A theoretical, numerical and experimental analysis of S-shape instabilities in reversible pump-turbines: Resultant strategies for improving operational stability

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Abstract. In order to be able to draw conclusions on preventing S-shape instabilities in reversible pump-turbines during normal operation, it is important to fully understand their complex nature. In particular for the design, it is essential to predict occurring instabilities as accurate as possible and to fathom their underlying mechanisms properly. Therefore, the first section of this work incorporates a short CFD study based on the design of a reversible pump-turbine scale model. The numerical investigations are done according to the methodology represented in [7], comprising unsteady flow simulations under steady and transient operating conditions. The hydraulic machine is spatially discretized in model size. The turbulence modeling is based on the Explicit Algebraic Reynolds Stress Model. Related measurements are executed on a test rig at ASTROE (Hydraulic Engineering and Laboratory) in Graz, Austria, providing experimental data of integral quantities at different guide vane openings. Besides the CFD study, this work mainly aims at comprehensively analyzing underlying mechanisms of S-shape instabilities in reversible pump-turbines. An analytical description of flow phenomena is intended to provide a better understanding of mechanisms such instabilities are based on. A synthesis results in potential strategies for improving operational stability at individual operating points.

1. Introduction

Since the share of volatile energy sources like wind and sun has continuously been growing during the last decade in the mix of power generation, reliable energy storage has become increasingly important in terms of stable energy supply. The discrepancy between energy consumption and random generation encouraged energy companies to realign their operational strategy towards load balancing. The need to extend storage capacity in particular regions is evident, but technologies additionally have to fulfill challenging requirements with regard to reliability, operational safety and response time. Reversible pump-turbines, for example, are a proven technology and are capable of balancing frequency fluctuations in the electric grid within a reasonable time period. As the electricity price is mainly determined by the supply and demand ratio, plant operators may be forced to operate their hydraulic machines at off-design conditions unless the price exceeds a certain limit. Where years ago purely peak efficiency and cavitation limits predominated, today calls for proposals additionally demanding continuously expanding operating ranges, restrictions in terms of maximum pressure amplitudes and stability...
margins in pump and turbine mode. In order to verify whether all customers’ requirements are met, a variety of preliminary investigations has to be done before commissioning. This procedure incorporates a precise prediction of flow phenomena even at off-design operating conditions to obtain a comprehensive understanding of physics within the hydraulic machine. Model tests are quite expensive and thus, CFD has been forced to become a comparatively economical alternative to experiments. The combination of rising computer performance and cutting-edge tools facilitates a better prediction of extraordinary operational scenarios than in the years before. Although numerical methods and computing power have been continuously improved, computational costs are still the major factor limiting the usability of CFD. It is, therefore, a key issue to keep a balance between resolving and modeling or, in other words, between result’s accuracy and computational efforts.

In general, reversible pump-turbines are designed as pumps since decelerated flow rather tends to detach than accelerated flow. The pump design is thus more sensitive to flow separation than a turbine’s. The interaction of local flow separation and secondary flow may arouse instabilities [3]. In turbine mode, such instabilities are called “S-shape” and are indicated by a positive slope in the $Q_{ED} - n_{ED}$ or $T_{ED} - n_{ED}$ diagram, see figure 1. Relevant dimensionless coefficients read:

$$Q_{ED} = \frac{Q}{D_1^2 \sqrt{gH}}, \quad n_{ED} = \frac{nD_1}{\sqrt{gH}} \quad \text{and} \quad T_{ED} = \frac{T}{\rho D_1^3 gH}. \quad (1)$$

Further important dimensionless numbers represent the flow coefficient $\psi$, the pressure coefficient $\phi$, the Thoma number $\sigma$ and the normalized pressure $\tilde{C}_p$:

$$\psi = \frac{gH}{0.5 \cdot u_1^2}, \quad \phi = \frac{c_{m1}}{u_1}, \quad \sigma = \frac{NPSH}{H} \quad \text{and} \quad \tilde{C}_p = \frac{p - p_0}{0.5 \cdot \rho u_1^2}. \quad (2)$$

S-shape instabilities induce diametrically acting pulsations of pressure and discharge which even excite self-oscillation if their frequency corresponds to the resonance frequency of the entire hydraulic system. In that state, the energy input exceeds dissipation or damping. In the worst case scenario, the current operating point switches back and forth between turbine and reverse pump mode which strongly increases the potential risk of occurring water column separation. Several contributions have already been published about this topic, see [1, 2, 5, 9, 11, 12].

Since the design of a reversible pump-turbine is a compromise between pump and turbine, it is challenging to guarantee a high efficiency and a completely stable operating range at once. Taking the entire hill chart into account, S-shape instabilities cannot be entirely suppressed – at least not as long as a significant drop in efficiency must be avoided. Major efforts have been made either to reduce their intensity or to shift their onset towards operating conditions where hydraulic machines are typically not in operation. However, there are certain scenarios at which S-shape instabilities might cause problems, namely during start-up or in case of load-rejection.
2. Numerical Setup
The following investigations are executed based on the experimental data of a reversible pump-turbine model, see figure 2. Corresponding machine specifications are given in table 1. The applied CFD methodology is adopted from a previous sensitivity study introduced by Lenarcic et al. [7]. The spatial discretization comprises all relevant components of the entire hydraulic machine, namely a spiral case (SC), stay vanes (STV), guide vanes (GV), a runner (RN) and a draft tube (DT).

Table 1. Main specifications of the reversible pump-turbine model.

| $n_{q}$ | $N_{STV}$, $N_{GV}$ | $N_{RN}$ |
|--------|-----------------|---------|
| min$^{-1}$ | - | - |
| 34.3 | 22 | 9 |

Corresponding attributes of grid quality are shown in table 2. Therein, a determinant of 1 characterizes fully orthogonal cells. The SC and STV grids are merged together without any interface in between. All remaining components are interconnected via General Grid Interfaces, providing transient rotor-stator interactions between rotating runner and adjacent stationary domains.

Table 2. Grid quality of the reversible pump-turbine model.

| Determinant | Min. angle | Max. angle | No. of nodes $|^{10^{6}}|$ | $y_{mean}@SNL$ |
|-------------|------------|------------|----------------|----------------|
| SC-STV | 0.11 | 5.5 | 175.9 | 2.6 | 7.6 |
| GV | 0.38 | 22.0 | 157.3 | 3.0 | 3.7 |
| RN | 0.23 | 7.4 | 172.9 | 2.5 | 5.3 |
| DT | 0.68 | 39.3 | 146.2 | 0.3 | 18.1 |

The inlet boundary condition defines a discharge and comprises a bulk velocity profile, a turbulence intensity of $I_T = 0.05$ and a turbulent length scale of $L_t = 0.01m$. The rotational speed of runner $n$ is constant for all computations. At the outlet, a so-called opening boundary condition is prescribed, incorporating an area-averaged static pressure of $p_{mean} = 0$ and allowing a re-entering flow that reduces numerically oscillations at the outlet. The extensions at the in- and outflow keep the influence of the boundary conditions moderate. The commercial code Ansys CFX 18.1 is used and combined with the Explicit Algebraic Reynolds Stress Model (EARSM). The high-resolution scheme is applied to interpolate advection terms, whereas the upwind difference scheme describes turbulence transport through individual cell boundaries. The temporal discretization is set to 4 time-steps per guide vane pitch. An abortion criterion for the averaged values of residuals limits the maximum number of inner iteration loops to 8-10 steps. Unsteady CFD simulations are performed under steady and transient operating conditions, starting from best efficiency point, through speed-no-load (SNL), down to operate at zero-discharge (Q0). The maximum simulation duration of the unsteady simulations under steady operating conditions corresponds to 15 runner revolutions. For the purpose of significance, quantities are averaged over the last 6 runner revolutions before starting evaluation.
3. Results
3.1. Model Conception
S-shaped characteristics basically correspond to stable pump characteristics, however, with one exception: They occur in turbine (brake) mode and are thus undesirable. In order to prove this assumption, the phenomena must be observable along the S-curve which are attributable to the operating principle of a reversible pump turbine acting in pumping mode. For this reason, unsteady CFD simulations are performed under steady and transient operating conditions at the guide vane opening angles $\alpha_0 = 9^\circ$, $\alpha_1 = 12^\circ$ and $\alpha_2 = 20^\circ$. Related results are shown in figure 3, revealing characteristics from the best efficiency point, through speed-no-load, down to zero-discharge. While the $\alpha_2 = 20^\circ$ characteristic is reasonably predicted, the S-shape is underestimated at $\alpha_0 = 9^\circ$ and $\alpha_1 = 12^\circ$. This might be attributable to a lack in numerical accuracy causing inadequate flow conditions through the guide vane channels. As a consequence, exciting phenomena remain off and the slope of characteristic becomes damped. It can also be seen that the region of hysteresis constitutes a very labile part of characteristics. Incipient inter-blade vortices excite unsteady flow phenomena in the runner channels and thus, intensify scattering in integral quantities.

A comprehensive analysis identifies different types of vortical structures to be mainly responsible

![Figure 4. Position of monitoring points (MPs) for the detection of static pressure; adapted from [8].](image)

Figure 5. Normalized pressure $\tilde{C}_p$ computed along the S-curve at $\alpha_1 = 12^\circ$.

Figure 6. Normalized pressure $\tilde{C}_p$ computed along the S-curve at $\alpha_2 = 20^\circ$. 
for the onset of unstable operation. These phenomena are based on the interaction of a momentum transfer from the runner to the fluid (= pumping) and inappropriate flow conditions at the runner inlet (= a small incidence flow angle $\beta_1$). The momentum transfer originates in the runner channels, whereas the small incidence flow angle is a product of constant runner speed and low discharge. These two mentioned mechanisms control onset and intensity of S-shape instabilities. The following investigations analyze the occurring phenomena in more detail. Monitoring points (MP) are positioned within the hydraulic machine in order to detect static pressure fluctuations, see figure 4.

As can be seen in figure 5 and figure 6, S-shape instabilities originate at the runner inlet. At the positions MP1 and MP2, the ensemble average of static pressure stagnates or even increases despite a diminishing flow rate (= pumping), whereas it decreases downstream in the runner passage at MP3 (= generating). This can be led back to the fact that a rotating pump-turbine runner establishes a pressure gradient in its individual channels which counteracts the main flow through the machine in turbine mode. The correlation between runner rotation and pressure increase is given by the formulation of the radial equilibrium equation:

$$\frac{c_u^2}{r} = \frac{1}{\rho} \frac{dp}{dr}$$  \hspace{1cm} (3)

Under the assumption of a two-dimensional rigid body rotation $c_u = r\Omega$, an equilibrium condition exists at the runner inlet between radial-acting momentum of the main flow and centrifugal forces due to runner rotation:

$$c_m \frac{\partial c_m}{\partial r} = -r_1\Omega^2$$  \hspace{1cm} (4)

Equation 4 represents a condition to roughly predict whether individual parts of runner channels act in pumping or turbine mode. If the corresponding centrifugal forces exceed the radial-acting momentum of a fluid particle then it is reversed. If this effect occurs over the majority of runner inlet, then the main flow is throttled and characteristics become S-shaped. In the worst case scenario, this mechanism forces the discharge to completely change its main direction through the hydraulic machine so the reversible pump-turbine starts operating in reverse pump mode. The rotating blockage drags the surrounding fluid which is why the main flow starts accelerating in the vaneless space. During synchronization, the flow enters the vaneless space at small incidence. Together with the runner blockage this forms a symmetric vortex toroid in the vaneless space that additionally obstructs the main flow. Its effect is indicated by a different ensemble average of static pressure between MP0 and MP1 along the S-curve at $\alpha_1 = 12^\circ$, see figure 5. The vortex formation is supported by the inflow through the guide vanes and propagates according to the fluid movement in a bend. As it requires sufficient space to fully develop itself – in this context – a distinction must be drawn between small and large guide vane openings. In case of large opening angles (e.g. $\alpha_2 = 20^\circ$), the momentum exchange between guide vanes and fluid is even more intensified than at small openings (as the solid guide vanes block the circulating flow) so the vortex toroid becomes completely interrupted. The inflow into the runner is more inclined which in turn strengthens the runner blockage along the S-curve. As a result, the ensemble average of static pressure increases at the runner inlet (MP1) and approaches the same value as observed in the guide vane channels at MP0, see figure 6. All these effects gain a high level of flow losses within the hydraulic machine which can be quantified by the entropy production rate based on the contribution of Herwig et al. [4] and Kock et al. [6], see figure 7. The corresponding results reveal different types of vortex formations occurring at unstable operation: The development of inter-blade vortices indicates pumping in individual runner channels [10]. They occur once an equilibrium condition exists between radial-acting momentum of the main flow and counteracting pressure gradient due to runner rotation, see equation 4. In addition, a cross-channel vortex is
caused by a momentum transfer from the runner to the fluid which additionally obstructs the main flow at the runner inlet. These phenomena together develop pulsating backflows into the vaneless space, causing the mentioned vortex toroid at small guide vane openings and the high level of entropy production rates at large openings. It therefore requires a radial-acting momentum through the guide vanes that is capable at overcoming the stable structure of the vortex toroid in the vaneless space and the pumping phenomena in the runner.

In order to fathom the underlying mechanisms of S-shape instabilities in an analytical way, an approach has been developed which approximates the pressure coefficient at zero-discharge for different guide vane openings:

\[
\psi_{Q0} = f \left( \frac{D_2}{D_1}, \frac{D_0(\alpha)}{D_1}, \zeta \right)
\]  

(5)

representing an analytical model to describe the equilibrium condition between reverse pump and turbine brake mode. It is mainly affected by the capability of a rotating pump-turbine runner to build up a pressure increase by itself. Its value depends on the runner diameter ratio

\[
\frac{D_0(\alpha)}{D_1}, \zeta
\]
$D_2/D_1$, the guide vane design ($\zeta$) and its opening ratio $D_0(\alpha)/D_1$. The $\zeta$-function is thereby based on empirical data. As can be seen in figure 8, the pressure increase reaches its maximum value at a fully closed wicket-gate and is continuously reduced towards large guide vane openings. A larger distance between runner and guide vanes therefore causes a higher pressure increase. This fact corresponds very well to the assumption that the vortex toroid requires a certain space to become fully stable in space and time.

If the analytical model derived from equation 5 is correct, then a correlation exists between the runner’s tendency for pumping and the intensity of incipient S-shape instabilities in turbine mode. According to its formulation, the related slope of turbine characteristic is a function of the equilibrium condition at zero discharge and not vice versa, see figure 9. This is a very important aspect in terms of transient responses during an emergency shutdown. The main question for the countermeasures needed to be clarified is therefore, if it is possible to stabilize S-shaped characteristics not only at speed-no-load for synchronizing, but also in a transient way down to reverse pump mode without weakening pump stability.

3.2. Resultant Strategies for Improving Operational Stability

As already mentioned, S-shape instabilities are accompanied by different vortex formations occurring in the runner channels and the vaneless space. They represent an additional obstruction and prevent the flow to soundly pass the hydraulic machine. However, if the radial-acting momentum of the inflow through the guide vanes exceeds the counteracting pressure increase, then these vortex formations become interrupted and the blockage collapses. This condition remains intact as long as the kinetic energy of the main flow suffices to overcome the counteracting effects. Such a mechanism is reached by means of so-called Misaligned Guide Vanes (MGVs), a proven technique to stabilize unstable characteristics. They interrupt the circulating flow in the vaneless space, partially bypass fluid to an adjacent guide vane channel and thus, improve flow conditions downstream at specific runner passages according to the velocity triangles in figure 10. A detailed study about their operating principle is given by Lenarcic et al. [8].

However, another possibility to improve operational stability is to vary the rotational speed during start-up. Although variable-speed machines are mainly used to adjust power input, this specific machine type is capable of synchronizing at operating conditions where characteristics are typically less or not S-shaped. Its operating principle is described in figure 11. Reducing the rotational speed (lower $n_{ED}$) shifts the operating range towards smaller guide vane openings.

![Figure 10. Operating principle of Misaligned Guide Vanes (MGVs); adapted from [8].](image-url)
Figure 11. Operating principle of a variable-speed machine. Reducing the rotational speed ($n_{\text{var}} < n_{\text{const}}$) shifts the synchronization (operating range) to smaller guide vane openings where characteristics are typically not S-shaped at speed-no-load.

so that synchronization takes place in a stable manner. Compared to a fixed-speed machine, the operating range might be extended and the actual synchronization procedure is completed much faster. As the operation of variable-speed machines requires an additional static frequency converter, there still exist economic aspects, such as equipment and service costs, which discourage customers to generally deploy this specific configuration in pumped-storage power plants.

Incipient cavitation at the low pressure side (at $D_2$) reduces the capability of a rotating pump-turbine runner to build-up a pressure increase by itself. If the cause of S-shape instabilities is based on the fact that reversible pump-turbines are designed as centrifugal pumps, then all measures that reduce operational stability in pump mode might diametrically influence stability in turbine mode. As the pressure increase of a centrifugal pump collapses in case of incipient cavitation, it can be assumed that the same mechanism occurs in reverse pump mode. This shifts characteristics to higher $n_{ED}$ causing a maximum level of stabilization at the equilibrium condition between turbine brake and reverse pump mode where pumping starts dominating flow physics, see figure 12 (left). Its intensity is blended in turbine brake mode as

Figure 12. Influence of Thoma number ($\sigma$) on S-shape instabilities. Model conception (left) and experimental results at two different $\sigma$ levels (right).
pumping increasingly affect characteristics with diminishing discharge. A possible explanation therefore might be that pumping effects no longer suffice to block runner channels over a wide range of circumference. The radial-acting momentum of the main flow overcomes counteracting phenomena and characteristics become less S-shaped. The degree of stabilization is thus a function of static pressure at the runner outlet. This assumption correlates very well with the model conception mentioned above. For the purpose of confirmation, model tests are carried out on a test rig at ASTROE in Graz, Austria. Figure 12 (right) shows related results of such a change in suction conditions depending on the Thoma number $\sigma$. As therein depicted, the level of cavitation has a major impact on the slope of characteristics. A reduced Thoma number enhances operational stability particularly in regimes of dynamic processes at the transition between turbine brake and reverse pump mode which emphasizes the relevance of a reasonable choice of pressure levels in transient studies and for model acceptance tests.

The model conception introduced in Section 3.1 concludes with the assumption that a larger distance between runner and guide vanes causes a higher pressure increase at the transition between turbine brake and reverse pump mode. This in turn intensifies S-shape instabilities, see figure 9. Provided that the mentioned approach is correct, it should therefore be possible to stabilize S-shaped characteristics by lowering the diameter ratio $D_Z/D_1$ (guide vane pivot/runner inlet diameter). Figure 13 depicts the experimental data of a reversible pump-turbine model based on two different diameter ratios $(D_Z/D_1)_{small} = 1.12$ and $(D_Z/D_1)_{large} = 1.14$. It clearly shows a relation between S-shape intensity and diameter ratio, meaning that a smaller diameter ratio $D_Z/D_1$ increases operational stability significantly.

4. Conclusion

In general, reversible pump-turbines are designed as centrifugal pumps which is why a pressure gradient incessantly acts against the main flow in turbine mode. The runner already ”pumps” at the best efficiency point, however, pumping intensifies from turbine mode to reverse pump mode at which the direction of the main flow through the machine becomes reversed. The intensity of S-shape instabilities is thus attributable to the runner’s tendency for pumping. This might be a reason why a high-head pump-turbine rather tends to develop S-shaped characteristics than a low-head pump-turbine. If the introduced model conception is correct, then all runner modifications that improve operational stability in turbine mode contrarily decrease stability in pumping mode. It is therefore obvious to focus the countermeasures on the symptoms (= vortices
in the runner channels and the vaneless space) and not on the causes of S-shape instabilities (=pumping).

The flow analysis revealed the existence of different vortical structures at unstable operation which all are attributable to a pumping runner. At small guide vane openings, the weak momentum exchange between guide vanes and fluid additionally causes a circulating flow in the vaneless space acting as a kind of porous wall and blocking the through flow. It collapses once the radial-acting momentum of the main flow through the guide vanes exceeds the counteracting pressure gradient caused by the rotating pump-turbine runner. At large openings, this vortex toroid becomes interrupted by the solid guide vanes. This, however, produces a much higher level of entropy in the vaneless space than at small guide vane openings.

Different techniques are available which have already been proven to be effective for compensating S-shape instabilities. Misaligned guide vanes, for example, interrupt the vortex toroid in the vaneless space and improve flow conditions downstream at the runner inlet. However, solely varying the runner speed shifts synchronization to operating conditions where characteristics are not unstable. The main disadvantage of this method is the need for an additional frequency converter. Another effective way to improve operational stability in turbine mode is lowering the level of static pressure at the runner outlet. Corresponding results obtained on a test rig at ASTROE are exhibited in this work, providing experimental data of integral quantities at different guide vane openings. Concerning this matter it could be confirmed that a reduced Thoma number enhances operational stability particularly during transient processes in case of load-rejection at the transition between turbine brake and reverse pump mode. Besides that, further model tests are aimed at examining the sensitivity of the rotor-stator distance on S-shape instabilities. The experimental results proof the assumption that a smaller ratio of guide vane pivot to runner inlet diameter ($D_2/D_1$) improves operational stability significantly. This fact agrees very well with the model conception introduced in this work predicting intensified S-shape instabilities due to a larger distance between runner and guide vanes. To summarize this work, a variety of strategies have been presented covering different mechanisms to compensate S-shape instabilities without the need for any modification to the hydraulic design.

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