

Condition monitoring of a prototype turbine. Description of the system and main results.

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Abstract. The fast change in new renewable energy is affecting directly the required operating range of hydropower plants. According to the present demand of electricity, it is necessary to generate different levels of power. Because of its ease to regulate and its huge storage capacity of energy, hydropower is the unique energy source that can adapt to the demand.

Today, the required operating range of turbine units is expected to extend from part load to overload. These extreme operations points can cause several pressure pulsations, cavitation and vibrations in different parts of the machine.

To determine the effects on the machine, vibration measurements are necessary in actual machines. Vibrations can be used for machinery protection and to identify problems in the machine (diagnosis). In this paper, some results obtained in a hydropower plant are presented. The variation of global levels and vibratory signatures has been analysed as function as gross head, transducer location and operating points.

1. Introduction

Hydropower is the most flexible source of power generation available and it is capable of responding to demand fluctuations in minutes. Hydroelectric generating units are able to start up quickly and operate efficiently almost instantly, even when used only for one or two hours [1]. As a result of this flexibility, hydropower is an ideal complement to new renewables.

The required operating range of turbine units today has increased dramatically. In this new scenario, more cases of damage have been reported and a more advanced and effective vibration monitoring is necessary [2]

Vibration monitoring is the process of determining the condition of machinery while in operation. The state of a machine is determined by the acquisition and processing of information and data acquired. Depending on the criticality of the machine, condition monitoring can be ongoing or intermittent. The former reacts more quickly to a sudden change and, therefore, better protects machines from sudden faults that cannot be predicted; the latter is cheaper and avoids the permanent installation of sensors, often a difficult task if not considered in the design stage.

In this paper the vibration monitoring in a hydraulic turbine is analysed. The study has been done from data obtained by on-line monitoring system working from January 2016.

2. Monitoring system

The turbine unit studied belongs to hydropower Mica located in British Columbia, Canada. It is a Francis turbine with 500MW of maximum power, rotating at 125rpm. It is a vertical machine with two radial bearings and one axial bearing. The runner has 16 blades and the distributor 20 vanes.



Vibration sensors, pressure transducers and an OneProd MVX monitoring system were installed in the unit.

For vibration monitoring two accelerometers located at 90° were installed in each bearing. Pressure transducers were situated in draft tube and spiral casing. Signals from proximity probes from Bently-Nevada system were also acquired, see Figure 1.

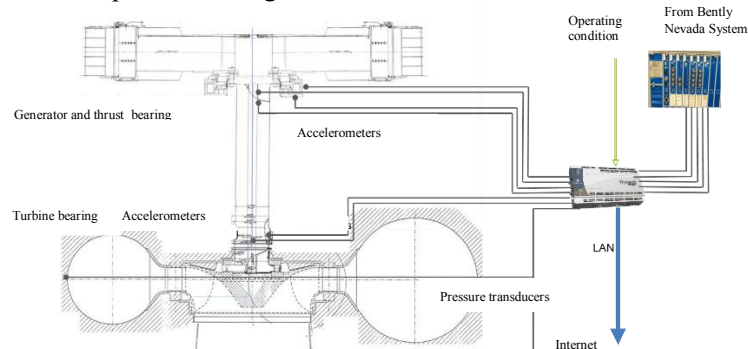


Figure 1. On-line measurements sketch

The system enables databases to store the vibrations from sensors connected to the MVX. Each database values can be stored in the time period desired (every hour, every two hours, once a day...). The databases of the same group can be classified according to the machine operation conditions. The acquisition system will be located inside a cabinet near the turbine floor. From the cabinet an Ethernet cable will connect the acquisition system with a computer.

3. Analysis of turbine operation

Gross head (difference of levels between higher and lower reservoir) changes significantly with season. There is a minimum level in spring (month of May) and a maximum level in autumn (month of October). The turbine studied has limited their operation between 300MW and 430MW see Figure 2.

Several problems appear if the machine works outside of these limits

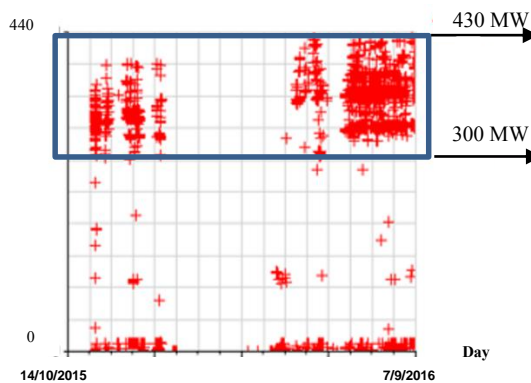


Figure 2. Power of turbine from January 2016

Anyway, vibration levels change with operating conditions in normal operation (between 300MW and 400MW). The variation of global levels and vibratory signatures has been analysed as function of the gross head. The study has been done from data obtained by on-line monitoring system working from January 2016 to September 2016.

In Figure 3 the change in the overall vibration levels (RMS values from 2 to 500Hz) with load and with head in normal operation (300MW to 400MW) has been represented.

It can be seen that vibrations are more important in the turbine bearing and in the draft tube and reaches maximum values at 300 MW. In addition, overall levels increase when head increases for all accelerometers.

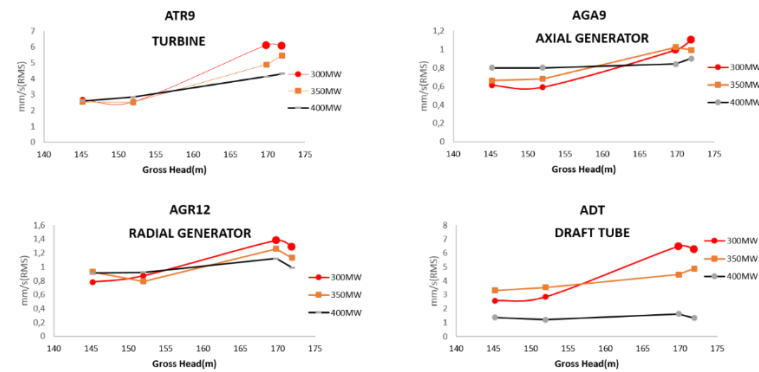


Figure 3. Overall vibration levels (RMS value of mm/s) of function as function of the head: A-TR9 Accelerometer Turbine bearing radial, A-GA9 Accelerometer Generator bearing Axial, A-GR12 Accelerometer Generator bearing Radial, A-DT Accelerometer Draft Tube.

To know the origin for this increase, the vibration signatures (from 0 to 500Hz) has been analyzed for different gross heads and loads for turbine and generator bearing see Figure 4.

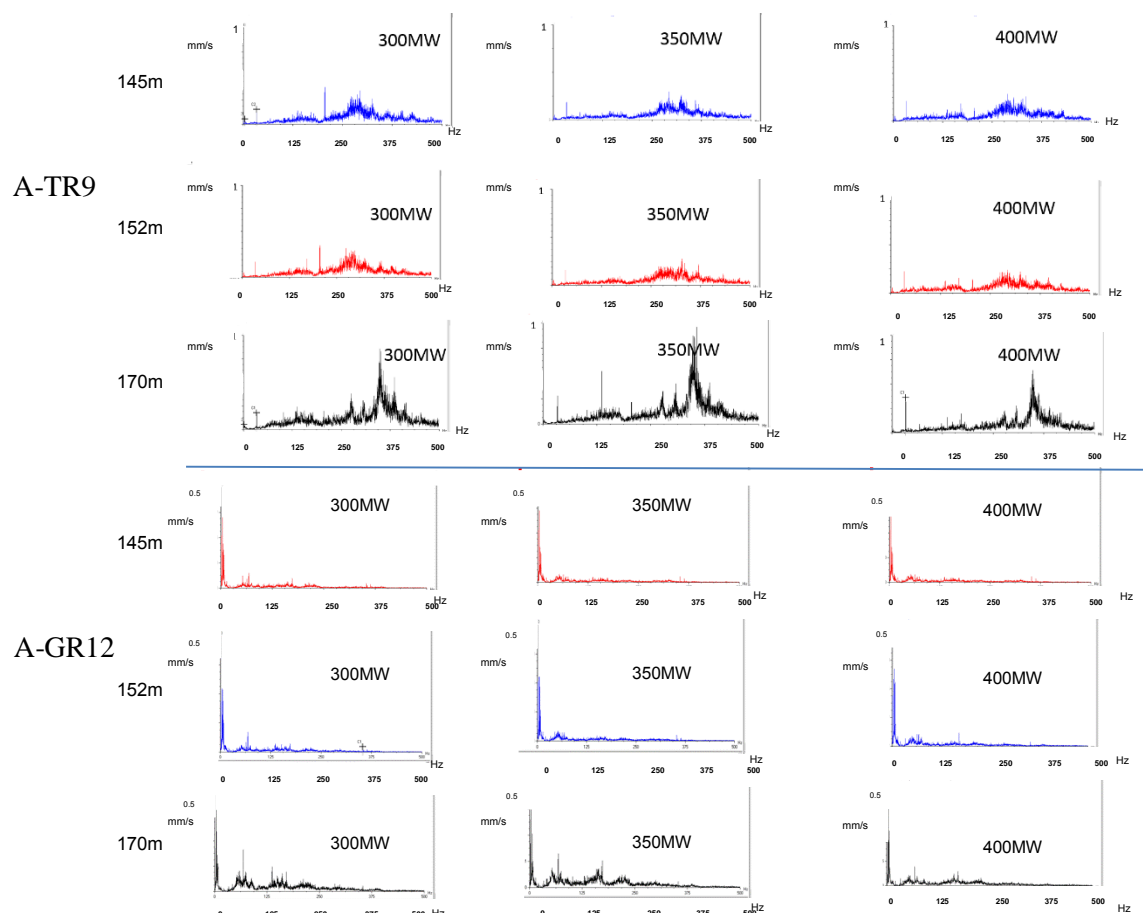


Figure 4. Vibrations signatures (RMS mm/s) for different gross heads and loads for A-TR9 (up) and for A-GR9 (down)

Vibrations can be generated by different type of excitation forces of hydraulic, mechanical and electromagnetic origin. Hydraulic excitations can have periodic characteristics like rotor/stator

interaction (RSI) or random characteristics like turbulence and cavitation. The final vibration amplitudes depend on the excitation type and on the response (hydraulic and mechanical) of the system. The main periodic excitation in hydraulic turbines are generated by RSI which is generated by the interference between the rotating blades of the runner and the stationary guide of the distributor. (RSI) [3,4]. Its frequency is:

$$f_b = n \cdot Z_b \cdot f_f \quad (1)$$

Where f_f is the rotating frequency of the runner, Z_b is the number of blades and n is the order of harmonic. Taking a look at the vibration spectra in the turbine and generator bearings (Figure 4), it can be seen that for a constant load (for example 300MW) vibrations increase when gross head increases especially in the band between 250Hz and 350 Hz. For a constant head, vibration signatures don't change significantly; minimum values of vibrations are observed at 400MW which is near the design point where flow turbulence should be at minimum. Vibrations in generator bearing are lower than in the turbine bearing because flow excitations act basically on the turbine bearing.

In Figure 5 the vibration signal measured in the turbine bearing at 300 MW with 152m of head gross is shown. Some periodic peaks can be identified at 34.21Hz and its harmonics generated by RSI and a broad band vibration almost in all the frequency range indicating that turbulence/cavitation is the predominant excitation. The system response enhances vibrations in some of the frequency bands.

In Figure 6 vibration signals from generator bearing at 300 MW with 152m of head gross are shown. For this position, the main peaks are detected at the rotating frequency of 2.18Hz (f_r) and its harmonics are detected. Also, natural frequencies at around 4,8 and 7.81Hz are a little bit excited. Based in a FEM model, this are rotor natural frequencies.

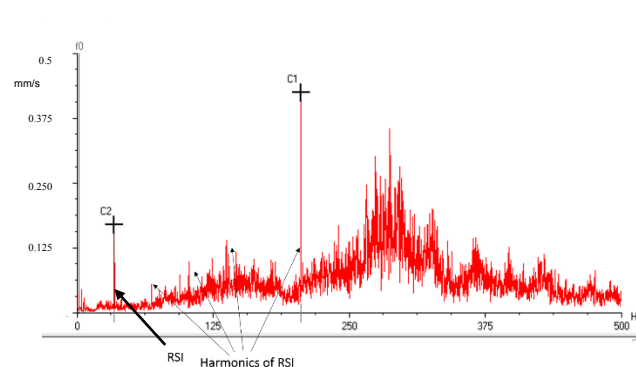


Figure 5. Spectral signatures for A-TR12

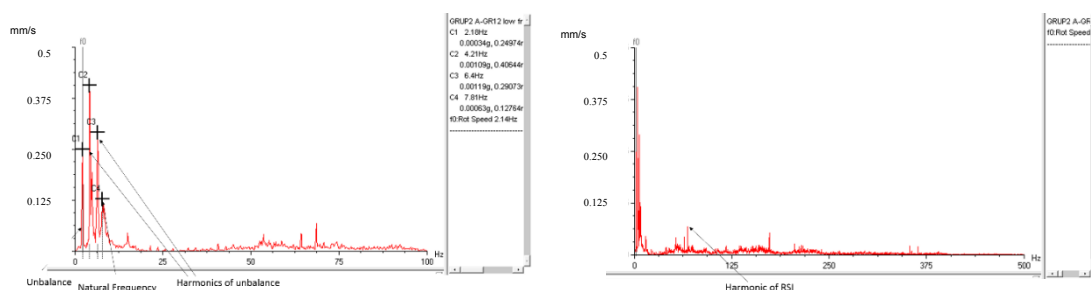


Figure 6. Spectral signatures for A-GR12 bearing from 0 to 100 Hz (left) and from 0 to 500Hz (right)

When the machine operates at very low load (for example at 100MW), vibrations increase in the radial and axial accelerometer located in generator as shown in Figure 7. The natural frequencies are more excited than before because when operating at very low loads the hydraulic excitation increases.

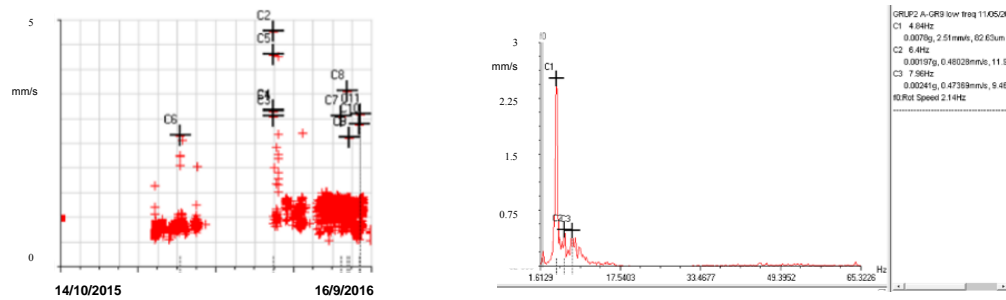


Figure 7. Overall levels RMS (mm/s) (right) and power spectrum (left) for AGR-9

In Figure 8 a waterfall spectra is shown for the machine working from start up to maximum load. When the load reaches 200MW and 250MW a part load instability appears and a power instability is produced. Also, an overload instability can be detected at 470 MW when gross head is maximum.

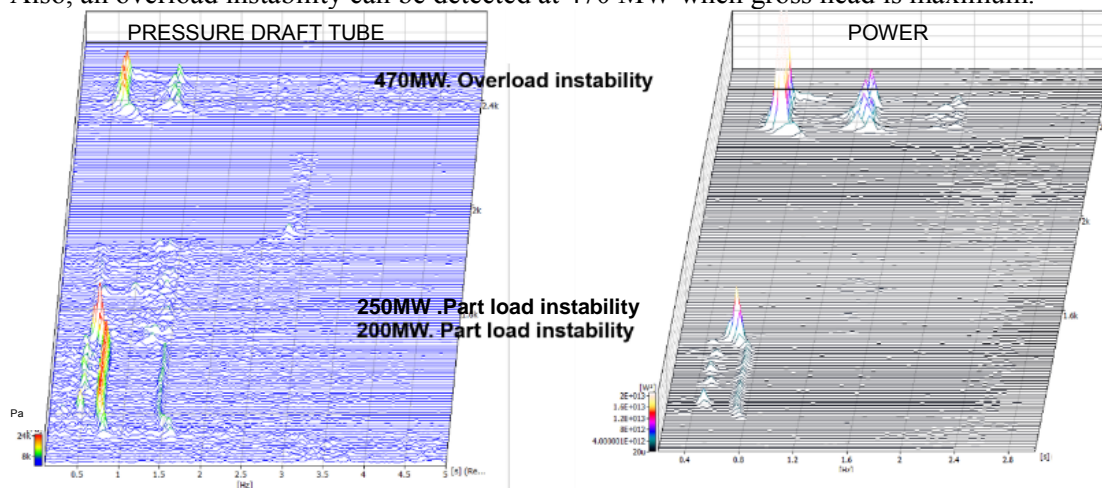


Figure 8. Waterfall spectra working machine from start –up to maximum load

4. Conclusions

This paper presents a preliminary analysis of the measurements obtained from an on-line monitoring system installed in a Francis turbine located in British Columbia (Canada). When the machine is working at normal operating conditions (300-430MW) overall vibration levels increase when head gross increases. Maximum values of vibrations are reached at 300MW with maximum gross head. Most important amplitudes are measured in the turbine bearing and in the draft tube. Predominant excitation is turbulence for all cases. Vibration amplitudes also increase when the machine works at very low loads. At part load (200-250MW) and at overload (470MW) instabilities appear inducing power fluctuations.

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