Model test with sensor equipped Francis runner for Part Load Operation

Florian von Locquenghien¹, Peter Faigle¹, Thomas Aschenbrenner¹

¹Voith Hydro Holding GmbH & Co. KG, Alexanderstraße 11, 89522 Heidenheim, Germany

E-Mail: Florian.Locquenghien@voith.com

Abstract. With the need for flexible operation Francis runners are exposed to various operating conditions outside the traditional operating range. These machines have to be designed for long time Part Load Operation to meet hydraulic and structural requirements. Deep Part Load Operation mainly consists of stochastic loads on the blades. Calculation methods for the dynamic stress at Full Load Operation are well established, whereas the calculation of the dynamics during low load operation are in the focus of the current research to increase the prediction accuracy. Through a model test of a Francis runner with multiple sensors in the rotating frame, knowledge is gained about low load operating conditions. The sensors consist of both pressure transducers and strain gauges. On the basis of the results for multiple operating conditions a comparison between pressure sensors and strain gauges is made and connected to flow phenomena. Prototype strain gauge data is utilized to prove the feasibility of the data gained in model size.

1. Introduction

In the past decade hydro power has been subjected to reviews as solar and wind power developments have been spreading substantially and stepping into competition. Grid stability by sufficient control and reserve of power is a major demand on hydro power [1].

Due to the volatile power received from solar and wind power plants, the energy market requires highly variable hydro power machines and fast changes of operating regimes. High numbers of Start-ups, Shut-downs and operation beyond traditional guaranteed operating ranges can have significant influence on the fatigue life of hydro power unit components. Stress is a result of both geometry and load induced by flow. Especially peak stress is dependent on local geometry (e.g. notches). A robust design is defined through a runner that withstands the complex load spectrum for a given lifetime.

To realize 0-100 % operation the static and dynamic stresses for multiple conditions need to be captured during the design. Different operating conditions can reduce the fatigue life of the machine as operation cycles [2], dynamics due to RSI [2] and especially long time Part Load Operation. Since simulation methods for the stochastic behavior of Part Load Operation are still part of current research and development, different approaches have to be utilized. Prototype strain gauge measurements deliver the most reliable results in terms of fatigue, however, can only be conducted after a machine is build. Hence it is desirable to predict principal loads on basis of an advanced model test. Furthermore, with this approach phenomena can be investigated and influences of design studied.

A model test with multiple sensors in the stationary and rotating frame of a model Francis runner has been performed. This includes both pressure sensors and strain gauges on the runner blades. With existing measurements of the prototype machine the feasibility of the results can be proven.
2. Operating Conditions
The flow phenomena in a Francis turbine can be separated by operating zones dependent on flow. They have impact on the pressure fluctuations and therefore the load on the runner. Stable flow conditions around the Best Efficiency Point (BEP) result in low dynamic loads. The vortex ropes can be distinguished between full load and Part Load vortex rope. Both are mainly visible in the draft tube cone (compare Figure 1). Interblade vortices appear for lower discharge. These are vortices that originate in the crown region of each channel and extend past the discharge edge of the blade into the draft tube cone (channel vortex) [4]. The visibility is dependent on cavitating volume and therefore the tailwater pressure. A phenomenon that is more related to system dynamics is the Higher Part Load (HPL). Higher frequency pressure fluctuations, introduced by the rotating vortex rope, lead to a self-excitation of the system. Due to its driving physical quantities, the impact cannot directly be transposed from model to prototype. The phenomena are explained in more detail in [5].

3. Model Test Setup
The proof of hydraulic parameters through a model test is well accepted as most of the parameters are transposable by hydraulic similarity laws [6]. Therefore, with similar flow conditions the load on the blades is comparable to the prototype. The major challenge is the transfer of the structural behavior of a model runner, as eigenfrequencies and stiffness do not scale accordingly.

Since the joints of a sectored model runner would lead to an undefined stiffness, a “monoblock” runner is mandatory for a sensor equipped model test to grant a defined stiffness and a geometry that can be investigated through simulation models (Figure 2). The runner material is congruent to the prototype runner material (CA6NM). The model runner has been manufactured according to the standards used for hydraulic model testing with very high accuracy. Due to the sensor application, the crown and band sections of the model runner show deviations in comparison to the prototype, however, these deviations have been accounted for in the simulation models. The hydraulic faces are not influenced by the sensor application.
In total 30 sensors have been installed in the rotating system with 16 pressure transducers and 12 strain gauges on multiple blades. Accessibility and limitations through blade thickness is a major challenge and a process to securely apply sensors had to be developed. The sensors are partly buried in the blade to not influence the flow. Observations did not show any influence e.g. cavitation due to the sensors. It was chosen to introduce pressure sensors on both sides of one blade to see the load distribution on one blade. The sensors for strain gauge (SG) testing were spread across three blades close to crown and band, at locations of the expected highest dynamic stresses. The locations were defined through a FEA which has been already proven through prototype measurement. Additionally standard pressure sensors were installed in the stationary system such as Draft Tube Cone (DTC) or Vaneless Space (VS) at IEC conform positions. Figure 3 shows the sensor allocation. The odd numbers on the blade refer to the sensors on the suction side and the even ones to the pressure side.

The runner design is of a mid range specific speed (nq=40-60) machine with available prototype measurements. The model hydraulic contour is the same, whereas fillets, mechanical crown and band shape are those of a standard model machine. The test rig utilized for the investigations is a standard test rig for hydraulic model testing. A large test program was performed with multiple operating conditions. The data presented refers to the conditions in accordance with an available prototype measurement. Since the behavior in the runner is of most interest, the reference elevation for the Thoma number is set to the outlet edge of the runner [6].

4. Model Test Results

Figure 4 shows the characteristic peak-to-peak (char. p-p) results for the sensors in the rotating frame and additionally the sensor DTC and VS. The results of the pressure transducers are normalized to head and the strain is normalized to the strain of SG 2 during Normal Operation. The flow is normalized to the Best Efficiency Point at the specific head. It has to be noted that very sensitive sensors were used which were known to have a high risk of failure, but were needed for thickness and placement requirements. The sensors P4 and P5 were lost after some testing, therefore no results for both sensors are plotted.

For all sensors the flow phenomena at Deep Part Load, Part Load, Normal Operation and Full Load can be identified based on visual observations and frequency analysis. The phenomena overlay to some extent.

**Figure 2.** “Monoblock” model runner with sensors.

**Figure 3.** Sketch of sensor allocation for pressure transducers and strain gauges.
Figure 4. Char p-p of rotating and stationary pressure / strain sensors (odd numbers suction side/even numbers pressure side).
The pressure transducers on the blades show in general similar characteristics as the sensors in the stationary system. The pressure fluctuations from Higher Part Load have equal amplitudes on both pressure and suction side. Since the strain gauges are integral indicators of the load on the blade, the described characteristic of the HPL adds no dynamic load to the blade and therefore are not present in the strain gauge signal. The strain gauge at the band (SG2) has a good correlation to the pressure fluctuation in the Draft Tube Cone and the strain gauge at the crown (SG1) shows a similar characteristic to the pressure fluctuation in the Vaneless Space.

**Figure 5.** FFT of signals from model testing for different Part Load Operation normalized to the nominal runner speed (A,B,C according Figure 4).
Three different flow regimes were analyzed in more detail. Position A refers to a mainly stochastic flow for all sensors (Figure 5). The pressure transducers towards the band show higher amplitudes as the channel vortex originates upstream close to the crown and leaves the runner near the band. Since the channel vortex is present in each individual flow passage, the load on both sides of the blade are effected. This matches the visual observations. A broad band of stochastics leads to increased char. p-p values.

Position B refers to a region where both a developed Part Load vortex rope and a channel vortex can be observed. The Fast Fourier Transformation (FFT) shows that the Rheingans frequency and higher harmonics are present, both in the rotating and stationary frame. Additionally a stochastic band is seen. The stochastic load ($>2 \times f_{Ra}$) defines 82% of the dynamic strain for SG2. Observations have shown that at this operating condition a “dancing” channel vortex is present, whereas, for lower loads the vortex is more stable. With the movement of the vortex it is likely to hit a pressure sensor directly and therefore induces high changes in pressure. Two examples for this are the pressures sensors at P13 and P15, which show very high char. p-p values. Since the pressure sensors only record the very local pressure the high values in the char. p-p are not alarming. The strain signals do not show increased values, as it is the answer to the load on the entire blade.

Position C refers to an operating condition where the Part Load vortex rope is dominant and the channel vortex only starts to develop. Due to the impact region of the vortex rope, it is dominant at SG1.

5. Comparison to Prototype

For the comparison between model and prototype strain gauge measurements by statistical values (e.g. char. p-p) the sampling rate and signal length have to be adapted. Since a model test is under laboratory conditions, a much higher sampling rate and relative recording length can be realized (compare Table 1). The data from the model test was post processed with a down sampling to have the same number of samples per revolution and cut for the length of identical number of revolutions.

| Table 1. Signal Processing of prototype and model |
|-----------------------------------------------|
|                              | Prototype | Model         |
| Sampling Rate | 2048 Hz | 30 000 Hz      |
| Speed          | n       | 9.68 x n       |

For both the model runner and the prototype runner the eigenfrequencies were determined. For the model runner an experimental modal analysis has been performed. A detailed description can be found in [7]. For the prototype runner a modal analysis in water is available. The eigenfrequencies of interest are represented by vertical lines in Figure 6 (higher eigenfrequencies exist, but are not visualized). Due to an increased stiffness the eigenfrequencies of a model runner shift to higher frequencies compared to the change in the nominal speed from prototype to model.

For SG 2 and the corresponding prototype position a detailed FFT has been performed for the operating condition A (DPL). Figure 6 shows the results normalized to the individual nominal runner speed. Both show a typical stochastic behavior, however, they show a different characteristic. Evidently a region with higher amplitudes is shifted between model and prototype. Additionally multiple regions of higher order frequencies are present with lower amplitudes. Apparently the first region of amplification match an accumulation of eigenfrequencies.

Figure 7 shows the char. p-p values for both the model and prototype test. The characteristic values have a probability of 97% and are normalized to the model value at $Q/Q_{BEP} = 14\%$. Each tested operating condition refers to one data point. For the model test a lot more operating conditions could be tested, since no limitations in time or grid requirements were present. The values at the position of the maximum stress of the prototype are approximately seven times larger than those at the strain gauge location of the model. Through scaling the model values by this factor, a qualitative comparison of the data can be performed. Both campaigns show a good correlation for the Part Load range. Towards the BEP the data differs. This difference was expected, as Rotor-Stator-Interaction is the dominant phenomena in this
operating range and the prototypes’ eigenfrequencies are closer to the gate passing frequency than those for the model runner. For the Part Load operating conditions the frequency characteristic change with operating condition, however, a constant scaling factor is reasonable according to the results. The differences between model and prototype frequencies does not influence the char. p-p values, otherwise different behavior with increasing flow would be expected. Through repeating tests the reproducibility was proven. Minor differences can be explained based on the stochastic character of the time signal.

![Figure 6. FFT results of both measurement and simulation for model (top) and prototype (bottom) for a strain gauge in comparison to the eigenfrequencies.](image)

A second comparison is made based on a Rainflow counting of the time signals, as this is of major interest for a fatigue assessment. Figure 8 shows the results for $Q/Q_{BEP} = 28\%$ with dominant stochastic load and $Q/Q_{BEP} = 62\%$ with both stochastic load and part load vortex rope. Each signal has been normalized to its individual char. p-p value. For both model and prototype identical number of revolutions were used. For the model test results two adjacent time frames were chosen resulting in $t_1$ and $t_2$. The Rainflow counting has been performed with a constant class size, resulting in >100 comparable classes for each time signal.

Both operating conditions show good agreement between prototype and model data. For the highest amplitudes with < 10 cycles a discrepancy is present, however, the influence on the chosen time frame can be seen comparing $t_1$ and $t_2$. In the lower diagram of Figure 8 both operating conditions from the prototype are plotted. Although it is a different operating condition and normalized to a different char. p-p value, they show a good correlation for amplitudes >10 cycles. This indicates that the composition of the stochastic load is very similar, independent of the operating condition. Additionally the different frequency characteristic between model and prototype has only a minor impact on the Rainflow counting as basis for a fatigue calculation.

To use a strain gauge measurement from a model test for the prediction of the fatigue life of a prototype machine, e.g. before the actual manufacturing, a scaling has to be established. As seen in Figure 7, a constant scaling for stochastic load seems reasonable based on this measurement. This scaling factor can be derived through finite element calculation of both runners, either based on the pressure distributions derived from transient simulations or an artificial pressure distribution that represents the stress pattern of the stochastic load. Both can further be used for the extrapolation from the strain gauge to the fatigue hot spot. Differences in the local geometry (e.g. crown, band, fillet size) are included in the FEA model, hence the influence on the maximum local peak stress is determined.
Figure 7. Comparison of strain at band between model and prototype.

Figure 8. Results of Rainflow counting for two operating conditions normalized to the individual char. p-p of each signal with >100 Rainflow classes and identical number of revolutions.
6. Summary
A model test with multiple sensors on the runner has been performed. At Part Load Operation the stochastic pressure load on the blade is dominant and can be seen on all sensors. For the pressure sensors in the stationary system and the strain gauge sensors similar behavior for char. p-p was found. Whereas the strain gauge results are the reaction to the integral load, the pressure sensors in the DTC relate to more averaged flow from the blade passages. The pressure sensors on the blade show the local pressure and for Part Load Operation the dynamics can be clearly related to the channel vortex.

For the investigated runner design both the char. p-p values and the Rainflow counting of the prototype and model measurements show qualitative matching results. It is feasible to use strain gauge model test data to predict the fatigue life once a scaling method is established. Neither the char. p-p behavior over flow nor the Rainflow matrix are significantly impacted by the differences in the frequency characteristic. This will need deeper investigation to be accurately addressed in a scaling process.

A robust design is not only dependent on the results of local pressure fluctuations, but even more on the geometry itself. Local peak stress occurs due to notch mechanisms in the transitions from blade to crown or band, hence, they are highly dependent on the structural design. A robust design with regard to fatigue life can already be accomplished through an intelligent structural design that can withstand the loads.

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