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# Behavior of Francis turbines at part load. Field assessment in prototype: Effects on the hydraulic system.

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**Abstract.** At present, hydropower plants play an important role because of their flexibility in the generation of electricity. Today, the required operating range of turbine units is expected to extend to deep part load. At part load the draft tube vortex rope is generated and, in some cases, can produce large pressure pulsations limiting the operation of the turbine. The draft tube vortex rope has been much studied.

In this paper, the effects of the part load behavior on the hydraulic system in existing power plants has been studied. Different hydropower units with Francis turbines has been selected for this purpose. All of them present strong vibrations in penstocks and other machine components at part load. Vibrations, pressure fluctuations and other parameters were measured in different positions of the turbine, the draft tube and the penstock. Signals were acquired during transients and for several operating loads. The signals in time and frequency domain and other signal treatments for these cases are shown.

## 1. Introduction

Years ago, the turbines used to operate around the best efficiency point and avoiding part load and overload operation. Today, due to the development of renewable energy, the extension of the operating range of hydraulic machines to off-design conditions is more and more requested by hydropower operators, even if the lifetime of the machine is reduced [1].

When machines operate at off-design conditions many problems may appear. Strong pressure pulsations, instabilities in the draft tube and excessive turbulence can generate excessive vibration levels that can damage the machine.

Flow instabilities in the draft tube are basically caused by the interaction of the vortex rope with the overall system [1-4].

Vortex cavitation is one of the most typical types of cavitation in Francis turbine. Because the blades in Francis turbines are fixed, they cannot adapt to changes in the flow conditions what affects the characteristics of the flow leaving the runner. Then, when the turbine is operating at off-design conditions, the flow leaving the runner has a rotating component and forms a vortex.

Because this vortex is generated at the exit of the runner where the pressure is at the minimum, the pressure inside the vortex rope can reach the vapor pressure, and then cavitation appears which may induce pressure fluctuations that propagate in the whole hydraulic system [5-6]. The interaction between this excitation source and the system in case of resonance may lead to high-pressure fluctuation, power swing and large level of vibrations [7-10].



As the occurrence of resonance is a problem for the extension of the operating range of Francis turbine at partial load, a better understanding of the consequences of this phenomenon is necessary in actual hydropower plants.

In this paper, three different cases found in existing power plants are studied. First, two cases of pipe vibration generated as a consequence of resonance between the rotating vortex rope and the hydraulic system are shown. In the first case the excitation is generated by the vortex rope and the second by a twin vortex [3][8]. Second, a case where the resonance produced large machine vibrations is discussed.

## 2. Experimental investigation

### 2.1. Description of the selected turbines

Three different hydropower units with Francis turbines have been selected. All of them presented strong vibrations in penstocks and other machine components at part load.

The main characteristics of the hydropower plants selected are showed in the following table (N: revolutions/minute, W: maximum power (MW), Q: flowrate (m<sup>3</sup>/s), H: net head (m)).

**Table 1.** Main characteristics of the power plants.

Hydropower 1	Hydropower 2	Hydropower 3
N=250rpm	N=360rpm	N=300rpm
W=11MW	W=11MW	W=35MW
Q=25m <sup>3</sup> /s	Q=7,2m <sup>3</sup> /s	Q=15m <sup>3</sup> /s
H=53m	H=112m	H=211m

### 2.2. Instrumentation

Several sensors were installed in the different parts of the machines and their signals were acquired simultaneously using an acquisition system Brüel&Kjaer LAN XI Type 3053. Sensors were placed in the hydraulic and mechanical systems of the machine.

For the hydraulic system, pressure sensors (WIKA S-10) were installed in the draft tube, spiral casing and penstock. In the mechanical parts, accelerometers (Kistler type 8752A5) were located in the stationary parts of the machine (bearings and draft tube walls), and displacement probes (VK-202A) in turbine bearing.

### 2.3. Testing procedure

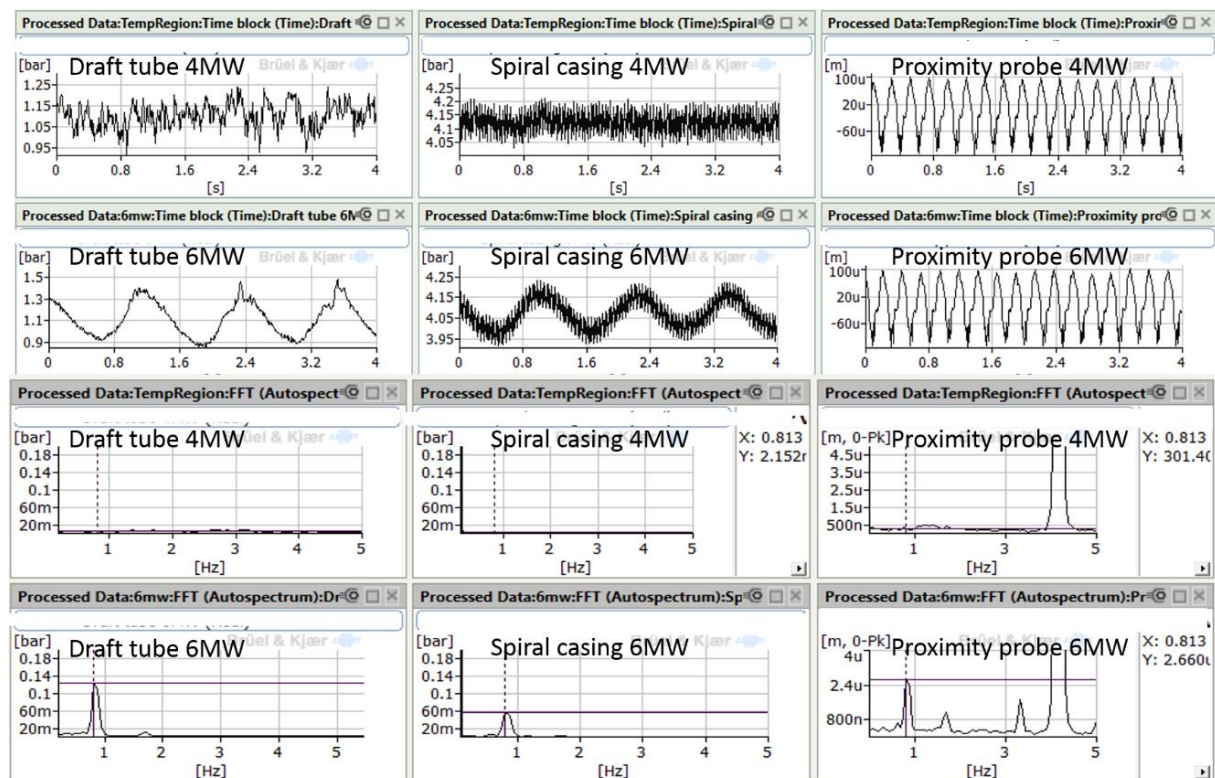
Machines were operated at its whole range of operation. Slow ramp-up and ramp-down (from minimum to maximum power and vice versa) were done, as well as different steps at constant power during 10 minutes in order to have steady conditions of the machine.

## 3. Results obtained

### 3.1. Hydropower 1

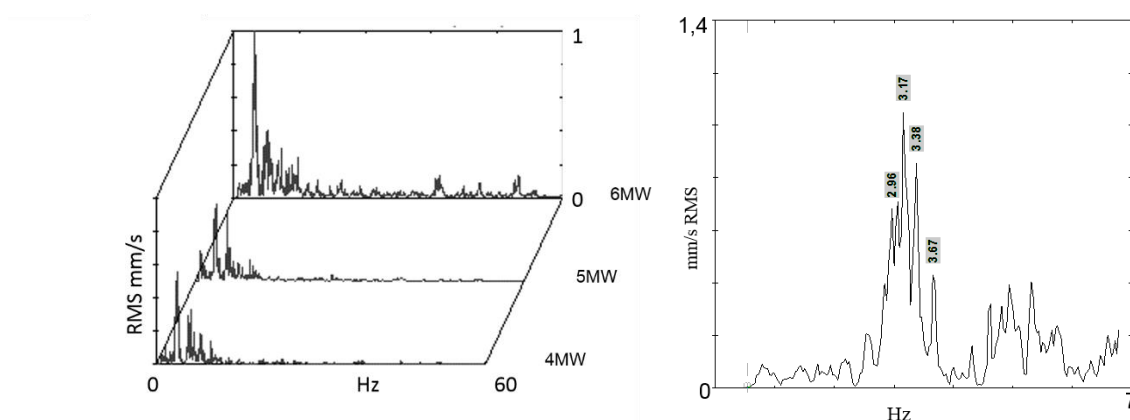
Vibrations in penstock between 6 and 8 MW related with vortex rope frequency were detected.

In Figure 1, pressure fluctuation versus times in draft tube and in spiral casing and displacement versus time in the shaft has been represented. Also, power spectrum has been represented for the same cases and positions. When machine achieves 6 MW a vortex cavitation appears. The frequency of this perturbation is 0.813Hz ( $\cong 0.2 \cdot$  rotational frequency) (see Figure 1.)

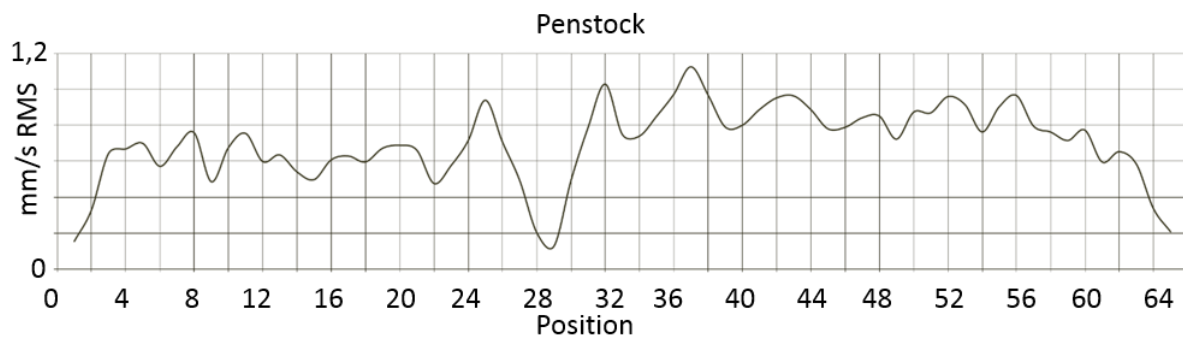


**Figure 1.** Time signal and power spectrum for draft tube pressure fluctuation, spiral casing pressure fluctuation and proximity probe for 4MW and for 6MW.

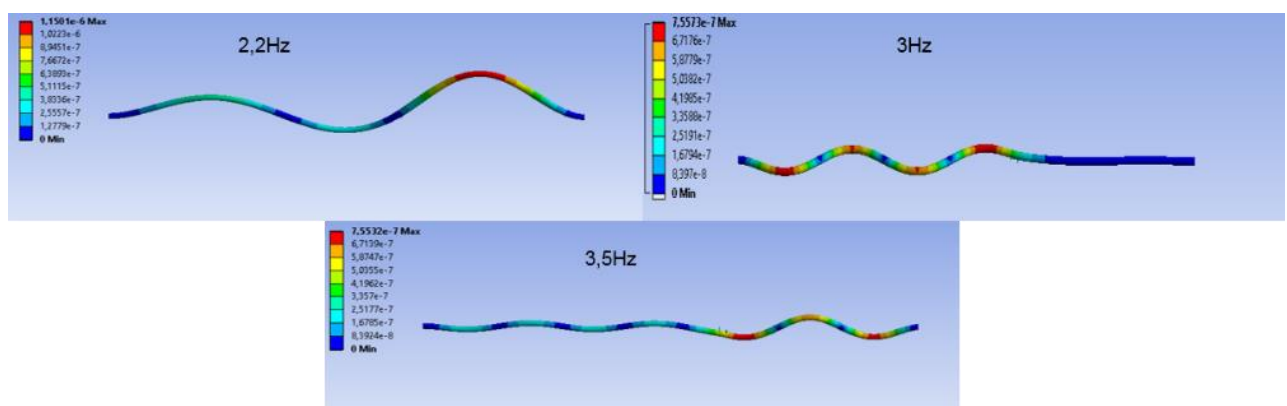
For this power plant, a direct relationship between the apparition of vortex rope frequency and vibrations in the penstock has been found. In Figure 2. (left) power spectrum for penstock is shown for 4, 5 and 6 MW. At 6 MW largest vibrations in the pipe are detected. The apparition of vortex rope frequency excites natural frequencies in the penstock. In Figure 2. (right) main frequencies detected at 6MW are showed. In Figure 3. vibrations measured at 6 MW along the pipe can be seen. In addition, a numerical study of the natural frequencies of the penstock with water has been done with ANSYS to verify the natural frequencies. First mode shapes has been represented in Figure 4.



**Figure 2.** Power spectrum vibrations for penstock at 4MW, 5MW and 6MW (left) and power spectrum at 6MW (right).



**Figure 3.** Vibration (mm/s RMS) of the penstock at 6MW.

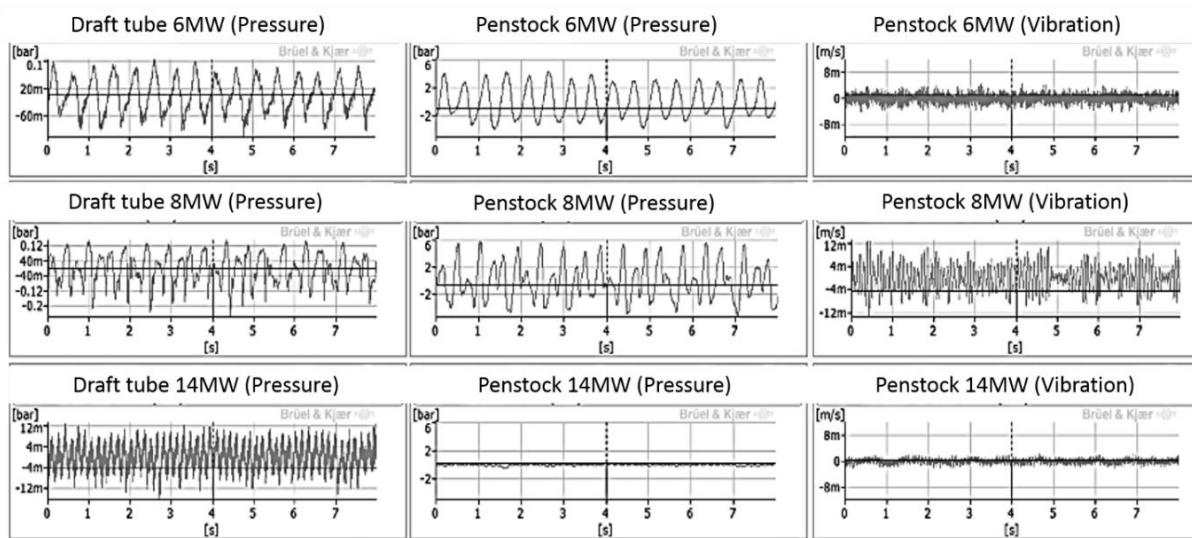


**Figure 4.** First mode shapes and natural frequencies of the penstock.

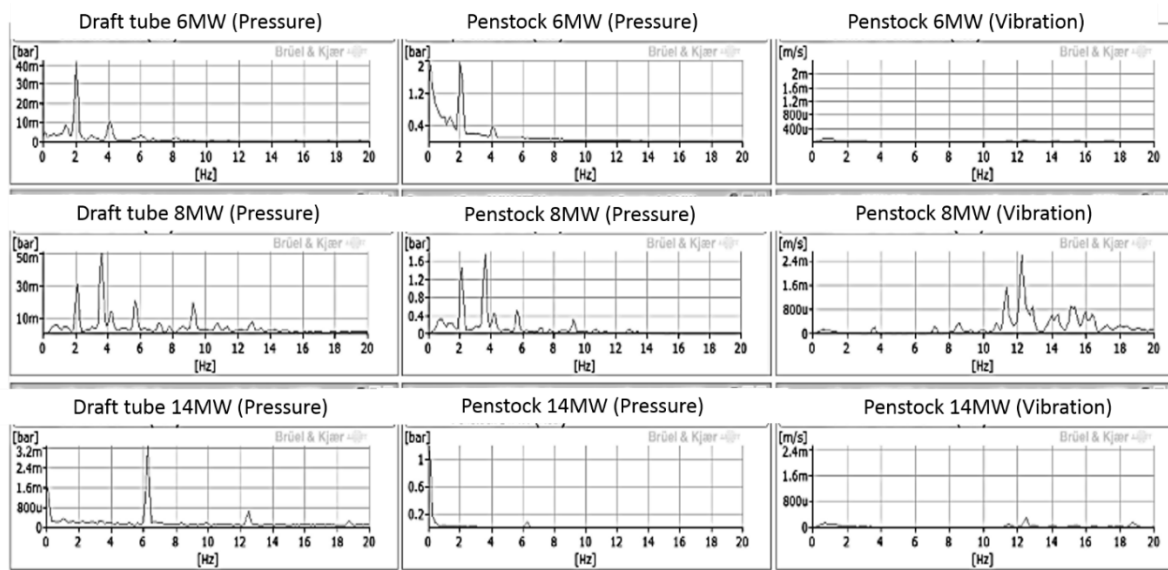
### 3.2. Hydropower 2

Vibrations problems in the penstock of this power plant appeared when machine worked between 7.5MW and 9MW.

The amplitude of pressure fluctuations versus time and power spectrum for pressure transducers located in the draft tube and penstock are showed in *Figure 5*. and *Figure 6*. It can be seen than the frequency at 0.3 times the rotational frequency and its harmonics appears for 6MW and 8MW. This frequency is the vortex rope frequency. At 8MW a twin vortex showed up at 3.625Hz. The resonance produces high vibrations in penstock.



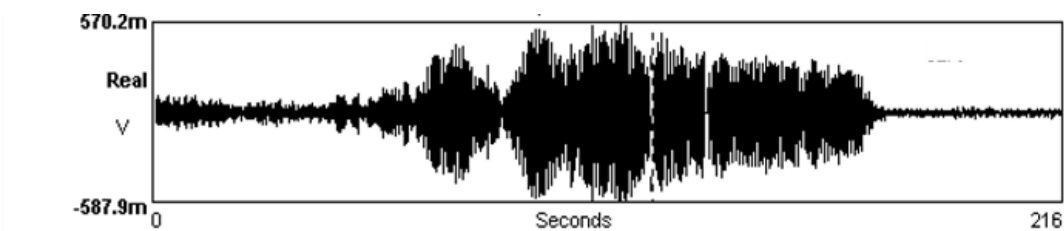
**Figure 5.** Time signal of the pressure fluctuation in draft tube and penstock and vibrations in penstock (6MW top, 8MW middle and 14MW bottom).



**Figure 6.** Power spectra of the pressure fluctuation in draft tube and penstock and vibrations in penstock (6MW top, 8MW middle and 14MW bottom).

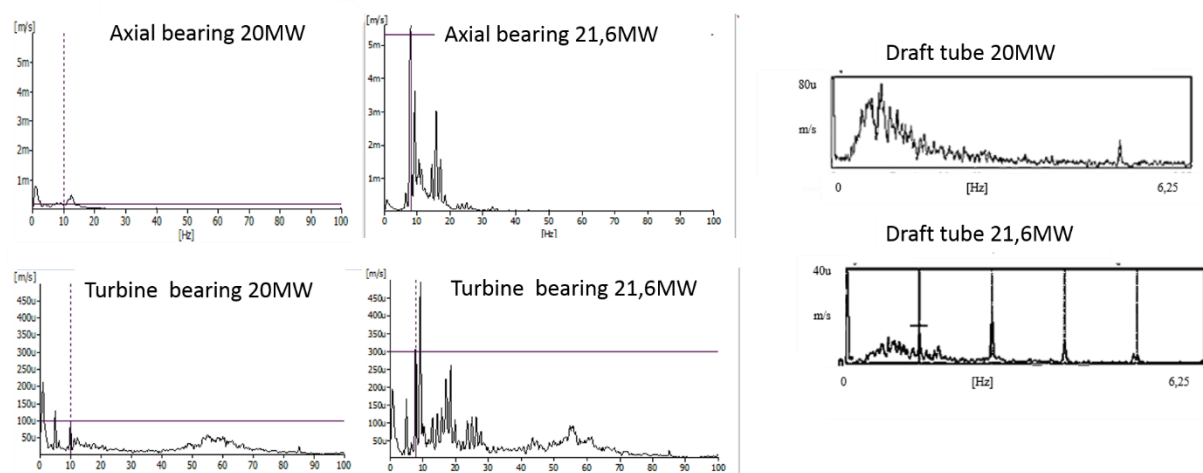
### 3.3. Hydropower 3

In this power plant, serious vibrations occurred when machine achieved about 21.6MW. In this case, a very important vibration in axial direction appeared (see **Figure 7.**).



**Figure 7.** Time signals for vibration in axial generator bearing.

In Figure 8, vibrations in axial and turbine bearing and in draft tube at 20MW and 21.6MW is represented. It can be seen that a frequency at  $\sim 0.2$  times rotational frequency is detected in the draft tube who corresponds at vortex rope frequency. This flow perturbation amplified by resonance excites a natural frequency of the machine and causes strong vibrations.



**Figure 8.** Power spectra of the vibration in axial and turbine bearing (left). Draft tube vibration (right).

#### 4. Conclusions

Three different cases showing the effects of part load resonance on the pipes and machine have been presented.

In the first cases, an important increase in the penstock vibrations related with the vortex rope frequency were generated that produced leakage in the pipe connections.

In the last case, the rotor natural frequencies were excited by the pressure fluctuation. Large vibrations were generated in the turbine bearing and in the thrust bearing that were not identified during the operation of the machine. The result was an accelerated wear in both bearings.

The resonant conditions were well detected with pressure transducers located in the penstock in the two first cases. An accelerometer in the trust bearing is also convenient to see the effects of the resonance in the machine.

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