Deep Part Load Flow Analysis in a Francis Model turbine by means of two-phase unsteady flow simulations

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Abstract.
Hydropower plants are indispensable to stabilize the grid by reacting quickly to changes of the energy demand. However, an extension of the operating range towards high and deep part load conditions without fatigue of the hydraulic components is desirable to increase their flexibility. In this paper a model sized Francis turbine at low discharge operating conditions \( \frac{Q}{Q_{BE}} = 0.27 \) is analyzed by means of computational fluid dynamics (CFD). Unsteady two-phase simulations for two Thoma-number conditions are conducted. Stochastic pressure oscillations, observed on the test rig at low discharge, require sophisticated numerical models together with small time steps, large grid sizes and long simulation times to cope with these fluctuations. In this paper the BSL-EARSM model (Explicit Algebraic Reynolds Stress) was applied as a compromise between scale resolving and two-equation turbulence models with respect to computational effort and accuracy. Simulation results are compared to pressure measurements showing reasonable agreement in resolving the frequency spectra and amplitude. Inner blade vortices were predicted successfully in shape and size. Surface streamlines in blade-to-blade view are presented, giving insights to the formation of the inner blade vortices. The acquired time dependent pressure fields can be used for quasi-static structural analysis (FEA) for fatigue calculations in the future.

1. Introduction
The European energy mix undergoes massive changes as more and more new renewable energy e.g. generated by wind and solar penetrate into the energy market. Their unlimited precedence injection and their volatility in power output lead to an imbalance between energy production and consumption. Hydropower plants play an important role in stabilizing the grid by responding quickly to sudden changes in the energy production and demand. An extension of the operating range towards off-design condition including deep-part load is desirable to add flexibility. However, conditions outside the originally indented operating range may lead to harmful loading due to pressure oscillations and cavitation phenomena. Key for long lifetimes and save operations of hydraulic components is the successful prediction of the loads on the prototype. Steady state flow simulations using CFD and structural analyses based on Finite Element Methods (FEM) are applied successfully during the design phase of the hydraulic components, but become inaccurate towards off-design conditions. Single phase-flow simulations for deep part load conditions were published by Magnoli[1]. Wack et al.[2] focused on the cavitation phenomena studying mesh size and outlet boundary treatment. Yamamoto et al.[3]
investigated experimentally the inner blade vortices by means of high speed visualizations at different speed factors.

2. Numerical setup
The computational domain reaches from the spiral case inlet to the draft tube outlet and was discretized by 14.54m hexahedral cells. Runner crown bores and side gaps were neglected to simplify the geometry. Simulations where carried out with the commercial CFD solver ANSYS CFX. Advection and temporal terms were discretized by high-resolution methods and second-order backward Euler, respectively. For two-phase modeling, vapor and liquid water are treated as continuously with constant properties. Because small vapor bubbles are convected with the main flow due to their low inertia the homogeneous multiphase model was employed. The Rayleigh-Plesset cavitation model was implemented to account for phase change. Stochastic flow fields at deep part load conditions demand for advanced turbulence treatment. In this study the so called explicit algebraic Reynolds stress model (BSL EARSM) is used. EARSM models have a comparable computational effectiveness as standard two-equation models and are numerically more robust then Reynold stress models (RSM)[5]. Derived from RSMs they account for many flow phenomena without solving transport equations for all six unknown stresses[4]. The BSL EARSM is an extension of the standard two-equation models and includes the nonlinear relation between the Reynolds stresses and the mean strain-rate and vorticity tensors. Additionally, curvature correction was employed because of the known insensitivity of eddy-viscosity models to streamline curvature and system rotation[6]. The measured mass flow was prescribed at the spiral case inlet. At the draft tube outflow an opening boundary condition with option “Entrainment” was set and a relative pressure corresponding to the measured Thoma-number was specified. The case was initialized by decreasing the Thoma-number and the time step size linearly ($\Delta t = 0.9^\circ$).

3. Numerical results
The investigated operating points are presented in table 1. Since the mass flow and the wicket gate opening is prescribed the turbine head is a simulation result. Comparison to the measured head yields large discrepancies (an underestimation of 9.87% and 10.12% for condition 1 and 2 respectively). However, the deviations in torque are less then 1.2% for both Thoma-numbers.

| Table 1: Operating conditions | Cond 1 | Cond 2 |
|-------------------------------|-------|-------|
| $Q_{ed}$                      | 0.05445 | 0.05442 |
| $n_{ed}$                      | 0.2881  | 0.2883 |
| $\sigma$                      | 0.11    | 0.07   |
| $\alpha$                      | 5       | 5      |

3.1. Flow visualization
The inner blade vortices, main source for the dynamic loads acting on the runner blade, originate from the flow misalignment at the runner leading edge for low discharges and small wicket gate
openings. In figure 1 surface streamlines of the transient averaged velocity at three constant span surfaces are presented in blade-to-blade view. The inner blade vortices can be clearly identified at approximately one third of the blade chord length and attached to the suction side. From the runner hub to half span height the vortex is axial parallel which becomes obvious when comparing the streamlines at 10% and 50% span. Near the hub the vortex blocks the main flow almost entirely. Near the shroud the vortex turns radial and follows the shroud contour.

3.2. Cavitation phenomena
The channel vortices rotate with high velocities forming low pressure zones in the vortex core. Depending on the Thoma-number the corresponding pressure may drop below the vapor pressure forcing the liquid to evaporate. For the investigated conditions large differences in the vapor volume are noticeable. Figure 2 shows the runner from below for one instance in time. Cavitation is visualized by a red iso-surface of the vapor volume fraction at 10%. For the higher Thoma-number, small cavitation regions are predicted only in one half of the runner channels, wherein for condition 2 almost every channel shows cavitation and the vapor volumes are remarkably larger. The differences in vapor volume inside the channels indicate fluctuations in the flow field and an interaction with the rotating low pressure zones inside the draft tube (cf. figure 5).

![Figure 2: Runner from below with iso-surfaces at 10% vapor fraction colored in red (Condition 1 left, Condition 2 right).](image_url)

3.3. Pressure oscillations
To compare the simulation results with measurement data, 5,128 simulated runner revolutions, resolved by 2048 time steps, were used. The available measurement data covers 1600 runner revolutions resulting in clear peaks for all deterministic frequencies. For time spans comparable to the simulation time, large differences among them were identified. Therefore an arbitrary time span was selected and used for all comparisons. Figure 3 shows the time signal and the corresponding frequency spectra for pressure sensor SS_TE2, positioned on the blade suction side (cf. meridional view in figure 3). The FFT reveals a broad band of low frequencies between 0-10 times the runner frequency which is resolved successfully by the simulation. While the amplitudes for $\sigma = 0.11$ are slightly underestimated, the simulation overestimates these for the lower Thoma-number. Rotor-stator interaction becomes visible as a peak at $20 \cdot f_{runner}$ and gets damped by the increasing vapor volume, in contrast to the amplitudes of the low frequency which are amplified. Both effects are captured by the simulation. By plotting the pressure amplitude decomposed into distinct frequencies on the runner suction side the origin for the above discussed low frequency fluctuation can clearly be assigned to the occurrence of the inner blade vortex whose trace is clearly visible. Figure 4 shows the results for ten frequencies between 0.2 and 2.0 times $f_{runner}$ at $\sigma = 0.11$. The contour plots show that the highest amplitudes arise...
upstream of the measuring position SS_TE2, which means that slight positional deviations in the measuring point placement and the prediction of the vortex shape would result in large deviations for the predicted amplitudes.

![Figure 3: From left to right: Measuring positions in meridional view, time signals and frequency spectra for both conditions (SS_TE2).](image)

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![Figure 4: Pressure plots on blade suction side decomposed into distinct frequencies.](image)

Figure 4: Pressure plots on blade suction side decomposed into distinct frequencies.

Measuring position CI1N is located in the draft tube cone as marked in the sketch of figure 5. Next to it the recorded time signals and the corresponding FFTs are presented. A very good correlation can be stated for $\sigma = 0.11$. For $\sigma = 0.07$ the runner rotational frequency is dominant in the measured signal, which is less pronounced in the simulation result. Four pictures in figure 5 illustrate the origin of the low frequency fluctuations observed in the draft tube cone. Iso-surfaces visualize low pressure structures rotating underneath the runner. The formation and collapse result in a varying amount of these structures which yields the broad band of frequencies.

4. Conclusion
Two unsteady two-phase simulations at deep part load with different Thoma-numbers are performed. Deviations up to 10% are found for the simulated head whereas the torque is in good agreement with the measurements. Deviations can be partly explained by the neglect...
of the runner side gaps and the crown bores. Pressure signals are compared as well. For the lower Thoma-number an increase in amplitude was observed. An acceptable correlation for all measuring locations was found as the low frequencies and the rotor-stator interaction were captured. Multiphase modeling reveals size varying vapor bubbles in the center of the inner blade vortices over time. The simulation approach proves its applicability to low discharge conditions including cavitation. For future work, advanced numerical methods for turbulence treatment like scale resolving simulations could be used to reduce the uncertainties associated with turbulence modeling. The loads calculated by the simulation can be mapped on a finite element model for fatigue calculations.

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