Francis turbine dynamic stresses measurements at model and prototype scales

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Abstract. A strain gauge and pressure measurement campaign on a model runner with a modern design was performed. The reference prototype runner was commissioned in 2013 and strain gauge measurements on the prototype are available for comparison. The model runner was fabricated by stereolithography in a plastic composite selected to obtain mechanical homology. The instrumented model runner was tested from 14% to 100% of the maximum guide vane opening at various speed factors ($n_{fd}$) by combining different head levels and rotating speeds as well as varying the cavitation coefficient (sigma) so that the effect of each studied parameter can be identified and quantified. By adjusting the speed and the head of the model test, it was also possible to induce rotor-stator resonance of the runner and to confirm expected hydrodynamic damping behaviour. Finally, the model scale pressure and stress measurements are compared to the ones obtained by both prototype measurements and CFD/FSI simulations.

1. Introduction
Hydraulic turbines are now required to operate in extended operating ranges to fulfil market demands. The challenge to design fixed blade turbines for extended range operation is now widely discussed in the literature, most specifically the challenges in predicting the stochastic pressure loads along with the induced dynamic stresses occurring in such operating conditions. The use of model scale strain gauge measurements can be used to investigate the complex Fluid Structure Interaction (FSI) phenomena [10,12]. The use of a test rig allows for performing tests at many speeds, heads and setting conditions as well as performing flow visualization. Altogether, it is expected that such a measurement campaign brings valuable insight into the understanding of hydraulic phenomena occurring on Francis units operated at Part Load (PL), Deep Part Load (DPL) and Speed No Load (SNL).

The model was instrumented with eight uniaxial strain gauges on two runner blades and two pressure sensors in one runner blade channel. The model is fully homologous to the prototype runner which was commissioned in 2013 and subject to a prototype strain gauge measurement campaign. The model test was performed solely for scientific purposes as the prototype was found to be robust and suitable for long term operation. This paper presents the results of the SNL, DPL and PL stress measurements of both the model and the prototype runners. Rotor Stator Interaction (RSI) hydrodynamic damping is also obtained. Finally, the results from model and prototype measurements are compared to simulation results.
2. Model instrumentation setup

2.1. Test rig

The model test measurements were performed in 2018 at Andritz Hydro’s Lachine Laboratory in Canada on test rig 4. The test rig has a 522kW pump capable of achieving a maximum head of 45m and a maximum discharge of 1.38m³/s. A 373kW motor brake is used to create a dissipative torque for turbines measuring 300 to 500 millimeters in reference diameter. A 37kW vacuum pump can produce pressures down to 95kPa below absolute in order to recreate the desired sigma levels. The rig has both a low (1Hz) and a high speed (up to 10kHz) data acquisition system to read data from all instrumentation. For this project, a telemetry system attached to the rotating turbine shaft was employed to relay strain and pressure information back to the high-speed data acquisition system.

2.2. Runner fabrication

The runner was fabricated by stereolithography (3D printing) from a polymer composite. The blade surfaces were polished, and the crown and band seal surfaces were machined. The geometry includes fillets at junctions between blades and crown and band. The runner blades, crown and band are mechanically homologous to the prototype runner in areas of interest.

The material was selected so that the runner can be tested at resonance under RSI excitation. The prototype frequency ratio between the potentially excited mode and the excitation frequency is also replicated on the model. The properties of the plastic material were tested by 3 points flexion on instrumented samples. The Young Modulus was tested parallel and transverse to the printing direction, and values were found to be within 6% of each other.

2.3. Instrumentation

Pressure measurements were performed using 2 watch battery style surface mount sensors measuring 3mm in diameter and 1mm thickness. Entran model EPL-B03-7B sensors were used. They were stainless-steel absolute pressure sensors with a 0 to 7bar range, 0 to 60°C temperature compensation, and a special 4V excitation to go with our telemetry system.

![Sensors used for model measurements](image)

**Figure 1** – Sensors used for model measurements

Strain measurements were performed using 8 water proof 3 wire, quarter bridge strain gauges by Kyowa (model KFWS-2N-120-C1-23 L1M3R). The sensors were connected to a specially made, waterproof 32 pin connector in order to bring excitation to and signals from the sensors, to a telemetry system made by KMT as shown in Figure 2. Signals from the telemetry system were sent wirelessly to a Quantum X data acquisition module. All signals were sampled at a frequency of 2400Hz.

2.4. Test plan

The test plan was designed to test the influence of many parameters on the Deep Part Load behaviour of the runner; head (16m < H < 32m), rotating speed (700RPM ≤ N ≤ 900RPM) and sigma (0.02 < σ < 0.12, σplant=0.095). For every speed, flow is varied at constant nED (0.29 < nED < 0.32, nED,proto = 0.30) and constant sigma. The summary of the nED constant curves, various guide vane openings at constant speed coefficients, performed for the deep part load investigation is presented in Table 1.
In the second phase of the test plan, a peak search at constant \( Q_{ED}/Q_{ED}^* = 1 \) with varied rotation speed is performed to plot the transfer function of the runner to the Rotor Stator Interaction (RSI) for the evaluation of the model runner natural frequency and damping.

![Telemetry system for rotor instrumentation](image)

**Figure 2** – Telemetry system for rotor instrumentation

**Table 1** – Deep part load dynamic stress investigation test conditions

| Speed [RPM] | \( n_{ED} \) | \( \sigma \) |
|------------|-------------|-------------|
| 700        | 0.320       | 0.091 0.110 |
|            | 0.300       |             |
|            | 0.292       | 0.091 0.110 |
| 800        | 0.320       | 0.110       |
|            | 0.292       | 0.110       |
| 900        | 0.320 0.300 | 0.110 0.110 |
|            | 0.020 0.056 | 0.091 0.110 0.121 |
|            | 0.292 0.020 | 0.056       |

3. **Model measurement analysis**

The strain gauge and pressure signals are processed using in-house signal processing algorithms. The average of the signals is removed using a detrend algorithm so that the calculated RMS values are representative of the dynamic portion of the signal only.

3.1. **Effect of Speed**

The results (RMS of dynamic portion of signal) from various \( n_{ED} \) constant curves with high and comparable sigma (0.11 < \( \sigma \) < 0.12) are plotted along \( Q_{ED}/Q_{ED}^* \) values in Figure 3 for the strain gauge with the higher stress amplitude. Reynolds number is as per convention of Hydraulic Laboratories and depends only on the tangential velocity of the runner and the kinematic viscosity which varies slightly with the water temperature during the tests.
Figure 3 – RMS strains as function of $Q_{ED}/Q_{ED}^\wedge$ for various $n_{ED}$ and Reynolds numbers

For each $n_{ED}$ constant curve, the tendency observed is very similar to the ones observed on many prototypes [2,7], where the dynamic stresses are typically higher from SNL to $Q_{ED}/Q_{ED}^\wedge \leq 0.4$ and decrease toward the high efficiency zone $Q_{ED}/Q_{ED}^\wedge \geq 0.6$.

The different signals are grouped around similar Reynolds number values or rotating speeds especially below $Q_{ED}/Q_{ED}^\wedge \leq 0.2$. Even if the model head varies significantly from $n_{ED}=0.29$ to $n_{ED}=0.32$ for a given Reynolds number it does not significantly influence the dynamic stresses measured close to speed-no-load. In summary, for a specific runner, the dynamic stresses near SNL depend strongly on Reynolds number but are relatively independent of test head.

3.2. Effect of sigma

The results at the highest Reynolds numbers (5.9E6 to 6.2E6) are used to compare the effect of sigma. RMS values of a typical strain gauge are plotted against $Q_{ED}/Q_{ED}^\wedge$ values for various values of $\sigma$ ($0.02 < \sigma < 0.12$, $\sigma_{plant}=0.095$) and $n_{ED}$ ($0.29 < n_{ED} < 0.32$, $n_{ED,proto}=0.30$) in Figure 4.

Figure 4 – RMS strains as function of $Q_{ED}/Q_{ED}^\wedge$ for various $n_{ED}$ and Re > 5.9E6
For the DPL zone \((0.3 < Q_{ED}/Q_{ED}^* < 0.5)\), smallest sigma values generate the lowest dynamic strains on the runner. The effect is the most pronounced at \(Q_{ED}/Q_{ED}^* = 0.4\), the point at which the largest dynamic strains have been measured for high sigma. For points of \(Q_{ED}/Q_{ED}^* < 0.4\), the variation of recorded RMS values is \(\pm 18.5\%\) above and below the average, which is less than the \(\pm 30\%\) variation observed with different Reynolds numbers. For this region of inter-blade vortices \(0.3 < Q_{ED}/Q_{ED}^* < 0.5\), it appears that a larger downstream suction head tends to reduce the appearance of dynamic stresses. While tests with higher submergence, either by having a higher test head or a higher \(\sigma\) (or both), exhibit larger stresses. From this result, it is apparent that the nature of the flow is characterized not only by the head and flow dimensionless parameters but also by the setting \((\sigma)\) of the machine.

3.3. RSI and damping analysis

The same dataset, curves with different \(n_{ED}\) values and \(\sigma\) \((0.02 \leq \sigma \leq 0.12)\), is used to obtain the amplitude of the signal at the RSI frequency. The RSI amplitude is obtained by calculating the average power spectrum of the signals using 1024 samples and a Hanning window. In order to compare results from different test heads, the strains are divided by the test net head, then strains per mWC are presented. For constant rotation speed and diameter, RSI stresses are scalable directly with the net head. The data selected is for the higher Reynolds numbers corresponding to \(N = 900\) RPM. At this speed, the excitation frequency is slightly above the natural frequencies of the excitable modes of the runner.

The response curves obtained are presented in Figure 5. The tendencies observed are similar to the ones obtained from various prototype tests for \(Q_{ED}/Q_{ED}^* > 0.6\) where the RSI stresses decrease almost linearly with the increase of the flow. It is shown in other papers [2, 5] that this variation is due to an increase of the hydrodynamic damping with the flow velocity.

![Figure 5 – Measured RSI strains per mWC at 900 RPM and for various \(\sigma\) values](image)

In the region between \(0.45 \leq Q_{ED}/Q_{ED}^* \leq 0.70\), the observed behavior may highly vary from the one typically seen on prototype measurements. The \(n_{ED}\) and the \(\sigma\) values are driving the amount of vapor in the flow which is showing to have an importance for the lowest \(Q_{ED}\) at which the RSI stresses show their full amplitude. Also, in some cases, an interaction between the rope and the RSI pressure can decrease the RSI stress amplitude in a significant way.

In the second test, a peak search with variation of the rotation speed at constant guide vane opening and \(\sigma\) is performed to plot the transfer function of the runner strains. The RSI strain amplitudes per mWC as a function of the excitation frequency are presented in Figure 6 for five strain gauges where dots are the measured points and lines are the transfer function best fits. The fitted parameters are force...
divided by the stiffness (F/K), the natural frequency (f_n) and the damping (ζ). Strain gauges 1, 2 and 3 are on one blade while strain gauges 4 and 7 are on a second blade.

![Image](image1.png)

**Figure 6** – Transfer function parameter fitting (n_{ED} = 0.30, Q_{ED} = 0.21, σ = 0.109)

The two blades exhibited different natural frequencies, with a 6% difference. One possible explanation may be due to geometrical variations due to the manufacturing process. The resulting natural frequencies and total damping values are presented in Table 2. The hydrodynamic damping is obtained by subtracting the plastic material damping from the obtained total damping. The plastic material damping was determined experimentally. The uncertainty on the values is calculated for a 95% confidence interval.

**Table 2** – RSI fitted parameters

|       | Rotating frequency for resonance [RPM] | Total damping ζ [%] | Hydrodynamic damping ζ_{hyd} [%] |
|-------|---------------------------------------|---------------------|----------------------------------|
| Blade 1 | 847 ± 11.8                          | 17.8 ± 6.6          | 15.7 ± 6.9                      |
| Blade 2 | 792 ± 23.0                          |                     |                                  |

The damping ratio is defined as the ratio between the damping coefficient and \(\sqrt{KM}\). As the damping (C) is induced by hydrodynamic forces it will not change with a different material. Therefore, the damping ratio for a steel runner at the same scale can be calculated from the following equation with constant relative error. As per modal analysis, 20% of the modal mass is attributed to the structure. It is assumed that the mode shape is not affected by the runner material change. This damping value can be extrapolated to the prototype scale by scaling with the flow velocity. [9]

\[
ζ_{hyd,m,plastic} = \frac{ζ_{hyd,m,plastic}}{\sqrt{E_{steel}P_{steel}}} = 3.53\%
\]
4. Prototype measurements

4.1. Prototype test conditions

The prototype stress measurements were performed shortly after the commissioning of the units with the intent of confirming the robustness of the runners predicted at the design stage. They were performed at the conditions presented in Table 3. The tested conditions, except for the Reynolds number, are within the value ranges tested on the model.

Table 3 – Prototype test conditions

| Parameter      | Prototype test values |
|----------------|-----------------------|
| nED            | 0.30                  |
| σ              | 0.110                 |
| QED            | 0.02 to 0.24          |
| Reynolds number| 2.46E8                |

4.2. Prototype instrumentation

The prototype was instrumented with 10 strain gauges on two blades (5 locations per blades), see Figure 7 for illustration. The strain gauge position and orientation was determined using FEM analysis to enable reliable extrapolation to the peak stress locations.

4.3. Stress measurement results

The measured stresses are presented in Figure 8. The stresses are normalized with the maximum measured dynamic stress. In the graph, results from the model strain gauge tests from the same nED and σ, are also added for reference. The results from both tests are normalized independently. The extrapolation of the results from the model to the prototype scale is discussed in the next section.

The tendency observed from both datasets are very similar. Dynamic stresses at the crown tend to have a higher increase from \( Q_{ED}/Q_{ED}^* = 0.2 \) to 0.4 than at the band, which is seen on both the model and the prototype. The sudden transition from the high stress zone to lower dynamic stresses occurs in both cases at very similar \( Q_{ED} \) values, which happens to be at slightly lower \( Q_{ED}/Q_{ED}^* \) than the rope. An important conclusion from our data is that the modal and prototype stress distributions as a function of flow are very similar.
5. Comparison of measurements with simulations

With the purpose of using the results from strain gauge measurements on a model for the design of new prototypes, an investigation on the scalability of the pressure dynamic loads is performed. CFD and FSI calculations are performed at model, prototype and at an intermediate scale (Dth = 35% of prototype) so that results at more than two scales are available.

5.1. Pressure Spectra comparison

The CFD and FSI methods are discussed in more detail in the literature [3,4,8,11]. The cross spectra from the simulated results and measurements are calculated using the pressure signals from both pressure sensors located in the runner channels. The spectra are shown in Figure 9. The positions of the sensors are homologous for all scales.
The comparison of the spectra obtained by simulation and the ones from the measurements exhibit different behaviour for the relative frequency \( f/n > 10 \) where the CFD simulation does not have enough spatial and temporal resolution to resolve the turbulence scales. Andritz’ internal FSI method solves this lack of resolution for the prediction of the runner stresses [3,11].

As discussed in previous publications [4,11], the CFD-FSI calculation for stochastic loads tends to be overly-conservative for smaller machines with low stresses but accurate for larger machines. The FSI analysis performed at the prototype scale of the machine investigated here shows good agreement with the strain gauge measurements in Figure 10.

5.2. Extrapolation from model to prototype scale

Scalability of turbulent pressure fluctuations is widely studied. Most of the publications focus on thin foils, pipes or pipe bundles for either the aviation or nuclear industries. For some cases, it was shown that the wall pressure fluctuations could be extrapolated using the natural logarithm of the Reynolds number. The logarithmic regression between the Reynolds number and the normalized stress is presented in Figure 10. This exercise assumes that the stress fluctuations are proportional to the wall pressure fluctuations.

![Figure 10 - Extrapolation from model to prototype measurements: logarithmic regression with Reynolds number](image)

The regression from all the points (in yellow), including all scales, measurements and simulations, has good quality with \( R^2 > 0.95 \). However, the regression considering only model test points has a quite different slope. Then if model test results are used to extrapolate stresses from the model scale to the prototype scale an underprediction of about 40% is obtained, which indeed is not acceptable for stress prediction and fatigue analysis of the prototype.

Other extrapolation methods were investigated. While some are promising, classical parameters such as the tangential speed and the head did not lead to satisfactory results. Extrapolation between scales for similar hydraulic conditions is possible, however extrapolation to other conditions or other turbine designs may not be possible without the contribution of CFD due to the complexity of the flow.

6. Conclusion

Strain gauge measurement campaigns on a mechanically homologous Francis turbine at model and prototype scales were performed. The model was tested at 3 different Reynolds numbers, with variations of \( n_{E D}, Q_{E D} \) and \( \sigma \). The exact conditions of the prototype test, except for Reynolds number, were replicated at the model.

From the results of the model strain gauge measurements it can be concluded that the SNL and DPL stresses strongly depend on the Reynolds number, or the rotating speed. As expected, higher stresses are
found at higher Reynolds numbers. The $\sigma$ value also influences the strain level especially in the inter-blade vortex zone. It must be noted that the $\sigma$ values at which lower dynamic stresses were found are not typical of real prototype machines.

The strain measurements from the model test show similar tendencies as the prototype strain measurements when observed on an $n_{Ed}$ constant curve. However, the absolute scalability of model to prototype strains are still under investigation.

Measurement on a mechanically homologous model runner allows for the testing of many conditions which are not possible to test on the prototype. Such measurement campaigns can provide valuable information, not only specific to a design, but also for the scientific aspects pertaining to the design of runners for 0-100% operation. Similar measurement campaigns combined with cutting edge flow visualization techniques should provide highly valuable data for the study of these FSI phenomena.

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