Study on S-shape instabilities and pressure pulsations in reversible pump-turbines: Conflictive mechanisms behind a varying rotor-stator distance and development of a new guide vane design for improving transient behavior during turbine start-up and in case of load-rejection

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Abstract. Increasing demands towards a more flexible and reliable operation of backbone systems for electrical grid stability has been leading to an upsurge of pumped-storage in the last years. As technological requirements concerning reversible pump-turbines have been undergone substantial transformations with respect to operational stability and noise emission, hydraulic designs are prompted to change according to market needs and requirements. One of the main factors threatening the mechanical structure of components represents the unstable operation in pump and turbine mode. Instabilities are typically accompanied by an intense increase in pressure pulsations that can be a main reason for vibration issues and resonance excitations in reversible pump-turbines. Avoiding instabilities during operation is, therefore, one of the major targets for competitive designs. This paper emphasizes the relevance of a thorough analysis of S-shape instabilities and provides an explanation of their underlying mechanisms based on analytical, numerical and experimental methods. It also addresses conflictive mechanisms behind a varying rotor-stator distance that could occur in terms of operational stability and rotor-stator interaction. The introduction of a new 3D-shaped guide vane design represents a further key outcome of this study. With respect to a varying rotor-stator distance, special attention has been paid to the description of influencing factors for turbine stability and pressure pulsations using three rescaled versions of an existing guide vane design. Series of model tests have been conducted on a test rig at ANDRITZ in Linz, Austria, revealing whether a measure is competitive based on the layout of a 2x330MW variable-speed pump-turbines plant.

1. Introduction
Reversible pump-turbines undergo a plurality of load cycles during their lifetimes. While in pump mode a sequence of guide vane openings allows the pump to start-up, in turbine mode, this degree of freedom is not available. During turbine start-up, the intersection between net head and the pump-turbine’s runaway-curve defines the guide vane opening at which synchronization with the electric grid takes place. It, therefore, needs stable characteristics at speed-no-load conditions between minimum and maximum head to accomplish the start-up procedure. Depending on the
hydraulic design, the slope of characteristics might become unstable, or S-shaped, representing a positive derivative in the $Q_{ED} - n_{ED}$ or $T_{ED} - n_{ED}$ diagram. The corresponding dimensionless coefficients are defined at the IEC 60193 [5] by

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

The underlying phenomenon is called S-shape instabilities causing changes in the runner speed which can be attributed to inversely phased oscillation of pressure and flow within the hydraulic machine. If their amplitudes exceed a certain tolerance band, the governor is not able to successfully complete the synchronization (see figure 1, blue). The start-up procedure needs then to continue as long as the synchronization has not been finished. Load-rejection scenarios entail a much higher risk for failure damage than synchronization issues, because S-shape instabilities cause critical values of hydrodynamics during such transient maneuvers. A critical view needs to be taken on the dynamic values of maximum pressure in the spiral casing, minimum pressure in the draft tube, and maximum runaway speed. These values generally involve high risks and are hardly predictable prior to model tests. The maximum pressure in the spiral casing is a consequence of the dynamic water-hammer pressure. If pressure in the draft tube drops to vapor level, water-column separation may occur with a certain probability (worst-case scenario). Such an incident is depicted in figure 1 (red) representing a layout of a 2x330MW variable-speed pump-turbine plant. When the machine units 1 and 2 reject their loads successively (unit 1 before unit 2), instantaneous operating conditions cause a minimum draft tube pressure in unit 2. This is caused by the interaction of the shut-off head that unit 1 builds up in reverse pump mode, the inertia of flow in the tailrace water tunnel, and the operation of unit 2 in turbine mode. The speed rise in the penstock due to the waterhammer also increases the dynamic runaway speed of the runners and heavily stresses the mechanical structure of the rotor generator. Its highest value represents one of the key criteria for the generator design and depends, among others, on the plant layout, and the operational stability of the reversible pump-turbine. The necessity to keep hydrodynamics within certain thresholds is a prerequisite for hydraulic designs. As the rotor-stator interaction (RSI) has become increasingly important, particularly in terms

![Figure 1](image_url)

**Figure 1.** Synchronization at speed-no-load (SNL, blue); S-shape instabilities cause changes in runner speed that prevent the governor from completing synchronization. Successive load-rejection (red); S-shape instabilities cause critical values of hydrodynamics in the spiral casing and draft tube.
of vibration avoidance and noise emission, the magnitude of pressure pulsations is taken into account as well. Based on model test data, different load-rejection scenarios are checked prior to the release of the prototype’s components to ensure a safe and reliable operation of the final layout. Further details regarding S-shape instabilities and pressure fluctuations can be found in [1–3, 6–10].

The aim of this paper is to present a comprehensive analysis of underlying mechanisms of S-shape instabilities based on analytical considerations, CFD, and model test results. The CFD study follows the guideline published in [6], referring to a methodology that is suited to properly predict S-shape instabilities. All unsteady CFD simulations are executed under steady and transient operating conditions. The corresponding model tests are witnessed on a test rig at ANDRITZ in Linz, Austria, providing experimental data of integral quantities and dynamic pressure pulsations in the vaneless space for different designs. The introduction of a new 3D-shaped guide vane design, incorporating special features to improve operational stability during synchronization and transient maneuvers in turbine mode, represents one of the key aspects of this study. Using an asynchronous closing law of four individual guide vanes serves as a reference of comparison in terms of stabilization. The expected correlation between a varying rotor-stator distance and the intensity of S-shape instabilities is surveyed for rescaled versions of an existing guide vane design. The competitiveness of all designs is experimentally assessed with respect to the fulfillment of guarantees.

2. Reversible Pump-Turbine Model Machine and Numerical Setup

One reversible pump-turbine model machine is investigated in the course of this paper. Its prototype specifications are given in table 1. A pressure tap is positioned in the stationary frame of the vaneless space at the hub (see figure 7). The transient study is based on the layout of a 2x330MW variable-speed pump-turbine plant, consisting of two machine units interconnected with a single penstock and tailrace water tunnel (see figure 1). The settings of the CFD study are consistent with the guideline published in [6] comprising a methodology based on the discretization of the entire model machine (see figure 2, left). The spatial discretization includes a spiral case (SC), stay vanes (STV), guide vanes (GV), a runner (RN) and a draft

| Orthogonality | Min. Angle | No. of Nodes | y_{mean} |
|---------------|------------|--------------|----------|
| SC-STV        | 0.11       | 5.5          | 2.6      | 7.6     |
| GV            | 0.38       | 22.0         | 3.0      | 3.7     |
| RN            | 0.23       | 7.4          | 2.5      | 5.3     |
| DT            | 0.68       | 39.3         | 0.3      | 18.1    |

Figure 2. Physical model of the investigated reversible pump-turbine and its grid quality.
tube (DT). Grids of spiral casing and stay vanes are stitched together, whereas General Grid Interfaces interconnect all remaining components. Corresponding attributes of grid quality are shown in figure 2 (right). Therein, an orthogonality of 1 represents a fully orthogonal cell. The commercial CFD code ANSYS CFX 18.1 is used together with the Explicit Algebraic Reynolds Stress Model (EARSM). A Dirichlet boundary condition defines the mass flow rate, turbulence intensity \( I_T = 0.05 \) and turbulent length \( L_t = 0.01 \text{m} \) at the inlet. The rotational speed of the runner \( n \) remains constant for all computations. An opening boundary condition at the outlet incorporates an area-averaged static pressure of \( p_{\text{mean}} = 0 \). A blended second-order high-resolution scheme is used to interpolate advection terms, whereas a first-order upwind scheme describes turbulence transport. The time step size is set to 4 time steps per guide vane pitch. The CFD simulations are performed under steady and transient operating conditions at the guide vane opening angles \( \alpha_0 = 9^\circ \) and \( \alpha_1 = 20^\circ \). Monitoring points are homologous to the model machines. Flow phenomena are explicitly investigated at speed-no-load (SNL), along the S-curve, down to zero-discharge (Q0). The maximum number of iteration steps is limited as follows:

- 15 runner revolutions for the unsteady simulations under steady operating conditions
- 150 runner revolutions for the unsteady simulations under transient operating conditions

All quantities are averaged over the last 6 runner revolutions.

3. Model Conception of S-shape Instabilities

Layouts of modern pumped-storage power plants are comprehensive and demand a high degree of flexibility in operation. While years ago, peak efficiency and cavitation were emphasized from a technological perspective, these days, a flexible and reliable operation has become of utmost importance. The necessity of reversible pump-turbines to perform highly efficient as pump and turbine has always been decisive in a competitive environment. However, there typically exists a kind of trade-off between both operating modes that needs to be taken into consideration during design. The connection between pump and turbine can be led back to the reversible nature of pump-turbines. The model conception introduced in [8, 9] explains in an analytical way the imaginary connection between (reverse) pump and turbine (brake) mode. It associates

![Figure 3.](attachment:figure3.png)

![Figure 4.](attachment:figure4.png)

Figure 3. Connection between the reverse pump's shut-off head and S-shape instabilities (model conception).

S-shaped characteristics with stable pump curves and highlights reverse pumping to be mainly responsible for the emergence of S-shape instabilities. This can be led back to the fact that
a rotating pump-turbine runner builds up a pressure counteracting the main flow through the machine in turbine mode. The value of pressure increase is approximated by the formulation of the radial equilibrium equation:

\[
\frac{c_u^2}{r} = \frac{1}{\rho} \frac{dp}{dr} \tag{2}
\]

Assuming a two-dimensional rigid body rotation \( c_u = r\Omega \), an equilibrium condition between the radial-acting momentum of the main flow and the centrifugal forces due to runner rotation can be established at the runner inlet:

\[
\left| c_r \frac{\partial c_r}{\partial r} \right| = \left| -r_1 \Omega^2 \right| \tag{3}
\]

Equation 3 features a condition to roughly predict whether a runner channel acts in reverse pump or turbine mode. A fluid particle entering a runner channel is reversed if the centrifugal force (or the reverse pumping) exceeds its radial-acting momentum in turbine mode. If this happens over large parts of the runner inlet, then the main flow is throttled and characteristics become steep (see figure 4). The main flow recirculates from the runner into the vaneless space building up regions with intense turbulent mixing. As a result, characteristics become S-shaped.

Due to the nature of a rotating pump-turbine runner, reverse pumping already exists at the best efficiency point in turbine mode. This mechanism gains with diminishing discharge and forces the main flow to change its flow direction in reverse pump mode. While in turbine and turbine brake mode, the momentum of the main flow suffices to overcome reverse pumping, these effects become dominant at the vicinity of zero discharge. The intensity of S-shape instabilities, therefore, depends on the shut-off head a pump-turbine is able to build up in reverse pump mode (see figure 3). Taking this model conception into account, it becomes obvious that turbine stability is decisively affected by reverse pump operation. Therefore, the fundamental cause of

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure5.png}
\caption{Streamlines with values of velocity at \( \alpha_0 = 9^\circ \) (left) and \( \alpha_1 = 20^\circ \) (right) in turbine brake mode (S-shape); relative eddies block runner channels (a); stable vortex toroid in the vaneless space block main flow through guide vanes at small guide vane openings (b); asymmetrical flow field supports rotational instabilities to be developed at large guide vane openings which additionally increase unstable behavior (c).}
\end{figure}

S-shape instabilities originates at the runner inlet and is attributable to a momentum transfer from the runner to the fluid (= reverse pumping). Due to the small flow angle \( \beta_1 \) at the runner inlet at low discharge, the radial momentum of the main flow does not suffice to overcome the
counteracting pumping effects in the runner channels. While reverse pumping only relates to the tendency of a pump-turbine runner to convey fluid against the main flow direction, the phenomena in the vaneless space strongly depend on the interaction between runner and guide vanes. The mechanisms behind S-shape instabilities at small and large guide vane openings require to be analyzed separately from each other.

Reverse pumping and small openings combine themselves to develop a stable vortex toroid in the vaneless space that propagates according to the fluid movement in a rectangular bend (see figure 5). It is continuously fed by the inflow through the guide vanes and additionally blocks the main flow. A larger rotor-stator distance expands the vortex formation in the vaneless space as a consequence of stronger secondary flows. This intensifies S-shape instabilities. In case of large guide vane openings, the momentum exchange between runner, fluid and guide vanes is much higher than at small openings. Operating conditions at speed-no-load become incidental, whereas those in turbine brake and reverse pump mode gain in relevance. The considerable spatial, unsteady asymmetry in the flow field supports rotating instabilities to be induced in turbine brake mode. Such a phenomenon additionally strengthens unstable behavior. With regards to the model conception mentioned before, the superimposition of flow recirculation and increased rotor-stator interaction improves the pump stability in radial pumps [4], while reverse pumping contrarily deteriorates the turbine stability. Fathoming this conflictive mechanism between (reverse) pump and turbine (brake) mode has become one of the key premises for hydraulic designs in reversible pump-turbines.

As the literature [4] illustrates a combination of flow recirculation and flow interaction between runner and surrounding stator to be mainly responsible for the inception of instabilities in radial pumps it could be assumed that in reversible pump-turbines a mix of reverse pumping and RSI consistently follows this theory. However, distinctions from this mechanism have to be made regarding the influence of RSI. Once the flow-induced coupling between runner and guide vanes becomes weak, the development of stable vortex formations disproportionately compensates the diminishing pressure increase in the vaneless space of reversible pump-turbines and keeps characteristics S-shaped (or even intensifies instabilities, see figure 6).

### 4. Results

#### 4.1. Sensitivity Study on Rotor-Stator Distance

The interaction between runner and guide vanes induces regions of pressure inbetween the opposite blade rows. The interacting blade pair causes a perturbation in the flow field that

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**Figure 6.** Influence of the rotor-stator distance $D_0(\alpha = \text{const.})/D_1$ on S-shape instabilities.

**Figure 7.** Guide vane trailing edge diameter $D_0(\alpha)$ at $\alpha = \text{max.}$: Reference design GV1 (green); rescaled versions GV2 (blue), GV3 (red) and GV4 (orange).
excites oscillations in circumferential direction. The corresponding pressure wakes propagate up- and downstream through the vaneless space. They are characterized by a specific frequency depending on the blade number and rotational speed. At part-load conditions, these pressure fluctuations become more stochastic and are no longer dominated by the RSI, but by unsteady flow phenomena. For investigative purposes, three rescaled versions of an existing guide vane design have been manufactured and tested. Their turbine trailing edges (indicated by \( D_0 \)) are gradually shifted outwards, whereas the pitch circle diameter (\( D_Z \)) remains constant (see figure 7):

- GV1: Reference design
- GV2 (based on GV1): Rescale of guide vane around pivot point
- GV3 (based on GV2): Rescale of guide vane around machine axis
- GV4 (based on GV3): Rotation of pivot point around machine axis

**Figure 8.** Pressure coefficient at zero-discharge (\( \psi_{Q0} \)) for \( \alpha = 6^\circ, 18^\circ \) vs. rotor-stator distance (\( D_0/D_1 \)); experimental results.

**Figure 9.** Difference in pressure coefficient between zero-discharge (\( \psi_{Q0} \)) and speed-no-load (\( \psi_{SNL} \)) for \( \alpha = 6^\circ, 18^\circ \) vs. rotor-stator distance (\( D_0/D_1 \)); experimental results.

**Figure 10.** Turbine stability margin at synchronization vs. rotor-stator distance (\( D_0/D_1 \)); experimental results.

**Figure 11.** Pressure pulsations (PP) in pump and turbine mode vs. rotor-stator distance (\( D_0/D_1 \)); experimental results.

These rescales keep the comparison as consistent as possible. A pressure tap is positioned in the vaneless space at hub-side for all corresponding measurements (see figure 7). The variation
in $D_0/D_1$ is realized using the four related guide vane designs. Taking a closer look at the experimental results in figure 8, it can be seen that a larger rotor-stator distance causes a higher pressure increase at the transition between turbine brake and reverse pump mode (i.e. a higher shut-off head of the reverse pump). Figure 9 clearly shows a relation between S-shape intensity and rotor-stator distance, meaning that a smaller diameter ratio $D_0/D_1$ increases operational stability significantly (indicated by $\Delta \psi = \psi_{Q0} - \psi_{SNL}$). An improvement in the stability margin for turbine synchronization is also feasible by lowering the rotor-stator distance (see figure 10). Generally speaking, reducing the rotor-stator distance improves turbine stability during synchronization and in case of load-rejection. In contrast to that, a large rotor-stator distance decouples the interacting components and damps pressure pulsations in pump and turbine mode (see figure 11).

4.2. Innovative Guide Vane Design

The model conception introduced in the section before concludes with the assumption that S-shape instabilities can be attributed to a momentum transfer from the runner to the fluid (= reverse pumping) interacting with different vortex formations in the vaneless space that additionally block the through flow. This means that the main flow is able to overcome the counteracting pressure gradient once its radial-acting momentum is sufficiently strong at the guide vane outlet. A conventional guide vane has the disadvantage that its hydraulic design

![Figure 12. Conventional (top) and new 3D-shaped guide vane (bottom).](image)

![Figure 13. Arrangement of misaligned guide vanes (MGVs) and asynchronous closing law in case of a successive load rejection.](image)

is not flexible enough to meet all requirements of turbine stability, efficiency, cavitation, and regulation range simultaneously, whereas a 3D-shaped guide vane design (3D-GV), that combines individual profiles differing from each other along the channel width, is able to do so (see figure 12). However, improving turbine stability in case of load-rejection represents its main feature. This objective can be achieved by increasing the radial-acting momentum of the main flow through the guide vane according to equation 3, so that the fluid particles are able to overcome reverse pumping at the runner inlet even at low discharges:

$$\left| c_r \frac{\partial c_r}{\partial r} \right|_{3D-GV} > \left| c_r \frac{\partial c_r}{\partial r} \right|_{\text{conventional GV}} > -r_1 \Omega^2$$

(4)
An experimental comparison between a conventional guide vane design, misaligned guide vanes (MGVs), and the 3D-shaped guide vane design is depicted in table 2 and figure 14. For this purpose, an arrangement of four point-symmetrically arranged MGVs has been tested on the pump-turbine model machine (see figure 13). Starting from the full-load operating condition at a guide vane opening of $\alpha = 27^\circ$, a successive load rejection is simulated using a synchronous and an asynchronous closing law. While the guide vanes always follow the red line (synchronous closing law), the four MGVs are moved according to the blue colored line (asynchronous closing law). Focusing on transient maneuvers based on the layout of a 2x330MW variable-speed pump-turbine plant, the model test results show that the new 3D-shaped guide vane design reduces the maximum pressure in the spiral casing and the maximum closing time on the basis of a linear

**Table 2.** Relative deviations from reference values; Comparison between conventional guide vane design (= reference), conventional guide vane design using misaligned guide vanes (MGVs), and the 3D-shaped guide vane design (3D-GV).

|                          | Conventional Guide Vane | MGVs | 3D-GV |
|--------------------------|-------------------------|------|-------|
| WEIGHTED PUMP EFFICIENCY | 0%                      | 0%   | -0.05 |
| PRESSURE PULSATIONS (PUMP) | 0%                      | 0%   | +0.8  |
| WEIGHTED TURBINE EFFICIENCY | 0%                      | 0%   | +0.03 |
| PRESSURE PULSATIONS (TURBINE) | 0%                      | 0%   | +0.3  |
| STABILITY MARGIN @ SYNCHRONIZATION | 0%                      | ≥230 | ≥25  |
| MAX. RUNAWAY SPEED (TRANSIENT) | 0%                      | 0%   | 0%   |
| MAX. SPIRAL CASING PRESSURE (TRANSIENT) | 0%                      | -50  | -50  |
| MIN. DRAFT TUBE PRESSURE (TRANSIENT) | 0%                      | ≥75  | ≥75  |
| MAX. CLOSING TIME (LINEAR) | 0%                      | -50  | -50  |

**Figure 14.** Four-quadrants characteristics; conventional guide vane design (gray), misaligned Guide Vanes (MGVs, blue) and the 3D-shaped guide vane design (3D-GV, green).
closing law by more than 50%. It also enhances the minimum draft pressure by about 75% reducing the probability of harmful water-column separation during a successive load rejection scenario. The new guide vane, therefore, reaches the same level of stabilization in terms of transients as the misaligned guide vanes. The synchronization stability margin increases by approximately 25% compared to the conventional design, whereas the misaligned guide vanes gain a value of more than 230%.

5. Conclusion

This paper introduces a model conception intended to explain the underlying mechanisms of S-shape instabilities in reversible pump-turbines based on analytical, numerical and experimental methods. It concludes with the assumption that S-shape instabilities can be attributed to a momentum transfer from the runner to the fluid (= reverse pumping) interacting with different vortex formations in the vaneless space that additionally block the main flow. Model tests have been conducted to demonstrate the influencing factors of a varying rotor-stator distance on S-shape instabilities and the magnitude of pressure pulsations in the vaneless space. The experimental campaign comprises three additional rescales of an existing guide vane design. They reveal a correlation between an increasing rotor-stator distance and intensified S-shape instabilities. In contrast to that, a large vaneless space decouples the physical interaction between runner and stator components and reduces pressure pulsations significantly. In order to improve turbine stability during synchronization and in case of load rejection, a 3D-shaped guide vane design has been developed. Compared to a conventional design, various hydraulic profiles are combined in one body forming a three-dimensional shape of the guide vane. Its design increases the radial-acting momentum of the main flow through the guide vanes, so that reverse pumping can be overcome at the runner inlet even at low discharges. This improves turbine stability during synchronization and in case of load rejection, while maintaining competitive values for performance parameters such as hydraulic efficiency, pump stability, and pressure pulsations.

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