Numerical study of hydraulic axial force of prototype pump-turbine pump mode's stop with power down

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Abstract. Pumped storage power station is regarded as a key component in modern social power system. It usually requires pump-turbine start and stop frequently. The hydraulic axial thrust is one of the most essential effects on the stable operation during start and stop. The hydraulic axial thrusts of pump-turbine pump mode’s stop with power down based on 1D transient flow simulation and 3D CFD simulation were carried out in this paper. The hydraulic axial thrusts of two operation conditions show different trends in early stage. In later stage, results show similar characteristics with load-rejection process of turbine mode. The components of $F_z$ including the crown, band and blade reach an obvious peak value with unit operation status change. This study provides the reference for the safety and stable operation of pump storage power stations.

Keywords: Pump-turbine, Hydraulic axial thrust, Simulation, power down

1. Introduction

Pumped storage power station is regarded as a key component in modern social power system[1]. During the electricity peak, units generate power in turbine mode. During the off-peak, units work in pump mode to reserve the electricity power to potential energy. Pump storage power station plays the influential roles in power grid dispatch, which usually works with renewable energy like wind power and solar power. Based on the operating characteristic, pump-turbines usually need start and stop daily. The hydraulic axial thrust is one of the most essential effects on the stable operation during start and stop. Especially when pump mode stops with power down, the hydraulic axial thrust change suddenly which will influence the structure stability and strengthen of the bracket, axial and runner [2, 3]. It is extremely possible for a lift accident while the upward hydraulic axial thrust exceeds the unit weight. It may cause deformation of support bracket even damage while the downward thrust is big or strongly pulsating [4-6].

Many researchers have focused on the hydraulic axial thrust in hydraulic turbines in engineering and research. It’s clear that the hydraulic axial thrust is caused by the pressure difference [7-10]. Pressure in the crown and shroud leakage is usually higher than that in the runner, as the size of leakage is extremely small compared to the runner passage [4, 11, 12]. The pressure difference between pressure and suction side of blades is also one of the reason of hydraulic thrust. In the study from D.Y. Liu[11], an analytical method was presented to calculate hydraulic axial force based on the leakage size and the angle velocity.
of flow in leakage. This method can calculate the hydraulic force without complex simulation, but strictly request the ability of tests. Based on Genetic Algorithms (GA) method and the database of experiment and CFD simulation results, an hydraulic axial force formula for Francis turbine with condition parameters, runner diameter and coefficient K is presented by X. Zhao [13]. From study of Z. Li for the turbine mode shutdown [14], we found that the pressure level increases before the guide vanes with the guide vanes close, while it decreases after the guide vanes due to the water hammer effect. Based on the studies of the pressure development in the transient process, pressure distribution in conditions of start-up and load rejection changed strongly [15, 16]. In this study, we researched the hydraulic axial thrust of pump-turbine pump mode’s stop with power down based on 1D transient flow simulation and 3D CFD simulation. The development of hydraulic axial thrust was shown in this paper. The relationship between the operation parameters and hydraulic axial thrust was analysed. Finally, the mechanism of hydraulic axial force of pump-turbine was explained. This research can provide references for engineering, pump-turbine design and research.

2. Numerical Method

2.1. Method of 1D Hydraulic Transient Flow Simulation

The 1D simulation of unsteady flow in pipe is based on the continuity equation and momentum equation[17].

\[ V \frac{\partial H}{\partial x} + \frac{\partial H}{\partial t} - V \sin \alpha + \frac{a^2}{g} \frac{\partial V}{\partial x} = 0 \]  

(1)

\[ g \frac{\partial H}{\partial x} + V \frac{\partial V}{\partial x} + \frac{\partial V}{\partial t} + fV|V| \frac{2}{2D} = 0 \]  

(2)

Where \( H \) is piezometric head, \( V \) is average velocity, \( g \) is gravity acceleration, \( f \) is Darcy-Weisbach friction factor; \( \alpha \) is pipeline slope; \( D \) is diameter of pipe; \( a \) is speed of pressure pulse.

The boundary conditions including the pump-turbine, reservoir and valve is determined by the test results. Based on method of characteristics (MOC), the fundamental functions (1, 2) is closed and analytical. In this study, considering the large storage capacity, the water level of reservoirs is regarded as constant. The results of 1D simulation were used as the initial boundary condition in 3D CFD simulation.

2.2. Method of 3D Turbulent Flow Simulation

The 3D CFD simulation is based on the Reynolds averaged Navier-Stokes (RANS) equations, combined with continuity equation and momentum equation considering turbulence effect [18]. The continuity equation and momentum equation are:

\[ \frac{\partial u_i}{\partial x_i} = 0 \]  

(3)

\[ \frac{\partial u_i}{\partial t} + u_j \frac{\partial u_i}{\partial x_j} = f_i - \frac{1}{\rho} \frac{\partial p}{\partial x_i} + \nu \psi^2 u_i \]  

(4)

where \( u \) is the flow velocity, \( t \) is time, \( x \) is the coordinate component, \( f \) is the body force, \( \rho \) is the density, \( p \) is the pressure, \( \nu \) is the kinematic viscosity.
In this study, the SST $k-\omega$ model which is an eddy viscosity model was applied to close the RANS equation [19]. The water is regarded as incompressible. The boundary conditions of pump-turbine inlet and outlet were based on the 1D simulation results.

3. Computational Model and Boundary Conditions

3.1 1D hydraulic Transient Flow Simulation model
The 1D hydraulic system has been modelled with reservoir, gate shaft, pump-turbine and tank, as shown in figure 1. This system adopt the arrangement of two-units with one diversion tunnel. Since the water level of reservoirs determine the overall pressure level, there were two conditions calculated in this study to consider the extreme conditions of reservoirs. The condition parameters are shown in table 1, where $H_{\text{maxL268}}$ is condition with maximum head and lower reservoir level 268 m, $H_{\text{minL292}}$ condition with minimum head and lower reservoir level 292 m. At $t=0$ s, the power down and the pump-turbine stop suddenly with the guide vane is unable to close normally. The calculation duration is 60 s. The time step was set as 0.005s in the entire process.

![Figure 1. The schematic map of the 1D hydraulic system](image)

| Condition | Upper reservoir level [m] | Lower reservoir level [m] | Head [m] | Guide vane opening [°] |
|-----------|--------------------------|--------------------------|----------|------------------------|
| $H_{\text{maxL268}}$ | 744.46 | 268 | 485 | 15 |
| $H_{\text{minL292}}$ | 715 | 292 | 434 | 26 |

3.2 3D CFD Model and boundary conditions of flow components
In this paper, the study focuses on a prototype pump-turbine of a pumped storage power station. The 3D models of the whole flow filed were constructed, including draft tube, runner, guide vane, stay vane, volute and runner as shown in figure 2. Particularly, the runner crown and shroud leakages, seals, pressure balancer and pressure balance chamber, which have significant influence on hydraulics thrust, were accurately modelled. In following description, the combine zone of crown and shroud leakages, seals, pressure balancer and pressure balance chamber will be regarded as the runner outflow. Hybrid meshes of tetrahedral and hexahedral meshes were used to balances the memory cost and mesh quality. Based on the hydraulic efficiency and pressure pulsation, the mesh independence check is performed. The final meshes of flow filed in this study have about 6.87 million nodes and 9.89 million elements.

Based on the results of 1D hydraulic transient simulation, the 3D steady simulation of pump mode’s stop with power down condition of pump-turbine at typical time points was calculated. The boundary of draft tube inlet was set as the mass flow rate acquired from 1D simulation. The boundary of volute outlet was set as the static pressure acquired from 1D simulation. All the solid walls were set as no-slip.
General grid interfaces were set to connect and transfer data between each two steady domain. The convergence criterion was set that the RMS residual of continuity equation and momentum equation should be less than $1 \times 10^{-4}$. The hydraulic axial thrusts in this simulation include the $z$-direction force on the internal and outside surface of crown and band, and the section and pressure surface of blades. For brevity, the total hydraulic axial thrust was symbolized as $F_z$.

![Figure 2. Three-dimensional modeling and mesh schematic map of pump-turbine](image)

### 4. Results and Discussion

#### 4.1 Total hydraulic axial thrust

The 1D hydraulic transient simulation of the pump-turbine was performed in pump mode’s stop condition with power down, starting from steady conditions $H_{\text{max}}=1268$ and $H_{\text{min}}=292$. All the parameters are in relative value that $Q^*=Q/Q_r$, $H^*=H/H_r$, $H_{\text{in}}^*=H_{\text{in}}/H_r$, $n^*=n/n_r$, $A^*=A/A_{\text{opt}}$, where $Q_r$, $H_r$, $n_r$ are rated flow speed, rated head and rated rotate speed, $H_{\text{in}}$ is the pressure at inlet. Figure 3 shows the 1D transient relative value including $Q^*$, $n^*$, $A^*$, $H^*$ and $H_{\text{in}}^*$. As the guide vane is unable to close normally, the guide vane opening maintained the initial status in duration. Based on the boundary conditions acquired from 1D simulation, the 3D CFD simulation performed at typical time points in stop process. The relative value of hydraulic axial thrust of pump-turbine, $F_z^*=F_z/\rho g m$, are shown in Figure 3, where $m_t$ is the weight of the entire shaft system. In this paper, the downward of axial thrust is defined as positive.
As shown in figure 4, the hydraulic axial thrust is almost upward during pump mode’s stop with power down. Comparing the condition $H_{\text{max}} L_{268}$ and $H_{\text{min}} L_{292}$, the hydraulic axial thrust in condition $H_{\text{min}} L_{292}$ is larger, which is the result of the higher water level of downstream reservoir in conditions $H_{\text{min}} L_{292}$. At $t=0$ s, $F_z^*$ is 0.39 and 0.16 (upward) respectively in condition $H_{\text{max}} L_{268}$ and $H_{\text{min}} L_{292}$. It shows the status of hydraulic axial thrust during pump mode operation. In $0\sim4.5$ s, $Q^*$ gradually decreases to zero without power. It means the water stop flow up and it is going to flow down due to the effect of gravity. Meanwhile, $H^*$ suffers around 40% decline. In this process, $F_z^*$ of $H_{\text{max}} L_{268}$ and $H_{\text{min}} L_{292}$ shows a reverse trend with 0.15 magnitude. Under the inertia effect, the runner continues to rotate in pump mode although the water flow down during $4.5\sim15$s. Then, $n^*$ declines to zero. Obviously, the flow is extremely turbulent. $F_z^*$ of $H_{\text{max}} L_{268}$ shows a flat trend while $F_z^*$ of $H_{\text{min}} L_{292}$ increase 0.1. After that, runner rotates in turbine mode with $n^*$ increase continuously until runaway condition. Meanwhile, $H^*$ fluctuate widely with around 50% against the rated head. $Q^*$ fluctuates between positive and negative and gradually decrease. $F_z^*$ shows a strong relationship with $Q^*$, which is similar with that in load-rejection process of turbine mode.

### 4.2 Components hydraulic axial thrust

To further analyse the development of hydraulic axial thrust during stop with power down, the components of $F_z^*$ located on crown, band and blade are shown in figure 5 respectively. We can find that $F_z^*$ on crown is always positive, while $F_z^*$ on band is negative. It means the hydraulic axial thrust outside runner is larger than inside. At typical time points like $t=4.5$s, 15s, 38s and 45s, the components
of $F_z^*$ reach an obvious peak value with unit operation status change. $F_z^*$ on blade shows fluctuation between positive and negative, because the operation status of runner transform between turbine and pump modes, which causes the relative magnitude of pressure between pressure and suction surface change.

![Figure 5. Components hydraulic axial thrust](image)

5. Conclusion
To find out the characteristics of hydraulic axial force during pump mode’s stop with power down, the 1D transient hydraulic simulation and 3D CFD calculation have been performed. The hydraulic axial thrust is almost upward during the process, and the hydraulic axial thrust in condition $H_{\text{min}}L_{268}$ is larger than in $H_{\text{min}}L_{292}$. In early stage, $F_z^*$ of two conditions show different trend since the runner still rotate in pump mode. In later stage, $F_z^*$ show consistent trend with flow rate when the runner has rotate in turbine mode. The components of $F_z^*$ reach an obvious peak value with unit operation status change. However, this study have limits in terms of the transient characteristic. we can focus on the hydraulic thrust based on transient CFD simulation in whole transition. This study will be helpful for pump-turbine units in reducing the large axial force and enhance the operation stability.

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