

PAPER • OPEN ACCESS

Investigation of a fast transition from pump mode to generating mode in a model scale reversible pump turbine

To cite this article: C Stens and S Riedelbauch 2016 *IOP Conf. Ser.: Earth Environ. Sci.* **49** 112001

View the [article online](#) for updates and enhancements.

You may also like

- [Study of Sulfur-Based Electrodes By in Situ Characterization Techniques](#)
Quentin Lemarie, Hassane Idrissi, Eric Maire et al.
- [Electrodeposition of Al-Ti Alloys from EmimCl-AlCl₃ Ionic Liquid Containing TiCl₄](#)
Takahiro Kizawa, Taisuke Nozawa, Takehiko Kumagai et al.
- [Comparison of Electrochemical and Electrothermal Models for Lithium-Ion Batteries Used in 3D Simulation](#)
Clemens Fink and Bernhard Kaltenegger



ECS
The
Electrochemical
Society
Advancing solid state &
electrochemical science & technology

DISCOVER
how sustainability
intersects with
electrochemistry & solid
state science research

Investigation of a fast transition from pump mode to generating mode in a model scale reversible pump turbine

C Stens and S Riedelbauch

University of Stuttgart, Institute of Fluid Mechanics and Hydraulic Machinery,
Pfaffenwaldring 10, 70550 Stuttgart, Germany

christine.stens@ihs.uni-stuttgart.de

Abstract. Pumped storage power plants are an efficient way to store energy at a large scale. In the last years, the changes between pump and turbine mode have become more and more frequent and the necessity of fast changes has increased. This paper analyses the flow in a model scale pump turbine during a fast transition from pump mode to generating mode by means of CFD. Results will be compared between two different mesh sizes and between simulation and measurement. A linear variation of rotational speed over time is chosen. A time-dependent flow rate through the machine is prescribed at the inlet. Due to the varying conditions, a fully transient analysis is carried out using the open-source code OpenFOAM®. The state of the machine at certain points of time during the transient is compared to the results for steady state simulations with identical boundary conditions. To characterize the phenomena in the guide vane channels, torque on selected guide vanes is evaluated as well as pressure at predefined locations. In the runner, pressure sensors are evaluated near the leading edge on pressure and suction side. In the draft tube, four dynamic pressure sensors in a plane below the runner are analysed. Frequencies and amplitudes are compared to simulation.

1. Introduction

A growing share of renewables in the electric grid leads to an imbalance between the production and consumption of electric energy. Pumped storage power plants offer an efficient way to store excessive energy on a large scale in times of high production and release it again when production falls below demand. To pass from one operating regime to the other, a reversible pump turbine needs to change its rotational direction and the flow direction reverses.

The more fluctuating production there is, the more desirable it becomes to be able to perform fast transitions between the operating modes without damaging the machine. This requires knowledge about the flow phenomena during a transient and the resulting mechanical loads and relevant frequencies. Some typical and possibly detrimental phenomena in pump turbines were investigated in [1].

CFD simulations for time-varying conditions have been carried out by various authors. For Francis and pump turbines, many of these are related to runaway (e.g. [2, 3, 4]). Others concentrate on start-up [5] or speed no load conditions [6]. The latter is of particular interest for Francis type pump-turbines, as these often show an S-shaped characteristic in generating mode.

In this paper, we investigate a transition from pump mode to generating mode in a Francis type reversible pump turbine without closing the guide vanes, a condition that is closely related to a conventional pump failure. To facilitate experiment and simulation, a linear variation of rotational



speed is chosen. A preliminary numerical study by the authors [7] was carried out with a rather coarse mesh and analysed the predominant flow phenomena during the transition.

The present work numerically and experimentally extends the previous analyses. In numeric analysis, steady state operating points are simulated for model verification and comparison with the transient. Additionally, all simulations are performed with the original mesh and a refined version to investigate how much relevant information can be drawn from a coarse mesh with its limited computational effort. For defined locations and variables, simulated results are compared with experimental data obtained from a model scale pump turbine in a test rig.

2. Simulation Setup

The pump turbine consists of 24 guide vanes and seven runner blades. For all simulations presented in this work, a full model including the spiral, twin cascade, runner and draft tube is employed, as not all expected flow phenomena show rotational symmetry. Simulations are run with the open source code OpenFOAM® for mesh sizes of 3 million and 20 million cells. Results for the initial state are mapped from the coarse to the fine mesh and ten runner revolutions are simulated before the start of the transient. The time step is kept constant for the whole transient and equals 3.6° per time step at the original rotational speed. The simulation uses OpenFOAM®'s k- ω -SST model.

For the model test, a transition time of 8 s is chosen. Data are shifted so that the beginning of the transient is at 1 s. The flow rate is determined by the test rig conditions. For simplicity, a linear change in rotational speed of both the pump turbine and the test rig pump are chosen. Under these conditions, flow rate rapidly reverses from pump to turbine flow direction passing zero at $t = 3.07$ s, whereas the increase is significantly slower during the rest of the transient. Flow rate and rotational speed from experiment are filtered and directly prescribed as boundary conditions for the simulation via lookup-tables. Flow rate is always prescribed at the respective inlet, i.e. the simulation is interrupted when it reaches zero and boundary conditions are adjusted accordingly.

Results include head, guide vane torque and pressure at selected locations. These quantities are compared to experimental data. Head is evaluated as in experiment, i.e. from flowrate and pressure values in the spiral and the draft tube.

3. Simulation of steady state operating points

For comparison, a set of steady state operating points along the transient is investigated. Two points are simulated and measured in each quadrant of the characteristic. Rotational speed is chosen to be 100%, 75%, 50%, 25%, -25% of the original value. Additionally, the state after the end of the transient is investigated. Time step is equal to the one chosen for the transient and constant for all rotational speeds, so that the lowest rotational speeds are resolved with 0.9° per time step. For the comparison, operating points are defined by a constant guide vane opening and flow rate.

Figure 1 presents the measured and simulated operating points together with the curve from the four quadrant characteristic. As flow rate and rotational speed are prescribed in simulation and guide vane opening is identical in measurement and simulation, differences between the curves are caused by the difference between simulated and measured head. The last value at maximum n_{ED} is calculated in experiment and simulation from the transient after all values have stabilized.

At the beginning and end of the transient, the difference in head between simulation with the fine mesh and experiment is 2.5 % and 2.3 %, respectively, which is considered a sufficiently good agreement to proceed with more complex operating points.

The second operating point is still in pump mode, but at a significantly lower flow rate. This leads to stall and consequently pressure fluctuations especially in the twin cascade. Still the finer mesh is able to reproduce the mean values of head and pressure in the guide vane channels and the runner with an error of less than 2 %. For the 3M mesh, the maximum deviation of head from experiment is 7 % in the runner. The 3M mesh underestimates and the 20M mesh overestimates the head and pressure, such that the difference in results between the meshes ranges from 2 % to 4 %. Furthermore, fluctuations are smaller in the 3M mesh than in the finer mesh and the experiment.

Both points in the pump brake quadrant are characterized by large pressure fluctuations in the guide vane channels, caused by the fact that the passing runner blades force the flow outward while the mean flow is inward. Fluctuations in head and in pressure at the runner leading edge remain comparable to the pump operating points. With a more complex flow field, simulation is less capable of capturing all the relevant phenomena and the deviation from experiment increases. For example, the difference in head is 7.5 % for the point with the lowest n_{ED} value in pump brake mode.

Unfortunately, torque measurement failed before the final measurements of the steady state operating points. Still, a comparison of normalized torque with the data from the measurement of the characteristic is presented in Figure 2 and shows a good agreement for the fine mesh. In pump brake mode, torque shows a higher mesh dependency, but the difference between results from the fine mesh and simulation remains comparable to other operating points. In generating mode at low flow rate and low rotational speed, both meshes are in very good agreement with measurement regarding head and reasonably well predict torque.

From the results of the preliminary investigations on steady state operating points, both meshes are expected to give reasonable results for the transient.

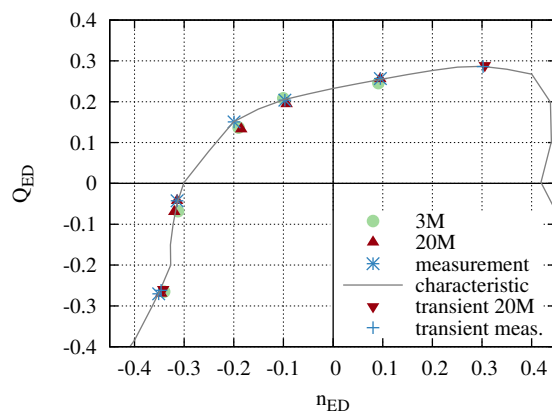


Figure 1. Simulated and measured steady state operating points.

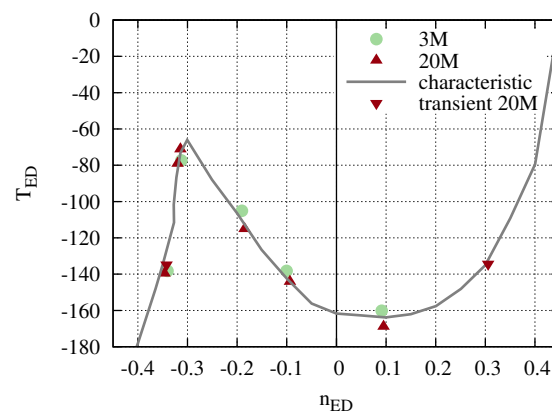


Figure 2. Torque for steady state operating points.

4. Simulation of the transient

The transition from pump to generating mode is simulated with both meshes. An analysis of global quantities gives an overview of the quality of the simulation results and is followed by a detailed analysis of the different sections of the machine. All pressures are evaluated against the pressure at the sensor positions at the end of the draft tube and normalized by the constant measured head at the beginning of the transient.

4.1. Convergence

To ensure convergence at each time step, the behaviour is monitored at the beginning of the transient and at a point during the pump instability. The simulations are run using a transient SIMPLE-scheme with fifteen outer iterations. The initial residual decreases in each iteration. However, in both cases, it can be shown that a maximum of eleven outer iterations and a final residual of $5e-5$ after the last iteration are sufficient to reach constant values for head, torque and pressure at the pressure sensors on the runner blades. Throughout the transient, residuals show no sign of divergence of the solution. During the pump instability, however, the number of iterations required to meet the target of $5e-5$ after the last outer iteration increases from six to twenty. Judging from this number combined with initial and final residuals, prescribing flow rate at the spiral case is numerically more stable than prescribing

it at the draft tube. This also holds true for pump brake mode with the large deviations between simulation and experiment.

4.2. Global result quantities

Figure 3 shows a comparison of experimental data and simulation in a four quadrant plot, together with the characteristic for the considered guide vane opening. Experimental data has been smoothed for clarity. The transient starts in the lower left quadrant in pump mode and then moves along the curve to the upper right corner. Dots along the transient mark the steady state operating points from section 3.

The operating point at the beginning of the simulation shows a good agreement as stated in section 3. However, shortly after the reduction of both flow rate and rotational speed starts, the simulation deviates from the experimental results. At the same time, a difference between transient results and the steady characteristic becomes visible in the measurement data. The agreement between simulation and experiment improves again as flow rate decreases, but only until reaching dissipation mode.

Obviously, this is the most critical and challenging part for CFD. While there is little difference between the meshes, there is a significant difference to the experimental data. Simulated steady state operating points show little differences to the transient results, although this difference is clearly visible in measurement.

Generating mode finally shows the best agreement between simulations and measurement. A finer mesh improves the accuracy of head prediction and leads to a better agreement as can be seen at the end of the transient. After rotational speeds of the test rig pump and the pump turbine reach their final values, there is still a small deviation between the characteristic and the transient that takes some time to disappear. Both meshes show that simulation can capture this.

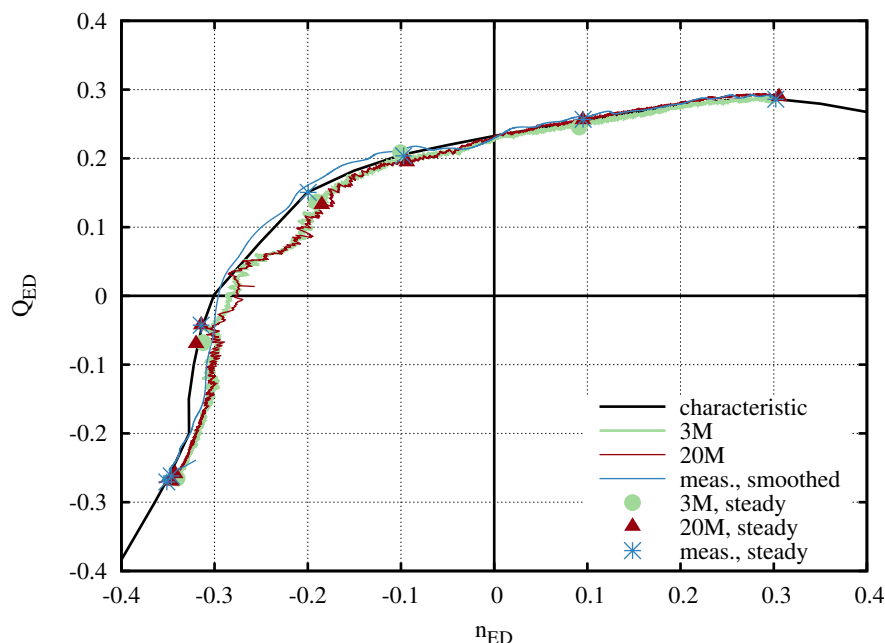


Figure 3. Comparison of measurement and simulation in four quadrant plot.

Between the meshes, the difference in simulated head depends on the operating mode. In stable pump mode, values for the coarser mesh are approximately 2.5 % of reference head higher than those predicted by the fine mesh. This reduces to a mean of zero with high fluctuations at the end of pump mode and in pump brake mode. For generating mode, a relatively constant offset of 4 % to 6 % of reference head is found. Differences between the fine mesh and experiment range from below 3 % in

generating mode over 4 % to 6 % in pump mode up to 11 % in pump break mode. Again, the offset is nearly constant within the particular operating regimes.

4.3. Results in the twin cascade

Pressure sensors in three guide vane channels and torque measurement on three guide vanes provide data for comparison with the simulation. To give an example for the pressure measurements, Figure 4 shows the results at the sensor positioned at 90° from the spiral case inlet over time. As the reference level is set at the draft tube exit, a comparison between experiment and simulation in the guide vane channels reflects all differences between the two in the draft tube and the runner additionally to those in the guide vane channels.

From the beginning of the transient until 2.5 s, simulation captures the slope of the pressure curve, but overestimates the absolute value. This is in agreement with the fact that in the same time interval, head is overestimated as shown in Figure 3. During pump brake mode, large fluctuations appear, whose amplitudes are well captured in simulation, while the simulated mean is still slightly higher than measured values. The dominating frequency corresponds to the blade passing frequency of the runner. Measurement also shows significant amplitudes at higher frequencies that are not captured by simulation.

In generating mode, simulation with the fine mesh agrees very well with measured data, while values from the coarse mesh continue at a higher level. The frequency of the fluctuation in experiment corresponds to blade passing frequency and its first harmonic and is found in simulation for both meshes with similar amplitudes.

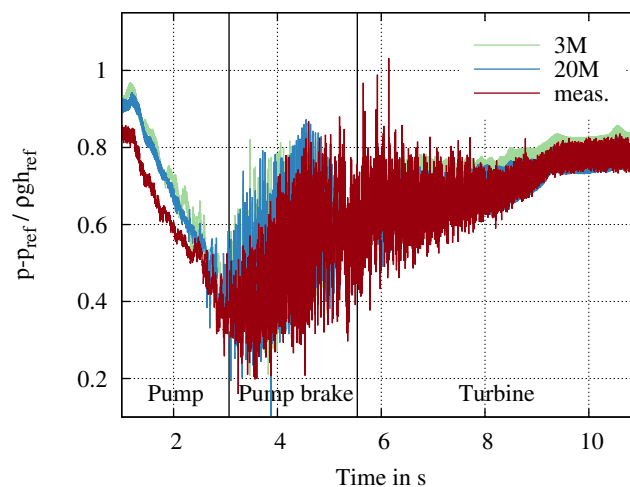


Figure 4. Pressure in a guide vane channel over time.

Previous studies with the coarse mesh showed evidences of rotating stall at low flow rates in the pump quadrant. Three parameters are used in the present paper to track the phenomenon: mass flow through each of the guide vane channels, torque on selected guide vanes and pressure at three locations on the top of the section. The latter two are compared to experimental results.

As an example, Figure 5 shows the evaluation of torque on the single guide vanes. For both meshes, disturbances can be tracked through adjacent channels over time. The same holds true for the other two criteria, flow rate and pressure at the pressure sensors. The guide vanes are numbered against the direction of rotation in pump mode, so that a passing of a minimum or maximum from high to low numbers over time in Figure 5 indicates that the phenomenon rotates with the runner. The frequency corresponds to approximately 8 % of the runner frequency. It slows down between 2.5 s and 3 s before disappearing completely as flow rate approaches zero.

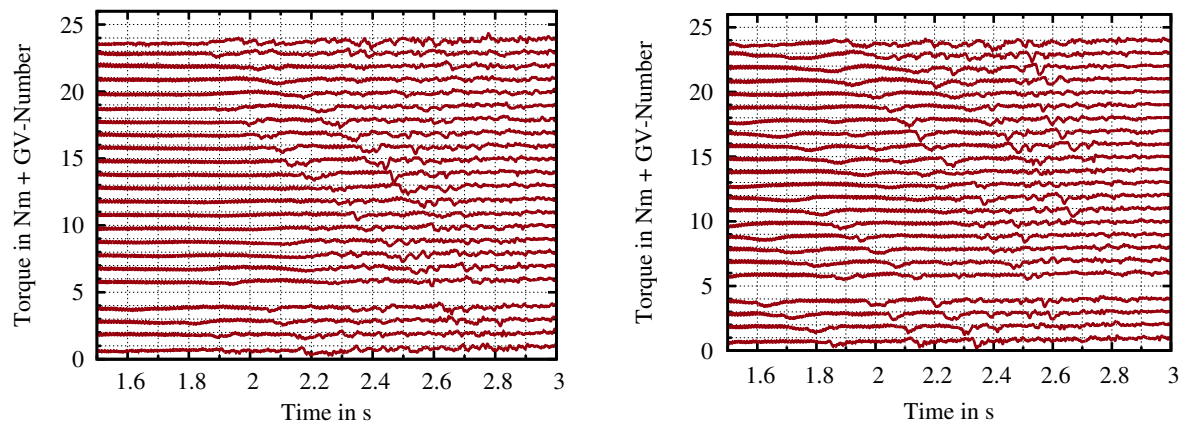


Figure 5. Disturbances in the guide vane torque signal passing through the channels during the pump instability. The torque signal has been offset by the respective guide vane number. Left picture: fine mesh, right: coarse mesh.

Figure 6 compares the pressure at the pressure sensors in the guide vane channels over time at low flow rates in pump mode. For simulations as well as measurement, at the beginning, the signal of all three sensors has a similar mean value and shows fluctuations according to the passing of the runner blades. This is followed by a period where the single sensors differ from each other, but blade passing is still visible. At very low flow rates, pressure fluctuations become stochastic. An analysis of the flow field shows that during the second phase, single channels are affected by larger turbulent structures, whereas the third phase shows randomly distributed structures in the majority of channels.

As can be seen from the figure, the onset of phase two is predicted too early by simulation. However, a finer mesh delays the onset and leads to a better agreement with measurement. Comparing absolute values at the individual sensors, simulation overestimates the pressure by nearly 10 % at $t = 2$ s, independently of the mesh. With decreasing flow rate, the agreement improves.

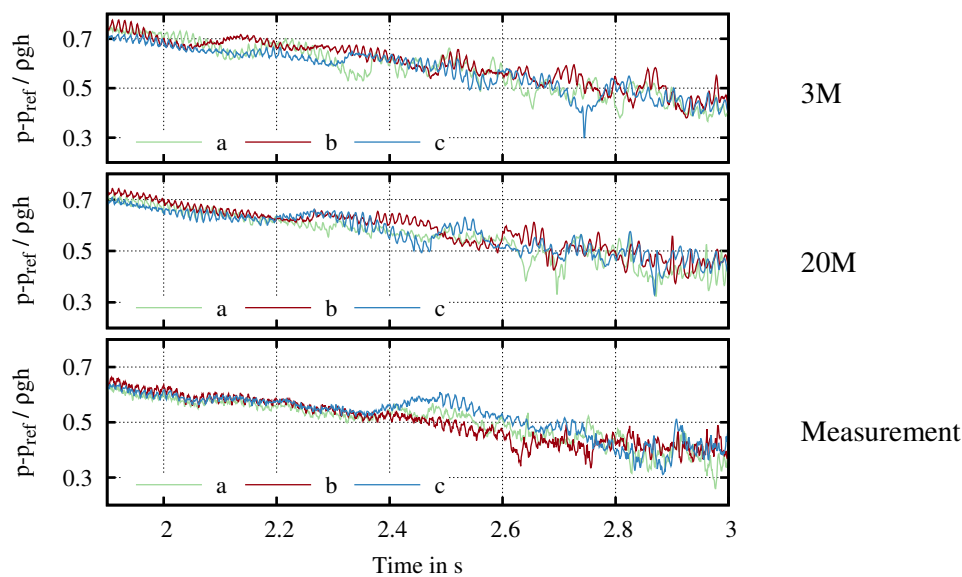


Figure 6. Pressure in the guide vane channels in simulation and measurement.

4.4. Results in the runner

In the runner, pressure is evaluated at two pressure sensors near the leading edge of the runner blades, one on the pressure side and one on the suction side. The signals over time for simulation are shown in Figure 8 and Figure 9, respectively.

On pressure side, the highest fluctuations occur at the end of pump mode and in pump break mode, where flow is forced outward in the guide vane channels when the runner blade passes, but the overall flow direction is inward. Still, fluctuations stay small compared to the pressure fluctuations in the guide vane channels.

As in head, a constant offset exists between the two meshes in generating mode at the pressure side sensor. On suction side, the offset disappears between 8.5 s and 8.6 s, where the curve for the finer mesh jumps back to the one of the coarser mesh. This coincides with the disappearance of high amplitude fluctuations in measurement. The sudden change in pressure in simulation results from the fact that in the upper part of the suction side, flow is able to follow the blade contour, while in the lower part of the channel, it detaches from the suction side, resulting in the pressure distribution depicted in Figure 7. The jump signals that the border between the two has moved further downward and the sensor has passed from the stall zone to one with attached flow.

In the coarse mesh, the general flow is comparable to the fine mesh, but the pressure gradient along the channel height is less steep between the two zones. Thus the pressure sensor more gradually passes from one zone to the other, resulting in a constant rise of mean pressure in Figure 9.

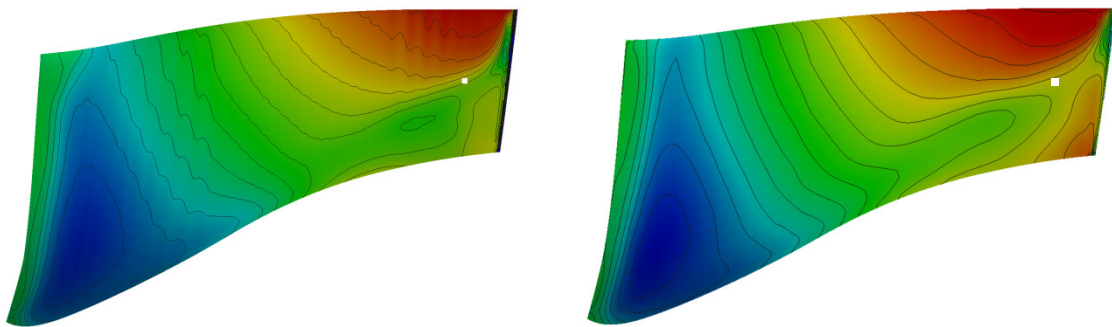


Figure 7. Pressure distribution on the suction side at $t = 8.55$ s for the 20M mesh (left) and the 3M mesh (right). Contour lines have a distance of $0.03 h_{ref}$. The 20M mesh shows as steeper gradient around the pressure sensor, leading to the jump in the pressure signal in figure 9. The white square signals the position of the pressure sensor.

Figures 10 to 13 show the corresponding FFT results for measurement and the 20M mesh. The blade passing frequency of the 24 guide vanes appears as peaks with linearly varying frequency starting from 370 Hz in both measurement and simulation. Their amplitude is approximately constant in stable pump mode before it nearly disappears at low flow rates. In pump brake mode, amplitudes are significantly higher, while in generating mode, the amplitudes are again constant at a low level.

Besides blade passing frequency, various low frequencies with high amplitudes are found. On the pressure side, this affects mainly pump mode and pump brake mode, while fluctuations on the suction side persist also in generating mode. The latter behaviour is not reproduced by simulation, where fluctuations in generating mode have significantly smaller amplitudes. However, simulation does capture some relevant peaks between 8 s and 9 s, where measurement shows higher amplitudes not only at low frequencies, but over a broad range.

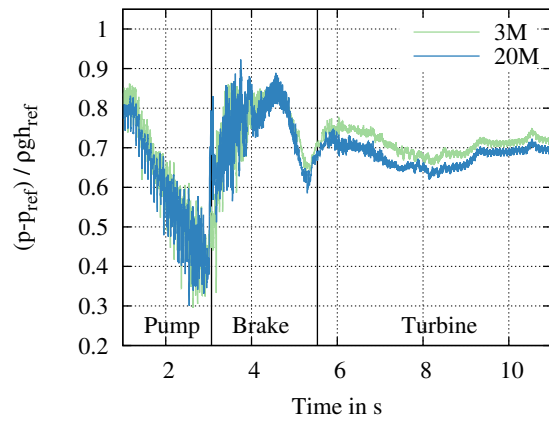


Figure 8. Pressure at LE pressure sensor, pressure side.

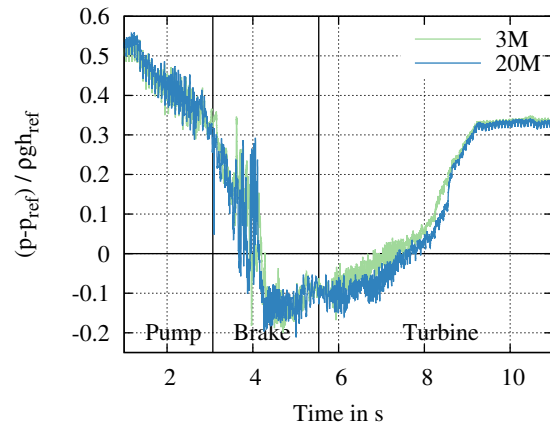


Figure 9. Pressure at LE pressure sensor, suction side.

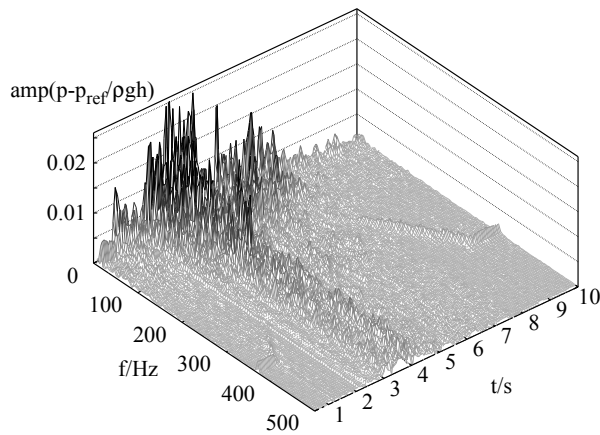


Figure 10. FFT of the pressure signal, LE pressure side, measured values.

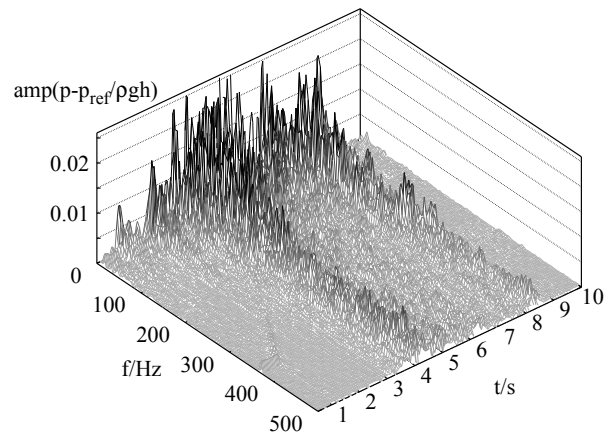


Figure 11. FFT of the pressure signal, LE suction side, measured values.

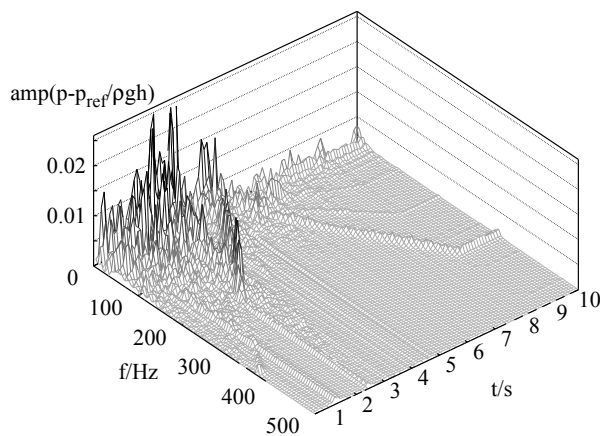


Figure 12. FFT of the pressure signal, LE pressure side, CFD with 20M mesh.

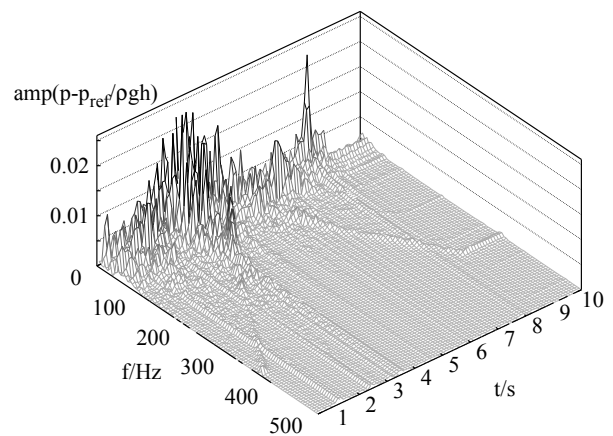


Figure 13. FFT of the pressure signal, LE suction side, CFD with 20M mesh.

4.5. Pressure fluctuations in the draft tube

Four dynamic pressure sensors below the runner positioned at 90° from each other are evaluated in both simulation and experiment. Figure 14 provides the result of an FFT of the signal of the first pressure sensor for the different meshes and the measurement. The behaviour is characterized by high amplitudes at low frequencies in the middle of the transient, but without a singular identifiable frequency that could give evidence for a vortex rope rotating at a defined speed. Again, as shown for the guide vane pressure signals in section 4.3, the onset of these fluctuations is predicted by simulation before it appears in the measurement. Simulation shows first peaks already in pump regime, while in measurement, they appear in pump brake mode. In generating mode, simulation clearly shows blade passing frequency of seven runner blades and its first harmonic. These are not easily identified in measurement.

Figure 14 additionally gives the simulated results for both meshes over time. Generally, there is a very good agreement between the curves. The coarse mesh slightly overestimates pressure in pump mode and underestimates the value in generating mode. It also predicts higher amplitudes of the fluctuations caused by the runner blades during the last two seconds of the transient.

An analysis of the flow velocities in the sensor plane confirms the general findings of the previous paper [7]. Four rotating low pressure zones in pump break mode collapse to two and finally one rope in generating mode. The main flow in the draft tube starts to rotate with the runner direction in pump mode and keeps this behaviour also in generating mode. At the end of the transient, a narrow ring at the draft tube wall is rotating with the runner direction.

The results for the flow in the draft tube are only slightly mesh dependent and the general form of the curves shows a good agreement for all time steps. The difference in absolute value between the meshes is approximately 2.5 % of reference head.

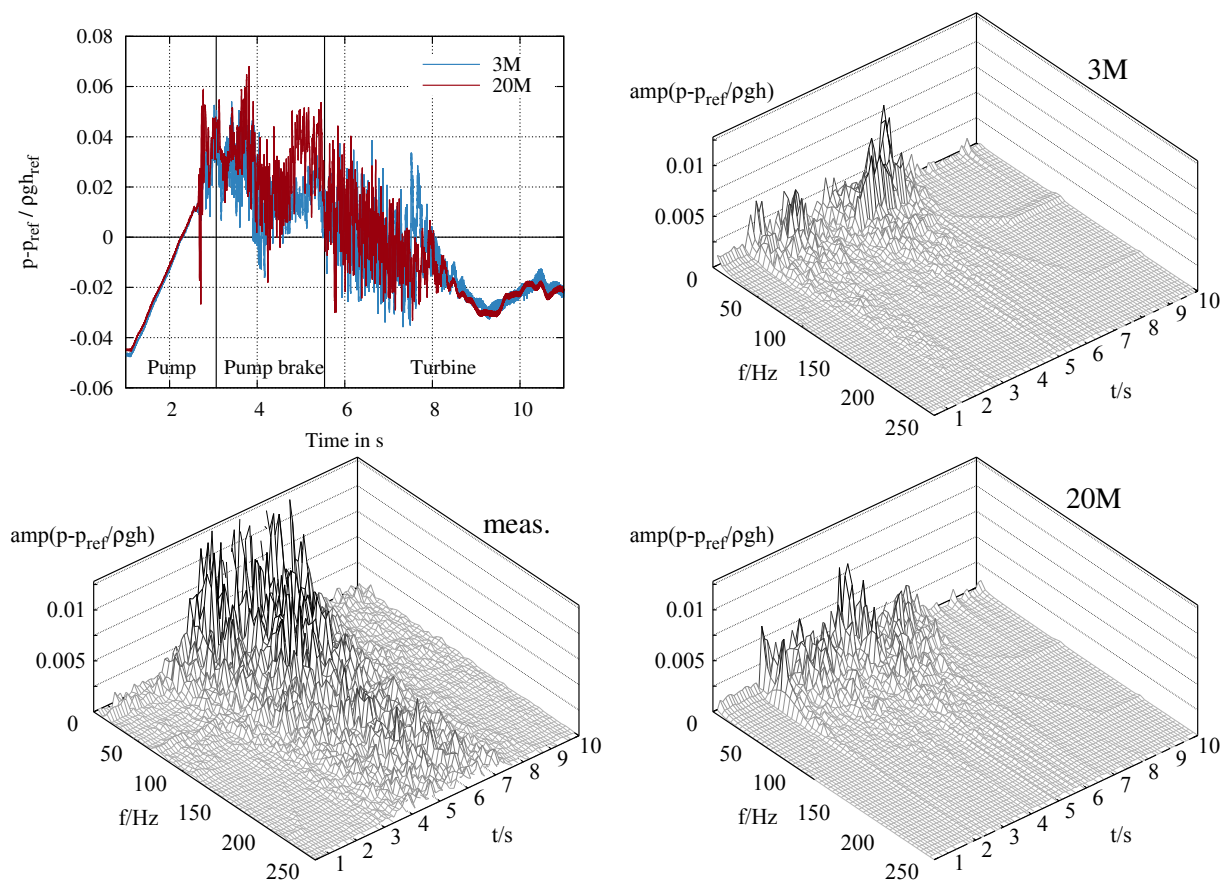


Figure 14. Values over time and FFT of the pressure signal in the draft tube below the runner.

5. Conclusions and Outlook

A comparison with experimental data shows that a URANS simulation is capable of capturing significant phenomena during the transient over a wide range of operating points. It correctly predicts that there are differences between steady state operating points and the operating points passed along the transient with the exception of pump brake mode.

Simulations were carried out with a coarse mesh of approximately three million cells and a fine mesh with 20 million cells. The coarse mesh generally captures the positions along the characteristic as well as the finer mesh and is well suited for an analysis of the general flow phenomena that appear during the transient. However, the onset of phenomena like stall in the guide vanes and fluctuations in the draft tube is predicted too early, which can be improved by a finer mesh. Furthermore, the 20 million cell mesh is better suited in regions of stable flow, especially in the turbine quadrant.

A critical point for simulation is pump brake mode with its high fluctuations and complex flow in guide vane channels and runner. Both in steady state and during the transient, there is a significant difference between simulated and measured head.

Future work will try to improve the predictions of simulation in pump brake mode. Furthermore, a mechanical analysis based on the generated CFD data will investigate the effects of the fast transition on the structure.

6. Acknowledgement

The authors would like to thank Andritz Hydro SA for providing the experimental data. 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). Part of this work was performed on the computational resource ForHLR Phase I funded by the Ministry of Science, Research and the Arts Baden-Württemberg and DFG ("Deutsche Forschungsgemeinschaft").

References

- [1] Gentner C, Sallaberger M, Widmer C, Braun O and Staubli T 2012 Numerical and experimental analysis of instability phenomena in pump turbines *IOP Conference Series: Earth and Environmental Science* **15** 032042
- [2] Cherny S, Chirkov D, Bannikov D, Lapin V, Skorospelov V, Eshkunova I and Avdushenko, A 2010 3D numerical simulation of transient processes in hydraulic turbines. *IOP Conference Series: Earth and Environmental Science*. **12** 012071
- [3] Li J, Yu J and Wu Y 2010 3D unsteady turbulent simulations of transients of the Francis turbine. *IOP Conference Series: Earth and Environmental Science* **12** 012001
- [4] Fortin M, Houde S and Deschênes C 2014 Validation of simulation strategies for the flow in a model propeller turbine during a runaway event *Proc. of 27th Symp. of Hydraulic Machinery and Systems* Montreal Canada
- [5] Nicolle J, Morissette J F and Giroux A M 2012 Transient CFD simulation of a Francis turbine startup *IOP Conference Series: Earth and Environmental Science* **15** 062014
- [6] Casartelli E, Mangani L, Romanelli G and Staubli T 2014 Transient Simulation of Speed-No Load Conditions With An Open-Source Based C++ Code *Proc. of 27th Symp. of Hydraulic Machinery and Systems* Montreal Canada
- [7] Stens C and Riedelbauch S 2015 CFD simulation of the flow through a pump turbine during a fast transition from pump to generating mode *Cavitation and Dynamic Problems: 6th IAHR Meeting of the Working Group* ed A Lipej A and S Muhic (Ljubljana, Slovenia) p 323