Dynamic stresses in a Francis model turbine at deep part load

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Abstract. A comparison between numerically obtained dynamic stresses in a Francis model turbine at deep part load with experimental ones is presented. Due to the change in the electrical power mix to more content of new renewable energy sources, Francis turbines are forced to operate at deep part load in order to compensate stochastic nature of wind and solar power and to ensure grid stability. For the extension of the operating range towards deep part load improved understanding of the harsh flow conditions and their impact on material fatigue of hydraulic components is required in order to ensure long life time of the power unit.

In this paper pressure loads on a model turbine runner from unsteady two-phase computational fluid dynamics simulation at deep part load are used for calculation of mechanical stresses by finite element analysis. Therewith, stress distribution over time is determined. Since only few runner rotations are simulated due to enormous numerical cost, more effort has to be spent to evaluation procedure in order to obtain objective results.

By comparing the numerical results with measured strains accuracy of the whole simulation procedure is verified.

1. Introduction
More and more new renewable energy sources like wind and solar power contribute to the European energy production. Since a certain amount of stochastic is attached to these energy sources highly variable operation of common power plants is required to compensate their fluctuations and to preserve the electrical grid stability. For this task hydraulic turbines and pump-turbines are especially suited. Therefore, they are no longer used only for electrical power generation at best efficiency condition but also for stand in if electrical energy from wind and solar drops spontaneously. As a consequence, the operating range of hydraulic turbines is extended towards deep part load. However, there are drawbacks to overcome. Since the turbine is operated further away from the design point in off-design conditions the fluid flow becomes more complex resulting in vortices with local pressure oscillations and cavitation phenomena, which can be harmful to the runner structure and lead to material fatigue. Therefore, it is essential to predict the loads on the turbine runner and the stresses for a fatigue assessment during design phase. Therewith, long operation time without material failure and time effort for repairs is ensured.

Recently, a model Francis turbine was equipped with strain gauges and measurements were performed for various operating points at the EPFL Lausanne [1]. Among other topics the influence of deep part load (DPL) operation on runner stress was investigated.
According to this measurement the loading pressures at one DPL operating point \((Q_{ed} = 0.05445, n_{ed} = 0.2881, \sigma = 0.11)\) on the turbine structure was determined by computational fluid dynamics (CFD) simulation \([2]\). Herein, the model includes the whole water passage from spiral case inlet to draft tube outlet. Since cavitation was expected at the operating point under investigation two-phase modeling was performed.

The pressures distributions of 2000 time steps at a sampling rate of 5334 Hz are mapped onto the structural model for further processing. Finite element analysis (FEA) is carried out for calculation of strains and stresses. Thereafter, time history of the strains and stresses especially at the locations of strain gauges in the experiment are determined. Since only numerical data for five runner rotations is available, comparison in the time domain is difficult. Therefore, numerical results will be transferred to frequency domain in order to obtain comparable information.

In the following, numerical setup for stress analysis, evaluation procedure as well as comparison between numerical results and measured strains will be shown.

2. Numerical stress analysis

A finite element model of the entire turbine runner (model scale) was generated, see figure 1. It was meshed with 1.57 million quadratic tetrahedron elements resulting in 2.70 million nodes. At the flange connection it is supported in axial and circumferential direction. On crown and band areas as well as runner blades pressures from CFD simulation are mapped. In the runner side chambers, which were not included in the fluid flow simulation, pressure is prescribed according to general distribution between pressure at vaneless space and draft tube considering the seal location. Modal analysis of the runner in air and water shows that the natural frequencies are much higher than the frequency range of interest. Therefore, a series of static computations is carried out to determine a pseudo-time series of the stresses.

3. Numerical and experimental results

From the results of the finite element analysis directional strain at locations of strain gauges (see Fig. 2) in their orientation direction is extracted for all blades. The strains are scaled to stresses by Young’s modulus in order to get more descriptive results. In the same way the strains of the test rig measurement are scaled.

Since the measurement data are slightly affected by the measurement system, frequency domain is chosen instead of time domain. In order to compare the data of 5 simulated runner rotations with the data of 800 measured runner rotations equivalent frequency spectrums have to be determined. These are finally used for the comparison of calculated and measured strains.
3.1. Numerical results

In figures 3 and 5 an excerpt of numerically obtained time series of scaled strains at pressure side and suction side at crown are shown. Here, the dynamics of rotor stator interaction (RSI) is clearly observable. Beside this, time signals show some irregular behavior, which differs also from blade to blade.

![Figure 3. Time history at pressure side close to crown](image)

![Figure 4. Frequency spectrum of figure 3](image)

![Figure 5. Time history at suction side close to crown](image)

![Figure 6. Frequency spectrum of figure 5](image)

In figures 4 and 6 the corresponding frequency spectrums are shown. Here, a good agreement for the stress amplitude at gate passing frequency is observed for different blades. For the range of 0 to 10 $f_n$ the general behavior seems similar, but locally there are large differences in the stress amplitude at one frequency. This characteristic arises from the fact that only a time series of 5 runner rotations is used for the Fourier transform, which is quite short. Therefore, an analysis to get objective values is performed in the next section.

3.2. Equivalent frequency spectrum

For the determination of frequency spectra that are comparable between numerical and measured data, one experimental data set of a strain gauge at the crown is analyzed in detail.

In figure 7 the frequency spectrums of 1875 samples that correspond to 5 rotations, of 30000 samples (80 rotations) and of the whole time signal are illustrated. Due to the different lengths of the time signals the frequency resolution of the spectrums is also different. The stress amplitudes that are related to a discrete excitation (e.g. RSI at 20 $f_n$) are comparable for all spectrums. For the whole frequency range, the amplitudes depend on the resolution of the spectrum, because stress amplitude of the coarsest resolution is distributed on different frequencies for the finer resolutions due to the stochastic nature of the stresses. Therefore,
all spectrums are brought to the same resolution by computing the energetic sum within each frequency range. Thereby, the spectrums of 80 runner rotations and 800 rotations are almost identical, see figure 8. However, the spectrum with the lowest number of samples still differ. In figure 9 a total of 16 spectrums with 1875 samples at different times of the measurement data are shown. Additionally, upper (Max) and lower (Min) bound as well as the root mean square value (Mean) is shown. It can be observed, that the spread of the amplitudes is quite high. Therefore, one single data set of 5 runner rotations is too short to be representative for the entire dynamic. If one compares the root mean square value of 16 data sets with the frequency spectrums of longer time histories as shown in figure 10 a very good agreement between all data is observed. Therefore, the calculated stresses at the strain gauge locations of the 16 runner blades are used for the comparison with the experimental data. Here, an average frequency spectrum is determined. From measurements, the resolution of the frequency spectrum is coarsened to obtain same resolution as from simulation.

3.3. Comparison of numerical with experimental results

In figures 11 and 12 the frequency spectrums of the stresses at the pressure and suction side in the vicinity of the crown are shown. In detail, strain gauges 05 and 06 are analyzed, cf. figure 2. In figure 13 the comparison for the strain gauge 12 at the suction side close to the band is shown. Unfortunately, all strain gauges at the pressure side were broken before this test. Overall, a good agreement between experimental and numerical results can be observed in the frequency range of interest which is up to 10 times \( f_n \). For higher frequencies low strain amplitudes are a result of low pressure oscillations that have already been observed in the CFD
simulation, see [2]. The difference in the RSI-stresses is not relevant, since it is out of scope in this examination. This phenomenon is captured well by other simulation approaches [3].

For the frequency $f_n$ there are some interferences from the measurement system that affect the measured amplitudes at this frequency. Also, the first harmonics are slightly influenced, as can also be seen from green line in figure 7.

4. Conclusion
A comparison of numerical obtained stresses in a Francis model turbine at DPL operation with experimental ones has been performed in frequency domain. Average frequency spectrums for all blades were used for the numerical data. The frequency spectrums of the measurement data were coarsened to the resolution of numerical ones in order to obtain objective spectrums.

A good agreement is observed for the numerical results at pressure side crown and suction side band between $f_n$ and $10f_n$. At the suction side near crown the amplitudes are slightly underestimated. The deviations above $10f_n$ are linked to the low numerical pressure oscillations from computational fluid flow simulation. Below $f_n$ longer numerical time series are necessary for more reliable amplitudes.

Acknowledgments
The research leading to the results published in this paper is part of the HYPERBOLE research project, granted by the European Commission (ERC/FP7-ENERGY-2013-1-Grant 608532)

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Figure 11. Measured and calculated strains at pressure side near crown

Figure 12. Measured and calculated strains at suction side near crown

Figure 13. Measured and calculated strains at suction side near band.