Numerical Analysis of Vibration Patterns in Hydropower Units

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Hydropower units are usually equipped with sensors that continuously deliver vibrational data that are suited for diagnostic purposes. The available data from normal stationary plant operation might be used to deduce the machine condition, discover present faults and to predict the future development of the machine’s vibration behavior. This information allows to optimize maintenance schedules and, therefore, minimize plant downtime. The best way to interpret vibration signals is still a topic of research and several methods, partly involving model-based monitoring, have recently been proposed \cite{1,2}. However, those methods are not directly suitable for hydropower units, as some model parameters like exact bearing clearance in the warmed-up machine are not accurately known and bearing non-linearity complicates the process. In this contribution, a suitable rotordynamic model is used to investigate vibrations in vertical hydropower units and their possible use for condition monitoring and predictive maintenance.

Most hydropower units share a common layout. A shaft connects generator and turbine. It is supported by three guide bearings, namely the upper generator guide bearing (UGB), lower generator guide bearing (LGB) and turbine guide bearing (TGB). Usually, they are of tilting-pad type, especially in modern plants. The rotor weight acts axially on a thrust bearing, which is neglected for the calculation of radial vibrations. The long shaft consists of two parts that are joined with a rigid coupling.

The shaft is modelled with Timoshenko-beam-elements, whereas the generator and the turbine are represented by their corresponding point mass and inertia properties. Electro-magnetic forces and hydraulic forces at the turbine are taken into account by using a negative spring element \(k_{\text{mag}}\) and a constant static force \(F_{\text{hydr}}\).

Spring elements with positive stiffness represent the connection of the bearing to the foundation. The actual bearing is considered with its oil film stiffness and damping. For horizontal machines, the gravitational force pushes the shaft into a certain operating position within the bearing shell, where the force-displacement relationship can be linearized. This is not applicable for vertical machines, where the operation position is unknown \cite{3}. It is necessary to calculate the oil film forces for a given operating position. Here, common approaches are based on the Reynolds equation. As a result, all calculations of the rotordynamic model are transient ones, constantly updating the bearing stiffness and damping in every time step according to the shaft’s position. The non-linear equation of motion can be written as \(M\ddot{x} + D(x)\dot{x} + K(x)x = f\).

In this work, the commercial software MADYN \cite{4} is utilized to solve the dynamic equation. The described model considers all important aspects of dynamic modelling of vertical hydropower units as mentioned in other publications \cite{5,6}.

2 Numerical Results

The analysis focuses on two common vibration causes that are difficult to distinguish: generator unbalance and coupling angular misalignment ("coupling bend") as indicated in Fig. 1. Whenever one of those imperfections is present and a rotor spin speed \(n_r\) is applied, the shaftline will show speed synchronous vibrations. Widely available sensors in hydropower plants measure shaft displacement relative to the bearing and bearing velocity. Those sensors will observe circular motions of the shaft within the bearing (the so-called orbit) for both types of imperfections. Evaluations can be done in terms of orbit radius

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(vibration amplitude) and angular position on the orbit at a given point in time, referred to as phase angle. Due to the bearing properties, shape deviations of the circular orbit that are resulting in higher harmonics and other effects might arise.

Several numerical calculations with varying model parameters have been conducted. Namely, the extent of imperfections, the bearing clearances and the hydraulic static load were modified within reasonable ranges. Following features could be observed, irrespective of the varying model parameters:

**Speed behavior.** Unbalance will cause nearly no vibration at slow rotor speeds. With increasing speed, the relative shaft vibration follows a resonance curve, see Fig. 2. This behavior is well-known from simple Laval rotor models [7]. A coupling bend will result in relative shaft movement even at slow speeds. With increasing speed, the oil flow within the bearing gap enhances, higher oil film forces arise and shaft vibration amplitude declines, see Fig. 3. This difference in speed behavior can be utilized to diagnose the vibration cause, but requires start-up data or data from very slow speed. For diagnosis based on stationary data recorded at normal operation this approach is not able to separate both effects.

**Phase angles.** A generator unbalance will result in a dynamic force on the generator in a specific direction in a given point in time. Thus, the generator bearing signals show the same phase angle. For coupling bend, forcing the non-straight shaft into three aligned bearings will result in a 180° phase difference between two neighboring bearings. This was observed to be true for the analyzed models with an accuracy of ±7°.

**Ratio of vibration amplitudes.** Despite of non-linear bearing behavior and varying model parameters, the ratios of absolute bearing vibration amplitudes appear to be nearly constant for one vibration cause, see Fig. 4. So even if bearing clearance and hydraulic static load are not exactly known, it is possible to determine those ratios for a given machine. The vibration amplitudes resulting from one imperfection can then be described as scaling-and-direction value $s$ times ratio vector $a = (a_1, a_2, a_3)^T$, $a_i$ being the vibration amplitude and phase of bearing number $i$. Furthermore, superposition appears to be permissible for the analyzed model, i.e. the absolute vibration amplitudes $c$ of a shaft with both types of imperfection can be adequately described by a sum of the two single-imperfection cases: $s_u \cdot a_u + s_b \cdot a_b \approx c$. The observed error was smaller than three percent. Note that all elements of this equation are complex numbers to account for the respective amplitudes and phase angles as explained in rotordynamic literature, e.g. [7]. Using this equation, the two scaling factors (corresponding to the extent of imperfection) can be calculated from the measured vibrations $c$ and the ratio vectors $a_u, a_b$.

### 3 Conclusion

With the presented features, it is possible to diagnose the two vibration causes generator unbalance and coupling bend from vibration signals for a given model, even when exact values of parameters like bearing clearance are not known. One has to keep in mind that the vibration signal at a real hydropower unit will be superimposed with considerable hydraulic flow induced vibrations, especially at the turbine guide bearing. Therefore, further research for distinguishing features that are robust against superimposed perturbation will be conducted and verification with field data will be done.

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