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Diagnostics of a hydraulic turbine failure

Monica Egusquiza[1]; Alex Presas[1]; David Valentin[1]; Carme Valero[1]; Beibei Xu[2]; Diyi Chen[2]; Eduard Egusquiza[1]

[1] Center for Industrial Diagnostics, Polytechnic University of Catalonia (UPC), Av. Diagonal, 647 – 08028 Barcelona (Spain)

[2] Institute of Water Resources and Hydropower Research, NorthWest A&F University, Saanxi Langyin, (PR China)

Abstract **– This paper presents the failure analysis and diagnostic of a hydraulic turbine. Shortly after a maintenance revision the thrust bearing of the turbine was destroyed when the turbine was put into operation. The damaged bearing was examined and the possible causes discussed. To identify the problem that led to the failure of the thrust bearing, a comprehensive on-site measurement campaign was done. Vibrations, pressures, temperatures and operating parameters were acquired at different operating conditions of the turbine. The analysis of the data allowed to determine the source of the problem and to implement a solution.**

Keywords – Hydropower, Failure analysis, Diagnostics, thrust bearing damage.

I. INTRODUCTION

One of the main benefits of hydropower comes from its ability to provide flexible power to the grid. Because of that, hydropower plants are used to compensate the random generation of wind and solar power plants, thus ensuring that power generation always matches demand. Due to the growing capacity of new renewable energies, hydropower units are working under off-design conditions and undergoing transients more frequently, what increases the risk of damage. In order to avoid large maintenance costs and, especially, the lack of availability of hydro turbines the detection of damage in incipient stages is of utmost importance. An analysis of history cases and damage symptoms is an essential step to ensure a sound condition monitoring.

A large percentage of hydropower plants are operated by Francis turbines. These machines are vertical shaft units with an electrical generator at the top and a hydraulic turbine runner at the bottom. The water flows into the runner through a spiral casing with movable guide-vanes that are used to regulate the discharge and the power delivered. Generally, these machines have three radial bearings and one axial bearing. The so-called upper and lower generator bearings support the radial forces produced by the electrical generator, while the turbine bearing supports the radial forces coming from the runner. The so-called axial or thrust bearing sustains the weight of the whole rotating structure and hydraulic axial thrust.

Hydraulic turbines have fluid film bearings. In a thrust bearing, a thin layer of oil separates the rotating sliding disk from a series of stationary thrust pads. Before the unit is put into operation, a high-pressure pump injects oil to the bearing through a hole located in the middle of each pad (i.e. hydrostatic lubrication). This produces an axial lift that separates the stationary and the rotating part and prevents them from mutual friction. Next, water flows into the runner and the unit starts to rotate. At nominal speed the bearing oil pump no longer operates, since the lift from the hydrodynamic lubrication between the tilting pads and the sliding disk is large enough to sustain the machine.

This paper describes and analyses the failure occurred in a Francis turbine. The unit studied is a small vertical shaft Francis turbine with three radial bearings and one thrust bearing, which rotates at 500 rpm and delivers a maximum power of 9 MW (Figure 1).

Fig. 1. View of the hydropower plant

II. FAILURE DESCRIPTION

After a trip motivated by an electrical fault (stator ground trip), the generator of the hydropower unit was inspected. During the inspection of the stator (the rotor was not disassembled) a malfunction was detected in a couple of stator bars. The fault found in the generator stator was considered of minor importance. Because of the large water discharge available from the river at that moment, the electrical company decided to put the machine back into operation without repairing the generator since that would have required an important amount of time and money.

After starting the unit and before reaching full power, a temperature alarm was triggered. The metal temperature of the thrust bearing overpassed the maximum value and the machine tripped. The thrust bearing was dismounted and upon inspection a huge damage was revealed; the Babbitt layer of the pads (soft material which is generally an alloy of tin and lead) was completely erased and the rotating disk had several scratches. Figure 2 shows a drawing of the thrust bearing and Figure 3 a picture of the dismounted thrust bearing pads with the remaining material scratched.

Critical conditions for thrust bearings take place during transients because the rotating speed is not high enough to create a large hydrodynamic force. This is why hydrostatic lubrication is provided in the beginning of the start-up. In this case, though, the damage occurred when the machine was rotating at nominal speed. In addition, this was caused by rubbing, which indicates a lubrication break-down. This type of damage produced during steady conditions is indicative of heavily loaded bearings. Other possible causes can be an inadequate supply of lubricant or misalignment, which leads to edge loading.

An excessive loading can be produced by a poorly aligned generator, but especially by the hydraulic axial thrust of the turbine. Hydraulic axial thrust is always present due to the difference in pressure between the runner crown and the band, and also because of the change in momentum of the flow. Axial thrust increases with the discharge (i.e. power). Pulsating axial thrust can also occur due to vortex rope instabilities.

After repairing the bearing, the turbine was put back into operation. This time, a vibration measurement was carried out to monitor the turbine behavior.

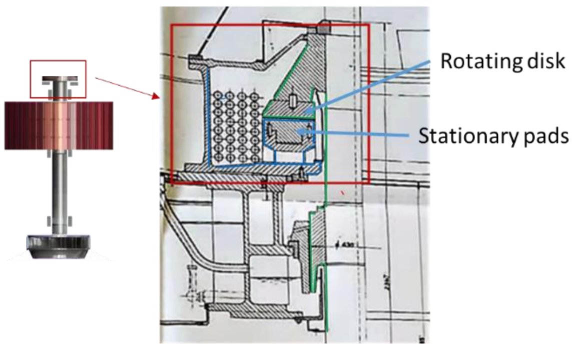

Fig. 2. Drawing of the upper generator bearings

Fig. 3. Bearing pads with damage

III. ON-SITE TESTS

In Figure 4, the measuring locations in the Francis turbine are shown. For the measurement, a multichannel acquisition system B&K with accelerometers, proximity probes and pressure sensors was used. In addition, the operating parameters (power, rotating speed, etc.) and temperatures were recorded from the Scada software of the power plant, as well as the axial displacement of the shaft related to the thrust bearing pedestal.

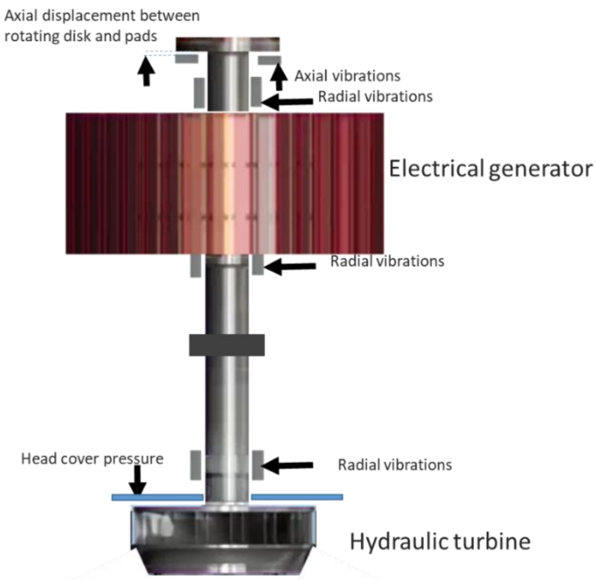

Fig. 4. Measuring positions during the tests

For the investigation, all the parameters were recorded during different operating conditions of the turbine. The measurement began with the start-up transient, during which the turbine increased steadily its rotating speed and the generator remained disconnected. Upon reaching the nominal rotating speed, the generator was electrically excited and synchronized with the frequency of the grid. After the synchronization, the power delivered by the turbine was gradually stepped up, starting from the minimum power.

At one point during the gradual increase of the turbine power, the machine presented the same behaviour observed during the thrust bearing incident. At a specific power, the thrust bearing temperature increased suddenly. In order to avoid the damage of the bearing components, the machine was stopped at due time and the hydrostatic lubrication from the oil pump was switched on.

The subsequent inspection of the bearing showed dry friction as in the previous event.

A. Analysis of SCADA data

In Figure 5, the evolution of the thrust bearing temperatures has been represented. When increasing power, the oil and metal temperatures increase slowly at first but when power reaches 80% the metal temperature jumps in a few seconds from 44,4 °C to almost 90 °C. Fortunately, hydrostatic lubrication was provided at the right time and the machine was stopped.

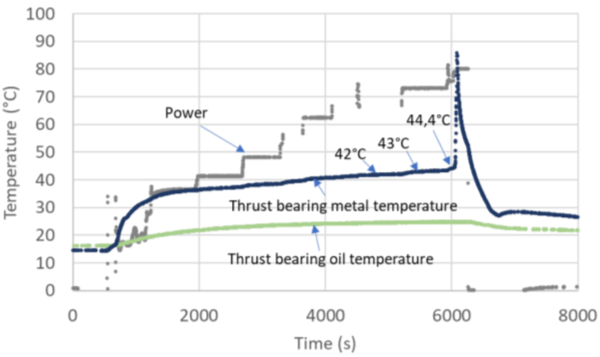

Fig. 5. Temperature evolution with power

B. Analysis of vibration data

For the diagnostic, all sensors were analysed using advanced techniques [1][2][3][4][5][6]. Francis turbines have a complex vibratory behavior due to complex hydraulic phenomena that depend on the operating conditions, such as vortex rope, hydraulic resonances, instabilities, cavitation and mechanical resonances of the runner and rotating train. Different detection techniques have been used to identify all of these phenomena. Improved condition indicators and sensor position has been optimized from the analysis of the dynamic behavior of the turbine. Other techniques like Neural Networks and Factor Analysis were used to optimize the monitoring and detection capabilities. Every phenomenon can be detected with different techniques and sensors. Further discussion is beyond the scope of this abstract.

In Figure 6 and 7, the spectral cascades during the start-up process have been represented. In the first part of the transient the machine is not excited electrically so the electromagnetic forces do not appear. The only excitation forces affecting the machine are of mechanical and hydraulic origin. When the generator is excited, an important increase in amplitude at the rotating frequency (8,4 Hz) and at twice the line frequency (100 Hz) occurs. This means that the generator is not working properly, and it produces an electromagnetic unbalance. The unbalance also enhances the axial vibration at the misalignment frequency (2x rotating speed). The predominant vibration symptoms in the selected vibration locations were related to damage produced by the generator.

The turbine behavior could be considered as normal. No abnormal vibrations were detected when the failure occurred.

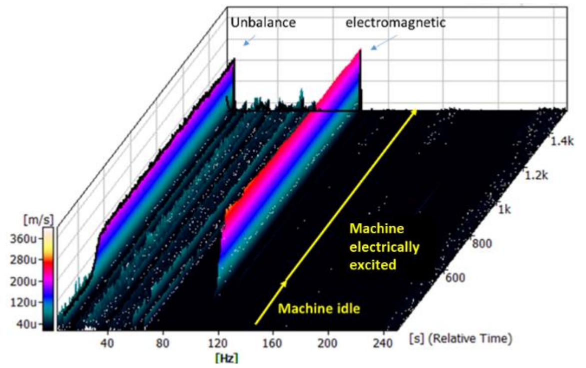

Fig. 6. Generator vibration during start-up and grid connection

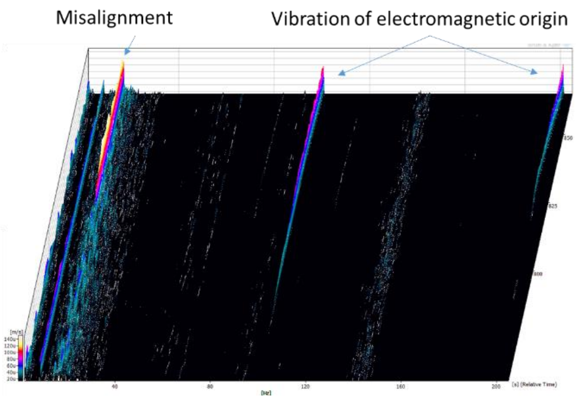

Fig. 7. Axial vibration in thrust bearing during start-up and grid connection

C. Analysis of axial thrust

To check the axial thrust, the pressure in the head cover was compared with the axial displacement of the shaft. In Figure 8, the variation of axial displacement and head cover pressure with load can be observed. It is seen that every time the power increases the pressure in the head cover also goes up. This leads to the runner having a larger pull-down force, which results in the clearance between the rotating and the stationary part of the thrust bearing being reduced. In the turbine studied, every increase in pressure produced an increase in the axial thrust that was leading to an excessive reduction of the bearing clearance.

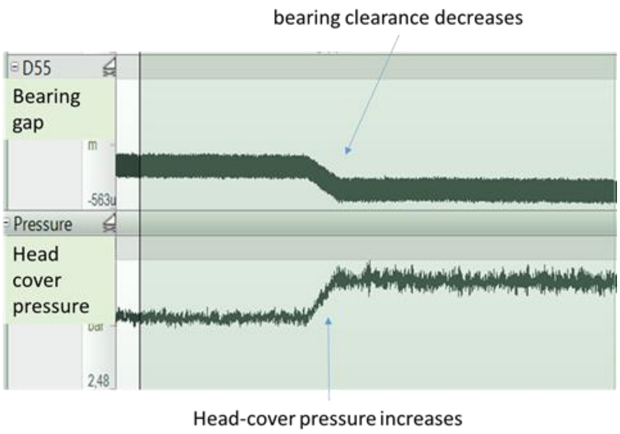

Fig. 8. Change in bearing clearance with water pressure when power changes from 1 MW to 2 MW

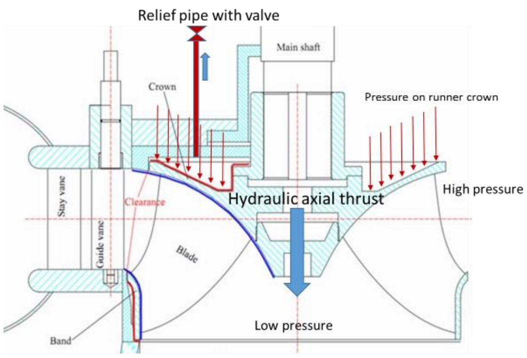

Fig. 9. Hydraulic axial thrust

In Figure 9, there is a schematic explanation of the axial thrust. The pressure acting on the runner crown is transmitted from the runner entrance (high pressure). On the opposite side of the runner the pressure is lower. This difference in pressure acts on the runner and produces a down-pull.

IV. DISCUSSION AND CONCLUSIONS

The vibration analysis indicated a faulty generator that produced radial and axial excitations. Although there is a considerable increase in axial vibration, the vibration levels seem not high enough to produce the damage detected in the thrust bearing.

What was considered more important was the increase in the hydraulic axial thrust of the turbine with power. This axial thrust produces a down-pull that has to be supported by the thrust bearing. A significant increase in axial thrust was detected when increasing power, which can be due to worn runner seals not able to sufficiently reduce the pressure on the head cover. At a certain power the axial thrust was excessive, which resulted in a large reduction of the bearing clearance. The combined effect coming from an excessive axial thrust and a large axial and radial vibration produced by the faulty generator caused the oil film rupture.

To allow the electric company to operate the turbine until an overhaul of the whole unit was possible, a mitigation method was provided. To reduce the axial thrust, a relief pipe with a valve was installed in the head cover (Fig. 9). By opening the valve, the head cover pressure (and thrust) could be controlled. The optimum pressure was obtained with the tests and the further cases of damage were prevented.

V. ACKNOWLEDGMENTS

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