Hydraulic Phenomena Frequency Signature of Francis Turbines Operating in Part Load Conditions

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Abstract. With the integration of renewable energies into the electricity grid, new requirements have been defined by power station operators. These changes bring new challenges for hydraulic turbine manufacturers, such as more flexibility during machines operation. In recent years, investigations have been focused on off-design conditions, since unsteady phenomena occur far from the classical Francis turbine operating range. This is especially the case at partial load, where dynamic stresses on the runner could impact the machine lifetime. The main objective of the study presented in this paper is to gain a better understanding of Francis turbine partial load flows. Thus, different runners are compared, based on test rig measurement campaigns already realized on model scale turbines. Pressure sensors and strain gauges signals are compared, using Fast Fourier Transform (FFT) analysis and Spatial Harmonic Decomposition (SHD). Finally, an attempt to classify these sets of frequencies is realized, according to their dynamical loading on the structure operating in off-design conditions and their impact on the turbine lifetime.

1. Introduction

As modern electricity grids include new renewable energy sources, hydroelectric power station operators need to meet new requirements in terms of flexibility and adaptability. The impact on hydraulic turbines manufacturers is to consider extended operating ranges for their machines. To address this request, off-design hydraulic and mechanical conditions are now better investigated. Francis turbines are known for their high efficiency at their Best Efficiency Point (BEP), and the wide range of heads and flow rates they cover worldwide. For more than a decade, the extension of their operating zone has led to the study of the partial load vortex rope [1]. This hydraulic phenomenon causes pressure fluctuations in the machine, mainly in the cone of the draft tube. Most of the technical specifications for Francis turbines still require low level of pressure fluctuations.

Going even deeper at partial load is a new challenge. In such regimes, Francis turbines encounter even more misadapted flows they were not designed for in the past. By increasing the time of operation at partial load, Francis runners are more often confronted to dynamic stresses on the blades, which may lead to lifetime reduction issues [2]. To develop a better understanding of such unsteady loadings and partial load flows, experimental data from reduced scale Francis turbines (called “model” turbine) have been compared. The objective of this study is to describe characteristics of the phenomena occurring at deep part load and their structural impact on the turbine runner.
2. Experimental Setup
The tests were performed in the GE Hydro Laboratory. Two kinds of instrumentation are considered in the present paper. Unless otherwise indicated, all the models presented thereafter are provided with similar configuration of sensors (Figure 1). The acquisition frequency is 2.4 kHz.

Firstly the study makes use of classical flush-mounted pressure fluctuation sensors on non-rotating parts of the scale model. Sensors in the vaneless space and cross sections of the draft tube cone will be used for comparisons.

Secondly, strain gauges are positioned directly on runner blades. The gauges are positioned at the connections of blades to the crown and the band of the runner. These zones are critical due to the stress concentration occurring in the fillet radii of the junctions when the blades are loaded. These sensors give information of the static and dynamic loading on the blades.

In addition, flush mounted pressure fluctuations sensors on the runner blades are available for one of the designs. Such embedded instrumentation gives access to information on the pressure field inside one rotating interblade channel.

3. Comparison of several runner designs at the rated head
In this first part, the purpose is to find if there are similarities between fluctuations signals recorded while different turbine models are operating at part load. The frequency content is analyzed by using Fast Fourier Transform (FFT).

3.1. Turbine configurations
Several designs are compared. Selected ones are noted here A, B, C, D and E, from low to high-head turbines. Two types of manufacturing process of the model runner are commonly used for such investigation, achieving either only hydraulic similitude with the industrial turbine, or both hydraulic and mechanical similitude.

3.2. Operating points
Dimensionless parameters are used to describe hydraulic turbines hill chart [3]:

\[ \psi = \frac{2gH_n}{U_2^2} \quad \text{and} \quad \varphi = \frac{Q}{U_2S_2} \]

with \( H_n \) the net head (m), \( Q \) the discharge (l/s), \( g \) the gravitational acceleration, and \( U_2 \) the peripheral velocity (m/s) of the runner outlet, which section is \( S_2 \) (m²).

For each design, the load variation at the rated head and at sigma plant value is performed (Table 1). At each steady operating point, a FFT analysis of the dynamic signals is performed. For the sets of measures, waterfall diagrams [3] are used, showing the evolution of one sensor spectrum amplitude and frequency along with the discharge coefficient. For each design, the frequency, the discharge coefficient and the amplitude are made dimensionless by dividing respectively by the rotation frequency of the runner \( f_0 \), the discharge coefficient at the best efficiency point \( \varphi_{opt} \) and the maximum spectrum amplitude obtained for the whole load variation. In the next section, waterfall diagrams from different sensors signals are compared, to highlight several frequency signatures of phenomena occurring at deep part load.
Table 1. Load variation parameters for the selected designs.

| Design | Type of turbine | \(Z_r\) \(^a\) | \(Z_g\) \(^a\) | \(\psi\) | \(\sigma_{\text{plant}}\) | Type of sensors |
|--------|----------------|----------------|----------------|--------|----------------|----------------|
| A      | Low Head       | 13             | 24             | 0.91   | 0.40           | Pressure (fixed), strain gauges |
| B      | Medium Head    | 15             | 24             | 1.62   | 0.17           | Pressure (fixed and embedded), strain gauges |
| C      | Medium Head    | 15             | 24             | 1.68   | 0.18           | Pressure (fixed), strain gauges |
| D      | Medium Head    | 15+15          | 24             | 2.57   | 0.11           | Pressure (fixed), strain gauges |
| E      | High Head      | 13             | 20             | 3.70   | 0.12           | Pressure (fixed), strain gauges |

\(^a\) \(Z_r\) and \(Z_g\) are the numbers of runner blades and guide vanes respectively.

3.3. Description of phenomena occurring at partial load

3.3.1. Large-band frequency signature. Figure 2 shows the waterfall diagram of the gauge with the highest levels of dynamic strain for each runner. Since the blades are thinner near the trailing edge junctions, all these strain gauges are positioned between 6\% and 12\% from the trailing edge in the chordwise direction. When reducing the discharge, large-band frequency signatures can be seen on designs A, B and C strain signals. The shape of this signature was generally correlated with the presence of interblade vortices (IBV). It consists of an increase of the spectrum amplitude on a whole range of frequencies. For design A, the signature is around 0.4\(\varphi_{\text{opt}}\), while for the designs B and C, it is around 0.3\(\varphi_{\text{opt}}\). For the runners D and E, the region of higher amplitude extends over 30\(f_0\), it is therefore not clear if the same phenomenon is occurring for these designs.

![Figure 2](image)

**Figure 2.** Large-band frequencies on dynamic strain for the selected runners.
3.3.2. “Claw-like” symmetric frequency signature. On pressure sensors located in the vaneless gap between the guide vanes and the runner, pressure fluctuations show a “claw-like” frequency signature on design B and C (Figure 3). Below $0.35\phi_{opt}$, two sets of several amplitude peaks are developing in opposite frequencies when the flow rate is increasing. Frequency differences between the peaks of each group seem to remain constant, and peaks of opposite frequency trend with the discharge can be identified. Both sets seem symmetrical compared to a vertical line corresponding to half of the blade passing frequency. All these elements suggest that a coherent phenomenon is causing this claw-like signature.

The design A shows some high amplitudes too, however a lack of discretization in the load variation records prevents from any conclusion on the frequency signature. For the higher head designs D and E, no signature of this type was found.

![Figure 3. “Claw-like” frequency signature in the vaneless space pressure signal.](image)

3.3.3. Deep part load low frequency signature. At even lower loads, low frequency dynamic loadings on the runner blades and pressure fluctuations on the draft tube cone walls were observed.

Firstly, on the strains gauges (Figure 4 on the bottom) two different trends are observed for the higher peak appearing at deep part load. The first one illustrated with the design B consists of a decreasing frequency peak around $f_0$ from $0.15\phi_{opt}$ to $0.3\phi_{opt}$. The second trend concerns the design E for which the peak frequency stays rather constant and around $0.2f_0$ on the same discharge range than mentioned before. Another interesting peak at $0.8f_0$ will be worth mentioning hereinafter.

Concerning draft tube cone pressure fluctuations (Figure 4 on the top), both designs are different. For design B, deep part load spectrums contain only small amplitudes peaks at lower frequencies, unlike what was on strain gauges signals. An upper part-load resonance can be found around $0.8\phi_{opt}$, but has only a small impact on the runner (see the bottom-left diagram of Figure 4). For design E, a constant frequency peak at about $0.2f_0$ is observed from roughly $0.45\phi_{opt}$ to $0.15\phi_{opt}$.
For design E, it was possible to use the Spatial Harmonic Decomposition [4] on four sensors located in the same horizontal cross section of the draft tube cone. A frequency analysis of the $P_0$ and $P_{-1}$ terms, representing respectively in phase pressure pulsation and a pattern with one-nodal diameter rotating in the runner direction, showed that the fluctuations peak at $0.2f_0$ in the draft tube was decomposed by the runner strain gauge in two parts, just like it was done for the vortex rope [4]. The $0.8f_0$ peak on the embedded strain gauge is linked to the rotating part of this phenomenon, while the $0.2f_0$ peak is present on strain signals, without changes in frequency from the stationary to the rotating frame instrumentation. Contrary to the vortex rope for which the rotating component is usually of major amplitude, the in-phase amplitude of the fluctuations ($P_0$) is higher than the rotating one ($P_{-1}$) for this $0.2f_0$ pulsation.

4. Frequency insight on design B test rig results
When trying to extend Francis turbines operating range at deep partial load below the part load helical vortex rope, the first phenomenon that the runner will experience, is the one associated with the large band frequency observed previously. This signature has been studied on the design B, since this reduced scale runner is equipped with on-board pressure sensors and strain gauges.

4.1. Signal frequency content analysis methodology
Classical representations used to describe fluctuations at partial load are the peak-to-peak values or the root mean square of the signals, as a function of the discharge. For finite energy signals, the signal power can be derived from the root mean square value (assuming an ergodic process):
\[ P_x = \frac{1}{T} \int_0^T |x(t)|^2 dt = \sum_{i=1}^N |x(t_i)|^2 \frac{dt}{T} = \frac{1}{N} \sum_{i=1}^N |x(t_i)|^2 = x_{rms}^2 \]

with \( x(t) \) the signal, \( T \) the acquisition time, \( dt \) the sampling period, \( N \) the number of samples, \( P_x \) the power of the signal and \( x_{rms} \) the root mean square value of the signal.

With the Parseval theorem, the signal can be represented in an equivalent way in the time or the frequency domains. Then, it is possible to decompose the signal power in several ranges of frequencies:

\[ P_x = \int_0^T |x(t)|^2 dt = df \int_0^{f_s} 2|X(f)|^2 df \]

\[ P_x = \int_0^{f_1} [\sqrt{2}|X(f)|df]^2 + \int_{f_1}^{f_2} [\sqrt{2}|X(f)|df]^2 + \cdots + \int_{f_{n-1}}^{f_n} [\sqrt{2}|X(f)|df]^2 \]

with \( X(f) \) the Fourier Transform of \( x(t) \), \( f_s \) the sampling frequency, \( df \) the frequency resolution, \( n \) the number of frequency ranges used in this frequency decomposition (\( n \) is equal to 5 for the rest of the study). The ranges are chosen according to the waterfall diagram to enclose observed signatures and isolate their contribution on the whole signal power.

The design B was equipped with embedded pressure sensors, located on one blade pressure and suction sides (PP and PS on Figure 5) in addition to strain gauges (SG). Strain and pressure fluctuations signals are compared in different operating conditions. They are made dimensionless by dividing by the net head.

**Figure 5.** Example of embedded sensors on the runner B.

### 4.2. Frequency content of the dynamic signals for the design B

To see the evolution of the signal frequency content during load variation, the signal spectral power (SSP) is plotted as a function of the relative discharge coefficient (thick black line) and placed next to the corresponding waterfall diagram (Figure 6). In addition, five grey lines show the spectral power evolution of each frequency range during the load variation. This allows to see at the same time the evolution of the total power and the frequency content of the signal.

This analysis is derived for the gauge with the highest levels of dynamic strain of the design B (Figure 6), which is located at the junction of the trailing edge with the band. When decreasing the discharge, a first local maximum of dynamic loadings is reach at \( 0.65 \phi_{opt} \) due to the presence of the part load vortex rope. It is worth mentioning that the majority of the spectral power is contained in the lower range of frequencies \([0 \cdots 1]f_0\), which includes the rope frequency. Then the loadings reach their absolute maximum around \( 0.31\phi_{opt} \), which corresponds to higher level of the large-band frequency signature identified earlier (Figure 2). For this operating point, the major part (roughly 40%) of the signal spectral power is coming from the range of frequencies \([7 \cdots 20]f_0\) which encloses the large-band frequency signature. On the right of the Figure 6, the runner outlet flow can be visualized, and developed cavitation interblade vortices can be seen near the band for this point of maximum strain of the blades.

The same analysis is applied on the embedded pressure sensors of the runner B (Figure 7). For the inlet sensor, a large-band signature can also be seen at \( 0.31\phi_{opt} \). It corresponds to a local maximum of pressure fluctuations but, unlike the dynamic strain, the spectral power is lower than the one of the
vortex rope regime. Even if there is a small increase of power caused by the range \([7 - 20]f_0\), the lower range \([0 - 1]f_0\) is dominant in this case. For the outlet sensor, no large-band signature was found; although interblade vortices are clearly visible in the runner channels near the sensor location (see Figure 6). Both pressure signals show clearly an upper part load resonance around \(0.8\varphi_{opt}\) but once again no impact is seen on the strain gauge. The great majority of the spectral power comes from the range \([1 - 4]f_0\), which encloses this phenomenon frequency.

![View of the runner outlet at maximum dynamic loadings (0.31\(\varphi_{opt}\))](image)

**Figure 6.** Design B strain gauge signal analysis for a load variation (rated head, sigma plant).

![Figure 7. Design B pressure sensors signal analysis for a load variation (rated head, sigma plant).](image)

5. Impact on turbine lifetime

The aim of this last section is to estimate the structural impact of dynamic loadings on the runner in deep partial load operating conditions. The dynamic stresses due to the vortex rope will be taken as a reference for the damage at partial load.

5.1. Load controlled fatigue testing

Fatigue analysis is based on material characterization. Load controlled fatigue testing are performed on different materials to derive their Wöhler curve (or S-N curve), representing the stress amplitude as a function of the number of cycles to failure. As the stress increases, the allowed number of cycles decreases:

\[
N_f = C \Delta \sigma^m
\]

with \(N_f\) the number of cycles for a given probability of survival (usually 95 or 99.9 %), \(\Delta \sigma\) the dynamic stress amplitude, and \(C\) and \(m\) are the tested material constants.
To describe the prototype material behavior, the Wöhler curve that is used is appropriate for welded structure in cast steel. Load controlled fatigue tests were conducted in water, under in-plane bending conditions with non-zero mean stress value.

5.2. The Rainflow counting method
To describe the blade loading temporal evolution, the commonly used Rainflow method is applied [5]. With this method, the signal is decomposed into several amplitude ranges and a number of cycles are determined for each range of stress amplitude. Then, the Miner’s assumption of linear cumulative damage is used: for each previously mentioned amplitude value, an elementary damage is defined as the ratio of the number of cycle encountered during the signal record and the number of cycles at failure. The damage is finally calculated by adding the elementary ones. This linear rule of cumulative damage has some limitations to reproduce such a complex physics, however it is a reasonable engineering tool used in industrial dimensioning processes.

\[
D = \sum_i d_i = \sum_i \frac{N_i}{N_{fi}} = \sum_i \frac{N_i}{c.\Delta \sigma_i}; \text{ fatigue damage}
\]

Generally, the value of 1 is chosen as a limit for the damage during dimensioning steps, which corresponds to a chosen probability of failure. For the full scale turbine, the rate of damage per unit of time can be derived by taking into account the ratio of runner rotation frequencies between the model and the prototype.

5.3. Comparison of fatigue damage at part and deep part load

5.3.1. Methodology and hypotheses. The first assumption is that scale model stress is scaled up to the prototype, by taking into account the net head ratio:

\[
\sigma_{11_{\text{prototype}}} = \frac{H_{\text{prototype}}}{H_{\text{model}}} \cdot \sigma_{11_{\text{model}}}
\]

The strain measurement is made on an isotropic material in plane stress conditions, so that:

\[
\sigma_{11} = \frac{E}{1 - \nu^2} (\varepsilon_{11} + \nu \varepsilon_{22})
\]

with \(E\) and \(\nu\) are the Young’s modulus and the Poisson ratio of the material, \(\sigma_{11}\) the normal stress in the first principal direction, \(\varepsilon_{11}\) and \(\varepsilon_{22}\) the strain in the first two principal directions.

The strain gauge gives the strain in one direction only. By neglecting the strain in the other one:

\[
\sigma_{11_{\text{prototype}}} \approx \frac{H_{\text{prototype}}}{H_{\text{model}}} \left( \frac{E_{\text{model}}}{1 - \nu^2_{\text{model}}} \cdot \varepsilon_{11_{\text{model}}} \right)
\]

With this methodology, the damage is calculated at the strain gauge location. The endurance limit was neglected to preserve information on the smaller amplitudes.

5.3.2. Comparing the vortex rope and deep part load phenomena. For the rest of the analysis, the most sensitive strain gauge of the design B will be considered. Four operating points are analyzed (Table 2): the best efficiency point (BEP), the part load maximum dynamic loadings in the vortex rope regime (PL), which is taken as a reference for the damage value; the deep part load occurrence of the large-band frequency signature (DPL), and the low variable frequency at quasi-no load (QNL). For these four operating points, the previously exposed fatigue damage method is applied. The results are listed in Table 2.

According to these results, the rate of damage increases during partial load operation. The vortex rope phenomenon has a real impact on the runner lifetime, since the rate of damage is more than 100 times higher than at the best efficiency point where the damage is practically negligible. At deep part load, when the dynamic loadings are at their maximum, the rate of fatigue damage gains roughly two orders of magnitude, while the stress peak-to-peak value is doubled. At lowest flow rate, the rate of damage is still high compared to the vortex rope, but still lower than for the large-band signature.
which occurs at higher frequency. As an order of magnitude, one hour of operation at $0.31\varphi_{opt}$ is equivalent to about ninety hours at $0.65\varphi_{opt}$.

The reason for higher rates of damage at deep part load can be explained by examining the Figure 8. When deriving the Rainflow counting method, the recomposed prototype stress signal is divided in several ranges of peak-to-peak amplitudes (on the horizontal axis). For each range, the number of counted cycles can be read on the vertical axis. The four colors are corresponding to the four operating points previously introduced (Table 2). For smaller stress amplitudes on the left of the green dashed line, the difference in damage between the vortex rope and the deeper partial load regimes is mostly linked to the number of cycles, which is multiplied by roughly 100, due to higher frequencies in the signal. However, the main changes are coming from the higher amplitudes that are not occurring when operating in the vortex rope regime (on the right of the dashed green line). The severe damage at deep partial load is mostly due to these increased amplitudes of the dynamic loadings on the runner blades.

| Operating point | $\psi$ (-) | $\varphi/\varphi_{opt}$ (-) | Relative Rate of Damage |
|-----------------|------------|-----------------------------|------------------------|
| $B_0$-BEP       | 1.62       | 1.00                        | $<1E-02$               |
| $B_0$-PL        | 1.62       | 0.65                        | $1.0E+00$              |
| $B_0$-DPL       | 1.62       | 0.31                        | $8.9E+01$              |
| $B_0$-QNL       | 1.62       | 0.22                        | $6.2E+01$              |

6. Conclusion
To address Francis turbine partial load issues, a better understanding of the flow inside the machine for this type of operating points is required. Several turbine designs were investigated by comparing signals acquired during test rig measurements campaigns. At partial load, it was possible to find similarities between pressure fluctuations in the flow and dynamic loadings on the runner blades of the different designs. The most complete instrumented runner was studied in detail by building specific tools to characterize the frequency content of partial load signals recorded in both stationary and rotating frames. For this design, the dynamic loadings on the blades reach a maximum when operating at deep partial load. At this operating point, the loadings are correlated with the occurrence of a large-band frequency signature in strain gauges signals, which contains the major part of the signal spectral power. A similar signature was found in signals of a pressure sensor located at the runner inlet. Investigations on the phenomenon causing such signature confirmed the presence of developed interblade vortices at the point of maximum dynamic stresses. Finally, a fatigue analysis based on the Rainflow counting method and Miner’s cumulative damage theory was realized. The rate of damage for the point of maximum large-band loadings is roughly two orders of magnitude higher than for the point of maximum vortex rope stresses. The same result was found for deeper part load lower frequency phenomena, with a rate of damage slightly reduced compared to the large-band frequency one. The phenomenon involving this large-band frequency signature is a major blocking factor that must be overcome in order to develop deep partial load operation and further investigations are recommended.
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References
[1] Avellan F., 2000, Flow Investigation in a Francis draft tube: The Flindt Project, Proc. of the 20th IAHR Symp., Hydraulic Machinery and Systems, Charlotte, USA.

[2] Lowys P. Y., Paquet F., Couston M., Farhat M., Natal S., Avellan F., 2002, Onboard Measurements of Pressure and Strain Fluctuations in a Model of Low Head Francis Turbine – Part 2: Measurements and Preliminary Analysis Results, Proc. of the 21st IAHR Symp., Hydraulic Machinery and Systems, Lausanne, Switzerland, Volume II, pp 873-880.

[3] Int. Electrotechnical Comission, 1999, Hydraulic turbines, storage pumps and pump-turbines – Model acceptance tests (Int. Standard IEC 60193), section 4.3: Pressure Fluctuations.

[4] Duparchy A., Guillozet J., De Colombel T., Bornard L., 2014, Spatial Harmonic Decomposition as a tool for unsteady phenomena analysis, Proc. of the 27th IAHR Symp., Hydraulic Machinery and Systems, Montreal, Canada.

[5] ASTM International, 2011, ASTM E 1049-85 Standard practices for Cycle Counting in Fatigue Analysis, West Conshohocken, PA, USA.