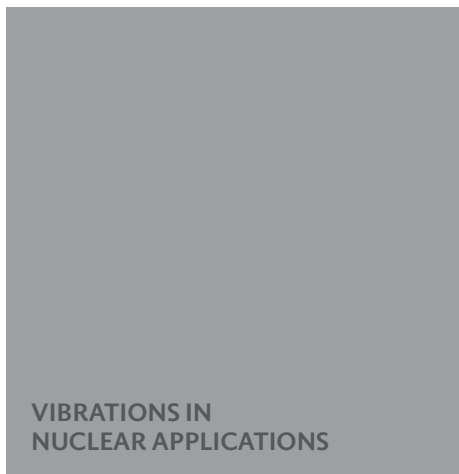


# LATERAL TURBINE AND GENERATOR VIBRATIONS

REPORT 2016:294





# **Lateral Turbine and Generator Vibrations**

## **Analysis and Mitigation**

**RAINER NORDMANN**

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## Foreword

Although turbines and generators are not safety related components, it is of course crucial to have them functioning properly to have a high availability and efficiency in the plant. During the years there have been different problems related to turbine and generator vibrations and therefore it is very important to improve the performance and minimize the risk for vibration problems.

In this project professor Rainer Nordmann from TU Darmstadt performed a mapping of vibration problems in turbines and generators in the nuclear power plants Forsmark, Oskarshamn and Olkiluoto. The experienced problems are grouped into problem areas and mitigating actions that can be taken are described. The results are presented in two separate reports, one describing lateral vibrations (2016:294) and one describing torsional and stator vibrations (2016:295). There is also a simplified guide that can be used to identify the cause of lateral vibrations.

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

## Sammanfattning

Syftet med projektet "Turbin- och generatorvibrationer - Analys och åtgärder" är att samla kunskap och erfarenhet vad gäller problem med turbin- och generatorvibrationer från de tre kärnkraftverken i Sverige (Oskarshamn OKG och Forsmark FKA) och Finland (Olkiluoto TVO) samt att studera hur vibrationsproblemen analyserades och åtgärdades.

Det insamlade materialet består av rapporter och information som erhållits från intervjuer med vibrations- och underhållsspecialister vid kraftverken. Materialet har sedan grupperats i tre olika huvudområden:

- Lateral vibrationsproblem
- Torsionsvibrationsproblemen
- Problem med statorvibrationer

Projektresultaten presenteras i två olika rapporter. I del 1 - denna rapport - diskuteras Lateral vibrationer (2016:294) och i del 2 diskuteras Torsions- och statorvibrationer (2016:295).

De laterala vibrationsproblemen har delats upp i undergrupper:

- 1X Lateral vibrationer i axlarna till följd av obalans
- Cykliska eller spiralformade laterala vibrationer
- Friktionsorsakade laterala generatorvibrationer
- Lateral vibrationer till följd av förändringar av havsvattnets temperatur
- Lateral vibrationer till följd av obalans i tröghetsmoment
- Instabila laterala vibrationer

Rotorsprickor behandlades inte i denna rapport, eftersom detta inte ingått i det underlagsmaterial som fått från kraftverken. Sprickor i bladen, se del 2. Se även del 2 (rapport 2016:295) för 2XRPM vibrationer på grund av generatorns magnetiska krafter.

I denna rapport presenteras, för var och en av undergrupperna, en teoretisk och tillämpad beskrivning av de specifika laterala problemen, beskriver hur dessa problem kan analyseras och föreslår vilka åtgärder som kan tillämpas för att lösa vibrationsproblemen.

Följande slutsatser kan dras:

- För varje specifikt vibrationsproblem gäller att en god kunskap om de fysiska förhållandena utgör en viktig bas för en bra lösning av vibrationsproblemet
- De analysverktyg som används i kraftverken (givare, signalbehandlingsenheter, analysenheter) är nödvändiga för analys av vibrationer och för att efter införda åtgärder säkerställa att problemen verkligen är åtgärdade.

- Ett samarbete med turbintillverkaren rekommenderas.
- Modellering och beräkningar kan ofta kan vara till stor hjälp för att tillse att rätt åtgärder införs

Ett praktiskt analysverktyg har utarbetats för att identifiera laterala vibrationsproblem utifrån problemets karaktär. Verktöget finns att laddas ned från Energiforsks hemsida.

Denna studie kan vara till stor hjälp för att överföra kunskap till ny personal, för att stödja planerade förändringar i turbinsträngen och hitta snabba lösningar när vibrationsproblem uppstår i turbiner

## Summary

The objective of the project “Turbine and Generator Vibrations – Analysis and Mitigation” is to assemble knowledge (reports) and experience in the area of turbine and generator vibration problems from the three Nuclear Power Plants in Sweden (Oskarshamn OKG and Forsmark FKA) and Finland (Olkiluoto TVO) and to study in some depth how vibration problems were investigated and mitigated.

The collected material from reports and the information obtained from interviews with vibration and maintenance specialists in the power plants was a good base for the grouping of vibration problem areas. As a result the following three main groups have been identified, independent from the power plants:

- Lateral Vibration Problems of the turbine-generator rotor trains,
- Torsional Vibration Problems of the turbine-generator rotor trains
- Stator Vibration Problems

The overall technical report is subdivided in two parts. In Part 1- this report - **Lateral Vibrations** are discussed, in the shorter Part 2 **Torsional Vibrations** and **Stator Vibrations**.

Each of the main vibration problem groups was further subdivided. As example the subgroups of Lateral Vibrations (Part 1) deal with the following problems:

- 1X 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

**Rotor cracks** were not treated in this report, this problem was not included in the delivered plant reports and not discussed in the meetings. **Blade cracks** see Part 2. **2XRPM vibrations** due to Magnetic forces of the Generator see also Part 2.

This technical report (Part 1) presents for each of the different subgroups a physical and practical description of the specific lateral vibration problem, it describes the applied investigation techniques, reports about the observed lateral vibrations in the power plants and suggests mitigation activities, which were used to solve the vibration problems.

It can be concluded, that

- for each specific vibration problem a good knowledge about the physical relations is an important base for a good solution of the vibration problem
- the used experimental tools in the power plants (transducers, signal processing units, analyzers) are necessary for the vibration analysis and for the control of the success of measures.

- a cooperation with the turbine manufacturer is in any case recommended
- numerical tools (support from manufacturer) can often be very helpful, in order to support mitigation activities

This study can be very helpful, in order to transfer knowledge to new personnel, to support planned changes of turbine trains and to find fast solutions when vibration problems occur in turbine trains.

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# 1 Introduction – Project description

## 1.1 OBJECTIVE OF THE PROJECT

The objective of this project is to assemble knowledge and experience in the area of turbine and generator vibration problems in three Nuclear Power Plants in Sweden (Oskarshamn OKG and Forsmark FKA) and Finland (Olkiluoto TVO) and to study in some depth how they were investigated and mitigated. The collected material and the information obtained from interviews with vibration specialists in the power plants were analysed and the project results were put together in this report.

The presented project results can be helpful for

- a better knowledge transfer to new personnel,
- for planned changes in turbine trains,
- for fast solutions when problems occur in turbine trains.

## 1.2 SCOPE OF THE TASK

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

1. To assemble information and documentation (e.g. reports) from the participating power plants: OKG, FKA, TVO
2. To group the information and documentation into different problem areas. The structure of this work will therefore be problem specific and not plant specific.
3. To produce a technical report with a physical and practical description of the different encountered problems and with a presentation of the applied investigation techniques and the mitigation activities.

The geographical locations of the three power plants are shown in Figure 1.



Figure 1 Geographical location of the three power plants.

## 2 Submitted Reports from 3 Power Plants

Subject of this investigation are the turbine generator shaft trains in the three power plants Oskarshamn, Olkiluoto and Forsmark. Each power plant has a number of units and each unit consists of a number of shaft trains with steam turbines and a generator, which are either running with 1500 rpm or 3000 rpm.

Oskarshamn (Sweden) has three units: O1, O2 and O3.

| Unit | Number of Shaft trains | Power   | Operating speed |
|------|------------------------|---------|-----------------|
| O1   | 1                      | 494 MW  | 3000 rpm        |
| O2   | 1                      | 661 MW  | 3000 rpm        |
| O3   | 1                      | 1450 MW | 1500 rpm        |

Olkiluoto (Finland) has two units: OL1 and OL2

| Unit | Number of Shaft trains | Power  | Operating speed |
|------|------------------------|--------|-----------------|
| OL1  | 1                      | 870 MW | 3000 rpm        |
| OL2  | 1                      | 870 MW | 3000 rpm        |

Forsmark (Sweden) has the three units F1, F2 and F3.

| Unit | Number of Shaft trains | Power                | Operating speed |
|------|------------------------|----------------------|-----------------|
| F1   | 2                      | F11, F12 each 510 MW | 3000 rpm        |
| F2   | 2                      | F21, F22 each 560 MW | 3000 rpm        |
| F3   | 1                      | 1230 MW              | 1500 rpm        |

From the three power plants several reports have been submitted to the author of this report for review:

Oskarshamn O1, O2, O3: 150 reports & attachments from years 2006 – 2015

Olkiluoto OL1, OL2 : 46 reports & attachments from years 2007 – 2015

Forsmark F1, F2, F3 : 15 reports & attachments from years 2004 - 2015

The reports and the corresponding attachments describe the Turbine & Generator vibration issues for each of the three plants. As a preparation for the meeting with discussions in the power plants a first grouping into problem areas has been undertaken, based on the documentation received.

### 3 Visits with Interviews at the 3 Power Plants

For each of the three power plants a one day meeting was arranged with the maintenance and vibration specialists of the power plants. In these meetings further explanations and discussions should help, to better understand the observed turbine and generator vibrations, described in the delivered reports. The discussions in the meetings should also help, to explain the applied analysis techniques and the mitigation methods for the solution of the vibration problems.

The author of this report prepared the meeting by reading the delivered reports, by grouping the contents in a first step into different vibration problem areas and to prepare questions regarding the applied analysis techniques and mitigation methods.

The meetings were held at the location of the three power plants:

Meeting in the power plant of Oskarshamn on December 14 – 9.00 am to 5.00 pm with the following participants:

Tobias Törnström, Elisabet Blom, Kent Andersson, Carl Möller, Robert Larsson, Rainer Nordmann and other engineers from the plant

Meeting in the power plant of Olkiluoto on December 16 – 9.00 am to 5.00 pm with the following participants:

Paulus Smeekes, Petri Lemettinen, Jako Rostedt and Rainer Nordmann

Meeting in the power plant of Forsmark on January 19 – 8.00 am to 5.00 pm with the following participants:

Ylva Vidhög, Magnus Adolfsson and Rainer Nordmann

As a result of the meetings in the three power plants with valuable discussions an improved grouping of the vibration problems was performed. By this clear grouping of the observed vibrations it was easier to describe the physics of the vibration phenomena, the investigation technique (analysis) and the method of mitigation. The performed grouping of vibration problems was also a good basis for further search of reports in direction of the vibration problems and for a good experience exchange with the other plants.

The result of the detailed grouping into Vibration Problem Areas is described in the following Chapter 4.1.

## 4 Vibration Problem Areas

### 4.1 GROUPING INTO VIBRATION PROBLEM AREAS

As a result of the detailed study of the delivered reports from the three power plants and the following visits in the power plants and interviews with the vibration specialists a more detailed grouping into vibration problem areas was possible.

The following groups have been identified:

#### **Lateral Vibration Problems**

- 1X Lateral Vibrations in shaft trains due to Unbalance
- Cyclic or Spiral Lateral Vibrations
- Friction Induced Lateral Vibrations in Generators
- Lateral Vibration due to changes of Seawater temperature, condenser vacuum and load
- Lateral Vibrations due to Unequal Moments of Inertia
- Unstable Lateral Vibrations

#### **Torsional Vibration Problems (see Part 2 of the report)**

- Torsional Vibrations due to Electrical Faults
- Torsional Vibrations due to Sub Synchronous Resonance SSR
- Blade Vibrations in Last Stage LPT due to Fluid Structure Interaction

#### **Stator Vibration Problems (see Part 2 of the report)**

- 2X End Winding Vibrations in the Generator Stator
- 2X Stator Core Axial Vibration
- 2X Stator Cooling Pipe Vibrations

The following tables illustrate in which of the power plants these vibrations have been observed.

Table 4.1.1 shows six different Lateral Vibration phenomena, which have been identified. Besides the classical 1X Lateral Vibrations due to unbalance forces, which often appear in power plant shaft trains the following other lateral vibrations have been observed: Cyclic or spiral vibrations, Friction induced lateral vibrations in generators, lateral vibrations due to changes of the sea water temperature, the condenser vacuum and the load, unstable lateral vibrations and finally lateral vibrations due to unequal moments of inertia. The three first vibration types are caused by some kind of unbalance (e.g. thermal bow, bow due to friction). Table 4.1.1 shows in which of the observed power plants these phenomena were observed. Further details of the vibration characteristics are described in the columns of the table.

| Vibrations                                                                     | Oskarshamn                                                                                   | Olkiluoto                                                                                      | Forsmark                                                                                          |
|--------------------------------------------------------------------------------|----------------------------------------------------------------------------------------------|------------------------------------------------------------------------------------------------|---------------------------------------------------------------------------------------------------|
| <b>1X Lateral Vibrations in Shaft Trains due to Unbalance</b>                  | 1X Vibrations are normal. Observe Run up curves. Critical speeds. Damping Long term behav    | 1X Vibrations are normal. Observe Run up curves. Critical speeds. Damping Long term behav      | Problem with F3 Gen.-rotor during warming in the balance machine before dismantling and assembly. |
| <b>Cyclic or Spiral Vibrations</b>                                             | O3: Cyclic vibrations since 2011 due to Oil seal system in Generator. 10 to 50 um. 12 hours. | OL1: Strong unstable cyclic vibrations after R108 at 1800 rpm. Labyrinth Seals?                | F21: Cyclic vibrations during Runout tests (RA10) at 500 rpm 100 um at B03. 1 hour. Steam Seal    |
| <b>Friction Induced Lateral Vibrations in Generators</b>                       | Not reported during visit                                                                    | OL2/R7: problem appeared with R7 Rotor Friction betw. Windings and Geno Rotor                  | Not reported                                                                                      |
| <b>Lat. Vibration due to change in sea water temperature/ Condenser vacuum</b> | 1X Vibration increase due to water on Rotor with increasing sea water temp. Thermal bow.     | Change of 1X Vibration during the year due to changing cond. Pressure +temp. Stat. Deformation | 1X Vibrations show cyclic behaviour. Rubbing. Period 1 day.                                       |
| <b>Unstable Lateral Vibrations</b>                                             | Not reported                                                                                 | Not reported                                                                                   | Bearing Instability at first bending mode (8 Hz) of the LP turbine                                |
| <b>Lateral Vibrations due Unequal Moments of Inertia</b>                       | Not at O3, because of 4 pole Generator 1500 rpm. O1, O2?                                     | 2X vibrations of 150 um at weight resonance of 2 pole Generator 350 rpm.                       | Not reported                                                                                      |

Table 4.1.1 Grouping of vibrations into six different lateral vibration phenomena

In a similar way torsional vibrations were considered. Table 4.1.2 shows three of the most important torsional vibration phenomena, which have been observed. They include Torsional vibrations due to electrical faults, torsional vibrations due to Sub Synchronous Resonance (SSR) and Blade vibrations in the last stage LPT. More details are described in Part 2 of the report.

| <b>Vibrations</b>                                    | <b>Oskarshamn</b>                               | <b>Olkiluoto</b>                                | <b>Forsmark</b>                                                                    |
|------------------------------------------------------|-------------------------------------------------|-------------------------------------------------|------------------------------------------------------------------------------------|
| <b>Torsional Vibrations due to Electrical Faults</b> | Not reported                                    | Not reported                                    | Disturbance in electrical grid. Step change. 2X and 1X vibrations. Coupling moves. |
| <b>Torsional Vibrations due to SSR</b>               | Not investigated in Oskarshamn. Grid is stable? | Olkiluoto has a SSR protection system (Fingrid) | Due to SSR a blade resonance of F3 was excited. Cracks in 2 LS blades.             |
| <b>Blade Vibrations in Last Stage LPT</b>            | Not reported                                    | Not reported                                    | Strong LS Blade Vibrations (70 mm axial) at run down after vacuum breaking         |

Table 4.1.2 Grouping of vibrations into three different torsional vibration phenomena

| <b>Vibrations</b>                                    | <b>Oskarshamn</b> | <b>Olkiluoto</b>                                                       | <b>Forsmark</b>                                                                       |
|------------------------------------------------------|-------------------|------------------------------------------------------------------------|---------------------------------------------------------------------------------------|
| <b>2X End Winding Vibrations in Generator Stator</b> | Not reported      | Strong End winding vibrations 100 Hz up to 60 mm/sec from 2000 to 2007 | 2X 100 Hz End Winding Vibrations in F2 2014). Cracks found.                           |
| <b>2X Stator Core Axial Vibrations</b>               | Not reported      | High Axial Stator Core Vibrations up to 50 mm/sec                      | F1 and F2 had problems after many years running in old Gen. Design.                   |
| <b>2X Stator Cooling Pipe Vibrations</b>             | Not reported      | High axial and horiz. Cooling pipe vibrations up to 48 mm/sec.         | Problems on old Stator design of F2 with high pipe vibrations, resonance and leakage. |

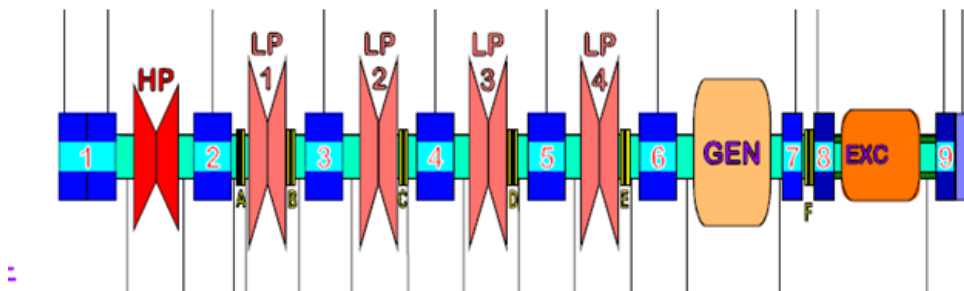
Table 4.1.3 Grouping of vibrations into three different Stator vibration phenomena

Finally Table 4.1.3 shows the identified Stator vibration problems in the power plants. The source of these vibrations are electrodynamic and electromagnetic forces acting in the Generator of the shaft trains. They usually occur as 2X vibration signals. More details are also described in Part 2 of the report.

## 4.2 ANALYSIS OF THE VIBRATION PROBLEMS

The base for the analysis of vibration problems in power plants is to measure absolute vibration velocities in mm/sec on the bearings and relative shaft vibrations in  $\mu\text{m}$  (displacements) close to the bearings or at shaft ends. Measurements usually are taken in two directions (horizontal and vertical or in other directions, e.g. 45 deg.). Measurements are sometimes also taken in axial directions on the bearings (e.g. Forsmark F3 all bearings). Forsmark F1 has axial transducers on bearings 5-6-7-8.

### Transducer Locations for Bearing Absolute Vibrations



### Transducer Locations for Shaft Relative Vibrations

Figure 2 Locations of the Vibration transducers in a shaft train

The transducer signals are available in the time domain. By signal processing different functions in the time or in the frequency domain can then be determined:

- time-history of the vibration signal
- vibration-orbits e.g. for one location of the shaft train
- vibration amplitudes and phase over short times or long times
- Frequency spectra to analyze the frequency content,
- e.g. 1X, 2X, 1/2X, higher Harmonics and broadband.

Typical presentations of lateral vibrations in the frequency domain are shown in the following Figure 3. They include

- Run up and Run down curves (Power = 0) with 1X amplitudes and phase versus rotational speed. 2X 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 plots are also used for normal operation at full speed and full load to evaluate changes during operation.

- Polar plot at low speed (e.g. 500 rpm), to check whether the amplitudes are stable and within a defined alarm circle before run up.

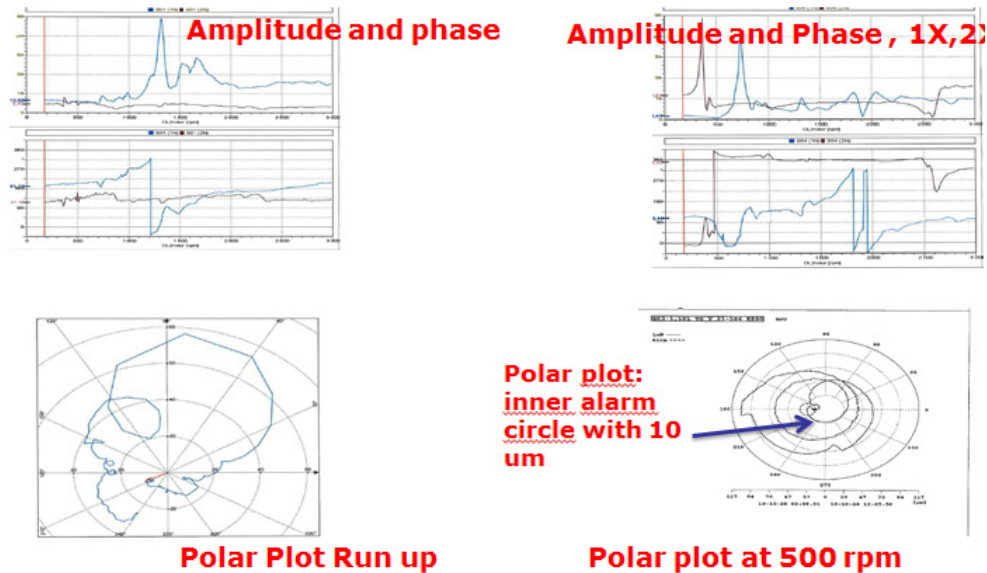


Figure 3 Typical presentations of vibrations for shaft trains in power plants

Besides the presentations in the frequency domain amplitudes and phase are also presented versus time, either short time or for a longer period. Examples are:

- Short term presentation (e.g. 1 week or 1 month) of vibration amplitudes and phase versus time. Which changes are observed versus time?
- Short term presentation (e.g. 1 week or 1 month) of static shaft position in the bearings. Which changes of static positions are observed versus time?
- Long term presentation (e.g. 1 year) of vibration amplitudes (and phase) versus time. Which changes are observed versus time?
- Comparison of Vibration values over years (Especially values taken at the revisions).

#### 4.3 MITIGATION OF THE VIBRATION PROBLEMS

In the following Table 4.3.1 a summary of the study “Vibrations in Turbines and Generators – Analysis and Mitigation” is presented, in this case especially for the group of lateral vibrations. This summary includes besides the identified vibration problem areas (4.1 Grouping) also the used Analysis Techniques (4.2 Investigations) and finally the applied Mitigation methods (4.3 Mitigation) in order to fix the vibration problem. A detailed discussion of these topics will later be presented in the chapters for the identified vibration problems.

In Part 2 of the overall report similar tables will be shown for the cases of torsional vibrations and stator vibrations.

| 4.1 Grouping                                                     | 4.2 Investigation                                                                                    | 4.3 Mitigation                                                                                                |
|------------------------------------------------------------------|------------------------------------------------------------------------------------------------------|---------------------------------------------------------------------------------------------------------------|
| <b>1X Lateral Vibrations in Shaft Trains due to Unbalance</b>    | Run up and Run down curves. Critical Speeds (peaks). Damping? Short term and long term behav.?       | Balancing with influence coefficients. Improve damping. Shift resonances by tuning.                           |
| <b>Cyclic or Spiral Vibrations</b>                               | Amplitude and phase versus time. Determine change of amplitudes and period. Polar plot. Rot.Vector.  | Avoid rubbing by adjusting seal Pressure control in oil seal system. Damping. Change load or Condenser vacuum |
| <b>Friction Induced Lateral Vibrations in Generators</b>         | Heat runs. Influence of temperatures, pressure and speed on vibrations (Geno).                       | Avoid unsymmetry in circumference for friction, pressure, temperature. Control winding temperature.           |
| <b>Lateral Vibrations due to change in Sea water temperature</b> | Investigate vibration versus time depending on condenser temp. , pressure and power                  | Protect rotor by sheet metal against water. Reduce active Power. Check process param. Adjust seals.           |
| <b>Unstable Lateral Vibrations</b>                               | Investigate Frequency spectra and look for half frequency components, e.g. from bearings or HP seals | Change bearing parameters, improve damping. Decrease power in case of HP seal instability.                    |
| <b>Lateral Vibrations due Unequal Moments of Inertia</b>         | Investigate lateral 2X and 1X vibration components in the range of .5 to 1.0 of 1st Geno Critical    | Slots in 2 pole Generator rotor. Improve damping                                                              |

Table 4.3.1 Applied Analysis Techniques and Mitigation Methods in the three power plants to solve Lateral Vibrations of the shaft trains

## 5 Lateral Vibration Problems – Analysis and Mitigation

### 5.1 1X – LATERAL VIBRATIONS IN SHAFT TRAINS DUE TO UNBALANCE

#### 5.1.1 Physical Description

To explain the the 1X Lateral vibrations in shaft trains we use a very simple rotating system (Figure 4), the so called Laval shaft (named after G. Laval, a Swedish engineer, 1845- 1913). It consists of a flexible, massless shaft with bending stiffness  $c$  and a disk with mass  $m$  and a mass eccentricity  $e$ . The disk is fixed in the center of the shaft. In this simple example the bearings are considered to be rigid. In real shaft trains they are flexible and have damping due to the oil film in the bearings. The rotor system is running with angular velocity  $\Omega$ . The simple dynamic system has a natural bending frequency  $\omega$ , which is determined by the square root of the ratio  $c/m$ .

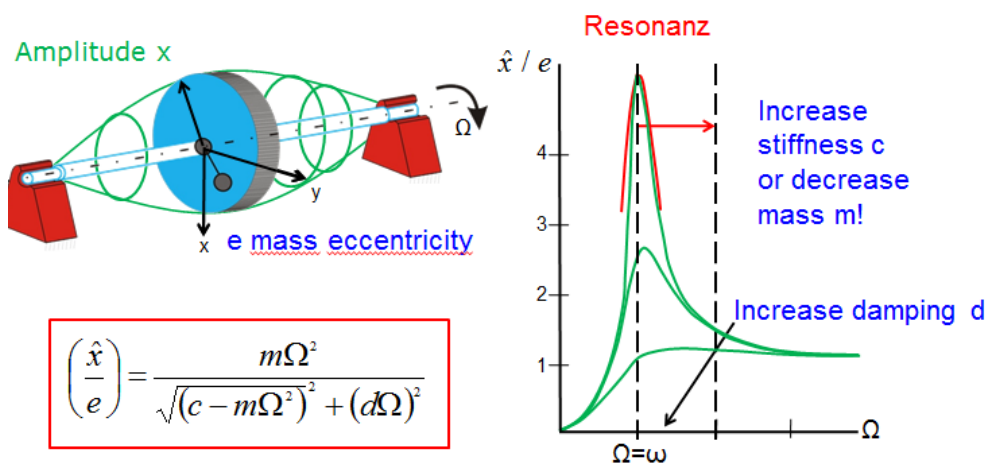


Figure 4 Dynamic behavior of a Laval Shaft, Run up curve and Critical Speed

Due to the unbalance ( $m \times e$ ) centrifugal forces excite the flexible shaft system when it is running with angular velocity  $\Omega$ . The resulting vibrations in the two directions  $x$  and  $y$  are harmonic time functions with the rotational frequency  $\Omega$ , where the vibration amplitudes and the phase (time shift between vibration and force excitation) depend on the frequency  $\Omega$ , the mass eccentricity  $e$  and the system parameters  $c$  and  $m$ . Besides  $c$  and  $m$  another important system parameter is the damping  $d$ , which helps to reduce the vibrations. The vibration behavior of the Laval Shaft with unbalance during a run up is described by the formula in Figure 4. The corresponding vibration amplitudes  $x/e$  versus the rotational frequency  $\Omega$  are presented on the right side of Figure 4 (Phase is not shown in this presentation). At low speeds the vibrations are small. They increase with running speed  $\Omega$  and reach their maximum, when  $\Omega$  is equal to the natural frequency  $\omega$ .

This is the case of the critical speed, when the rotor is running in a resonance condition. When the rotor speed is further increased, the amplitudes are decreasing again due to the so called self-centering effect.

An important question for the Vibration Engineer is, how the vibrations can be influenced (reduced)? As mentioned before they depend on the system parameters of the dynamic system  $m$ ,  $c$ ,  $d$ , on the excitation parameter-the mass eccentricity  $e$ - and on the rotational frequency  $\Omega$ . The rotational frequency  $\Omega$  should not be too close to the critical frequency  $\omega$ . This can be achieved by tuning of the parameters  $c$  and  $m$ . Furthermore it can be seen in Figure 4, that damping helps to reduce the amplitudes, especially in a resonance. In real shaft trains damping is mainly coming from the oil film bearings.

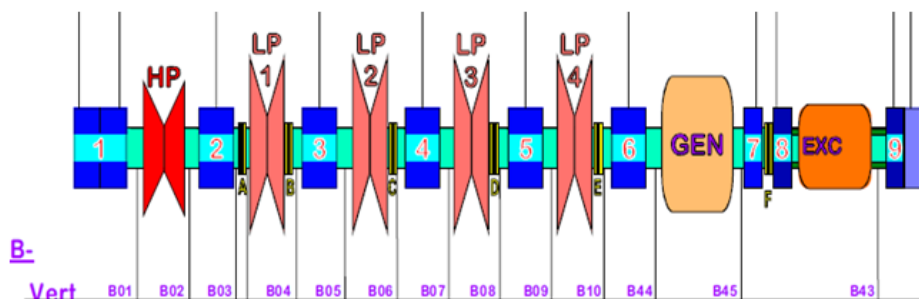
Regarding excitation, a very important parameter is the mass eccentricity  $e$ , which should be as small as possible. The reduction of  $e$ , respectively the unbalance  $m \times e$ , can be achieved by the procedure of Balancing! Later on we will see, that unbalance excitation may appear in different ways (see chapters 5.2, 5.3, 5.4).

The simple Laval shaft has only one critical speed (Resonance). However, the basic vibration character can already be explained by this rotor system. Real shaft trains consist of several part rotors (turbines and generator), rigidly connected via the couplings. They are multi-degree of freedom systems and may have several critical speeds. Their dynamic behavior is also very much influenced by the dynamic behavior of the oil film bearings with stiffness and damping coefficients.

### 5.1.2 Investigation Technique – Vibration Problem Analysis

The usual way of investigation for unbalance vibrations in a power plant is to measure the relative shaft vibrations in  $\mu\text{m}$  and the absolute bearing vibrations in  $\text{mm/sec}$  with well suited vibration transducers (see Figure 5). This was already shown in chapter 4.2.

#### Measurement of absolute bearing vibrations in mm/sec



#### Measurement of relative shaft vibrations in $\mu\text{m}$

Figure 5 Investigation Techniques in Power Plants – Measurements

Another way of investigation – in the machine design process as well in case of vibration problems in the plant – is the Numerical Analysis with the Finite Element method. An example is given in Figure 6, showing a run up curve up to 25 Hz with resonances between 10 to 15 Hz. In this case the shaft train is excited by an unbalance in the center of the ND3- rotor (LPT3).

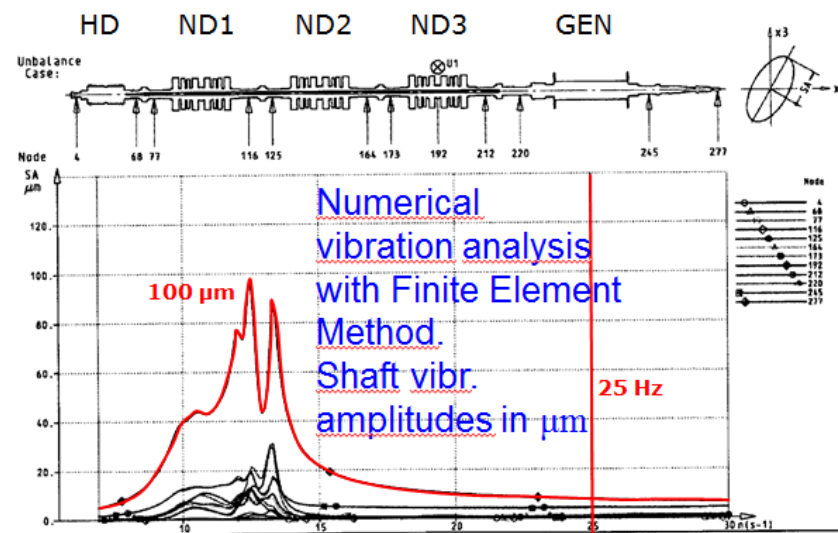


Figure 6 Investigation techniques in power plants – Numerical Analysis

### 5.1.3 Observed 1X-Lateral Vibrations in Power Plants

Figures 7 and 8 show typical observed 1X lateral vibrations in a power plant, in this case the vertical relative shaft vibrations during a run up to 3000 rpm, taken at the bearing locations B05 and B06 (LP2-rotor) in unit OL1 of Olkiluoto.

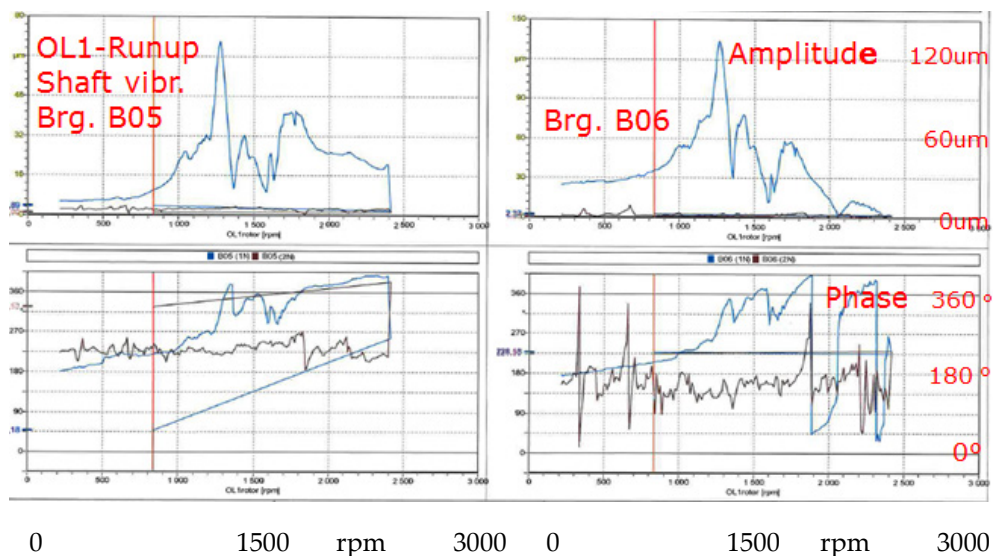


Figure 7 Amplitude and phase of vertical relative shaft vibrations (Run up) versus speed at bearing locations B05 and B06 (LP2 rotor) in power plant unit OL1

Figure 7 presents amplitude and phase versus the rotational speed, while in Figure 8 the polar diagram is used, showing a vector presentation of the vibrations. In the last case the rotational speed is a parameter along the polar plot.

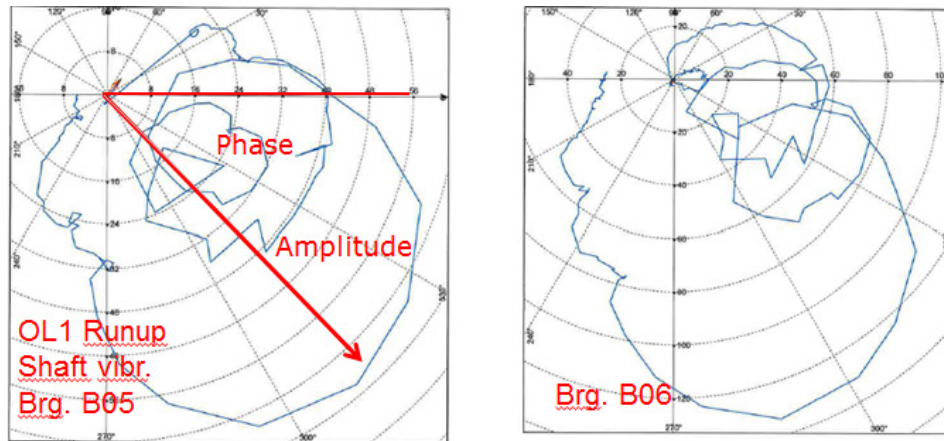


Figure 8 Amplitude and phase of vertical relative shaft vibrations (Run up) at bearing locations B05 and B06 (LP2 rotor) in power plant unit OL1. Shown as Polar Plot.

It was reported, that the lateral vibrations of unit Forsmark F3 are very low damped. The following Figure 9 shows for example the relative amplitudes of the shaft at the HP turbine (GS, horizontal) versus speed. The resonance at 478 rpm has a very sharp peak with amplitudes up to 420  $\mu\text{m}$ . If we consider the corresponding mode shape, it looks that the bearings are relative stiff compared to the bending stiffness of the rotor. Due to the high bearing stiffness, the damping contribution from the bearing is small because of restricted motion (damping needs velocity). The low damping may also be a disadvantage in case of destabilizing mechanisms (seals).

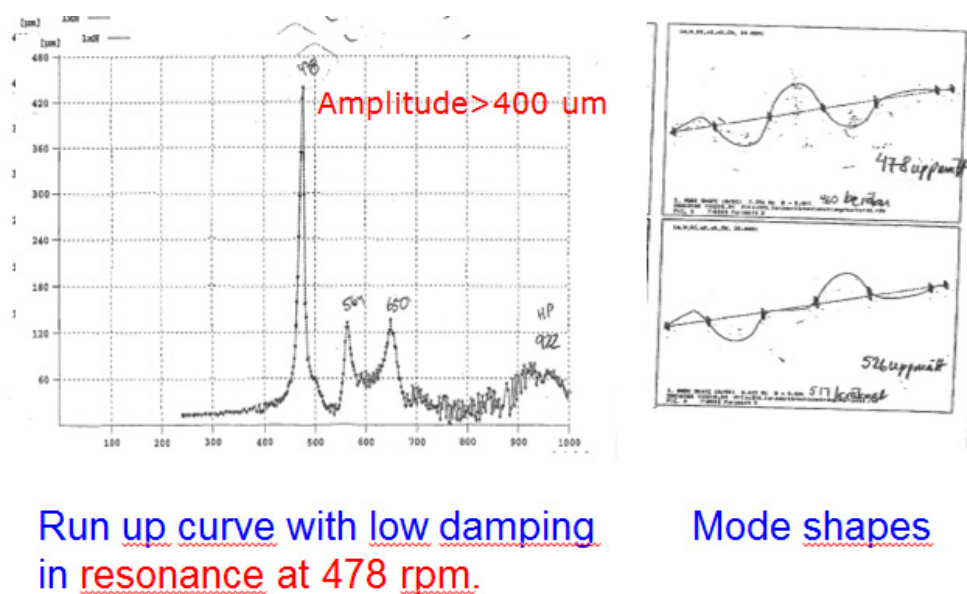
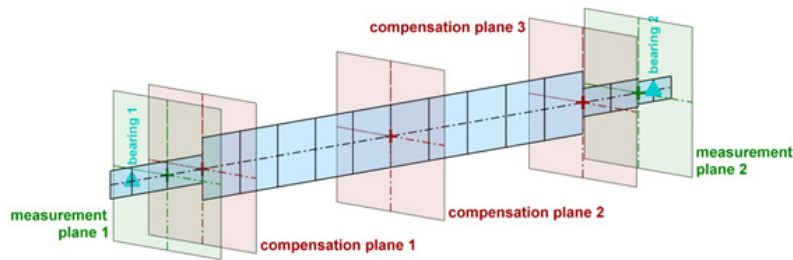


Figure 9 Run up curve of unit F3 (Forsmark) with low damping in resonance at 478 rpm. The right picture shows the corresponding mode shape.

#### 5.1.4 Solution – Vibration Problem Mitigation

If 1X-lateral vibrations in a shaft train are too high –related to any standard, e.g. ISO 7919-2, ISO 10816-2 or related to contract agreements – measures should be undertaken to reduce the vibrations. As described before vibrations can be influenced either by the system parameters – e.g. masses and stiffness of the shaft along the train but also by the dynamic parameters of the bearings (stiffness and damping) – or by the excitation parameters, e.g. by weights to compensate the original unbalance. Most often the vibration problem mitigation in case of 1X-lateral vibrations in shaft trains is balancing, where balancing with influence coefficients is a known and approved method. This method is also used in the three power plants of Oskarshamn, Olkiluoto and Forsmark. In order to briefly describe the procedure, we again use a simplified rotor system (see Figure 10), which can be considered as a part rotor, running in two bearings (for example one of the turbine rotors or a generator rotor).

$$M\ddot{x}(t) + Kx(t) = F(t) \qquad (K - \Omega^2 M) x(\Omega) = F(\Omega)$$



#### Balancing with Influence Coefficients

$$\alpha_{ik}(\Omega) = \frac{x_i(\Omega)}{U_k} \qquad F(\Omega) = U \cdot \Omega^2$$

$$U_k = m_k \cdot e_k$$

Figure 10 Balancing of a simple rotor in two bearings by means of Influence coefficients

To describe the balancing procedure in this example three compensation planes are selected, where compensation weights can be applied. The 1X-lateral unbalance vibrations can be measured in two measurement planes. Usually these planes are selected at the location of the bearings or at the shaft ends.

Frequency Response Functions (FRF) for the presented rotor system are the base for the introduction of influence coefficients (see Figure 10). They can be obtained experimentally by applying test unbalances at the 3 degrees of freedom  $k$  and determining the displacements at the degrees of freedom  $i$ :

$$\alpha_{ik}(\Omega) = \frac{x_i(\Omega)}{U_k}$$

The influence coefficients depend on the system parameters mass, damping and stiffness (shaft, bearings and foundation) and on the rotational frequency  $\Omega$ . In a numerical procedure the influence coefficients can be calculated from the Frequency Response Functions. The dimension of the matrix of influence coefficients is dependent on the number of displacement measurements (number of rows) and the number of compensation planes (number of columns). In the presented example we have introduced 3 compensation planes and 2 displacement measurements. Therefore the matrix in Figure 11 contains 6 influence coefficients, if one measurement (vertical or horizontal) is relevant for one measurement plane. As shown in Figure 11 the matrix of influence coefficients describes the relation between the 3 unbalances in the planes  $k$

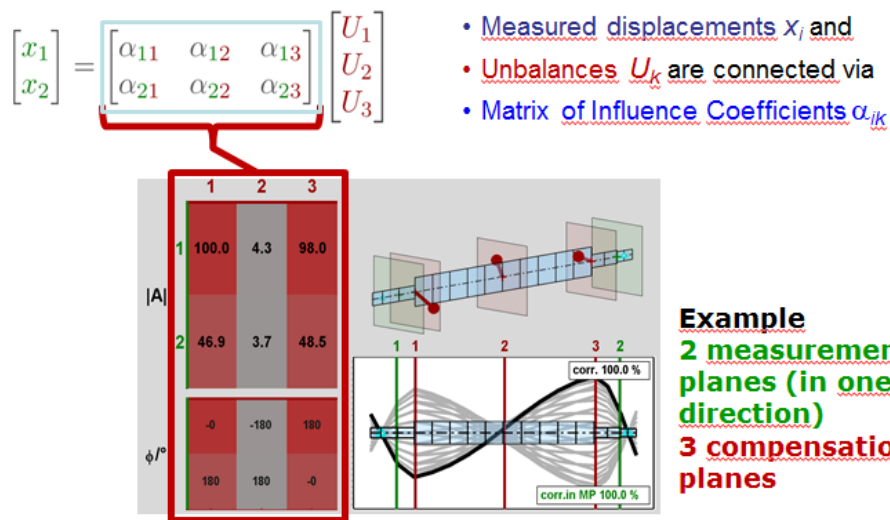


Figure 11 Matrix of Influence coefficients for a simplified rotor in two bearings.

In large power plants (e.g. Forsmark in turbines) there may be 40 transducers and 10-12 planes to build up the matrix of Influence coefficients and the 2 measured displacements in planes  $i$ . If the matrix of influence coefficients (amplitudes and phase) is known – determined by measurements in test runs or by numerical calculations – the shown relation can then be used to determine the unbalances acting in the compensation planes, based on measured displacements at a specific running condition with speed  $\Omega$ . Finally the determined unbalances determine the set of compensation unbalances in the 3 planes in order to balance the rotor. In the practical application some more details have to be known, which are not further described in this report. The described procedure is applicable to balance part rotors in a spin pit, but can also be used for on site balancing in the power plants, when the corresponding shaft train influence coefficients are available.

**Recommendation:** To determine the influence coefficients of a shaft train in a power plant can be very time consuming. However, due to the fact, that modern numerical FE-models are well developed and consider all important effects (e.g.

Bearing and Foundation dynamics) numerical procedures can also be used to determine the needed sets of influence coefficients very accurate and fast.

Figure 12 shows a LP steam turbine rotor of a shaft train running with 1500 rpm in preparation for balancing in a spin pit. On the right side of the Figure the location for balancing weights in a circumferential groove close to the last stage blades can be seen. Other weights, that are used on site are screw weights in numbered holes in each end of the rotor.

As a result of the balancing of the LP steam turbine rotor the vibration amplitudes in  $\mu\text{m}$  (0-peak) are presented as a run up curve (0 – 1500 rpm) in Figure 13. The amplitudes of the relative shaft vibrations at the bearings are less than  $7 \mu\text{m}$  at the critical speed of 600 rpm. At operational speed (1500 rpm) the amplitudes are about  $2 \mu\text{m}$

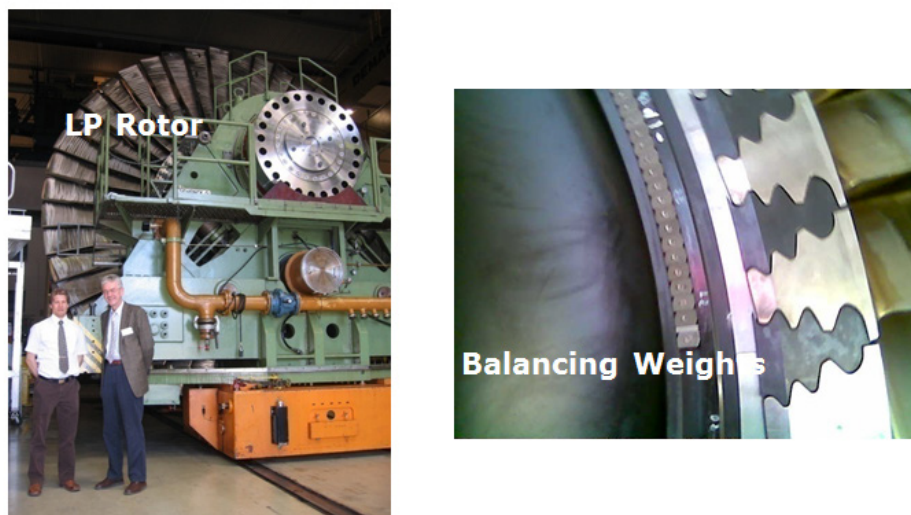


Figure 12 Preparation of a LP Steam turbine rotor for balancing in a spin pit

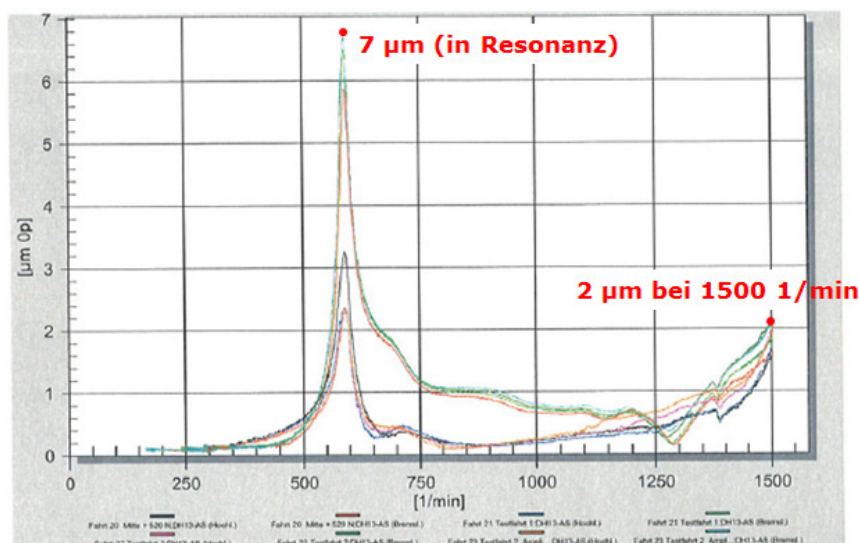


Figure 13 Vibration amplitudes (0-peak) of a LP steam turbine after balancing in the spin pit

## 5.2 CYCLIC OR SPIRAL LATERAL VIBRATION

### 5.2.1 Physical Description

Cyclic Vibrations occur due to weak rubbing between rotor and stator parts (e.g. brushes in slip ring units, bearings, labyrinth and oil seals). Local heating leads to a thermal unbalance which is superimposed to the mechanical unbalance. The resulting unbalance vector rotates. Cyclic vibrations can be stable or unstable with increasing amplitudes. The occurrence of this vibration phenomena depends on the dynamics of the rotor system – natural frequencies and damping – and on the ratio of heat input to heat output at the location of the weak rubbing.

Hot Spot = local rubbing point on the shaft,  
local heat input, thermal deflection, thermal unbalance,  
superposition of unbalances  
leads to Vector rotation

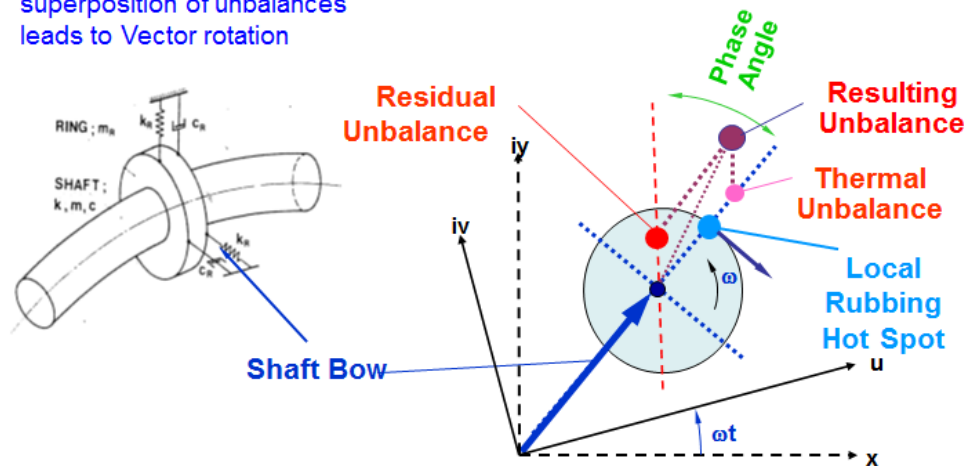
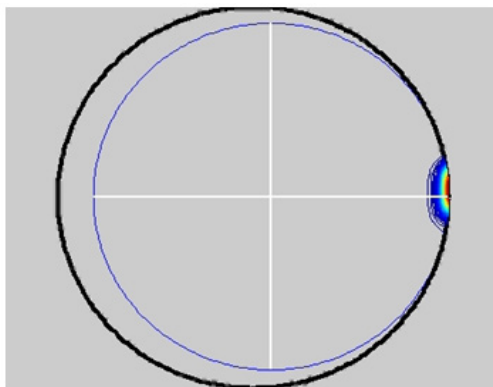


Figure 14 Physical description of cyclic vibrations



When the rotational frequency of the rotor coincides with the frequency of the rotor (shaft) vibration, rubbing will always appear at the same location of the rotor, which is the hot spot. This is relevant in case of unbalance vibrations. Due to the (rubbing) heat input at the same location the rotor will bow (Thermal bow). This thermal bow is equivalent to a mechanical unbalance.

Figure 15 Demonstration of the hot spot due to rubbing between rotor and stator (Seals, bearings, brushes).  
Local heat input causes a thermal bow or thermal unbalance

A hot spot may also occur due to friction in fluid bearings or other fluid filled clearances (e.g. in oil sealing ring systems of hydrogen cooled generators). Figure 16 demonstrates the hot spot in a narrow gap with higher friction due to fluid viscosity between the rotor and the stator. This phenomenon was observed by Kellenberger and Morton. Therefore it is sometimes named as the Kellenberger-Morton effect.

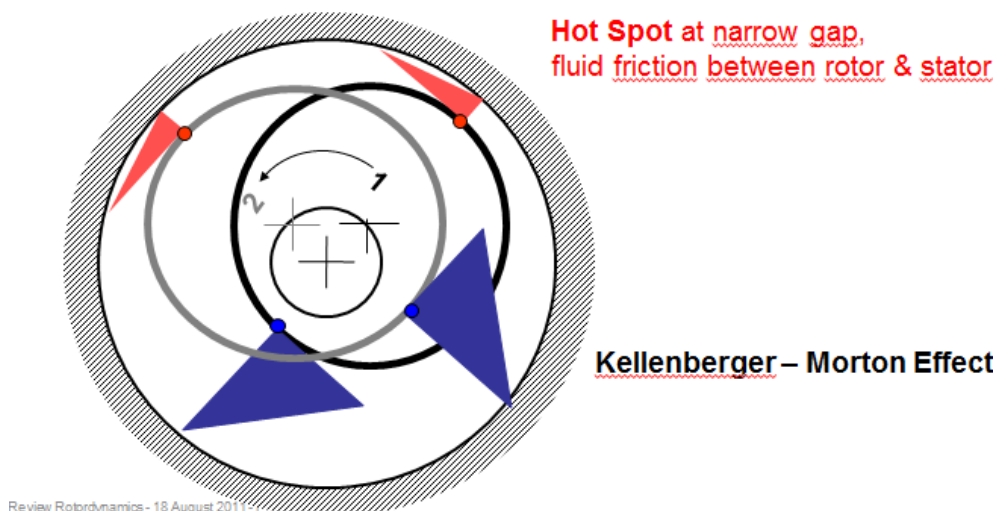


Figure 16 Hot Spot due to fluid friction between rotor and stator in bearings or other fluid filled clearances – Kellenberger-Morton effect.

### 5.2.2 Investigation Techniques – Vibration Problem Analysis

In the power plant 1-X shaft vibrations are permanently measured at the different transducer locations (Figure 17). The time signals can be transformed into the

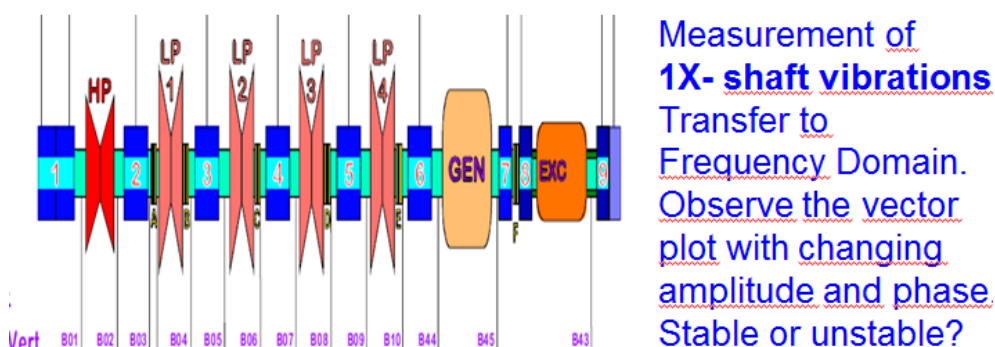


Figure 17 Experimental investigation of cyclic vibrations with input from measured transducer time signals and signal processing into the frequency domain.

The vector plot (amplitude and phase) can be observed, looking for changing amplitudes and phase. In this way possible cyclic vibrations can be identified.

In the design process of a machine and in case of cyclic vibration problems during the operation a numerical investigation is possible as well. Rotordynamic equations of motion with mechanical unbalance and superimposed thermal unbalance – due to rubbing – determine the vibration response. The thermal unbalance is determined by the thermal deformation of the shaft at the location of the Hot Spot. And this thermal deformation is a function of heat input  $p$  and the heat output  $q$  to the shaft. These relations are described by a special thermal equation. Depending on the rubbing mechanism different heat input models exist.

#### Rotordynamic equations– including a Thermal Bent

$$\mathbf{M}\ddot{\mathbf{x}} + (\mathbf{D} + \mathbf{G})\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} - \mathbf{K}^R \mathbf{x}_T = \mathbf{f}(t)$$

Stiffness matrix free-free rotor

Thermal deformation

#### Thermal Deformation $\mathbf{x}_T$ of the shaft

$$\mathbf{x}_T = \mathbf{T} \mathbf{x}_{T,HS}$$

$\mathbf{T}$  - Thermal deformation matrix

$$\mathbf{x}_{T,HS} = (\mathbf{x}_{T,h}, \mathbf{x}_{T,v})^T$$

$\mathbf{x}_{T,HS}$  - Thermal deformation at hot spot

Figure 18 Numerical investigation of cyclic vibrations with a FE Rotordynamic model including a thermal equation for the heat input/output at location of the Hot Spot

In case of cyclic vibration problems further investigations can be recommended:

- To study the heat input/output relations together with the manufacturer,
- to investigate the static shaft center lines for various conditions,
- to adjust the seal positions relative to the static shaft centerline,
- to check the seal clearances,
- to study the stability diagram for possible changes.

In Forsmark all steam seals are evaluated one seal at a time for example LP1 TS vert and LP1 horiz together with 2 vibration channels that show the increase of vibration, variation and rub. Evaluation of condenser pressure and load also with time. One can with this evaluation get help to understand which seal need to be moved, in which direction and how much.

### 5.2.3 Observed Cyclic Vibrations in Power Plants

The following cyclic vibrations have been observed in the three power plants:

**Olkiluoto:** Cyclic vibrations at run up after revision R 108, 29.05.2008

**TVO/OL1:** Strong increase of 1X vibration amplitudes up to 300  $\mu\text{m}$  at 1800 rpm, observed at the two bearing locations B01 and B02 of the High Pressure Turbine. The phase changes as well (see Figure 19). This event was probably no resonance but an unstable cyclic vibration. No clear indication about the location of hot spot, e.g. labyrinth seals? From the shown diagram the time period cannot be estimated.

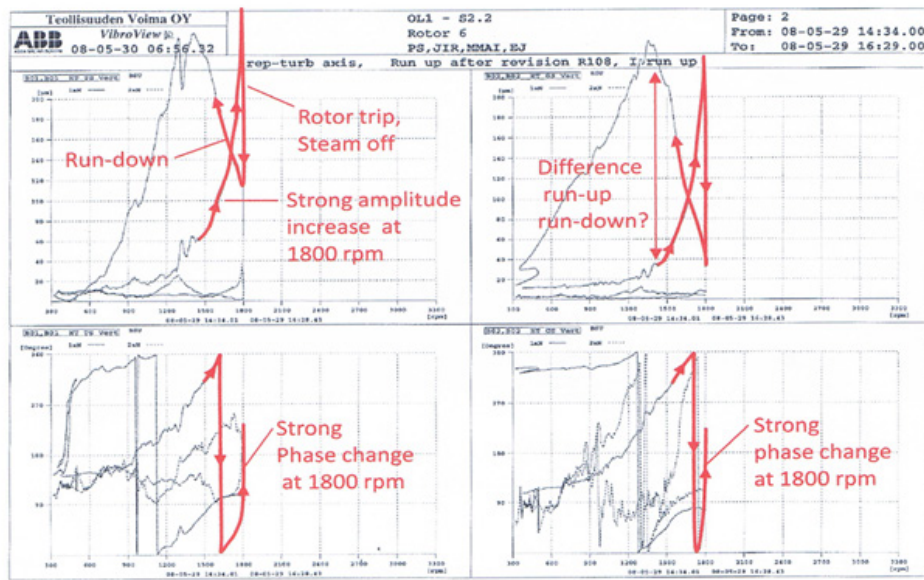


Figure 19 Cyclic vibrations at TVO/OL1 during run up after revision R 108

**Oskarshamn:** Cyclic Vibrations due to oil seal system in Generator, observed since 2011

**OKG/O3:** Cyclic vibrations have been observed continuously in OKG/O3, especially since the Tilt-Pad-bearings were changed to R1T-bearings. The source of vibration is assumed to be in the oil seal rings in the hydrogen cooled turbogenerator (Figure 20).

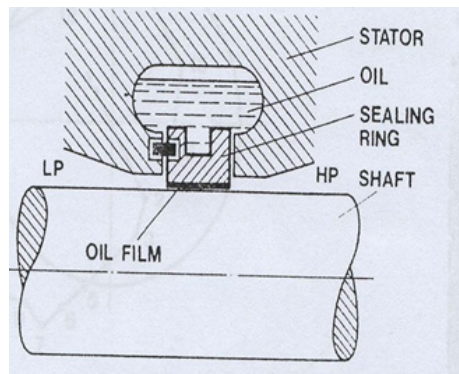


Figure 20 Oil sealing system in hydrogen cooled turbogenerators

It is possible, that the seal was not in an optimal position. Probably the generator steam seals were not adjusted accordingly after the bearing change. The cyclic 1X vibration is initiated in the generator, but can be observed also in the turbines (this depends on the mode shape of the rotor train). The following Figure 21 shows the change of 1X vibrations versus time at transducer location B20 (Generator Bearing, Exciter side). Amplitudes are changing between 10 to 50  $\mu\text{m}$ . The period of one cycle is about 12 hours.

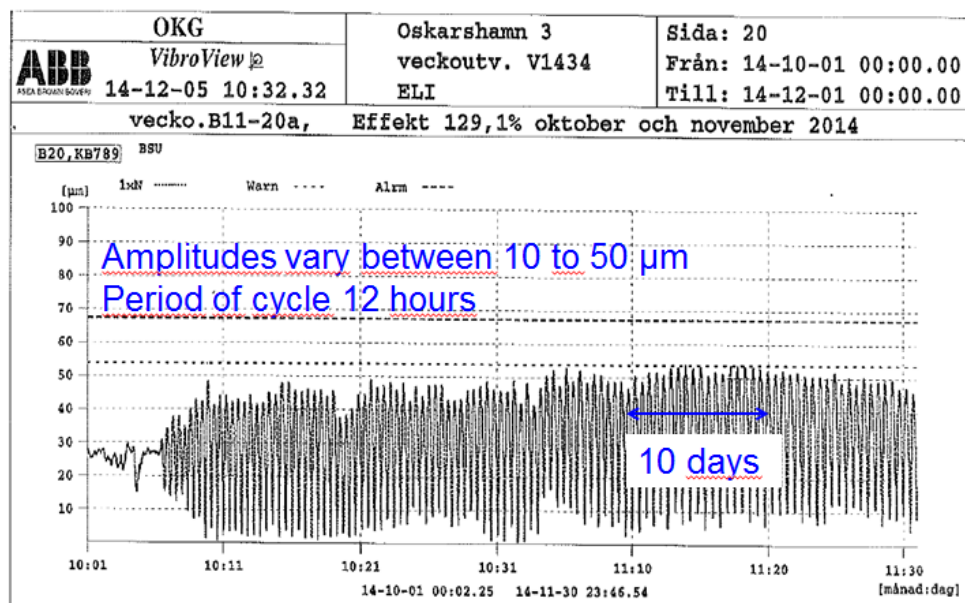


Figure 21 Observed cyclic vibrations in OKG/O3 in 2014

**Trial for explanation:** In gas cooled turbogenerators shaft sealing rings are fitted to prevent the pressurized gas in the stator space from leaking past the shaft into the surroundings. These rings which encircle the shaft are mounted in the stator so that they can follow small excursions but cannot rotate themselves. They slide on the shaft, the eccentricity of which causes them to translate in their stator mountings. The mountings are oil lubricated and their motion in the oil produces a force on the ring, which is transferred to the shaft. The heat arising from the resulting friction loss is partly absorbed into the shaft. The heat supply is not uniform round the shaft circumference and the resulting non uniform temperature rise causes the shaft to bow.

It has also been assumed, that pressure variations in the oil system might be the source of the problem.

**Forsmark:** Cyclic Vibrations during run out test at 500 rpm in FKA/F21, Oct. 2010

**FKA/F21:** Cyclic vibrations were observed in FKA/F21 during runout tests (28.Okt. 2010) at rotational speed 500 rpm in the frame of RA10. High 1X vibrations mainly appeared at transducer location B03(LT1TS) in vertical direction (see Figure 22). The vibration values at B03 reached values up to 100  $\mu\text{m}$ . The period of vector

rotation was one hour. It was assumed, that the origin of the hot spot was at one of the steam seals in the steam turbine LT1.

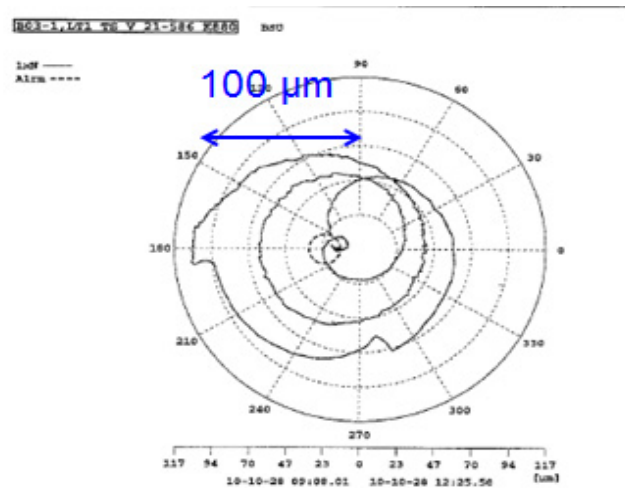


Figure 22 Cyclic vibrations in FKA/F21 at speed 500 rpm during runout tests. The small circle near the origin is the normal alarm circle. Normal alarm radius is 10 µm. Vibration values should be well inside this alarm circle.

#### 5.2.4 Solution – Vibration Problem Mitigation

For the three reported cases of cyclic vibrations the following solutions to mitigate the vibration problem have been applied, respectively can be recommended for a solution.

**Olkiluoto:** Cyclic Vibrations at Run up after revision R 108, 29.05.2008

After trip at high vibration amplitudes, the steam delivery was stopped and the vibrations dropped down. The event did not occur a second time.

**Oskarshamn:** Cyclic Vibrations due to oil seal system in Generator, observed since 2011

Avoid rubbing by adjusting the oil seal ring position, consider the static shaft line of the turbine train. Pressure control in the oil system helps to reduce the cyclic vibrations. Discuss with manufacturer the design and mechanism of the oil sealing system. Estimation of heat input and heat output.

Which mode shape is relevant? Improve damping in bearings.

Inspect the oil sealing system at next suited revision. To avoid trips at the upper level of amplitudes, carefully increase temporary the alarm limits.

**Forsmark:** Cyclic Vibrations during run out test at 500 rpm in FKA/F21, Oct. 2010

Avoid rubbing in steam seals by adjusting the seal positions relative to the static centerline of the shaft train. Improve damping in the rotor system. Inspect the steam seals at next revision.

Various test runs with different speeds 400 – 600 rpm resulted finally in a stable condition (e.g. test K 1:3). However problems occurred again with F21 in 2012.

### 5.3 FRICTION INDUCED LATERAL VIBRATIONS IN GENERATORS

#### 5.3.1 Physical Description

In Generators (see cross section in Figure 23) Friction Induced Lateral Vibrations can be caused by unsymmetrically distributed friction forces in circumference direction between the copper windings and the generator steel rotor. In case of this unsymmetrically force distribution a resultant moment may lead to a bow of the rotor. Such a friction induced bow is again a special case of unbalance, leading to 1X-Lateral Vibrations.

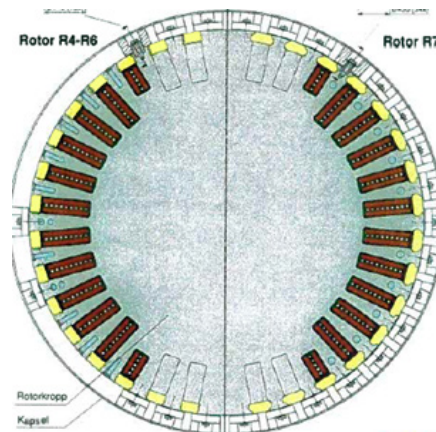


Figure 23 Cross section of a two pole turbogenerator with steel rotor and copper windings

Figure 24 shows the normal forces between the winding and the rotor. They depend on the temperature differences  $\Delta T_1$  (relative temperature between rotor and copper to the environment) and  $\Delta T_2$  (relative temperature between copper and steel). Furthermore they depend on the mounting pressure  $p$  and on the rotational speed  $\Omega$  (centrifugal forces).

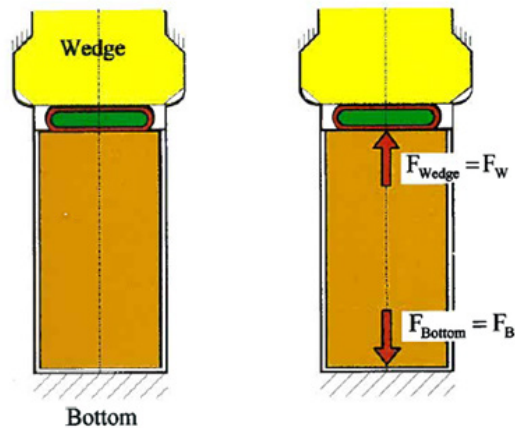


Figure 24 Normal contact forces between copper winding and steel rotor

The normal forces at the bottom and at the wedge of each slot can be obtained by superposition of the normal forces due to the mounting pressure, due to the temperature differences and due to the rotational speed. From the contact forces between the copper windings and the steel rotor the friction forces between these components can be derived, when the friction coefficient for the copper-steel surface is known.

For the determination of these friction forces it has first to be considered how the different materials (copper and steel) would be elongated due to the temperature effects  $\Delta T_1$  and  $\Delta T_2$ . Copper and steel have different heat thermal expansion coefficients. Figure 25 shows the free difference elongation  $\Delta l$  (no friction) between copper and steel in dependence of the two temperature differences  $\Delta T_1$  and  $\Delta T_2$ .

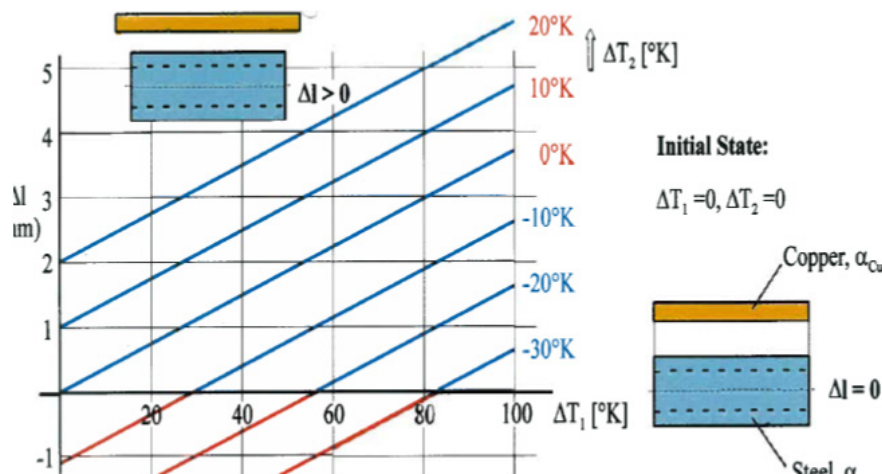


Figure 25 Difference elongation  $\Delta l$  between copper and steel in dependence of the temperature differences  $\Delta T_1$  and  $\Delta T_2$ . Free expansion without friction

The difference expansion can be avoided, when the friction forces between the two materials are strong enough (Figure 26).

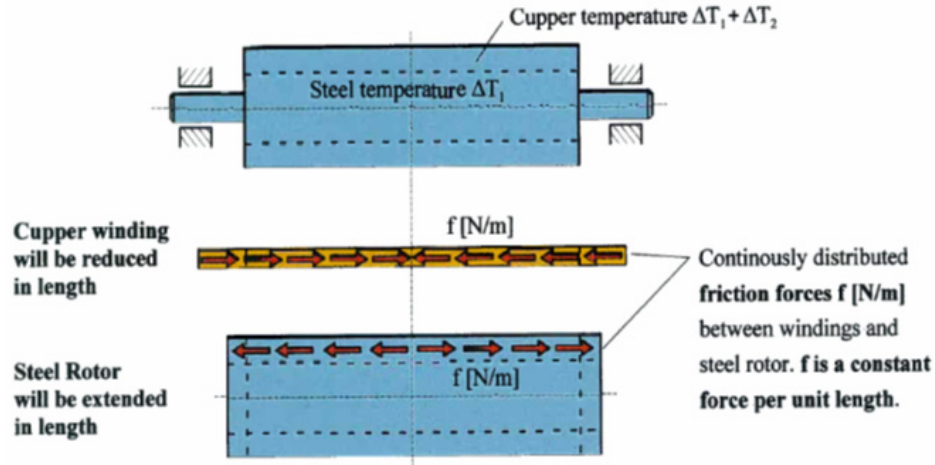


Figure 26 Friction Forces or compensation forces  $f$  N/m between copper and steel

The necessary compensation force (force per unit length) to avoid sliding is shown in Figure 27 as function of  $\Delta l$ . The force also depends on Young's modulus  $E$  of copper, the cross section area  $A$  and the length  $l$  (see Figure 27).

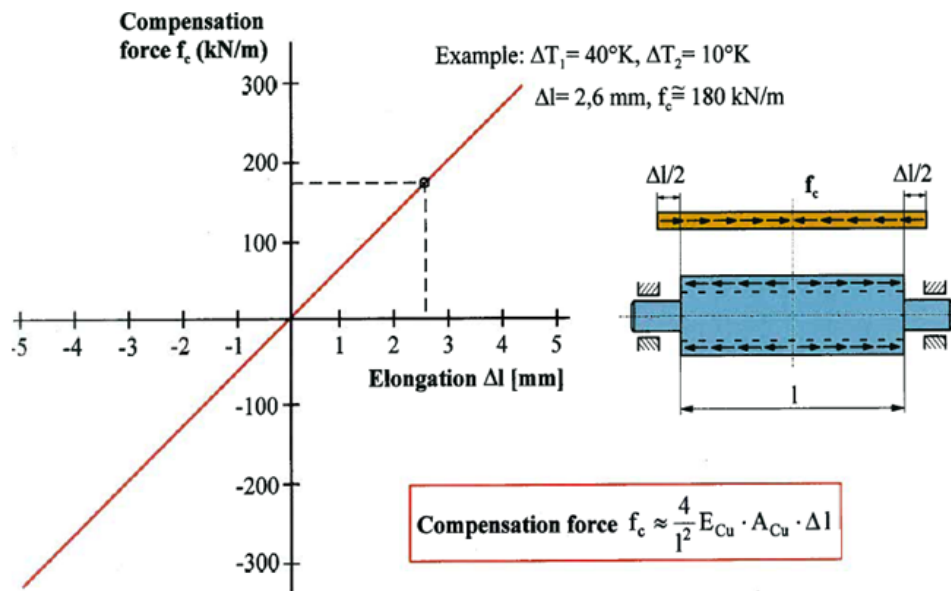
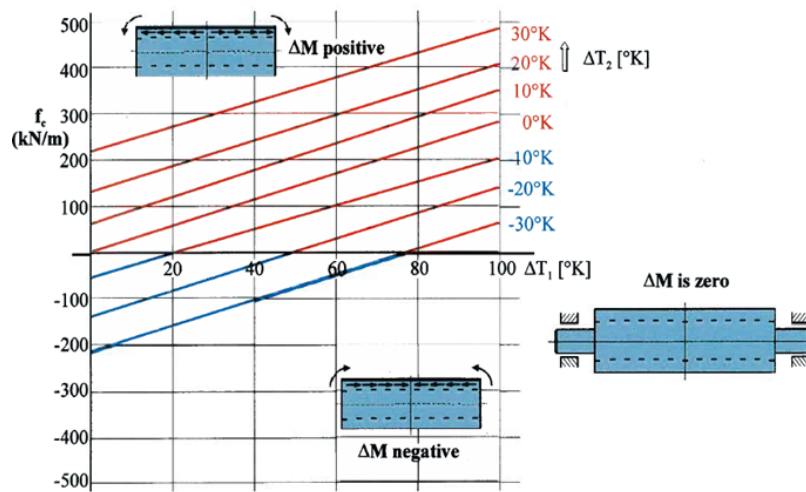


Figure 27 Necessary compensation forces in dependence of  $\Delta l$  to avoid relative movement (sliding) between the two materials copper and steel

The compensation force  $f(\Delta l)$  can also be presented as a function of  $\Delta T_1$  and  $\Delta T_2$ , as it is shown in Figure 28. This is related to Figure 25.



**Figure 28** Necessary compensation force in dependence of  $\Delta T_1$  and  $\Delta T_2$  to avoid a relative movement (sliding) between the two materials copper and steel

As long as the necessary compensation force for a special temperature condition  $\Delta T_1$  and  $\Delta T_2$  is smaller than the limit friction force (friction coefficient  $\times$  normal force) no sliding will occur.

Besides the necessary compensation force for given temperature differences  $\Delta T_1$  and  $\Delta T_2$  we now consider also the limit case, where the compensation force becomes equal to the limit friction force. This limit cannot be exceeded:

$$\text{Friction force} = \text{friction coefficient} \times \text{normal force} = \text{compensation force}$$

As we have seen before the normal force depends on the rotational speed, the mounting pressure and on the temperature differences  $\Delta T_1$  and  $\Delta T_2$ . Due to this fact also the limit friction force is dependent on these parameters and in addition on the friction coefficient  $\mu$ . In Figure 29 besides the temperature dependent compensation force two limit case curves for the friction force are presented for the two friction coefficients  $\mu = 0,1$  and  $\mu = 0,2$ . A constant temperature difference  $\Delta T_2 = -20^\circ\text{C}$  is assumed. As long as the compensation force is less than the friction limit force sliding is not possible and the two materials copper and steel are fixed together. When the compensation force reaches the limit friction forces we have to switch and to consider the limit friction curve.

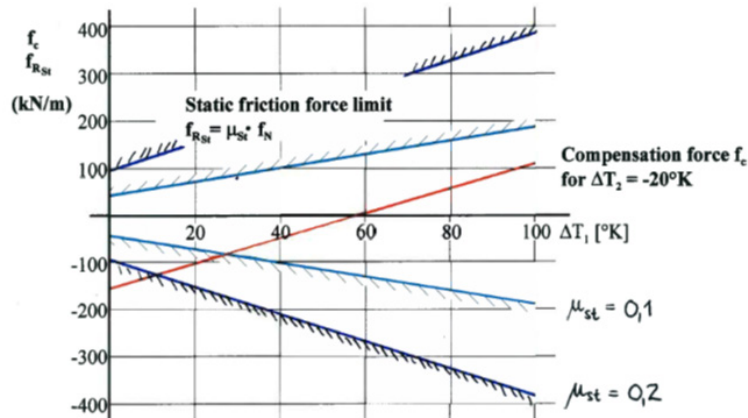


Figure 29 Compensation force and limit friction force in dependence of the temperature difference  $\Delta T_1$ .  $\Delta T_2$  is considered to be constant in Fig. 29. For the limit friction force two friction coefficients  $\mu = 0,1$  and  $0,2$  are considered.

### 5.3.2 Investigation Techniques – Vibration Problem Analysis

We analyze a two pole generator rotor with a cross section shown in Figure 30. We assume a constant speed of  $n = 700$  rpm and a constant temperature difference  $\Delta T_1$ . Young's moduli and the cross section areas are also considered to be constant. For the investigation in the following Figure 31 the temperature difference  $\Delta T_2$  can be varied.

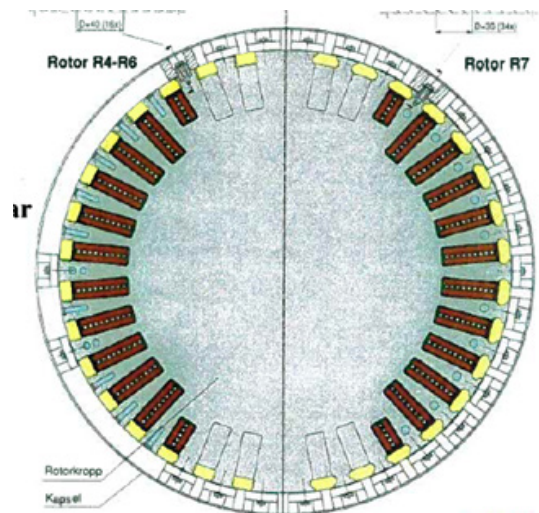


Figure 30 Investigation of Friction induced Lateral Vibration for a two pole turbogenerator with unsymmetrically distributed forces in circumference direction Speed  $n = 700$  rpm. Temperature difference  $\Delta T_1 = \text{constant}$ .

Figure 31 shows the compensation force in dependence of the temperature difference  $\Delta T_2$ . Also presented are two curves for the limit friction forces, one of them may be for the angle  $90^\circ$  and the other one for  $270^\circ$ . Possible reasons for the difference in these two curves can for example be: different mounting pressure for

90 and 270° and due to that different normal forces, different friction coefficients  $\mu$  for 90° and 270°, and due to that different friction forces

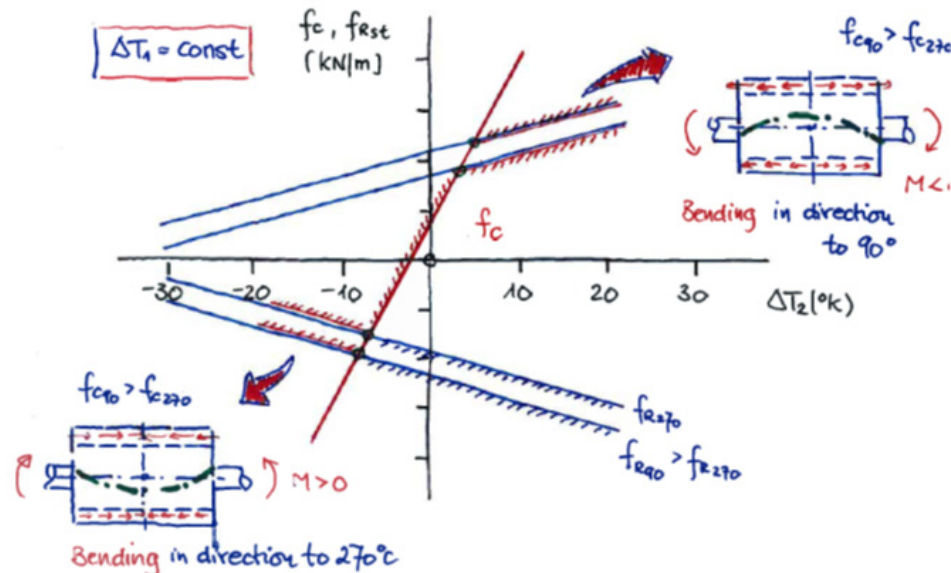


Figure 31 Compensation force and limit friction force as function of  $\Delta T_2$ . The following parameters are considered to be constant: speed  $n$ , temperature difference  $\Delta T_1$ , Youngs moduli  $E$ , Cross section areas  $A$ .

**Question:** In which cases Friction induced bending may appear? To answer this question we simplify the explanation for the two slots at 90° and 270° in Figure 31.

**Bending (bow) due to compensation force:** If the limit friction force is still not reached the compensation forces at 90° and 270° can be different due to different temperature differences  $\Delta T_2$  in these two slots.

**Bending (bow) due to limit friction forces:** The two limit friction forces in 90° and 270° may be different due to different friction coefficients in the two slots. Different mounting pressures can also be the reason (normal forces are different)

If we consider compensation forces or limit friction forces in one slot only (e.g 90°), the shaft will experience a bow (see Figure 32). If the force would appear in the same size also in 270° the overall bow would be zero. Only in case of different forces in 90° and 270° a bow will be seen.

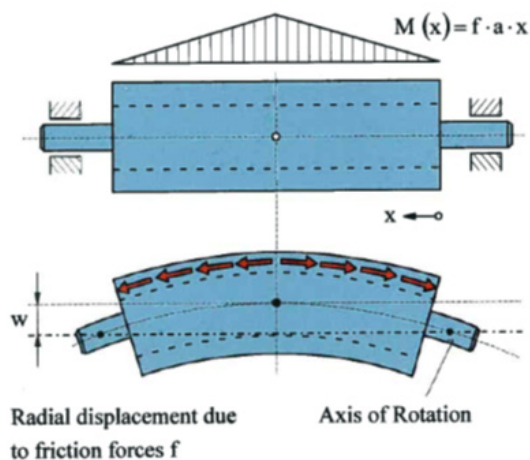


Figure 32 Friction induced bow (bending) in a two pole turbogenerator

### 5.3.3 Observed Friction Induced Lateral Vibrations in Power Plants

Figures 33 shows the protocol of a run up procedure in OL1 in the power plant Olkiluoto. Before running through the first critical of the generator at 600 to 700 rpm a vibration check was performed. The vibration readings at bearing location B44 and B45 contain information about the size of a possible bow. In case, that the vibration values are too high, the bow is controlled by changing the rotor temperature.

Figure 34 demonstrates the procedure. It starts with a first run up to 500 – 600 rpm, checking the vibration amplitudes at the bearing locations. By changing the rotor temperature + 3°C the vibration values were reduced and the following run up through the resonance was possible

Table 5. Runup log, VibroCam times.

| TIME               | RPM  | DESCRIPTION                                                                                                                                                                                                                                                                               |
|--------------------|------|-------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| 24.2.2015<br>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 33 Protocol for a run up of TVO/OL1 to pass the first generator critical

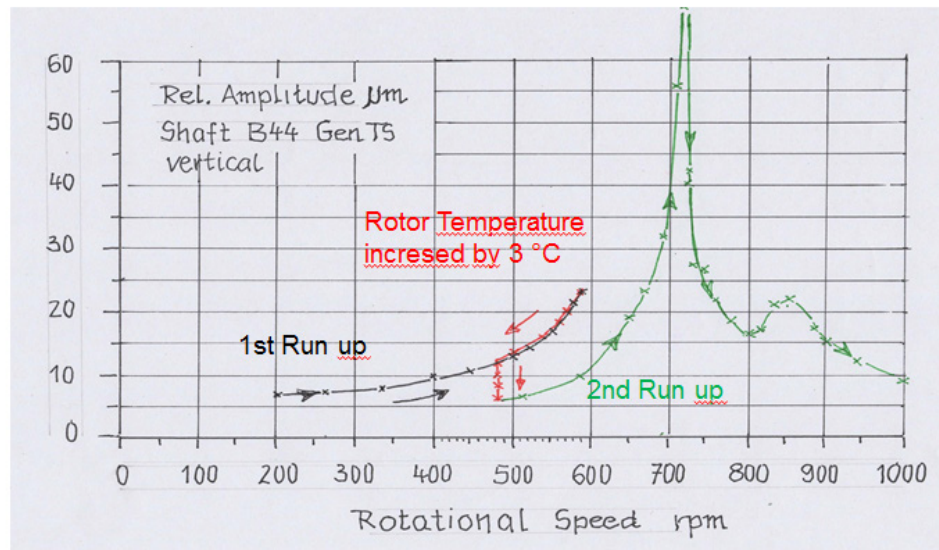


Figure 34 Run up of unit TVO/OL1 with vibration control via the rotor temperature

#### 5.3.4 Solution – Vibration Problem Mitigation

Avoid unsymmetry in circumferential direction, e.g. friction coefficients, mounting pressure, temperature. If possible, control temperature  $\Delta T_2$  by cooling or heating the winding. Testing of generator system by manufacturer, e.g. by Heat Runs.

### 5.4 LATERAL VIBRATIONS DUE TO CHANGES IN SEAWATER TEMPERATURE

#### 5.4.1 Physical Description

In the condenser of a power plant unit the steam from the LP turbines is cooled down and turned back to water. For this purpose seawater is used as cooling water (see Figures 35 and 36). When the seawater temperature is lower in winter, the temperature in the condenser becomes lower as well and with that the condenser and LP turbine pressure is also decreasing. This leads to a higher power output because of the higher steam pressure difference  $\Delta p$  in the unit. The changing seawater temperature influences the condenser vacuum. Therefore seawater temperature and the condenser vacuum both influence the lateral vibrations of a rotor train. When the load (MW) changes, the condenser vacuum changes as well.

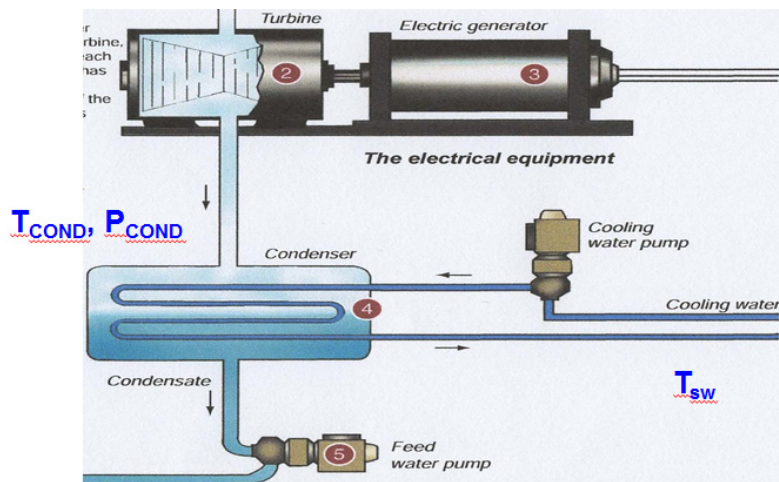


Figure 35 Cooling down of the steam of LP Turbines by sea water

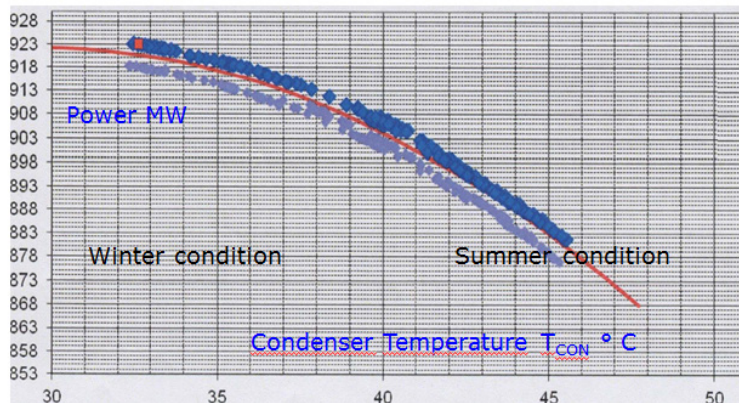


Figure 36 Active Power as a function of the condenser temperature

#### 5.4.2 Investigation Techniques – Vibration Problem Analysis

The lateral vibrations due to changes of the sea water temperature can best be investigated by measuring the relative shaft vibrations and the absolute bearing vibrations, respectively as described in 5.1.2. Different signal processing procedures and signal presentations may follow. Forsmark reported, if there is a rubbing the quickest way to change the situation and decrease the vibration is to change the load MW, but sometime it was not possible to change the load and then one can change the vacuum by for example stopping 1 or 2 cooling water pumps below the turbine/condenser.

#### 5.4.3 Observed Lateral Vibrations due to changes in Seawater Temperature

The following lateral vibrations due to Sea water changes have been observed in the three power plants:

**Oskarshamn:** Water falling down to the shaft, which leads to a shaft bow

OKG/O3: When the cooling water temperature becomes higher than 10 – 14 °C, the percentage of water in the LP turbine increases. Usually sheet-metals are used to avoid, that the high water content falls down to the shaft. It seems that water came down to the shaft in the O3 unit, leading to a thermal shaft bow with increasing 1X unbalance vibration. A current hypothesis from EDF, based on a numerical CFD study, is shown in Figure 37.

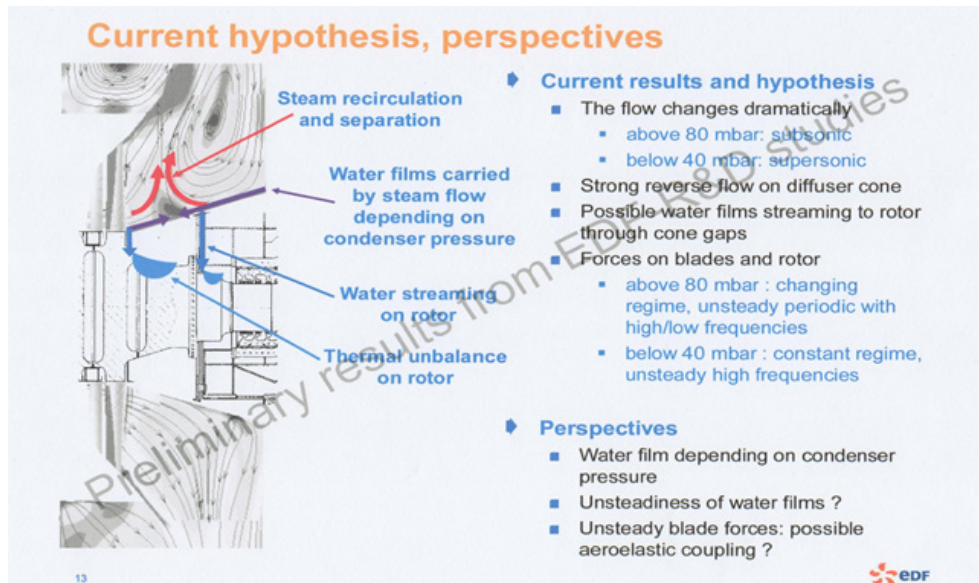


Figure 37 Hypothesis from EDF for Thermal Bow due to water falling down to the shaft

Figure 38 shows vibration velocities in mm/s for two bearings in OKG/O3 in Sept. 2014. The average vibration values follow the condenser temperature and pressure curves, e.g. decreasing vibrations with decreasing temperature and pressure and vice versa. This confirms the hypothesis of a thermal bow due to water streaming on the rotor. The superimposed cyclic vibration (first half of September) was assumed to have its origin in the oil seal system of the generator (see chapter 5.2.3). The period of one cycle is again about 12 hours. It seems, that also the cyclic part is influenced by condenser pressure and temperature. With increasing values the cyclic part subsides, but comes back with decreasing values (beginning of October). The relation is not quite clear and has to be clarified by further investigations. Of interest is also the influence of the power, where lower power has a stabilizing effect. The influence of the oil pump system needs also further clarification.

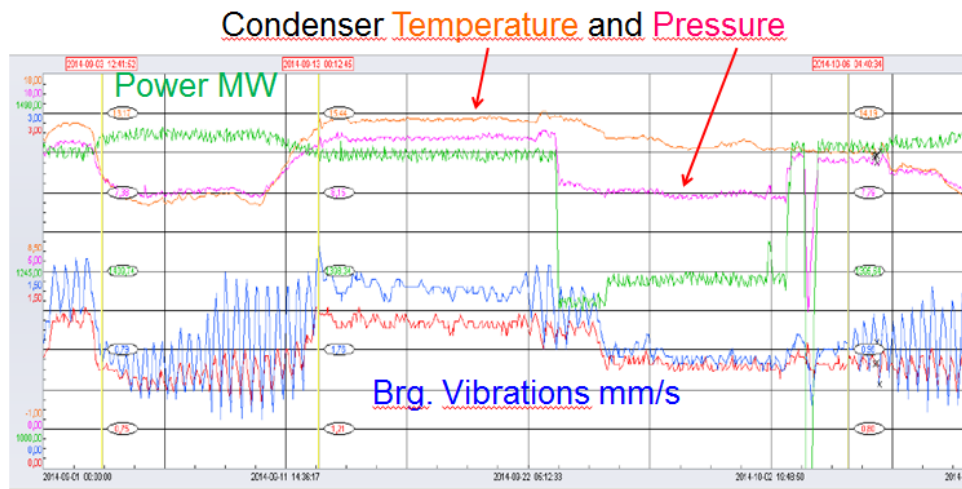


Figure 38 Bearing vibrations in OKG/O3 in relation to condenser temperature and pressure and in relation to power.

**Olkiluoto:** Changing lateral vibrations due to condenser deformation

TVO/OL1&OL2: Figure 39 shows the arrangement of the OL1/OL2 units with rotor train, foundation and condenser. Bearings 3 and 5 are mounted on the condenser, other bearings on the foundation. In case of changes of the seawater temperature the condenser temperature and pressure change as well, leading to a relative thermal condenser deformation. Due to this bearings 3 and 5 are moving relative to the shaft train and the static bearing forces will be different. The static bearing forces have an influence on the dynamic characteristics of the oil film. The vibrations of the shaft train will therefore change. Due to the thermal deformations it can also be assumed, that the static centerline of the shaft is changing.

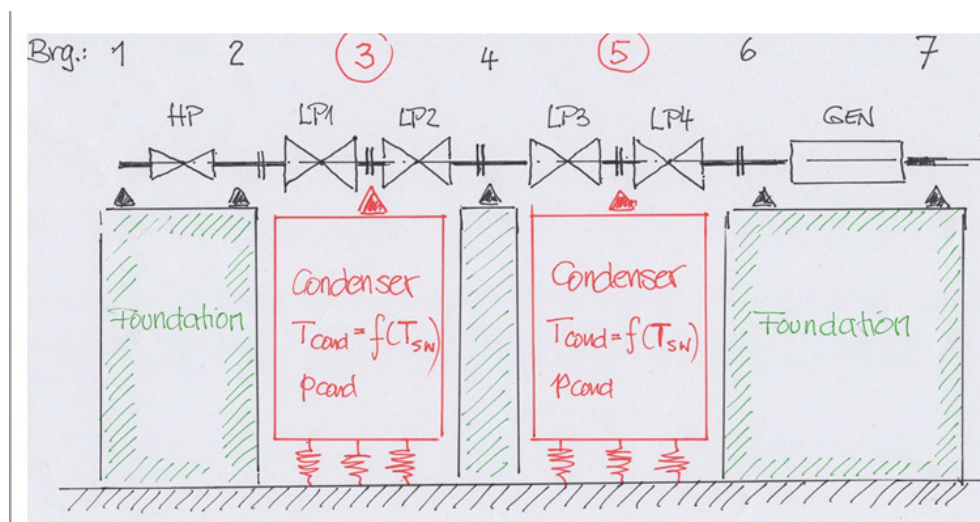


Figure 39 Arrangement of rotor, foundation and condenser in units OL1 and OL2

Figure 40 shows the seawater temperature versus time from 06/2012 – 05/2013 for the Olkiluoto plant. The variation is between 02°C and 22°C. The shaft vibrations of the High Pressure Turbine (as example) in the same period are between values of 30 to 42  $\mu\text{m}$ . These changes are not very high and are well known by the plant vibration engineers in dependence of the sea water temperature. Effects like in Oskarshamn (water fall down to the shaft) have not been observed in Olkiluoto.

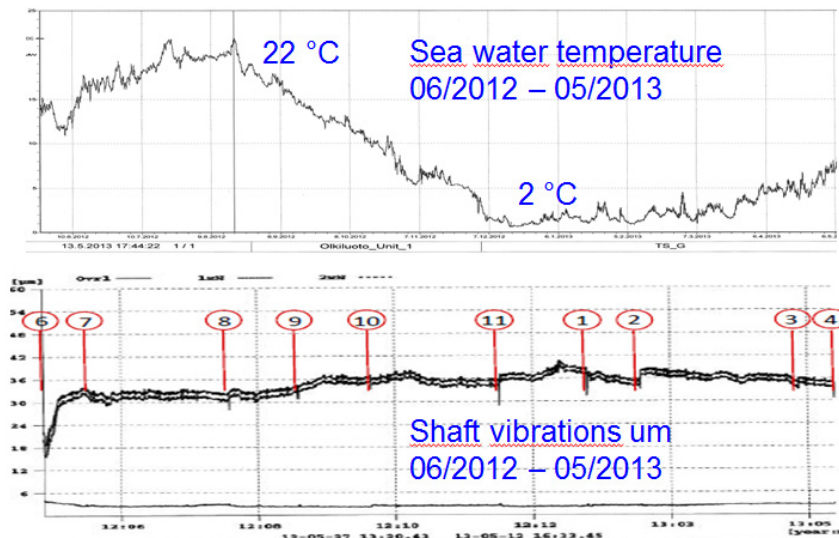


Figure 40 Influence of the sea water temperature on the shaft vibrations of OL1/OL2

**Forsmark:** Influence of sea water temperature on bearing vibrations of LT3 in F3

FKG/F3: The influence of the seawater temperature on lateral vibrations has also been observed in Forsmark at unit F3. Figure 41 shows the change of the sea water temperature at summer time and the increasing power versus time until full load. Figure 42 presents amplitude and phase of the horizontal relative shaft vibrations at the LT3 bearing (GS). For low power the 1X lateral vibrations are low between 10–20  $\mu\text{m}$  but increase to higher values up to 55  $\mu\text{m}$  at full load. In this state the vibration show cyclic characteristic with amplitudes changing between 5 and 55  $\mu\text{m}$ . The period of the cycle is 24 hours. In this case higher vibrations follow higher seawater temperatures. Seal rubbing is assumed by plant.

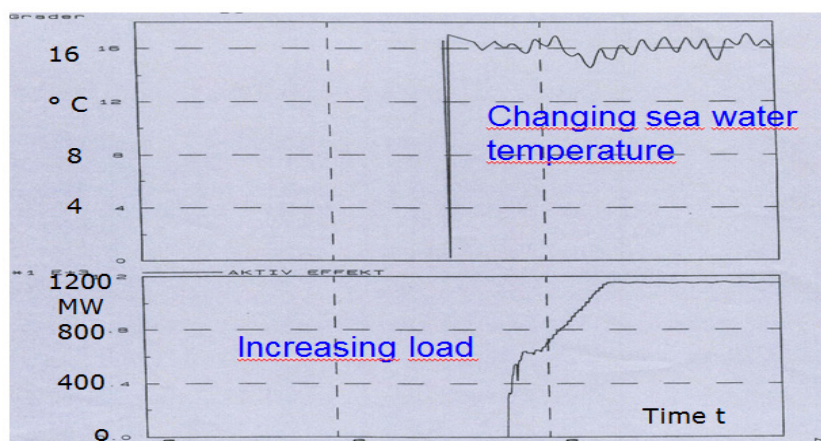


Figure 41 Changing sea water temperature and increasing load in FKG/F3 unit

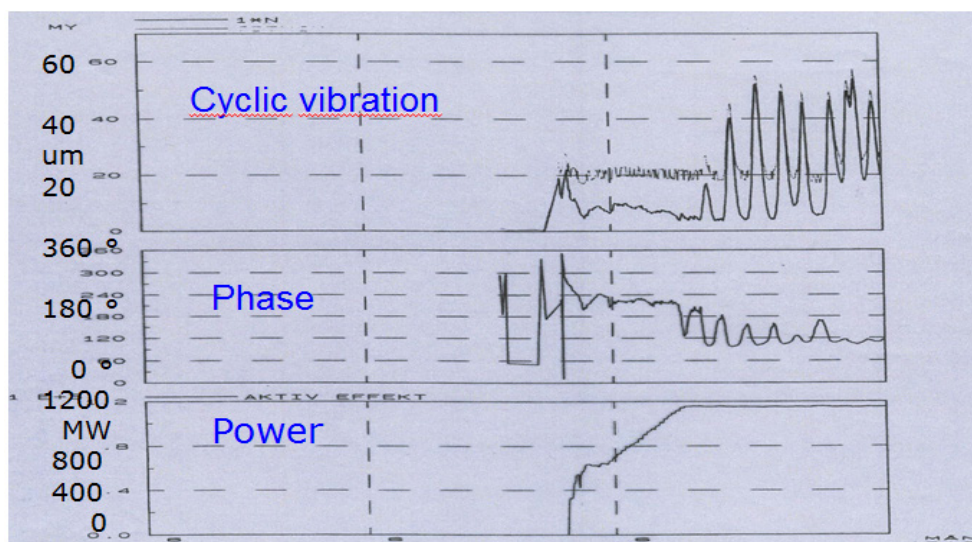


Figure 42 Lateral vibrations due to changing sea water temperature in FKG/F3

#### 5.4.4 Solution – Vibration Problem Mitigation

Avoid water film streaming down on the LP-rotor by protecting the rotor with sheet metal (sealing). To clarify, why the water on rotor leads to a thermal bow (hypothesis EDF). Reduce active Power. Avoid rubbing in steam seals by alignment of static rotor deformation after revision. Check influences of different process parameters, e.g. condenser temperature and pressure, active power, on the static movement of the shaft axis relative to the seals. Clarification with manufacturer about oil seal system design. Reduce heat input in oil seal system.

## 5.5 LATERAL VIBRATIONS DUE TO UNEQUAL MOMENTS OF INERTIA

### 5.5.1 Physical Description

Due to different stiffness values of the turbogenerator rotor in two directions (Figure 43), we expect two natural frequencies or two critical speeds. Furthermore, when the shaft rotates the weight of the disk is lifted two times per revolution (see Figure 44). Therefore, for the rotational frequency  $\Omega$  a frequency of excitation  $2\Omega$  is expected. When this frequency of excitation  $2\Omega$  is equal to a natural frequency of the turbogenerator rotor the so called weight resonance appears. Between the two critical speeds instability may occur.



Figure 43 Two pole turbogenerator rotor with Unequal Moments of Inertia

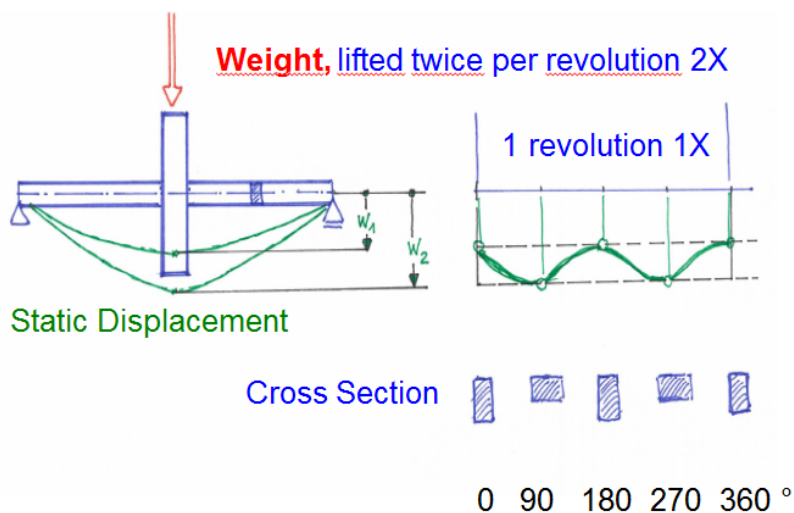


Figure 44 Static displacement of a shaft per revolution, when the shaft has unequal moments of inertia

Due to the unequal moments of inertia a rotor will have the shown dynamic behavior, when running up (see Figure 45). The run up curve shows amplitudes versus rotational speed with a weight resonance (green), two resonances due to unbalance excitation (yellow) and a red unstable area between these resonances. However instability will only occur, if the system damping is too small. As usual the unbalance vibration has a 1X component (rotational frequency), while the weight excited vibration has a 2X- component, as explained before in Figure 44.

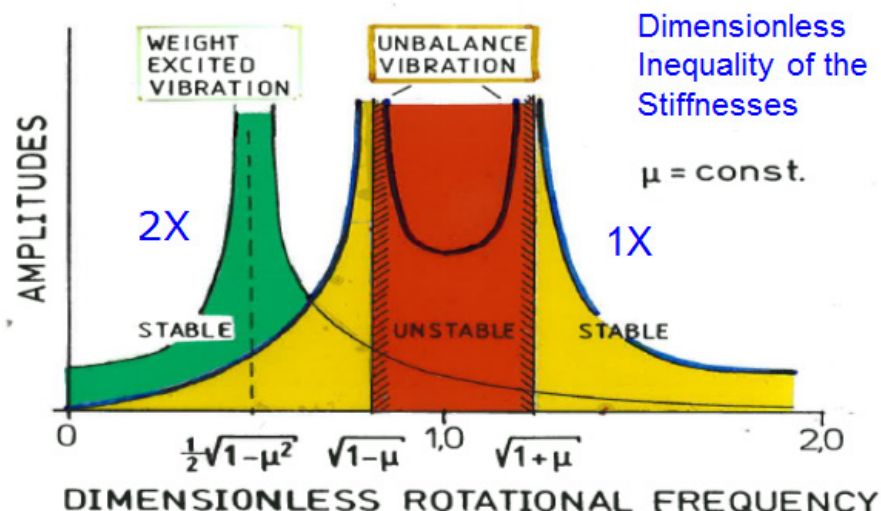


Figure 45 Runup of a simplified Jeffcott rotor with unequal moments of inertia

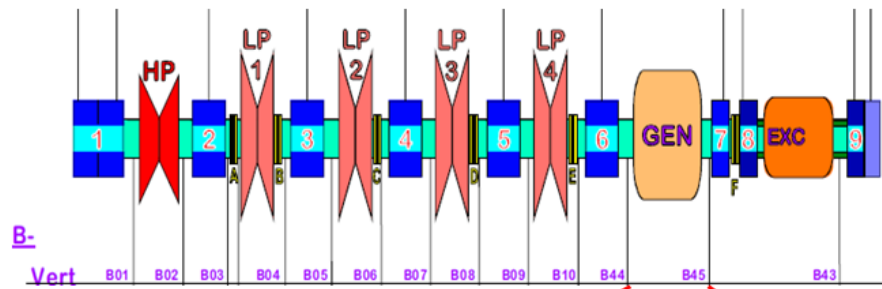
Note that 2X vibrations on the Generator rotor also can be excited by electromagnetic forces. This was often observed on the water cooled generators in Forsmark 1 and 2. It can clearly be seen when the machines are synchronized.

### 5.5.2 Investigation Techniques – Vibration Problem Analysis

Take vibration signals from transducers and analyze 1X and 2X components in dependence of the rotational speed  $\Omega$ , especially in the speed range from about 0.5 to 1.0 of the 1st generator bending natural frequency. The 1st generator bending natural frequency is usually in the range between 600 to 800 rpm, in the following example 720 rpm.

### 5.5.3 Observed Lateral Vibrations due to Unequal Moments of Inertia in Power Plants

Figure 46 shows the two measurement locations at the generator bearings of unit TVO/OL1, where the 1X and 2X relative shaft vibrations in vertical direction could be observed.



Measurement of relative shaft displacements with transducers B44 (Generator TS, vertical) and B45 (Generator ES, vertical)

Figure 46 Measurement locations at generator bearings in unit TVO/OL1 to observe 1X and 2X relative shaft vibrations (vertical direction).

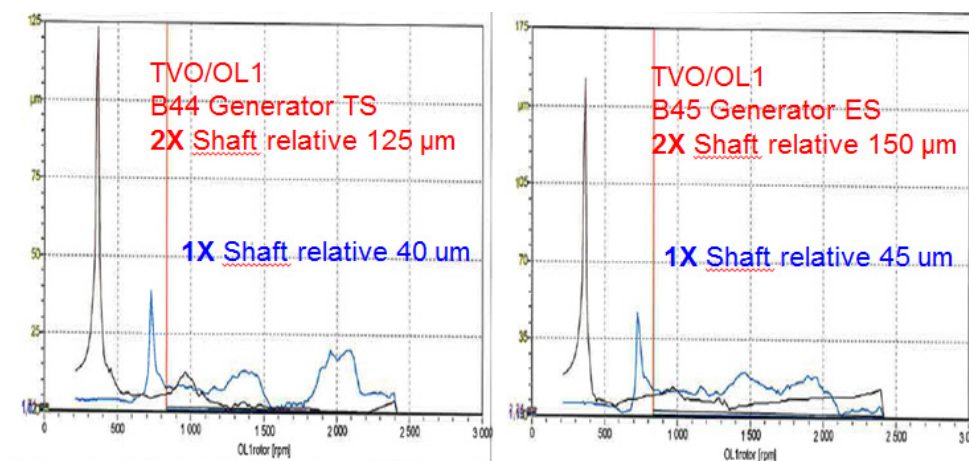


Figure 47 1X and 2X relative shaft vibrations (vertical direction) at generator bearings B44 (TS) and B45 (ES) of unit TVO/OL1

#### 5.5.4 Solution – Vibration Problem Mitigation

With slots in the generator rotor the different moments of inertia can be equalized. With that the effect of the weight resonance can be reduced. Damping also helps to improve the vibration behavior and avoids instability

### 5.6 UNSTABLE LATERAL VIBRATIONS

#### 5.6.1 Physical Description

Instability in steam turbine rotor trains can be caused by fluid structure interaction forces, e.g. in oil film bearings or in labyrinth steam seals. The cross coupled stiffness coefficients of the fluid are the destabilizing parameters, while the main damping coefficients are stabilizing. Unstable lateral vibrations usually appear with one of the lower natural frequencies corresponding to one of the first bending

modes of a steam turbine rotor. Oil film bearing instabilities can be identified by the half frequency whirl.

Other sources of instability are the so called Thomas forces or Alford forces, resulting from unequally distributed steam forces in circumference direction, which are caused by differences in the clearance losses in circumferential direction.

### 5.6.2 Investigation Techniques – Vibration Problem Analysis

The usual investigation technique in the power plant is to measure relative shaft vibrations in the time domain and to transfer them into the frequency domain. In case of instability the frequency spectrum is dominated by one of the bending natural frequencies of the turbine rotors.

Numerical analysis can help to predict possible instabilities by calculating the rotor train complex eigenvalues. The real part of an eigenvalue is a measure for the damping behavior and allows an evaluation of the system stability. The imaginary part is the value of the natural frequency.

### 5.6.3 Observed Unstable Lateral Vibrations in Power Plants

In the first reports from the power plants unstable vibrations could not be found by the author of this report. It was also confirmed in the meetings in Oskarshamn and Olkiluoto, that instability in the turbine trains are no problem in the considered power plants.

Later in the Forsmark meeting the following was reported: We had an instability problem since we changed to new Low Pressure Turbines in 2004. We call the phenomena – “Bearing Instability”. After the meeting on 23. February an E-Mail was sent from Magnus Adolfsson with the following content: “Regarding the behavior at F3 as we can call Bearing instability. That shows in FFT spectra at full load, we can see component in approx. 8 Hz on the proximitor probes (not far away from  $1/2 \times N$ ). 8 Hz is also the frequency for the first mode of the LP train.

The author of this report answered as follows: You assume a bearing instability in F3. However, a bearing instability most often appears closer to half of the running frequency, in our case at 12.5 Hz. Therefore I think that the 8 Hz vibration is something different. We know that the corresponding mode has large displacements in the LP turbines and that the mode has very low damping. One possibility could be, that we have a seal instability in one of the LP turbines or even from the HP turbine, which has also displacement contributions in the 8 Hz mode shape. Such a seal instability can also occur with a frequency unequal to  $1/2 \times N$ , e.g. with 8 Hz. Another possibility could be, that a broadband excitation from the process excites the 8 Hz mode with low damping.

A final response from Forsmark had the following content:

Note that the 8 Hz peak is the first mode of the LP:s affecting the whole rotor train with all rotors from HP – to Generator, although the highest amplitudes during runup are noted on the LP-rotors shaft proximity probes. The new LP-rotors are

much weaker than the original and the bearings that were modified-when the LP:s were exchanged- are now not enough loaded.

The 8 Hz resonance is very undamped. Bearing instability does not have to be exactly half the running speed – if there is a resonance near it can lock to that frequency. As this 8 Hz resonance is so dominating and undamped the bearing instability frequency will be seen at 8 Hz during running. Forsmark and Siemens are quite sure that this is a bearing instability phenomenon.

This vibration problem should further carefully be observed.

#### **5.6.4 Solution – Vibration Problem Mitigation**

In case of instability the source of the unstable vibrations has to be found. Possible measures in the fluid structure mechanism (Oil film bearing, labyrinth seals) have to be investigated: Change of bearing type, change of bearing or seal parameters like clearances, fluid temperatures, fluid viscosity. Increase of damping is usually a good measure to avoid instability.



# LATERAL TURBINE AND GENERATOR VIBRATIONS

The objective of this project is to assemble knowledge and experience in the area of turbine and generator torsional and stator vibration problems from the three Nuclear Power Plants in Sweden (Oskarshamn OKG and Forsmark FKA) and Finland (Olkiluoto TVO) and to study in some depth how vibration problems were investigated and mitigated.

The results are presented in two reports, one focused on lateral vibrations and one focused on torsional- and stator vibrations. For lateral vibrations, a tool to analyze and group the problem has been developed.

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