Numerical flow simulation of pump stability for pumped storage units with analysis and visualization of the dynamic flow patterns

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Abstract. With the increasing participation of alternative energy conversion sources in the energy market, the demand for energy storage and grid regulation services keep steadily growing. Pumped storage stations stand out currently as the most suitable and effective solution for these purposes. This leads to tougher requirements regarding the operation flexibility, regulation band and operating range of pumped storage units with reversible pump-turbines or ternary sets (with single or multi-stage pumps), operating at fixed or variable speed. Through the extension of the pump stable region, the pump regulation band can be extended (assuming the proper cavitation safety is provided) and in the case of reversible units the design of the hydraulic machine for turbine mode operation can be accordingly adjusted. The objective here is to model and examine details of the flow phenomena occurring as the pump goes through the unstable region at flows below minimum operating flow. A detailed understanding of the flow phenomena in this region can provide insights to improve the hydraulic design and extend the stable region of operation. Traditional model tests only provide data about the pump characteristic curve. The focus here will be transient computational fluid dynamic (CFD) simulations of a pump-turbine as it goes through the unstable low flow pumping region. The finite volume model includes the complete hydraulic machine from draft tube to spiral case and hybrid turbulence models were used, in this case scale adaptive simulation (SAS). For the analysis and assessment of the transient fluid flow at the different pump operating conditions, integral quantities such as the lift head were evaluated, the velocity field was visualized with flow streamlines, the pressure field with contour plots at the hydraulic surfaces and the vortical structures were identified, quantified and plotted using the \( \lambda \) criterion. The transient fluid flow calculation was started at an operating point in the pump stable operating range. The simulated pump flow was progressively reduced during the given numerical simulation until the set-in of the pump instability with the break-down of the lift head and pump flow continued to be reduced until the lift head started rising again. The numerical simulations were carried out for a low specific-speed reversible pump-turbine. Detailed model test results were available for the selected design and have been used to validate the numerical simulations.

1. Introduction
The demand for energy storage and grid stabilization services keep steadily growing together with the implementation of alternative energy conversion sources and the goal to maximize revenues in the energy market through flexible operation. Reversible pump-turbines and ternary sets are currently the most proved solution for these purposes. Still, these high expectations lead to tougher requirements concerning the extension of the operating range and regulation band. One
of the limitations regarding the operating limits and flexibility is the safety towards the pump stability limit. The design for improving the pump stability leads to trade-offs involving machine size, efficiency and cavitation limits. Deeper understanding of the pump stability behaviour with the associated fluid flow with the support of computational fluid dynamics (CFD) simulations can offer further possibilities to improve the hydraulic design and performance.

In ternary sets the pump is a separate hydraulic machine with fixed or regulable distributor. Reversible pump-turbines are mostly equipped with adjustable guide vanes. In case of fixed distributor, only one single pump characteristic curve is available for the hydraulic machine. With adjustable distributor, each guide vane opening leads to its own pump characteristic curve, as seen in Figures 1 and 2 for the energy coefficient and efficiency. With the regulating capability an envelope curve is defined varying the guide vane opening, in order to maximize the efficiency at each flow coefficient value as shown in the plots, taking into account the cavitation limits, the desired safety margin to the pump stability limit and smooth regulation curves.

If the pump characteristic curve is taken at the maximum achievable plant energy factor, $E_{nD_{max}}$, which corresponds to the maximum geodetic head, maximum number of units simultaneously in operation at the same penstock and lowest frequency variation range, as exemplified in Figure 3, different regions might be described.

Figure 1. Energy coefficient for individual pump characteristics and envelope.

Figure 2. Efficiency for individual pump characteristics and envelope.

Figure 3. Plant characteristics, selected pump characteristic and envelope.

Figure 4. Stability regions for individual pump characteristic.
As seen in Figure 4 there are different regions in the pump characteristic curve. The normal operating range is located at higher flow coefficient values, where the energy factor, $E_{nD}$, is monotonically increasing with decreasing flow factor, $Q_{nD}$, i.e. its derivative is strictly negative, $dE_{nD}/dQ_{nD} < 0$. The energy factor keeps increasing with decreasing flow factor till point 1, where local maximum is reached, $dE_{nD}/dQ_{nD} = 0$ and $d^2E_{nD}/dQ_{nD}^2 < 0$. At this point the flow becomes too low, leading to massive flow separation and blocking in the distributor, causing the break-down of the characteristic curve into the region where $E_{nD}$ decreases with decreasing $Q_{nD}$, $dE_{nD}/dQ_{nD} > 0$. The energy factor keeps decreasing with decreasing flow till the point 2, where a local minimum is reached, $dE_{nD}/dQ_{nD} = 0$ and $d^2E_{nD}/dQ_{nD}^2 > 0$. For lower flow factor values than at this point, the energy factor starts increasing again.

There are different definitions for the pump stability margin, depending on manufacturer and utility experience and procedures. This margin aims to account for additional safety considering the uncertainty on reservoir levels, water way head loss coefficients and eventual manufacturing tolerances. This should not be critical for machines with variable distributor opening, because reduced openings result in pump characteristic curves with large energy factors and higher stability limits, as observed for the several characteristic curves at Figure 1. For non-regulated pumps the margin has greater relevance.

The operation when the plant characteristic curve cuts the pump characteristic above point 2 results in multiple intersection points. In theory this could lead to unstable operating conditions with more than one possible operating point or difficulties at start-up. Therefore, the margin between the energy factor at point 2 and at the plant characteristic curve at the corresponding flow coefficient is often used as criterion. Less usual is the margin taking the energy factor at point 1. This is rather relevant to assess normal operation with increasing headwater and decreasing tailwater levels.

For the assessment of the flow patterns and properties along the pump characteristic curve especially at the pump stability limit region, numerical transient fluid flow simulations are performed for several operating points distributed along the experimentally measured curve of a model machine.

2. Model Test
The investigations are carried out for a reversible pump-turbine in the mid specific speed range, $n_{opt} \approx 40 \text{ min}^{-1}$ in pump mode. The hydraulic characteristics of the model machine were extensively measured at the test rig. The water passages of the model machine are homologous to those of the prototype from draft tube till spiral case. The tests were conducted according to the IEC 60193 standard [1]. The test rig and the model machine were instrumented for measuring among others differential pressure head, volume flow, rotational speed, main shaft torque, friction torque, leakage flow, water temperature, suction head, ambient pressure, guide vane opening and pressure pulsations. For the differential pressure head measurement 6 pressure transducers were
mounted at the low pressure measurement section near to the draft tube boundary and other 4 at the high pressure measurement section near to the spiral case boundary.

The measured pump characteristic curves with the energy factor, $E_{nD}$, and model efficiency $\eta_M$ as function of the flow factor, $Q_{nD}$, for different openings were of main interest for the comparison with the numerical simulations. From all measured points some were selected. They are listed in Table 1 with the flow factor, energy factor and model efficiency normalized in relation to the values at the optimum efficiency point. Operating points 1 and 2 lay on the envelope curve with varying guide vane opening. The energy factor at operating point 2 corresponded to the maximum plant value, $E_{nD,max}$. All following points kept the same guide vane opening belonging to the same individual pump characteristic curve. Operating points 5 till 13 were located around the pump stability region.

### 3. Numerical Setup

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

The computational mesh counted with around 16 million cells and part of it can be seen in Figure 5. The employed FVM made use of second-order interpolation schemes for the time dependent, convective and diffusive terms.

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

For the simulation of the transient effects in the fluid flow through the pump-turbine in pump mode, like flow instabilities, flow separation, rotating stall and rotor-stator interaction, 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 flow
separation and rotating stall, but also from flow incidence angle at the distributor and rotor-stator interaction.

The selected operating points for the numerical investigations were computed successively in a single transient simulation with variable geometry and inlet boundary conditions, avoiding the handling of several individual finite volume models involving setup and post-processing.

The guide vane opening changed between the first two operating points belonging to the envelope curve. The rotational movement of the guide vanes walls was prescribed as boundary condition and the mesh deformation was calculated based on the displacement diffusion principle. The mesh displacement was described by a diffusion transport equation in the computation field, \( \partial (\Gamma_\delta \partial \delta / \partial x_i) / \partial x_i = 0 \). To limit the mesh deformation near to the wall boundaries and preserve the mesh resolution and quality in the boundary layer, the local diffusion is inversely proportional to the distance to the moving boundary, \( \Gamma_\delta = (1/d)^K \), \( K > 0 \). It has the effect, that the mesh near to the moving wall remains almost undeformed, while the stream region absorbs the most part of the mesh deformation caused by the displacement of the mesh boundaries.

The discharge measured during the model test was used for the operating points along the simulation as shown in Figure 6 with the normalized flow factor and simulation time, \( t/T \), normalized in relation to the rotation period, \( T \). The proper evaluation of each operating point required stable transient flow patterns without undue influences related to the change of the discharge value or numerical fluctuations. The variation of the prescribed inlet flow was performed along 1 rotation. Considering the flow at the draft tube as quasi-stationary, the water volume present inside runner, guide vanes, stay vanes and spiral case domains took approximately 9 rotations to leave the finite volume model through the spiral case high pressure section for the lower discharge value. Latest after this amount of rotations the flow achieved stable transient conditions, after which additional 3 rotations were computed for the correct assessment of each operating point.

Since several time steps had to be computed until stable flow conditions were reached without contributing with data for the post-processing, the time step size was varied within each operating point as seen in Figure 7 with the objective to reduce the computational time. The time step size was smoothly varied to avoid numerical oscillations. Its size was increased for the intermediary iterations and reduced for the time steps to be post-processed, offering adequate time resolution.

Reliable turbulence models are of main importance for the precise calculation of transient fluid phenomena such as flow separation and rotating stall. For the numerical simulations of the

Figure 6. Flow coefficient variation along operating points and time.

Figure 7. Time step size variation within operating point.
transient fluid flow through the pump-turbine, the scale adaptive model (SAS) from Menter and Egorov [2] was the chosen turbulence model. As investigated by Magnoli [3, 4], Magnoli and Schilling [5, 6], Hasmatuchi [7], Benigni et al. [8] and Wunderer [9], hybrid turbulence models deliver accurate numerical solutions for the transient fluid flow through hydraulic machines with currently acceptable computational costs.

Figure 8 brings the hybrid turbulence model blend function distribution calculated for the first operating point plotted at the conformal plane \( v = 0.50 \). When the blend function assumes values close 0 the turbulence model behaves similar to URANS. With values approaching 1 the LES behaviour is active. In the figure it can be observed that in almost all the complete computational domain the mesh density and flow properties allowed the hybrid turbulence model to assume the LES behaviour, being adequate to properly simulate the transient fluid flow phenomena.

The turbulent flow viscosity, which brings the contribution of the turbulence modelling to the momentum transport equations, can be found in Figure 9. Its normalized distribution, \( \mu_T/\mu \), for the first operating point at the conformal plane \( v = 0.50 \) is shown. The order of magnitude of the turbulent viscosity was adequate, based on the investigations from Magnoli [3], to avoid artificial numerical dissipation, which could reduce the accuracy in the simulation of the transient effects.

The transient simulation of each operating point required 13 machine revolutions. The computation of one operating point took, in average, 59 hours in a Linux cluster with 72 parallel threads running at Intel Xeon X5680 processors with 3.33 GHz and 24 GB shared memory.

4. Numerical Simulation Results

The numerical simulation was carried for all operating points from 1 till 13. Monitor points were placed at the low pressure measuring section at the draft tube and at the high pressure measuring section at the spiral case at identical locations as found in the tested model machine. The head was determined in the same manner as for the model machine following the IEC 60193 [1]. The seal leakage losses and disc friction losses at the runner side chambers were considered for the efficiency and Euler head correction obtained from the numerical simulation. The leakage flow measured during the model test was applied and the disc friction was estimated with a similar procedure as done e.g. by Schilling [10].
The energy factor variation along the operating points and time can be found in Figure 10. During the imposed variation of the inlet discharge between the operating points, the flow was decelerated and the pressure as result of the momentum conservation. Fluctuations of the instantaneous head were observed along the complete simulation time and were related to the dynamic fluid flow effects taking place in the hydraulic machine.

With the decreasing discharge factor along the operating points, the energy factor first increased as expected in the stable pump operating range, until the maximum energy factor of the pump characteristic curve was nearly achieved at operating point 6. After this point the break-down of the pump characteristic curve took place and the unstable region was entered, as seen with the lower value of the energy factor at operating point 7. Increasing fluctuations of the energy factor could be partly observed as the unstable region was approached and became clearly pronounced after the pump characteristic break-down in the unstable region. The increase of the energy coefficient fluctuations was caused by the dynamic fluid flow effects arising from the reduced discharge, especially from the flow separation and defect flow incidence angle, being also the reason for the pump characteristic break-down.

The time averaged value of the energy factor was computed for the last three post-processed rotations of each operating point, allowing the numerical determination of the pump characteristic curve. The numerical results could be compared with the experimental measurements regarding the energy factor and model efficiency as function of the discharge factor as done in Figures 11 and 12. The maximum absolute error for the simulated head was 1.8% and 1.1% for the efficiency. Both calculated quantities, $E_{nD}$ and $\eta_M$, showed tight qualitative agreement with the experimental values with the same curve shapes.

The calculated efficiency curve could accurately reproduce the measured values also with the proper prediction of the efficiency break-down in the pump unstable region.

The simulated energy factor curve could predict the pump lift head break-down and the unstable region. However, the point, where the pump lift head occurred, differed between the CFD simulation and the model measurement. According to the experimental results, the pump characteristic break-down was found after operating point 5 at $Q_{nD}/Q_{nD_{opt}} = 0.7645$. In the numerical simulation the break-down took place after operating point 6 at $Q_{nD}/Q_{nD_{opt}} = 0.7232$. The numerically calculated pump characteristic remained longer stable with decreasing volume flow. The reason for this difference was probably the difficulty in measuring the exact peak of the energy factor curve during the model test. The massive flow separation caused by the reduced discharge is responsible for the break-down of the pump lift head. During the experiments oscillations arising from the dynamic fluid flow effects in the hydraulic machine itself, external
test rig fluctuations and governor parameters led the model machine to oscillate around and across the stability threshold. The result was successive massive flow separation followed by reattachment as the operating point oscillated, leading to earlier pump instability during the experiment. In the numerical simulation these external influences were not present, allowing the pump characteristic to exhibit the stable behaviour through wider discharge factor range.

Figure 13 brings the flow pattern evolution along the operating points as function of decreasing $Q_{nD}$. The projected velocity streamlines are seen at the conformal plane $v = 0.50$ for selected operating points at arbitrary time instants. In the stable region between operating points 1 and 6 smooth streamlines were observed. With decreasing discharge flow separation could be observed at the guide vanes and stay vanes at operating point 7. This was due to the defect incidence angle resulting from the reduced flow, to the curvature of the hydraulic surfaces and to the adverse static pressure gradient with the expanding vent opening in flow direction through guide vanes, stay vanes and volute. With further decreasing volume flow, separation also occurred in the runner as seen at operating point 10. The vortices arising from these conditions dissipated energy transmitted from the impeller to the fluid and blocked part of the hydraulic passages. The visualization of the flow patterns could relate the flow separation to the pump instability.

The total pressure distribution along the operating points with decreasing discharge can be seen in Figure 14 at arbitrary time instants. The difference of the local total pressure in terms of head, $p_T/(\rho_M g_M)$, to the operating point lift head, $H_M$, was converted to the local energy factor variation, $\Delta E_{nD}$, and normalized in relation to $E_{nD_{opt}}$. The energy transmission from the impeller to the fluid could be clearly seen with the continuously increasing total pressure in the runner. At operating point 1 in the stable operating range, the total pressure distribution was homogeneous through all guide vane and stay vane passages and volute with moderate losses. With decreasing discharge at operating point 6 the total pressure distribution became less homogeneous. The losses at the distributor were kept moderate, as long as no significant separated flow regions were present. At operating point 7, just after the pump characteristic break-down, the flow separation became massive. Some distributor channels were blocked by the resulting fluid flow vortical structures. The losses increased significantly, as indicated by the local total pressure drop, especially at the distributor but in the spiral case as well. The blocked channels led to strongly inhomogeneous flow field and rotating stall, as described by Giese, Jung and Hassler [11]. The flow pattern remained similar for decreasing discharge till
Operating point 1,  
$Q_{nD}/Q_{nD_{opt}} = 0.8830, \ E_{nD}/E_{nD_{opt}} = 1.0565$

Operating point 6,  
$Q_{nD}/Q_{nD_{opt}} = 0.7232, \ E_{nD}/E_{nD_{opt}} = 1.0990$

Operating point 7,  
$Q_{nD}/Q_{nD_{opt}} = 0.6852, \ E_{nD}/E_{nD_{opt}} = 1.0948$

Operating point 8,  
$Q_{nD}/Q_{nD_{opt}} = 0.6652, \ E_{nD}/E_{nD_{opt}} = 1.0945$

Operating point 10,  
$Q_{nD}/Q_{nD_{opt}} = 0.6053, \ E_{nD}/E_{nD_{opt}} = 1.0973$

Operating point 13,  
$Q_{nD}/Q_{nD_{opt}} = 0.5138, \ E_{nD}/E_{nD_{opt}} = 1.1071$

**Figure 13.** Operating points and instant projected streamlines at conformal surface $v = 0.50$. 
Operating point 1,
$\frac{Q_{nD}}{Q_{nD,\text{opt}}} = 0.8830$, $\frac{E_{nD}}{E_{nD,\text{opt}}} = 1.0565$

Operating point 6,
$\frac{Q_{nD}}{Q_{nD,\text{opt}}} = 0.7232$, $\frac{E_{nD}}{E_{nD,\text{opt}}} = 1.0990$

Operating point 7,
$\frac{Q_{nD}}{Q_{nD,\text{opt}}} = 0.6852$, $\frac{E_{nD}}{E_{nD,\text{opt}}} = 1.0948$

Operating point 8,
$\frac{Q_{nD}}{Q_{nD,\text{opt}}} = 0.6652$, $\frac{E_{nD}}{E_{nD,\text{opt}}} = 1.0945$

Operating point 10,
$\frac{Q_{nD}}{Q_{nD,\text{opt}}} = 0.6053$, $\frac{E_{nD}}{E_{nD,\text{opt}}} = 1.0973$

Operating point 13,
$\frac{Q_{nD}}{Q_{nD,\text{opt}}} = 0.5138$, $\frac{E_{nD}}{E_{nD,\text{opt}}} = 1.1071$

**Figure 14.** Operating points and instant normalized total pressure at conformal surface $v = 0.50$. 
operating point 13. At this operating point the losses were still high, but the flow distribution in the spiral case became more homogeneous allowing the head to slowly increase again.

As observed with the local total pressure distribution, considerable losses also appeared in the spiral case with reduced discharge. Although the flow in the volute seemed to be mainly smooth judging by the projected velocity streamlines at the mid conformal plane, it presented complex patterns including a multitude of eddies and vortical structures. For the flow visualization, the $\lambda_2$-criterion of Jeong and Hussain [12] allowed the identification of the vortex cores. This criterion relies on the second eigenvalue of the tensor $S_{ik}S_{kj} + \Omega_{ik}\Omega_{kj}$, i.e. $\det [(S_{ik}S_{kj} + \Omega_{ik}\Omega_{kj}) - \lambda\delta_{ij}] = 0$, $\lambda_1 \geq \lambda_2 \geq \lambda_3$. The condition $\lambda_2 < 0$ guarantees a local pressure minimum, being derived from the vorticity transport equation neglecting unsteady straining and viscosity. Isosurfaces of $\lambda_2$ characterize the vortices.

Figure 15 shows the $\lambda_2$ isosurfaces for operating point 1 at an arbitrary time instant. Numerous transient eddies, which could be computed with the SAS hybrid turbulence model, were distributed over the flow. Nevertheless, at a single arbitrary time instant no coherent vortical structures could be identified. This was due to the numerous small eddies resolved in the flow field. Nonetheless, the time-averaging of the flow revealed the vortices in the spiral
case as depicted in Figure 16 for operating point 1. Vortical structures describing helicoidal patterns, apart from the self-rotation around themselves, and extending through the spiral case could be identified. They were originated at the stay vane channels near to hub and shroud. The helicoidal flow pattern could also be identified by the intersection of the streamlines with the volute exit section in the figure. The identification of these vortices characterized the three-dimensional flow pattern in the spiral case. Associated to the several eddies, they contributed to the head losses in the spiral case.

5. Conclusion
The numerical simulation of the pump stability for a pump-turbine in the mid specific speed range offered close agreement to the experimental model test results. Operating points distributed along the pump characteristic curve in the stable operating range and at the pump unstable region were numerically computed. With adequate finite volume models and hybrid turbulence models it was possible to accurately predict the pump characteristic curve, including the break-down point and unstable operating region. This capability offers the possibility to numerically investigate and improve the pump stability limit for different design variants.

The deeper and further understanding of fluid flow leading to the pump unstable operation was also possible. Dynamic flow patterns found at the pump unstable region could be identified through the numerical simulations. With decreasing discharge, flow separation in the distributor and effects related to the defect flow incidence angle could be captured by the computational simulations. Rotating stall, helicoidal vortices in the spiral case and small eddies could be revealed with the post-processing of the numerical results.

Further applications of the employed numerical finite volume model and calculation procedure could be the prediction and improvement of pressure fluctuations at different machine locations and simulation of off-design conditions such as pump zero-discharge among others.

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