Experimental Identification and Study of Hydraulic Resonance Test Rig with Francis Turbine operating at Partial Load

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Abstract. Resonance in hydraulic systems is characterized by pressure fluctuations of high amplitude which can lead to undesirable and dangerous effects, such as noise, vibration and structural failure. For a Francis turbine operating at partial load, the cavitating vortex rope developing at the outlet of the runner induces pressure fluctuations which can excite the hydraulic system resonance, leading to undesirable large torque and power fluctuations. At resonant operating points, the prediction of amplitude pressure fluctuations by hydro-acoustic models breaks down and gives unreliable results. A more detailed knowledge of the eigenmodes and a better understanding of phenomenon occurring at resonance could allow improving the hydro-acoustic models prediction.

This paper presents an experimental identification of a resonance observed in a close-looped hydraulic system with a Francis turbine reduced scale model operating at partial load. The resonance is excited matching one of the test rig eigenfrequencies with the vortex rope precession frequency. At this point, the hydro-acoustic response of the test rig is studied more precisely and used finally to reproduce the shape of the excited eigenmode.

1. Introduction

Extending the operating range of Francis turbine forces the machine to experience pressure fluctuations, induced by the development of a cavitating swirling flow at the runner outlet, see [1]. These can propagate into the entire hydraulic system and induce consequently torque and power fluctuations which can be unacceptable under certain operating conditions. During reduced scale model tests IEC 60193, pressure fluctuations are measured in order to obtain the hydro-acoustic response of the test rig for deriving the pressure fluctuations sources, see [2]. In the case where there isn't significant interaction with the hydraulic system, pressure fluctuations amplitude can be transposed fairly well from the model to the prototype, see [3]. However, when the excitation frequency matches one of the hydraulic system eigenfrequencies, resonance characterized by large undesirable pressure and output power fluctuations occurs, potentially dangerous for the stability of the entire power plant. For this case, prediction of the pressure fluctuations amplitude by one-dimensional hydro-acoustic models becomes unreliable.

Zielke is one of the first to attempt to predict and simulate part load resonance. In [4], eigenfrequencies and eigenmode shapes of a pressurized piping system susceptible to be excited by vortex rope were computed by the method of transfer matrix. Brennen calculates in [5] the dynamic

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transfer matrix of a cavitating pump introducing a key parameter, called cavitation compliance. It represents the compressibility of the cavitating vortex and it is defined as the rate of the cavitation volume with change of pressure. In [6], Dörfler determines it experimentally for a Francis turbine as function of the Thoma number and predicts pressure and torque fluctuations in the entire hydraulic system using a transfer matrix method. Another approach to determine experimentally the transfer matrix of the turbine is possible, using an external excitation source, see [7]. Recently, Alligné [8] performs a stability analysis of a hydraulic system with Francis turbine and derives the eigenmodes shape and frequency in function of the wave speed in the draft tube. He notably highlights the deformation of the eigenmodes shape by the cavitation volume in the draft tube and the influence of the wave speed on the eigenfrequencies. This is also highlighted experimentally by Ruchonnet [9] in a simple system composed with one pipe and one bluff body inducing cavitating wake.

However, more precise prediction of resonance in hydropower plants needs a more detailed knowledge of eigenmodes shape and frequency and a better understanding of phenomenon occurring at resonance conditions. For this purpose, an experimental identification of a resonance in a close-looped hydraulic system with a Francis turbine reduced scale model operating at part load is described in this paper. One of the test rig eigenmodes is excited by the vortex rope precession excitation. More precisely, the eigenfrequencies of the hydraulic system are slightly modified increasing Thoma number. The resonance is then identified as the point featuring the maximum of torque and pressure fluctuations. The characteristics of pressure fluctuations around the resonance operating point are studied in both draft tube and test rig conducts. Finally, assuming that the point with the maximum hydro-acoustic response is effectively a resonance, the shape of the excited eigenmode is determined using the hydro-acoustic response of the test rig at this point.

2. Experimental set-up

Experimental investigations are performed on a Francis turbine reduced scale model of specific speed $\nu = 0.27$ installed on the EPFL test rig PF3. Pressure measurements are performed using 14 dynamic pressure sensors arranged throughout the first part of the hydraulic test rig, see figure 1. The draft tube is equipped with 6 pressure taps, whose 2 in the same cone cross-section, one in the elbow and 3 through the right channel of the diffuser. The rest of the pressure sensors is arranged in the first third of the test rig upstream the Francis turbine. Pressure fluctuations measurements are synchronized with torque fluctuations measurement and acquired simultaneously with a PXI-system using a sampling frequency of 1000 Hz

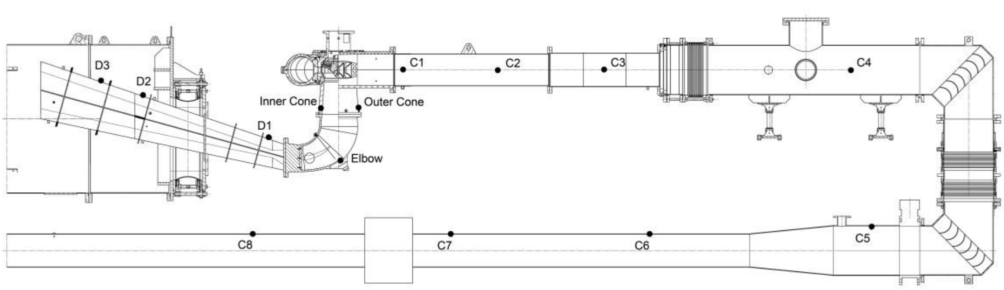

Figure 1. Sensors location on the PF3 test rig.

The turbine operating point selected for this investigation is given by table 1.

Table 1. Selected turbine operating point.

n_{ED}	Q_{ED}	n	E_M	Q_M	Fr
0.318	0.133	15 s^{-1}	272 $J.kg^{-1}$	0.27 $m^3.s^{-1}$	8.9

3. Results for the selected operating point

3.1. Hydraulic resonance identification

At the selected operating point, pressure fluctuations in the whole system and torque fluctuations are measured for different values of σ. It is modified stepwise, from 0.15 to 0.21 in steps of 0.01, then in steps of 0.005 around the identified resonance. The vortex rope precession frequency, identified as the excitation frequency equal to about 0.3 times the runner frequency, remains nearly constant in this narrow range of σ, whereas the test rig eigenfrequencies are modified consequently to the change of cavitation volume in the draft tube. The resonance is reached when the Thoma number allow matching one of the eigenfrequencies with the vortex rope precession frequency.

The Fourier Transform of each pressure fluctuations signal normalized by ρE is computed using Fast Fourier Transform. The spectral analysis of the pressure fluctuations measured in the draft tube cone and in the pipe upstream the turbine (position C2) highlights a maximum response of the system around σ = 0.20 at the frequency 5.05 Hz, see figure 2. This result is confirmed obviously by the spectra analysis of the torque fluctuations (figure 3). As an interpretation of this result, the point σ = 0.20 can be identified as a resonant point and the frequency of 5.05 Hz as one of the test rig eigenfrequencies. This result is only available for the identified point since the important influence of the cavitation volume on the hydraulic system eigenfrequencies.

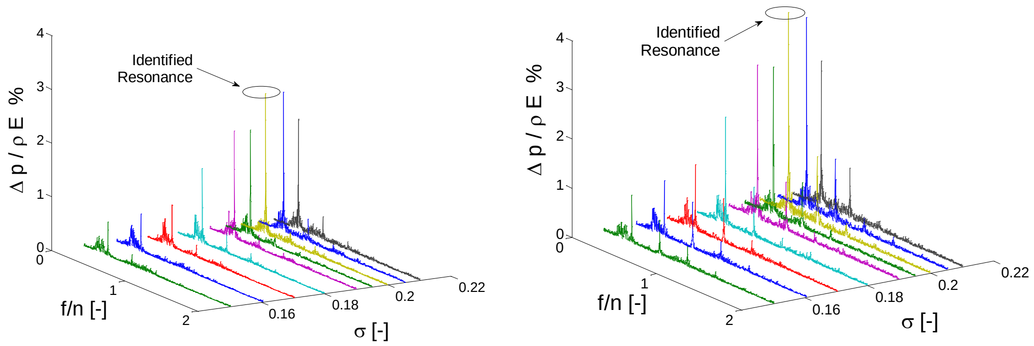

Figure 2. Waterfall diagram of pressure fluctuations at position C2 (left) and inner cone (right).

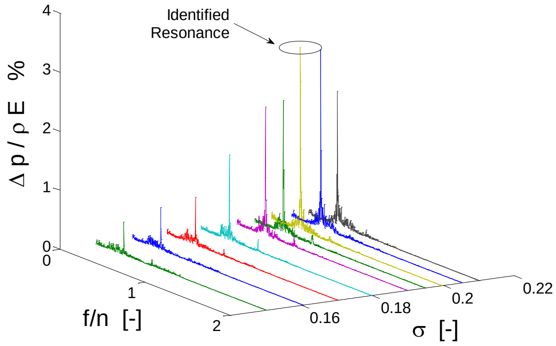

Figure 3. Waterfall diagram of torque fluctuations.

3.2. Pressure fluctuations characteristics around identified resonance condition

3.2.1. *Pressure fluctuations in the test rig conducts.* The pressure signals are filtered with a low-pass filter in order to study only the phenomenon of interest occurring at the vortex rope precession frequency. Figure 4 highlights the evolution of the pressure fluctuations at the two positions C1 and C8 with respect to σ. It is shown that pressure wave propagates in the whole hydraulic system with no loss of amplitude until the position C8, for all of the studied operating points. Out of resonance, the pressure fluctuations at the two positions seem to be very intermittent and the amplitude of pressure fluctuations is relatively small and nearly the same. Approaching identified resonance condition, the pressure fluctuations become more and more regular, with higher amplitudes which reach a maximum at $\sigma = 0.20$. Moreover, the amplitude measured at C8 becomes more important than the amplitude measured at C1, which could be an indication of the eigenmode shape.

The two pressure signals at C1 and C8 positions seem to have a nearly constant time offset for each point. Thus, at the identified resonance, the non-zero temporal phase shift doesn't indicate the development of a pure standing wave in the test rig, contrary to what is theoretically expected. This result must be confirmed by a more precise study of the phase shift between the two signals.

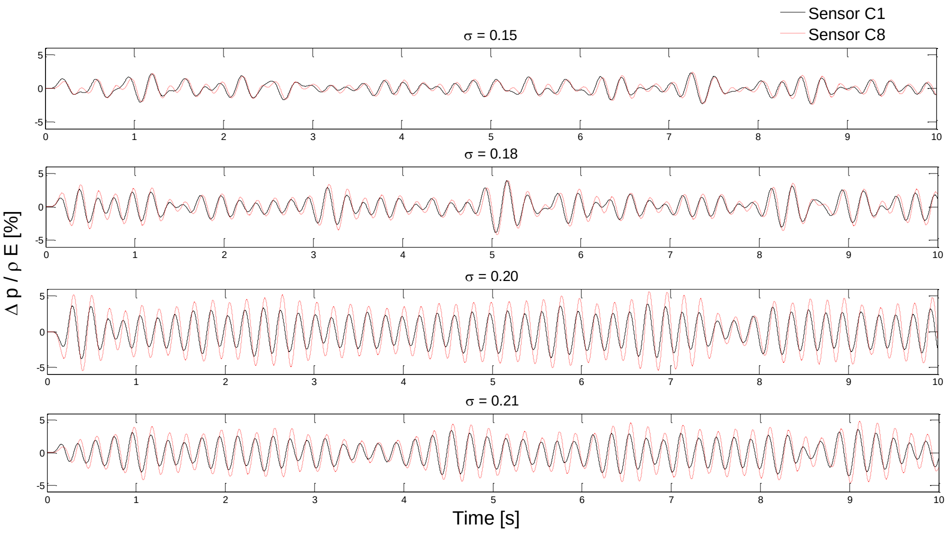

Figure 4. Pressure Fluctuations Factor for different σ values at positions C1 and C8, with low-pass filter at 10 Hz.

3.2.2. *Pressure fluctuations in the cone.* For lower Thoma number, the pressure fluctuations in the cone are very intermittent with low amplitude and are not in phase. This is explained by the precession movement of the vortex rope, which passes successively near the two sensors taps. Approaching the identified resonance condition, the signals are more and more regular with higher amplitudes and the temporal phase shift between the two signals seems to decrease. This observation can be confirmed by a decomposition of the pressure signals in the cone at the vortex rope precession frequency into synchronous and convective components, using a vectorial analysis in the frequency domain [10], see figure 6. The convective component represents the pressure fluctuations due to the rotation of the vortex rope, whereas the synchronous component is a synchronous perturbation due to the non-symmetric rotation of the vortex induced by the elbow influence. In [6], Dörfler assumed that this synchronous fluctuation results of both source excitation and hydraulic response of the system. It is the only fluctuation component interacting with the hydraulic system and susceptible to propagate into the whole system, whereas the convective component is a local pressure fluctuation.

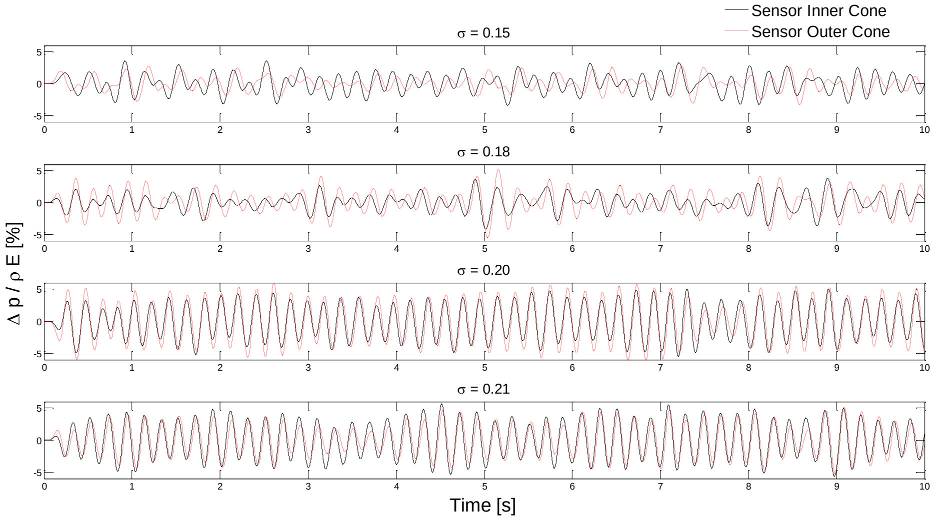

Figure 5. Pressure Fluctuations Factor for different σ values in the draft tube cone, with low-pass filter at 10 Hz.

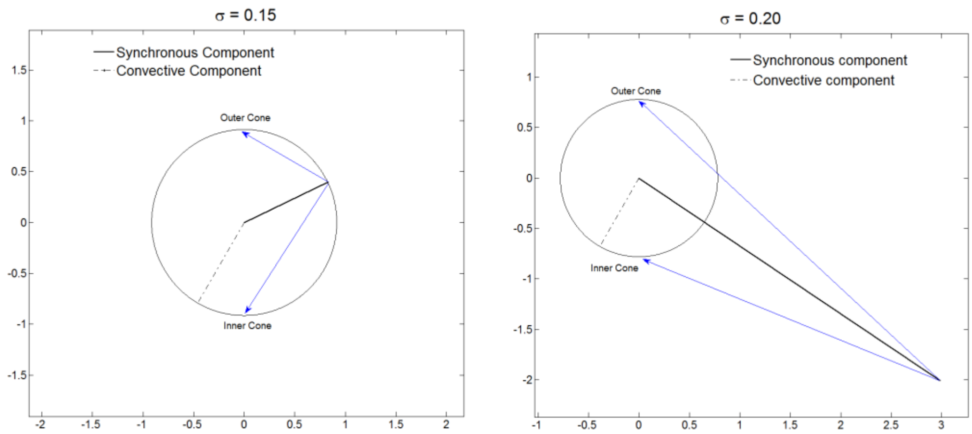

Figure 6. Vectorial decomposition of pressure fluctuations in the cone into synchronous and convective components.

It has been shown by Angelico and Muciaccia (see [10-11]) that the synchronous component becomes preponderant at resonance. This result is confirmed by figure 7. Out of resonance, the convective and synchronous components are nearly equal. Modifying Thoma number, the convective component remains nearly constant, as the vortex features practically the same volume in the straight range of studied σ – values. This observation confirms that the convective component is just a result of the vortex rope precession movement which doesn’t interact with the hydraulic system.

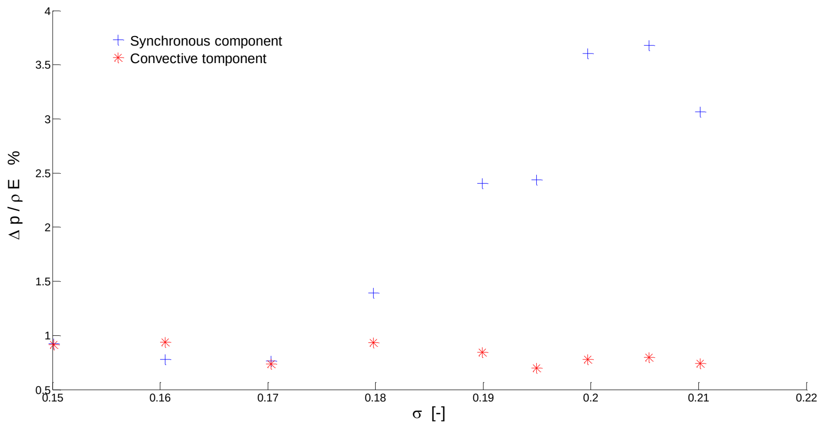

Figure 7. Influence of σ on the amplitude of the synchronous and convective pressure fluctuation in the cone.

On the contrary, the synchronous component becomes more and more important approaching resonance conditions. Its evolution features a peak around $\sigma = 0.20$. This result could be interpreted as follow. The amplitude of excitation sources doesn't increase approaching resonant condition, as the vortex rope features nearly the same behavior and volume for each sigma, but the response of the hydraulic system is maximal at the resonance. As expected, the synchronous component might be identified as a fluctuation resulting not only of the excitation source, but also of the hydraulic system response.

3.2.3. *Study of phase shift through the test rig.* The temporal phase shift at the vortex rope precession frequency between the sensors arranged in the test rig upstream the turbine is computed, and the corresponding wave speed is derived. For this purpose, a special treatment of the signals is necessary. Indeed, the measured pressure fluctuations in the entire test rig don't correspond to the result of a pure progressive or standing wave, but a set of progressive and retrograde waves. The different reflections – transmissions in the whole hydraulic test rig at the boundary conditions lead to the superimposition of standing and progressive waves, see [12]. The rate of standing and progressive waves depends to the boundary conditions. That is possible to decompose the resulting acoustic field into progressive and standing waves without supposition about the boundary conditions, and thanks to pressure fluctuations measurements at only two different locations, see [13]. This special treatment is being realized and results will be available for paper presentation.

3.3. Eigenmode shape determination.
If we assumed that the point $\sigma = 0.20$ corresponds to a resonance, the frequency of 5.05 Hz featuring a peak of amplitude can be identified as one of the eigenfrequencies of the hydraulic system. With this hypothesis, the amplitudes measured at this frequency can be used to reproduce the shape of the corresponding eigenmode, which can be probably identified as the first test rig eigenmode considering hydro-acoustic model results, see [8]. For this purpose, the amplitudes measured at each location are plotted in figure 8 with respect to the location of the sensors on the test rig. For the cone, only the synchronous component is plotted. The eigenmode shape in the turbine and the spiral casing isn't represented since the complex geometry in this part of the test rig doesn't allow having reliable results.

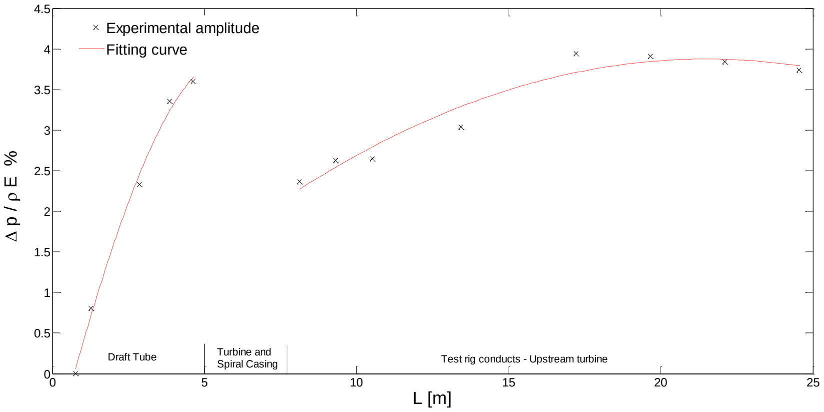

Figure 8. Determination of the first test rig eigenmode shape (f = 5.05 Hz) using hydraulic response at σ = 0.20.

As expected, the eigenmode features a pressure node at the end of the diffuser, where the boundary condition of constant pressure is imposed by the tank. It seems equally to feature a pressure anti-node in the pipe between positions C5 and C8, where maximum amplitudes are measured. The presence of the cavitation volume in the draft tube seems to deform consequently the eigenmode with the largest amplitude in the cone, but these results must be compared with a free-cavitation case.

4. Conclusion.

The hydro-acoustic response of a hydraulic test rig with Francis turbine operating at partial load is measured for one selected operating point with different σ-values. The point featuring the maximum hydro-acoustic response is identified as a resonant point, one of the hydraulic system eigenfrequencies matching the vortex rope precession frequency.

At this point, the synchronous component of the pressure fluctuations measured in the draft tube cone becomes preponderant over the convective component, which remains nearly constant and isn't influenced by the hydro-acoustic response of the system. In the conducts, the temporal phase shift between the different pressure signals indicates the superimposition of travelling and standing waves in the entire test rig, induced by the boundary conditions. Finally, assuming that the identified point is effectively a resonance, the hydro-acoustic response of the test rig at this point is used to reproduce the shape of the excited eigenmode. However, these results must be confirmed by further investigations and compared with results at other operating points, and notably a free-cavitation point.

Nomenclature

E_M	Model specific energy [J.kg^{-1}]	Q_M	Model discharge [m^3.s^{-1}]
Fr	Froude number [-]	T	Torque [N.m^{-1}]
n	Runner rotational frequency [s^{-1}]	σ	Thoma number [-]
n_{ED}	IEC speed factor [-]	ρ	Fluid Density [kg.m^{-3}]
Q_{ED}	IEC discharge factor [-]		

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Acknowledgments

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