

TORSIONAL AND STATOR VIBRATIONS IN TURBINES AND GENERATORS

REPORT 2016:295



VIBRATIONS IN
NUCLEAR APPLICATIONS



Torsional and stator vibrations in turbines and generators

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 diskuteras Lateral vibrationer (2016:294) och i del 2 – denna rapport - diskuteras Torsions- och statorvibrationer (2016:295).

Huvudgrupperna för torsions- och statorvibrationer har sedan delats upp ytterligare.

Undergruppen för Torsionsvibrationsproblem behandlar:

- Torsionsvibrationer till följd av elektriska fel
- Torsionsvibrationer till följd av subsynkron resonans
- Bladvibrationer till följd av excitation av vätska

Undergruppen för statorvibrationsproblem behandlar:

- Vibrationer i statorns ändlindningar
- Axiella vibrationer i statorn
- Axiella vibrationer i statorns kylrör

I denna rapport presenteras, för var och en av undergrupperna, en teoretisk och tillämpad beskrivning av de specifika torsionsvibrationsproblemen och statorvibrationsproblemen, 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 vara till stor hjälp för att tillse att rätt åtgärder införs

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 and
- Stator Vibration Problems

The overall technical report is subdivided in two parts. In Part 1 **Lateral Vibration Problems** have been discussed. This report is Part 2 with the content of **Torsional Vibration Problems** and **Stator Vibration Problems**.

Each of the main vibration problem groups was further subdivided. The subgroup of Torsional Vibration Problems deals with:

- Torsional Vibrations due to Electrical Faults
- Torsional Vibrations due to Sub Synchronous Resonance
- Blade Vibrations due to Fluid Excitation

and the subgroup of Stator Vibration Problems deals with:

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

The technical report (Part 2) presents for each of the different subgroups a physical and practical description of the specific torsional problems and stator vibration problems, 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.

As in Part 1 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 and
- 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. After this grouping of the observed vibrations the physics of the vibration phenomena, the investigation technique (analysis) and the method of mitigation was described. The 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 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 (see Part 1 of the report)

- 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, condenser vacuum and load
- Lateral Vibrations due to Unequal Moments of Inertia
- Unstable Lateral Vibrations
- Rotor cracks not considered (explanation see Part 1)

Torsional Vibration Problems

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

Stator Vibration Problems

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

The following tables 4.1.2 and 4.1.3 illustrate in which of the power plants the vibration phenomena **Torsional Vibrations** and **Stator Vibrations** have been observed.

Table 4.1.1 about **Lateral Vibrations** has already been presented and discussed in Part 1 of the report, showing six different lateral vibration phenomena. The table is not shown any more in this Part 2.

Table 4.1.2 presents 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. It is also shown in which of the observed power plants these phenomena were observed. More details are described in the following chapter 6.

Vibrations	Oskarshamn	Olkiluoto	Forsmark
Torsional Vibrations due to Electrical Faults	Not reported	Not reported	Disturbance in electrical grid. Step change. 2X and 1X vibrations. Coupling moves.
Torsional Vibrations due to SSR	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 .
Blade Vibrations in Last Stage LPT	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

Vibrations	Oskarshamn	Olkiluoto	Forsmark
2X End Winding Vibrations in Generator Stator	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.
2X Stator Core Axial Vibrations	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.
2X Stator Cooling Pipe Vibrations	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.

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Table 4.1.3 Grouping of vibrations into three different Stator vibration phenomena

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. They usually cause 2X vibration signals. More details follow in Chapter 7 of this report.

4.2 ANALYSIS OF TORSIONAL VIBRATION AND STATOR VIBRATION PROBLEMS

The base for the analysis of torsional vibrations and stator vibrations in power plants is to measure these vibrations with suited transducer types. For the case of torsional vibrations speed encoders at the shaft end can be used in order to measure vibration velocities (see Figure 2).

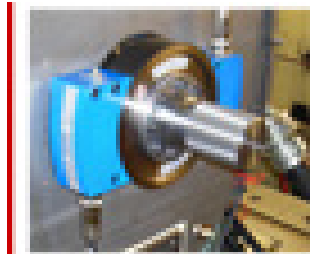


Figure 2 Speed encoder at shaft end to measure torsional vibration velocities

Torsional vibrations can also be measured by optical systems. Figure 3 shows a measurement setup for torsional vibrations using sensor targets for encoder based measurements.

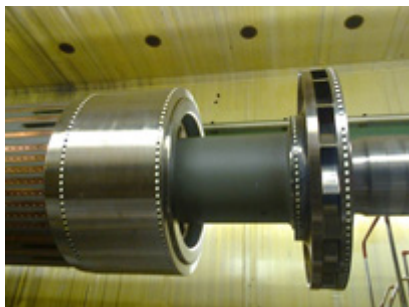


Figure 3 Measurement setup with sensor targets for torsional vibrations

Strain gauge arrangements can also be used to measure shaft torques. However, in most cases they are applied in test rigs but not in real operating shaft trains.

Usually 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 amplitudes and phase over short times or long times
- Frequency spectra to analyze the frequency content of the signal,
- e.g. waterfall diagrams.

For the case of stator vibrations, e.g. End Winding Vibrations usually accelerometers or vibration velocity transducers are used. Figure 4 shows transducer locations at the circumference of the end windings of a turbogenerator.

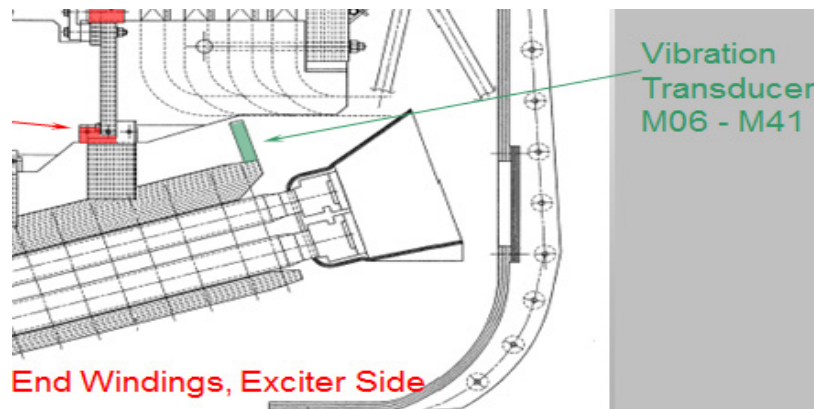


Figure 4 Vibration transducer location at end windings of a turbogenerator

4.3 MITIGATION OF THE VIBRATION PROBLEMS

In the following Tables 4.3.2 and 4.3.3 a summary of the study “Vibrations in Turbines and Generators – Analysis and Mitigation” is presented, in this case especially for the groups of torsional vibrations and stator 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 for torsional and stator vibration problems will later be presented in the chapters 6 and 7.

4.1 Grouping

4.2 Investigation

4.3 Mitigation

<u>Torsional Vibrations due to Electrical Faults</u>	<u>Transient Analysis of Electrical Faults. Shear stresses. Encoder measurements</u>	<u>Keep torsional natural frequencies outside limit range (ISO 22266). Lifetime assessment.</u>
<u>Torsional Vibrations due to SSR</u>	<u>Determine low torsional natural frequencies and allowable angular displacements</u>	<u>Protection System: Sub Synchronous Damping Controller (SSDC).</u>
<u>Blade Vibrations in Last Stage LPT</u>	<u>Blade Vibration measurements (BVM) on blades.</u>	<u>De-activate partial vacuum breaking during run down</u>

Table 4.3.2 Analysis Techniques and Mitigation Methods to solve Torsional Vibrations

4.1 Grouping**4.2 Investigation****4.3 Mitigation**

2X End Winding Vibrations in Generator Stator	Experimental investigation of mech. and electrical influence parameters	Avoid mechanical resonances close to 100 Hz. Mass and stiffness tuning.
2X Stator Core Axial Vibrations	Measurement of axial core vibrations	Consolidation of the stator core by inserted wedges
2X Stator Cooling Pipe Vibrations	Vibration measurements at cooling pipes in 3 directions	Avoid resonances close to 100 Hz. Tuning by masses. Add damping.

Table 4.3.3 Analysis Techniques and Mitigation Methods to solve Stator Vibrations

5 Lateral Vibration Problems – Analysis and Mitigation

The different Lateral Vibration phenomena, observed in the three power plants, have been described in detail in Part 1 of the report. We refer to this part of the report.

6 Torsional Vibration Problems – Analysis and Mitigation

6.1 TORSIONAL VIBRATIONS DUE TO ELECTRICAL FAULTS

6.1.1 Physical Description

Large Steam Turbine Power trains can strongly be influenced in their mechanical behavior due to interactions between the mechanical and the electrical systems. Figure 5 shows the different components of the mechanical and the electrical systems and their coupling to each other.

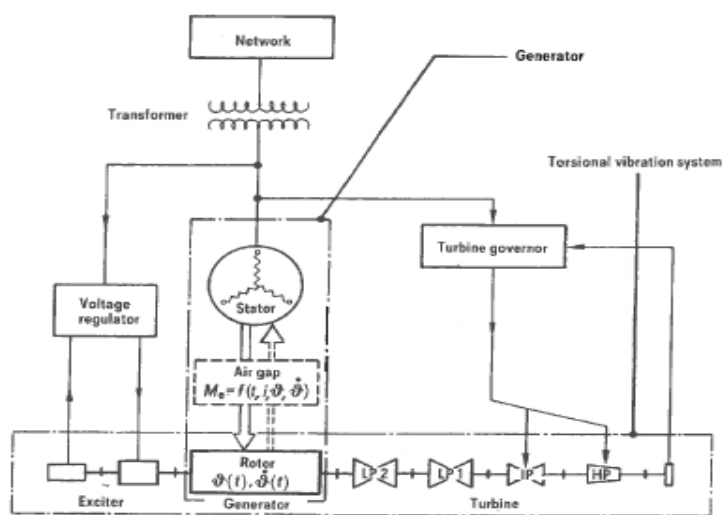


Figure 5 Electro-Mechanical Interaction in a Steam Turbine Power Train

In case of disturbances in the electrical system - such as 2-phase and 3-phase short circuits, faulty synchronization and negative sequence current - the torques in the air gap of the generator will be disturbed as well and due to this the shaft is excited to torsional vibrations with resulting internal torques and shear stresses in the shaft elements. The frequency content of the transient air gap torques in case of electrical faults in general consists of the network frequency and twice the network frequency. Torsional natural frequencies of the steam turbine shaft train should therefore not be too close to these excitation frequencies of the grid. This has to be considered in the design of the shaft train, where dynamic coupling effects with the last stage blades of the turbines also have to be considered. When torsional vibrations are excited due to electrical disturbances, the transient response needs long to decay. This disadvantage is caused by low damping in the torsional system.

A permanent excitation from the electrical grid may be caused by non-symmetrical electrical grid loading (see Figure 6). In this case the shaft line is excited with a 2X

grid frequency (100 Hz in the power plants of Sweden and Finland). As a design criteria the torsional natural frequencies of the shaft line (including Blade Rotor Interaction BRI) should therefore be outside a limit frequency range around the 2X grid frequency, defined in ISO 22266.

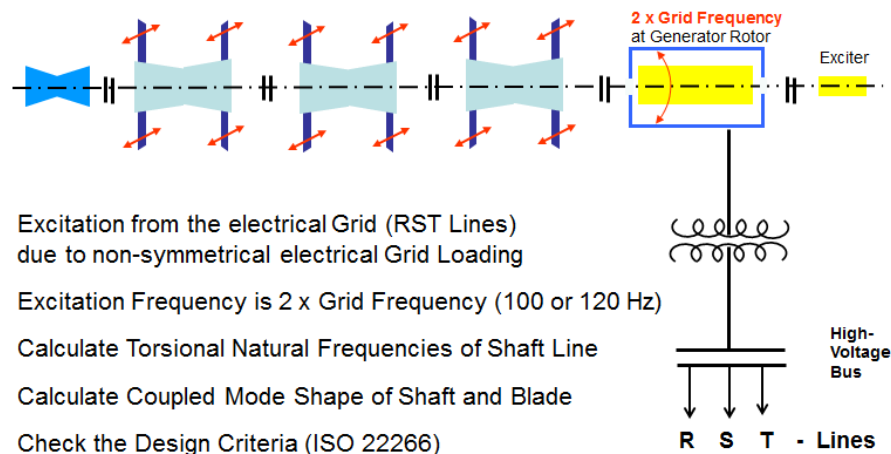


Figure 6 Excitation of the shaft line due to non-symmetrical Grid Loading

6.1.2 Investigation Techniques – Vibration Problem analysis

In the design process torsional vibrations of the shaft train are assessed by investigating the torsional natural frequencies and mode shapes, by a transient analysis of electrical faults, like short circuits and fault synchronization and by a negative sequence current excitation (2X grid frequency). The shear stresses due to the electrical faults are evaluated and the material utilization is considered.

An experimental investigation in the power plant during operation is only possible, when the shaft train is equipped with suited transducers (see chapter 4.2). Measurements taken permanently in the power plant can be used for a Life Time Assessment to estimate the Material Utilisation (see chapter 6.1.4).

6.1.3 Observed Torsional Vibrations in Power Plants

There were no reports about torsional vibrations due to electrical disturbances from the power plants in Oskarshamn and Olkiluoto. A problem was discussed during the Forsmark visit. 1X and 2X lateral vibration components were observed in unit F2 after a disturbance in the grid (e.g. short circuit?). At the moment there is no clear explanation for this frequency components in the lateral vibrations.

To find an explanation, here are some comments about the phenomenon:

Electrical faults may have the following frequency components :

2 phase short circuit:	1X grid frequency	2X grid frequency
3 phase short circuit	1X grid frequency	
Faulty synchronization	1X grid frequency	
Negative sequence current		2X grid frequency

If we assume for the Forsmark case a 2-phase short circuit, both frequency components should be seen in a torsional vibration signals. But here both frequency components have been observed in the lateral vibration signal. Reasons for this could be:

- The large air gap torque also acts at the generator housing and the foundation. Due to this also the bearings could have been excited with the 1X and 2X frequency components, which are seen in the proximity probes of the generator bearings.
- At the short time of the electrical disturbance there might be an impact on the coupling, leading to a unbalance change. This would increase the 1X lateral component. The 2X component could come from a nonlinear behavior in the oil film in case of large unbalances in the coupling.

For further clarification Forsmark delivered the following comments:

Forsmark 1 and 2 units get vibration changes when there are disturbances on the power grid or if there is a heavy thunder/lightning. This disturbance is always seen on the 2X vector on the shaft proximity probes on the Generators. This 2X changes are likely to depend of changes in the magnetic forces/field. Sometimes when the disturbance on the grid is large there may also be some coupling change showing at changing of 1X vector on the turbine train, both on shaft and bearing transducers.

6.1.4 Solution – Vibration Problem Mitigation

In the design process numerical calculations are performed in order to determine the stresses in the shaft train due to electrical disturbances. Predicted Stresses in case of disturbances should be lower than the allowable stresses. Regarding the negative sequence current the torsional natural frequencies of the shaft train should be outside a limit frequency range (ISO 22266).

If torsional vibrations can be measured e.g. by an encoder a lifetime assessment to estimate the material utilization could be installed in a shaft train. Figure 7 shows a model based estimation of the lifetime with a Monitoring System during operation.

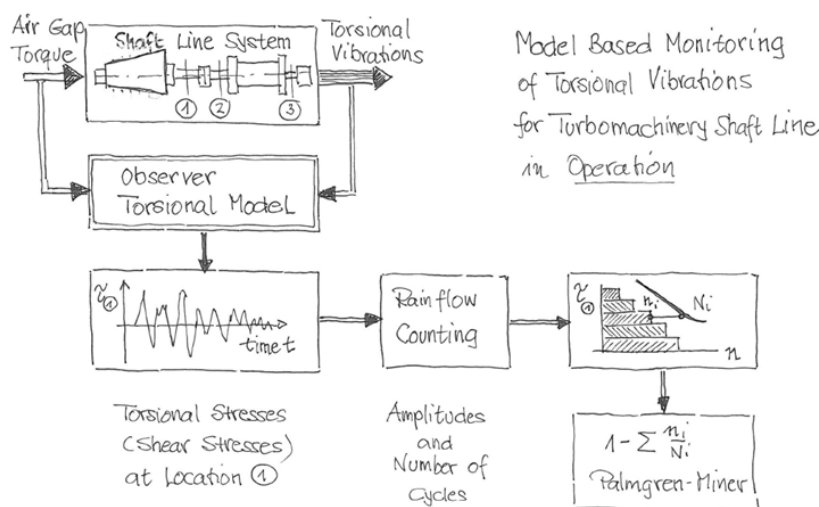


Figure 7 Life time assessment for a shaft train based on torsional vibration measurements

An Observer Torsional model is used in parallel to the real system. The observer determines at each time the torsional vibrations based on input data taken from measurements at the real system (air gap torque and torsional vibrations at defined locations). Stresses versus time can then be calculated with the model. By means of the Rainflow-Counting method the amplitudes and the number of cycles lead to a load-cycle diagram, from which the lifetime estimation by means of the Palmgren-Miner formula can be obtained. At each time during operation the remaining lifetime for the machine can be predicted.

6.2 TORSIONAL VIBRATIONS DUE TO SUB SYNCHRONOUS RESONANCE

6.2.1 Physical Description

Sub Synchronous Resonances are oscillations in the electrical and mechanical systems which occur, when the turbogenerator units feed into a network, the long lines of which are compensated by series capacitors. An electrical resonant circuit can be triggered by a fault in the electrical system and an exchange of energy can occur between the shaft and the inductive and capacitive elements of the “Generator-Transformer-Line-Network” circuit. The resulting currents generate Low Frequency Electrical Torques in the Generator air gap.

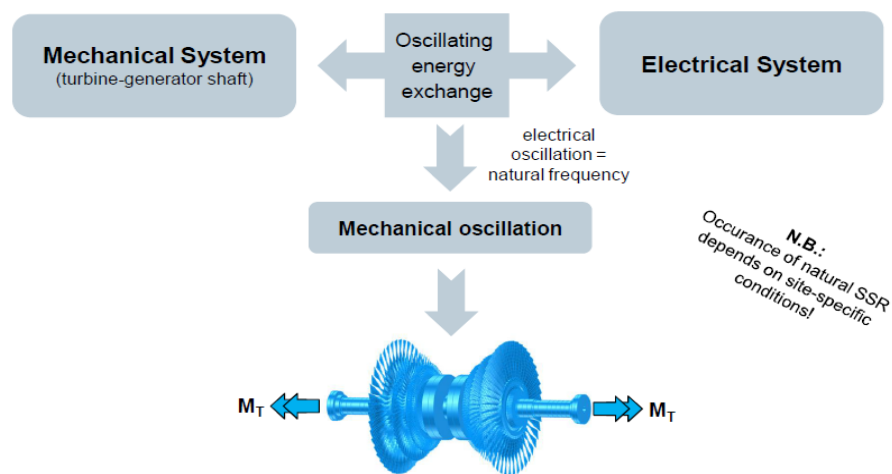


Figure 8 Sub Synchronous Resonance (from Siemens presentation)

If these frequencies are in the vicinity of one of the lowest Natural Torsional frequencies of the shaft line, the shaft assembly may be excited to strong Resonant Vibrations. These in turn are transmitted into the electrical system by the electromechanical coupling. The electro-mechanical damping of the coupled system may be low or even negative. If the damping is negative, there is an increase in the torsional vibrations and in the electrical torque, which may lead to high stresses and even to damage.

6.2.2 Investigation Techniques – Vibration Problem analysis

In order to protect the mechanical system of the shaft train against Sub Synchronous Resonances, control systems have been developed (see chapter 6.2.4). Such control systems need as input data the torsional natural frequencies of the shaft train in the low frequency range (sub synchronous) and in addition the maximum allowable angular displacements of the sub synchronous oscillation modes for the generator angle (measurement location) before any part of the shaft system has reached its fatigue limit. These numerical data have to be delivered by the manufacturer. As an example the following Table 6.2.1 shows the torsional natural frequencies of the shaft train TVO/OL1 in the low frequency range.

Mode N.	Frequency (Hz)	Description of modes
1	7.7	Exciter quill-shaft
2	9.3	Shaft-line 1st (quill-shaft in antiphase)
3	18.9	Shaft-line 2nd (LP3–HP–Gen.)
4	24.0	Shaft-line 3rd (LP2–HP–LP4+Gen.)
5	29.1	LP4-LP3-Gen.
6	35.7	HP-LP1
7	119.3	Exciter 1st mode
8	124.5	Generator 1st mode

Table 6.2.1 Torsional Natural Frequencies of TVO/OL1 in the low frequency range

The natural frequencies can also be measured when the shaft train is equipped with suited transducers to measure the torsional vibrations of the shaft train (chapter 4.2).

In Forsmark 3 a SSR system for protection and a TME Torsional Monitoring Equipment for measurement and analysis are available. Torsional natural frequencies can be measured by connecting a frequency analyzer to the TME system. This has been done several times – with original rotor train, after changing of the 3 LP:s, before and after changing of the HP and finally after changing of the Generator to be able to compare with calculated levels.

6.2.3 Observed Torsional Vibrations : Sub Synchronous Resonance

Torsional Vibrations due to sub synchronous resonances are only relevant in power plants, which are connected to a grid which is sensitive for this kind of excitation. It seems, that Oskarshamn is not relevant for sub synchronous resonances. Opposite to this in Olkiluoto and Forsmark control systems have been installed for protection. Due to SSR the blade resonance on Forsmark 3 (old LP-design) was excited and 2 last stage blades were later found to be almost half off with crack in the foot of the blades. The 2 blades with cracks were positioned 180 degrees apart.

Forsmark 3 is most sensitive because of the position in the power net.

6.2.4 Solution – Vibration Problem Mitigation

As mentioned before in the two power plants in Olkiluoto and Forsmark protection systems have been installed in cooperation with the responsible grid driver. In order to control the damping of sub synchronous torsional oscillations so called Sub Synchronous Damping Controllers (SSDC) and Sub Synchronous Torsional Interaction Supervision schemes (SSTIS) have been implemented. For a final design of these systems SSDC and SSTIS besides the lower torsional natural frequencies of the shaft train also the maximum allowable rotor angle displacement for the low oscillation mode shapes of the turbine generator is needed, as described in 6.2

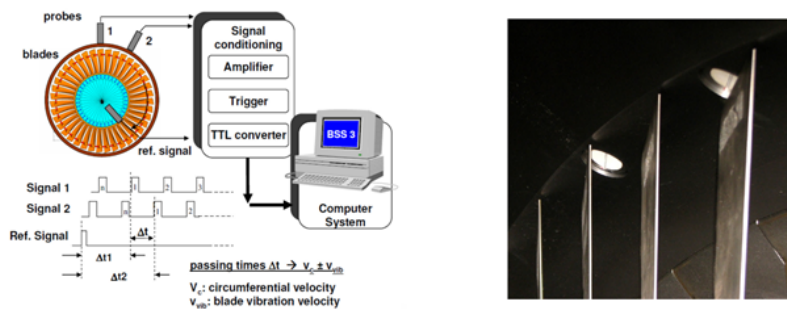
6.3 BLADE VIBRATIONS DUE TO FLUID EXCITATION

6.3.1 Physical Description

Blade Vibrations in large steam turbine shaft trains often appear as coupled vibrations with the torsional vibrations of the rotor system. This Rotor Blade Coupling occurs in the last stage blades of the LP Turbines and has to be considered in the case of Torsional Vibrations due to electrical faults and the Negative Sequence Current (see chapter 6.1). However, blade vibrations can also be excited by the fluid, e.g. by the steam in the turbines of the shaft train. Blade cracks in the blades or rubbing marks between the rotating blades and the stator may be caused by too high vibrations of the blades. High blade vibrations due to fluid excitation were observed in the Forsmark units F1 and F2.

6.3.2 Investigation Techniques – Vibration Problem analysis

Rubbing marks on the tips of some of the last stage blades of the retrofitted LP turbines of Forsmark F1 and F2 were discovered during inspections in 2008 and the following years. It was concluded, that these rubbing marks could be caused by too high vibration amplitudes of the blades. Therefore it was recommended to perform in-service blade vibration measurements (BVM) in order to identify the operating conditions at which the suspected high vibration amplitudes occur. Blade tip vibration sensors were installed on flows LP2GS and LP3GS of turbine T12 of Forsmark unit F1 turbines during the outage in August 2012 (see Figure 9).



- Sensors installed in turbine casing to enable very accurate measurement of the arrival time at the sensor location of each blade on each revolution
- 3 sensors on both LP2 GS and LP3 GS
- Blade vibration determined from the variation in the measured blade tip arrival time

Figure 9 BVM measurements on blades in Forsmark 1 in 2012

6.3.3 Observed Blade Vibrations due to Fluid Excitation

Blade vibration measurements were made during the start-up and at operation with maximum load. It could be demonstrated, that high blade vibrations occurred during a run-down with partial vacuum breaking on 8th September 2012 (see Figure 10).

Measured blade vibration during rundown with partial vacuum break

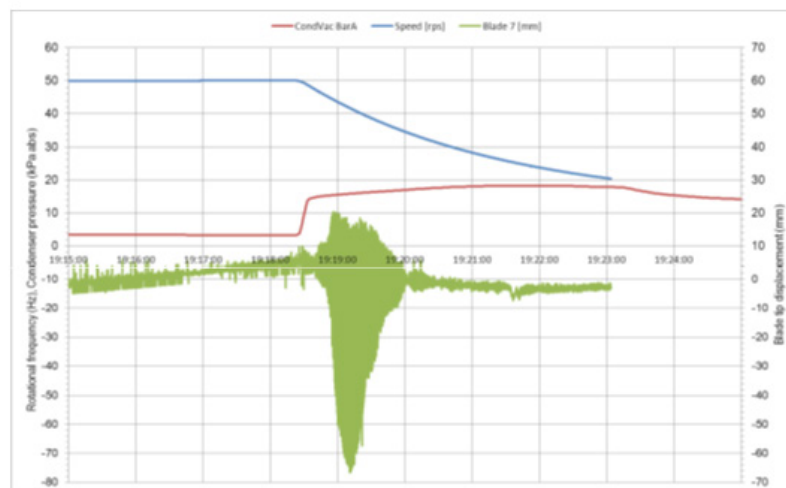


Figure 10 BVM – measurements on blades in Forsmark 1 with high blade vibrations during run down with partial vacuum breaking

The BVM system measures an “apparent” blade vibration, which is derived directly from the time-of arrival measurements of the blade tips when they pass the tip timing sensors. It is affected by variations in the axial position of the blade tip each time it passes a sensor, which can occur due to the axial component of the

blade tip vibration displacement. For modes of interest, this means that the apparent displacement is typically a factor of 3 larger than the real modal vibration displacement (the actual factor depends on the vibration mode and is determined by finite element vibration analysis).

The vibration amplitudes reached values up to 80/3 mm in axial direction. Such high vibrations can be avoided by de-activating partial vacuum breaking during run down on all four turbines of Forsmark units 1 and 2.

6.3.4 Solution – Vibration Problem Mitigation

To avoid high blade vibrations of last blade stages of LP turbines in Forsmark units F1 and F2 the partial vacuum breaking during run down should be de-activated. Further measurements should be taken for other critical operating conditions. If these testings show other operating regions with high vibrations, appropriate actions should be taken either to avoid these operating regions or to eliminate the vibrations in these regions by appropriate modifications to the blades.

7 Stator Vibration Problems – Analysis and Mitigation

7.1 END WINDING VIBRATIONS IN GENERATOR STATOR

7.1.1 Physical Description

Figure 11 shows the cross section of a TVO/OL1 generator stator (3000 rpm) with its main components: the rotor with copper windings for the magnetization and the stator core with copper windings and the End Windings on both sides (turbine and exciter side).

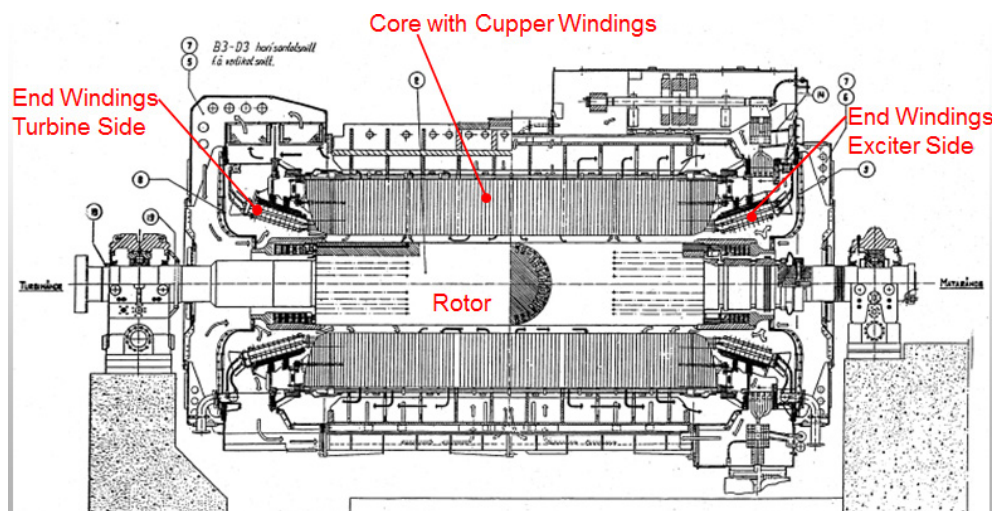


Figure 11 Generator Stator TVO/OL1 with its main components

A magnetic field with an elliptical shape in circumferential direction is built up due to the magnetization current I_{magn} in the rotor windings (Figure 12). At running speed with rotational frequency $f = 50$ Hz the magnetic field with a magnetization force F_{magn} excites the stator along the core with a frequency of $2 \times 50 \text{ Hz} = 100 \text{ Hz}$. The magnetic force depends on the exciter current in the rotor and on the magnetic circuit resistance.

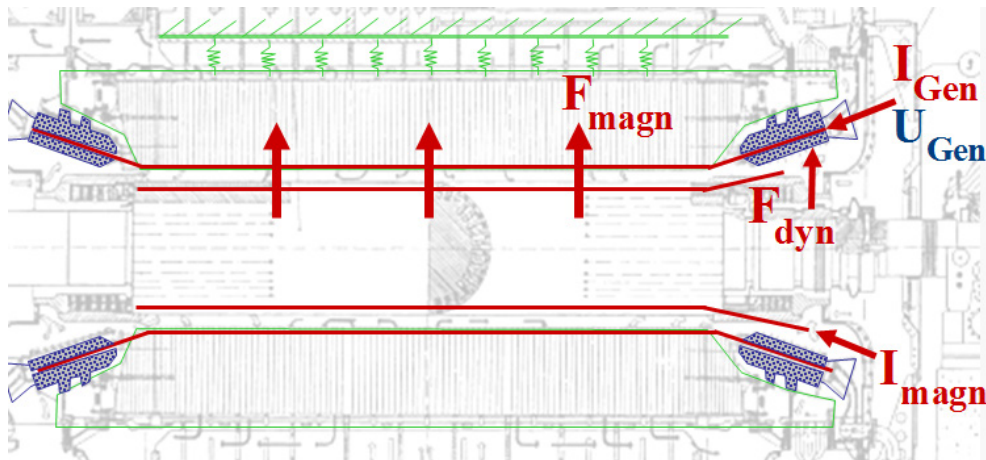


Figure 12 Electromagnetic and Electrodynamic Forces acting at the Generator stator

An additional electrodynamic force F_{dyn} excites the end windings directly at the location of the end windings, with 100 Hz as well. The electrodynamic force depends on the stator current and on the flux density in the end winding area. Both forces are responsible for the excitation of the end windings, as is explained in Figure 13.

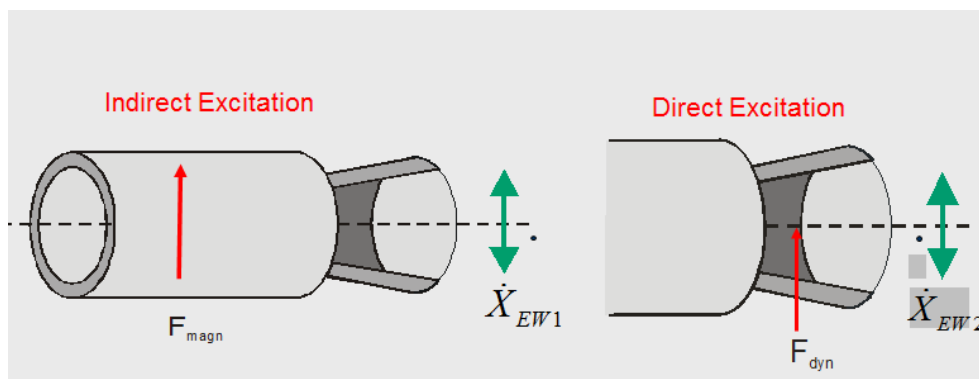


Figure 13 Two sources for the End Winding vibrations

The magnetic force first excites the stator core and the stator core transfers the vibration energy to the end windings (indirect excitation). The direct excitation is coming from the electrodynamic force. This is also pointed out in Figure 14, where the two excitation forces are presented again. In this Figure it is also shown, that the end winding vibrations depend very much on the dynamic characteristics of the end windings.

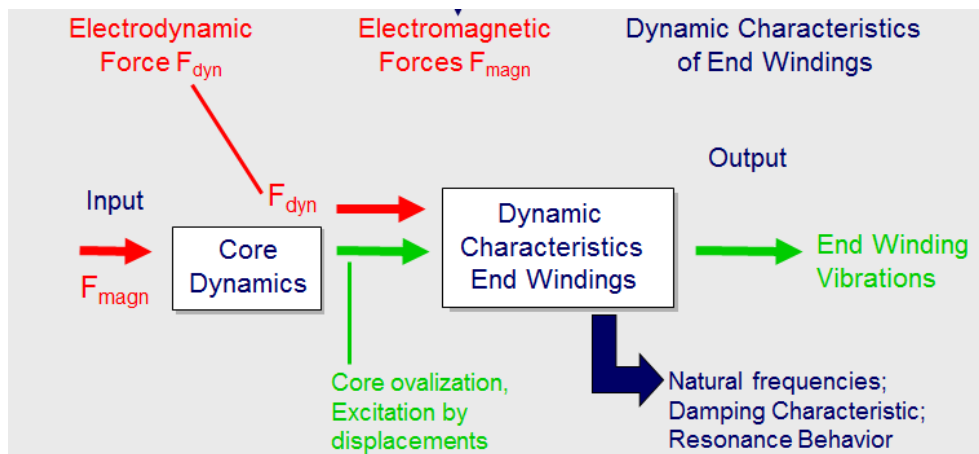


Figure 14 Forces and Dynamic Characteristics of the End Winding (Natural frequencies and damping) determine the End Winding vibrations

7.1.2 Investigation Techniques – Vibration Problem analysis

In Figure 15 it is once again shown, which parameters have an influence on the end winding vibrations. These are besides the electrical quantities also the mechanical parameters mass, stiffness and damping of the end winding system. An end winding resonance should in any case be avoided, because this would increase the vibrations due to resonance behavior.

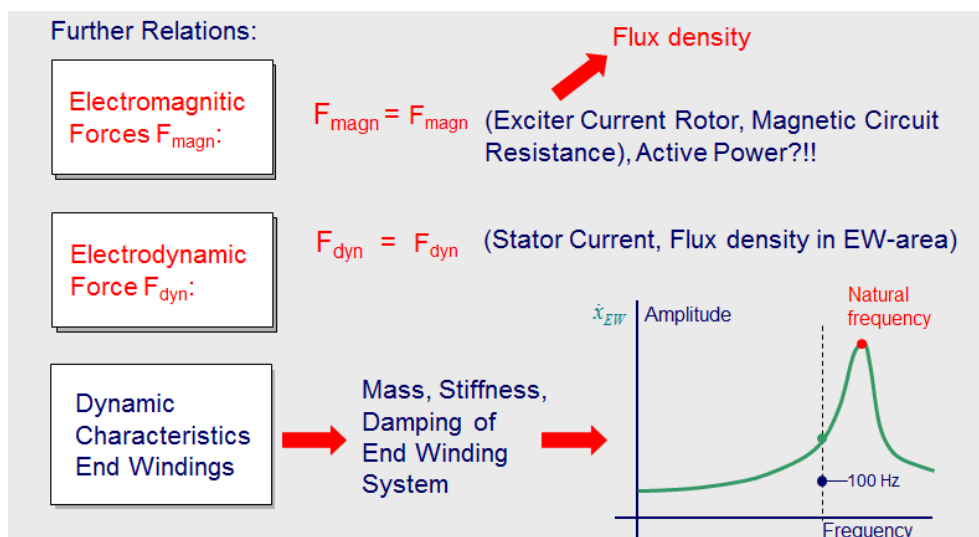


Figure 15 Electrical and mechanical parameters influencing the end winding vibrations

Figure 16 shows changes of some electrical parameters versus time (2 days) after a run up and their influence on the vibrations of the end windings.

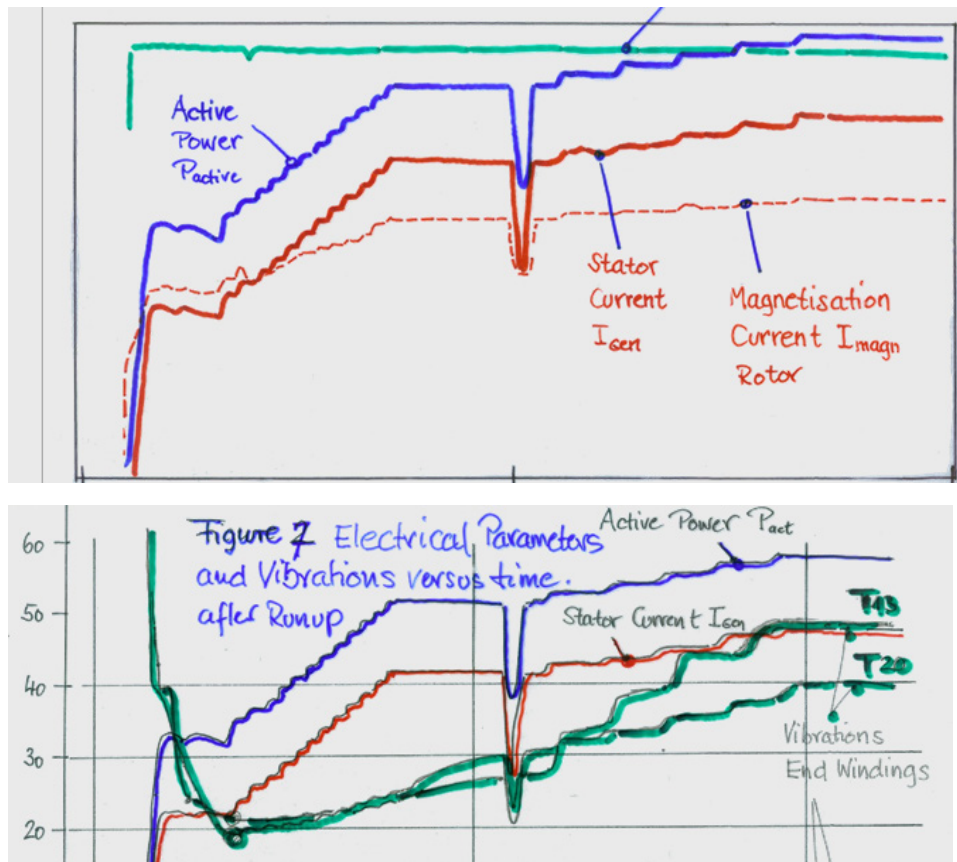


Figure 16 Electrical Parameters and End Winding vibrations over a time period of two days after a run up of the TVO/OL1 shaft train

The influence of the Active Power on the end winding vibrations is shown in Figure 17.

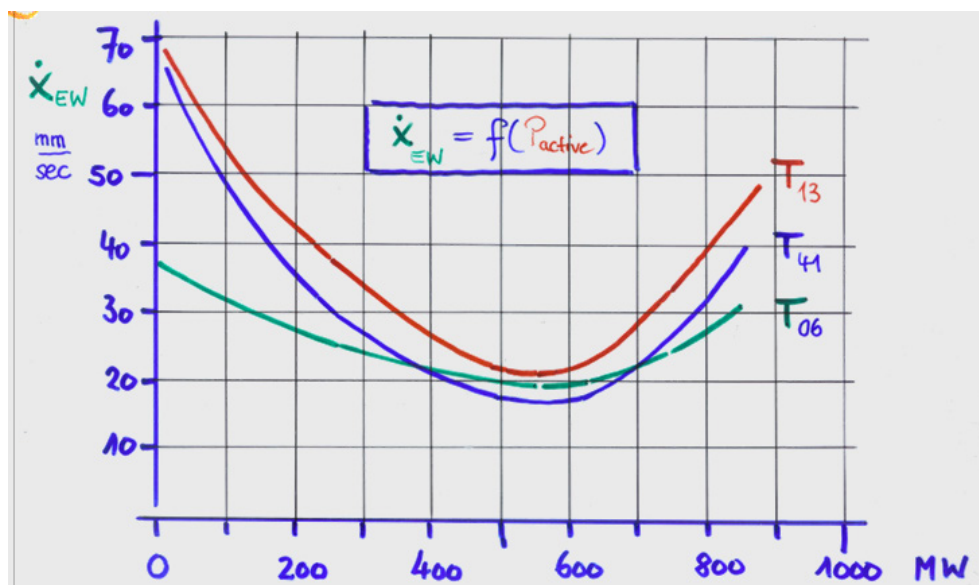


Figure 17 End Winding Vibrations in mm/sec at different locations (Turbine side: T06, T13 and T41) in dependence of the Active Power of the unit TVO/OL1

7.1.3 Observed End Winding Vibrations in Generator Stator

End Winding vibrations in the units OL1 and OL2 of the TVO power plant were observed permanently over a long time because of temporary high vibrations. For an observation several transducers were attached around the end windings as shown in Figure 4. In this way vibration mode shapes (e.g. elliptical shapes) could also be identified. As an example Figure 18 shows a typical elliptical mode shape.

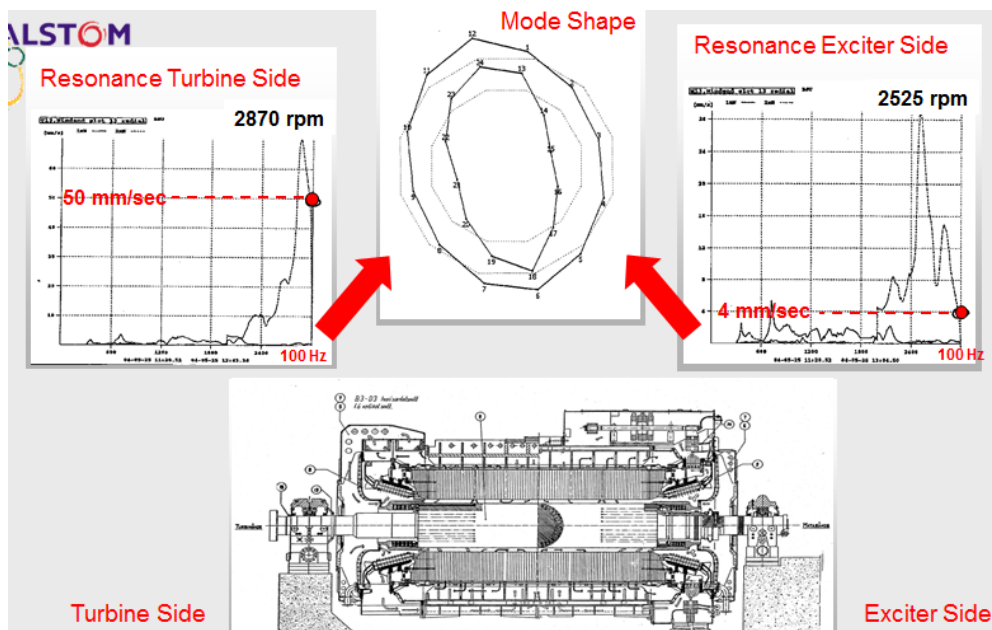


Figure 18 Observation of End Winding vibrations- Run up curve and mode shapes.

The two Frequency Response curves (Exciter side and Turbine side) in Figure 18 show the amplitudes of the 2X end winding vibrations for a special location in circumferential direction. It can be seen in the Turbine side diagram, that the resonance peak is close to 100 Hz, leading to an amplification of the vibrations.

Long term vibration amplitudes (example from 1996 to 2005) in mm/sec are shown in Figures 19 and 20 for the 6 circumferentially distributed transducers of both end windings in the unit OL1 in the TVO plant. The amplitudes are temporary high, with amplitudes up to 60 mm/sec.

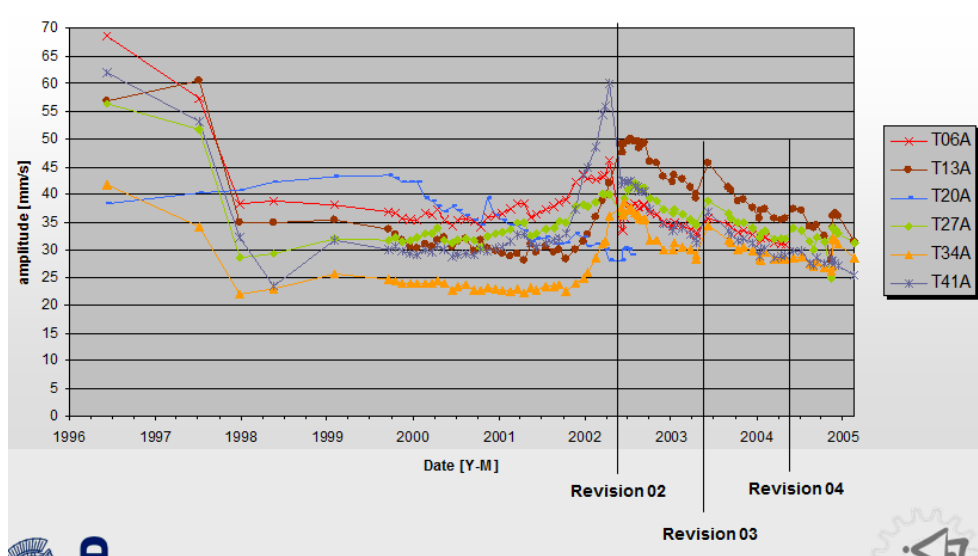


Figure 19 Long term observation (1996-2005) of vibration amplitudes in mm/sec for the End Windings on Turbine side of TVO/OL1. 6 Transducers in circumference.

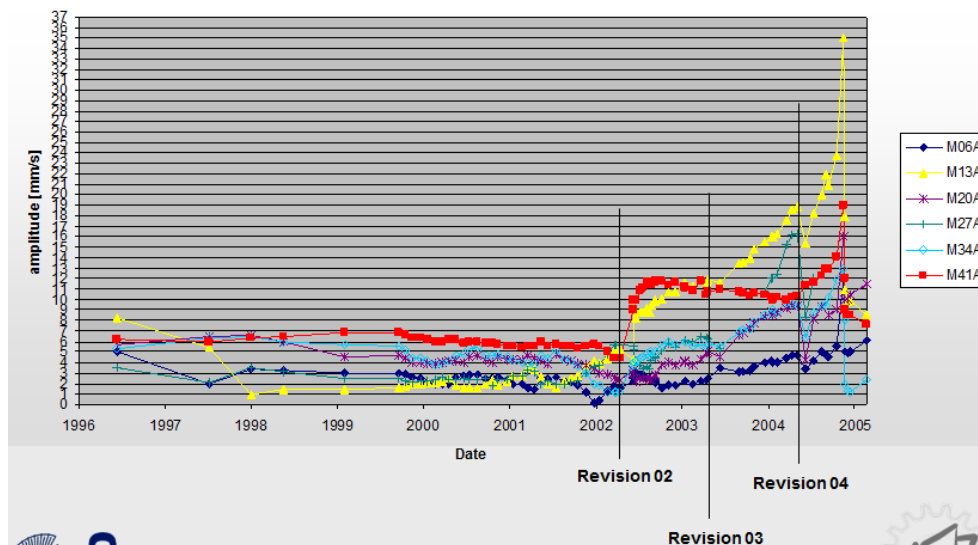


Figure 20 Long term observation (1996-2005) of vibration amplitudes in mm/sec for the End Windings on Exciter side of TVO/OL1. 6 Transducers in circumference.

End Winding Vibration problems were also reported from unit F2 in the Forsmark power plant with increasing amplitudes versus time, probably due to a decreasing resonance frequency into the 100 Hz range. Cracks were found in the end winding supports, which were repaired by injection.

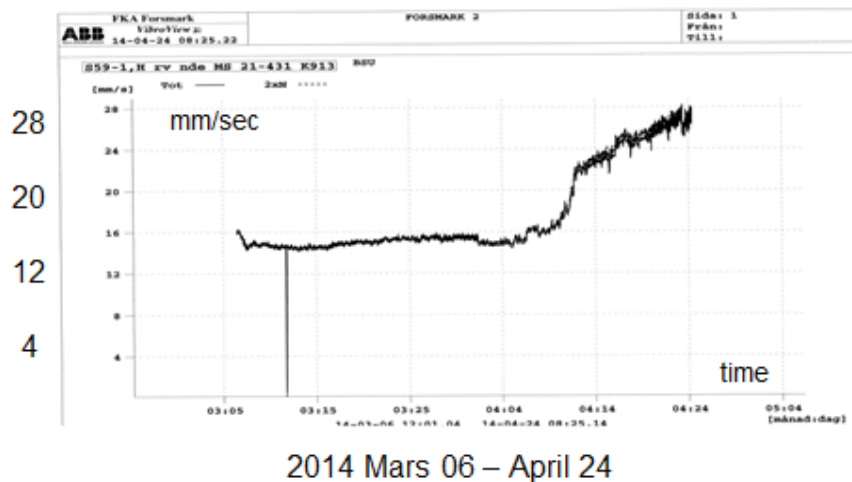


Figure 21 End Winding Vibration Problems in unit F2 of power plant Forsmark

7.1.4 Solution – Vibration Problem Mitigation

Parameters of the electrical system (stator currents, exciter currents, stator voltage, power) that have an influence on the end winding vibrations can hardly be changed because they are designed for an efficient power plant operation. A good solution is always to avoid mechanical resonances of the end winding system in the neighborhood of 100 Hz. This should already be considered in the design stage. One decision can be to use either a free end winding system or one that is supported. As a measure to solve an existing end winding vibration problem system tuning can be applied, e.g. by adding masses.

7.2 STATOR CORE AXIAL VIBRATIONS

7.2.1 Physical Description

The observed high axial 2X stator core vibrations (100 Hz) in the generator stator have probably the same excitation source as the end winding vibrations. Excitation is due to the electromagnetic and electrodynamic forces in the generator as described in 7.1. The vibration energy is transferred from its source to the generator stator, where especially axial vibrations can also occur due to coupled mode shapes between radial and axial directions.

A special problem occurred with the new TVO-generator stators S4 and S5 in Olkiluoto. Due to a bad stacking procedure at manufacturing of the core the needed core axial pressure was not achieved. This resulted in high axial vibrations up to 50 mm/sec with a high risk of failures in the core.

Forsmark 1 and 2 had also problems with high axial vibrations on the stator cores.

7.2.2 Investigation Techniques – Vibration Problem analysis

TVO: The problem with the high axial vibrations in the stator core due to low axial core pressure had to be solved by a consolidation. This was managed by moving and adding pressure in two air ducts in DE of the core, which was managed by use

of pressurized flattened copper tube. In the package between pressure ring and core wedges were inserted. It is expected that the result of the consolidation can finally be evaluated after 3 years of relaxation. During this time the pressure will come to a uniform pressure over the length. Due to this changes of the axial vibrations may occur.

7.2.3 Observed Stator Core Axial Vibrations

TVO: After the consolidation measures the observed axial vibrations at TVO have been in the range of 10 mm/sec, which is acceptable. Recommended limit values are 18 mm/sec (base risk).

Forsmark: 2X Stator core axial vibrations in Forsmark 1 and 2 after many years of running in the old design of Generator. Temporary fix was to cool the stator housing from outside. The reason to the axial vibrations was that the grip between stator core and stator structure (in Swedish stator stomme) had disappeared with time. Resonance problem.

7.2.4 Solution – Vibration Problem Mitigation

TVO: A consolidation of the stator core by introducing wedges in the stator core was the preliminary solution to reduce the axial vibrations of the stator core in Olkiluoto. A permanent observation of the axial vibrations is recommended

Forsmark 1 and 2: Temporary fix was to cool the stator housing from outside. First with blowing air, then by mounting cooling pipes on the stator frame. All 4 Generator stators were then exchanged to new design. No such problems have occurred with the new stator design in Forsmark 1 and 2.

7.3 STATOR COOLING PIPE VIBRATIONS

7.3.1 Physical Description

The observed high horizontal and axial 2X stator cooling pipe vibrations (100 Hz) at NDE of generator stator S5 (Olkiluoto OL2) during normal operation has probably the same excitation source as the end winding vibrations. Excitation is due to the electromagnetic and electrodynamic forces in the generator as described in 7.1. The vibration energy is transferred from its source to the generator ends, where the ring pipes are located.

Forsmark also had large problems especially on one of the old design stators on Forsmark 2 (TA21) with very high vibrations on piping, resonance and leakage.

No such problems have occurred with the new design of stators in Forsmark 1 and 2. No resonance in the piping with the new design.

7.3.2 Investigation Techniques – Vibration Problem analysis

TVO: Vibrations were measured with transducers, mounted on the horizontal surface of the upper flange of the water tube NDE of generator in October 2012. These measurements of vibrations in mm/sec were taken in axial, horizontal and vertical directions. Highest values in OL2-S5 occurred in horizontal direction with 48 mm/sec and in OL1-S4 in axial direction with 46 mm/sec. Investigations included also variable speed tests and impact tests to detect possible resonance locations close to 100 Hz.

Based on the measured vibrations the stresses in the piping system were checked and the vibration limits determined.

7.3.3 Observed Stator Cooling Pipe Vibrations

TVO: Observed stator cooling pipe vibrations are described in 7.3.2. The vibration situation today has improved in generator S5 since rotor R8 was changed to rotor R7. Compared to R8 the rotor R7 differs in the following design points: different slot sizes, different cooling fans, different air gap. The vibrations in stator S4 are still the same as before.

Forsmark: Forsmark also had large problems especially on one of the old design stators on Forsmark 2 (TA21) with very high vibrations on piping, resonance and leakage.

7.3.4 Solution – Vibration Problem Mitigation

TVO: Potential solutions recommended by Alstom in 2012:

Resonances of piping system close to 100 Hz should in any case be avoided. Stiffness/mass tuning to shift the natural frequencies of existing piping. Vibration isolation of the ring pipe from the end cover (Stator frame). Increase of damping in the system (viscous damper). These dampers were installed to reduce the vibrations.

Forsmark: New Design of Stator. No such problems have occurred with the new design of stators in Forsmark 1 and 2. No resonance in the piping with the new design.

TORSIONAL AND STATOR VIBRATIONS IN TURBINES AND GENERATORS

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.

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