Mechanical risks prediction on Francis runner by Spatial Harmonic Decomposition

F Bouloc, J Guillozet, F Duparchy, P.Y Lowys, A Duparchy
GE Renewable Energy, 82 Avenue Léon Blum, 38041 Grenoble Cedex 9, France
francois.bouloc@ge.com

Abstract. Extending the operating zone of Francis turbines toward low load is a major stake to allow the optimization of the electrical grid. The dynamic phenomena encountered at low load are potential sources of pressure fluctuations, power instability and runner fatigue. Traditionally, the peak to peak value of pressure fluctuations is used to assess these risks. However, this estimator is not sufficient to analyse separately the various dynamic phenomena and their impact on the stability of the turbine. In this paper the recent Spatial Harmonic Decomposition (SHD) method is used to analyse the pressure fluctuations through more relevant indicators. The evolution of these indicators along a load variation is compared with the associated runner strain measured with on-board gauges. It is shown that the use of the Spatial Harmonic Decomposition is a powerful tool to evaluate the risks for the industrial turbine and thus improve its behaviour and its reliability.

1. Introduction

Today, the stability and the lifetime of a turbine are mainly judged on reduced scale model by analysing peak-to-peak value of pressure pulsations measurements. The sensors are commonly located at the spiral case inlet, in the vaneless area and in the draft tube. For a given net head, these peak-to-peak values are then expressed as a percentage of the net head against the power output and supposed to transpose as such to the full scale machine. Such a rough peak-to-peak analysis suffers from at least one major drawback: it does not tell whether the pulsations will actually decrease the lifetime of components, as it encompasses various phenomena at different frequencies although the number of cycles is determinant for fatigue.

On the contrary, on-board strain measurement on reduced-scale design in mechanical homology with the full-scale machine enables to properly assess risk on runner lifetime in part load operation. GE Renewable Energy conducts these mechanical measurements in order to extend safely the operating zone, and also to reinforce its knowledge concerning the phenomenon that can impact prototype life time. But to correlate runner stresses with hydraulic phenomenon also requires a fine description of the implied phenomena. In that purpose, spatial harmonic decomposition (SHD) is used to have a better understanding of the structure of the flow and of the impact of each individual pattern on the stresses.

2. Experimental Set-up

The reduced scale model is a medium head Francis turbine with $Z_r = 15$ blades and a distributor with $Z_g = 24$ guide vanes. It was designed to achieve mechanical as well as hydraulic homology with the full-scale machine. It was tested in a closed loop test rig of the GE Renewable Energy Laboratory in
Grenoble. In order to evaluate the strain fluctuations in the runner, the reduced scale model was equipped with 8 strain gauges located on 2 blades. Alongside the on-board strain measurement, pressure fluctuation, and radial thrust fluctuation were recorded. Trailing Edge is the location of highest stresses on a runner blade; consequently the crown suction side trailing edge gauge (J1S) is used to analyze the result in this article. Its location is schemed on Figure 1. Pressure sensors are located as recommended by international norm IEC 60193 0, at the spiral case inlet, in the vaneless area between the wicket gate and the runner, and in the draft tube cone. As detailed in [1], various checking tests must be conducted during the model tests to ensure the good behavior of the sensors and distinguish hydraulic phenomenon from interaction with the test rig.

![Figure 1. Position of the strain gauge](image)

3. Strain measurement in the runner

Dynamic strains on the runner blades were measured along a load variation at constant net head and constant rotation speed $f_0$ covering the range of $0.4 < q < 1.2$ where $q = \phi/\phi_{opt}$. The results are given on Figure 2 in terms of Root Mean Square values of the dynamic signal (RMS). Results are non-dimensionalized by the maximum strain of the load variation. The maximum of the strain occurs at deep part load around $q = 0.4$. From this load, strain fluctuations are almost constant up to $0.6Q_{opt}$, and then decrease till $q = 0.8$. For $0.8 < q < 1.2$, strains at the outlet remain low.

![Figure 2. Evolution of strain fluctuations in the runner with load](image)

Three operating points will be further investigated throughout this article:
1. OP1: $q = 0.43$: Deep part load with presence of inter-blade vortices
2. OP2: $q = 0.65$: Part load, beginning of the rope zone
3. OP3: $q = 0.78$: Part load, end of the rope zone, with presence of an Upper Part Load Resonance – UPLR (identified on the pressure sensors).

Pictures of the flow under the runner for these 3 points are given on Figure 3.
a. OP1  
b. OP2  
c. OP3

**Figure 3.** Cavitating structures in the draft tube under the runner

The spectral analysis helps to identify the phenomena involved but it is necessary to understand how frequencies change between the stationary frame and the runner rotating frame. Indeed, a frequency is relative to the motion of the frame in which it is considered, thus a same phenomenon will be seen at different frequencies in a static frame or in a moving frame. For a purely axially pulsating flow, as both the draft tube and the runner are static in the axial direction, frequencies will be equal in both frames. On the contrary a rotating phenomenon will rotate at a different speed relatively to the draft tube (static frame) or relatively to the runner (rotating frame). Let’s consider a runner with a rotation frequency \( f_0 \) and a vortex with a rotation frequency \( f_v \), with \( f_0 > f_v \). At \( t=0 \) the runner rotating frame (\( Oxy \)) and the vortex (\( V \)) are aligned as described on Figure 4a. Their respective angular position is denoted as \( \alpha_0 \) and \( \alpha_v \). Frame (\( Oxy \)) and (\( V \)) will be aligned again when \( \alpha_0 = \alpha_v + 2\pi \) (Figure 4b). This position is reached for time:

\[
t = \frac{1}{f_0 - f_v}
\]

Hence the corresponding vortex frequency relative to the rotating frame is

\[
f_v^r = \frac{1}{t} = \frac{1}{f_0 - f_v}.
\]

This result can be extended for the case of a spatially cyclic phenomenon with \( k \) nodal diameters patterns. In this case the runner frame will be aligned with the \( k \)th node of the pattern when \( \alpha_0 = \alpha_v + 2\pi/k \), resulting in:

\[
f_v^r = k(f_0 - f_v)
\]

**Figure 4.** Vortex rotating in the runner rotating frame (left: one nodal diameter pattern ; right: two nodal diameters pattern)

The spectrum of the strain gauge is shown on Figure 5. On Figure 6 is also shown the cumulative distribution of signal power over the total power against the frequency. This latter representation helps to assess the contribution of each phenomenon to the overall fluctuations.

At OP2, the vortex rope signature is dominant with two harmonics \( f_{VR,1} = 0.78f_0 \) and \( f_{VR,2} = 1.56f_0 \). As explained above, this corresponds to a rope frequency of \( f_{VR} = 0.22f_0 \) in the static frame. The presence of this second harmonic may be due to the structure of the rope. As identified in [3] by spatial harmonic decomposition, the part load rope is usually composed of two pattern, the first having 1 nodal diameter, the 2nd having 2 nodal diameters. The two nodal diameters pattern will be seen with a frequency twice higher than the one nodal diameter pattern frequency. Another phenomenon is
characterised by the range \([15f_0-30f_0]\) where 35% of the signal power is to be found on that band (Figure 6). The influence of the guide vane passage at \(Z_0f_0 = 24f_0\) is hardly visible.

At OP3, the signature of the vortex rope is still present. Almost 50% the energy is owned by this phenomenon. At this load, the gap between the trailing edges of the guide vanes and the blades leading edge is small enough to create strain on the runner at \(24f_0\). The passage of the guide vanes represents 20% of the total energy.

At OP1, the hydraulic structures seem to be less organized than for OP3 and OP2. Strain gauge exhibit a peak close to the runner rotation frequency but the greatest part (70% to 75%) of the energy is spread on the band 5f0-20f0. This may be associated with Inter-blade Channel Vortices (IBCV) that are occurring at this load. It can be finally noticed that there is no significant influence of the guide vane passage.

On board strain measurements provide levels and frequencies of the dynamic phenomenon involved. This is crucial information to evaluate the actual structural behaviour of the runner and the impact for the fatigue. It thus allows to assess properly the possibility to extend the operating zone toward part load or high load without compromising the life time of the runner. This kind of extension of operating range was successfully applied on an industrial turbine as described in [4]. As a complementary approach, performing such mechanical tests on the reduced scale model can allow to optimize the definition of the continuous and temporary operating zone at the early stage of the project.

4. Pressure fluctuations – Standard Analysis
The classical analysis of pressure fluctuation consists in plotting the evolution of the peak-to-peak amplitude of the time signal of pressure fluctuations against the load or discharge. The evolution of this estimator is presented on Figure 7. We notice that the maximum of pressure fluctuation is located at \(q = 0.59\). It does not appear at the same position as the strain fluctuation maximum. OP3 is a local maximum of pressure pulsation although this point is close to be a strain fluctuations minimum in the runner. OP3 is a good example of poor correlation between pressure pulsation peak-to-peak estimator and strain fluctuations in the runner. Pressure fluctuations remain low from \(q = 0.87\) up to \(q = 1.20\).
Spectra and cumulative distribution of signal power are presented on Figure 8 and Figure 9 for the three operating points as before and for three sensors (spiral case inlet, vaneless area and draft tube cone). At point OP1, pressure fluctuations are high in the cone. The phenomenon responsible for this high amplitude is not well identified on the pressure fluctuation spectrum (Figure 8). The spectrum shows a peak at $0.7f_0$, on top of a narrow band of frequencies close to runner frequency. It is mainly visible in the cone, but also present in the gap and in the inlet. Another peak at $0.15f_0$ is visible from the inlet to the cone and is decreasing along the turbine. It lets suggest a pulsating flow coming from the upstream of the turbine. For the vaneless area, the cumulative distribution of signal power shows that the greatest part of the power is owned by a wide range of frequencies going from $1f_0$ to $16f_0$. The blades passage at $Z_r f_0$ accounts for 15% of the signal power at this operating point, for which the $Z_s f_0$ component was not visible on the strain gauges at OP1. This is another example of poor correlation between pressure pulsation peak-to-peak estimator and strain fluctuations.

At OP2, the fluctuations are mainly due to the partial load vortex rope, characterised by a peak in the spectrum at $f_{VR,S} = 0.2f_0$. The rope is responsible for 70% of the signal power in the cone.

At OP3 there is a sharp increase of pressure fluctuation. This pressure surge is present everywhere from the spiral case inlet to the draft tube. The spectrum show a high energetic component at $f_{UPLR} = 3.54f_0$, with a small modulations at $f_{UPLR} + f_{VR,S}$. This spectrum is the typical signature of the UPLR.
From this analysis, it is difficult to know whether or not the surge in pressure pulsation will impact the stability of the prototype, or the lifetime of the runner, although these two concerns may require different mitigation devices. The peak-to-peak estimator seems consequently not sufficient to estimate runner fatigue or prototype stability and to adapt properly the operating zone.

5. Spectral Harmonic Decomposition Analysis

5.1. SHD Method
Considering the limitation of the peak-to-peak approach, Spatial Harmonic Decomposition (SHD) was recently developed as a new analysis of instabilities in hydraulic turbines. It provides an expansion of the pressure pulsations into a sum of modes with increasing number of nodal diameters as pictured in Figure 10. The details of the mathematical development for the SHD may be found in [3] and another application to Francis turbine is available in [5]. The results of a SHD are:

- \( \hat{P}_0(t) \), a synchronous pattern, pulsating in the axial direction.
- \( \hat{P}_k(t) \) \& \( \hat{P}_{-k}(t) \) patterns with k nodal diameters

Their respective spectrum indicates at what frequencies the shapes are moving. The corresponding rotating velocity is equal to \( -\omega/k \) if \( \omega \) is the pulsation of the chosen frequency component.

In the rest of the article, \( P_0 \), will indicate the signal power contained in the in-phase \( \hat{P}_0(t) \) while \( P_k \) indicates the energy contained in both \( \hat{P}_k(t) \) and \( \hat{P}_{-k}(t) \). \( P_0 \) will be called the “in-phase contribution” of the pressure pulsations, whereas the other terms are sometimes grouped under the term “asynchronous” of the pressure pulsations.

![Figure 10](image)

**Figure 10.** Illustration of the two most common \( P_k \) patterns to be found in a Francis turbine (from 0)

5.2. Application to cone pressure pulsation

The SHD was applied to the cone of the draft tube where four sensors were equally spaced. Due to Shannon-Nyquist theorem, this instrumentation will permit to compute \( \hat{P}_0 \), \( \hat{P}_1 \), \( \hat{P}_{-1} \), \( \hat{P}_2 \) and \( \hat{P}_{-2} \). The SHD was applied to the load variation previously analyzed. The evolution on the load variation of their signal power \( P_0 \), \( P_1 \) and \( P_2 \) is plotted on Figure 11, the spectra are presented on Figure 13.

At deep part load (OP1) below \( q = 0.6 \), \( P_0 \), \( P_1 \) and \( P_2 \) are of the same order of magnitude, and are rather constant. For each mode, the energy is spread on a band of frequencies from 0 to \( 2f_0 \), but the maximum are not located at the same frequency.

From \( q = 0.6 \) which corresponds to the beginning of the part load rope zone, \( P_1 \) is rising sharply and shows a maximum at OP2, like the strain fluctuations in the runner. This contribution is the vortex rope: as it can be seen on Figure 13, there is an important peak at \( 0.22f_0 \) for \( \hat{P}_{-1} \), which correspond to an asynchronous pattern at \( 0.22f_0 \) in the runner direction.

In all the vortex rope zone from \( q = 0.6 \) to \( q = 0.9 \), \( P_1 \) is still the main contribution to pressure fluctuations (Figure 11). At OP3, \( P_0 \) is increasing sharply, while the \( P_1 \) contribution presents a small increase. At this particular point, \( P_0 \) becomes the main contribution of the pressure pulsations. This indicates the plane wave nature expected from a hydro-acoustic resonance like the UPLR. The small increase of \( P_1 \) is due to the non-linear combination of the UPLR with the asynchronous pattern of the
vortex rope. On Figure 13, it can be seen that \( P_0 \) holds \( f_{UPLR} \), while \( P_1 \) holds the component at \( f_{UPLR} + f_{VR5} \).

Figure 12 presents the comparison of in-phase part and asynchronous parts with strain fluctuations measured on the outlet crown edge suction side. The comparison of the asynchronous parts and the strain fluctuations in the runner shows a good consistency. It is interesting to note that the asynchronous part brought by UPLR (modulation at \( f_{UPLR} + f_{VR5} \)) do not have any impact on the strain fluctuations. In any case, the in-phase part \( P_0 \) is poorly correlated with the strain fluctuations. Duparchy et al. in [3] also exhibit a resonance similar to the UPLR that has no impact on the runner strain fluctuations. Therefore, it can be stated that the asynchronous phenomena are responsible for strain fluctuations in the runner, while purely pulsating phenomena like the UPLR have no impact on the runner deformations.

![Figure 11: Variation of the SHD RMS contribution with the load](image1)

![Figure 12: Comparison of strain fluctuations with the rotating part of the pressure pulsation](image2)

Figure 11: Variation of the SHD RMS contribution with the load

Figure 12: Comparison of strain fluctuations with the rotating part of the pressure pulsation

![Figure 13: SHD modes spectra (0 – 16f0)](image3)

Figure 13: SHD modes spectra (0 – 16f0)

Axial thrust and radial thrust are also impacted by hydraulic instabilities; it is thus interesting to compare them with the SHD contributions.

On Figure 14, \( P_1 \) is compared with radial thrust. Superposition of both spectra is displayed on Figure 15. Correlation of radial thrust and rotating patterns shows a strong consistency at the beginning of the vortex rope zone. In particular at OP2, frequencies of fluctuations of radial thrust and \( P_1 \) are identical. This consistency decreases along the rope zone, even if the tendency is still good. At OP3, the vortex rope is still the dominant phenomenon responsible for radial thrust fluctuations as it can be seen on the spectrum. It is interesting to note that the radial thrust is not sensitive to 2 nodal
diameters phenomena. Indeed, the patterns create a symmetric pressure field with both X and Y axis, and is hence not supposed to create radial thrust. Both of them also show a slight impact of the UPLR, which is visible on the spectrum, due once again to the modulation of the UPLR at $f_{UPLR} + f_{VRS}$. Between 0.7$f_0$ and 3.5$f_0$, there are some phenomena influencing radial thrust that are not due to asynchronous patterns of the pressure distribution. At deep part load (OP1), $P_1$ does not represent correctly anymore the evolution of radial thrust: spectra are there uncorrelated.

Figure 14: Comparaison of radial Thrust fluctuation with SHD asynchronous contribution

Figure 15. Spectra of radial thrust fluctuations and rotating patterns (0 – 6f0)  

Figure 16. Cumulative distribution of signal power for radial thrust fluctuations and rotating patterns (0-6f0)

Figure 17 presents the comparison of axial thrust pulsation with the in phase contribution. Correlation is rather good in the range [0.65-0.8] corresponding to a vortex rope zone, where both of them are similarly impacted by the UPLR. At OP3, spectrum of the axial thrust fluctuations presented in Figure 18 shows that not only $f_{UPLR}$ is present but also the modulation of the UPLR at $f_{UPLR} + f_{VRS}$. This modulation is not carried by the in-phase $P_0$ contribution, but by the rotating contribution. Between 1$f_0$ and 3.5$f_0$, axial thrust shows a higher level of fluctuations that the in-phase or the rotating contribution. The SHD from cone pressure sensors cannot explain by itself all the phenomena acting on the axial thrust fluctuations.

At OP1, it can be seen that the low frequency pulsating pattern (0,15f0) as a direct impact on the axial thrust. More experimental data on deep part load are required to see if $P_0$ may allow to correlate correctly the axial thrust on a large zone, and to characterize more precisely pulsating phenomenon.
Measurements of runner strain, pressure, and thrust pulsation were conducted on a load variation. The experimental data shows that peak-to-peak pressure pulsations in stationary parts and strain fluctuation in the runner were poorly correlated. In particular, some phenomena pulsating axially like hydro-acoustic resonances have a strong impact on pressure pulsations but are not visible on runner strain fluctuations. The peak-to-peak value of pressure pulsations does not seem sufficient to properly assess the stability and the mechanical behaviour of the industrial turbine.

In comparison with the usual peak-to-peak approach on individual sensors, the SHD allows to distinguish phenomena by their shape and their motion. The impact of the phenomena on the stability and on the mechanical behaviour of the runner depends of these space-time features. Hence the SHD is a useful tool to have a better evaluation of the actual impact on the industrial machine of the pressure pulsations measured in the draft tube cone. It was shown that axially pulsating phenomena have mainly an impact on the axial thrust, but have a negligible impact on strain fluctuations in the runner. On the contrary, rotating phenomena like the part load vortex rope generate strain fluctuations in the runner, as well as radial thrust.

The SHD applied on cone pressure sensors is powerful at explaining the main phenomena above \( q = 0.6 \). Nevertheless it does not allow, for the moment to clearly identify all the phenomena above \( q = 0.6 \).
that take place in the deep part load zone below $q = 0.6$. The problem for this zone is that some instability is confined in the runner and escapes the detection by the standard sensors in the stationary part. Further test with additional sensors preferably in the runner would allow to distinguish more structures, and hence to have a better understanding of this operating zone. Understanding of the various phenomena and their impact being still in progress, on board strain measurement remains the most powerful tool to fully optimise the definition of continuous and temporary operating zone by evaluating finely the fatigue of the runner.

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Notations

| Symbol | Unit | Description |
|--------|------|-------------|
| $\hat{p}_k(t)$ | (Pa) | Signal power of $\hat{p}_k(t)$ and $\hat{p}_{-k}(t)$ |
| $P_k$ | (Pa) | Single SHD pattern |
| $f_{VR}$ | (Hz) | Part load Vortex Rope frequency |
| $f_s$ | (Hz) | A frequency in the stationary frame |
| $Q_{opt}$ | (l/s) | Optimum flow rate for the net head considered |
| $Z_r$ | (-) | Runner blade number |
| $k$ | (-) | Number of nodal diameters |
| $f_0$ | (Hz) | Runner rotation speed |
| $f_{UPLR}$ | (Hz) | Upper Part Load Resonance Frequency |
| $f_r$ | (Hz) | A frequency in the rotating frame |
| $Q$ | (l/s) | Flow rate |
| $Z_s$ | (-) | Guide vane number |