Radial dampers impact on shaft vibration at resonance

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Abstract. During commissioning of a refurbished mid-size hydropower unit in Sweden, high shaft vibration occurred due to resonances. The refurbishment included a runner upgrade from 38 MW to 45 MW, a new stator and refurbished rotor poles. One of the requirements was that the dynamic behaviour of the machine should not be affected in a negative way despite a reduction in runner mass. In spite of the requirements, severe resonance problems became apparent during re-commissioning of the unit. Extended commissioning measurements were conducted to identify the cause of the high shaft vibrations. It was found that the resonance problem occurred when the machine reached 90% of its synchronous speed and the vibration levels increased with increased rotor speed; i.e., it was not possible to test the protection equipment for runaway. The runner was identified as the excitation source with an increased excitation frequency in the range of 5.4–7.4 Hz. Linear rotodynamic analysis of the rotor system was performed by the turbine and generator manufacturer as well by the owner of the unit. The analysis showed that the system has several natural frequencies with low damping in the 5.4–7.4 Hz range. This can cause instability problems if the excitation frequency coincides with the eigenfrequencies of the rotating system. It was not possible to eliminate the excitation force from the turbine, and, due to the bandwidth of the excitation force, it was not likely that enough change in frequency for critical eigenmodes could be achieved by adding mass and stiffness to the rotating and support structures. The measure most likely to have a positive effect on the resonance problem was to add damping to the system. Extended numerical simulations were performed to investigate the effect radial dampers would have on the unit if they were mounted between the upper bracket and the concrete structure. The simulations were performed in software developed for transient simulation of vertical machines with non-linear hydrodynamic guide bearings and non-linear dampers. The radial dampers were found to reduce response in the range of 5.4–7.4 Hz, in accordance with the simulations. Radial dampers were installed in the machine, and it was possible to finalize commissioning with acceptable vibration levels.

1. Introduction
In 2017 a vertical hydropower unit with a Kaplan turbine was refurbished in northern Sweden. The refurbishment included a new runner, which was upgraded from 38 MW to 43 MW, a new stator and refurbished rotor poles. The bearings were renovated without geometry change.

During the commissioning of the unit, resonance problems occurred when the machine reached 90% of its synchronous speed, and the vibration problems increased as rotor speed increased. It was not possible to test the protection equipment for runaway due to high vibration amplitudes in the turbine and lower generator bearings. The runner was identified as the excitation source, and the excitation frequency from the turbine was in the range of $2.8\Omega_0 - 3.9\Omega_0$, where $\Omega_0$ is the nominal rotor speed.

To solve the resonance and vibration problem at start up, shutdown and operation at synchronous speed with no generator load, different combinations of runner angle and guide vane opening were...
tested. As a complement, air injection was also tested. None of these alternatives had a noticeable impact on the vibration levels. The runner angle was by mechanical design limited to a minimum angle of 6 degrees.

It was not possible to eliminate the excitation force from the turbine, and, due to the bandwidth of the excitation force, it was not likely that a large enough change in frequency could be attained for critical eigenmodes by changing mass and stiffness of the rotating system and support structure. The action most likely to have a positive effect on the resonance problem was to add stability to the system. Numerical simulations were performed using software developed for simulation of vertical machines with non-linear hydrodynamic guide bearings and non-linear dampers to evaluate the influence of dampers. In the simulations the dampers were placed between the upper generator bearing bracket and the concrete structure.

Only a few papers have been presented that address rotordynamic modelling and measurements made on hydropower systems. In 2004, Hofstad [1] presented some alternative rotordynamic modelling methods for hydropower units. Hofstad also pointed out the importance of including bearings, seals and brackets in the rotordynamic analysis. The clearance of the bearings affects the bearing’s stiffness and damping. Someya [2] compiled the results from numerous experimental evaluations on journal bearings. Typical bearings for vertical hydropower units were not included in these experimental evaluations. Tilting-pad bearings in Swedish hydropower units consist of 8–24 pads, and in extreme cases up to 48 pads. In 2003, Gustavsson and Aidanpää [3] presented a method for measuring bearing loads, and later Gustavsson et al. [4] presented a method for determining the stiffness and damping from the measured bearing loads. Synnegård et al. [5] presented a method to include visco-elastic dampers in hydrodynamical units.

![Figure 1. Schematic figure showing the rotating system in the hydropower unit.](image-url)
2. Nomenclature

2.1. Designations

Table 1. Nomenclature.

| Symbol | Description |
|--------|-------------|
| $\Omega_0$ | Nominal speed [rpm] |
| $\Omega$ | Rotational speed [rpm] |
| $f(t)$ | Unbalance load vector [N] |
| $f(t)_{\text{exc}}$ | Excitation load vector [N] |
| $f(t)_{\text{D}}$ | Damping load vector [N] |
| $G$ | Gyroscopic matrix [Ns²/m] |
| $C$ | Damping matrix [Ns/m] |
| $K$ | Stiffness matrix [N/m] |
| $K_T$ | Stiffness bearing matrix [N/m] |
| $K_{xx} \ldots K_{yy}$ | Bearing stiffness coefficients [N/m] |
| $C_{xx} \ldots C_{yy}$ | Bearing damping coefficients [Ns/m] |
| $\varepsilon$ | Eccentricity [%] |
| UGB | Upper generator bearing |

2.2. Machine properties

Properties of the hydropower unit are shown in Table 2.

Table 2. General machine data.

| Property | Value |
|----------|-------|
| Turbine type: | Kaplan |
| Nominal output: | 45 MW |
| Nominal speed ($\Omega_0$): | 115.4 rpm |
| Head: | 23 m |
| Mass runner: | 60 metric tons |
| Mass rotor: | 197 metric tons |
| Magnetic pull generator: | 310 MN/m |

3. Measurement setup

A series of on-site measurements were performed during the commissioning of the hydropower unit. The commissioning test was interrupted several times due to high shaft vibration levels or control system failures. The main focus during the commissioning test was on minimizing the vibration levels during start and stop as well as in the event of emergency stop from high loads, rather than a regular commissioning.

Each bearing was equipped with radial displacement sensors between the shaft and bearing housing, in both the x- and y-directions. The shaft displacement was also measured at the intermediate shaft in between the lower generator bearing and the turbine guide bearing as well as between dampers and the upper bracket. The vibrations of the bearing housings were monitored at each bearing position. All data was simultaneously sampled at 640 Hz. Anti-aliasing filters, with a cut-off frequency of 200 Hz, were used in order to avoid aliasing.
4. Numerical model

4.1. Rotor description
As in all finite element calculations, the end result is implicitly dependent on how the various components in the mechanical structure have been represented. Within the Swedish hydropower industry there are a variety of different philosophies regarding how bearings and their surrounding structures should be modelled in rotor-dynamic calculations. The components included in the rotor-dynamic calculation performed in this paper are the rotating structure, bearings, bearing brackets and dampers. The eight dampers are connected between the upper bracket arms and the foundation; see Figure 1.

The stiffness of the bearing brackets was calculated by the generator manufacturer and the turbine manufacturer respectively using finite element analysis. A prescribed load was applied to the housing, first in the x- then in the y-direction. The displacement of the bearing housing was calculated in each direction, and from the applied load and the calculated displacement, the stiffness of the bearing brackets was calculated.

4.2. Bearing model
For a vertical machine with tilting pad bearings, the stiffness and damping vary periodically depending on the number of pads in the bearing [6]. Bearing properties for the two tilting pad bearings and the turbine sleeve bearing were calculated in the “Journal bearing analysis” module in the rotor-dynamic analysis software Rappid [7]. The solver is based on Navier-Stokes. It determines the journal’s operating position from a prescribed load, or solves the reaction load from a prescribed journal position. The bearing properties in this case were performed for different eccentricities of the rotor in the bearing clearance. The calculated oil film stiffness and damping were combined with the pivot pin, i.e. the component connecting the bearing segment to the bearing housing. In the rotor-dynamic calculations, the stiffness and damping terms, i.e. $K_{xx}$, $K_{yy}$, $K_{xy}$, $K_{yx}$, $C_{xx}$, $C_{yy}$, $C_{yx}$ and $C_{xy}$, were included.

![Bearing Stiffness UGB $K_{xx}$ and $K_{yy}$ LOP](image)

**Figure 2.** Bearing stiffness, upper guide bearing $K_{xx}$ and $K_{yy}$ in combination with pivot pin stiffness.
Nässelqvist et al. [8] described a method for calculating the bearing coefficients using polynomial fit to describe the stiffness and damping for load on pad, LOP, and load between pad, LBP, as function of eccentricity $\varepsilon$. Synnegård et al. [9] improved the model by including the cross-coupling stiffness and damping in the equations. The polynomial describing the bearing coefficients for a given load angle and eccentricity is calculated according to equations 1 and 2.

$$
K_{ij} = \frac{K_{ij}^{\text{LOP}} + K_{ij}^{\text{LBP}}}{2} + \frac{K_{ij}^{\text{LOP}} - K_{ij}^{\text{LBP}}}{2} \begin{cases}
\cos(n\alpha) & \text{for } i = j \\
\sin(n\alpha) & \text{for } i \neq j
\end{cases}
$$

(1)

$$
C_{ij} = \frac{C_{ij}^{\text{LOP}} + C_{ij}^{\text{LBP}}}{2} + \frac{C_{ij}^{\text{LOP}} - C_{ij}^{\text{LBP}}}{2} \begin{cases}
\cos(n\alpha) & \text{for } i = j \\
\sin(n\alpha) & \text{for } i \neq j
\end{cases}
$$

(2)

Since the bearing coefficients are dependent on the eccentricity and the direction in the fixed coordinate system $\alpha$, the bearing coefficients must be transformed at each time-step from the local coordinate system to the global (fixed) coordinate system. In the analysis, it is assumed that the system can be treated as a quasi-static load case and the transformation of the bearing components can be treated as a transformation between two static coordinate systems at each time-step. The stiffness and damping matrix can be transformed according to equation (3) and (4)

$$
K_T = TK_B T^T
$$

(3)

$$
C_T = TC_B T^T
$$

(4)

where

$$
T = \begin{bmatrix}
\cos(\alpha) & -\sin(\alpha) \\
\sin(\alpha) & \cos(\alpha)
\end{bmatrix}
$$

(5)

where $K_B$ and $C_B$ are the bearing stiffness and damping matrices in the local coordinate system, while the $K_T$ and $C_T$ are transformed bearing data in the global (fixed) coordinate system.

4.3. Damper model

Tests with eight dampers installed between the upper bracket and the foundation were performed to see if it was possible to reduce the vibration during start and stop. Two different dampers from ITT Enidind Inc. were tested [10]. The two tested dampers were model VESXD30-100 with a damping constant of $C = 19.6 \text{kN(mm/s)}^{0.2}$ and VESXD100 $C=50.0 \text{kN(mm/s)}^{0.2}$. The visco-elastic damper model is introduced as a non-linear force in the analysis, where the total damping constant for the system is $C_{39}=39.2 \text{kN(mm/s)}^{0.2}$ and $C_{100}=100 \text{kN(mm/s)}^{0.2}$ for the two cases. The index of the damping constant indicates the total damping constant

$$
f_{\text{damp}} = C \text{sign}(\dot{x}) |x|^{0.2}
$$

(6)

where the constant $C$ is the damping constant for the two cases with $C=C_{39}$ or $C=C_{100}$. 


4.4. Excitation force
The unbalance force applied on the generator rotor and the turbine is assumed to be \( G_{2.5} \). Besides this, an excitation force is applied to the turbine with variable frequency running from \( 2\Omega_0 \) to \( 4\Omega_0 \), where \( \Omega_0 \) is the nominal rotational speed of the rotor. The magnitude of the excitation force in the analysis is set to 100 kN acting in a specific direction. The equation for the excitation force can be written as

\[
f_{\text{exc}} = 1.67m_{\text{turb}} \cos(\Omega_0 (2 + 0.0015t) t)
\]

where \( m_{\text{turb}} \) is the mass of the turbine.

Calculations of the hydropower unit’s rotor response versus the mass unbalance in the turbine and generator rotor as well as a disturbance force acting on the turbine were performed with an in-house code for simulation of vertical rotors with hydrodynamic guide bearings. The rotor geometry is built up with discrete numbers of beam elements and connected nodes; see Figure 1. The program solves the rotor displacement as a function of time with nonlinear bearings, nonlinear external dampers and a time-dependent excitation force. The equations of motion (1) below describe a method for solving the rotor response problem.

\[
M \ddot{x} + (G\Omega + C + C_T) \dot{x} + (K + K_T) x = f(t) - f_{\text{exc}}(t) - f_D(x)
\]

where \( M, G, C \) and \( K \) are mass, gyroscopic, damping, and stiffness matrices respectively. \( C_T \) and \( K_T \) are the damping and stiffness matrix for the bearing updated for each rotor position. \( f(t) \) is the unbalance force, \( f_{\text{exc}}(t) \) is the excitation force acting on the turbine and \( f_D \) is the damping force from the external dampers.

5. Results from measurements
This section presents measured shaft displacement at emergency shutdown from 26 MW and at synchronous speed without load on the generator. The figures present data measured at the upper generator bearing, lower generator bearing, at the intermediate shaft and at the turbine guide bearing. Upper generator bearing and intermediate shaft sensor positions are marked with a * in Figure 1. Figure 3 presents shaft displacements at synchronous speed without load on the generator. The blue lines represent displacements without dampers, the red lines represent displacements when the dampers have a damping constant of \( C_{39} \) and the green lines represent displacements when the dampers have a damping constant of \( C_{100} \).
Figure 3. Shaft displacement at upper generator guide bearing (a) and at lower generator guide bearing (b). The associated FFT diagrams are shown in (c) and (d).

Figure 4 shows the range of data used in the FFT analyses shown in Figures 5 and 6 for an emergency shutdown. Figure 5 visualizes the dampers’ impact on shaft vibrations in transient operating modes. The figure presents shaft displacements with C100 dampers (red lines) and with dampers C39 dampers (blue lines). Figure 6 presents the frequency characteristics of the shaft displacements during emergency shutdown.

Figure 4. Normalized rotational speed and guide vane opening at emergency shutdown from 26 MW.
Figure 5. Shaft displacement, (a) upper generator bearing and (b) intermediate shaft, at emergency shutdown from 26 MW.

Figure 6. FFT on shaft displacement; (a) upper generator bearing and (b) intermediate shaft. Data from Figure 5, at time 10–17 s.
6. Numerical results

The simulation time to increase the frequency of the excitation force from $2\Omega_0$ to $4\Omega_0$ is 666 seconds. Excitation force acting on the turbine with fixed direction is $1.67 m_{\text{urb}}$, which corresponds to 100 kN. Results from the numerical simulation presented in the figures below have 200 Hz resolution, and the internal solver has a sampling speed of 10 kHz.

In the diagrams, rotor response at four different positions on the shaft is presented. The positions are the exciter, centre of generator rotor, shaft position between lower generator guide bearing and turbine guide bearing and centre of turbine. Blue curves (C0), represent results without dampers, red curves (C39) represent simulation with damper type VESXD 30-100 and green curves (C100) represent results with dampers VESXD100.

![Diagram showing rotor response at different positions](image)

Figure 7. Rotor response at exciter (a), generator rotor (b), intermediate shaft (c) and turbine (d).
7. Conclusion

The aim of the paper was to investigate whether visco-elastic supports can be used to reduce vibration in hydropower rotors. The presented results from the measurements in Figures 3–6 and the simulation results in Figure 7 demonstrate that the installation of visco-elastic dampers significantly reduce the large vibration levels in the rotating system. The dampers were installed behind the upper generator bracket, which was not the ideal location. A better location for the dampers would have been on the intermediate shaft since the shaft deflection is largest there; i.e., the efficiency of the dampers would have been the greatest in that position. An interesting alternative corrective action would have been reconstruction of the bearing to include some sort of squeeze-film dampers in the bearings.

Overall, the installation of dampers appears to be very promising for hydropower applications. Some key benefits include:

- Large restoring forces for low velocities
- Easy to install; no major modifications necessary since the dampers are located between the bearing bracket and the foundation/generator chamber
- The dampers have a low cost
- Low maintenance and long service life

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