Influence of Hydraulic Design on Stability and on Pressure Pulsations in Francis Turbines at Overload, Part Load and Deep Part Load based on Numerical Simulations and Experimental Model Test Results

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Abstract. Francis turbines have been running more and more frequently in part load conditions, in order to satisfy the new market requirements for more dynamic and flexible energy generation, ancillary services and grid regulation. The turbines should be able to be operated for longer durations with flows below the optimum point, going from part load to deep part load and even speed-no-load. These operating conditions are characterised by important unsteady flow phenomena taking place at the draft tube cone and in the runner channels, in the respective cases of part load and deep part load. The current expectations are that new Francis turbines present appropriate hydraulic stability and moderate pressure pulsations at overload, part load, deep part load and speed-no-load with high efficiency levels at normal operating range. This study presents series of investigations performed by Voith Hydro with the objective to improve the hydraulic stability of Francis turbines at overload, part load and deep part load, reduce pressure pulsations and enlarge the know-how about the transient fluid flow through the turbine at these challenging conditions. Model test measurements showed that distinct runner designs were able to influence the pressure pulsation level in the machine. Extensive experimental investigations focused on the runner deflector geometry, on runner features and how they could reduce the pressure oscillation level. The impact of design variants and machine configurations on the vortex rope at the draft tube cone at overload and part load and on the runner channel vortex at deep part load were experimentally observed and evaluated based on the measured pressure pulsation amplitudes. Numerical investigations were employed for improving the understanding of such dynamic fluid flow effects. As example for the design and experimental investigations, model test observations and pressure pulsation curves for Francis machines in mid specific speed range, around \( n_{opt} = 50 \text{ min}^{-1} \), are reported, analysed and commented here. The analysis of experimental results allowed the identification of designs and configurations, which can improve the machine stability.

1. Introduction
Dynamic fluid flow phenomena, as e.g. full load instability, rotor-stator-interaction, draft tube instabilities and channel vortex, are responsible for pressure oscillations in hydraulic turbines. The level of the pressure pulsation amplitude is commonly associated to the hydraulic stability of Francis turbines. Common requirements from hydraulic energy producers are that the pressure pulsations amplitude remains below predefined limits.
Depending on the operating point and specific speed of Francis turbines, different dynamic fluid flow effects may take place in the machine. At overload, full load instability may arise in very special cases. At full load operation, the rotor-stator-interaction may become important, especially at low specific speed machines. The draft tube instabilities at part load are associated with the vortex rope rotation in the turbine draft tube and sometimes higher part load effects might be present. Channel vortices in the turbine runner can appear at deep part load and can also lead to pressure pulsations in the turbine.

The regions in the turbine hill chart, where the different dynamic fluid flow phenomena normally take place, can be seen as example in Figure 1. Depending on the turbine volume flow, each of the dynamic effects becomes dominating.

As mentioned above, the dynamic fluid flow through the hydraulic machine is responsible for the pressure oscillations. Typical amplitudes of the pressure pulsations, $\Delta H/H$, are schematically presented in Figure 2 and associated to the different flow phenomena. The guide vane opening, $\Delta \gamma$, can be related to the fluid flow. Some dynamic effects, as higher part load and full load instability, could be especially challenging for the machine smooth operation, but they are not directly tranposable from the model to the prototype and very seldom present at the prototype. Deep part load, where the runner channel vortex might be encountered, is also at the origin of typically high pressure oscillation amplitudes.

The several dynamic effects in Francis turbines are associated with typical flow patterns at the turbine runner, which can be observed during the machine model test and to a certain extent even be simulated and visualised with modern computational fluid dynamic (CFD) techniques. Typical fluid flow patterns can be found in Figure 3, where pictures of the cavitating runner
outlet flow could be seen. Depending on the driving phenomenon, different flow characteristics are observed.

The numerical simulation and experimental measurements are important tools for the optimisation of the pressure pulsation level in hydraulic machines. Francis turbines in the mid specific speed range offer an interesting example for this kind of research, as long as most of these dynamic effects are present even if in very moderated levels.

2. Numerical Simulation

The numerical simulation of the fluid flow through the hydraulic machine is an important tool in the design and optimisation of Francis turbines. In modern turbine design, this is an essential part in the optimisation process in regard to efficiency and cavitation safety. Before a promising turbine design is released for model test, up to even hundred geometrical variants are investigated using computational fluid dynamics (CFD) simulations of the stationary fluid flow.

Full load instability, vortex rope rotation, higher part load, channel vortex and s-shape instabilities are strongly unsteady phenomena, which are at the origin of pressure pulsations in the hydraulic machine. Recent progresses in the numerical simulation techniques made it possible to accurately predict the unsteady fluid flow in hydraulic turbines. Some examples, are the numerical prediction of pressure pulsations in Francis turbines at full load, part load and deep part load performed by Magnoli [1], the part load calculation by Krappel et al. [2], the numerical simulation of higher part load in the model machine by Magnoli and Schilling [3], the CFD calculations of rotating stall and s-shape instability done by Hasmatuchi and Avellan [4] or the rotor-stator-interaction calculation from Yan et al. [5].
Even if CFD simulations of these dynamic effects are currently possible, today they are still extremely computationally costly calculations, requiring extremely long times for the calculation of even single operating points. For this reason, nowadays they still cannot be integrated in the iterative design of hydraulic turbines and several investigations regarding pressure pulsations are performed experimentally. However, the numerical simulation of the transient effects taking place in the Francis turbines can contribute to increase the know-how and understanding of these phenomena and they can also deliver new insight in the design of Francis machines to minimise pressure pulsations.

For example, in the development of hydraulic turbines in the mid specific range with \( n_{q_{\text{opt}}} \approx 50 \text{ min}^{-1} \), several numerical simulations were performed, with the objective of improve the machine pressure pulsation behaviour at deep part load with extremely reduced volume flow. At this operating condition, the channel vortex becomes present in the turbine runner and the fluid flow is highly unsteady. Figures 4 and 5 bring an example of the simulation of such conditions. The streamlines showing the channel vortex rotation around itself can be seen in Figure 4, while magnitude of the vortex core is visualised in Figure 5 with the characterisation method of Jeong and Hussain [6].

The computational analysis and visualisation of the channel vortex at deep part load provided additional understanding of this effect and its relation with the runner geometry. The acquired know-how could be used for the model tests of new turbine geometry, which could reduce the pressure pulsation level at deep part load, through the reduction of the size and intensity of the channel vortex. The comparison of the original design and the modified design could be later on carried out experimentally as seen in the sequence.

The higher part load effect and the full load instability could already be numerically simulated as done by Magnoli and Schilling [3] for the former and by Alligné et al. [7] and by Mössinger, Conrad and Jung [8] for the latter. Nevertheless, the quantitative computational prediction of
the pressure pulsations generated by these both phenomena still has to be improved. The reasons are the challenge in modelling the compressible cavitating fluid flow in the draft tube cone and the influence of the plant Thoma coefficient, $\sigma_{pl}$, and of the power house hydraulic circuit. In both cases the experimental investigation during the model test is still the more accurate alternative for industrial applications. Nevertheless, no exaggerated importance should be given to these two effects, as long as they are not directly transposable from the model to the prototype and their occurrence in the prototype is extremely rare.

3. Experimental Investigation
The improvements in the hydraulic design of Francis turbines in the mid specific speed range, $n_{q_{opt}} \approx 50 \text{ min}^{-1}$, could be verified with experimental model tests of different runner geometries. The effect of these modifications could be identified especially at the part load and deep part load operating conditions. The influence of different design alternatives on the full load instability was experimentally observed and measured with the usage of different runner deflectors.

The reduction of the pressure pulsation level from the modified hydraulic design in comparison to the original one can be seen in selected charts in Figure 6 and 7. The characteristic amplitude peak-to-peak of the pressure pulsations, $\Delta H/H$, at the spiral case inlet and the draft tube cone as function of the guide vane opening, $\Delta \gamma$, for selected net heads are shown in the diagrams.

The knowledge about the deep part load and part load behaviours acquired during the numerical simulations, offered the possibility to modify the runner meridian contour, the runner outlet flow and the flow in the draft tube cone. These modifications could reduce the pressure pulsation level in the draft tube cone at deep part load and part load, as observed in Figure 7. At individual operating points, the pressure pulsation characteristic amplitude could be reduced by over 25%.
In addition, the changes in the hydraulic design were able to reduce the small indications of higher part load (HPL) present in the original model design, in spite of its limited importance for the prototype. The local peak in the values of the pressure pulsations between $\Delta \gamma \approx 17.0^\circ$ and $\Delta \gamma \approx 18.0^\circ$ is characteristic of higher part load (HPL). They could be found in the complete machine, at the measuring points in the spiral case and in the draft tube cone, as depicted in Figures 6 and 9. With the modified design, the higher part load effect was eliminated.

Figures 8 and 9 show the pressure pulsation level in the vaneless space for the original and the modified runner geometry. The improvements in the pressure pulsation level could be experimentally observed mainly at the deep part load portion of the hill chart, where the channel vortex is the dominating unsteady phenomenon in the fluid flow through the hydraulic turbine. This reduction was obtained with changes in the blade inlet geometry, with the objective to reduce the size and intensity of the channel vortex. The changes were suggested by the numerical simulation of the runner operation with CFD and could be confirmed during the model test. The pressure pulsation characteristic amplitude in the vaneless space could be reduced by over 30% at some operating points at deep part load.

Another investigation carried out with help of model tests concerned the full load instability, which very seldom may take place at overload operating conditions. As discussed before, the numerical simulations of this dynamic effect require large computational resources and times with limited accuracy when compared to model test results. During the experimental measurements,
different deflector geometries can be very quickly tested and their effectiveness can be assessed. Figure 10 brings a generic example of runner deflector for a low specific speed machine.

Runner deflectors are commonly used to eliminate the higher part load (HPL) effect, although this effect is mostly present only in the model and not in the prototype, since it is not directly transposable due to differences in the hydraulic circuit and Froude number. Depending on the shape of runner deflectors, their length and diameter, they might be more or less effective. However, their geometry may in few cases lead to the appearance of full load instability in the model. This effect has been experimentally investigated. Three runner deflectors with different shape, length and diameter were tested with the same hydraulic model machine. The results can be found in Figure 11.

Deflectors 2 and 3 were responsible for introducing full load instability in the model machine. They caused punctual increases of $\Delta H/H$ in the draft tube cone by around 80%. On the other hand, Deflector 1 counted with a special geometry, which could eliminate any trace of full load instability. At the same time it was also effective against the appearance of higher part load. These are effects, which may arise in the model machine. As already mentioned, they are not directly transposable to the prototype and extremely seldom appear.

4. Conclusion
The pressure pulsations are of great importance for the operational stability of Francis turbines, especially at off-design operating points. Operating conditions at deep part load, part load and overload are gradually becoming more important in the energy generation market. At these conditions, it is expected that the hydraulic turbine still offers smooth operation as far as possible. In this context, the numerical prediction and experimental investigations are important elements for the optimisation of the turbine behaviour.

Modern computational simulation techniques offer the possibility to better understand the dynamic phenomena being responsible for the pressure oscillations. Even though if these methods still cannot be integrated in the iterative optimisation process, they could contribute with additional knowledge. It made possible to optimise the dynamic behaviour of Francis turbines in the mid specific speed range, around $n_{opt} = 50 \text{ min}^{-1}$.

The experimental measurements during the model tests could confirm that the modifications in the hydraulic design could bring improvements related to the pressure pulsation level. Different runner meridian contour geometry, runner outflow velocity distribution and blade inlet
Figure 11. Pressure pulsation characteristic amplitude peak-to-peak at rated head at the draft tube cone headwater side.

draft tube cone headwater side.

genera...ect amplitude in the vaneless space and draft tube cone.

The combined iterative optimisation of the runner geometry with CFD simulations, base research of the dynamic fluid flow effects in the hydraulic turbine and the experimental investigation during the model tests are important tools, which were used for the increase of the hydraulic stability of Francis turbines.

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