Comparison of model measured runner blade pressure fluctuations with unsteady flow analysis predictions

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Abstract. An accurate prediction of pressure fluctuations in Francis turbines has become more and more important over the last years, due to the continuously increasing requirements of wide operating range capability. Depending on the machine operator, Francis turbines are operated at full load, part load, deep part load and speed-no-load. Each of these operating conditions is associated with different flow phenomena and pressure fluctuation levels. The better understanding of the pressure fluctuation phenomena and the more accurate prediction of their amplitude along the hydraulic surfaces can significantly contribute to improve the hydraulic and mechanical design of Francis turbines, their hydraulic stability and their reliability.

With the objective to acquire a deeper knowledge about the pressure fluctuation characteristics in Francis turbines and to improve the accuracy of numerical simulation methods used for the prediction of the dynamic fluid flow through the turbine, pressure fluctuations were experimentally measured in a mid specific speed model machine. The turbine runner of a model machine with specific speed around $n_{q,\text{opt}} = 60 \text{ min}^{-1}$, was instrumented with dynamic pressure transducers at the runner blades. The model machine shaft was equipped with a telemetry system able to transmit the measured pressure values to the data acquisition system. The transient pressure signal was measured at multiple locations on the blade and at several operating conditions. The stored time signal was also evaluated in terms of characteristic amplitude and dominating frequency.

The dynamic fluid flow through the hydraulic turbine was numerically simulated with computational fluid dynamics (CFD) for selected operating points. Among others, operating points at full load, part load and deep part load were calculated. For the fluid flow numerical simulations more advanced turbulence models were used, such as the detached eddy simulation (DES) and scale adaptive simulation (SAS). At the different operating conditions, distinct flow phenomena were available for evaluating the accuracy of the simulation method, e.g. rotor-stator interaction, rotating vortex rope and runner channel vortex. The experimentally obtained pressure time signals at the runner, characteristic amplitude and frequency at several operating points offered the possibility to assess the precision in the prediction of the pressure fluctuations in the Francis turbine using the numerical simulation methods.

1. Introduction
There is an increasing tendency to operate hydraulic turbines more and more at off-design conditions with the objectives to maximize the owner benefit regarding the energy trading market and to regulate the electrical grid. With the extension of the turbine operating range,
some machines are subjected to full load and part load as well as even deep part load and speed-no-load.

In order to obtain further knowledge about the highly dynamic and complex transient fluid flow through the hydraulic turbine at these operating conditions, transient pressure measurements were performed in the model runner of a Francis turbine. The chosen machine counted with mid specific speed, \( n_{q,\text{opt}} \approx 60 \text{ min}^{-1} \), and was named as FT 60. This type of experimental investigation was already conducted in the past as done by Avellan et al. [1], but in a limited extent and extremely seldom.

The model runner blades were equipped with dynamic pressure transducers and the runner and shaft made use of a telemetry system for transmitting the measured data. Several operating points were tested inside the turbine continuous operating range. The experimental acquired data provided new material for the analysis of several operating conditions, where runner-stator interaction (RSI), rotating draft tube vortex rope (DTI) or runner channel vortices (RCV) were present.

Moreover, the measured data also allowed the evaluation of numerical simulation methods based on computational fluid dynamics (CFD) for the prediction of the dynamic fluid flow through the Francis turbines. The numerical simulations were carried out with advanced hybrid turbulence models such as detached eddy simulation (DES) and scale adaptive simulation (SAS).

The analysis concentrated on the measurements and prediction of the pressure oscillation amplitude at the runner blades as parameter for the continuous improvement of the hydraulic and mechanical design of Francis turbines, their hydraulic stability and their reliability.

2. Experimental Setup

For the FT 60 Francis turbine, transient pressure measurements were performed at the model runner blades and crown. The tests were conducted at the Laboratory for Hydraulic Machines (LMH) of the École Polytechnique Fédérale de Lausanne (EPFL).

The model machine was mounted on one of the universal test rigs, operating in a closed hydraulic system. The model head, \( H_M \) and volume flow, \( Q_M \) were controlled independently for testing any operating point on the model hillchart, \( (n_{1_M}', Q_{1_M}') \). The cavitation coefficient, \( \sigma \), could be set to match the corresponding homologous powerplant conditions.
The model turbine geometry was hydraulic homologous to the prototype machine, including the instrumented runner blades.

A total of 28 dynamic pressure transducers were mounted on the runner blades and band. Four rows from crown to band with three sensors on the blade pressure side (PS) and another three on the blade suction side (SS) were installed along the blade length, going from the trailing edge pressure side to the trailing edge suction side. Additionally four sensors were inserted along the band. The location of the pressure transducers at the blade can be seen in Figure 1. The physical installation of the dynamic pressure transducers at the model runner blades and of the wiring is shown in Figure 2. Due to the mechanical design constraints, not all pressure transducers could be brought in the same blade and therefore were distributed over four blades.

The sensors are numbered from 1 to 28. Their location at the blades is defined by the pair of coordinates \((u, v)\). Both coordinates are normalised from 0 to 1. The \(u\) coordinate starts at the trailing edge on the pressure side, \(u = 0.00\), and runs to the trailing edge on the suction side \(u = 1.00\). The \(v\) coordinate runs from crown to band, going from \(v = 0.00\) to \(v = 1.00\). The transducers were placed at the runner blades at the coordinates, \(u = 0.05, u = 0.25, u = 0.40, u = 0.60, u = 0.75, u = 0.95\) and \(v = 0.20, v = 0.40, v = 0.60, v = 0.80\).

For the data transmission between the dynamic pressure transducers in the rotating runner to the data acquisition system a telemetry system was used. Part of the electronic was mounted inside the runner crown, as seen in Figure 3. The wiring was routed through the inner part of the turbine shaft up to its top, where the transmitting part of telemetry system was mounted, as observed in Figure 4.

The telemetry system was only able to transmit a limited number of channels simultaneously. For this reason, each operated point had to be tested several times in order to obtain the data from all pressure transducers. The signals had to be synchronised afterwards during post-processing.

The pressure transducers were calibrated by immersing the instrumented runner connected to the telemetry system in a tank, where a known and accurate controlled pressure was applied.
Table 1. Selected operating points.

| Operating Point       | $n'_{1,M}/n'_{1,\text{opt}}$ | $Q'_{1,M}/Q'_{1,\text{opt}}$ |
|-----------------------|-------------------------------|-------------------------------|
| 1 Full Load           | 1.107                         | 1.206                         |
| 2 Part Load           | 1.107                         | 0.938                         |
| 3 Deep Part Load      | 1.107                         | 0.769                         |

3. Experimental Results

In total 114 operating points at 6 different project prototype heads inside the continuous operating range were measured. Additionally, 21 points outside the operating range were also tested.

For a short analysis here and comparison to the numerical simulations 3 operating points were selected at the same head, the first one at full load, the second one at part load and the third one at deep part load. The important dynamic fluid flow phenomena at these points should be respectively roto-stator interaction (RSI), rotating draft tube vortex rope (DTI) and runner channel vortex (RCV). Data about these operating points is found in Table 1.

The transient pressure history, $p(t)$, for each pressure transducer was measured and stored at each operating point during 10 s with an acquisition rate of 2400Hz. Figure 5 brings an example of part of the transient pressure history for the pressure pulsation transducer at $u = 0.95, v = 0.80$ at the runner inlet edge near to the band at deep part load. The time, $t$, is normalised to the runner revolution period, $T_n$.

The pressure pulsation data was post-processed with focus on the pressure oscillation amplitude and frequency. The pressure oscillation amplitude was evaluated in terms of the peak-to-peak characteristic amplitude with a confidence factor of 97%, in order to eliminate spurious external influences on the model machine. The pressure oscillation amplitude is expressed in percent as $\Delta H/H$, being equal to $\Delta P/(\rho g H)$, where $\Delta P$ is the peak-to-peak characteristic amplitude and $H$ the tested head. The presented results were normalised as $(\Delta H/H)^* = (\Delta H/H) / (\Delta H/H)_{\text{max}}$, where $(\Delta H/H)_{\text{max}}$ was defined as the highest measured peak-to-peak pressure oscillation amplitude considering all measuring locations and over the complete hill chart, even beyond the normal operating range.

The frequency spectrum of the transient pressure history could be obtained with the Fourier transform and the dominating frequencies are normalised and denoted by $f_i/f_n$, $i = 1, 2, 3, \ldots$, where $f_n$ is the runner rotating frequency. Figure 6 shows the measured pressure pulsation frequency spectrum for the sensor at $u = 0.95, v = 0.80$ at deep part load.

The analysis of the measured data confirmed that at full load, part load and deep part load the dominant effects in the transient fluid flow were respectively the rotor-stator interaction, rotating draft tube vortex rope and runner channel vortex.

At full load the higher pressure oscillation amplitudes were found near to blade inlet and decayed in direction to the trailing edge. The pressure side experienced more dynamic pressure load than the suction side. This behaviour was caused by the kinematic interaction between the rotating runner and the stationary guide vanes and by the shock of the passing guide vanes at the runner blades in its rotating reference frame.

At part load the transient flow was characterised by the rotating vortex rope in the draft tube cone. This effect could be observed by the high measured values on the suction side of the runner blades near to the trailing edge. The important pressure pulsation oscillations were caused by the rotating dynamic pressure field caused by the vortex rope in the vicinity of the blades suction side.
At deep part load there was the set-in of the runner channel vortex. This was noticed by the increased pressure oscillation amplitude near to the blade inlet edge. Larger pressure oscillation amplitudes were also observed in the direction of the band trailing edge, following the runner channel vortex shape.

As example of the measured pressure oscillation amplitude distribution over the runner blades, the measured values of \((\Delta H/H)^*\) at deep part load can be observed in Figure 7. The location of the pressure transducers, forming a measuring mesh, is marked at the blade meridian view. The variation of the pressure oscillation amplitude interpolated inside the measuring mesh is shown as contour plot.
4. Numerical Simulation Setup

The numerical simulations were carried out with the finite volume method (FVM) applied to the complete turbine with all its hydraulic components: spiral case, stay vanes, guide vanes, runner and draft tube. The calculations were conducted in a coupled manner, i.e. considering all the turbine components and interfaces between them, as long as the dynamic effects in the fluid flow through the turbine arise from the interaction between its components, caused by the rotating runner and the stationary parts.

Different computational mesh densities were tested and optimized to deliver accurate results with acceptable computation times. The final computational mesh counted with around 18 million cells and part of it can be seen in Figure 8. The employed FVM made use of second-order interpolation schemes for the time dependent, convective and diffusive terms.

The computational domain was artificially extended at the spiral case inlet and draft tube outlet with the objective to eliminate any boundary effects on the turbine inlet and outlet sections. As inflow boundary condition the volume flow was prescribed together with 5% turbulence intensity. At outlet a reference integral pressure level was prescribed and allowed any eventual backflow.

For the simulation of the instationary effects in the fluid flow through the turbine, like the rotor-stator interaction and the flow instabilities in the draft tube, adequate time resolution and turbulence models were needed. Approximately 400 time steps were calculated for each machine revolution, in order to capture not only the effects from the draft tube instabilities, but also from the rotor-stator interaction and in the runner channel.

Considering the turbulence modelling, the URANS (unsteady averaged Navier-Stokes equations) model introduced, as expected, excessive artificial dissipation in the flow simulation and turned out to be unable to reproduce the highly transient effects in the turbine, especially in the draft tube. Therefore, adequate and more sophisticated turbulence models had to be employed.

Hybrid turbulence models as detached eddy simulation (DES) and scale adaptive simulation (SAS) were already repeatedly and successfully employed in the past for the numerical simulation of the transient fluid flow with accurate results. Cases of precise numerical simulations making use of hybrid turbulence models, as for example SAS, in hydraulic turbines were brought...
among others by Magnoli [2, 3], Magnoli and Schilling [4, 5], Hasmatuchi [6], Benigni et al. [7] and Wunderer [8].

For the numerical simulations of the transient fluid flow through the FT 60 turbine, the SAS turbulence model was chosen. As investigated by Magnoli [2], the SAS and DES turbulence models have similar accurate performance for the transient flow simulation in Francis turbines. Through numerous publications, SAS is currently gaining in popularity compared to DES.

The scale adaptive model (SAS) from Menter and Egorov [9] attempts to resolve the turbulent eddies, switching from the large eddy simulation (LES) behaviour at fine mesh resolution regions to URANS in regions where the eddies are smaller than the grid resolution. This hybrid behaviour is achieved with the modification of the turbulence dissipation frequency, $\omega$, with the limiter $F_{SST-SAS}$, based on the von Kármán length scale, $L_{vK}$. The kinetic turbulence and turbulence length scale transport equations are transformed in the form of the equations in the $k$-$\omega$ SST model original formulation as shown below.

$$\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\bar{c}_j \rho k)}{\partial x_j} = \tau_{ij} \frac{\partial \bar{c}_i}{\partial x_j} - \beta^* \rho \omega k + \frac{\partial}{\partial x_j} \left[ \left( \mu + \sigma_k \mu_T \right) \frac{\partial k}{\partial x_j} \right]$$

(1)

$$\frac{\partial (\rho \omega)}{\partial t} + \frac{\partial (\bar{c}_j \rho \omega)}{\partial x_j} = \gamma \nu_T \tau_{ij} \frac{\partial \bar{c}_i}{\partial x_j} - \beta \rho \omega^2 + \frac{\partial}{\partial x_j} \left[ \left( \mu + \sigma_\omega \mu_T \right) \frac{\partial \omega}{\partial x_j} \right] + 2 \left( 1 - F_1 \right) \rho \sigma_\omega \frac{1}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} + F_{SST-SAS}$$

(2)

The influence of the von Kármán length scale on the turbulence dissipation frequency is controlled by the additional term $F_{SST-SAS}$ in relation to the original turbulence dissipation frequency transport equation in the $k$-$\omega$ SST model. The additional term is defined with two limiters:

$$F_{SST-SAS} = F_{SAS} \max \left( T_1 - T_2; 0 \right)$$

(3)

$$T_1 = \tilde{\zeta} \rho k S^2 \frac{L}{L_{vK}}, \quad T_2 = \frac{2pk}{\sigma_\Phi} \max \left( \frac{1}{\omega^2} \frac{\partial \omega}{\partial x_j} \frac{\partial \omega}{\partial x_j}; \frac{1}{k^2} \frac{\partial k}{\partial x_j} \frac{\partial k}{\partial x_j} \right)$$

(4)

In URANS regions $T_1 \approx T_2$ and the original $k$-$\omega$ SST model is recovered. In LES zones the limiter $T_2$ tends to zero and the term with the von Kármán length scale, $T_1$, is largely dominating. It corresponds to the sink term in the turbulence length scale transport equation and it controls the reduction of the turbulent viscosity, $\mu_T$, assuring the LES behaviour.

The interpolation scheme has also to be modified for the application of the SAS model. In the regions where SAS assumes the LES characteristic, the interpolation function must reproduce the CDS scheme. While, in other regions, where the URANS behaviour is dominant, the interpolation function must be second-order UDS. This blend between these two interpolations schemes is based on local flow characteristics and is described by Travin [10].

The transient simulation required approximately over 30 machine revolutions, depending on the operating point and on the prescribed initial solution, until the stable transient flow patterns were established. The computation of one machine revolution took, in average, 12 hours in a Linux cluster with 72 parallel threads running at Intel Xeon X5680 processors with 3.33 GHz and 24 GB memory.

5. Numerical Simulation Results

The three selected operating points, full load, part load and deep part load, were numerically simulated as described above. These different operating conditions constituted an interesting
benchmark for the prediction capability of the numerical simulation model. They counted with different transient fluid flow effects taking place in the hydraulic turbine, rotor-stator interaction, rotating draft tube vortex rope (DTI) and runner channel vortex (RCV).

The simulated pressure oscillation time history and frequency spectrum was compared to the experimentally obtained results. As example for the typical achieved numerical simulation accuracy, the numerical calculated results for the sensor located near to the blade trailing edge at the suction side near to band, \( u = 0.95, v = 0.80 \), are plotted together with the measurement results for deep part load in Figures 9 and 10.

This operating point and pressure transducer were taken as example, since they were responsible for the highest measured pressure oscillation amplitudes at the runner within the normal operating range. The simulated value \( \left( \frac{\Delta H}{H} \right)^* = 0.47 \) was tightly close to the measured value of \( \left( \frac{\Delta H}{H} \right)^* = 0.49 \). As seen in Figures 9 and 10, the pressure variation time history and frequency spectrum also showed tight agreement between the simulated and experimentally obtained values.

For the extensive evaluation of the accuracy of the numerically calculated pressure oscillation amplitude, its distribution over the blades pressure and suction sides surfaces were compared to the experimentally determined values. The experimental and simulated \( \left( \frac{\Delta H}{H} \right)^* \) distribution over the runner blades at full load can be respectively seen in Figures 11 and 12. The results at part load can be found in Figures 13 and 14. For deep part load the comparison is available in Figures 15 and 16. The experimentally determined pressure oscillation amplitude at the blade measuring locations was interpolated and extrapolated over the complete blade surface with the Kriging method as explained e.g. by Davis [11].

The experimental and numerically simulated pressure oscillation amplitude distribution showed roughly the same tendency for the three investigated operating points. At full load due to the rotor-stator interaction the largest amplitudes were located near to the blade leading edge and decayed considerably fast in direction to the blade body and trailing edge. The rotating vortex rope in the draft tube cone dominated the pressure oscillation distribution at part load, with larger pressure oscillation amplitudes at the blade suction side near to the trailing edge. At deep part load the runner channel vortex caused the larger amplitude values near to the band in direction to the trailing edge, following the same pattern as the vortical structure. For this mid specific speed Francis turbine, the highest amplitudes were observed at deep part load, followed by part load and full load.
As observed in Figures 11, 13 and 15, the few number of measuring points at the runner blade surface offered only a coarse mesh for the determination of the pressure oscillation amplitude field. Even with the Kriging interpolation method, no smooth distribution could be generated. Moreover, in some portions of the blade, near to its extremities, the pressure oscillation amplitude field had to be extrapolated.

On the other hand, the fluid simulation results could provide a fine distribution field for \( (\Delta H/H)^* \) as plotted in Figures 12, 14 and 16. With the tight agreement between the measured and calculated pressure oscillation amplitude values at the single pressure transducer locations, as seen e.g. in Figure 9, an effective procedure for future projects would be to verify and calibrate...
the simulation results with the data obtained at the sensors. For further analysis, calculation and design, the validated and calibrated transient simulation results could be reliably used.

6. Conclusion
The transient pressure measurement at the model runner blades were seldom performed in the past. They required a complex experimental arrangement, in order to allow the pressure measurement, data transmission and acquisition.

The transient pressure measurement at the FT 60 model turbine at numerous operating points offered the possibility to obtain further knowledge about the complex transient characteristics of the fluid flow through Francis turbines.

The deeper analysis and understanding of the transient fluid flow the hydraulic can possibly lead to further improvements on the hydraulic and mechanical turbine design, machine stability and reliability.

The experimental data about the pressure oscillation amplitude at the turbine runner also permitted the comparison with numerical predictions of the transient fluid flow through the Francis machine performed with CFD and advanced turbulence models. The available experimental data and comparison to the numerical results can improve the prediction accuracy of the numerical models.

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