Article

Flow Deflection between Guide Vanes in a Pump Turbine Operating in Pump Mode with a Slight Opening

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Abstract: During the startup and shutdown processes of a reversible-pump turbine (RPT) working in pump mode, abnormal sounds and vibrations usually occur in the distributor when the guide vanes (GVs) are at a slight opening (max opening of about 6%). The objective of this paper is to apply a three-dimensional numerical CFD method to study the unsteady flow behavior in the guide vane region of a pump turbine operating in pump mode. The dynamic meshing technique is introduced to simulate the startup and shutdown processes, and it is shown to be critical in accurately capturing the details of the flow pattern variations. In addition, the RNG k-epsilon two-equation turbulence model is applied and the governing equations are discretized with the finite volume method. Moreover, the boundary conditions are set through the calculation of the transient process of the power station. The results show that the main flow between the GV is deflected during the startup and shutdown processes. In the shutdown process, the deflection occurs when the guide vane opening (GVO) is between 1.99 and 5.32 degrees, on average. In the startup process, the deflection occurs when the GVO is between 2.83 and 4.11 degrees, on average. In these processes, the velocity field and pressure field change dramatically. Simultaneously, the hydraulic torque (HT) on the GV has a sharp change. The abrupt change in the HT leads to vibrations and abnormal sounds.

Keywords: pump turbine; pump mode; slight opening; flow deflection; dynamic meshing technique

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1. Introduction

A reversible-pump turbine (RPT) is designed to pump water from a lower reservoir to a higher reservoir by using the surplus energy in a power grid. Furthermore, water has to go down to generate electrical energy through the RPT at peak hours or in case of an emergency. Hence, an RPT has to change working conditions between the pump mode and turbine mode [1]. In order to meet the requirements of a power grid, an RPT usually works in a low-load off-design working condition. Due to the RPT's special way of operating, it is hard to guarantee the stability of the units. Facing these challenges, many experts have made a lot of contributions. The unstable phenomena that occur in an RPT under runaway conditions (S-shaped characteristic in the turbine) are associated with fluctuations in the head and discharge of the system. This unstable behavior has been addressed by some authors [2–7] who believed that the nature of the vortex structures can be the mechanism that leads to potentially unstable characteristics.

Based on previous works, C. Gentner [8] performed a flow survey with CFD and PIV, and the results of the simulation and experiment had a great agreement, which strongly supported the earlier explanation of the mechanisms. In addition, some other authors pointed out that the reasons for the S-shaped characteristics are related to the development of rotating stall in the runner channels [9–12], or they are influenced by secondary flow in all parts of the RPT, and especially in the vaneless space [13]. Reference [14] performed

a depth study of the onset and development of the unsteady phenomena that cause the unstable behavior of a pump turbine during a load-rejection scenario with servomotor failure, and their work revealed that, only when the unsteady phenomena evolved in a fully developed rotating stall that was characterized by a well-defined frequency, the unstable behavior of the RPT in turbine mode could be limited or eliminated with proper design criteria that aimed to move this critical operating point at lower flow rates.

Unstable phenomena such as rotating stall always come with the vibration of the units [5–7], which is harmful to the stability of the turbine and other devices. In addition, especially during the startup or shutdown process, the resonance of the units also frequently occurs, which is usually caused by rotor–stator interference (RSI) [15], self-excitation [16], or pressure pulsation [17]. To improve the stability and guarantee the successful startup of the units, misaligned guide vanes (MGVs) have been implemented and have achieved the expected results [18].

However, a few years ago, in both the Tianhuangping pumped-storage power station [19] and Yixing pumped-storage power station [20], abnormal sounds and vibrations occurred at the distributor of the RPT, which was working in the pump mode, and the guide vane opening was very small. In order to resolve this problem, B. Nennemann [21] conducted detailed research about the abnormal phenomena in the Yixing pumped-storage power station by using a 2D periodical CFD simulation. Consequently, the author claimed that unexpected bi-stable flow conditions and a self-excited torsion-mode flutter vibration were discovered in the GVs. Moreover, the vibration problem could be successfully eliminated by modifying the shape of the vanes. To address the issue of Tianhuangping, H. G. Fan [22] studied the HT of the GVs of the RPT during the startup and shutdown processes in the turbine mode with a 2D periodical CFD simulation. His work indicated that a repeated reversal of fluid occurred when the GVs worked with a slight opening during the shutdown process in the turbine mode. This phenomenon finally resulted in a dramatic increase in the HT, which caused the vibrations.

The aforementioned research on vibrations and abnormal sounds occurring at the distributor of an RPT was all performed with 2D periodical CFD simulations; however, the flow in an RPT is much more complicated when the working conditions are far from the optimal operation point and the aperiodicity of the flow is obvious [23]. Therefore, a 2D periodical CFD simulation is imperfect for dealing with this issue, and it cannot capture all of the details of the flow behavior. Based on previous studies, this paper describes the performance of a 3D CFD simulation of the startup and shutdown processes of the unit of the Tianhuangping pumped-storage power station when working in the pump mode. In the present study, the dynamic meshing technique was implemented, and more details of the flow behaviors were captured.

2. CFD Methodology

The RNG k-epsilon two-equation turbulence model has high precision and accuracy in solving high-Reynolds-number clearance flows. We found out that the deflection of the main flow can also be captured with the RNG k-epsilon model. Thus, the RNG k-epsilon two-equation turbulence model is adopted in this study. Due to the large amounts of calculations, we will perform a comparative study of other turbulence models in the future.

2.1. Geometry and Mesh

In this paper, the simulation was performed by using the commercial CFD solver ANSYS Fluent. The geometry of the prototype pump turbine of the Tianhuangping pumped-storage power station was implemented, and it was the same as that in previous work [23]. A geometric model is shown in Figure 1, and the characteristics of the parameters are listed in Table 1.

Figure 1. Pump turbine's (a) geometry and (b) GV number.

Table 1. Parameters of the pump turbine.

Parameters	Value
D_2 (mm)	2045
b_0 (mm)	262
Runner blade number	9
Guide vane number	26
Stay vane number	26
Rated speed (rpm) Head (m)	500,610

The dynamic meshing technique was used to control the movement of the GVs. Considering the geometry of the GVs and the deformation form of the mesh, a prism mesh was applied in the guide vane domain and stay vane domain (shown in Figure 2). In addition, in order to resolve the near-wall flow, a boundary layer mesh was implemented. In this case, the max Y^+ was about 100 in the guide vane domain. A tetrahedral mesh was used in the rest of the domains due to the complex geometries.

Figure 2. Mesh of the guide vane domain.

To ensure the accuracy of the results and to keep the computational cost at the minimum level, five sets of different meshes with 7.5 million elements, 8.9 million elements, 9.5 million elements, 11 million elements, and 12 million elements were used in a mesh independence study. The results will be discussed in the “Mesh and Time-Step Independence” section.

2.2. Boundary Conditions

The inlet discharge laws under the 610 m head, which were obtained from a transient process simulation of the Tianhuangping pumped-storage power station (shown in Figure 3), were selected for the boundary condition settings in this study. A velocity inlet and pressure outlet were implemented. A constant runner rotational speed was set at 500 rpm, while the opening rate of the GVs was 1/60 s and the closing rate was 1/25 s. In this case, $Re = 10^6$.

Figure 3. Cont.

Figure 3. Discharge law of the transient process simulation of the Tianhuangping pumped-storage power station: (a) discharge law of the startup process; (b) discharge law of the shutdown process.

2.3. Parameter Nondimensionalization

In this paper, the Reynolds number and the dimensionless methods of the results are defined as follows:

$$Re = \frac{2\pi n R D_2}{\nu} \quad (1)$$

$$K_v = \frac{V}{\sqrt{2gH}} \quad (2)$$

$$C_p = \frac{(p - p_{ref})}{\rho g H} \quad (3)$$

$$C_m = \frac{M}{0.5\rho V_{ref}^2 A_{ref} L_{ref}} \quad (4)$$

Here, Re is the Reynolds number, K_v is the velocity coefficient, C_p is the pressure coefficient, and C_m is the moment coefficient (a positive moment direction was defined as the closing direction of the guide vane). In addition, the rated rotational speed $n = 500$ rpm, outlet radius of the runner $R = 1.0225$ m, outlet diameter of the runner $D_2 = 2.045$ m, water density (20 °C) $\rho = 998.2$ kg/m³, water viscosity (20 °C) $\mu = 1.003 \times 10^{-3}$ Pa.s, head of the pump turbine $H = 610$ m, reference pressure $P_{ref} = 0$, reference velocity $V_{ref} = 1$ m/s, reference area $A_{ref} = 1$ m², and reference length $L_{ref} = 1$ m; in this paper, $Re = 6.55 \times 10^6$.

2.4. Mesh and Time-Step Independence

Steady and unsteady calculations were performed with a constant guide vane opening (2°) with different mesh numbers and time steps. The number of nodes ranged from 7.5 million to 12 million, and the time step ranged from 1×10^{-4} to 5×10^{-3} s. The amplitude of the HT coefficient of the GVs was used as the criterion for this independence study. Figure 4a indicates that when the mesh number is greater than 11 million, the amplitude of the hydraulic torque only has a slight change; thus, 11 million mesh elements were adopted in this paper.

Figure 4. Independence study results: (a) the time step and (b) the mesh size.

A mesh with 11 million elements was used in the calculation of the time-step independence. Figure 4b shows that the amplitude of the hydraulic torque coefficient in the GVs is stable when the time step is below 5×10^{-4} s; thus, the time step was set to 5×10^{-4} s.

2.5. Numerical Model Validation

Mesh settings with openings of 3° and 6° were chosen for the numerical model validation. These meshes were both deformed from the initial mesh. The comparison between the results and the test data from the site shows that they are consistent with each other (the results for the 3° opening mesh are shown in Figure 5a; the results of the 6° opening mesh are shown in Figure 5b).

2.6. Calculation Approach

This work was carried out in the following steps, and Figure 6 shows a flowchart of our work:

- (1) **Steady simulation:** In this step, the GVO was fixed at 2° and kept constant. The discharge law of the startup process was set as the inlet velocity boundary condition. The steady calculation was considered to be converged when the HT coefficient of the GVs converged, since the HT is the most important parameter in this case.
- (2) **Transient simulation:** Before a dynamic process simulation, it is very necessary to carry out a transient simulation to develop the transient flow field. Thus, we carried out a transient simulation with a constant GVO, which was kept at 2° , and the rotational

speed (500 rpm) remained constant. The initialization was based on the result of step 1, and the boundary condition was kept the same as that in step 1. Finally, we set the transient simulation results of 10 complete runner revolutions as the initial flow field.

- (3) Startup process simulation: The result of step 2 was used for the initialization of the startup process calculation with a dynamic meshing technique in the guide vane domain. The spring-smoothing method was selected, and the GVO increased with an opening rate of $1/60$ s from 2° to 6.5° .
- (4) Shutdown process simulation: The mesh of the last time interval of step 3 (GVO of 6.5°) was set as the initial mesh for the shutdown process calculation. The discharge law of the shutdown process was set as the inlet velocity boundary condition. In addition, a transient simulation with a constant GVO (6.5°) was carried out. Ten runner revolutions were completed with a constant rotational speed (500 rpm). The result of this simulation was used for the initialization of the shutdown process calculation. In the shutdown process calculation, the GVO decreased with a closing rate of $1/25$ s from 6.5° to 1.5° .

(a)

(b)

Figure 5. Numerical model validation results for different GVOs of (a) 3° and (b) 6° .

Figure 6. Workflow of the calculations.

3. Results and Discussion

3.1. Flow Deflection in the Guide Vane Domain

Figure 7 shows an example of a typical flow state in the shutdown process ($GVO = 6.5^\circ \sim 1.82^\circ$), including the velocity contour and streamline at the $Z = 0$ plane (middle plane of the GV domain). It can be observed that the main stream in the GV domain is deflected when it passes through the gap in the GVs.

At $GVO = 6.5^\circ$, the water from the runner outlet flows out along the channel between the movable guide vanes, and its direction is consistent with the overflow channel. When the GVO decreases to 3.1° , it can be seen that the direction of the water flow is no longer towards the right side of GV #4, but is deflected by nearly 90° and flows towards the fixed guide vane. The main flow in the deflection process is very unstable. When the GVO is further reduced to 1.82° , the water flow is completely attached to the right wall of GV #5. According to the streamline distribution, a clockwise circulation is formed between the movable guide vane and the fixed guide vane.

The reverse process is found during the startup, as shown in Figure 8. At $GVO = 2^\circ$, the water flow from the runner outlet is completely attached to the right wall of GV #5. When the GVO increases to 3.3° , it can be seen that the direction of the water flow is no longer attached to GV #5, but is deflected by nearly 90° , and the water flows towards the fixed guide vane. There are vortices with opposite rotation directions on both sides of the main stream. When the GVO is further decreased to 4.45° , the main stream flows out along the channel between the movable guide vanes, and its direction is consistent with the overflow channel.

We gave definitions for the two flow types in the pump mode, which are sketched in Figure 9. Type I is defined as the flow type with a direction consistent with the incoming flow direction, while Type II is the flow type with a deflected direction that is attached to the guide vane head. This belongs to the state of the transition between the two.

Figure 7. Global velocity contours of the shutdown process (Z = 0 plane).

Figure 8. Global velocity contours of the startup process (Z = 0 plane).

Figure 9. A schematic diagram of the flow pattern definitions.

The main flow in the gap of GVs #26 and #1 was chosen to demonstrate the formation of these two flow patterns. Figure 10 shows the velocity resolution of the main flow near the tailing edge of GV #26. V_r is the radial velocity, and V_θ is the tangential velocity. In the plot, a subscript of 1 stands for a small opening, and 2 stands for a large opening. It can be seen that, near the tailing edge of the GV, the radial velocity is greater than the tangential velocity. At the small opening, the incidence angle is too large, and this large incidence angle can lead to flow separation near the tailing edge of GV #26. Then, this separation forces the main flow to turn toward the leading edge of GV #1. Because the curvature radius of the GV surface is too large, the Coanda effect [24,25] can make the main flow adhere to the surface of GV #1 so that the main flow stays in Type II. However, at the large opening, the incidence angle can be reduced with a change in the vane angle; thus, the flow separation is weakened and the main flow eventually stays in Type I.

Figure 10. Analysis of the resultant velocity: (a) small opening; (b) large opening.

3.2. Flow-Field Characteristics of Startup and Shutdown

Because the flow pattern of each guide vane has little difference, we selected some guide vanes (GV03, GV04, GV05, GV06) for the flow-field analysis. In order to analyze the characteristics of flow deflection between moving and stationary guide vanes in detail, Figure 11 shows the local velocity contours and pressure contours of the shutdown process. During this process, the GVs gradually closed from a max opening of 19.69% to a max opening of 5.91% (decrease in the GVO from 6.5° to 1.95°).

Figure 11. Cont.

Figure 11. Local velocity contours (**left**) and pressure contours (**right**) of the shutdown process ($Z = 0$ plane, $GVO = 6.5^\circ \sim 1.95^\circ$).

When $GVO = 6.5^\circ \sim 4.58^\circ$, the main flow stays in Type I. At $GVO = 6.5^\circ$, the direction of the main flow is towards the left side of the fixed guide vane directly in front of the flow channel. The velocity coefficient k_v is about 0.9. The velocity at the throat between the two GVs is the highest where k_v is about 1.2. Therefore, the pressure here is also the lowest relative to the flow field inside and outside the GV. The velocity on the right side of the main stream is very low with $k_v \approx 0.1$.

As the GVO decreases, the main flow tends to shift away from the left movable guide vane, as shown at $GVO = 4.58^\circ$. When the GVO decreases, the velocity at the throat gradually increases and the pressure further decreases, which is caused by the decrease in the cross-sectional area of the overflow.

In the range of $GVO = 4.58^\circ \sim 2.08^\circ$, the main flow is in the transition stage between Type I and Type II. When $GVO = 3^\circ$, the direction of the main flow is deflected by nearly 90 degrees. At the same time, there is a high-pressure area at the left GV, where $c_p \approx 0.8$. The trailing-edge of the main stream impinges on the other fixed guide vane (transferred from left to right) and flows along the fixed guide vane wall to both sides. With the decrease in the GVO, the deflection angle of the main flow increases further and changes to flow in the flow channel between the fixed guide vanes, as shown at $GVO = 2.08^\circ$. At this time, the pressure coefficient inside the GV increases to $c_p \approx 1.2$.

When the GVO further decreases to 1.95° , the main flow state changes to Type II. At this opening, the main stream flows around the leading edge of the GV. The throat position becomes the boundary between the high-pressure area and low-pressure area.

Moreover, the main flow stays in Type I when the GVO is more than 6.24° , and the flow pattern stays in Type II when the GVO is less than 1.95° . The pressure plot shows that the pressure at the tailing edge of the GV is higher than that at the leading edge of the GV; thus, the water coming from the runner domain pushes the GV to close, and the resultant moment should be positive. Furthermore, with the decrease in the GVO, the HT must theoretically increase.

During startup, the flow goes through the process of opposite deflection, as shown in Figure 12. With the increase in the GVO, the main flow starts deflecting from Type II to Type I. Unlike in the shutdown process, when $GVO = 2.8^\circ$, the main flow is still in Type II, while the main flow is converted to Type I.

Figure 12. Cont.

Figure 12. Local velocity contours (**left**) and pressure contours (**right**) of the startup process ($Z = 0$ plane).

The pressure plot shows that the pressure at the trailing edge of the GV is larger than the pressure at the leading edge of the GV; thus, the resultant moment on the GV helps the GV to close. In this paper, the moment that forces the GV to close is defined as a positive moment ($HT > 0$).

The behaviors of the deflections during the startup and shutdown processes are not the same. Figure 13 shows a comparison of the GVO ranges of the deflections between the shutdown and startup processes. The specific deflecting range of the shutdown process is larger than that of the startup process. For the shutdown process, the deflection angle ranges from 1.99° to 5.32° on average. For the startup process, the range is reduced to 2.83° to 4.11° on average.

Figure 13. A comparison of GVO ranges of the deflections between the shutdown and startup processes.

During the shutdown process, the GVO of the starting deflection fluctuates greatly on different guide vanes. However, for the ending deflection, the fluctuation is small.

3.3. Hydraulic Torque on the Guide Vanes

Several GVs were chosen for analysis in this part because the torque change trends of most GVs are similar.

Figure 14 presents the GVO- C_m plot for GV03, GV04, GV05, GV06, and GV10 during the simulation of the startup process. In the beginning, C_m gradually decreases with the increase in the GVO, but when the GVO goes up to 2.8° , C_m abruptly increases. This sudden increase in C_m is indicated by the red line. When the GVO is about 4.05° , which is indicated by the blue line, C_m of GV10 is larger than that of the others. As can be seen in Figure 15, when $\text{GVO} = 4.05^\circ$, the deflection speed of the main stream of GV10 is faster than that of GV06, so the torque of GV10 at this time is greater than that of the other GVs. When the GVO is around 4.22° (indicated by the green line), the flow deflection is complete, and the trend of C_m slows down. In addition, it can be seen that during the deflection process (GVO ranging from 2.8° to 4.2°), the values of C_m have phase differences between each other. This difference is caused by the unstable flow behavior of the main flow during the deflection process, as mentioned above.

Figure 14. GVO- C_m plot of the startup process.

By comparing the C_m plot with the velocity contours of the main flow (Figure 12), it can be seen that the main flow deflection and the increase in C_m occur at the same opening. When the deflection occurs abruptly, the velocity near the GV surface on the side of the stay vane decreases quickly; thus, the pressure on the surface of the GV increases. In addition, the resultant moment on the GV suddenly increases.

The GVO- C_m plot for GV01, GV03, GV04, GV05, and GV06 obtained from the calculation of the shutdown process shows that C_m increases with the gradual decrease in the GVO. However, when the GVO ranges from 5.26° to 3.15° (which is indicated by the red line and blue line in Figure 16), C_m has a turbulent trend, and the GV near the nose of the spiral case (GV01) has a large moment value. This phenomenon is due to the instability of the main flow when the GVO is in the specific range (from a max opening of 19.39% (about 6.4°) to a max opening of 5.91% (about 1.95°)) that was mentioned before. The velocity near the surface of the GV fluctuates, and this fluctuation leads to the turbulent fluctuation of C_m . When the GVO is less than 3.15° , C_m has a stable trend. Moreover, when the GVO

is about 2.2° , C_m descends sharply, as indicated by the green line in the plot. In addition, the main flow stays in Type II when C_m flattens (which is indicated by the yellow line in Figure 16).

Figure 15. Comparison of deflection speeds between the main flow of GV10 (right) and that of GV06 (left) at $GVO = 4.05^\circ$.

Figure 16. GVO- C_m plot of the shutdown process.

It can be inferred that the deflection led to a sudden change in the velocity near the GV, and the pressure on the GV's surface abruptly changed. Consequently, the change in pressure resulted in a sudden increase in the C_m ; if the transmission of the distributor cannot immediately adapt to this change, there will be a relative displacement between the shaft of the GV and the friction device, which will produce violent friction. This violent friction will lead to vibrations and abnormal sounds.

4. Conclusions

Numerical studies were presented for the startup and shutdown processes of an RPT in the pump mode when GVO is between 1.5 and 6.5 degrees. A dynamic meshing technique

was used to investigate the flow deflection of the main flow between the guide vanes. The primary findings include:

1. During the startup and shutdown processes of the RPT, flow separation occurred at the tailing edge of the GV when the GVO was less than the 6% max GVO. This separation forced the main flow to turn to the other side, and then the main flow adhered to the surface of the GV because of the Coanda effect; thus, the main flow stayed in Type II. With the increase in the GVO, the separation was weakened, so the main flow stayed in Type I. Therefore, the main flow was deflected from Type II to Type I during the startup process. During the shutdown process, the main flow was deflected from Type I to Type II.
2. When the main flow was deflected away from the surface of the GV or adhered to the surface of GV, the velocity field had an intense change near the GV. In addition, the pressure field also changed. Consequently, the hydraulic torque sharply changed at this moment. If the components of a distributor cannot adapt to this dramatic change, the violent friction will lead to vibrations and abnormal sounds.
3. During the startup and shutdown processes, the deflection of the main flow happened in a specific GVO range. The differences in the discharge laws of these two dynamic processes cause the shutdown process to have a wide GVO range for deflection.
4. Moreover, the position of the GV also has an influence on the deflection and the hydraulic torque. The deflection of the main flow at different GVs did not occur at the same GVO, and the hydraulic torque on the GV near the nose of the spiral case had the maximum hydraulic torque magnitude.

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Nomenclature:

ρ	density
V	absolute velocity
V_r	radial velocity
V_θ	tangential velocity
ε	turbulent energy dissipation rate
S_{ij}	Strain-rate tensor
μ_{eff}	equivalent viscosity coefficient
Z	axis Z
HT	hydraulic torque

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