart procedure of this project works with linear probability procedures, organized in Excel tables. However an improvement with more sophisticated probability procedures based on Bayesian Networks seems to be possible and should be investigated in future studies in the Plants. First trials with Bayesian Networks are promising, results will be presented at the Energiforsk seminar in November.

		Mitigation method													Relative probabilities based on investigated																										
		MoV1: Balancing by Influence	MoV2: Improve Damping by Oil Film	MoV3: Avoid rubbing in seals, Exc. Brushes and Bearings	MoV4: Avoid Unsymmetry in Circumference for friction, pressure	MoV5: Protect Rotor against water	MoV6: Change Bearing Type or Bearing Parameters (Clearance, viscosity,..)	MoV7: Change Seal Parameters	MoV8: Change vertical Bearing positions to reduce Misalignment	MoV9: Slots in two pole generators	MoV10: If possible, repair the crack by	MoV11: Counter Balancing or insert of a	MoV12: Check clearances in Bearings	MoV13: No Mitigation needed																											
Recommended method, sum		3	3	2	4	2	2	2	2	4	2	2	2	5																											
%		0	5	0	0	0	5	0	2	0	0	0	3	0																											
Easiness weighted %			33				33		13			20																													
Vibration Phenomenon		0																																							
	Slow Change of Unbalance	5				3			2			2	2																												
	Sudden Change of Mechanical Unbalance	5	2		3	3						2																													
	Cyclic Vibrations. Mech. &Therm. Unbalance		3	5																																					
	Change of Oil Film Bearing Coefficients						5		4																																
	Friction induced Mechanical Bow	2	3		5							2																													
	Change of Sea Water Temp. & Cond. Deform.												5																												
	Change of Seawater Temp. / Thermal bow	3	3			5			2		3																														
	Oil Film Instability in Bearings		5				5		2			3																													
	Labyrinth Seal Instability in HP or LP Turbines		4	2				5																																	
	Instability in Steam Turb. Clearance Excitation		4	2			2	5																																	
	Misalignment in Shaft trains I - Instability								5																																
	Misalignment in Shaft trains II - Change 1X vibr.								5																																
	Misalignment in Shaft trains III - Banana Orbit								5																																
	Generator Rotor with Uneq. Moments of Inert.		2							5																															
	Transverse Shaft cracks in Turbine trains										5																														
	Radial or Angular Coupling Errors		3	3								5																													
	Support Stiffness changes in Turbine trains			2								5																													
	Turbine train is running in a Resonance condition		3	3			3																																		

Figure 46. Structure of Matrix M4, Input from identified Vibration Problem, Output is recommendation for Mitigation of the problem.

6 Appendix – Physical Description of new Lateral Vibration Problems

6.1 LATERAL VIBRATIONS IN SHAFT TRAINS DUE TO CRACKS

6.1.1 Introduction

Cracks in large turbine trains were first time detected in the years of around 1970. Due to this fact research studies were started to explain the Lateral Vibration Phenomena caused by cracks. In comparison to crack-free shafts the vibration signals of shafts with a crack have besides the 1xN- vibration also higher components, especially 2xN- and 3xN- components. Furthermore additional resonance effects occur, particularly the weight resonance, where the rotational frequency is $\frac{1}{2}$ of the first natural bending frequency. Shaft cracks were observed mainly in heavy horizontal rotors of shaft trains, e.g. in low pressure turbines and generator rotors, which have high 1xN alternating bending stresses due to the rotor weight. Crack detection is not an easy task, therefore it is very important to know the often small differences between a shaft with and without a crack. In order to show the influence of a crack on the lateral vibration behavior, we consider a simple rotor model together with a even simple crack model.

6.1.2 Physical Description of Lateral Vibrations in case of a Cracked Rotor

We consider the cross section of a shaft with a transverse crack with depth a , as presented in Figure 48. The diagram shows in a simplified form the two dimensionless bending flexibilities H_{22} and H_{33} (Angular displacement/Bending moment \times Youngs modulus \times Radius R^3) in the main and the cross direction versus the related crack depth a/R . For small crack depths ($a/R < 0.5$) the cross flexibility can be neglected. In this case we have only one factor describing the crack flexibility.

For the first moment it seems strange, that the flexibility H_{22} increases so strong in Fig. 4.8. However H_{22} is only a small part of the following overall bending flexibility of the considered Laval shaft. The complete flexibility consists of two parts, one from the shaft and one from the cracked zone.

We now consider the simplified rotor (Laval shaft) in Figure 49, which consists of a flexible shaft of length l and a mass m in the center of the rotor. Very close to the mass we assume, that the shaft has a crack (similar to Fig. 48). The crack is modelled by a hinge-joint, presenting a breathing crack when the shaft is rotating. If the crack is in the compressive zone the shaft flexibility corresponds with the one for the shaft without crack. If the crack is in the tension zone the flexibility is increased by the amount of Δh_{ζ} .

Bending flexibility in the compression zone: h_0 (flexibility of shaft without crack)

Bending flexibility in the tension zone: $h_0 + \Delta h_{\zeta}$

Translatoric Bending flexibility of crack: $\Delta h_{\zeta} = H_{22} \times l^2 / E \times R^3 \times 16$

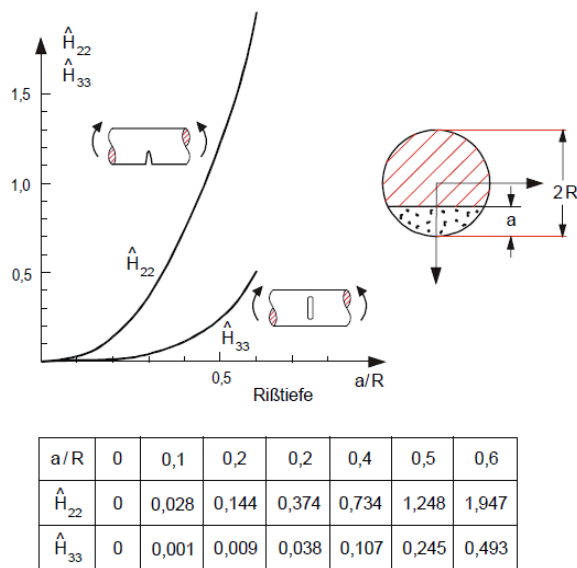


Figure 48. Dimensionless Main- and Cross Bending flexibilities of a splitted crack versus the dimensionless crack depth.

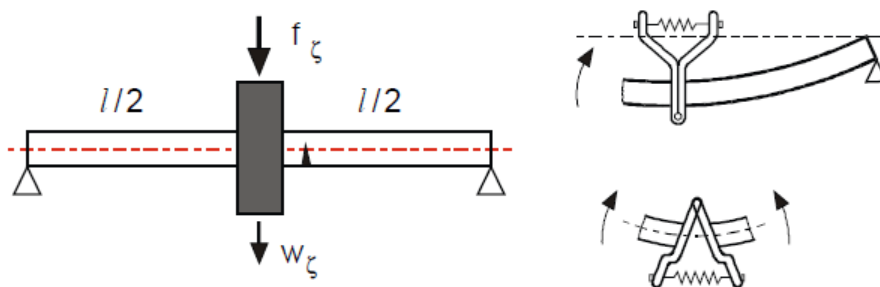


Figure 49. Laval shaft with a flexible shaft and a mass in the center. The crack close to the mass is presented as a hinge-joint (see picture on the right).

In a rotating coordinate system with the coordinates ζ (direction of transverse crack) and η (perpendicular to the direction of the crack) we can express the lateral displacements in dependence of the lateral forces by the following relation with the flexibilities h_0 and Δh_ζ , as defined before:

$$\begin{Bmatrix} w_\zeta \\ v_\eta \end{Bmatrix} = \left(\begin{bmatrix} h_0 & \\ & h_0 \end{bmatrix} + f(t) \cdot \begin{bmatrix} \Delta h_\zeta & \\ & 0 \end{bmatrix} \right) \cdot \begin{Bmatrix} f_\zeta \\ f_\eta \end{Bmatrix} \quad (6.1)$$

$f(t)$ is a control function for the case of the rotating shaft. When the crack is closed in the compression zone the control function is $f(t) = 0$, the shaft behaves like a “round” shaft (see also Fig. 49 with the hinge-joint). When the crack is completely opened in the tension zone the control function is $f(t) = 1$ and the flexibility in ζ -direction is increased by Δh_ζ . The crack mechanism has considered to be as a

nonlinear case, which will lead to higher Harmonics in the vibration response, which will be seen later.

Stiffness Matrix of the Laval Shaft with a Crack

For the derivation of the equations of motion the flexibility relation 6.1 is now transferred into a stiffness relation 6.2 (still in the rotating coordinate system) by means of an inversion.

$$\begin{Bmatrix} \mathbf{f}_\zeta \\ \mathbf{f}_\eta \end{Bmatrix} = \left(\begin{bmatrix} s_0 & \\ & s_0 \end{bmatrix} - \mathbf{f}(t) \cdot \begin{bmatrix} \Delta s_\zeta & \\ & 0 \end{bmatrix} \right) \begin{Bmatrix} \mathbf{w}_\zeta \\ \mathbf{v}_\eta \end{Bmatrix} \quad \boxed{6.2}$$

$$\mathbf{f}_{\text{rot}} = \left[\mathbf{S}_{0,\text{rot}} + \Delta \mathbf{S}_{\text{rot}}(\mathbf{u}_{\text{rot}}, t) \right] \mathbf{u}_{\text{rot}}$$

In these equations s_0 is the stiffness of the shaft without crack, and Δs_ζ represents the stiffness loss of the shaft due to the crack. $\mathbf{f}(t)$ has the same meaning as before. As mentioned, the Stiffness matrix 6.2 is still defined for the rotating system. However, the Vibrations in the plant (Detection) are measured with sensors in a fixed coordinate system. We therefore transform the stiffness matrix 6.2 from the rotating system into the fixed coordinate system. This is handled by means of a transformation matrix, which relates the displacements and forces between the two coordinate systems. The well known transformation matrix (not shown here) is a 2X2 matrix with harmonic sin- and cos-functions with rotational shaft frequency Ω .

Applying the transformation process, the following stiffness matrix for the fixed coordinate system will be obtained.

$$\mathbf{S}_0 + \Delta \mathbf{S}(t) = \begin{bmatrix} s_0 & \\ & s_0 \end{bmatrix} - \frac{1}{2} \mathbf{f}(t) \cdot \Delta s_\zeta \cdot \begin{bmatrix} 1 + \cos 2\Omega t & \sin 2\Omega t \\ \sin 2\Omega t & 1 - \cos 2\Omega t \end{bmatrix} \quad \boxed{6.3}$$

The stiffness matrix consists of two parts. The first part expresses the stiffness of the shaft without the crack (stiffness s_0). The second part is on the one hand time dependent due to the transformation matrix and on the other hand due to the time dependent control function $\mathbf{f}(t)$. As shown before $\mathbf{f}(t)$ becomes 0, when the crack is closed (crack in compression zone) and it becomes 1 for the open crack (crack in tension zone). Whether the crack is in the compression or in the tension zone depends on the deformation vector \mathbf{u} of the shaft, the stiffness matrix can therefore be expressed as shown in formula 6.4.

$$\mathbf{S} = \mathbf{S}(\mathbf{u}, t) = \mathbf{S}_0 + \Delta \mathbf{S}(\mathbf{u}, t). \quad \boxed{6.4}$$

Equations of Motion of the Laval Shaft with a Crack

In order to determine the vibration response of the simple Laval Shaft with a crack (Figure 50) we start with the equations of motion 6.5, considering the special type of Stiffness Matrix, presented in equations 6.2 to 6.4.

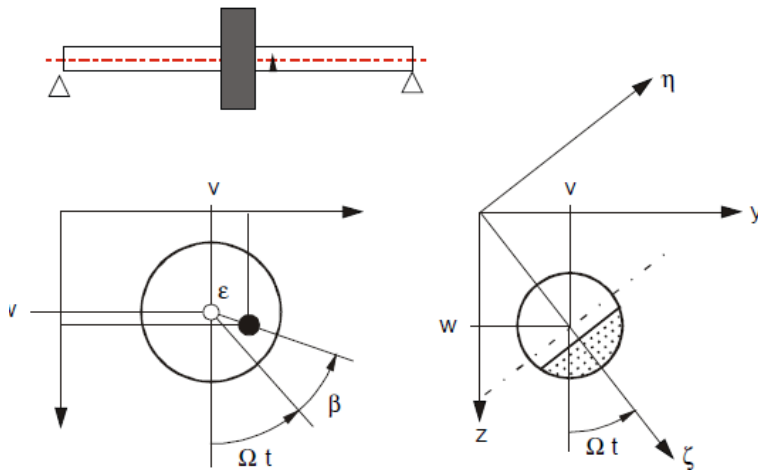


Figure 50. Laval Shaft with a Crack, Equations of motion.

$$\begin{bmatrix} \mathbf{m} & \\ & \mathbf{m} \end{bmatrix} \cdot \begin{Bmatrix} \ddot{\mathbf{w}} \\ \ddot{\mathbf{v}} \end{Bmatrix} + \begin{bmatrix} \mathbf{d} & \\ & \mathbf{d} \end{bmatrix} \cdot \begin{Bmatrix} \dot{\mathbf{w}} \\ \dot{\mathbf{v}} \end{Bmatrix} + \begin{bmatrix} s_{11} & s_{12} \\ s_{21} & s_{22} \end{bmatrix} \cdot \begin{Bmatrix} \mathbf{w} \\ \mathbf{v} \end{Bmatrix} = \begin{Bmatrix} \mathbf{mg} \\ 0 \end{Bmatrix} + \varepsilon m \Omega^2 \begin{Bmatrix} \cos(\beta + \Omega t) \\ \sin(\beta + \Omega t) \end{Bmatrix}$$

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{D}\dot{\mathbf{u}} + \mathbf{S}(\mathbf{u}, t)\mathbf{u} = \mathbf{p}_0 + \mathbf{p}_u$$

6.5

As mentioned earlier, the Stiffness Matrix for the cracked shaft is time dependent and non-linear. On the right hand side we recognize the Unbalance excitation \mathbf{p}_u and the weight as a static force \mathbf{p}_0 . The last one has a strong influence on the dynamic behavior. In case of the crack sensitive heavy Low Pressure Turbine and Generator rotors the static displacement due to the weight \mathbf{u}_0 is usually much larger than the vibration displacements $\Delta\mathbf{u}(t)$. This means, that opening and closing of the gap is mainly determined by the static displacement \mathbf{u}_0 . Figure 51 shows the weight dependent opening and closing of the gap versus the angular displacement.

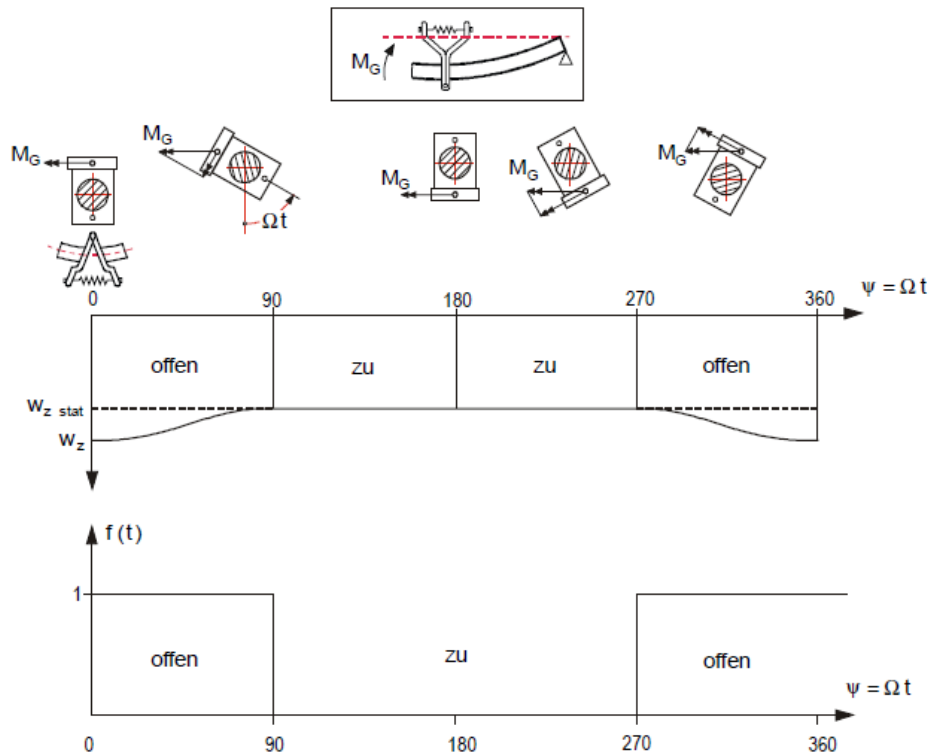


Figure 51. Opening and closing of the gap versus the angular displacement.

Considering the fact discussed before and some further matrix arrangements lead finally to the equations of motion 6.6 for the vibration response of the Laval Shaft with a transverse crack. They describe the vibration terms $\Delta \mathbf{u}(t)$, caused by the two time dependent excitation forces on the right hand side.

$$\begin{bmatrix} \mathbf{m} & \\ & \mathbf{m} \end{bmatrix} \cdot \begin{Bmatrix} \Delta \ddot{\mathbf{w}} \\ \Delta \ddot{\mathbf{v}} \end{Bmatrix} + \begin{bmatrix} \mathbf{d} & \\ & \mathbf{d} \end{bmatrix} \cdot \begin{Bmatrix} \Delta \dot{\mathbf{w}} \\ \Delta \dot{\mathbf{v}} \end{Bmatrix} + \begin{bmatrix} \mathbf{s}_0 & \\ & \mathbf{s}_0 \end{bmatrix} \cdot \begin{Bmatrix} \Delta \mathbf{w} \\ \Delta \mathbf{v} \end{Bmatrix} \\ = \frac{1}{2} \Delta s_{\zeta} f(t) \begin{bmatrix} 1 + \cos 2\Omega t & \sin 2\Omega t \\ \sin 2\Omega t & 1 - \cos 2\Omega t \end{bmatrix} \cdot \begin{Bmatrix} w_{\text{stat}} \\ 0 \end{Bmatrix} + \varepsilon m \Omega^2 \begin{Bmatrix} \cos(\Omega t + \beta) \\ \sin(\Omega t + \beta) \end{Bmatrix}$$

$$\mathbf{M} \Delta \ddot{\mathbf{u}} + \mathbf{D} \Delta \dot{\mathbf{u}} + \mathbf{S}_0 \Delta \mathbf{u} = -\Delta \mathbf{S}(t) \mathbf{u}_0 + \mathbf{p}_u, \quad \boxed{6.6}$$

The first term is the excitation due to the crack. This excitation depends on the stiffness loss due to the crack Δs_{ζ} , the static displacement due to the weight w_{stat} , on the harmonic time functions with frequency 2Ω ($2xN$ excitation) and on the non-linear control function $f(t)$ for the hinge joint (gap opening and closing). If we have a closer look on the forward and backward Harmonics of the first crack dependent excitation we find the following participation factors:

	Participation		Participation
Forward excitation: + 1 Ω	+ 0,318	Backward excitation: - 1 Ω	0,106
	+ 2 Ω		- 2 Ω 0,000
	+ 3 Ω		- 3 Ω -0,021

The second excitation is the known Unbalance excitation with frequency Ω (1xN excitation).

Vibration Response of the Laval Shaft with a Crack

The vibration response of the Laval shaft with a crack can now easily be determined by solving the equations 6.6. For the first type of excitation (the crack dependent excitation without the unbalance forces \mathbf{p}_u), we obtain the following results, which are presented in Figures 52 to 55.

Figure 52 shows the dimensionless amplitudes of the three forward vibrations with frequencies of 1 Ω , 2 Ω and 3 Ω and the backward vibration with frequency -1 Ω versus the dimensionless rotational frequency Ω/ω_0 . Damping was assumed as $D = 0.05$. The diagram shows the resonances at rotational frequencies $\Omega = \omega_0, \omega_0/2$ and $\omega_0/3$ (ω_0 is the natural frequency of the rotor without the crack).

In Figure 53 the same vibration response is presented for the 1 Ω , 2 Ω and 3 Ω forward vibrations in a different way, showing now the circular and elliptical orbits along the rotational frequency Ω . In reality the three separated Frequency response functions in Figure 52 and also the three separated orbit presentations in Figure 53 are superimposed. Showing this superposition leads to the orbits for the different dimensionless rotational frequencies shown in Figure 54. The information in all of these presentations for the vibration behavior can be used for the detection and identification of a rotor crack.

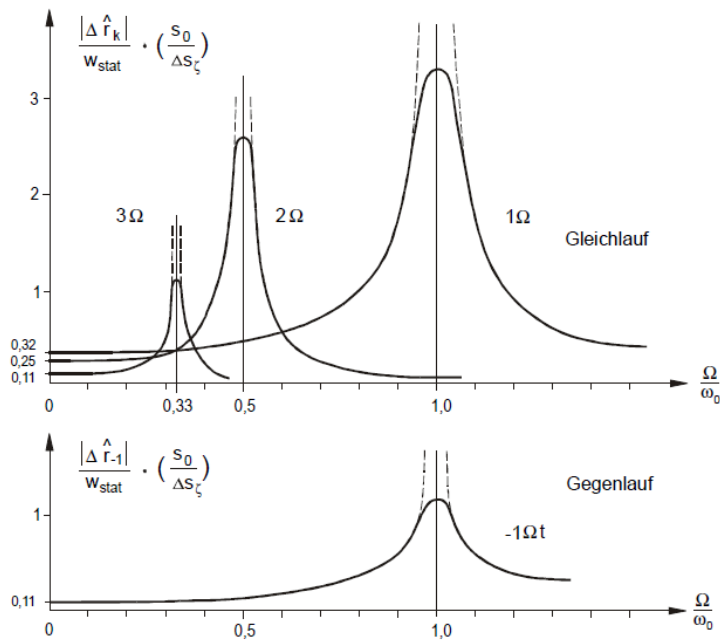


Figure 52. Dimensionless +/- 1Ω , $+2\Omega$ and $+3\Omega$ Vibration Amplitudes of a Laval Shaft with a Crack versus the dimensionless Rotational Frequency.

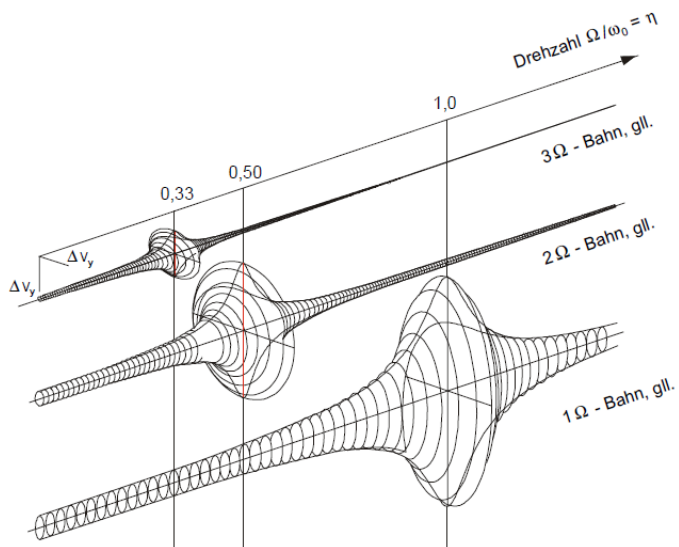


Figure 52. Elliptical forward Orbits of 1Ω , 2Ω and 3Ω Vibrations of a Laval Shaft with a crack versus the dimensionless rotational frequency Ω/ω_0 .

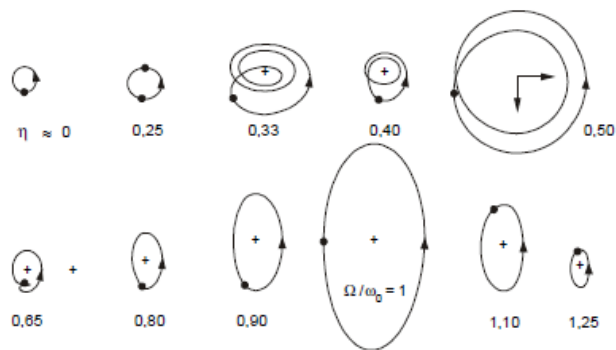


Figure 53. Vibration Orbits at shaft center for a Laval Shaft with a crack for different dimensionless rotational frequencies Ω/ω_0 . Crack dependent excitation only.

The double sided frequency spectrum with forward and backward frequencies, as shown in Figure 54, can also be a very helpful tool for the crack identification. We recognize the different forward and backward frequencies in dependence of the rotational frequency Ω .

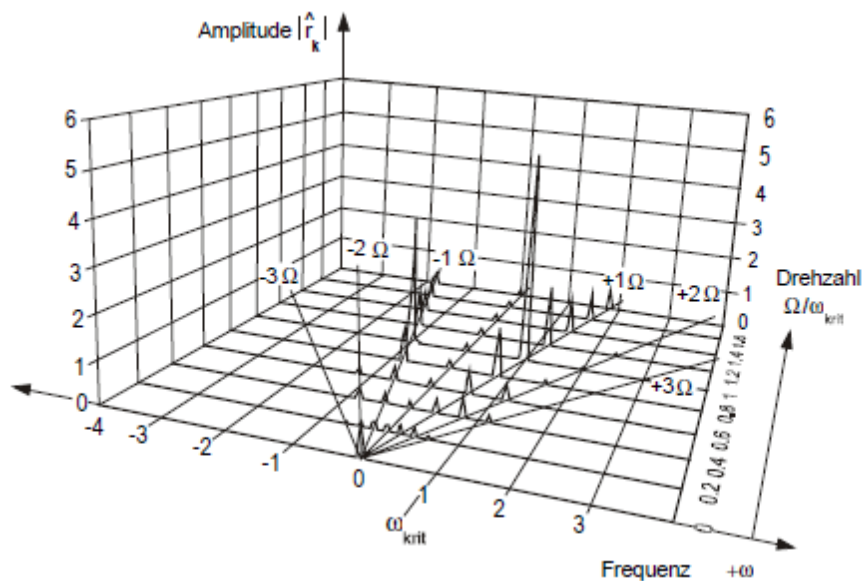


Figure 54. Double sided Frequency Spectrum for a Laval Shaft with a crack for different dimensionless rotational frequencies Ω/ω_0 .

Up to now we have only considered the response of one part of the excitation, the crack dependent excitation. This response has to be superimposed by the response of the second excitation, the Unbalance excitation. The Unbalance response consists only of 1Ω components. Depending on the phase angle of Unbalance-force and Crack-force the superimposed 1Ω response can be amplified or reduced compared to the crack response only. The other frequencies in the response 2Ω , 3Ω are not influenced by the Unbalance.

6.1.3 Examples of Methods to evaluate Rotor Cracks from Vibration Measurements

To monitor and to evaluate cracks in rotors the following methods have been used in Swedish Power Plants and UK Power Plants with good results. They are based on vibration measurements of horizontal and vertical absolute bearing vibrations and relative shaft vibrations (vector measurements), where the 1xN and 2xN amplitudes and phases are monitored and evaluated. In some cases higher Harmonics were also monitored and evaluated.

Turbine-Generator Vibration Monitoring during Operation: Running on Load

This method is useful, when the turbogenerators do not run up and down very often, for example in Nuclear Power Plants. A sign for a developing crack are changes on both 1xN and 2xN vibration components. The vectors normally change in one direction.

A typical example is described in an article “**On the vibration behavior of a cracked rotor**” by Leif R.K. Nilsson D.Eng., STAL-LAVAL Turbin AB, Finspong, Sweden.

1981 the manufacturer STAL-LAVAL measured and evaluated the vector change during operation, when the crack developed and finally stopped the machine in time. The bearing amplitude change with time was not too large, but the phase angle changed. The vector change were only around 1-2 mm/sec rms. Shaft vector change on the Generator was around 20-25 microns peak. At the final run down the Monitoring of the above Generator showed increased levels of vibrations and the Critical Speed also decreased due to the rotor crack.

Turbine-Generator Vibration Monitoring during Run down/Run Up

In England machines, that ran up and down often were investigated around 1980. Vibrations and phase reference were recorded during run up and run down and 1xN and 2xN vectors were evaluated from the run up and run down measurements.

If vector changes in 1xN and 2xN were detected – an evaluation of 3xN and 4xN components was also performed, in order to verify that there was a propagating crack in the rotor.

For the 3xN- component the vector change on 1/3 of the first critical was studied.

In the same way the vector change on 1/4 of the first critical for the 4xN component.

The levels on the 3xN and 4xN peaks were not so high – but when the crack was propagating one could easily see the change of speed (frequency) for those peaks.

One example was in an AGR nuclear power station in England which had a crack in the Generator rotor which was discovered by these evaluations of 1xN, 2xN, 3xN and 4xN vectors from run up and run down. The machine was stopped in time.

6.2 LATERAL VIBRATIONS IN SHAFT TRAINS DUE TO MISALIGNMENT

6.2.1 Introduction

Misalignment occurs, when the bearings of a shaft train are not in the right vertical position for a moment free coupling of the rigid flanges of two turbine rotors or a turbine rotor and the generator. Figure 55 shows, which types of Misalignment may occur. The two rotors are in perfect alignment, when their shaft center lines are collinear at the coupling (top in Fig. 55). Parallel Misalignment (middle in Fig. 55) is the case, when the shaft centerlines are offset, but parallel. Angular Misalignment (bottom of Fig. 55) occurs, when the shafts are not parallel and have a relative angular displacement between the shafts. The most common situation in practice is a combination of parallel and angular Misalignment. The following example demonstrates the case of angular Misalignment.

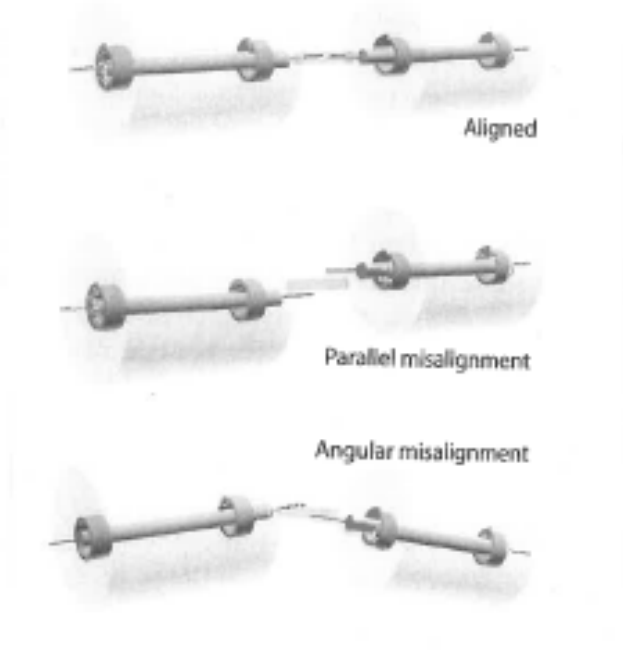


Figure 55. Types of Misalignment for two rotors with a coupling.

6.2.2 Physical Description of Lateral Vibrations in case of Angular Misalignment

Figure 56 demonstrates the case of angular Misalignment for two part rotors and four bearings. In this example the vertical location of Bearing No. 4 (belongs to Rotor 2) is too high in the uncoupled case (see upper figure in 56) and due to this a relative angular displacement between the two coupling flanges occurs (the coupling itself shall be error free).

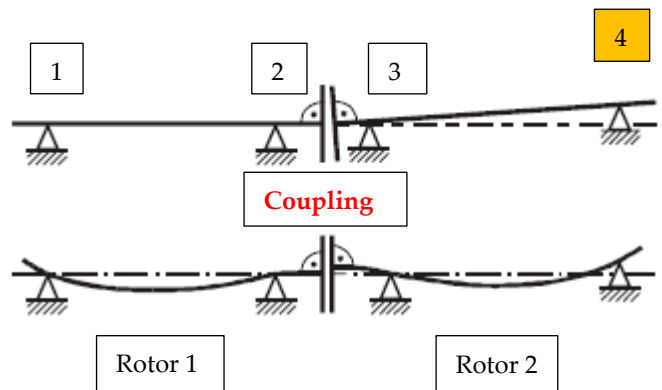


Figure 56. Misalignment in case of two part rotors and four bearings.

When the two shafts are now rigidly connected together, a static moment in the coupling is needed and this also leads to a static bending line in the two rotor shafts and will also change the static Bearing forces (see lower figure in 56).

The Bearing forces are on the one side dependent on the weights of the rotors - same as before coupling the rotors - and in addition on the compulsive forces (moments) in the coupling. It is important to note, that the bending line is a static one, which is fixed in space. When vibrations appear, they will be relative around this static bending line. The simple example has shown that in a first consideration Misalignment is a static phenomenon and cannot directly be considered as a vibration problem. However, as we have seen in the simple example, Misalignment can lead to a change of the static bearing forces and the static displacements and this will finally also lead to a change in the dynamic characteristics of the rotor system. For example, due to the change of the static forces in the bearings, the static journal displacements and the dynamic coefficients of the oil film will change as well. Figure 57 points out the static shaft displacement curve in dependence of the Sommerfeld number. If the static bearing force F_{stat} increases (decreases) the Sommerfeld number changes in the same direction. Correspondingly the shaft moves to higher or lower eccentricities within the bearing clearance. With this move the bearing is either higher loaded or lower loaded and the dynamic bearing coefficients (stiffness and damping) change as well and influence the vibration behavior of the shaft train.

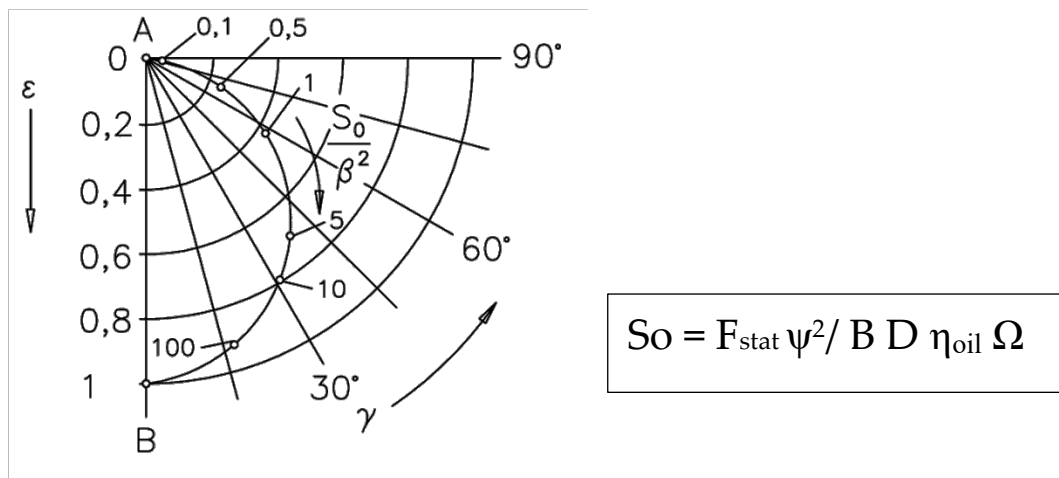


Figure 57. Static shaft displacement curve within a circular bearing in dependence of Sommerfeld number S_0 .

In steam turbine trains of Power Plants there are usually several oil film bearings arranged. In case of Misalignment, the rotor shaft train has to be considered as a statically undetermined system. Compared to the known weight dependent bearing forces in a well aligned shaft train, the bearing forces may decrease or increase due to Misalignment, as mentioned before. This again means that the bearings can be higher or lower loaded, compared to the aligned situation, which corresponds to the nominal design. And depending on the loading condition the vibration behavior will be quite different.

Figure 58 compares the shaft positions in the bearings for the cases of an aligned and a misaligned shaft line. Misalignment shifts the load among the bearings. In this example Bearing No. 2 experiences a very high load and the shaft has a high eccentricity, usually also with high oil film temperatures. Contrary Bearing No. 3 is essentially unloaded and is sensitive for oil film instability. Figure 59 shows the orbit and the frequency spectrum for such an under loaded case due to Misalignment. We can recognize the half frequency whirl due to instability besides the rotational frequency due to Unbalance.

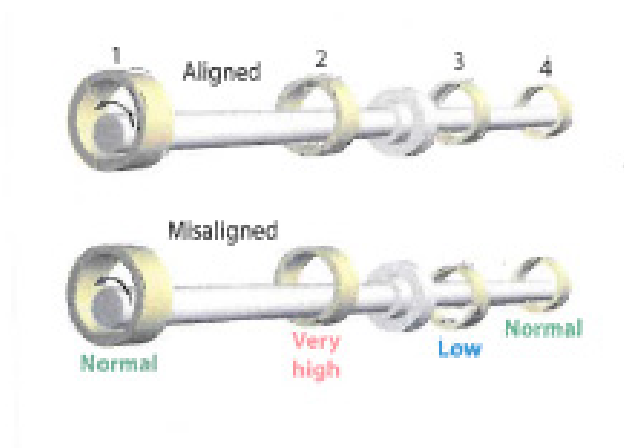


Figure 58. Average shaft position in an aligned machine versus a misaligned machine.

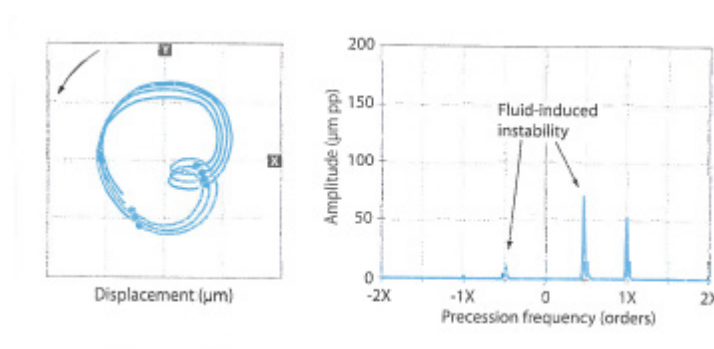


Figure 59. Orbit and Spectrum for the case of Oil Film -induced Bearing instability due to Misalignment.

If Misalignment leads to a higher loaded bearing with higher eccentricities the unbalance excitation with 1Ω may lead to a response with 1Ω and 2Ω frequency components due to the nonlinear oil film behavior at higher eccentricities. The elliptical orbit, known from the linear theory will now be deformed to a so called Banana Orbit, when the shaft comes closer to the bearing sleeve.

The 2Ω frequency component is very often a symptom for Misalignment with higher bearing loading. The relation of the 2Ω to the 1Ω amplitudes can be a measure of the severity of the Misalignment (Figure 61).

In the Rotor Dynamic literature two explanations can be found, why Misalignment may lead to a $2xN$ frequency component. The first one was just described above: High Bearing loads may lead to a nonlinearity in the bearing with a $2xN$ response component, excited by $1xN$ Unbalance. However, in one research project it was stated, that the nonlinear oil film produces only small $2xN$ components.

Another explanation for the $2xN$ component is based on the changing Bending flexibility of the shaft with double rotational frequency ($2xN$). This changing bending flexibility may be caused due to Misalignment in the coupling parts of shaft trains with several bearings.

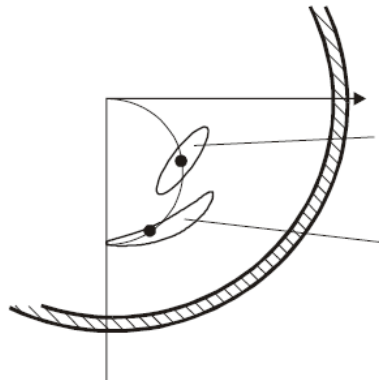


Figure 60. Deformation of an elliptical orbit in the Bearing to a Banana Orbit.

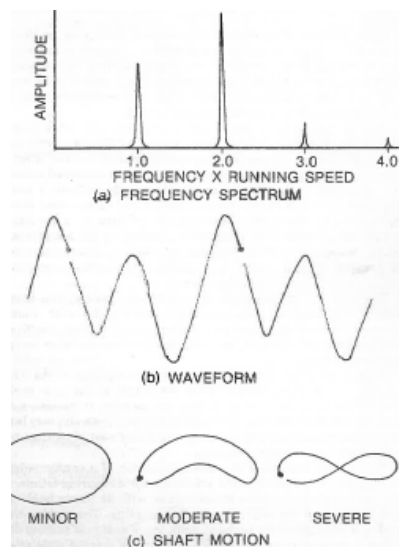


Figure 61. Characteristics of Misalignment with 1Ω and 2Ω Vibration frequencies.

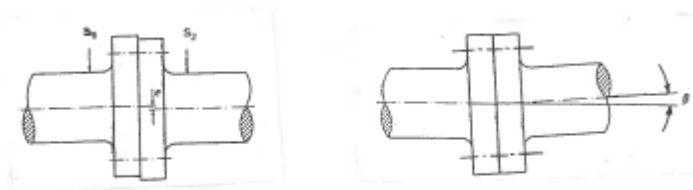
6.2.3 An Example for Vibrations in a Shaft Train due to Misalignment

Ylva Vidhög has reported in the CISM Conference Proceedings “Diagnostics of Rotating Machines in Power Plants” (1993, page 254 – 257) about a change in the couplings between two LPT rotors with observed vibration vector changes of the $1xN$ and $2xN$ frequency components. Such vibration changes may occur, when one rotor has been lifted during an overhaul and the couplings had been opened. However, a coupling change can also occur, when there is a disturbance in the Electrical Power net and higher voltages as usual are present. The resulting Vibration vector changes have been observed rather common in the Nuclear Power Plants in Sweden due to Electrical Work on the Power lines or due to lightning.

6.3 LATERAL VIBRATIONS IN SHAFT TRAINS DUE TO COUPLING ERRORS

6.3.1 Introduction

Due to manufacturing defects in couplings radial eccentricity errors as well as angular errors (axial eccentricity) may occur in rigid couplings (Figure 62). Due to these errors $1 \times N$ - frequency dependent vibrations can be excited.



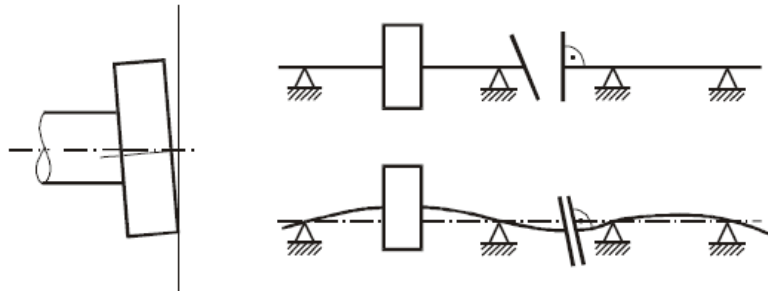
Radial eccentricity

Angular error (axial eccentricity)

Figure 62. Radial eccentricity and angular error (axial eccentricity) in rigid Couplings.

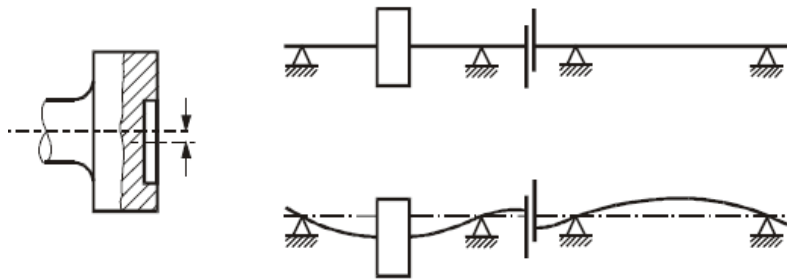
6.3.2 Physical Description of Lateral Vibrations due to Coupling Errors

Figures 63 and 64 show once again the two types of coupling errors: Angular Coupling error and Radial eccentricity error. For each of these error types the two shafts are presented in the uncoupled state as well as in the coupled state. When the two shafts are bolted rigidly together, a bend or bow will be produced in the shaft system. Contrary to misalignment, where the error occurs in the nonrotating (static) Bearing system (wrong vertical Bearing location), coupling errors occur in the rotating shaft system. As a consequence the produced bend is stationary for the shafts or in other words the bend or bow rotates with the shafts. Like an unbalance the bow causes therefore lateral vibrations and also rotating transverse forces in the bearings with rotational frequency Ω . Shaft vibrations and dynamic bearing forces depend on the size of the bow and on the rotational frequency. In order to show the effect of a bend due to coupling errors, we consider again the simple Laval shaft as before. For more complex systems like real steam turbine trains in a Power Plant, Finite Element (FE) routines can be used in order to study the influence of coupling errors. Such FE Analysis can be performed by manufacturers of turbine trains.



Angular Coupling error Shafts in decoupled (top) and coupled state (bottom)

Figure 63. Shaft system with a rotating bend (bow) due to an angular coupling error.



Radial Eccentricity error Shafts in decoupled (top) and coupled state (bottom)

Figure 64. Shaft system with a rotating bend (bow) due to a radial eccentricity.

Equations of motion of a Laval Shaft with a Bow

In this study for the simple Laval shaft (see Figure 65) we take into account the bow “a” at the location of the disk besides the unbalance $m\varepsilon$, with the mass of the disk m and the mass eccentricity ε . A bow may have different causes (plastic deformation, thermal effects, etc.). Here we assume, that the bow is caused by a coupling error.

If the shaft rotates with very low speeds ($\Omega \ll 1$, natural shaft bending frequency), the orbit of the bow “a” can be observed (see Figure 65, left side), which can be described by the two displacement components.

$$w_0 = a \cos \Omega t$$

$$v_0 = a \sin \Omega t$$

6.7

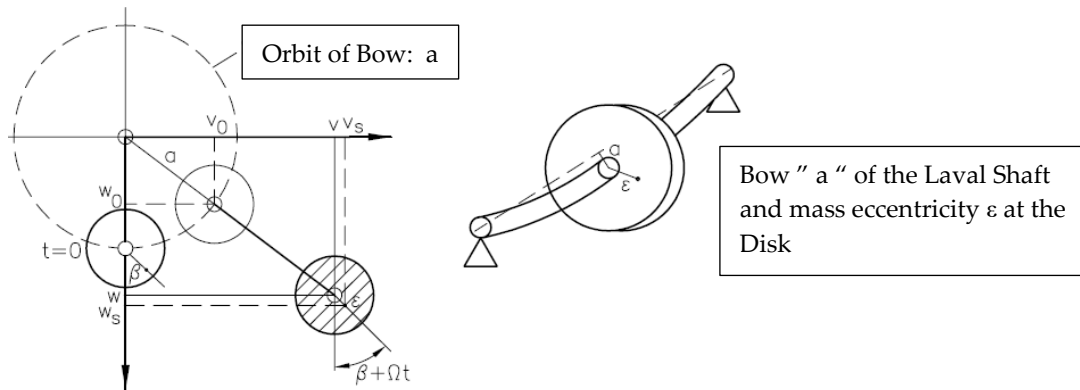


Figure 65. Laval Shaft with a bow " a " and the unbalance me. ε is the mass eccentricity.

For arbitrary rotational speeds the vibrations w (vertical) and v (horizontal) can be described by the equations of motion, considering the bow as well as the unbalance as excitation on the right hand side (see equations 6.8).

$$m\ddot{w} + sw = m\epsilon\Omega^2 \cos(\Omega t + \beta) + sa \cos \Omega t$$

$$m\ddot{v} + sv = m\epsilon\Omega^2 \sin(\Omega t + \beta) + sa \sin \Omega t \quad .$$

6.8

Unbalance Excitation Bow Excitation

The excitation force due to the bow (stiffness s multiplied with the bow a) is not directly dependent on the rotational frequency Ω as in case of the Unbalance. The two linear equations can be solved independently for the two excitation cases and can then be superimposed to the complete solution.

Vibration Response of a Laval Shaft with a bow excitation

The vibration response for the two coordinates $w(t)$ and $v(t)$ due to the bow excitation alone can be expressed in the following form

$$w_v(t) = a \frac{\omega^2}{\omega^2 - \Omega^2} \cos \Omega t = a \frac{1}{1 - \eta^2} \cos \Omega t$$

$$v_v(t) = a \frac{\omega^2}{\omega^2 - \Omega^2} \sin \Omega t = a \frac{1}{1 - \eta^2} \sin \Omega t \quad .$$

6.9

With the bow a , the bending natural frequency ω and the rotational frequency Ω . The frequency ratio is $\eta = \Omega/\omega$. As in the case of unbalance the vibration response is harmonic with the frequency Ω . Due to the assumption of isotropic bearing stiffness the orbits are circular in this case. The radius of the circular orbit of the vibration response is shown in Figure 66 in dependence of the dimensionless rotational frequency Ω/ω .

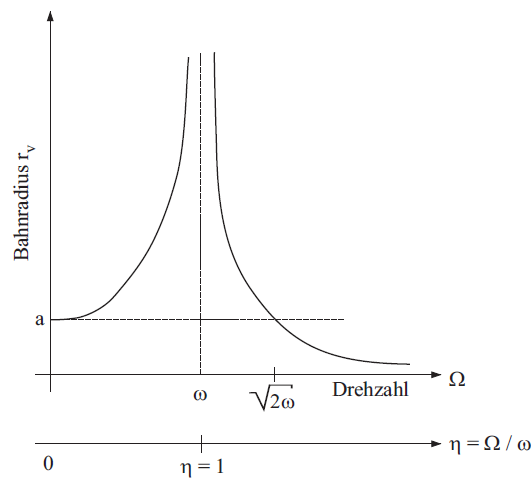


Figure 66. Radius of the circular orbit of the vibration response due to a bow.

As in the case of unbalance the vibrations are high at resonance frequency $\Omega \sim \omega$. Contrary to unbalance there is a vibration response even at very low speeds, which is equal to the bow “a”.

The superposition of the vibration responses due to Unbalance and due to the Bow leads to the following expression 6.10 for the sum of the vibrations.

$$w(t) = \varepsilon \frac{\eta^2}{1-\eta^2} \cos(\Omega t + \beta) + a \frac{1}{1-\eta^2} \cos \Omega t$$

6.10

$$v(t) = \varepsilon \frac{\eta^2}{1-\eta^2} \sin(\Omega t + \beta) + a \frac{1}{1-\eta^2} \sin \Omega t .$$

The expression shows, that Counter-Balancing would be a possible Mitigation to reduce the vibrations of a bow due to coupling errors. With Counter-Balancing the vibration response due to the unbalance shall eliminate the vibration response due to the bow.

6.4 LATERAL VIBRATIONS IN SHAFT TRAINS DUE TO SUPPORT STIFFNESS CHANGES

6.4.1 Introduction

If the vibrations of a turbine train are changing, this may be caused by a change of the excitation (e.g. unbalance forces) or by a change of the system dynamics. Change of the system dynamics may be on the Rotor-Oil-Film-Bearing side or on the Supporting structure side. It is well known, that the overall stiffness or flexibility of the supporting structure with the components of bearing housing, pedestal and foundation may strongly influence the dynamic behavior of the overall system. There may be manifold causes for a vibration change, if the

supporting structure is involved. Vibration problems may for example occur, when the Rotor-Bearing-System is not well connected to the foundation (Figure 67).

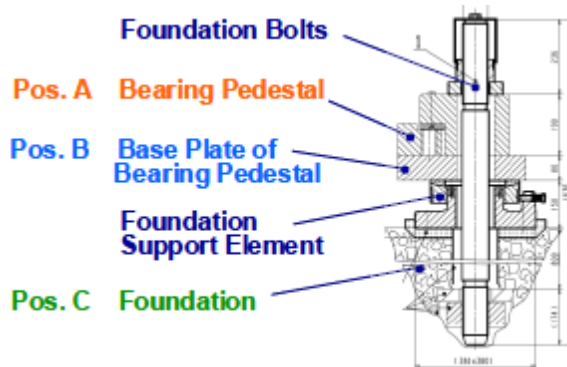


Figure 67. Connection of the Bearing pedestal with base plate to the foundation with foundation bolts.

This often happens for a singular bearing only, showing high vibrations concentrated around this bearing. Resonance may amplify the vibration amplitudes. For the identification of the vibration problem it is necessary to have a clear view about the force transfer from the shaft via the bearings and the bearing pedestals to the foundation. From a machine dynamics point of view it is important to know the dynamic behavior of the complete coupled system. Therefore it is also important to know the support stiffness in vertical and horizontal direction and their influence on the dynamic behavior of the complete system. The vibrations of the system may increase continuously over a long period. Due to settlement of the concrete the foundation bolts loose pre-stress. Settlement usually leads to a reduction of the horizontal and vertical stiffness values.

6.4.2 Physical Description of Lateral Vibrations due to Support Stiffness Changes

In order to demonstrate the influence of support stiffness changes on the lateral vibrations, we use the simple Laval Rotor as before, but consider now flexible supports. As a first approach we introduce spring elements at the bearing locations with different stiffness values for the horizontal and vertical direction s_h and s_v (see Figure 68).

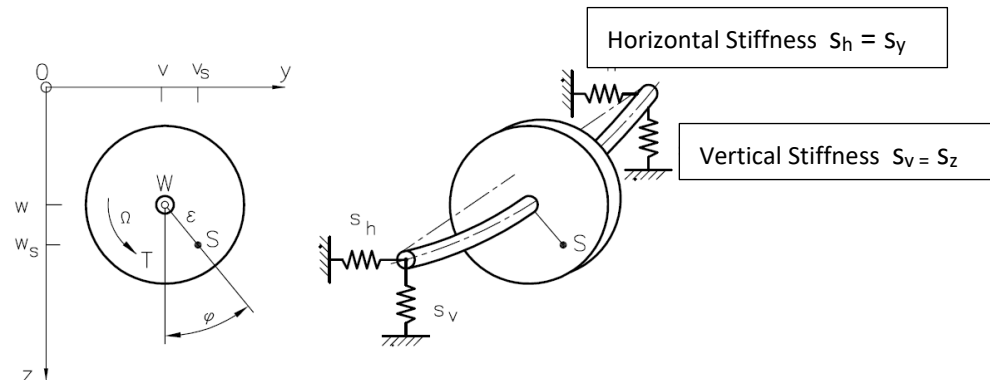


Figure 68. Laval shaft, supported in spring elements with horizontal and vertical stiffness.

In Figure 69 we compare the bending natural frequency of the Laval rotor running in flexible supports ω_z with the bending natural frequency of the Laval rotor in rigid supports ω_0 . In the example the vertical direction z is considered only. However, the result is applicable for the horizontal direction as well.

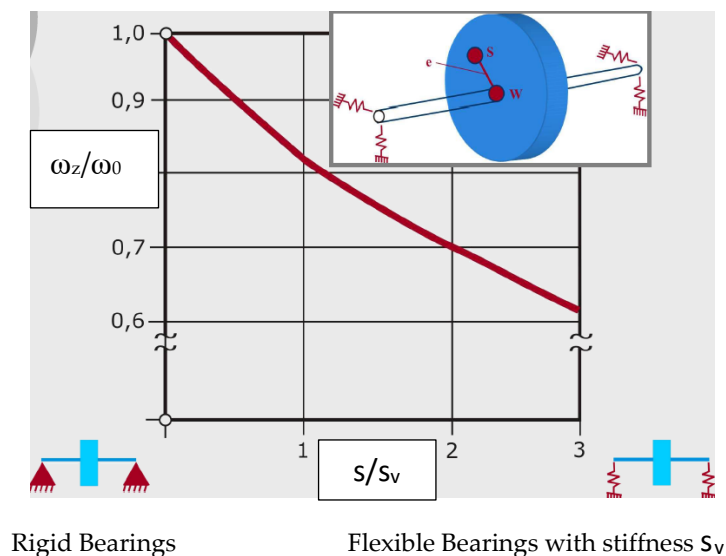


Figure 69. Bending natural frequency of the Laval rotor in flexible supports. Support stiffness $s_v = s_z$, Shaft stiffness s .

The diagram points out, that the bending natural frequency decreases with a decreasing support stiffness. This means that the critical speeds and resonances are also reduced. When the support stiffness values in horizontal and vertical direction are different, the bending natural frequencies for the two directions will be different as well.

Based on the simple equations of motion for the Laval shaft with anisotropic flexible support stiffness and unbalance excitation 6.11.

$$\begin{aligned} m\ddot{w} + s_z w &= \varepsilon m \Omega^2 \cos \Omega t \\ m\ddot{v} + s_y v &= \varepsilon m \Omega^2 \sin \Omega t \end{aligned} \quad \boxed{6.11}$$

We obtain the amplitudes w_ε and v_ε of the unbalance vibration response for the horizontal and vertical direction, as described in 6.12.

$$\begin{aligned} \hat{w}_\varepsilon &= \varepsilon \frac{\Omega^2}{\omega_z^2 - \Omega^2} = \varepsilon \frac{(\Omega/\omega_z)^2}{1 - (\Omega/\omega_z)^2} = \varepsilon \frac{\eta_z^2}{1 - \eta_z^2} \\ \hat{v}_\varepsilon &= \varepsilon \frac{\Omega^2}{\omega_y^2 - \Omega^2} = \varepsilon \frac{(\Omega/\omega_y)^2}{1 - (\Omega/\omega_y)^2} = \varepsilon \frac{\eta_y^2}{1 - \eta_y^2} \end{aligned} \quad \boxed{6.12}$$

ω_z and ω_y are the bending natural frequencies for the vertical and horizontal direction and η_z and η_y the corresponding dimensionless rotational frequencies Ω/ω_i ($i = z, y$).

Figure 70 shows the two bending natural frequencies and the the amplitudes w_ε and v_ε versus the rotational frequency Ω for the Laval rotor with different support stiffness values in horizontal and vertical direction.

Depending on the operational point of design, vibrations may decrease or increase in case of support stiffness changes. It is a question, in which direction the resonance curves are shifted due to the support stiffness changes.

For more complex systems like real steam turbine trains in a Power Plant, Finite Element (FE) routines can be used in order to study the influence of stiffness changes in the support system. Such FE Analysis can be performed by manufacturers of turbine trains.

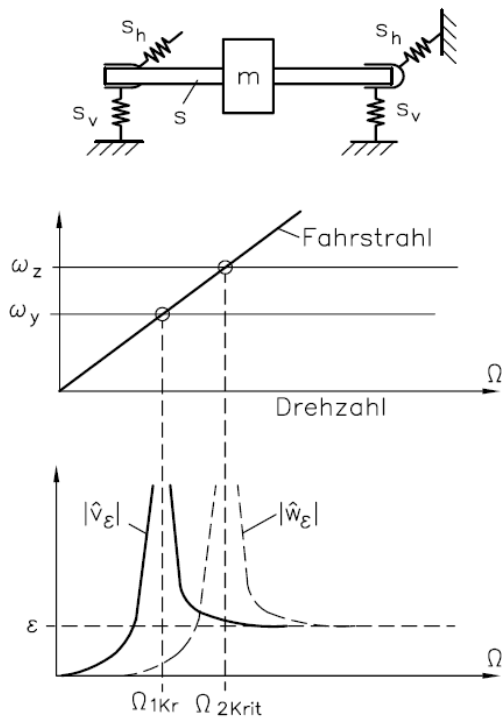


Figure 70. Bending Natural frequencies and Vibration response due to Unbalance in case of anisotropic Support Stiffness.

DIAM, A MATRIX TOOL FOR TURBINE AND GENERATOR VIBRATIONS

The findings of this project can be very helpful, to solve lateral turbine and generator vibration problems in power plants in a more efficient way as before. Matrices have been developed to assist in the detection, investigation, analysis and mitigation of lateral turbine- and generator vibrations. The results of the project can also be very useful in transferring knowledge to new personnel in the power plants.

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