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DIAGNOSTIC OF CAVITATION IN HYDRAULIC MACHINERY

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An experimental investigation has been carried out to diagnose cavitation in actual hydraulic turbines from the analysis of structural vibrations and hydrodynamic pressures. The frequency content and the amplitude modulation of the signals have been studied in real Kaplan, Francis and Pump-turbine prototypes for detecting some types of cavitation like the leading edge cavitation and the draft tube swirl. The first type is erosive and the second one limits the stability of operation. The results found in the selected machines are presented. General conclusions have been obtained about the most adequate sensor, measuring location, signal processing and analysis for each type of cavitation.

Keywords: cavitation, hydraulic turbines, monitoring.

1. INTRODUCTION

The occurrence of cavitation in hydraulic turbines can in some cases lead to damaging operating conditions. Typical problems are flow instabilities, excessive noise and vibrations, reduction of machine efficiency and erosion of solid surfaces. The long term operation of the machine under one of these negative effects obviously increases the maintenance costs and the losses of revenue during plant overhaul. Furthermore, in extreme cases even a catastrophic failure of the machine can take place unexpectedly.

Nowadays, the trends towards power concentration for new designed machinery are favouring the negative consequences of cavitation. Firstly because the propensity for cavitation and its intensity are being enhanced by increasing the rotating speeds and secondly because the mechanical structures are much weaker. Another point is the fact that the power market favours the operation of the hydraulic machines in conditions far from their best efficiency point where there is more risk of cavitation occurrence.

Therefore, it is expected that more cavitation problems will appear from now on. Since remedies are very difficult to apply in existing units, especially if they have to be low cost and effective, the best alternative consists in developing diagnostic techniques for detecting and evaluating this kind of situations. The most suitable methods would be based on external measurements that do not interfere with the machine operation, say vibrations, acoustic emissions and pressures. If diagnostic methods are developed then with this information it will be possible to apply maintenance policies based on prediction and to avoid previously unknown harmful situations.

2. METHODOLOGY

2.1. Test cases

The current experimental investigation has been carried out with three different hydro turbines corresponding to one Kaplan, one Francis and one Pump-Turbine. The three of them are vertical shaft machines.

The Kaplan turbine has a nominal flow rate of 225 m^3/s and a net head of 34 m. The maximum output power is about 73 MW and the rotating speed N is 125 rpm, so its fundamental frequency f_f is 2,08 Hz. The number of runner blades, Z_b, is 6 and the number of guide vanes, Z_v, is 24. It has a turbine bearing and, just above it, a thrust bearing. Both are sleeve bearings. This unit suffers cavitation erosion on the blade inlet suction sides as detected during plant overhaul.

The Francis turbine has a nominal flow rate of 28 m^3/s and a net head of 51 m. The maximum output power is about 11 MW, N is 250 rpm (f_f = 4,16 Hz), Z_b is 15 and Z_v is 24. The turbine rotor is assembled to the foundation through sleeve bearings. There are 3 guide bearings, one in the turbine shaft and two in the generator shaft, above and below it. It has a turbine bearing and, just above it, a thrust bearing. It must be noted that in this case the turbine bearing is made of rubber and works without oil film. This unit suffers from severe cavitation erosion at the middle of the blade next to the band in the extrados that is obviously due to inlet cavitation.

The Pump-Turbine is a high head single stage reversible machine. It can reach a maximum output load of about 112 MW. The total head is 400 m and the flow rate is 24 m^3/s in pumping mode and 31,5 m^3/s in turbine mode. The N is 600 rpm (f_f = 10 Hz), Z_b is 7 and Z_v is 16.

2.2. Instrumentation

A complete measurement and recording set-up has been used for these field tests. Vibrations have been monitored with general purpose piezoelectric accelerometers and the dynamic pressures with piezoelectric pressure sensors. A photoelectric tachometer probe has been used for contact-free detection of the shaft rotation. The output signals of the transducers have been conditioned prior to their recoding with a RACAL V-Store tape recorder.

2.3. Techniques

The proposed methodology is based on the monitoring and analysis of cavitation induced vibrations and pressures [1]. The vibrations are recorded on external parts of the machine such as the guide bearing pedestals, the guide vane central arms and the draft tube man door The pressures are measured from flush-mounted transducers in the draft tube wall.

For the diagnostic of cavitation occurrence the selection of the most adequate measuring positions on the machine is of relevant importance. Moreover, measurements have to be carried out at different operating conditions since the flow hydrodynamics can change significantly depending on them. Therefore, it has an influence on the type and characteristics of the cavitation that takes place [2].

In the frequency domain, the study of the high frequency content of the vibrations is a well-known technique to detect erosive cavitation activity [3,4]. Then, to identify the type of cavitation provoking the damage it is necessary to use specific signal processing techniques like the demodulation of band pass filtered signals [5,6]. The results from this technique can be improved with the use of synchronous time averaged signals [7]. Finally, joint time-frequency analysis is also a promising tool that permits to detect the impulsive forces produced by cavity collapses. Detection of other forms of cavitation that generate strong pressure fluctuations is easier since low frequency results are a suitable tool for that.

Once cavitation has been detected and identified, the next step consists in the diagnosis of the possible risk of erosion or damage. For cavitation erosion prediction there is a method based on the dynamic calibration of the transmission path (transfer function) between the eroded regions and the most sensitive measuring position [8]. The knowledge of this function should permit to infer the amplitude of the forces acting on the material due to the collapse of cavities from the measured vibrations [9].

3. RESULTS

3.1. Diagnostic of leading edge cavitation in Kaplan

The frequency content of the vibrations in the turbine and thrust bearings has been analyzed in the whole operating range of the machine from 10 to 56 MW. The averaged power spectral densities (PSD) of the acceleration signals measured in radial direction on turbine bearing are shown at the top of Fig. 1. It is observed that broadband vibration is present above 4 kHz and that the maximum levels are around 8 kHz for all the output power conditions. It must be noted that the amplitude levels in the turbine guide bearing are considerably higher than in the thrust bearing that is located farther from the runner.

The spectra are roughly parallel for all the output loads but the strongest increase in the vibration amplitude is detected when reaching maximum load. The radial vibrations measured in the same bearing at 90º give similar results than the ones shown in Fig. 1. The axial direction on the thrust bearing is the worst one with the lowest amplitudes and no significant information.

The signals have been band pass filtered between 5 and 10 kHz using a second order Butterworth digital filter. The envelope of the filtered signal has been calculated using the Hilbert transform. Then, the averaged PSD of the analytic signals has been calculated and the results for the turbine guide bearing are plotted at the bottom of Fig. 1.

Since leading edge cavitation development is affected by the rotor-stator interaction between blades and guide vanes [2], the cavitation induced high frequency noise should be modulated in amplitude by the synchronous frequencies related to this interaction. Particularly, blade passing frequency ($f_b = f_f{\cdot}Z_b$ = 12,5 Hz) and guide vane passing frequency ($f_v = f_f{\cdot}Z_v$ = 50 Hz) should govern the cavity dynamics. Actually, a significant frequency peak at f_v appears at maximum load. For lower operating conditions such modulation does not exists. This indicates that inlet cavitation occurs mainly when the machine operates at maximum guide vane opening and flow rate.

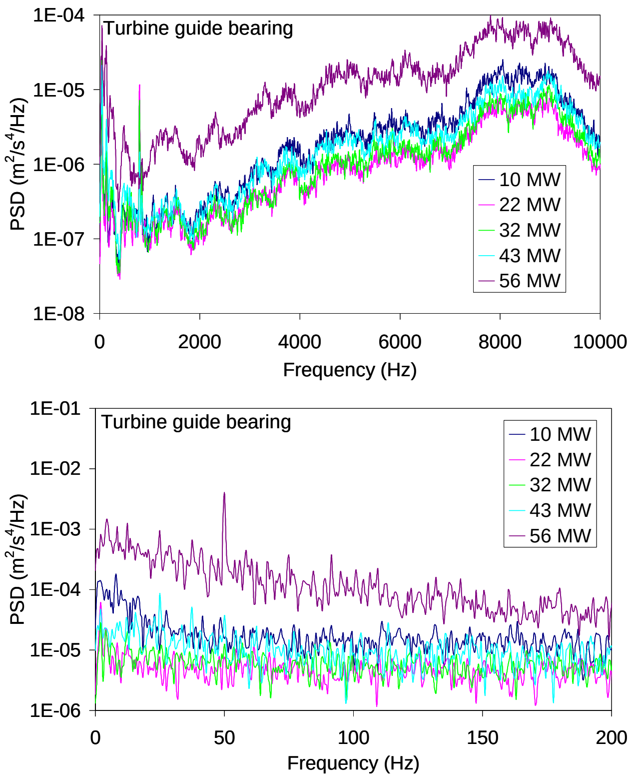

Fig. 1. Power spectral density of raw accelerations (top) and of modulations in the band from 5 to 10 kHz (bottom) measured in radial direction in the turbine guide bearing as a function of output power.

The induced vibrations measured on the turbine guide bearing give the best results from the point of view of detection and confirm the suitability of this measuring position. A detailed observation of the modulation spectrum at 56 MW indicates the presence of other peaks at f_b and its harmonics, $2f_b$ and $3f_b$, apart from the already commented f_v that predominates in the whole frequency band. So leading edge cavitation on the blades of the Kaplan runner is well identified. Therefore, it can be concluded that the nearest bearing to the runner is the most adequate for detecting cavitation. The farther the measuring location, the more difficult the diagnostic.

3.2. Diagnostic of leading edge cavitation in Francis

High frequency vibrations have been measured in four positions around the Francis runner. Two accelerometers have been mounted in radial directions separated by 90º on the turbine guide bearing pedestal (B13 and B12), another one has been mounted in axial direction on the central arm of a guide vane (GV) and the last one has been mounted on the external side of the draft tube wall next to the man door (DT). The Francis turbine has been operated from its minimum, 4 MW, to its maximum, 9 MW, output load.

At the top of Fig. 2 the auto power spectra of the accelerations measured simultaneously in all the positions when the machine was operating at 9 MW are plotted. As expected, the two sensors on the bearing give similar results. They show the largest amplitudes at the highest frequencies. Meanwhile, the sensors in GV and DT show a completely different pattern with the largest levels at lower frequencies.

At the bottom of the same Fig. 2 the auto power spectra of the envelopes of the filtered acceleration signals between 15 and 20 kHz are shown. The main modulation frequencies are f_v and f_b. It is important to notice that again the results from the bearing positions are very similar independently of the angular location of the sensor. Larger differences are found when they are compared with the rest.

The evolution with turbine output load of the envelopes of the filtered accelerations between 15 and 20 kHz detected in position B13 is presented at the top of Fig. 3. From those spectra it is clear that the modulation at certain hydrodynamic frequencies starts at around 8 MW with the appearance of f_v. Then, at 9 MW its amplitude increases significantly and other peaks such as f_b and its third harmonic $3f_b$ also appear.

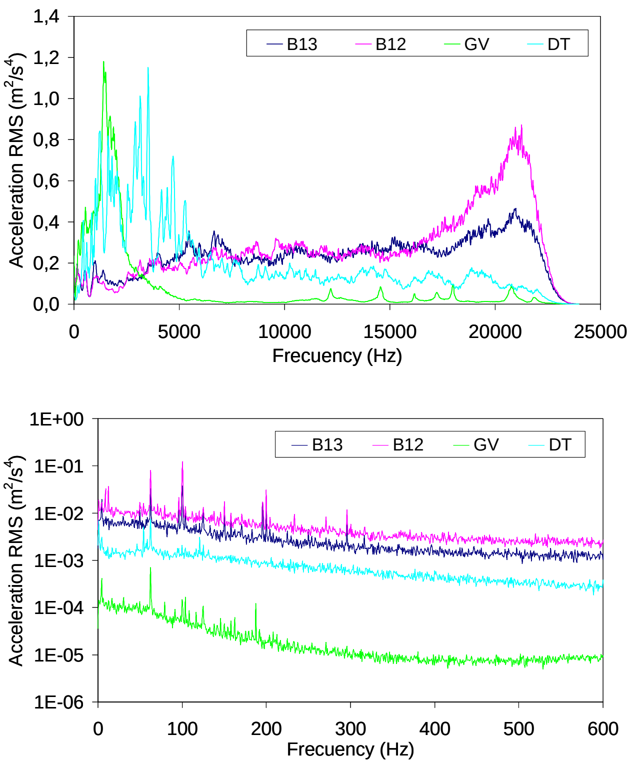

Fig. 2. Auto power spectra of raw accelerations (top) and of modulations in the band from 15 to 20 kHz (bottom) in bearing B13 and B12, guide vane GV and draft tube DT at 9 MW.

Finally, at the bottom of the same Fig. 3 the evolution of the amplitude of the modulation at the guide vane passing frequency f_v has been plotted against output load for all the measuring positions. It is clear that the best location for detecting this modulation is the turbine guide bearing. On the contrary, locations such as the guide vane and draft tube seem to be less adequate.

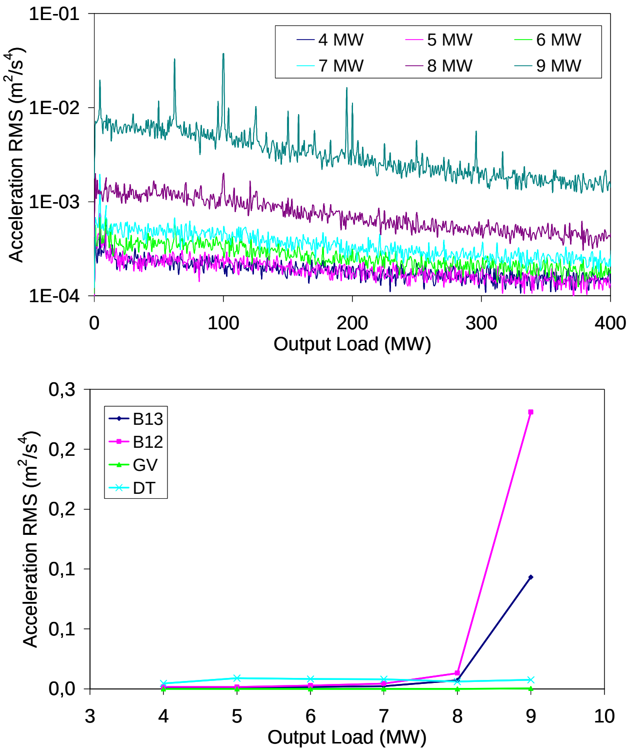

Fig. 3. Auto power spectra of modulations in the band from 15 to 20 kHz in bearing B13 for all the output loads (top) and amplitude of the modulating frequency f_v in bearing B13-12, guide vane and draft tube DT also for all the output loads.

From the diagnostic point of view, it is of our opinion from the current results that the sensors located on the guide bearing pedestal are the most sensitive to erosive cavitation acting on the blades. Firstly because they detect high frequency excitation that is a characteristic feature of cavitation collapse activity. Secondly because these signals are modulated at f_v which does not happens in the GV and DT signals. These latter ones appear to be mainly modulated at f_b. In this case, the conclusion would be that this Francis turbine is suffering erosive leading edge cavitation mainly at maximum output load as indicated by the sharp increase of the amplitude modulation of the vibrations measured on the turbine guide bearing.

3.3. Diagnostic of draft tube swirl in Pump-Turbine

Three dynamic pressure transducers have been flush-mounted in the draft tube wall of the pump-turbine. Two of them, P7 and P8, have been located at the same height with P7 separated +90º from P8 in the sense of shaft rotation and the third one has been positioned in a lower plane and separated -45º from P8. The test have been conducted during turbine mode operation at 50%, 75% and 100% of the maximum output load. At 50 %, partial load condition, a pressure pulsation at 3,1 Hz dominates the signal that corresponds to 0,31 times the fundamental frequency (f_f = 10 Hz). This strong fluctuation disappears at higher flow rates as it is seen at the top of Figure 4 where the auto power spectra for P6 are plotted at 50% and 100%. A strong coherence (around 1) is found between all the transducers at this particular frequency peak at partial load. The phase shift between pressure pulsations at 3,1 Hz are calculated between the two sensors located at 90º, P7 and P8, and a value around 175º is found. Therefore, these results confirm the detection of a cavitating vortex core with an helical shape in the draft tube at 50% load that rotates in the same direction as the runner at 0,31 times the runner rotating speed.

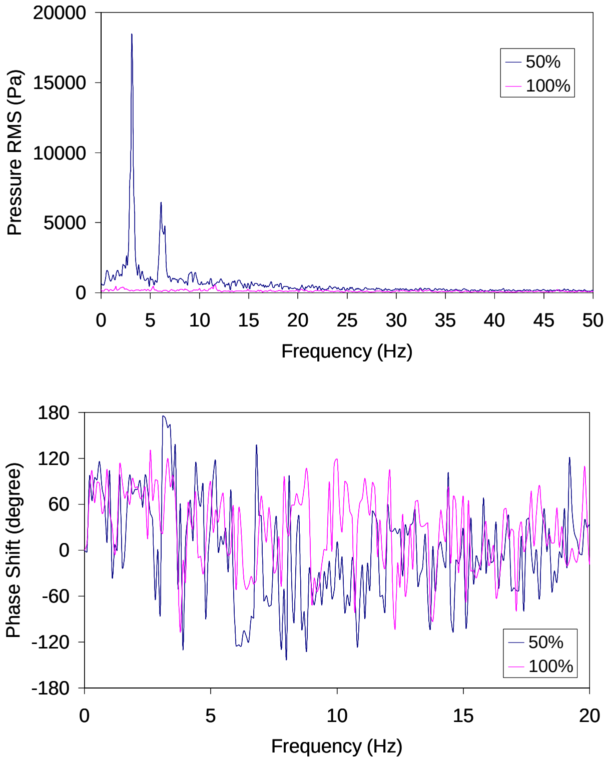

Fig. 4. Top) Auto power spectra of draft tube pressure P6 at 50% and 100% output load. Bottom) Phase shift between pressures P7 and P8 at 50% and 100% output load.

3. CONCLUSIONS

The diagnostic of cavitation problems in large hydraulic turbines has been addressed. In particular, erosive leading edge cavitation and unstable draft tube swirl have been investigated from vibration and pressure measurements respectively.

A Kaplan and a Francis turbine, both suffering erosion on the blades, and a Pump-Turbine with a strong draft tube swirl have been selected for the tests.

Cavitation erosion on the blades increases the level of vibrations from several kilo Hertz to the tenths of kilo Hertz. The best measuring location is the closest bearing to the runner. No privileged angular positions around the shaft are identified.

The comparison of the high frequency content of the vibrations at various output loads serves to identify the operating conditions with highest risk of cavitation erosion. However, to confirm the diagnostic it is necessary to determine the main modulation frequencies at these conditions. In the case that the guide vane passing frequency or the blade passing frequency appears, and in absence of other peaks not related to hydrodynamic phenomena, it can be assumed that erosive cavitation is taking place.

Unstable behaviour due to strong draft tube swirl can be easily identified from pressure measurements on the draft tube. In this case the symptom is the occurrence of a pressure fluctuation at around 30% of the rotating speed.

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