Numerical investigations into the transient behaviour of a model pump-turbine during load rejection process

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Abstract: Three-dimensional numerical simulations were carried out to investigate the transient behaviours of a model pump-turbine during load rejection processes. The influences of guide-vane closing schemes on the transient evolutions of flow patterns, pressure fluctuations and runner forces were investigated. The results show that when the trajectory of operating point moves through the region with backflows occurring at the hub side of the runner inlet, pressure fluctuations in the vane-less space and unstable radial force on the runner become severe, which are due to the enhanced momentum exchange between the sound inflow and stalled backflows. Furthermore, closing the guide-vanes more quickly in the first reach of piecewise procedures can effectively suppress the development of backflows at the initial stage of load rejection process, resulting in significant decreases of pressure fluctuations and runner forces.

1 Introduction

Pump-storage power is widely used in frequency control and peak regulation of power grids. Francis pump-turbines are the key components of pump-storage plants [1-2]. To fulfill the regulatory demands, the pump-turbines are forced to undergo transient processes between start and stop frequently, even up to 2500 cycles per year for a unit [3]. If the grid parameters fluctuate beyond the manageable limit, the turbine-generator unit may experience unexpected load rejection. A high-head Francis pump-turbine usually suffers higher pressure fluctuations, larger dynamic forces and more severe vibrations than a conventional Francis turbine, because of its S-shaped characteristics and special geometric parameters [4, 5]. Some serious accidents in pumped-storage plants, such as rotor-stator crash and blade crack, happened suddenly during the load rejection processes [6]. These extreme unstable behaviors may be caused by sudden change of flow fields and pressure fluctuations.

In case of load rejection, the high inertia of water mass will accelerate the runner rotational speed up to 140%-170% of the rated speed, and the operating point of a pump-turbine will passes through the turbine-working, turbine-braking and reverse-pumping regions before the guide-vane being closed [7, 8]. As a consequence, the sharp increase of
water-hammer pressure occurs in the spiral case, and sudden pressure drop and surging emerge in the draft tube, which may lead to damage of penstock and collapse of vortex cavity, respectively [9-11]. A lot of studies have been conducted to investigate the impacts of S-shaped characteristics on the transient pressure during load rejection by experimental measurements, analytical calculations and numerical simulations, and some approaches have been proposed to reduce the water-hammer pressure, such as improving the closing schemes of guide-vanes and inlet valves [12-18]. However, the affecting mechanism of deteriorated flow fields on the sudden changes of pressure fluctuations within the turbine and dynamic asymmetric forces on the runner are still not clear.

These paper aims to investigate the changes of flow fields, pressure fluctuations and runner radial forces of a model pump-turbine during load rejection and reveal the mechanism of sudden changes of pressure fluctuations and runner radial forces. Here, the total load rejection processes of a pumped-storage station model with two guide-vane closure schemes were predicted by three dimensional (3D) CFD simulations method. The transient rotational speed was obtained by the hydraulic-force coupling method and the guide-vane motion was accomplished by dynamical mesh technique. The mechanism of suppressing the sudden changes of pressure fluctuations and runner radial forces during the load rejections by changing guide-vane closure schemes were explained, and the conclusions could provide a reference for pump-turbine manufacturers and operators in terms of improving the transient performance.

2 Numerical method

2.1 Pumped-storage station model

The pumped-storage station model is composed of upstream tank, penstock, a Francis pump-turbine and downstream tank. A schematic of the model is shown in figure 1. The basic parameters of the pipe system have been reported by Zhang [19], and the main parameters of pump-turbine is given in Table 1. Two sets of guide-vane closure schemes were shown in figure 2. The guide-vane opening (GVO) for Case1 was linearly closed within 7s; the GVO for Case2 was closed with a multi-phase. The 75% GVO was first closed within 1.2s, and then kept constant for 2.3s, finally closed within 3.5s.

![Figure 1. Pumped-storage station model](image)

**Table 1. Parameters of the model pump-turbine**

| Parameter                        | Value         |
|----------------------------------|---------------|
| Specific speed $n_s = n\sqrt{Q/H^{3/4}}$ | 29.17 m\cdot Kg |
| Runner inlet diameter $D_1$      | 0.280 m       |
| Runner outlet diameter $D_2$     | 0.1409 m      |
| Number of runner blades $z_b$    | 9             |
| Number of stay vanes $n_{sv}$    | 20            |
| Number of guide vanes $n_{gv}$   | 20            |
| Rated rotational speed $n_0$     | 1000 r/min    |
| Rotational inertia $GD^2$        | 1.6532 kg\cdot m² |
| runner inlet blade angle         | 20 degree     |
2.2 Geometry and mesh generation

Hybrid meshes were generated for different domains by software ANSYS ICEM 14.0. Tetrahedral grids were used in the spiral-case; wedge grids in the guide-vane and stay vane; structured hexahedral grids in the runner, draft-tube, penstock and water tanks. The total numbers of meshes is about 10.8 million.

2.3 Numerical schemes

3D numerical simulations were carried out by commercial software ANSYS FLUENT 14.0. Under load rejection condition, the internal flow in the high speed rotating impeller is quite unstable, which is strongly influenced by the rotor-stator interaction. Meanwhile, complex flow separations will occur at the runner inlet for the large variation of attack angle. The SAS-SST (Scale Adaptive Simulation-Shear Stress Transport) model was applied to simulate the flow field in the pump-turbine. It is a hybrid turbulence model, which includes the von Karman length-scale in the transport equation of the turbulence eddy frequency. Therefore, the SAS-SST turbulence model could provide better solutions for the off-design condition due to considering various scale vortices [20].

The boundary conditions were defined as follows: total pressure was defined at the inlet of upstream tank; static pressure was used at the outlet of the downstream tank; no-slip wall condition was used for all the walls. Fluid coupling and sliding mesh method were used to simulate the transient rotational speed during the runaway process.

For the transient simulation, the time-step was set to 1.875E-4s, the maximum number of iterations per time-step was set to 40, and the convergence criteria of residuals at each time-step were 1.0E-5. SIMPLEC algorithm was chosen to achieve the coupling solution for the velocity and pressure equations. Second order discretization schemes in time and in space.
were used.

2.4 Layout of monitor points

The time-varying pressure and velocity signals were acquired at six monitor points. They are located in the spiral case inlet (SC), the gap between the stay-vanes and guide-vanes (SG), vane-less space between runner blades (HS, MS, and SS) and guide-vanes and draft tube inlet (DT), as shown in figure 4. The three monitor points in the vane-less space are located on the shroud side (SS), mid-span (MS) and hub side (SS), respectively.

3 Results and Analysis

3.1 Transient properties during the load rejection process

During load rejection processes, guide-vanes are closed according to the prescribed sequence. The relative values of spiral case pressure, draft tube pressure, discharge, rotational speed of 3D-CFD simulation are compared with the results of 1D simulation by the code of TOPSYS, whose simulated accuracy on the macroscopic parameters of hydraulic transients has been widely verified by field tests and model experiments[14,15]. As shown in figure 5., the results of 3D-CFD method agree well with those of 1D method.

Figure 6 shows the dynamic trajectories (in red and green) traced by operating point and the static characteristic curves of model test (in black) under coordinate of rotating speed $n_{11}$ and unit discharge $Q_{11}$. The equal-opening curves were obtained from the model test. The $n_{11}$ and $Q_{11}$ were calculated by equations (1) and (2):

$$n_{11} = n_{D_{H}} / \sqrt{H}$$

(1)

$$Q_{11} = \frac{Q}{D_{H}^{2}} \sqrt{H}$$

(2)

Figure 5. Comparison of results of 3D-CFD and 1D simulations
Figure 6. Comparison of dynamic trajectories under two different closing schemes

The fluctuations on dynamic trajectories are evident due to the prominent variations of turbine operating water head. As shown in figure 6., the fluctuations for Case2 are slighter than those for Case1, indicating that the fluctuations can be diminished by changing the guide-vane closure schemes. The dynamic trajectory of Case1 has a sudden increase in fluctuations when the operating point begins to enter the S-shaped region, which means that the flow patterns inside the turbine abruptly lose the stable state and become extremely deteriorative. For Case2, more quicker guide-vane closing speed leads to higher pressure rise in the spiral case and lower pressure drop in the draft tube before $t=1.3s$. However, the increase of runner rotational speed is slower. As a consequence, the influence of water head increasing on the variation of unit speed $n_{11}$ is larger than that of runner rotational speed. Therefore, the change tendency of $n_{11}$ for Case2 is opposite compared to that for Case1 in the first guide-vane closing phase.

3.2 Flow evolution differences under different guide-vane closing schemes

The aforementioned fluctuations on dynamic trajectories are closely rated to the development of unstable flow in the vane-less space under the rotor-stator interaction (RSI) between the runner blades and guide-vanes. The variations of flow velocity in the vane-less space can reasonably demonstrate the characteristics of flow evolution during the load rejection processes. Figure 7. and figure 8. show the variations of normalized tangential $V_t$ velocity and radial velocity $V_r$ of three monitor points (HS, MS, and SS) in the vane-less space for Case1 and Case2. In addition, the Savitzky–Golay finite impulse response smoothing filter was used to extract the low frequency component of transient velocity [15]. The normalized velocities were calculated by equations (3) and (4):

$$V_t = \frac{60U_t}{\pi n_0 D_1}$$  \hspace{1cm} (3)

$$V_r = \frac{60U_r}{\pi n_0 D_1}$$  \hspace{1cm} (4)

where $U_t$ is the tangential velocity, $U_r$ is the radial velocity, $n_0$ is the initial runner rotational speed, and $D_1$ is the runner inlet diameter.

As shown in figure 7., when the runner inflow is sound, the rise of runner rotational speed increases the velocity fluctuations slightly and makes the inflow velocity angle progressively deviate from the runner blade inlet angle. This will result in the flow deviating from the curvature of the blade surface and separations growing at the pressure sides. For Case1 at $t=1.3s$, the radial velocity at hub side abruptly turns to be negative, leading to a sharp increase of tangential velocity at the corresponding location. Moreover, the velocity fluctuations become much more severe than the initial values. The negative radial velocity means emergence of backflows at the runner inlet, causing local recirculation flows in the vaneless space. At approximately $t=2.1s$, the location of backflows turns to mid-span, the intensity of velocity fluctuations gest highest and then decrease...
with lower discharge and smaller GVO. During this migration process of backflows, the radial velocity at shroud side turns to be negative in a very short time, which is marked by blue ellipse. The typical instantaneous flow patterns during the transient process for Case 1 were shown in figure 9.

The velocity fluctuations in the vane-less space under RSI should be driven by runner rotational speed, the movement of the guide-vanes and moment exchange between the sound inflow and stalled backflow. When the backflows occur at the hub side of the runner inlet, the discharge is about 80%-90% of the initial value and the radial velocity at other sections is progressively increasing, therefore the sound inflow can transfer enough energy to the stalled backflows. Although the guide-vanes are closed with decreased discharge, the abundant moment exchange and increased rotational runner speed can cause stronger RSI between runner blades and guide-vanes, resulting in higher velocity fluctuations. When the discharge is too low to provide sufficient energy to the moment exchange between guide-vane and runner flows, the intensity of RSI become attenuate.

For Case2, quicker guide-vanes closing speed make the tangential velocity increase faster, but the velocity fluctuations are very weak. When the guide-vanes stop moving at \( t=1.2 \)s, the pump-turbine enters into runaway state, the flow patterns at the runner inlet have sudden changes with the backflows occurring at the midspan instantaneously, as shown in figure 8. This phenomenon suggests that the movement of guide-vanes can dominate the development of flow patterns at the runner inlet. If the guide-vanes stop moving, the development of flow patterns within the turbine will be dominated by the inherent flow characteristics of the pump-turbine under runaway condition at the corresponding GVO.

![Figure 7. Transient variations of velocity in vane-less space of Case1](image1)

![Figure 8. Transient variations of velocity in vane-less space of Case2](image2)
The low frequency of velocity fluctuations are mainly caused by the unstable vortices in the runner channels. When the backflows at runner inlet turn to midspan, one rotating stall was established in the runner channels, as shown in figure 10. The decreasing discharge and increasing runner angular speed cause the development and enlargement of stall cell. However, after the runner rotational speed reaching maximum, the rotating stall begins to disappear as the guide-vanes is further closed.

3.3 Pressure fluctuations differences under different guide-vane closing schemes

In order better observe the variations of pressure fluctuations during transient process, the pressure fluctuations was extracted and normalized by equation (5):

$$C_p = \frac{p - \bar{p}}{0.5 \rho u_i}$$

where $p$ is the instantaneous pressure signals, $\bar{p}$ is mean pressure values, $\rho$ is the water density, $u_i$ is the tip velocity of the runner blade leading edge.

Figure 11. shows variations of normalized pressure fluctuations at different locations for Case1 and Case2. As shown in the figure, the pressure fluctuations in vane-less space between guide-vanes and runner are strongest, indicating RSI lead to largest pressure fluctuations in the pump-turbine. The time marked by the red dash line in the figure is when the backflows occurs at the runner inlet. For Case1, the amplitudes of the pressure pulsations increase linearly and slightly until the backflows occur at the runner inlet. Thereafter, the amplitudes increase significantly with low frequency signals emerging and developing. The amplitudes reach maximum at about 2.1s, when the backflows at runner inlet migrate from the hub side to the mid-span. The maximum amplitude is near 4 times larger than that of the initial steady state. As the guide-vane is closed further, the amplitude decreases gradually because the momentum exchange between guide-vane
inflows and runner backflows begin to decay with lower flow rate. For Case2, the intensities of pressure fluctuations at all locations are suppressed dramatically. Their maximum amplitudes only increase to approximately 1.2 times the initial values.

A time-frequency analysis was performed on the transient pressure fluctuations by Short Time Fourier Transform (STFT) [20]. Figure 12. shows the spectrogram of transient pressure fluctuations at the location MS in vane-less space for two the Cases. At initial conditions, the dominant frequency in the spectrogram is blade passing frequency (BPF=150Hz=18fn). Immediately after the load rejection, the amplitudes at BPF and its harmonics increase gradually with increasing runner rotational speed because of stronger RSI between the runner blades and guide-vanes. For Case1, when the backflows occur at the runner inlet (t=1.3s), a lot of low frequency signals with high amplitudes began to dominate the spectrogram. At the same time, the amplitudes at BPF and its second harmonics also have notably increases. While for Case 2, the amplitudes at all frequencies are far less than those for Case1.

Figure 11. Comparison of transient pressure fluctuations at different locations for Case1 and Case2

Figure 12. Spectrogram of transient pressure fluctuations at the location MS in vane-less space
3.4 Radial force differences under different guide-vane closing schemes

The asymmetrical flows in the turbine contribute to strong radial force fluctuations, resulting in rotor-dynamic instabilities [3, 20]. Figure 13 shows the comparison of radial forces fluctuations in x- and y-direction during the load rejection processes for the two Cases. For Case1, when there is no backflows occur at the runner inlet, the radial force fluctuations is slight and does not depend on the runner rotational speed. Immediately after backflow occurring at the hub side of runner inlet, the radial force fluctuations increased sharply (figure 13(a)). The development and growth of backflows lead to more non-uniformed flow in the runner channels, result in significant increase of the radial force fluctuations. However, for Case2, the radial force fluctuations were greatly reduced because the strong backflows at the runner inlet at the initial stages of load rejection were suppressed.

![Figure 13](image_url)

Figure 13. Comparison of radial forces fluctuations for Case1 and Case2

4 Conclusions

The transient characteristics of a pump-turbine during load rejections with two different guide-vane closure with two guide-vane closure schemes were investigated by 3D-CFD simulations. The affecting mechanism of deteriorated flow fields on the sudden changes of pressure fluctuations within the turbine and dynamic asymmetric forces on the runner are revealed. The major conclusions are as follows:

1. The rise of runner rotational speed increases the velocity fluctuations slightly and makes the inflow velocity angle progressively deviate from the runner blade inlet angle, which will result in the flow separations and backflows growing at the runner blade pressure sides.
2. The energy contained in the moment exchange between the sound inflows and stalled backflows in the vaneless space determine the intensity of velocity and pressure fluctuations under rotor-stator interaction.
3. The movement of guide-vanes can dominate the development of flow patterns at the runner inlet. Fast closing the guide-vanes at the initial stage during the load rejection process can effectively suppressed the development of unstable backflows at the runner inlet, resulting in smaller pressure fluctuations in the vane-less space and radial force fluctuations on the runner.

Acknowledgments

This work was supported by the National Natural Science Foundation of China (Grant No. 51579187) and Natural Science Foundation of Hubei Province (Grant No. 2018CFA010).

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