Detection and analysis of part load and full load instabilities in a real Francis turbine prototype

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Abstract. Francis turbines operate in many cases out of its best efficiency point, in order to regulate their output power according to the instantaneous energy demand of the grid. Therefore, it is of paramount importance to analyse and determine the unstable operating points for these kind of units. In the framework of the HYPERBOLE project (FP7-ENERGY-2013-1; Project number 608532) a large Francis unit was investigated numerically, experimentally in a reduced scale model and also experimentally and numerically in the real prototype. This paper shows the unstable operating points identified during the experimental tests on the real Francis unit and the analysis of the main characteristics of these instabilities. Finally, it is shown that similar phenomena have been identified on previous research in the LMH (Laboratory for Hydraulic Machines, Lausanne) with the reduced scale model.

1. Introduction

One of the advantages of using large hydraulic turbines for power generation is that they output power can be adjusted depending on the instantaneous energy demand of the grid. In order to regulate the output power, these units have to operate out of its design conditions. Particularly, many Francis units, which are nowadays the most installed hydraulic turbines in power plants, have to work out of its best efficiency point for long periods of time. Nevertheless, some operating points can be unstable, limiting the effective operating range of the unit. For this reason, it is of paramount importance to accurately predict (during the design phase of the unit) and to determine (during the operation of the installed unit) these particular points.

Inside the HYPERBOLE project (FP7-ENERGY-2013-1; Project number 608532) a large Francis unit has been investigated by means of numerical models [1, 2] and experimentally on both the reduced scale model of this unit [3-6] and the real prototype [7]. The real unit has a rated power of 444MW and it is located on the British Columbia in Canada. The tests on the reduced scale model (complete model with scale 1/16) were performed in the Laboratory of Hydraulic Machines (EPFL) in Switzerland.

The unstable points of this unit were firstly predicted based on the numerical and experimental studies made on the reduced scale model. As a final step of the HYPERBOLE project, a measurement campaign on the real unit has been performed. One of the goals of this campaign was to determine and to investigate the unstable operating points of the real prototype of the investigated unit.

In this paper, the experimental data obtained during the prototype tests are analyzed. The main focus of the analysis is firstly to determine and to analyze the main unstable operating points of the unit. Finally, it is observed that similar phenomena was found in the tests made on the reduced scale model.
2. Experimental tests

The on-site tests in the real prototype planned inside the HYPERBOLE project took place in November 2016. The objective was to measure different parameters of the unit such as pressure fluctuations, stresses on the runner, vibrations on the bearings and power fluctuations for many operating conditions. Schematically, the sensors where installed according to Figure 1.

![Figure 1. Overview of the installed acquisition system](image)

As seen in Figure 1, the sensors installed were: strain gauges and pressure sensors on the runner, pressure sensors on the draft tube and spiral case & penstock, accelerometers on the bearings and other stationary parts and strain gauges on the shaft. Furthermore operating signals such as Power or Wicket Gate Opening and Gross Head were also monitored.

All the signals were simultaneously acquired with a B&K LAN XI module (Pulse module). The sampling rate was selected at 4096Hz. All the data were stored in the central computer. Figure 2a shows the active power of the unit and gives an overview of the operating points measured. For the analyses presented in this paper, firstly the slow ramp-up (approximately 1 hour) was analyzed. To have such a characteristic with the signal of Power, the operating gross head was approximately constant and the wicket gates were opened very slowly. The second part analyses the stabilized points, for which the operating parameters of the unit were maintained constant for approximately 5 minutes. This permits to analyze each particular condition with more details.

![Figure 2. Overview of the testing day with the signal of Power](image)
3. Results

3.1. Detection of unstable points during the slow ramp-up

Analysing one of the pressure sensors located on the draft tube (Figure 3) one can identify possible unstable points. In Figure 3, the part load instability corresponding to the resonance condition between the vortex rope precession and the natural frequency of the unit can be identified for $P = 255$ MW. In a similar way the full load instability has been investigated. The main operating characteristics of these points and the coefficients $n_{ED}$, $Q_{ED}$ are summarized in Table 1.

In order to analyse the main characteristics of these two unstable conditions, these are studied with more detail during the steady conditions (see Figure 2).

![Figure 3. Time-Frequency analysis during the slow ramp up of the tests. Unstable point with maximal pressure fluctuation detected at 255MW](image)

| Unstable point | Main frequency | Power | $Q_{ED}$ | $n_{ED}$ |
|----------------|----------------|-------|----------|----------|
| Part Load      | 0.625 Hz       | 255.6 MW | 0.136    | 0.276    |
| Full Load      | 0.8125 Hz      | 471 MW  | 0.2533   | 0.279    |

3.2. Analysis of the unstable points during the steady conditions

For the operating points shown in Table 1, a FFT (Fast Fourier Transform) analysis of the time signals has been performed. The steady conditions (see Figure 2) are now considered. For the part load and full load instability a signal of approximately 180 seconds is taken for the analysis. The analysis is performed with a resolution of 1/16Hz (window of 16 seconds weighted by a Hanning) and by averaging all the signals in the frequency domain.

For these two unstable points (Figure 4a), a large power fluctuations of the unit are observed. These large fluctuations of the output power are not convenient for the grid. The period of oscillation of these time signals for these two unstable conditions corresponds to the period detected with the pressure sensor at the respective unstable points for the slow ramp up (see Table 1).
One of the pressure sensor installed on the spiral case, detects an oscillation in pressure of approximately 0.5 bar (RMS value) for both unstable points (Figure 4b). The same frequencies are clearly detected with an accelerometer installed on the draft tube wall (Figure 4c). Regarding the strain gauge sensor installed on the runner, there is a difference in the autospectrum when comparing the two instabilities (Figure 4d). For the full load instability, the frequency detected with this sensor on the rotating frame is the same as the frequency detected with the stationary sensors. For the part load instability the peak of 1.5 Hz dominates instead of the peak of 0.625 Hz. According to the part load and full load analyses on the reduced scale model, the first unstable point is excited by a vortex rope with a precession motion while for the full load instability, the excitation source is an axial vortex rope. This precession produces differences in the frequency content of the pressure field on the rotating and stationary frame. The main frequency on the rotating frame can be calculated according to 
\[ f_{\text{vortex,rot}} = f_{\text{runner}} - f_{\text{vortex,stat}} \]
where \( f_{\text{vortex,rot}}, f_{\text{vortex,stat}} \) are the vortex rope frequency from respectively the rotating and stationary frame and \( f_{\text{runner}} \) is the rotating speed of the machine. Similar phenomena was found in [9].

For the unstable point 1 (Figure 4), the ratio between the vortex frequency and the rotational frequency of the runner is \( f_{\text{vortex,rot}} / f_{\text{runner}} \approx 0.29 \). Favrel et al. in [4] found approximately the same ratio for the same range of \( \frac{Q}{Q_0} \). According to [4, 5] (tests in the reduced scale model) this unstable point could be explained by a resonance caused by a coincidence of \( f_{\text{vortex,stat}} \) with one natural frequency of the hydraulic system.

For the full load unstable point, the study made by Müller et al. in [8] is taken into account. In that case, \( Q_{\text{ED}} \) was approximately the same as the \( Q_{\text{ED}} \) for the full load instability point identified on the prototype and \( n_{\text{ED}} \) was slightly larger. Same as in the prototype tests, a large pressure fluctuations, due to an axial oscillating vortex rope were detected with a pressure sensor installed on the wall of the draft tube. The similarity in the shape of the pressure signal can be clearly seen in Figure 5.

Figure 4. Frequency analysis of the instabilities. a) Power, b) Pressure, c) Accelerometer and d) Strain gauge on the runner.
4. Conclusions

The data obtained after the prototype tests of the HYPERBOLE project, has been analysed. The analyses were focused on the detection and the study of two unstable operating points found during the tests. For the tests, the operating head of the unit was approximately constant and the wicket gates were moved from its minimum opening to almost its maximum.

These unstable points have been confirmed analysing many types of sensors located on the stationary frame and on the rotating frame (runner). Furthermore, the results obtained in the prototype tests seems to be in good qualitative agreement with the ones from the reduced scale model test.

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