Measurement and Analysis of Damping in Hydropower Units

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Abstract

Hydropower and nuclear power are the most important sources of energy in Sweden. Half of Sweden’s total energy production is produced in the hydropower plants along the Swedish rivers. The size of the hydropower units in these power plants varies. The biggest hydropower unit in Sweden, Harsprånget G5, has a rotating structure that weighs over 1000 ton and generates almost 500 MW. Hydropower units use the kinetic energy of the falling water and transform it to electrical energy. In a hydropower unit as big as the G5, approximately 500 m$^3$ water passes through the turbine each second. The enormous flow of water through the turbine and the enormous mass of the rotating structure results in huge loads on the bearings that supports the rotating structure.

The characteristics of generator bearings and the loads acting on it has been analysed in this thesis. A method to determine bearing data from measured loads and displacements in the generator bearing has been developed and evaluated. The loads are measured with a method that allows bearing loads to be measured without disassemble the bearing. Disturbances that affect these measurements has also been analysed. MATLAB were used for mathematical computation and analysis.

Data measured at the hydropower units were analysed and some disturbances were found. Most of the measured data were affected by 50 and 100 Hz disturbances. To avoid the influence of these disturbances appropriate filters should be used. Some strange frequency characteristics were also found in the measurements performed with the inductive displacement sensors. The huge magnetic field that surrounds the hydropower unit probably affects the inductive displacement sensors. Inductive displacements sensor should therefore be avoided in displacement measurements near generators at hydropower units. It is possible to calculate the bearing data for the generator bearing if the displacements and loads in the generator bearing are measured correctly. The bearing data for the generator bearing is calculated in the generator axles static position. The static position of the generator axle depends of the load cases that affect the hydropower unit.
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1 Introduction

1.1 Background

In today’s society is the dependence of electrical energy a fact. For each year that passes the total consumption of electrical energy increases. The hydropower plants along the Swedish rivers produces approximately 50 percent of all the electrical energy consumed in Sweden. Sweden’s largest hydropower units are located along Luleåälven and are owned by Vattenfall AB. Most of these hydropower units are rather old and mechanical failures have occurred. These failures cost a lot of money, both in restorations and loss of income. With the energy taxes of today, the cost for each day of standstill is approximately 1 million SEK for a big hydropower unit. To be able to predict and prevent mechanical failures different kinds of measurements are performed at the hydropower units.

1.2 Problem Description

A method to calculate bearing data from measured displacements and loads in generator bearings needs to be developed and evaluated. The properties of the bearing are changed depending on the load and the condition of the bearing. To determine the mechanical properties of the hydropower unit and the generator bearing load and displacement measurements should be performed. The measurements will be performed in the unfavorable environment that occurs near hydropower units, huge magnetic fields and thermal expansions disturb the measurements. The measured signals need to be analyzed and appropriate hardware and software filters needs to be constructed.

1.3 Purpose

The purpose of this thesis is to analyze and evaluate some of the measurements performed at three big hydropower units in the northern part of Sweden, determine a method to calculate bearing data for generator bearings in hydropower units and calculate the bearing data for the measured data. Disturbances that affects the measurements should also be identified and analyzed, filters that eliminates as much as possible of the disturbances should be created.

1.4 Method

Information and knowledge regarding this thesis was obtained through literature studies, information search at databases and consultation with employees at Vattenfall Utveckling AB. The treated data consist of measurements performed at the hydropower stations during the last three years. Mathematical models was analytically determined and implemented in MATLAB. The results of the mathematical models applied on the measured data are presented in this report.
2 Theory

2.1 Hydropower

Most of the big hydropower plants are placed along rivers in the northern part of Sweden. Five of the six hydropower units that are rated over 200 MW are placed along Luleåälven and all these are owned by Vattenfall AB. The mechanical properties of three hydropower units are analysed in this thesis, H1, H2 and H3. H1 and H2 are rated above 200 MW and H3 are rated at approximately 70 MW. These three hydropower units are placed along Luleåälven.

The mechanical properties of the hydropower units depend on the type of hydropower unit and the geographical properties that surrounds the hydropower unit. The hydropower unit need to transform kinetic energy to electrical energy and are therefore dependent of the head. The choice of turbine type in hydropower units also depends on the heads.

![Figure 2.1. Relation between turbine choice and vertical drop.](image)

Both hydropower unit H1 and H2 are equipped with Francis turbines and H3 is equipped with a propeller turbine. A propeller turbine is a Kaplan turbine with a fixed angle of the propellers runner blades. Different types of turbines have its optimal efficiency levels at different heads. The head in to hydropower unit in H3 is approximately 30 meters and the head in to H1 and H2 are approximately 60 meters. Figure 2.1 shows the range for the different turbine types. Kaplan turbines are used in hydropower units with low heads, Francis turbines are often used in hydropower units with a head between 50 – 500 m. For hydropower units with very high heads are Pelton turbines the most suitable turbine type. There are no large hydropower unit in Sweden with a head higher than 250 m. Pelton turbines are therefore not used in Sweden’s large hydropower units.

Hydropower unit H1, H2 and H3 are all relatively modern hydropower units. Especially H3 that was rebuilt in 1999, it was equipped with a Powerformer and its Kaplan turbine was replaced with a propeller turbine. The Powerformer is a combined generator and transformer that directly produces 50 Hz electrical energy.
The amount of electrical energy \( (P) \) that each hydropower unit produces depends on the head \((h)\), the flow through the turbine \((Q)\) and the efficiency level \((\eta)\) for the hydropower unit. See formula in Equation 2.1.

\[
P = g \cdot \eta \cdot h \cdot Q \quad [kW]
\]  

(2.1)

Hydropower units have generally very high efficiency levels, often above 90 percent. Hydropower unit H1 and H2 are both rated above 200 MW and have heads of approximately 60 meters. The flow through each turbine is there for above 400 m\(^3\) water per second. In the smaller hydropower unit H3 is the water flow through the turbine approximately 300 m\(^3\) per second. The flow through the highest rated hydropower unit in Sweden, G5 in Harsprånget, is approximately 500 m\(^3\) per second. These huge water flows through the turbines results in large forces acting on the turbines.

### 2.1.1 Vibration in Hydropower Units

A hydropower unit rated at 100 MW should be regarded as a relatively large hydropower unit. The turbines and generators in hydropower unit H1 and H2 have masses of approximately 500 tons each. The total weights of the rotating structures in these hydropower units are over 1000 tons. Its driving frequency, \( \omega_{dr} \), is relatively low, 83 rpm, but the moment of momentum of the rotating structure is huge. The rotor diameter of approximately 14 m also results in a quite high circumference velocity for the rotor. Unbalance in rotor or turbine causes huge radial loads on the radial bearings. Deformation of the turbine or turbulence below the turbine causes also loads on the bearings. H1, H2 and H3 are all equipped with an upper and lower generator bearing, an axial bearing and turbine guidance bearing. In H1 and H2 are the lower generator bearings and the axial bearings assembled as a combined axial and radial bearing. The axial loads are consumed by the axial bearing while the upper and lower generator bearing and the turbine guidance bearing consume the radial loads. Only the radial vibrations and loads in the generator bearings have been analyzed in this thesis.
Figure 2.2 shows a picture of a hydropower generator similar to the generators at H1 and H2. Notice the size of the person walking in the left part of the picture. The generator bearings placed are placed above and below the rotor.

The hydropower units are balanced and optimized for a spin speed at its driving frequency. Spin speeds below $\omega_{dr}$ can cause high loads in the generator bearings. Stationary operation is there for more lenient for the hydropower unit. Starts and stops wear on the bearings and other parts in the surroundings that are sensitive for vibrations. In the last decade the amount of start and stops has increased for the hydropower units. The production of electrical energy is more optimized today than it was 10 years ago and this result in a higher number of starts and stops.

A more specific description for the loads acting on the generator baring and its characteristics will be presented later in this report.

### 2.1.2 Damping and Stiffness in Generator Bearings

This report presents a method to calculate the damping and stiffness in radial generator bearings. The damping and stiffness for the generator bearing will be calculated in the generator axles static position. Depending on the load case the axles static position varies. The bearing and the forces acting on it can be describes as a simple two degree of freedom system, 2-DOF-system, Figure 2.3.
Figure 2.3. Schematic Figure of a 2-DOF system

The axle has its centre of displacement at the distance $e$ from the centre of the bearing. Around that centre of displacement the axle rotates with a dynamic eccentricity. The static load acting on the generator bearing causes the static displacement, $e$, in the generator bearing. The static loads dependent of the load cases over the turbine and the air gap in between rotor and stator in the generator. These loads mainly cause the static and dynamic displacements in the generator bearing. The size of the dynamic loads, displacements and displacement velocities determines the damping and stiffness for the generator bearing in the static position for the generator axle. Different types of bearings have different damping and stiffness properties. The most common types of generator bearings in hydropower units are tilting pad bearings, see chapter 2.1.3. All three of the hydropower units analyzed in this report are equipped with that type of bearings. The method to calculate bearing data that is developed in this report will there for be adapted for tilting pad bearings.

To be able to calculate the bearing properties for these bearings the loads on and displacements in the generator bearing need to be known. Also the velocity of the displacement and the phase between force, displacement and displacement velocity need to be known. Each of spoke in H1, H2 and H3’s generator brackets are equipped with strain gauges. The strain gauges register the strain in each spoke. Figure 2.4 shows a schematic picture over the generator bracket and the angles between the spokes.
From the strain in the spoke, the spokes geometry and its material properties the resulting load acting on the bracket can be calculated. The loads in the generator bracket are calculated according to Equation 2.2 and 2.3 [1] and these are divided in $x$ and $y$ components.

\[ F_x = -E \cdot A \sum_{i=1}^{n} \varepsilon_i \cdot \cos(\varphi \cdot i) \]  
\[ F_y = -E \cdot A \sum_{i=1}^{n} \varepsilon_i \cdot \sin(\varphi \cdot i) \]

$E$ is Young’s modulus, $A$ is the cross section area for the spokes, $\varphi$ is the angle between the spokes, $\varepsilon_i$ is the strain in the spoke and $n$ is the number of spokes. In the hydropower stations the spokes of the bracket consist of I-beams. The number of spokes and its cross section area varies. The upper generator bracket in hydropower unit H3 consist of 12 spokes and its cross section area is approximately 25000 mm$^2$, the lower bracket consists of 8 spokes with a cross section area of approximately 40000 mm$^2$. H1 and H2 are larger power units and the upper generator brackets have 18 spokes and cross section area of approximately 25000 mm$^2$. 

Figure 2.4. *Schematic picture over the generator bracket*
Figure 2.5. Load and displacement in generator bearing, x-direction.

The generator brackets also has mechanical properties that affect the properties for the generator bearing. These properties can easily be determined, the stiffness is calculates form Equation 2.4 \[2\] and the damping only consists of structural damping and can there for be neglected.

\[
k_B = \frac{AE}{L} \sum_{i=1}^{n} \cos(\phi \cdot i)
\]  

(2.4)

$L$ is the length of the spoke and the other constants were declared above. The vertical stiffness in the I-beam can be neglected.

Figure 2.5 shows the force acting on the generator bracket is equal to the force acting on the generator bearing. $F_x$ and $F_y$ represents also the forces acting on the generator bearing. One advantage with this method of measuring bearing loads is that there is no need to disassemble the generator bearing. In conventional load measurements at generator bearing are load cells assemble between the bearing pads and the bearing bracket. Loses in production costs a lot of money and to replace the existing bearing with a bearing equipped with load cells takes a lot of time.

As mentioned previously in this chapter the forces and displacements in the generator bearing can be described as a simple 2-DOF system. The equation of motion of that system is given by Equation 2.5

\[
\begin{bmatrix} F_x(t) \\ F_y(t) \end{bmatrix} = \begin{bmatrix} k_{11} & k_{12} \\ k_{21} & k_{22} \end{bmatrix} \begin{bmatrix} x(t) \\ y(t) \end{bmatrix} + \begin{bmatrix} c_{11} & c_{12} \\ c_{21} & c_{22} \end{bmatrix} \begin{bmatrix} \dot{x}(t) \\ \dot{y}(t) \end{bmatrix}
\]  

(2.5)

$x(t)$ and $y(t)$ are the dynamic displacements relative the centre of rotation. According to Figure 2.5 is $x(t) = x_A - x_{BB}$ and the same relations applies for $y(t)$. $\dot{x}(t)$ and $\dot{y}(t)$ are the displacement velocities in each direction. The dynamic forces acting on the system are divided in to $x$ and $y$ components, $F_x(t)$ and $F_y(t)$. The parameters left to determine in Equation 2.5 are the stiffness terms, $k_{11}$ and $k_{22}$, the stiffness cross coupling terms, $k_{12}$ and $k_{21}$, the damping terms, $c_{11}$ and $c_{22}$, and the cross coupled damping, $c_{12}$ and $c_{21}$.

\[
F_x(t) = k_{11} \cdot x(t) + k_{12} \cdot y(t) + c_{11} \cdot \dot{x}(t) + c_{12} \cdot \dot{y}(t)
\]  

(2.6)

\[
F_y(t) = k_{22} \cdot y(t) + k_{21} \cdot x(t) + c_{22} \cdot \dot{y}(t) + c_{21} \cdot \dot{x}(t)
\]  

(2.7)

Equation 2.6 and 2.7 are equivalent to Equation 2.5 in non-matrix form. To solve the bearing data for the system described in Equation 2.6 and 2.7 the equation system that
follows must be solved. The known parameters are \( F_x(t), F_y(t), x(t), y(t), x'(t), y'(t) \) and the bearing data are the unknown parameters.

\[
\mathbf{F}_x = \begin{bmatrix} F_{x_1} \\ F_{x_2} \\ F_{x_3} \end{bmatrix} \quad \mathbf{XY}_1 = \begin{bmatrix} x_1 & y_1 & \dot{x}_1 & \dot{y}_1 \\ \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot \\ x_4 & y_4 & \dot{x}_4 & \dot{y}_4 \end{bmatrix} \quad \mathbf{KC}_1 = \begin{bmatrix} k_{11} \\ k_{12} \\ c_{11} \\ c_{12} \end{bmatrix}
\]

\( \mathbf{KC}_1 = \mathbf{XY}_1^{-1} \cdot \mathbf{F}_x \) (2.8)

\( \mathbf{KC}_1 \) contains the damping and stiffness parameters \( k_{11}, k_{12}, c_{11} \) and \( c_{12} \). The damping and stiffness parameters \( k_{22}, k_{21}, c_{22} \) and \( c_{21} \) are achieved by solving \( \mathbf{KC}_2 \) in Equation 2.9.

\[
\mathbf{F}_y = \begin{bmatrix} F_{y_1} \\ F_{y_2} \\ F_{y_3} \end{bmatrix} \quad \mathbf{XY}_1 = \begin{bmatrix} x_1 & y_1 & \dot{x}_1 & \dot{y}_1 \\ \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot \\ x_4 & y_4 & \dot{x}_4 & \dot{y}_4 \end{bmatrix} \quad \mathbf{KC}_2 = \begin{bmatrix} k_{21} \\ k_{22} \\ c_{21} \\ c_{22} \end{bmatrix}
\]

\( \mathbf{KC}_2 = \mathbf{XY}_1^{-1} \cdot \mathbf{F}_y \) (2.9)

Tilting pad bearings with an even numbers of pads are assumed to be symmetrical [6]. This results in: \( k_{11} \approx k_{22} = k; k_{12} \approx k_{21} = k; c_{11} \approx c_{22} = c; c_{12} \approx c_{21} = c \). The cross coupling terms in tilting pad bearings are also assumed to be neglectable [6]. \( k \), and \( c \) are set to 0.

For rotating machinery is Equation 2.7-9 better described in polar form. The damping and stiffness of the bearing will then be calculated according to the shafts angular position. The new relation between load, displacement, stiffness and damping that applies for the system are:

![Figure 2.6. Relation between forces in the generator bearing.](image-url)
\[ F_k = r \cdot (k_{11} \cos(\omega t + \phi) + k_{22} \sin(\omega t + \phi)) \]
\[ \phi = \text{phase between force, } F(t), \text{ and displacement, } D(t) \]
\[ F_c = r \cdot (c_{11} \cos(\omega t + \theta) + c_{22} \sin(\omega t + \theta)) \]
\[ \theta = \text{phase between } d(F(t))/dt \text{ and } d(D(t))/dt \]

\[ F^2 = k^2 r^2 + c^2 \dot{r}^2 + 2 \cdot kcr \dot{r} \cos(\phi - \theta) \] (2.10)

\[
\mathbf{F}_{\text{ext}} = \begin{bmatrix} F_1^2 \\ F_2^2 \\ F_3^2 \end{bmatrix}, \quad \mathbf{r}\phi = \begin{bmatrix} r_1^2 \\ r_2^2 \\ r_3^2 \end{bmatrix}, \quad \dot{\mathbf{r}} = \begin{bmatrix} \dot{r}_1 \\ \dot{r}_2 \\ \dot{r}_3 \end{bmatrix}, \quad \mathbf{KC}_4 = \begin{bmatrix} k^2 \\ c^2 \\ 2 \cdot k \cdot c \end{bmatrix}
\]

\[ \mathbf{KC}_4 = \mathbf{r}\phi^{-1} \cdot \mathbf{F}_{\text{ext}} \] (2.11)

The first element in \( \mathbf{KC}_4 \) are the stiffness squared for the generator bearing in a certain angle. The second element in the array are the damping squared.

### 2.1.3 Tilting Pad Bearing

The upper and lower generator bearings in the hydropower units are tilting pad bearings. These bearings are symmetric bearings with an even number of pads. Generally, radial tilting pad bearings superior stability characteristics compared to other radial bearings. In rotating machineries that requires maximum stability are therefore tilting pad bearings are often used. The high rotodynamic stability comes from the reduction of the cross coupled damping and stiffness that occurs when the pads are free to tilt about their individual pivot points. There exist two types of tilting pad bearings, rocker-pivot and spherical-pivot. The pads in the rocker-pivot bearing can pivot about the axial coordinate, while the pads in the spherically-pivoted bearing can pivot both the axial and tangential coordinates. A spherical-pivot bearing has therefore a higher tolerance to shaft misalignment and also tends to be more durable than the rocker-pivot bearing. The tilting pad bearings in the analyzed hydropower units are spherical-pivot pad bearings.
2.2 Signal and Sampling Theory

When it comes to measure and sample signals there are one very important and basic rule that applies, the sampling theorem [3]. According to the sampling theorem the sampling frequency, $f_s$, must be at least twice as high as the highest frequency contents in the signal of interest, $f_{signal,max}$. The signal must be lowpass filtered below half the sampling frequency, otherwise aliasing can arise [5]. It is therefore important to know and limit the frequency contents of the interesting part of the measured signal. The sensors used in the measurements must also be well adapted for the type of measurement they are to perform. The most important factors are the sensors maximum sample rate and its resolution. To be able to analyse the characteristics of the measurements, the measured data should be generated for at least 15 periods of the lowest frequency of interest. In hydropower units are often $\omega_{dr}$ the lowest frequency of interest, turbulence below the turbine can however cause bearing loads that consist of frequency components below $\omega_{dr}$. The total numbers of samples in the measured data file must then at least be $15*\frac{f_s}{\omega_{dr}}$ samples.

2.2.1 Aliasing

If a measured signal is undersampled, $f_s < 2 f_{signal,max}$, aliasing will appear. The Figures below shows the effects of aliasing. Figures 2.6a represents the signal $x(t) = \sin(2\pi \cdot 1.4t) + \sin(2\pi \cdot 52t) + \sin(2\pi \cdot 95t)$ in frequency domain, the signal is sampled at 100 Hz. The solid lines in the figure are the frequency representation of $x(t)$ and the dashed lines are where the aliased frequencies over $f_s/2$ will appear. Figure 2.7b shows the frequency contents of the aliased signal, $x_a(t) = \sin(2\pi \cdot 1.4t) + \sin(2\pi \cdot (f_s - 52) \cdot t) + \sin(2\pi \cdot (f_s - 95) \cdot t)$, when $x(t)$ is sampled at 100 Hz.

![Figure 2.7a. Effect of undersampling in the frequency domain.](image)
Figure 2.7b. Result of undersampling in the frequency domain.

Figure 2.7a,b represents the original signal and the signal that the undersampling will results in. The appearance of these two signals is quit different but if these signals are sampled at 100 Hz the data file will look exactly the same. In Figure 2.8 b are the samples that will appear in the data file are circled.

Figure 2.7a. $x(t)$ and $x_A(t)$ in time domain.
Figure 2.7b. *Effect of undersampling in the time domain.*

The figures above shows the importance of a correctly chosen sampling frequency and the importance of knowing the frequency characteristics of the measured object.

### 2.2.2 Other Existing Error Sources

Even if the measurements are correctly performed and equipments used in the measurements are well suited for the measurement, the results can still be hard to interpret. It’s very difficult to predict how the environment where the measurements are performed in will affect the measurements. In measurements performed in hydropower stations everything vibrates due to the hydropower units rotating mass of 500 – 1000 tons. The vibrations generated by the rotating machinery also cause vibrations in the floor and walls. When the measurements are performed it’s difficult to measure the absolute vibrations and displacements, but the surrounding structure are assumed to be relatively still. The generator of the hydropower unit are huge, it generates hundreds of MW in electrical power. The huge magnetic field causes vibrations of the generator and the fields affect the equipment used for the measurements.
3 Approach / Accomplishments

3.1 Method for Measuring Vibrations

From the measurements performed at the hydropower units the displacements of the generator axle and strains in the spokes of the generator brackets were analysed. To measure the displacement of generator axle laser sensors and inductive sensors were used. The generator axles in these hydropower units are cylindrical axles with a 1-2 meters diameter and rotate normally with \( \omega_{dr} = 1-2 \) Hz. The static and dynamic displacements of the generator axle are normally with in 0.5 mm, extreme load cases can cause higher displacement amplitudes.

The strains in the generator brackets spokes are measured with strain meters. These strain meters register the instantaneous strains in the spokes. As mentioned earlier in the report the spokes in the generator brackets consist of a number of I-beams. The number of I-beams depends of the size of the hydropower unit and the cross section area of the I-beam. These beams have often relatively large cross section areas and to achieve noticeable strains in these beams high axial loads in the beam are needed. From the strain in the spokes the bearing load is calculated according to Equation 2.1-2. Depending of which hydropower unit the measurements are performed on and which load case the hydropower unit is exposed to the strains in the spokes varies between 1-10 \( \mu \)-strain. H3 runs very smooth and has very low strains in the spokes of the generator bracket. H2 was balanced in 2002, before the balancing were static and dynamic strains above 10 \( \mu \)-strain registered in the bracket. That together with a cross section area of approximately 25000 mm\(^2\) for each of the 18 spokes results in huge loads acting on the generator bearing. These huge loads caused the generator bearing to break down despite the fact that the bearing was only 3 years old.

To calculate the dynamic load that acts on the bearing, a mean strain in each spoke is calculated, \( \bar{\epsilon} \), and this is done for each measurement. The mean strain for each spoke is subtracted from the instantaneous strain in each spoke. Then the dynamic load is calculated, see Equation 3.1-2. It’s the same procedure to calculate the dynamic displacement in the generator bearing. The mean displacements of the measurement are subtracted from the measured displacements.

\[
F_{dx} = -E \cdot A \sum_{i=1}^{n} (\epsilon_i - \bar{\epsilon}) \cdot \cos(\varphi \cdot i) \tag{3.1}
\]

\[
F_{dy} = -E \cdot A \sum_{i=1}^{n} (\epsilon_i - \bar{\epsilon}) \cdot \sin(\varphi \cdot i) \tag{3.2}
\]

To calculate the static strain in the generator bearing the strains in the spokes must be subtracted with a reference measurement. The reference measurement is a measurement performed when there are almost no load at the turbine and no magnetic load over the generator. When the power unit rotates slower than 15 % of its \( \omega_{dr} \), the brakes are activated. The reference measurement must therefore be performed just above that speed. Another problem in the calculations of static load is that the thermal expansions effects the measurements. The strain levels in the I-beams, caused by the static loads, are of the order of \( \mu \)-strain. The temperature changes in power units can cause thermal expansions in the beams of the same order. The reference
measurements are therefore performed close to the time of the measurement of the actual load case. The strains in the measurement of the actual load case are subtracted to the calculated mean strain of the reference measurement, see Equation 3.3-4. To calculate the static displacement the measured mean displacement is subtracted to the mean displacement of the reference measurement. In the calculations of the static displacement there is no need to consider the thermal expansion, its affect on the results insignificant.

\[
F_{sx} = -E \cdot A \sum_{i=1}^{n} (\varepsilon_i - \bar{\varepsilon}_i) \cdot \cos(\varphi \cdot i) \\
F_{sy} = -E \cdot A \sum_{i=1}^{n} (\varepsilon_i - \bar{\varepsilon}_i) \cdot \sin(\varphi \cdot i)
\]  

(3.3)  
(3.4)

\( F_{sx}, F_{sy} \) represents the static load in the generator bracket, \( \varepsilon_z \) is the strain in spoke \( i \) at the reference measurement.

### 3.1.1 Strain Gauges

Hottinger Baldwin Messtechnik, HBM, manufactured the strain gauges used to register the strain in the spokes of the generator brackets. A strain gauge is a type of resistor and its resistance changes when the strain meter is stretched, see Figure 3.1. If the copper wires in the strain gauge is stretched the strain gauges total resistance increases. The change in resistance is linear to the strain gauges change in length. This relation only applies the strains that the strain gauge is specified for. The strain gauges resistance is 350 ± 0.35 % ohm and its gauge factor, \( k \), is approximately 2. The gauge factor varies for each strain gauge.

![Strain gauge](image)

Figure 3.1. Strain gauge (measure strain in y-direction).

The strain gauges are configured in a bridge formation, a full Wheatstone bridge. Wheatstone bridge is especially useful to measure small changes in resistance.
Two different bridge configurations were used in the analyzed hydropower stations. The generator brackets spokes at H3 has two strain gauges connected to the side of the spoke, \( R1 \) and \( R3 \). The other two strain gauges, \( R2 \) and \( R4 \), are connected to a separate metal plate. One end of this metal plate are fixed to the spokes, the other end is left unconnected. The strain gauges bonded on the lose metal plate will only register the thermal expansion of the plate, non of the strains cause by the loads acting on the generator bearing. The strain gauges bonded on the lose plate will there for compensate for the thermal expansion affecting the spokes and the relation between \( V_I \) and \( V_U \) will only depend of the loads on the generator bearing.

At H1 and H2 strain gauges \( R1 \) and \( R3 \) has the same function as in H3. The other two strain meters, \( R2 \) and \( R4 \), are replaced with two precision resistors with specifications \( 350 \pm 0.1\% \) ohm. A disadvantage with this bridge configuration could be that they don’t have the same differential amplification properties as the bridge used in H3. The disturbances that the strain gauges are sensitive for will affect all the strain gauges in
H3 and will therefore not be amplified. If the disturbances that were critical for the strain gauges doesn’t affect the resistors \( R2 \) and \( R4 \) in H1 and H2, the disturbance will be amplified and will affect the measurements. Most of the disturbances that affect the strain gauges are relatively high frequencies compared to the interesting contents of the measures signal. The different in results between these two bridge configurations becomes therefore insignificant. For both bridge configurations the following equations applies:

\[
\Delta V_U = \frac{\Delta R_1 - \Delta R_2 + \Delta R_3 - \Delta R_4}{4 \cdot R} \cdot V_i ; R = R_1 = R_2 = R_3 = R_4
\]

(3.5)

\[\varepsilon = k \cdot \Delta V_U \]

(3.6)

Results from measurements performed with the Wheatstone bridge are shown in the Figure 3.4. The interesting part of the signal is in the frequency range between 1 – 10 Hz. The first peak at approximately 1.4 Hz represents the power units \( \omega_{dr} \). Two more peaks, the first and second multiples of \( \omega_{dr} \) follows. Higher up in the frequency spectra two more peaks are found, one at 50 Hz and another at 100 Hz. These two peaks are assumed to be disturbances. From these measurements the loads in the generator bearing are calculated according to formula 2.1-2.

![Figure 3.4. Strain in the upper generator brackets spoke. Measured at H2 and sampled at 500 Hz](image)

A probable cause of the disturbances at 50 and 100 Hz are the long straight electrical wires that extend from the strain meters to the data collection unit. These wires also passes by the generator.

### 3.1.1.1 Dummy

As mentioned in the previous chapter, the 50 and 100 Hz strains registered in the generator bracket spokes was assumed to be disturbances. To verify these assumptions a dummy was constructed. The dummy is a full Wheatstone bridge with all the strain gauges are connected to a loose steel plate. The dummys task is to only register the electrical disturbance that affects the strain gauge, no strains are to be registered. The steel plate is placed near one of the Wheatstone bridges that are fixed to one of the generator brackets spokes. One end of the steel plate is fixed to the spoke, the other ends are left lose. The dummy should register the same disturbances...
as the Wheatstone bridge on the spoke, except for strains caused by the load on the generator bearing. The dummy will only register disturbances and thermal expansions. The thermal expansion has a very low frequent characteristic and that can be seen in Figure 3.5. It has two obvious peaks at 50 and 100 Hz and some low frequency content.

The peaks at 50 and 100 Hz are most likely disturbances and the low frequent content of the signals results from the thermal expansion in the steel plate. Except for these peaks in the frequency spectra, the measured dummy signal is very smooth.

3.1.2 Displacement Sensors

The static and dynamic displacement of the generator axle needs to be determined. The total mass of the rotating structure is over 1000 tons. Large displacements at high frequencies do not exist in the rotating structure of hydropower units. The measured signal should consist of frequencies related to the driving frequency of the turbine.

3.1.2.1 Inductive Displacement Sensors

One of the sensor types that were used during the displacement measurements of the generator axle is an inductive displacement sensor manufactured by Contrinex. This sensor type used is characterized by a large sensing range and good accuracy. As seen in Figure 3.6, the transmission behavior for the sensor isn’t entirely linear. In the measurements is the stationary point set to a distance of 3 mm. As long as the power unit is running as it should the dynamic displacement is kept below 1 mm. The sensor is relatively linear in the displacement region 2.5 – 3.5 mm and is there for assumed to be linear.
The results from the inductive displacement sensors were not satisfying. Frequency characteristics for displacement measurements the sensors are shown in Figure 3.7.

It’s not likely that the frequency characteristics that Figure 3.7 shows is the true characteristics of the generator axles displacement. The hydropower unit rotates with 1-2 Hz and weighs approximately 1000 ton and is therefore not likely to have high frequency variations. For each multiple of $\omega_{dr}$ there is a peak in the frequency spectra. These peaks are probably caused by the huge magnetic fields that affect the inductive sensors. In this measurement the sampling frequency is 500 Hz and no aliasing filter are used. It’s not likely that these peaks are the results of aliasing. The aliasing should be dependent of the sampling frequency and not the driving frequency of the power unit. Also measurements sampled at 460, 230 and 100 Hz has the exact same frequency characteristics. A turn is used in the production of the generator axle, during the turning of the axle can periodic irregularities arise. Theses irregularities are periodic but will only cause occasional peaks in the frequency spectra.
Figure 3.8. *Time dependence in the frequency characteristics for the measured displacement in the upper generator bearing in H2. Displacement measured with inductive sensors and sampled at 500 Hz.*

The strange peaks that occurs in the displacement measurement with the inductive sensors are also continuous through time, see Figure 3.8. Other possible sources of errors are that the inductive sensor is too big. The displacements in the upper generator bearing are measured at the top of the generator axle. The generator axle cover only the half of the inductive sensor and the top of the generator axle are also a bit uneven. Figure 3.9 shows the top of the generator axle and the inductive sensors.
If the frequency characteristics for the upper and lower generator bearing are compared obvious similarities can be seen. The displacements measured at the lower bearing are measured at the middle of the generator axle. The similarities in the frequency characteristics can there eliminate that the size of the inductive sensor causes errors in the measurement. The irregular frequency characteristics must be explained by the sensors sensitivity of large magnetic fields and are less appropriate to perform displacement measurements in hydropower units.

### 3.1.2.2 Laser Sensors for Displacement Measurements

At the last measurements at H2 the displacements of the generator axle were measured with both laser and inductive sensors. The similarities between the laser sensors and the inductive sensors were investigated. Both sensors were placed at the same position on the generator bracket. The frequency characteristics of the measurements performed with the laser sensors were a lot more pleasant than the frequency characteristics of the inductive sensors, see Figure 3.10.
Figure 3.10. Frequency characteristics for the displacement in the upper generator bearing at H2. Displacement measured with laser sensors and sampled at 500 Hz.

Figure 3.10 shows an obvious peak at the frequency that represents the generator axles rotating frequency $\omega_{dr}$, some lower peaks at the first and second multiples of $\omega_{dr}$ and also some small disturbances at the higher frequencies. The similarity between the displacements frequency characteristics, Figure 3.10, and the strains frequency characteristics, Figure 3.4, is obvious. One decisive difference between the frequency characteristics in the strain and displacement measurements are the 50 and 100 Hz disturbances. The laser sensor does not show any sensitivity for the magnetic fields that surrounds the generator. If Figure 3.10 is carefully studied a small peak at 50 Hz is found. This peak probably represents the true vibration that the huge magnetic field causes.

If it is assumed that all the important contents of the measured displacements are below 10 Hz and all other frequencies can be neglected. The similarities between the displacements measured with the inductive and laser sensors will then be evident. See Figure 3.11. The displacements measured with the laser sensors shows a smoother orbit.
3.1.3 Data Acquisition Hardware

At the measurements in H2 a new data acquisition unit was used. This data unit was constructed to perform measurement at Akkats hydropower unit. Akkats had a serious break down in the summer of 2002 and the restoration of the hydropower unit took over two year. The total expenses of these mechanical break down ended at several hundred millions and to prevent these failures to happed again an advanced data acquisition unit was built. Before Akkats hydropower station were put in action again a lot of measurements were performed at the hydropower unit. The new data acquisition unit was used to collect all the data during the measurements.

This unit consists of two identical data acquisition units, each with 96 channels. There are two different investor that financed the built of the data unit. After the measurements at Akkats are finished the unit will be split in to two separate 96 channels units. The first 32 channels on each unit consist of four National Instruments SCXI-1140. SCXI-1140 is an 8-channel simultaneous sampling differential amplifier module. These are equipped with sample and hold function and can handle a sample rate up to 20 kHz. The first two SCXI-1140 cards are also equipped with each SCXI-1143 card. SCXI-1143 is an 8-channel programmable lowpass filter modules that is ideal for antialiasing applications. The filter type in SCXI-1143 is an 8th order Butterworth filter with a filter range of 10 Hz to 25 kHz. The remaining 64 channels on each unit are equipped with two sets of SCXI-1102B cards. The SCXI-1102B is a 32-channel analog input module. It has a very high maximum sampling rate, 333 kS/s,
and is equipped with antialiasing filters. The filters are RC-filters and have fix cut of frequency at 200 Hz. All these SCXI modules are housed in the SCXI-1001 chassis. More information of these components from National Instruments can be found at their homepage [4].

In other words, the measuring unit has 196 channels, 64 of these have a sample and hold function and 32 of these channels also has an adjustable lowpass filter. The 132 channels that hasn’t the sample and hold function has a fixed 200 Hz lowpass filter. Schematic block diagram for the acquisition unit is shown in Figure 3.12.

Figure 3.12. One of the two equivalent data collection units used at the measurements performed at hydropower unit H2.

At the measurements performed at H2 channel 17-31 and 65-80 was used. Channel 17-30 measured displacements in the generator bearing, accelerations at the generator bearing and pressure in the tube. For the strain measurement channel 31 and 65-80 was used. The reason for this channel distribution is unknown, but channels with adjustable lowpass filters were deliberately avoided.

At the measurements performed at H3 hydropower station an old data collection system was used. It was a 96-channels unit based on three SCXI-1100. SCXI-1100 is a 32-channels analog input module. Handles up to 240 kS/s at full bandwidth and has no antialiasing filter.

3.1.3.1 Sampling

Some the measurements that were performed at hydropower unit H1 and H2 were sampled at 100 Hz. The reason why this sampling frequency was used is unknown but it is less appropriate for these measurements. The cut off frequency of the antialiasing
filter is twice the sampling frequency, the antialiasing filter is therefore useless in these measurements. As discussed chapter 3.1.1 has the strain measurements has obvious disturbances at 50 and 100 Hz. That combined with no correctly adapted antialiasing filter and a sampling frequency at 100 Hz enable aliasing to arise. The disturbances in the frequency region just below 100 Hz should be critical for the measurements sampled at 100 Hz. The aliasing that these disturbances causes should arise in the frequency region 0-5 Hz, i.e. the frequency region where the most important part of the measured signal is represented.

To analyse the aliasing effect on the measured data two different dummy measurements was studied. One measurement sampled at 100 Hz and the other was sampled at 500 Hz. In the measurement sampled at 500 Hz shouldn’t disturbances around 100 Hz affect the low frequency contents of the measured signal. These two measurement are shown in Figure 3.13 and 3.14.

![Figure 3.13. Measured strain in the dummy sensor sampled at 100 Hz.](image)

![Figure 3.14. Measured strain in the dummy sensor sampled at 500 Hz](image)

The measurement sampled at 100 Hz shows a high representation of low frequent signals, especially in the frequency region 0-2 Hz. In the other measurement, the one sampled at 500 Hz shows only a small peak at $\omega_{dr}$, approximately 1,4 Hz. The high concentration of the low frequent signal in the 100 Hz measurements is probably caused by aliasing. The 100 Hz measurements can also be affected by thermal expansions in the steel plate and that causes also an increased concentration of low frequent signals. On the basis of the test schedule from the measurements should the machine have similar thermal conditions.
Figure 3.15. Measured load in H1 upper generator bearing. Both measurements are performed at the magnetic load of 210MW. The channel used for the data acquisition is equipped with a 200Hz antialias filter and the measured data is low pass filtered at 2Hz.

In the last measurements performed at H1, two measurements with identical load cases were performed. The only differences between the measurements were the sampling frequency. The measurements were performed at a magnetic load of 210 MW and both measurements were performed relative close in time. Figure 3.15 shows the results of the measurements. The channel used for the data acquisition is equipped with a 200Hz antialias filter and the measured data is low pass filtered at 2Hz. The measured loads in the measurement sampled at 200 Hz is allot smaller than the measurement sampled at 100 Hz. Due to incorrect settings of the antialias filter is it possible that the disturbances just below 100Hz can affect the measurements, see Figure 3.16. The thermal expansion can also affect the load levels, however since both of the measurements were performed within one and a half hour on a warm machine the thermal expansion effect should be insignificant.
Figure 3.16. Visualization of incorrect antialiasing filtering. The “true” signal is \( x(t) = \sin(2\pi 1.4t) + 0.5\sin(2\pi 98.6t) \). If the measured signal is sampled at 100Hz (black circles in the figure) without the antialiasing filter the result will be \( x_{100Hz} = 1.5\sin(2\pi 1.4t) \). If the measured signal is sampled at 100Hz and 200Hz and both are low pass filtered at 2 Hz the result will be \( x_{100Hz} = 1.5\sin(2\pi 1.4t) \) and \( x_{200Hz} = 1\sin(2\pi 1.4t) \).

Better relation between sampling frequencies and antialias filters were used at the measurements performed at hydropower unit H3. The sampling frequencies of these measurements vary between 230–720 Hz. There are no noticeable disturbances above 100 Hz but to avoid aliasing the sampling rate of the measurements should be at least twice the frequency of the antialiasing filter i.e. sampling frequencies above 400 Hz. The disadvantages with these high sampling frequencies are the size of the data file. At least 15 periods of the \( \omega_{dr} \), approximately 1-2 Hz, and a sampling rate of 720 Hz generates a lot of samples. It is also appropriate to down sample the measured data afterwards.
3.2 Software Filtering in MATLAB

All of the measurements performed at the hydropower units are more or less affected by the disturbances in its surroundings. As seen previously in this report the inductive sensors are more sensitive to disturbances. But even the measured data from the other sensor have a need to be filtered. See Figures 3.17-18.

Figure 3.17a. Displacement measured with inductive sensors
Figure 3.17b. Displacement measured with laser sensors

Figure 3.18. Load in the upper generator bearing calculated from measurements performed with strain meters.

The filtering of the measured data is performed in MATLAB. The Filter Design and Analysis Tool (FDATool) is used to design the filters. It’s an interface for designing and analyzing filters. In FDATool it is possible to design digital FIR or IIR filters by setting filter performance specifications, by importing filters from your MATLAB workspace, or by directly specifying filter coefficients.
The filtering will not be performed in real time and the processor speed of the computer that performs the filter computations is relatively high. A higher order FIR filters is therefore designed and used for filtering.

To determine which part of the signal that should be filtered away, analysis of the dummy signal is useful. The dummy is supposed to only register disturbances that affect the strain meters.

### 3.2.1 Dummy Characteristics

The conclusions that can be made from the measurements performed with the dummy sensor are that the strain gauges are sensitive for disturbances. In hydropower units the huge magnetic fields at the frequencies 50 and 100 Hz are critical for the strain measurements. The dummy was fixed at a spoke, spoke 13, during the measurement at H2. It’s assumed that the magnetic field that affects the Wheatstone bridges at the other spokes is the equal to the effect it has on the Wheatstone bridge in spoke 13. The 50 and 100 Hz contents in all the measured strain signals is therefore assumed to be disturbances and should be filtered away. In Figure 3.9 a small peak at 50 Hz was discovered despite the laser sensor is assumed to be insensitive to the magnetic disturbances. If it is a small 50 Hz displacement in the generator bearing is it a high probability that the generator brackets spokes also should experience some small 50 Hz strains. Still these loads are assumed to be very small and the disturbances are very big which makes it impossible to separate the true 50 Hz signal form the disturbance.

### 3.2.2 Filtering Results

For the measurements performed in H2 a special filter is created. The filter is created from the frequency characteristics of the measured displacements, strains and the dummy signal. As seen in Figure 3.4 and 3.9 the most dominant frequencies except for the 50 Hz disturbances is the driving frequency of the turbine and its first two
multiples (1.4, 2.8, 4.2 Hz). To eliminate noise and other disturbances at the signal is filtered with a Multi Band Pass (MBP) filter with band pass frequencies between 1.2-1.6, 2.6-3.0, 4.0-4.4 Hz. Figure 3.20 presents the magnitude and phase response for the filter.

Figure 3.20. MBP filter characteristics.

The filter is designed for the 100 Hz measurements at H2 and the order of this filter is 2000. That’s a very high filter order and results in a very steep filter. The steepness in the filter is needed due to the signal is sampled at 100 Hz and the interesting part of the signal lies between 1 – 5 Hz. The alternative to the high filter order is to down sample the signal. It is very easy and quick to create filters of high order in FDATool, and because of this, the measured data isn’t down sampled.

To design a filter with this characteristic for the measured data sampled at 500 Hz, a filter of extremely high order is needed and impractical to implement. To be able to create a filter of reasonable order, the measured data need to be down sampled with at least a factor of 5. The data sampled at 500 Hz doesn’t contain enough data to be down sampled that much, the data acquisition time was to short. Despite the five time higher sampling frequency contains the data file sampled with 500 Hz less data than the data file sampled at 100 Hz.

The results of the filtration of the data sampled at 100 Hz are shown below. Other types of filter were also used but the MBP filter shown in Figure 3.20 gave the best result.
Figure 3.21. Unfiltered and filtered frequency characteristics for the load in H2 upper generator bearing.

Figure 2.22. Unfiltered and filtered frequency characteristics for the displacement in H2 upper generator bearing.


4 Results

In this chapter are two of the measurements performed at H2 and one of the measurement performed at H1 analysed. One of the measurements from H2 was performed before the generator was balanced and the other measurement was performed after the balancing. A measurement performed at H3 in 2002 was also analysed. The purpose with the measurement performed at H3 in 2002 was to evaluate a method to measure bearing loads using strain gauges [1]. The loads and displacements in the generator bearings will be analysed and its damping and stiffness will be calculated.

4.1 Displacements and Loads in Generator Bearings

The size of the loads and displacements that arise in the generator bearings are caused by many different parameters. The size of the hydropower unit, the size of its generator and turbine has of cause an effect on the load that acts on the generator bearing. Unbalance and defects on the turbine and generator also has an evident affect on the size of the loads and displacements in the generator bearings. Type of turbine, flow through the turbine and magnetic loads over the generator are other parameter that affects the loads and displacements. The relation between the displacement and loads are dependent of the damping and stiffness in the bearings.

4.1.1 Loads and Displacements in H3

As mentioned earlier in this report the hydropower unit H3 is equipped with a propeller turbine. The difference between a propeller turbine and an ordinary Kaplan turbine is the ability to adjust the angle of the turbine blades. The blades are fixed for a propeller turbine and adjustable for the Kaplan turbine. The blade angle at H3 propeller turbine is optimized for full flow through the turbine.
If it isn’t full flow through the turbine turbulence can arise below the turbine. This turbulence below the turbine is called vortex rope and causes higher loads on the
generator bearings. The vortex rope affects especially the lower generator bearing. Figures 4.1-2 shows loads in the generator bearings and its dependence of different load case at the turbine. The lower generator bearing experiences a lot higher loads when the load over the turbine is lowered from 100% to 75%. If the frequency characteristics for these load cases are analyzed a peak at 0.63 Hz is found, see Figure 4.3 and 4.4. This peak grows when the load over the turbine are lowered from 100% to 75%. This peak represents the loads caused by the vortex rope.

Figure 4.3. Frequency characteristics for the load on H3 lower generator bearing, full load over the turbine.

Figure 4.4. Frequency characteristics for the load on H3 lower generator bearing, 75% load over the turbine.

To minimize the loads on the generator bearings the load at the turbine should be 100% and loads around at 75% should be avoided. During starts and stops of the hydropower unit the critical loads at 75% are passed but only for a short period of time. The propeller turbine has five blades and that can be derived from the shape of the load figures, Figure 4.1 and 4.2, and the frequency characteristics for the load in the generator bearing. Both the figures over the bearing loads frequency
characteristics, Figure 4.3 and 4.4, show a peak at the frequency $5\omega_{dr}$. When the load over the turbine is at 75% are the loads caused by the vortex rope dominant. In this load case no four leaf flower shape can be seen in the load plot for the lower generator bearing. The four leaf flower formation in the load plots appears when the measured loads consist of the frequencies $\omega_{dr}$ and $5\omega_{dr}$. When full load is applied on the turbine these two frequencies becomes dominant and the flower shape orbit appears. At full load the load over the lower generator bearing also decreases to one third of the loads measured at 75% flow.

4.1.2 Loads and Displacements in H2

The other hydropower unit that was analyzed, H2, is a totally different hydropower unit. It is a lot larger power unit, has a Francis turbine and generates almost 3 times more power. The total mass of the rotating parts in H2 is approximately 1000 tons and the generators diameter is approximately 14 m. Unbalance in this huge rotating structure causes huge dynamic loads in the generator bearings. The loads on the generator bearing shown in Figures 4.5 and 4.6 shows the difference between a balanced and an unbalanced H2. Before the balancing in 2002 very large loads were measured in the generator bearing and the new generator bearing only lasted for 3 years. The normal lifetime for a generator bearing is at least 30 years. A new upper generator bearing replaced the old one and the rotating structure was balanced. A mass of 260 kg was needed for the balancing of the machine and the mass was placed at the circumference of the rotor.

![Dynamic load in upper generator bearing before balancing](image)
Before the hydropower unit was balanced the static and dynamic load levels in the generator bearing were approximately 30 tons. The balancing of the hydropower unit decreased the static and dynamic loads to 3-7 tons. This new load levels must be seen as normal for a hydropower unit of this size. The balancing of the hydropower unit will probably also result in a much longer lifetime for the generator bearing.

Figure 4.6. Dynamic load in upper generator bearing after balancing.
4.2 Calculated Damping and Stiffness in Generator Bearings

In chapter 2.1.2, *Damping and Stiffness in Generator Bearings*, a method to calculate bearing data is described. After the measurements have been filtered according the theory described previously, the bearing data is calculated according to Equation 2.13 and the results can be seen in Figure 4.7. Some of the calculated stiffness, $K$, is unreasonable high. If the condition number of the inverted matrix $r_{\phi_2}$ in Equation 2.13 is studied an obvious correlation between the high condition number and unreasonable values of $K$ can be found. The same relation exists between high condition numbers and unreasonable damping values.

![Stiffness normalized](image1)
![Condition number for matrix $r_{\phi_2}$](image2)

Figure 4.7. *Condition number of the $r_{\phi_2}$-matrix vs. calculated stiffness.*

The calculated damping and stiffness with high condition numbers are neglected and only the calculated bearing data with low condition number are of interest. The calculated damping and stiffness are described as a function of rotation angle. 90 degrees angle represents the y-axis and are up streams in the hydropower station. The mean damping and stiffness are calculated over 2.5 degrees intervals. This minimizes the influence of calculation errors that slipped through despite the elimination of samples with high condition number.
Figure 4.8. The stiffness in H2 upper generator bearing.

The calculated stiffness for H2 upper generator bearing is approximately $4 \times 10^9$ N/m, Figure 4.8. The stiffness in the bearing is almost the same in all directions. For the calculated damping is the pattern more irregular. The damping is approximately 10% of the stiffness and that gives a damping just below $4 \times 10^8$ Ns/m, Figure 4.9.

Figure 4.9. The damping in H2 upper generator bearing
The bearing data are calculated from measured data at full magnetic load over the generator bearing and full flow through the turbine. For this load case, the static displacement in the generator bearing is approximately 0.4 mm at the angle 250 degrees. The measure data are sampled at 100 Hz and this data file is chosen because it contains a lot of data.

Figure 4.10. *Static and dynamic displacement in H2 upper generator bearing.*

The static and dynamic loads are individual for each load case, also the static displacement of the generator axle changes in different load cases.

Hydropower unit H3 is a lot smaller than H1 and H2. Also the generator bearings in H3 are a lot smaller. Figure 4.11 shows the calculate stiffness for the upper generator bearing.
Figure 4.11. *Stiffness for upper generator bearing in H3.*
5 Conclusion

In this report the characteristics and properties for generator bearings in hydropower units were analyzed. A method to calculate bearing data from measured displacement and load in generator bearings were also developed. The damping and stiffness for a generator bearing in its static position is possible to determine with this method. Results of these calculations are shown in Figure 4.9-11 in the previous chapter. As seen in these figures the stiffness and damping plots are a bit irregular despite the measured data are averaged over 2.5 degrees intervals. To achieve smooth plots for the exact damping and stiffness the data files must be sampled at a high sample rate and for a longer period of time. The disturbances also need to be minimized and the filtering process needs to be optimized. But the need of knowing the exact damping and stiffness for generator bearing for a certain angle are relatively pointless. It is more useful to know the approximate amplitude of the bearing data, the bearing data’s change in characteristics depending of load cases and the shape of the bearing data depending of the angle of the generator axle. For these tasks the method described in this report is suitable and a relatively few strain and displacement samples are needed.

The most important part in achieving accurate bearing data using the method described in this report is the acquisition of the measured data samples. To measure the strain and displacement data correctly and use appropriate equipment during the measurement are two very important issues. The frequency characteristics of the inductive displacement sensors are bad and these sensor types are less appropriate for displacement measurements at generator axles. If displacement measurements are performed with both laser sensors and inductive sensors, the laser measurements should be used and the measurements performed with the inductive sensors are less important. In this report the importance of correctly chosen sampling frequencies and antialiasing filters was also analyzed. Some of the measurements performed at H1 and H2 were sampled at 100 Hz without antialiasing filter or with antialiasing filter with a cut off frequency above 100 Hz. That in a combination with a high concentration of disturbances at 50 and 100 Hz resulted in aliasing. Figure 3.15 in chapter 3.1.4.1 shows the difference between two equivalent load cases measured sampled at 100 and 200 Hz. The effect of the aliasing is obvious and a 100 Hz sampling frequency should be avoided and correctly defined antialiasing filters should be used. If aliasing has occurred when recording the data, it is impossible to regenerate a data file with correct data afterwards.

6 Future work

This thesis was focused on calculating bearing properties for tilting pad bearings. A suitable extension of the present work would be to investigate the possibilities to apply these bearing theories on regular journal bearings i.e. bearings with cross coupling terms. Moreover it would be interesting to investigate how successful these theories would be when applied to bearings with cross coupling terms.
7 References