

State of the art hydraulic turbine model test

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Abstract. Model tests are essential in hydraulic turbine development and related fields. The methods and technologies used to perform these tests show constant progress and provide access to further information. In addition, due to its contractual nature, the test demand evolves continuously in terms of quantity and accuracy. Keeping in mind that the principal aim of model testing is the transposition of the model measurements to the real machine, the measurements should be performed accurately, and a critical analysis of the model test results is required to distinguish the transposable hydraulic phenomena from the test rig interactions. Although the resonances' effects are known and described in the IEC standard, their identification is difficult. Leaning on a strong experience of model testing, we will illustrate with a few examples of how to identify the potential problems induced by the test rig. This paper contains some of our best practices to obtain the most accurate, relevant, and independent test-rig measurements.

1. Introduction.

In hydraulic turbine development, model tests are essential, their initial role was to cast predictions on the performances of the industrial machine however, now it plays the additional role of addressing a large amount of contract guarantees. For several years, the quantity of measurement as well as the technologies used on test rig shows steady progress, and provides access to this information with an increasing level of accuracy. In this context, the responsibility of the experimenter is to ensure the relevance and accuracy of the measurements in question. And particularly, to make certain that the measurements are not distorted by the test rig, test conditions or any other mechanical issue. The latter is necessary to ensure the transposition of the industrial machine

The following article deals with tools used to ensure the relevance of measurements illustrated on several examples extracted from Francis and Pump-turbine reduced scale model test measurements managed in agreement with the IEC code 60193[1] and on 4 different test rigs of the GE hydraulic laboratory (France).

2. Dedicated analysis tools

In addition to a good knowledge of the hydraulic characteristics of turbine, we employ several tools to ensure the relevance of measurements and to detect the impact of test rigs

2.1. Reynolds effect

The Reynolds effect consists in changing the test conditions (turbine speed, net head, discharge, pump speed and consequently the Reynolds number), at the same hydraulic point defined by the same Q_{ED} , n_{ED} , guide vane opening γ , under the same cavitation conditions σ (Thomas number). We note that during a Reynolds effect the Froude number is not conserved.



The Reynolds effect test allows the use of sensors on a large range. This range of measurement should at least cover the calibration range on the ongoing test (including the right measurement of axial thrust, wicket gate torque...). During the Reynolds effect, the operation of the test rig also results in a strain on the mechanical parts of the model, and the wide range of operating conditions, allows for the detection of potential mechanical resonance.

What we expect during a Reynolds effect is a constant normalised number. This is because a normalised value should not depend on the test condition. It means that the efficiency referred to a constant Reynolds [1]: η^* (model efficiencies referred to a Reynolds $Re_{ref}=7.10^6$ in our case) should stay constant regardless of the test condition. Maintaining the same logical reasoning, the pressure pulsation divided by the net head should also stay constant as illustrated by the following figure 1:

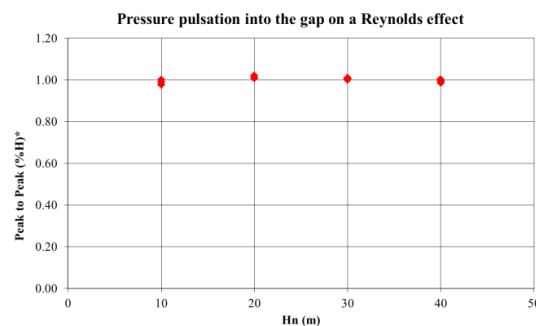


Figure 1. Pressure pulsation at the gap between guide vane and runner during a Reynolds effect.

The conservation of pressure pulsation divided by the net head during a Reynolds effect is the demonstration of a zero-impact of the test rig on pressure pulsation measurement. Given this status, a likely prediction can be forecasted of prototype pressure pulsation (Figure 1).

2.2. Sigma break curve

On a test model we modify the σ number step by step, decreasing the pressure over the free surface of the downstream reservoir. This change of sigma on the same operating point allows us to modify the cavitation condition and so, in case of rope, to change its gas volume.

2.3. Follow-up point

On a test rig, the same model can be tested for several weeks or months. The follow-up point is a chosen point close to the best efficiency point (BEP) to ensure a stable hydraulic. The recording of efficiency and dynamic measurements for this point is recorded at least thrice daily.

With the follow-up point, we verify the stability of sensors calibrations daily as well as the mechanical health of the model and make sure that no alteration has occurred since the first day of testing. This tool allows to detect a problem in measurement, or a calibration deviation even if the previous calibration is still valid and has been done just a few days ago. This tool can also help to identify a mechanical problem or a breakdown during a test. The follow-up point is used as a warning guide to help the experimenter ensure the relevance and accuracy of its measurements.

2.4. Signal processing

Further dynamic measurements are recorded each day on a test rig. The following chapters/paragraph is an overview of our tools used in dynamic data post processing and analysis.

2.4.1. FFT

The Fast Fourier Transform is a computational tool which facilitates signal analysis such as power spectrum analysis and filter simulation by means of digital computers. It is a method that efficiently computes the discrete Fourier transforms of a series of data samples (time series). The FFT is based on the hypothesis of stationary signals.

2.4.2. SHD mathematical tool [2]

The Spatial Harmonic Decomposition is a mathematical tool presented in paper [2]. The SHD combines information coming from a set of sensors in order to give a spatial description of the measurement. The SHD is used to distinguish synchronous and rotating part of hydraulic phenomenon.

2.4.3. FRF

The Frequency Response Function is a system characteristic which answers to a measured stimulation. The FRF is composed on the coherence computation, computation phase and ratio computation.

3. Research of relevant efficiency measurement on test rig.

The first aim of model tests is the prediction of hydraulic performances (power and efficiency). Nowadays, most of the contracts specify performance guarantees in the 0.01% and penalties from a 0.1% deviation between specification and measurements. Consequently, the accuracy of the measurement and transposition hypothesis have a direct impact on the market.

Following the IEC code 60193[1], the way to predict industrial machine performances is by using the efficiency transposition. This transposition is based on the dependence of relative scalable losses on Reynolds number, and so the independence of efficiency from scale model tests conditions.

The contingencies that can affect the quality of efficiency measurement are mechanical issues like assembly problems or measurement related problems like sensor deviation or perturbation.

3.1. Assembly problem detection.

The respect of a perfect hydraulic homology between model and industrial machine is the prerequisite to transpose test results. It leads to very small clearances on the model and particularly at the labyrinths seals. As a result, model assembly in the model testing process becomes dicey and the assembly quality has to be strictly controlled and monitored.

Around the best efficiency point, an assembly problem can be detected with a non-stable η^* during a Reynolds effect, explained by the non-scalable losses due to frictions. Figure 2, shows the detection of a friction problem (left) by the dependence of the η^* to the tests conditions. The second graph shows the results after the resolution of the identified problem (right). Actually, without friction, the efficiency referred to a constant Reynolds is independent on the test conditions and Reynolds number which validate the scale effect formula [1].

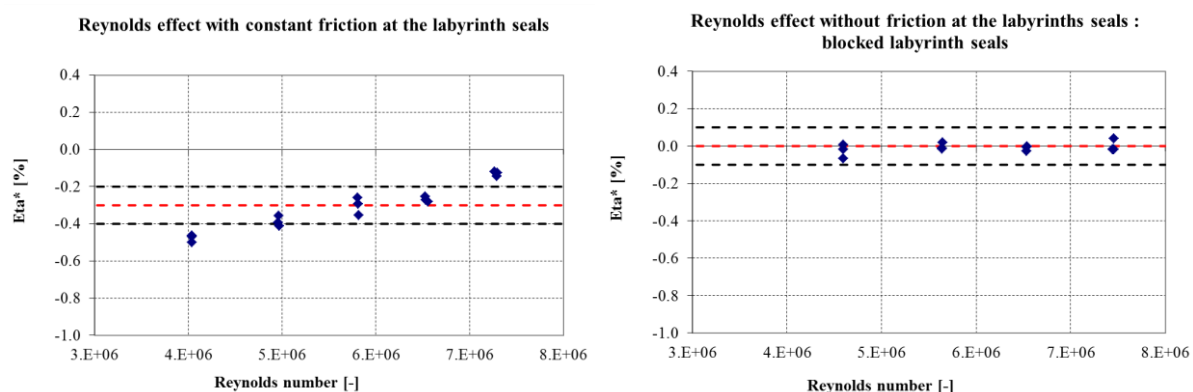


Figure 2. Determination of a friction problem due to a loose labyrinth ring on a Reynolds effect test.

To delve further into our diagnostic, the radial thrust measurement [6] investigation is a good way to determine the assembly quality and the health status of the mechanical parts. Indeed, on the same

Reynolds effects, friction is detected in figure 3 via analysis of dynamic radial thrust ($Fr_{ED\ dyn}$) waterfall.

In fact, at the BEP point, dynamic radial thrust should describe a pure unbalanced mass characterized by a unique emergence at f_0 whose magnitude is function of the square ratio of the model rotation speed (figure 3 right). In case of friction, the FFT signature on radial thrust is a shock signature with the presence of 10 harmonics (figure 3 left).

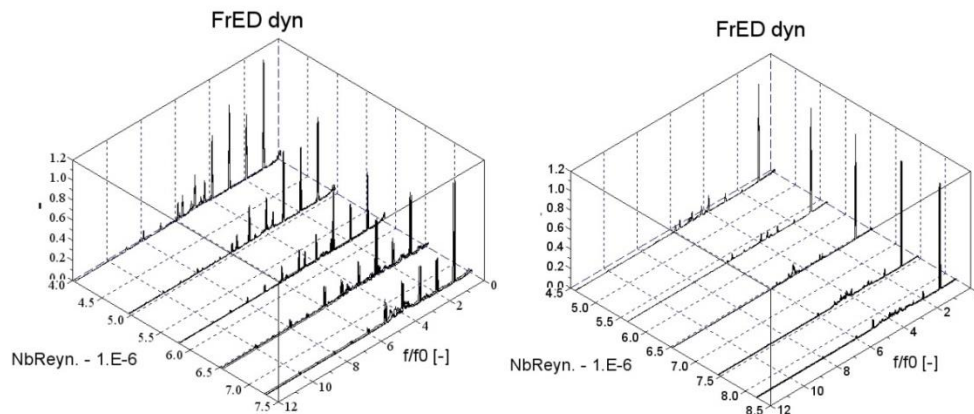


Figure 3. Detection of a friction problem on a Reynolds effect tanks to Radial thrust analysis

The Reynolds effect is a powerful tool to validate the model assembly, the efficiency measurements and all the dynamic measurements like pressure fluctuation, thrust, wicket-gate torque regarding the dimensionless factors. A constant factor is observed except in case of resonance (part 4) or measurement problems.

3.2. Measurement problem detection

The follow-up point tool enables us to bring out measurement deviation.

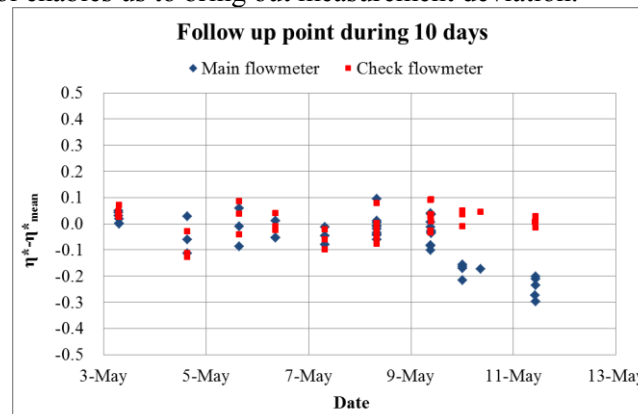


Figure 4. Determination of a measurement problem on an electromagnetic flowmeter, ascertained by the follow-up point.

In the case illustrated by Figure 4, we observe an abnormal decrease of the η^* on the follow up point (blue). After this assessment, we launched an investigation and called into question the relevance of the efficiency measurement since the 9th of May. On all of our test-rigs, all the sensors involved in the efficiency computation are doubled. Using the second flowmeter to compute the efficiency, we notice that the anomaly disappeared (figure 4 red). The various controls carried out on the flowmeters pipe line concluded that the position of the valve located upstream the flowmeter moved by 2 degrees inducing a 0.2% deviation of the main flow measurement, despite the 11 diameters pipe length

separating the valve from the flowmeter. For the experimenter, the knowledge of his test rig, circuit and impact for example of the valves on the discharge measurement is indispensable.

When applied to large volumes of measurements, the Reynolds effect test, follow up point and FFT tools enable us to detect and correct test-rig issues.

4. Dynamic measurements and test rig interaction

The stability and lifespan of a turbine is commonly evaluated through pressure pulsations. In a context where contractual guarantees on these measurements become more and more severe, cover larger operating conditions with an increasing number of sensors, a specific critical analysis on accuracy is a key point.

To ensure the relevance of measurements, the experimenter has to be careful to avoid test rig impact like pump, generator regulation parameters, mechanical parts natural frequencies or hydro acoustic resonance. This requires a deep knowledge of the test rig itself and of the involved hydraulic phenomena. For this purpose the classical contractual indicator (peak to peak 97%) is by far insufficient [3]. On a set of practical case studies, we will illustrate the specific methodology developed by G.E, using the analysis tools presented previously.

4.1. Detection of mechanical resonances due to shaft excitation.

With following example, the pressure pulsation under 30m net head observed on a take of load shows an amplification at the partial load vortex rope zone. (See Fig 5 at $Q_{ED}/Q_{ED\ opt}=0.6$).

This unusual Francis turbine behaviour finding leads to further investigations.

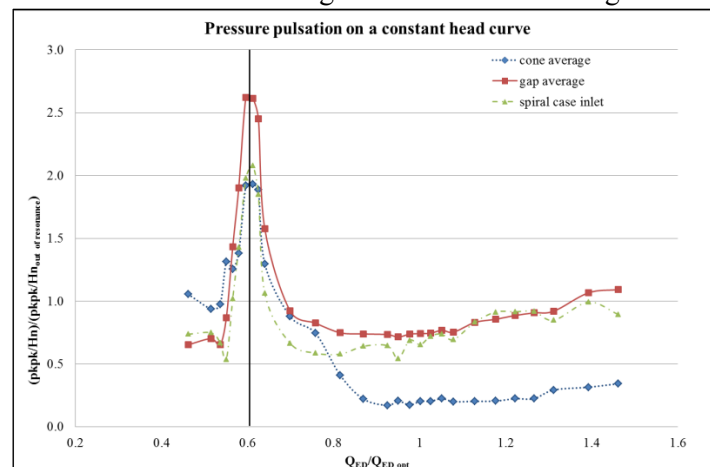


Figure 5. Pressure pulsation in all the model parts and identification of an exaltation

We conducted an analysis of the various frequency levels on a Reynolds effect in order to identify if the source of pressure pulsation amplification is hydraulic or external.

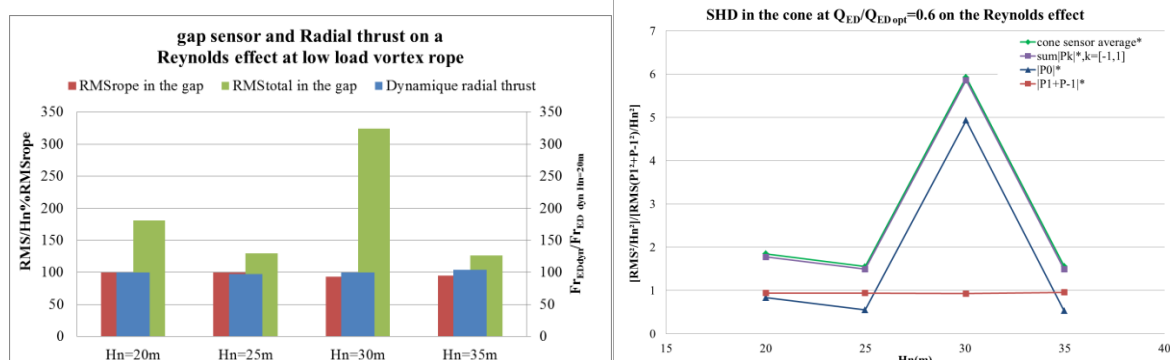


Figure 6. Frequency analysis of pressure pulsation at the gap between guide vane and runner,

dynamic radial thrust (left) and SHD based on 4 sensors located in the draft tube cone (right) on a Reynolds effect at $Q_{ED}/Q_{EDopt}=0.6$.

In figure 6 the analysis of FFT frequency levels show an abnormal dependence of the total pressure pulsations level on the tests conditions (Figure 6, left green) specially at $H_n=30m$. The fig 7 at this head shows that the levels are mainly generated at 27Hz which is known to be the first torsional mode of the shaft line.

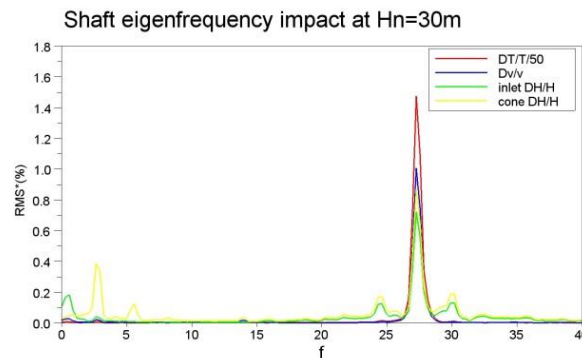


Figure 7. Torsion eigenfrequency of the shaft impact on torque, speed and pressure fluctuation for all the model locations under 30m net head.

At this stage, the prior determination of the mechanical eigenfrequencies of the test rig is mandatory to conclude to a torsional resonance. In our laboratory, the mechanical eigenfrequencies of our test rigs are determined thanks to shaker or vibration transient analysis. In such a case, we advise the measurement of the pressure pulsation at heads not impacted by resonance and if that cannot be achieved, a second option will be to filter pass band its effects.

It is also noticeable on figure 6 that some parameters are not impacted by the shaft excitation: the rope energy (left, red), the dynamic radial thrust [6] (left, blue) and rotating part or convective part $P1+P-1$ of the SHD [2][4]. This pointed out that among the available dynamic indicators, some provide a more relevant indication of the machine solicitation than the basic peak to peak approach.

As demonstrated in this part, the Reynolds effect test allows us to detect test rig resonance on dynamic measurements and consequently to choose the appropriate test conditions (head). In this case the resonance doesn't impact the energy of the rope itself but only the total energy of the signal. In case of interaction between rope energy and the resonance source, the $P1+P-1$ part of the SHD is a better indication as demonstrate in the paper [4] and in the following part.

4.2. Hydro-acoustic resonance detection

Various operating conditions lead to diphasic flows that creates condition's for the apparition of hydro-acoustic waves. In the following example, we will elaborate on the best approach in recognizing these types of resonance on a vortex rope.

Depending on the sigma number and Froude number, this rope can create a hydro-acoustic resonance [5], [7] and [8]. This hydro-acoustic resonance originates at the change of rigidity of the system (change of eigenfrequency of the model and test rig) due to the evolution of the rope gas volume depending on the Thomas number [4]. Fig 8 shows a Sigma break curve on a low load vortex rope.

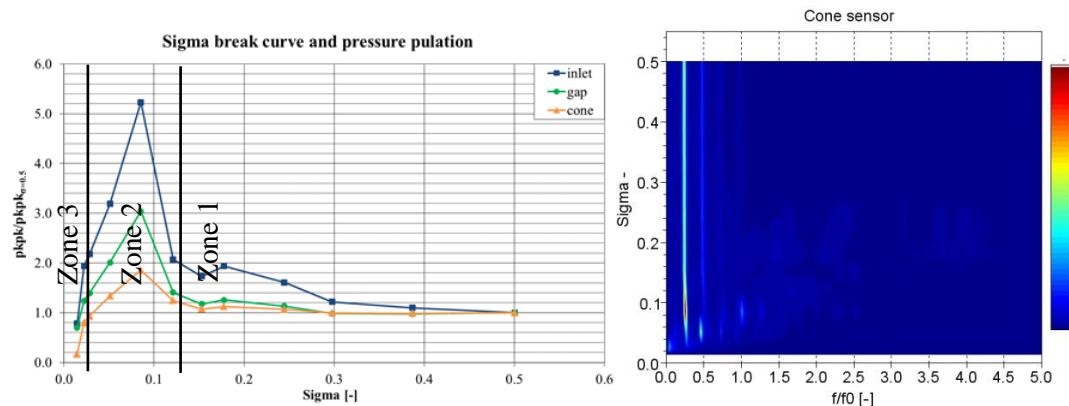


Figure 8. The pressure pulsation evolution during a sigma break curve in all the model parts and color waterfall in the draft tube cone.

In the cone, the peak to peak value is amplified by nearly 200% at sigma 0.09 (Figure 8). A FFT analysis allows to identify that the rope energy amplification is responsible to the total energy exaltation which occurred at a hydro-acoustic resonance. We can observe that the energy of the rope is similar to the transfer function of a one liberty degree oscillating system (Figure 8). We observe three zones: total transmission without deformation (zone 1), resonance and amplification (zone 2) and attenuation at low sigma (zone3).

When dealing with transposition to the full scale machine, the only precaution is to measure pressure pulsation out of resonance (in the zone 1) or to find other parameters less sensitive to tests conditions in order to characterize the industrial machine stability (as the part P1+P-1 of SHD paper [4]).

Pressure pulsations are responsible for head, torque, power fluctuations but also for mechanical stresses on the turbine blades [5] whereas hydro-acoustic waves are responsible for head, torque and power fluctuation only.

4.3. Detection of generator regulation parameters disturbances.

As a hydro-acoustic resonance, the generator regulation is another source of pressure pulsation exaltation, as well as head and torque fluctuation. With this example, during a constant head curve, an uncommon exaltation of pressure pulsation is measured on the full load rope illustrated by figure 9.

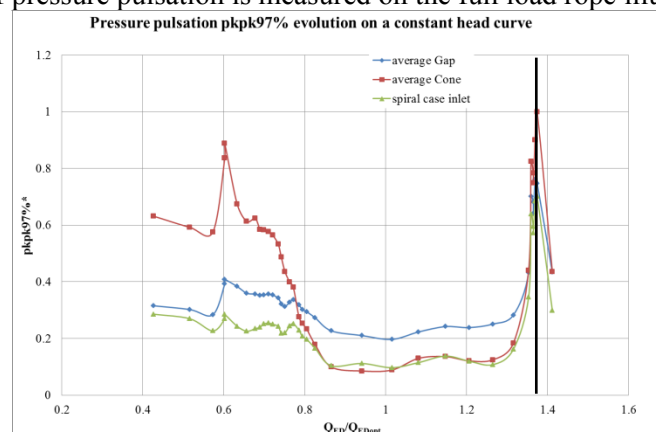


Figure 9. Pressure pulsation evolution on a constant head in the whole model at $H_n=30m$.

The abnormal level of full load rope energy conducted to test the independence of this fluctuation amplification against the test conditions was facilitated by Reynolds effect. To analyse the Reynolds effect, we have employed frequency analysis at the full load rope point.

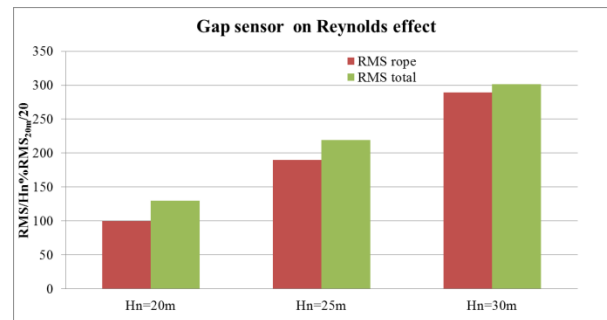


Figure 10. Frequency analysis of pressure pulsation in the gap between guide vane and runner on the Reynolds effect at the same point of full load rope ($Q_{ED}/Q_{ED\ opt}=1.37$).

The dependence of gap pressure pulsation, total or rope energy to the test conditions point out that the pressure pulsation measurements are not fully transposable to the industrial machine. We are faced with resonance impacting the rope energy. Moreover an analysis of a sigma break curve for this point doesn't highlight a hydro-acoustic resonance at such Thomas number (like in previous part).

Due to the detection of the rope frequency in the generator speed signal we performed a FRF between the torque and speed generator measurements which when conducted identified the problem as stemming from the regulation parameters. In our case illustrated on Figure 11, the excitation source (rope) occurs at a frequency where a strong coupling between speed generator and torque exists, and the regulation parameters of the generator amplify the absolute level of pressure fluctuation because of a resonance effect.

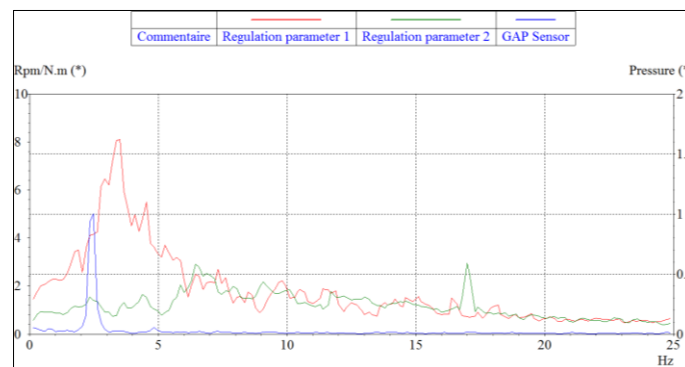


Figure 11. FRF coupling between torque and speed generator with two different regulation parameters.

Finally, as a result of the modification of the generator regulation parameters, the peak to peak pressure pulsation into the draft tube cone was divided by 300% (change from configuration 1 to 2, figure 12).

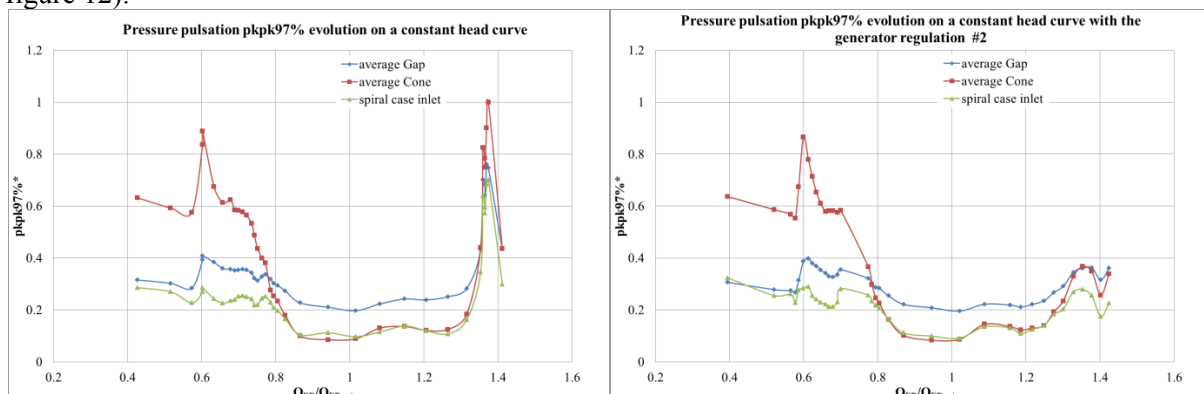


Figure 12. Pressure fluctuation depending on the generator regulation parameters represented by Peak to Peak 97% on the constant head curve (regulation 1 on the left and 2 on the right).

The Reynolds effect allows us to detect a test rig impact, by the dependence of pressure pulsation energy to the test conditions. The FRF conducted to change the generator regulation parameters. Dynamic measurements of parameters like generator speed, torque fluctuation in parallel to the dynamic acquisition of pressure pulsation are necessary to identify such resonance.

In this case, in presence of a test rig impact, neither the total energy of the signal (peak to peak value) nor the rope energy is a relevant indicator to characterize the hydraulic phenomena. We can conclude that the generator regulation could be responsible for a large increase of pressure pulsations.

4.4. Pump interaction in dynamic measurements.

Another source of disturbance of pressure pulsation measurements comes from the low frequencies phenomenon. The main difference between industrial machine and test rigs is the piping circuit and the presence of pump. Among the possible sources of low frequencies, we suspect that the test rig pump can induce low frequency disturbances which can propagate to the entire water conduit [1].

To confirm the pump responsibility on one of our test rig equipped with two pumps, we used the not adapted BP (high discharge) pump and the adapted HP pump (low discharge) on a same hydraulic point, under the same test conditions. Generally test rigs are equipped with only one pump and so cannot always operate close to its optimum point. (With one pump, this identification can be attained via the Reynolds effect. The only precaution is to ensure that the range of the Reynolds effect discharge is large enough to ensure the measurement of at least one point under stable pump conditions). (Figure 13)

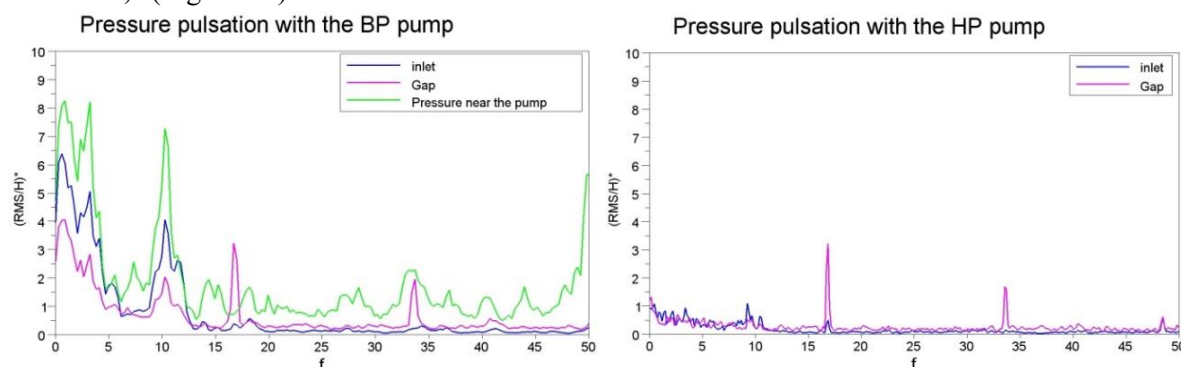


Figure 13. Comparison of pressure pulsation low frequencies in the whole model measured with a not adapted pump (left graph) and an adapted pump (right) to test conditions.

The excitation source is clearly the pump for two reasons: first, between the two measurements at the spiral case inlet and Gap location only the pump changed, the disturbances affect the entire model. In second, the dynamic pressure measurement near the pump shows more important amplitude than at the inlet of the model, validating the source of the excitation.

In our case the unsuitable pump generates 5 times more pressure pulsation at the inlet than using an adapted pump. When the test rig has only one pump, it is clear that this type of disturbances can appear, the only way to decrease this impact is to change the test conditions (Reynolds effect). Nevertheless, the pump and test rig circuit will always have an impact on pressure pulsation measurements at the model inlet. It appears that the impact of the pump is lower into the gap where specific frequencies due to Rotor stator interaction appear, than at the model inlet where the only remarkable frequencies are low frequencies disturbances.

The spiral case inlet pressure fluctuation measurements is a good indicator of test rigs impact on pressure measurements (identification of pump resonance), but is not representative of the industrial

machine inlet pressure fluctuation. Consequently, a contractual peak to peak value is not pertinent at this location regarding the observed phenomena.

5. Conclusion.

To ensure the life expectancy of industrial machine, in the context of constant evolution of methods and technologies used to perform model test, our main objective is the quality and accuracy of measurements. To obtain this certitude we developed methods of verification, and analysis tools. Applied on a large volume of measurements, these tools enable us to detect and correct test-rig issues. Keeping in mind that the principal aim of model testing is the transposition of the model measurements to the customer's machine, test-rig impact like pump, generator regulation, mechanical part eigenfrequencies, hydro-acoustic resonance are limitations. In other words, it is clear that the model measurements can be impacted by the test-rig.

We have also shown the limitation of pressure pulsation peak to peak 97 % values despite historically used to describe the machine stability. We observe that the sensors position and criteria have to progress in order to take into account the presented problems.

Even if we cannot prevent of all type of test rig impacts to progress in this way, and aware of these difficulties and limits, we develop the mathematical tools SHD [2] to identify and separate test rig impact from the transposable parameters. To preserve customer's turbine and assure the machine stability, we have identified mechanical measurements less sensitive to test conditions and more representative of runner stresses [5].

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