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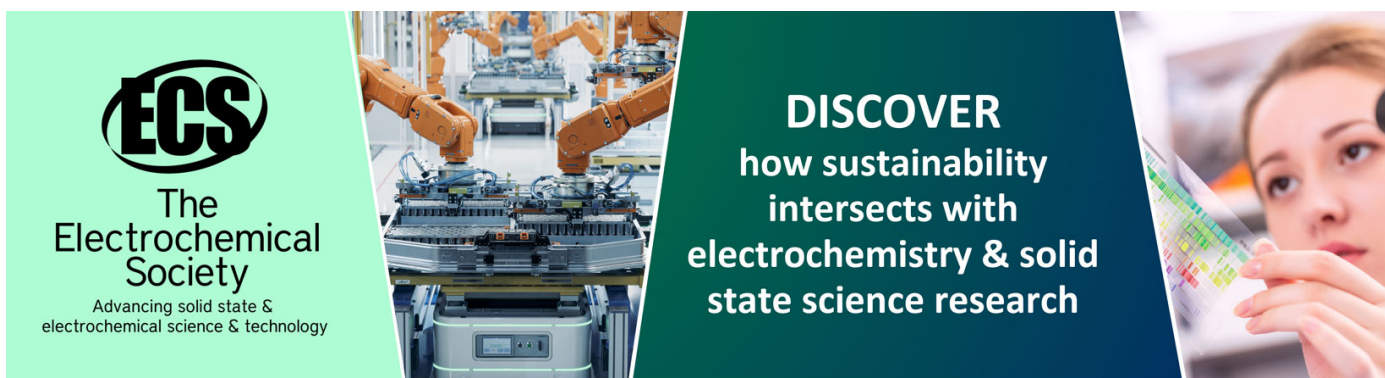
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Stress predictions in a Francis turbine at no-load operating regime

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Abstract. In the operation of hydraulic turbines, no-load and very low load conditions are among the most damaging. Even though there is no power generation, there is still a significant amount of energy which has to be entirely dissipated, mainly in the runner, where the flow is quite complex, with large scale unsteady and chaotic vortices resulting from partial pumping. This paper presents different approaches to perform stress analyses at low load conditions on a Francis turbine, taking into account the pressure fluctuations on the runner blades due to the large stochastic flow structures inherent in no-load operating regimes. With appropriate mesh density and time step, unsteady computational fluid dynamics (CFD) simulations using the SAS-SST turbulence model can be used on a Francis runner to predict the pressure fluctuations with reasonable accuracy when compared to measurements. These calculated pressure loads can then be used to predict the dynamic stresses with finite-element analyses (FEA). Different approaches are discussed ranging from quasi-static single-blade models to full runner time-dependent one-way fluid-structure interaction (FSI). Pros and cons of the different modelling strategies will be discussed in a detailed analysis of the structural results with comparisons to experimental data. Once the time signal of the stochastic stress at no-load conditions is obtained, the runner fatigue damage related to this operating condition can be estimated using different tools such as time signal extrapolation and rainflow counting.

1. Introduction

During its lifetime in operation, a Francis turbine will go through different operating regimes that will each affect its life expectancy and reliability to a different degree. While it is now well known that some transient and dynamic regimes can be damaging for the runner [1][2], the quantitative contribution of each of these operating conditions remains to be ascertained. In most Francis turbines, it has been observed that startup, runaway, no-load and very low load regimes are the most damaging [3]. Among these, no-loads are probably the less studied and understood hydraulic regimes even though they are quite frequent and can occur over extended periods of time.

However, the absolute damage level caused by no-load regimes varies from one runner to another. One objective here is to understand the dynamic regime under such a flow condition to determine if it is critical or not for the life expectancy of a specific turbine. Under a no-load regime, there is no power



generation even though a non-negligible flowrate still exists through the turbine. This leads to a significant amount of energy that has to be dissipated by the flow itself. This occurs mainly in the runner because of the significant recirculation taking place due to partial pumping. The end result, most of the time, is a quite chaotic flow with large-scale unsteady vortices in the interblade channel and under the runner. It has been observed that this flow regime can generate high dynamic fluctuations on the runner blades.

In the context of either design or operational optimization, assessing the cost of operating under no-load conditions for a specific turbine requires the capacity to adequately model the dynamic loads to capture stochastic loads leading to fatigue damage. These loads may then be used to evaluate mechanical cyclic stresses, critical to the fatigue damage mechanism. The whole idea is to generate a stress signature from numerical modelling sufficiently representative of the stochastic behaviour and to use it in a life-assessment tool.

From a previous paper [4], it has been shown that with appropriate computational fluid dynamics (CFD) modelling it is possible to capture the unsteady pressure load with a satisfying accuracy. It was also shown that the dynamic strains on the structure could then be determined with acceptable accuracy using a specific mechanical model. In [4], no details on the mechanical model were given and the focus was kept on the CFD modeling. This paper proposes to look at two alternative mechanical models that could facilitate the evaluation of mechanical stresses. First, a time-dependent one-way fluid-structure interaction (1-way FSI) approach neglecting the effects of water added mass is carried out. Secondly, a less computationally expensive quasi-static analysis is performed. The results are compared with runner strain gauge measurements and the capacity to properly evaluate the fatigue life of the runner is discussed.

2. CFD summary

As described in [4], the CFD setup used to perform the unsteady simulations included the spiral casing, the entire distributor, the complete runner and the draft tube, (see figure 1 left). The flow was resolved with a high resolution scheme for advection and a second order backward Euler time marching scheme within Ansys CFX.

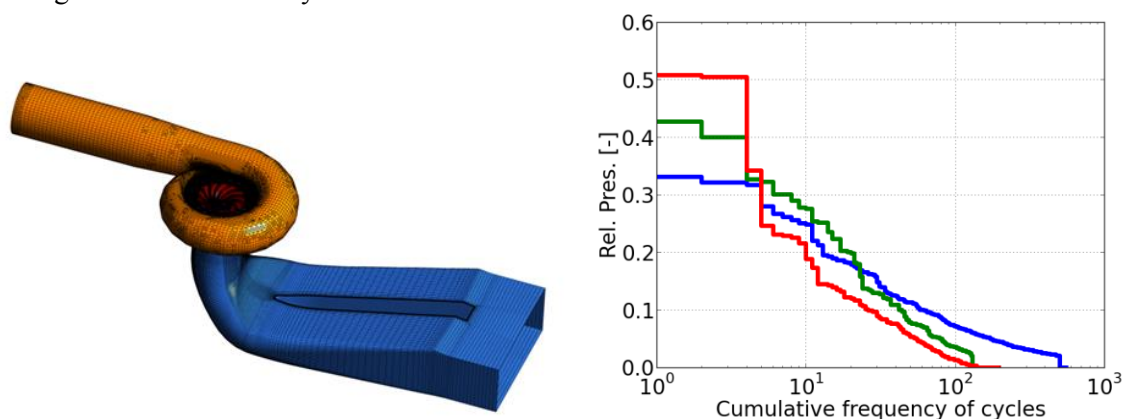


Figure 1. CFD domains (left) and rainflow diagram of measured (blue) and predicted (red and green) pressures (right).

From [4], it has been shown that SAS-SST, a scale-resolving turbulence model, allows resolution of the disorganized flow associated with a no-load operating regime significantly better than a standard RANS $k-\epsilon$ model. Many large vortices responsible for the stochastic pressure fluctuations are captured by the SAS model and with the appropriate mesh and timestep, the results have been found satisfactory in terms of ranges of pressure fluctuations for life-fatigue assessment. To do so, the modelling must be able to produce a temporal signature representative of the experimental one. Figure 1, on the right, shows a rainflow diagram comparing measured and predicted data for one pressure

probe. It shows a similar tendency between experiments and prediction, with a little over-prediction of the largest events of low occurrence, responsible for most of the fatigue damage.

3. Experimental data

Unsteady pressures were recorded at various operating regimes, including no-load, with 12 pressure probes located on the middle streamline of a Francis turbine blade. These pressure measurements are used to validate the CFD model (Figure 2). More details are available in the previous paper [4] where they were used to validate the CFD calculations.

Uniaxial strain gauges were also installed on the blade to measure the mechanical behaviour during the commissioning of the runner. The location of the strain gauges is shown in figure 2. The measured strains can then be used to validate our predictive approach of fluid-structure interaction at the no-load regime.

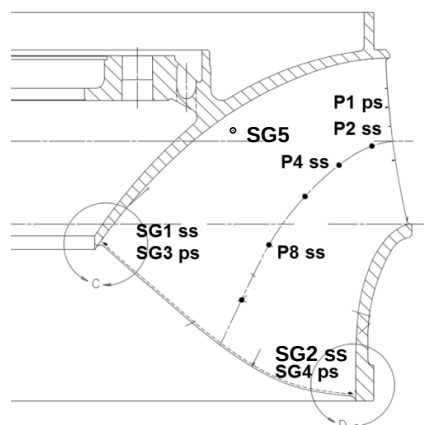


Figure 2. Pressure probe and strain gauge locations on the Francis runner blade

3.1. Strain gauge measurements at no load

Strain gauge measurements were performed for several minutes under different operating conditions. It can be observed in figure 3 that the time history is stochastic and does not repeat from one time window to another. The use of statistics is therefore the best approach to characterise the measured signal and to make comparisons between experimental and numerical results.

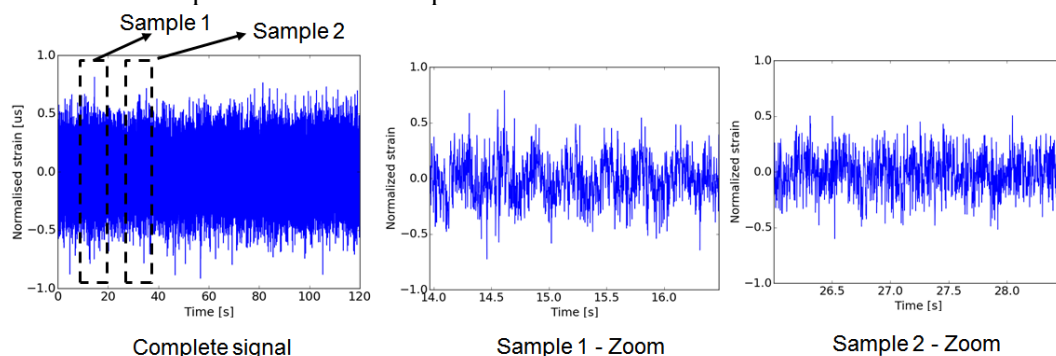


Figure 3. No load strain gauge measurement time signal (SG3)

3.1.1. Sample probability and convergence. Samples of a similar duration to the simulation (2.2s) are extracted from the complete measured signal. The measured maximal strain amplitudes of each signal sample are plotted as a histogram in figure 4. The same procedure is applied using larger samples (16s). Even though the number of counts is clearly different (fewer samples of 16s), it demonstrates that larger samples tend to include higher strains resulting from rare events. The larger the sample, the higher the probability of obtaining larger maximum strain amplitudes close to convergence.

Statistically speaking, it can also be noted, assuming a Gaussian distribution, that the scatter of the maximum strain amplitude is reduced when the sample is enlarged.

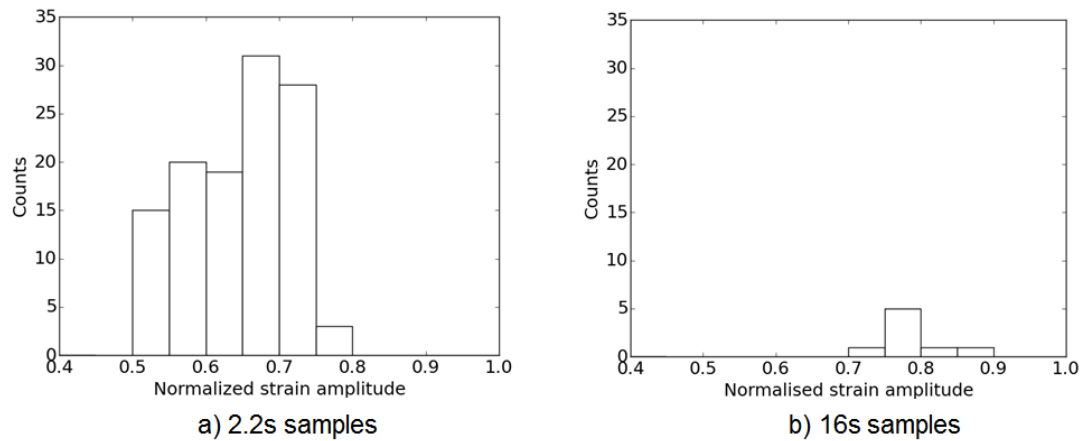


Figure 4. Strain amplitude histograms for various sample size (SG3)

3.1.2. Signal extrapolation. In order to compare simulation and experimental results on the basis of the maximum amplitude cycle, the period of time selected should be large enough so that the difference between samples decreases to an acceptable level. It has to be considered that given the random nature of the process, there will always be variations between samples, even with very large lengths.

In order to obtain significant simulation data, it is necessary to extrapolate the results obtained for short time simulations. In this case, signals are extrapolated using the extreme values theory and a Monte Carlo simulation [5]. Values exceeding pre-determined thresholds for minimum and maximum are used to calculate the statistical distribution of the signal, which is then used to generate longer-duration signals.

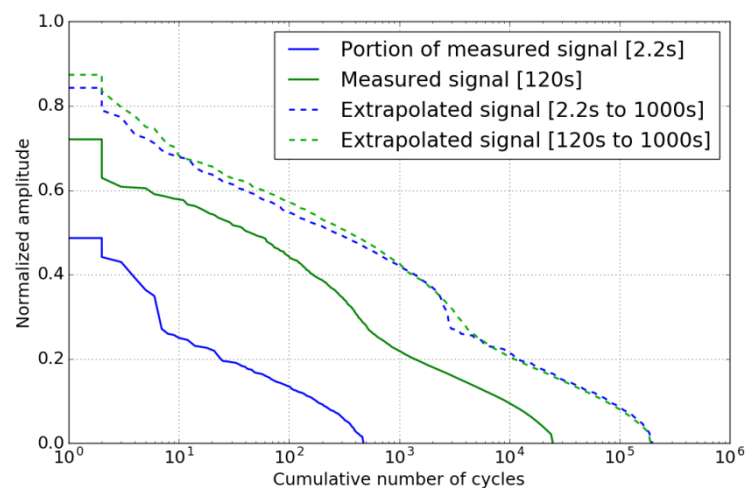


Figure 5. Rainflows of measured samples and extrapolated data

The extrapolation methodology is also applied to the experimental data. Two samples from the same signal, but of different lengths, are extrapolated to a reference length of 1000s. The extrapolated rainflows, shown in figure 5, have similar shapes and maximum amplitudes, making the comparison possible between short simulation signals and experimental results.

4. Mechanical analysis results

Pressures from CFD calculations are applied on the simplified mechanical models studied. As the deformations are expected to be small, they should not affect the fluid flow and thus 1-way FSI simulations, both quasi-static and transient, have been retained in this study.

4.1. Numerical setup

Mechanical FEM simulations presented here have mostly been performed using Ansys Mechanical 15.0, which is part of the Workbench suite. Considering the CFD setup used, which was summarized in section 2. , and the expected hydraulic behavior, a complete Francis runner prototype with 13 blades, complete band and crown, has been modelled and meshed with quadratic tetrahedron elements.

To obtain a good compromise between the cost of calculation and accuracy, the mesh generated with the built-in mesh generator inside Ansys Mechanical has been targeted to get approximately 100k nodes per blade, with more refinement in the fillet at the junctions with crown and band (mesh 1). A more refined mesh of a single blade has also been used and a 360° mesh with just one refined blade, following the approach suggested by [5] was built accordingly (mesh 2). This second full runner mesh enables a lighter overall mesh (allowing more time steps to be simulated with the same memory requirements) while keeping sufficient refinement for one blade of interest. The full runner meshes in those two configurations can be seen in figure 6.

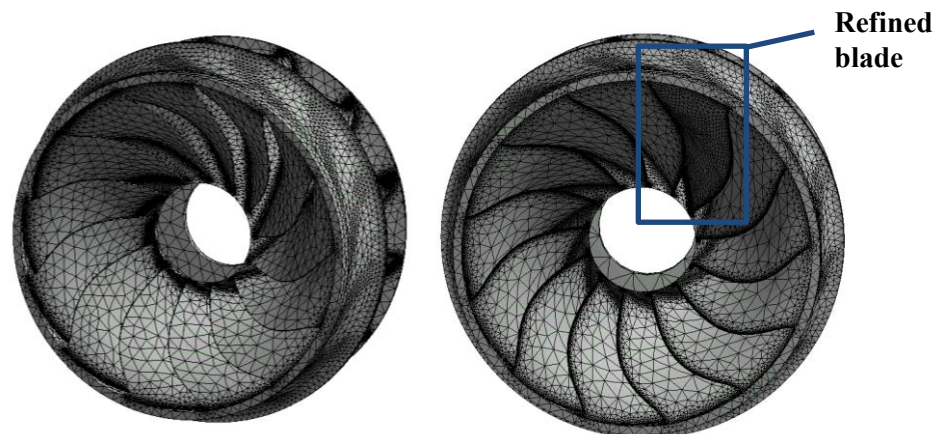


Figure 6. Full runner mechanical mesh 1 (left) and full runner mesh 2 with a refined blade (right).

As the idea is to assess dynamic stresses, the centrifugal and gravity loads are not included in the model. A zero-displacement boundary condition is imposed at the coupling flange. In both static and dynamic simulations, the only load is the pressure load. Default interpolation profile preserving with triangulation was used in all cases.

To manage all the dynamic loads in time-dependent simulations efficiently (one pressure load on all blades at each time step), an extension supported by Ansys is used. It enables the pressure on the blade to be converted within Ansys CFX at each time step into a pressure load that will be tagged to be loaded into Mechanical at the corresponding time step. This extension configures a transient mechanical analysis with a list of time steps and the corresponding pressures are loaded automatically. In doing so, it is easier to use the same CFD results for many mechanical simulations, as opposed to the built-in 1-way FSI implementation which requires the fluid and structure to be co-simulated at the same time in order to manage the temporal evolution of many time steps. The disadvantage of the proposed approach is the huge requirement in memory, which keeps increasing as the simulation time advances. Large transient simulations, such as the ones performed in this work, are thus memory- and time-consuming. They required approximately 5-6 days of calculation distributed on 64 cpus. Moreover, post-processing is not parallelized and requires at least 128Gb of memory.

The same time steps were used in mechanical and fluid simulations, which means that each CFD pressure load was used as an input in time-dependent mechanical simulation. Depending on the frequencies of interest, sub-sampling of the CFD load could be used to decouple the time step requirements of these FSI simulations which so far have been driven by the CFD. Acquisition frequency of the experiments could also become a criterion in the process of validation.

4.2. Time-dependent approach versus quasi-static approaches

In weighting the simulation cost-benefit, one could ask if the time-dependent mechanical simulations are required at all to feed the fatigue model or if a simpler approach could be used. A good proxy to evaluate the loads on the blade is the dynamic blade torque, which integrates the pressure field fluctuations over the whole blade. One hypothesis that was investigated was to check if the pressure loads corresponding to the maximal and minimal torque over a defined period of time may lead to infer peak-to-peak stress and strain ranges with a quasi-static approach using the same mesh. If so, a much less computer intensive approach could be used, requiring no temporal integration and a reduced number of loads (for example, one could choose as few as two loads to get the largest cycle only). In Figure 7, static and dynamic strains are compared at five locations between time-dependent and several quasi-static simulations. Static strains represent the instant of maximal torque and dynamic strains correspond to the difference between the instant of maximal and minimal torque.

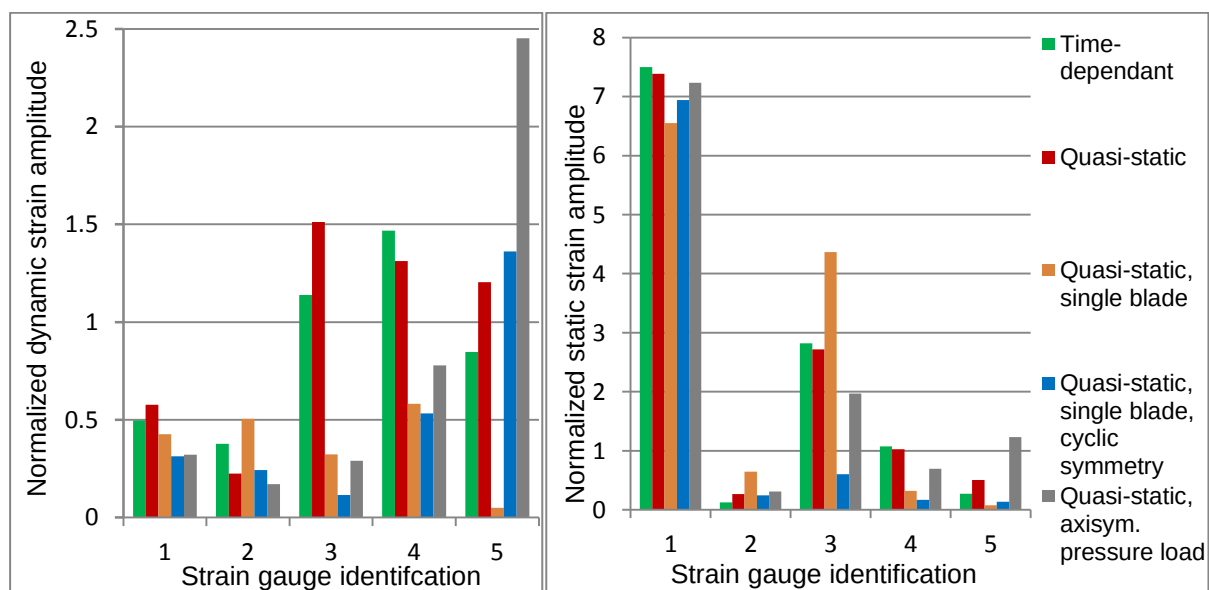


Figure 7. Comparison of different numerical approaches for five strain gauge locations for both dynamic (left) and static (right) strains.

When compared to the time-dependant simulation (green), quasi-static simulation with a complete runner (red) gives a reasonable approximation, although some more statistical considerations need to be taken into account for fatigue damage calculation. As the solver requires a full runner model in transient mode (no cyclic symmetric available with time-integration within Ansys Mechanical), the complete geometry was used at first in both quasi-static and dynamic approaches. It was also consistent with the CFD simulation setup where the complete machine was modelled. Nonetheless, by using a quasi-static approach, it is not mandatory to model the whole runner. The simulations (with the pressure loads from maximal and minimal blade torque) were thus conducted with a single blade, with or without the use of cyclic symmetry boundary conditions, and with an equivalent mesh size. Results are also shown in Figure 7. While some results are in the right range, most of them show wide discrepancies with the full runner results. Consequently, it seems to disqualify the quasi-static approach with single-blade models (orange and blue) to evaluate the effect of stochastic loads like

those occurring under no-load conditions. Finally, a full runner quasi-static approach (gray) with an axisymmetric pressure load is also compared. It consists of copying the load seen by the high-torque blade onto all the other blades. Once again, it can be seen that this approach does not deliver results corresponding to the complete pressure load. In all cases studied, simplified boundary conditions do not lead to satisfying results. Thus, both complete geometry and complete load seem to be required to get an adequate range of dynamic strains.

4.3. Time-dependent simulations and comparison with experimental results

In the validation process, predictions are compared to available measurements. Consequently, validation is performed only at selected locations where strain gauges are available. First it is important to note that the measured point do not necessarily correspond to the hot spot location. Second, there is no way to make sure that static and dynamic hot spots are the same. In Figure 8, a static strain field corresponding to maximal blade torque is presented. The largest deformations for this runner are found at the junction between the blade and the crown, near the trailing edge. Dynamic strains shown correspond to the range between the instant of maximal and minimal torque on the instrumented blade. For this runner, the regions with higher static stresses correspond to those with higher dynamic stress ranges. Nevertheless, since the dynamic pattern is different from the static one, this might not always be true.

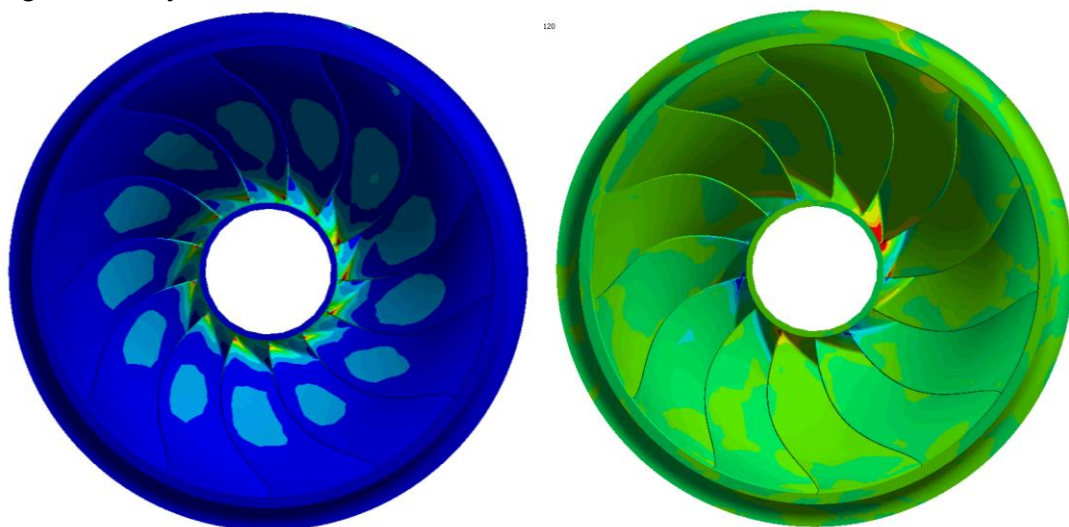


Figure 8. Normalized static (left) and dynamic (right) strains in a time-dependent simulation.

Strains obtained from the time-dependent 1-way FSI simulation vary according to stochastic pressure loads in a fairly disorganized way. Figure 9 shows five strain signals representing the deformation in the location and direction of the corresponding uniaxial experimental strain gauges. The highest static deformations are seen by the SG1 while the highest dynamic range is seen on SG3. Two mesh configurations are shown for SG1, although the signals almost match at this scale.

To allow comparison with experiments, signals must be compared on comparable timeframes. To do so, the extrapolation method used for experimental data is also applied to numerical results. Moreover, to get more insight from a short-duration numerical simulation, time-dependent signals from each strain gauge are recorded on each blade, thus allowing multiplication of the length of each signal by 13. Strain gauge time signals presented in Figure 9 are the concatenated signals. As the focus is on stochastic signals of a symmetric structure, using spatial variability is a way to increase temporal signal length. Concatenated longer signals increase the extrapolation precision because the probability of including large-amplitude/low-occurrence events increases.

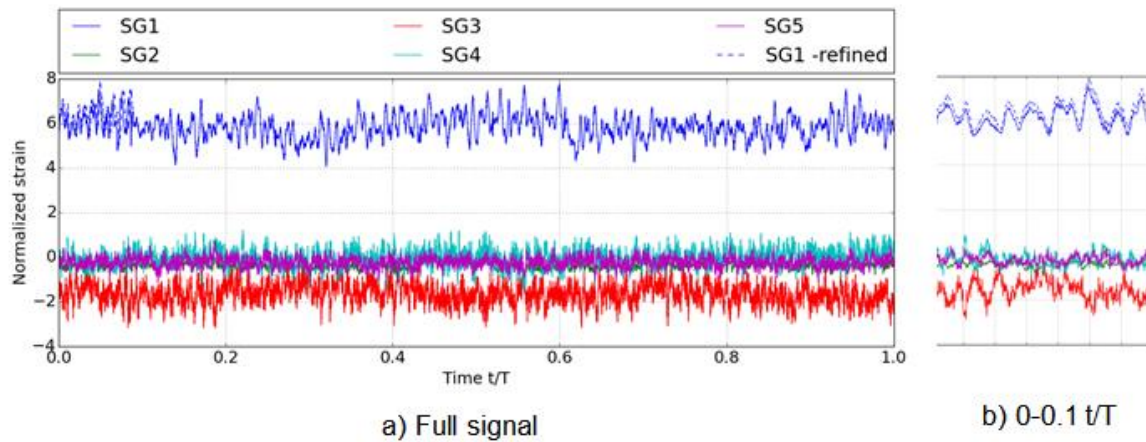


Figure 9. Normalized strain from numerical simulation with mesh 1. Time signals from 13 blades concatenated for all five strain gauges.

After both experimental and simulation time signals have been extrapolated to 1000 seconds and convergence of the extrapolation has been confirmed, comparison can be performed. Figure 10 presents the peak-to-peak amplitude for each strain gauge over the whole extrapolated signal for two experimental datasets and one numerical prediction coming from the 1-way FSI time-dependent approach with mesh 1. This largest amplitude is the critical data for the life assessment. At first, it can be seen that there is already a spread in both sets of measurements. Due to the stochastic nature of the flow, dynamic strains under no-load regime present inherent variability. On the other hand, numerical simulations offer just one deterministic answer with the approach presented here. Results are satisfying for some of the strain gauges, and not at all for some others.

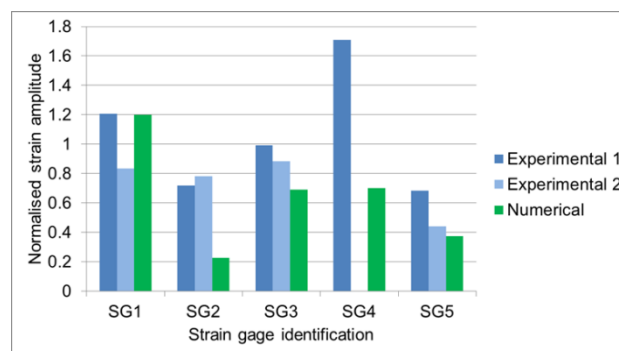


Figure 10. Comparison of normalized strain amplitude from numerical and experimental results for five strain gauge locations.

Maximal amplitude does not tell the whole story. The complete deformation spectrum of SG3 is presented in Figure 11. The lack of high frequency resolution in the numerical prediction is obvious. However, the content of the lowest frequency is fairly well captured. High amplitude with very low frequency (1/revolution) and rotor-stator interaction (66 Hz) appear in this signal.

Differences in frequency spectrums from simulation and experiments can be partly explained by uncertainties on geometry and strain gauge locations, fabrication tolerance versus identical blades in the numerical model and added-mass effect neglected in the simulation. In such a stochastic flow, turbulence certainly plays an important role exciting natural modes in a broad frequency range from 0 to hundreds of hertz. Simulations would benefit from more refined meshes, smaller time steps, better turbulence modelling and possibly cavitation modelling to capture this. But the computational cost

required would be much higher. Nevertheless, the most important phenomena are present to at least start to predict fatigue with such modeling.

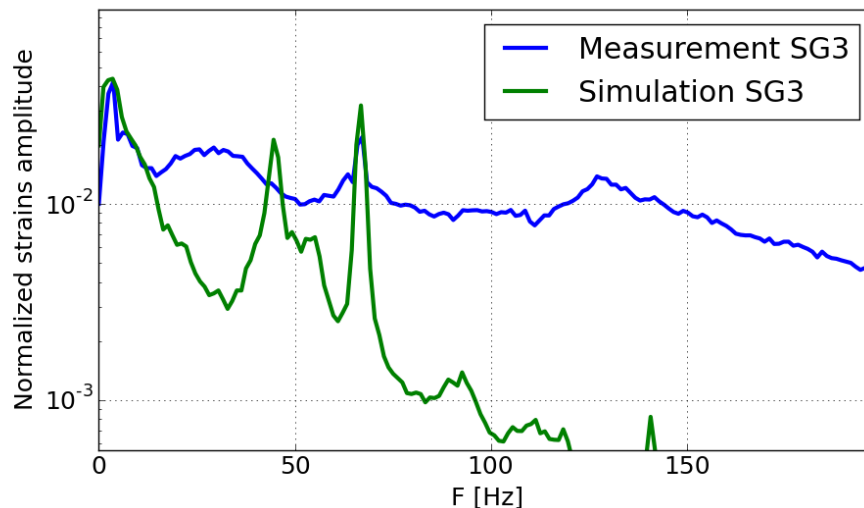


Figure 11. Comparison of frequency spectrum for a strain gauge in both numerical simulation and experiments.

Rainflow diagrams are another useful way to compare the signals with respect to fatigue evaluation. They enable the classification of fluctuations using their amplitude. It is assumed that, if the tendency is similar between the measured and predicted strain ranges (particularly in the low occurrence-events), the fatigue life assessment can be done using the prediction, thus validating the modelling approach. Figure 12 shows rainflow diagrams at two locations (SG1 and SG3). Once again, the spread between experimental data sets is visible. Even though numerical simulation under-predicts across most of the spectrum, it is not so far from the range of uncertainty, particularly in the case of SG1. The tendency is well captured, but without experimental data, the numerical approach is not accurate enough to be used as an effective tool for life-assessment yet. However, it remains a powerful tool to help better understand the physics at play. Since the limitations are already known, the approach will be further improved in the future.

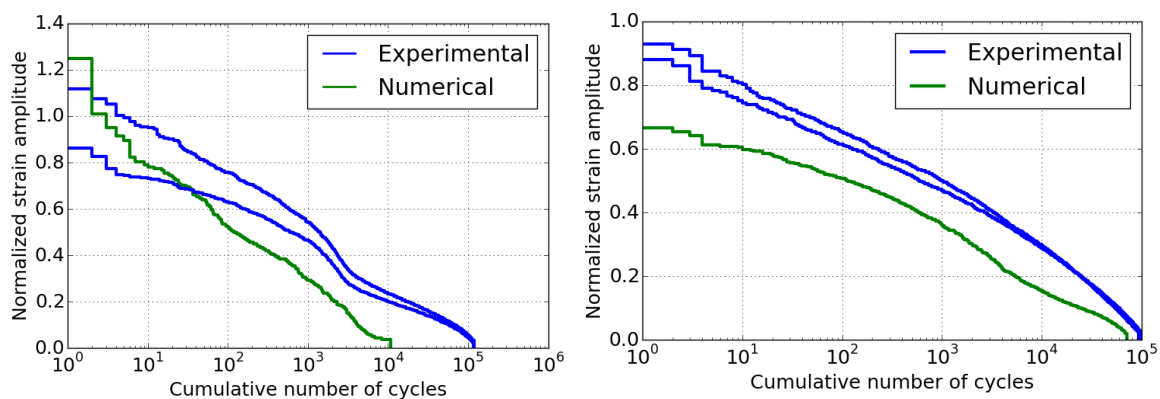


Figure 12. Rainflow diagrams at two strain gauge locations for numerical and experimental results (SG1 on the left, SG3 on the right).

5. Discussions and conclusion

This paper presents an attempt to predict Francis turbine strains and stresses under a no-load regime with a stochastic behaviour. The approach presented is a mechanical FEM analysis in the time domain using a large number of pressure loads obtained from an unsteady CFD simulation. This 1-way FSI

setup is compared with experimental values using strain gauge data from five locations on a Francis blade. To enable such comparisons, it has been shown that signal extrapolations with Monte Carlo simulation and extreme value theory for similar timeframe signals are appropriate. It enables direct comparison on a similar basis between simulations and experiments with a large difference in duration. By looking at rainflow diagrams and frequency spectra, strengths and weaknesses of the combined fluid/mechanical setup have been identified. Large amplitudes of low-frequency events can be well captured and converted into loads and mechanical response for most of the SG locations. However, higher frequency content is not present in the response, likely due to the load limitation (limits of the CFD modelling) and the time-resolution used. Discrepancies can also be attributed to the added-mass effect that was neglected, thus affecting the mode shapes and their frequencies under stochastic load.

A major disadvantage of the proposed method is the computational cost associated with it. In fact, many of the uncertainties listed in this paper could be circumvented with more computational resources. However, this might not be the optimal solution for the time being. A 2-way FSI approach would just be more expensive for unknown benefits. The temporal setup leads to high requirements both in time and memory (disk storage and active memory). This paper also presented some comparisons with simpler quasi-static models. The quasi-static approach offers a really low-cost alternative when it can be applied, but it has been shown that, as a minimum, it requires the 360° load and the complete runner to obtain satisfying results. It also remains dependent on the indirect choice of largest load cycle.

An alternative approach using 1.5 dimensional coupling, including the added mass effect of surrounding water on runner natural modes was used by [4]. It was demonstrated that it could predict the measured strains with much better accuracy. The description of this 1.5D coupling approach is beyond the scope of this paper. However, it is clear that further development should look into that direction.

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