LOW FREQUENCY PRESSURE PULSATIONS AT HIGH LOAD IN A FRANCIS TURBINE

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Due to energy market demands on ancillary grid services the hydro units are subject to off design operation, with more frequent starts and stops. The Francis turbine, operated at off design has a large swirling component at both high and low load, potentially damaging the rotor elements. The paper presents a case study of Francis turbine developing low frequency pressure pulsation at high load, in condition of poor aeration of draft tube. Also, are presented vibration measurement in connection with this particular case.

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Keywords: pressure pulsation, Francis turbine, vibration

1. Introduction

Recent years have marked redesign of the electricity market in order to increase security of supply and integrate a multitude of new producers, in particular renewable energy sources.

New challenges face hydroelectric power plants as the main provider of ancillary grid services and peak load energy. Therefore, hydro units have to be increasingly operated outside the optimum design point which cause additional stress.

In particular, in the case of Francis turbines operated at off-design point, at both high and low load, the flow leaving the runner has a swirling component that is the main cause of pressure fluctuations in the draft tube [1]. These pressure pulsations generate shocks and vibrations with failure potential both in rotating and static elements of the turbine [2]. The draft tube pressure pulsation may be decomposed in a "synchronous" and "asynchronous" part considering their spatial relationships, propagation mechanism and origin of physical phenomena [3]. The "synchronous" part is associated with a plane pressure wave propagating through the draft tube producing power swing, and axial vibration (runner, shaft, generator support bracket). The asynchronous component presents a pressure pattern rotating about the circumference of the draft tube and produce radial vibration of the shaft and low frequency components in the draft tube [1].

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At high load the asynchronous pulsation is very low and almost all of the pulsation in the draft tube cone is synchronous.

When units operate at full load a vapour/air void, similar to a surge chamber connected to the draft tube, may be created under the runner cone, rotating in the opposite direction of the runner and acting like mass oscillator [4]. There are reported cases [5] when power and pressure oscillations restricts the maximum available power output. Operation in condition of high pressure pulsation may cause structural vibration, cracks or failure of the runner blades [6], and even resonance of the powerhouse.

One solution to reduce draft tube surges is the admission of air into the draft tube conventional from the center of the runner cone [1,7]. Other methods include injection of air through the runner head cover or air duct attached to the runner cone [2,7], or using a water jet issued from the crown tip [8].

The aim of this paper is to report a case study of Francis turbine developing low frequency pressure pulsation at high load, in condition of poor aeration of draft tube. Also, vibration measurement in connection with this particular case are presented.

2. Francis turbine specifications and experimental set up

Specifications of Francis Turbines are: 50 MW nominal power, 144 m maximum head, 125 m design head, 47 m^3/s nominal flow, 300 rot/min nominal speed (main frequency $f_n = 5$Hz). The runner has 17 blades with the reference diameter D=2300 mm and the turbine distributor consists of 9 stay vanes with 18 guide vanes the wicket gate. The turbine has a self-adjusting air admission, one direction valve type, with no specific transducer for air flow measurement.

The hydro unit is part of a powerhouse with six hydro generators (4 x 27.5 MW, 2 x 50 MW) each equipped with Francis turbines; all generating units qualify for ancillary grid services. The hydropower facility consists of a concrete gravity dam, tunnel under pressure (4700 m length, 7 m diameter), surge tank, two penstocks, stilling basin and tailrace channel. The length of each penstock is 154 meters, 4 m diameter; the right one has four branches for each of 27.5 Mw units (1 to 4) and the other and has two for 50 MW units (5 and 6). The head can vary within a year between 106 – 144 m.

The experimental set-up used three pressure transducers located as follows (presented in figure 1): 1. Pressure on draft tube cone in median section (label P_{DT}) 2. Pressure on spiral case (label P_{SC}) 3. Pressure at the penstock base, (label P_M). Other three vibration transducers were located: 1. Absolute vibration of the draft tube cone, in radial position near the lower ring (label Vib DT)
2. Absolute vibration of the upper bracket that hold the generator thrust bearing in axial direction (label Vib Ax) 3. Relative vibration of the shaft measured near the

turbine bearing – radial position (label Vib T). Guide vane opening variation was linear during the measurements between 15% - 85 %. Gross head was 135 m, flow at 85 % guide vane opening was around 43 m^3/s. Two channel synchronous recording were performed in order to estimate the correlation between channels.

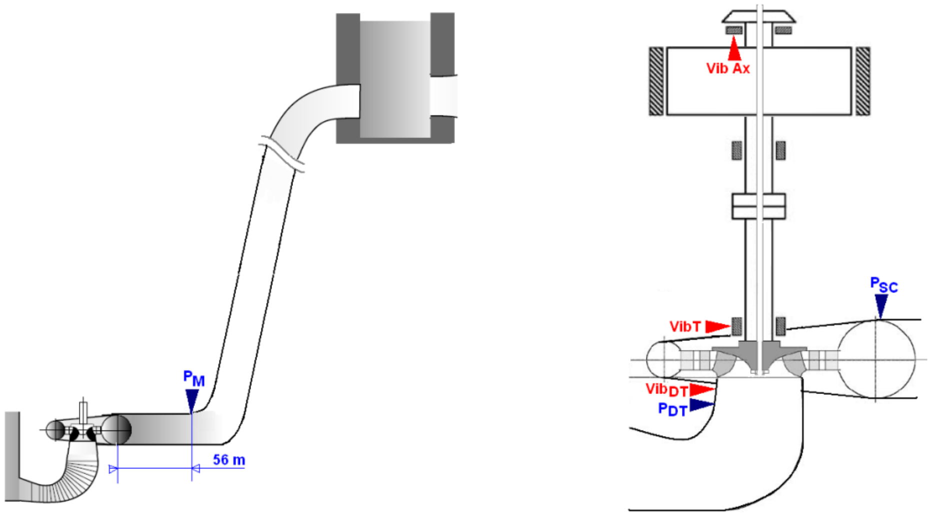

Fig. 1. Turbine, penstock and surge tank. Pressure transducer P_M placed at the elbow penstock limit (left). Turbine and hydro generator section with measuring points (right).

3. Experimental investigation

During high load operation of the unit strong mechanical vibrations and noise were noticed. Axial vibration was also present on the upper bracket that hold and support the generator combined radial and thrust bearing.

Pressure measurement performed in the spiral case highlight a pulsation with a frequency between 2.250 Hz - 2.375 Hz, $(0.45 - 0.475) \cdot f_n$, amplifying with guide vane opening over 75%. Superior harmonics tend to develop with increasing flow as in figure 2 (right). Figure 2 (left) presents the pressure autospectrum measured on the spiral case, as a function of guide vane opening. Pressure pulsation at partial load, at the same frequency band, are also present with amplitude lower than high load operation

The large value of pressure pulsation detected in spiral case induced the possibility that it might be the result of a hydraulic conduit natural frequency [10]. A pressure measurement (noted P_M in figure 1) at the base of the penstock, elbow limit, was performed. The distance between spiral case and the penstock elbow is around 56 m. The pressure autospectrum measured at the penstock base point location, as a function of guide vane opening, has a similar pattern, as it is presented in Figure 3 (left). Frequency lines of hydraulic self-oscillations along with antiresonance are present above $1.2 \cdot f_n$. Some of them are excited by the harmonics of low frequency order.

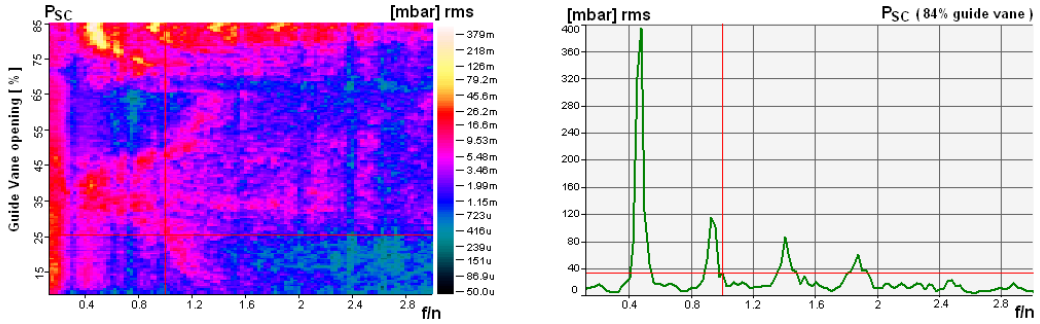

Fig. 2. Waterfall diagram of the autospectrum of pressure pulsation on the spiral case (P_{SC}), as a function of guide vane opening and reduced frequency *f/n* (left). Autospectrum of pressure pulsation on spiral case (P_{SC}), 84 % guide vane opening (right)

At high load the amplitude of main pressure perturbation frequency is attenuated with 4 dB. The frequency component with dominant amplitude continue to be at 2.375 Hz when guide vane opening is over 75 % as in figure 3 (right).

Pressure pulsation measurement performed on draft tube has different amplitudes but the same distribution of frequency as on spiral case as presented in figure 4 (left).

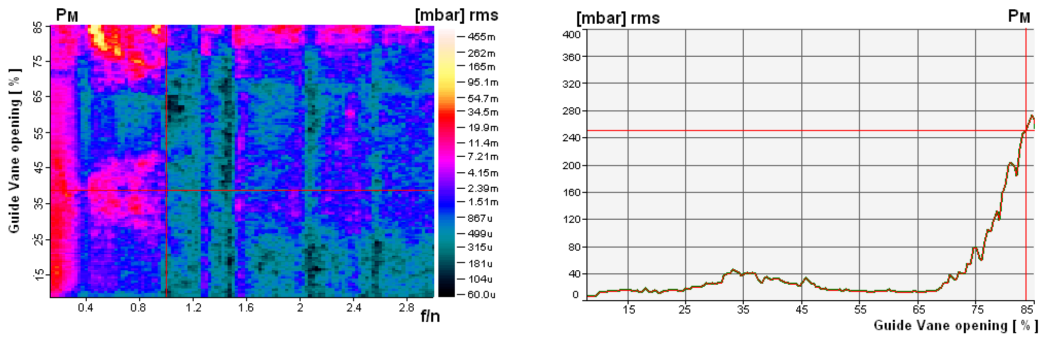

Fig. 3. Waterfall diagram of the autospectrum of pressure pulsation on penstock location (P_M) as a function of guide vane opening and reduced frequency *f/n* (left). RMS value of pressure pulsation (P_M) in low frequency band as function of guide vane opening (right)

Based on two channel measurements performed simultaneous it may be estimated the autospectrum and cross-spectrum between the signals. Several functions derived from these three spectra are available as coherence, cross-correlation or frequency response function.

Coherence ($\hat{\gamma}_{xy}^2$) measures the degree of linear relationship between two signals at a given frequency and is estimated by

$$\hat{\gamma}_{xy}^2(f) = \frac{|\hat{G}_{xy}^2(f)|}{\hat{G}_{xx}^2(f)\cdot\hat{G}_{yy}^2(f)} \tag{1}$$

$\hat{G}_{xx}(f)$ and $\hat{G}_{yy}(f)$ are the estimated auto spectral density functions of $x(t)$ and $y(t)$ and $\hat{G}_{xy}(f)$ is the estimated cross-spectral density function between $x(t)$ and $y(t)$ [11].

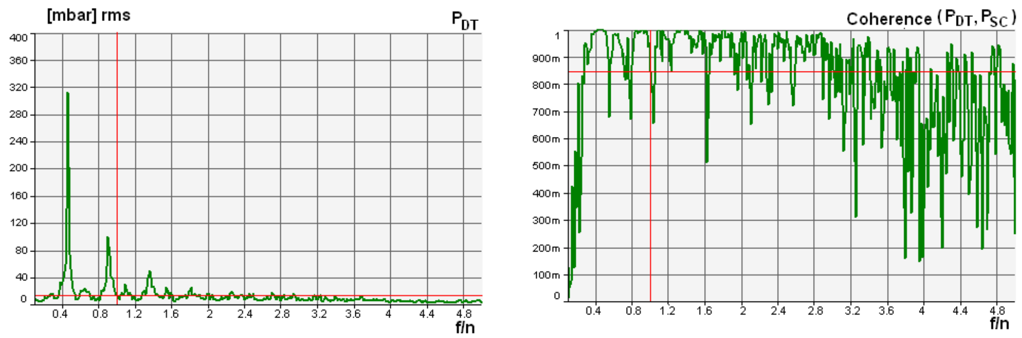

Fig. 4. Autospectrum of pressure pulsation on draft tube cone (P_{DT}), as function of reduced frequency *f/n*, 82 % guide vane opening (left). Coherence between pressure pulsation on draft tube (P_{DT}) and spiral case (P_{SC}) in low band frequency, 82% guide vane opening (right)

In Fig. 4 (right) coherence function estimated for pressure pulsation signals P_{DT} and P_{SC} is presented. In low frequency range $(0.35 - 3.2)\cdot f_n$ almost all component has coherence near one so the pressure instability is extended in a large bandwith.

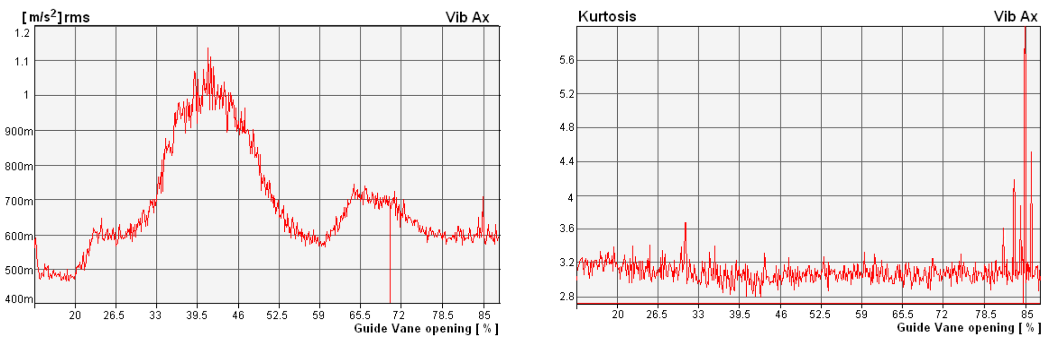

Fig. 5. RMS value of axial vibration (Vib Ax) as function of guide vane opening (left). Kurtosis of axial vibration (Vib Ax) as function of guide vane opening (right)

Axial vibration was measured on upper bracket that hold the generator thrust bearing. In figure 5 (left) is presented RMS value of acceleration as a function of guide vane opening. The maximum value is around the partial load, and this is assumed to be due the high frequency components and different transmissibility of the systems involved.

Kurtosis is the fourth moment normalized dividing by the appropriate power of the standard deviation [12]. For normal distribution kurtosis is 3, larger for impulsive signals. Kurtosis is estimated for axial vibration signal, Vib Ax, at quasi stationary guide vane opening and is almost equal to three, meaning that axial vibrations have a normal distribution. At high load (figure 5, right) kurtosis value tend to be higher than three as an effect of shocks induced by pressure instability on the runner.

Once measurements have been completed condition of the runner blades was dye checked after the draft tube cone has been dismantled. Cracks were found on the trailing edge near the band (presented in figure 6).

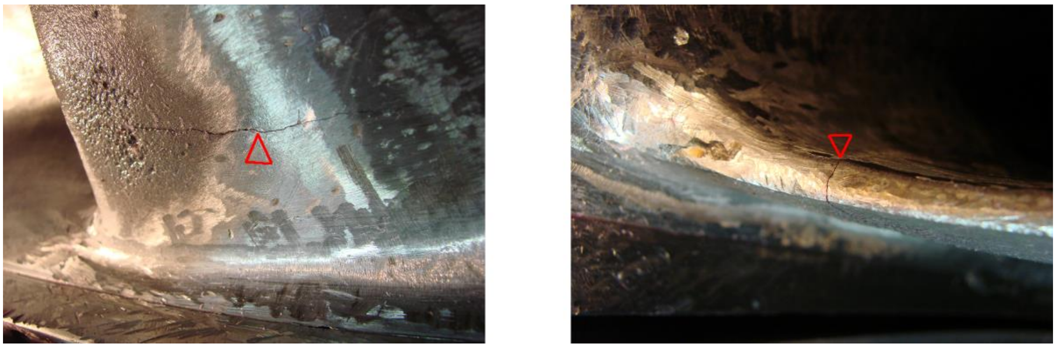

Fig. 6. Cracked runner blade at the band limit (left). Trailing edge crack at the blade/band limit (right)

The air valve was verified and proper air flow was restored. Low frequency pressure fluctuation at high load were reduced significantly. However, high pressure pulsation is still present at partial load. Vibration measurements performed near the turbine bearing indicate that radial shaft vibration has a low frequency component with high amplitude.

In figure 7 (left) is presented the autospectrum diagram as function of guide vane opening. The vortex induced vibration has a frequency between 0.85 Hz – 1.6 Hz with higher amplitude at 1.365 Hz $(0.273 \cdot f_n)$. In vibration spectrum a residual frequency at 2.5 Hz is still present at high load but the amplitude is lower than main rotation component. At partial load draft tube cone absolute vibration has a distinctive pattern as it is presented in figure 7 (right).

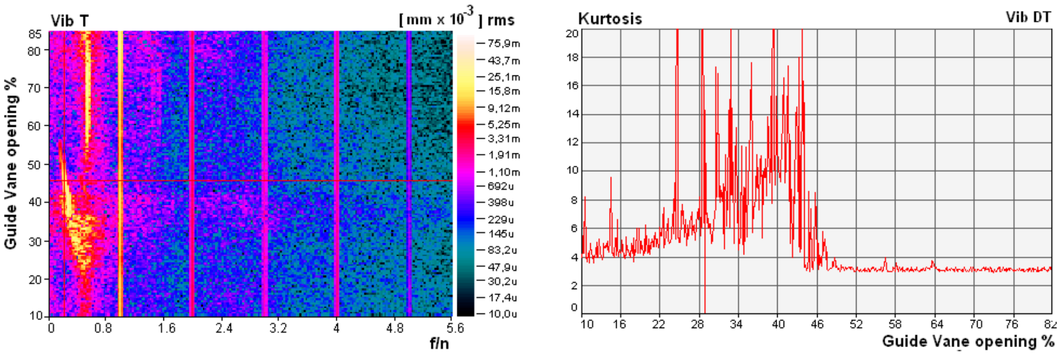

Fig. 7. Waterfall diagram of the autospectrum of radial shaft vibration on turbine bearing (Vib T), as function of guide vane opening and reduced frequency *f/n* (left). Kurtosis of absolute vibration on draft tube cone (Vib DT) as function of guide vane opening (right)

Kurtosis is far over 3 at partial load, and near 3 in all operation range 50 % - 82 % guide vane opening. Impulsive vibration is distinctive mark to partial load operation.

6. Conclusions

Pressure pulsation at high load has a low frequency component of high amplitude expanded in draft tube, spiral case and penstock. Pressure instability induces noise, axial impulsive vibration and may crack runner blades. Air admission is beneficial and effectively reduce pressure pulsation. Radial shaft vibration is affected by the partial load pressure pulsation.

Further studies are necessary to assess air admission effect on efficiency,

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