Analysis of dynamic performance in a pump-turbine during the successive load rejection

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Abstract. Successive load rejection is one of the most crucial transient processes for calculating the maximum pressure in spiral case and maximum rotating speed of runner. One-dimensional mathematical models were established for a pump-storage power station and characteristics method was used to study the successive load rejection. The condition in which the maximum pressure occurs was chosen for three-dimensional unsteady calculation. The boundary conditions at spiral-casing inlet and draft tube outlet were determined by using the one-dimensional result. Based on numerical simulation, the axial hydraulic thrust and transient flow characteristic during the successive load rejection process were analysed. The axial hydraulic thrust during the successive load rejection process was mainly caused by the complex flow in the runner. The complex flow was affected both by hydraulic disturbance and the successive load rejection process.

1 Introduction

At present, the layout of multiple units per penstock is widely used in pumped storage power stations. That is, a main water diversion pipe is connected to multiple units through a bifurcation pipe. In the process of hydraulic transition, the units affect each other. The successive load rejection occurs when one of the units rejects the load under the layout of multi-unit with one diversion tunnel in a multi-parallel hydropower system. For pump-turbine units, there is an unstable “S” region on the characteristic curve. The units will enter the braking condition or the reverse pump condition when load rejection occurs, and the flow characteristics become unstable, especially in the case of successive load rejection.

With respect to the transient pressure in combination conditions, the successive load rejection was studied before. Many Researchers found the successive load rejection condition could make the negative pressure in the draft tube more serious than that in simultaneous load rejection. Zhang et al [1] analyzed the pressure change at draft tube under the combined condition of successive load rejection based on the different layout arrangement of water conveyance system. The research indicated that when the units in the same hydraulic unit
reject the load successively, the change of pressure drop at draft tube inlet was great in the different interval time. And the dangerous interval time was when the first unit reject load and its flow achieve zero; the pressure of the latter unit draft tube inlet would drop to the minimum. Zhang et al[2] verified the time of the flow rate through the former load rejection unit being decreased to zero could not always be taken as the most unfavorable time for the latter unit during the successive load rejection, and an analysis on the sensitivity must be made for the specific problems. Liu et al[3] studied the mechanism of rapid pressure dropping in the first turbine unit during successive load rejection. The results indicated that when the first unit was operated on the slope inflection segment of the S-shaped characteristic curve and the flow in the second unit was increasing, and the flow increase would induce a large head on the first unit that caused a rapid pressure drop at the draft tube inlet.

Other researchers found if the tailrace tunnel was short in the hydropower station and the draft tube pressure would not reduce too much. But the successive load rejection may result in high speed of the runner and larger pressure in the spiral case in a long diversion-type hydropower station [4]. A lot of calculation in pumped storage power station verified the maximum pressure at the spiral case occurring under the successive load rejection condition. Serious pressure change may cause severe fluctuation, and it was closely related to the safety and stability of the pumped storage power station. The flow characteristics and dynamics characteristics inside the turbine during the transient process should be investigate.

At present, three dimensional computation fluid dynamic simulations could well perform the inner flow distribution and pressure fluctuation characteristics. A lot of researches have been got by using the three dimensional simulation during load rejection.

Liu et al[5] performed three-dimensional unsteady method to study the transient process during load rejection. And the fluid coupling and dynamics mesh method were used to calculate the rotational speed. The result showed the larger pressure fluctuation region was closed to the runner during the load rejection and the stalls and reverse flow in the runner resulted in the decrease of the torque at turbine mode and turbine-braking mode.

Liu et al[6] analyzed the dynamic characteristics like runner rotation speed, torque and flow rate during load rejection process of a reversible pump-turbine. There were two low frequency-high amplitude frequency bands, which occur around the zero-torque condition and near the maximum reversed flow condition. Besides, the rotor-stator interaction and reversed flow played an important role in the pressure pulsation.

Fu et al[7] simulated three-dimensional transient turbulent flow in a pump-turbine during the load rejection process using the calculation method of coupling the flow with the rotor motion of rigid body. The dynamic closing process of the guide vanes was simulated with the dynamic mesh technology. The results showed that there were severe unsteady vortex flows in the vaneless space near the conditions under which the hydraulic torque on the runner equaled to zero. When the pump-turbine operated into the maximum reverse discharge condition in the reverse pump operating process, the unsteady vortex flows in the vaneless space were instantaneously impacted into the region between the guide vanes and the stay vanes by the sudden reverse flows.

At the same time, a dynamic instability in the load rejection process was presented[8]. All the performance characteristics and the pressure fluctuated sharply near the operating condition points, where hydraulic torque on the runner was equal to zero or reverse flow was
maximum at reverse pump conditions. The dynamic instability in the load rejection process was mainly caused by the vortex flow in the tandem cascades regions.

In this paper, the dynamic performances during the successive load rejection process were investigated. One-dimensional mathematical models were established for a pump-storage power station and characteristics method was used to study the successive load rejection. The condition in which the maximum pressure occurs was chosen for three-dimensional unsteady calculation, and transient flow and pressure characteristic and the axial hydraulic thrust during the successive load rejection process were analyzed.

2 Computational models and Numerical methods

2.1 One-dimensional Numerical method

Following equations are fundamental equations for one-dimensional unsteady flow in pipes.

Continuity equation:

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

(1)

Momentum equation:

$$g \frac{\partial H}{\partial x} + V \frac{\partial V}{\partial x} + \frac{\partial V}{\partial t} + f V |V| \frac{D}{D} = 0$$

(2)

where $H$ is piezometric head, $V$ is average velocity at pipe, $a$ is speed of pressure pulse, $D$ is diameter of pipe, $\alpha$ is pipe line slope, $f$ is Darcy-Weisbach friction factor, $g$ is gravity acceleration, $t$ is time, $x$ is coordinate.

The equations for calculation are obtained by characteristic method. Fig. 1 shows the computation grid, where $i$ is the serial number of calculation points and $j$ is the time step number.

$$H_i^{j+1} = \frac{(C_P + C_M)}{2.0}$$

(3)

$$Q_i^{j+1} = \frac{(C_P - H_i^{j+1})}{B} = \frac{(-C_M + H_i^{j+1})}{B}$$

(4)

Equations (3) and (4) are used for the transient calculation of pipes, reservoir and other elements.

![Fig.1. XT grid for solving single-pipe problem](image)

2.2 three-dimensional Numerical method

The three-dimensional incompressible turbulent flow is using CFD based on the N-S Equations. Following equations are fundamental equations.

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0$$

(5)

$$\frac{\partial (\rho u_i)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \mu \frac{\partial u_i}{\partial x_j} - \rho u_i u_j \right) + S_M$$

(6)

where $u$ is velocity, $t$ is time, $x$ is coordinate, $\rho$ is density, $\mu$ is dynamic viscosity, $p$ is
pressure, $S_m$ is generalized source term of the momentum conservation equation. The SST $k-\omega$ model is used as the turbulence model to close the governing equation by establishing the relationship between Reynolds stress and eddy viscosity, and the turbulence field can be solved out.

2.3 One-dimensional computational model

The following presents the one-dimensional calculation models. The one-dimensional calculation model is a long diversion-type station. It consists of the upstream reservoir, upstream surge chamber, pump-turbine units, downstream surge chamber and downstream reservoir. The hydro-power system is the layout of three pump-turbine units with one diversion tunnel, shown in Fig.2. The length of upstream pipeline is about 2253m and the length of downstream pipeline is about 1392m.

2.4 Three-dimensional computational model

The three-dimensional calculation models consist of five components: spiral casing, stay vanes, guide vanes, runner and draft tube, as shown in Fig. 3. To make a reliable computation, the clearance between the hub and the cover, and the clearance between the shroud and the bottom ring are also considered. The runner has 10 blades and the guide vane can open from 0 to 22 degree. The design flow rate of the pump-turbine at turbine mode is about 70.5m$^3$/s, the design head is 653m, and the design rotation speed is 500rpm.

The structural grids are generated in all computational domains, shown in Fig.4, and the mesh details of flow domains are given in Table.1.

2.5 Boundary conditions

The boundary conditions were established according to the one-dimensional computational data. One-dimensional mathematical models were established based on Fig.1
and characteristics method was used to study the successive load rejection. The successive load rejection occurred when 2# and 3# units rejected their load simultaneously; 1# unit rejected its load after 4 seconds. The pressure of 2# and 3# units caused by the quick closure of the guide vanes affected 1# unit through bifurcated pipes, causing the flow discharge and pressure change. The maximum pressure at 1# unit spiral case occurred at this condition. The dynamic performance of 1# unit are analyzed in this paper.

Table.1. Mesh details of flow domains

| Component       | Mesh nodes number | Mesh elements number |
|-----------------|-------------------|----------------------|
| Spiral casing   | 332712            | 355069               |
| Draft tube      | 696001            | 747442               |
| Guide vane      | 446607            | 397980               |
| Runner          | 1171402           | 6494342              |
| Stay vane       | 316064            | 274755               |
| Leakages        | 1771714           | 1549258              |
| Total           | 4734500           | 9818846              |

In this study, the water level is 776.1m at upstream reservoir and 75m at downstream reservoir. The pressure changes at 1# unit spiral casing inlet and draft tube outlet could be got by the one-dimensional computation. Thus, the total pressure and static pressure boundary conditions at the inlet of the spiral casing and the outlet of the draft tube were given respectively at three-dimensional computation. At the same time, the guide vane opening and the unit speed were also obtained by the one-dimensional computation and could be as a three-dimensional boundary condition.

In this process, several points, such as maximum speed point and minimum flow discharge point, were selected for calculating to obtain the variation of axial hydraulic thrust. The calculated working points were given at next section.

3. Results and discussion

3.1 Analysis of the one-dimensional calculation results in the successive load rejection process

Fig. 5 showed the changes of 1# unit relative rotational speed, relative flow discharge and relative guide opening in the process of successive load rejection. Fig. 6 showed the pressure changes at the spiral casing inlet and draft tube outlet. Due to the pressure wave propagation caused by simultaneous load rejection of 2# and 3# unit, the flow discharge of 1# unit increased in 4 seconds, and the spiral casing inlet pressure also increased. After 1# unit load rejection, the flow discharge decreased and as the guide vanes closed, the spiral casing inlet pressure continued to fluctuating increase. But when the reverse flow in reverse pump condition was near maximum the spiral casing inlet pressure seriously pulsated and was at minimum values.

It could be seen that there was a same variation trend between the spiral casing inlet pressure and rotational speed. When the rotational speed changed higher, the spiral casing inlet pressure also increased; otherwise inlet the spiral casing pressure was also decreased when the rotational speed changed lower.

The unit parameters are a set of comprehensive parameters which can be used to evaluate the performance characteristics of the pump-turbine. The unit flow and unit rotational speed of pump-turbine are defined as equation. (7) and (8), respectively.

\[
\eta_f = \frac{nD}{\sqrt{H}}
\]  

(7)
Where $Q_{11}$ represents the unit flow discharge, $n_{11}$ represents the unit rotational speed, $Q$ represents the flow discharge, $H$ represents the transient head of pump-turbine, and $D$ represents the nominal diameter of the runner. The 1# unit comprehensive characteristic $Q_{11}$-$n_{11}$ of the pump-turbine in the successive load rejection process was given at Fig. 7.

The 1# unit comprehensive characteristic $Q_{11}$-$n_{11}$ in the successive load rejection process fluctuated constantly. This result showed the comprehensive performance characteristics were dynamically unstable during the load rejection process. The three-dimensional calculated working points were given on the 1# unit comprehensive characteristic $Q_{11}$-$n_{11}$.

![Fig.5](image1.png)

**Fig.5.** The changes of 1# unit relative rotational speed, flow discharge and guide opening in the process of successive load rejection.

![Fig.6](image2.png)

**Fig.6.** The pressure changes at the 1# unit spiral-casing inlet and draft tube outlet in the process of successive load rejection.

### 3.2 Analysis of axial hydraulic thrust results in the successive load rejection process

Firstly, we define the relative axial hydro-thrust $f_z$ as equation (9)

$$f_z = \frac{F_z}{|F_z(t=0)|}$$

Where the $F_z$ represents transient axial hydraulic thrust in the successive load rejection process, $F_z(t=0)$ represents the absolute value of axial hydraulic thrust at the initial moment before the load rejection.
The variation axial hydraulic thrust of 1# unit in the successive load rejection process was shown in the Fig.8. The positive and negative values of $f_z$ represented the direction of the thrust, and the positive values represent the downward direction, meanwhile the negative values represent the upward direction. It could be seen that the direction of 1# unit axial hydraulic thrust was always upward in the successive load rejection process, and the values of axial hydraulic thrust were all negative.

The variation trend of axial hydraulic thrust was basically opposite to that of relative flow discharge. However, the magnitude of axial hydraulic thrust had the same changing trend with the relative flow discharge of the 1# unit. In the initial stage of the successive load rejection, due to hydraulic disturbance, the flow discharge of the 1# unit increased and reached the peak value at 2.9s, meanwhile, the axial hydraulic thrust reached the maximum value at upward direction. When the flow discharge of the 1# unit decreased gradually, the value of the axial hydraulic thrust also approached to zero gradually. At the time of 11.0s, the unit had entered the reverse pump condition, and the unit flow was a negative value. And at this point, the value of axial hydraulic thrust was smallest and closest to 0. Then, as the unit left the reverse pump area, the value of axial force increased gradually.

**Fig.7.** The 1# unit comprehensive characteristic $Q_{11-n_{11}}$ of the pump-turbine in the successive load rejection process

**Fig.8.** The transient axial hydraulic thrust in the successive load rejection process
Furthermore, the runner surface is divided into three parts, respectively named OA, IN, and OB, as shown in the Fig. 9. $f_{OA}$ and $f_{OB}$ are defined as the ratio of the axial hydraulic thrust on OA and OB surfaces and the absolute value of the initial moment of the axial hydraulic thrust, shown as equation (10-12).

\[
\begin{align*}
    f_{OA} &= \frac{F_{OA}}{|F_{OA}(t=0)|} \\
    f_{OB} &= \frac{F_{OB}}{|F_{OB}(t=0)|} \\
    f_{IN} &= \frac{F_{IN}}{|F_{IN}(t=0)|}
\end{align*}
\]

**Fig.9.** The runner surface  
**Fig.10.** The transient axial hydraulic thrust on OA surface and OB surface

The axial hydraulic thrust on OA surface and OB surface were shown in the Fig.10. The direction of the axial hydraulic thrust on OA surface and OB surface were opposite, and the axial hydraulic thrust on OA surface was always downward direction. It could be seen that the axial hydraulic thrust on OA surface and OB surface were related to the pressure at the spiral casing inlet of the 1#unit. The loss in the water distributor and spiral casing was very small, when the spiral casing inlet pressure increased, the pressure in the bladeless zone also increased. In addition, as the pressure increasing at the spiral-casing inlet, the unit speed also rose, so a high-speed rotating water ring was gradually formed in the bladeless zone, and blocked the upstream flow, resulting in a higher pressure in the bladeless zone. As the clearance was connected with the bladeless zone, the pressure in the clearance increased in this process.

The relative pressure is defined as the ratio of the transient pressure to the downstream reservoir water level $H_d$, shown as equation (13)

\[
p' = \frac{p}{\rho g H_d}
\]

The distribution of the relative pressure of OA and OB surface at 0s, 2.9s, 11s were shown in the Fig.11. As can be seen, the pressure in the clearance gradually increased, resulting in the gradual increase of axial hydraulic thrust during the load rejection.

The axial hydraulic thrust on the IN surface was shown in the Fig.12. Since the axial hydraulic thrust of the band and the front of the blade were downward direction, while the crown and the back of the blade were upward direction, the transient axial hydraulic thrust on
the IN surface was the data after the superposition of the two. In addition, the transient flow in the runner was affected by the characteristics of flow discharge, so the variation of axial hydraulic thrust was more complicated.

At the point B and D, 1# unit speed rose and reached a maximum value in the successive load rejection process, and the spiral casing inlet pressure also reached or approached a larger value, so the pressure on the front and back of the blade were both increased. However, most of the energy of the fluid flow was not used to act on the blade, and the differential pressure between front and back of the blade was very small, resulting in the magnitude of the axial hydraulic thrust was also small.

**Fig.11.** The distribution of the relative pressure of OA and OB surface

When the 1# unit was at low speed, such as point A, C and E, the differential pressure between front and back of the blade was larger, so the magnitude of the axial hydraulic thrust was also larger. Compared with point C and E, the magnitude of the axial hydraulic thrust reached its maximum value at point A. Because of the hydraulic disturbance, the flow discharge of 1# unit increased in 4 seconds. At this time, the differential pressure between front and back of the blade was the largest.

**Fig.12.** The transient axial hydraulic thrust on the IN surface

4. Conclusion

In this paper, one-dimensional and three-dimensional numerical simulation were both carried out to simulate the transient characteristic in a pump-turbine during the successive load rejection process. Based on the numerical results, the transient flow characteristic and the axial hydraulic thrust in a pump-turbine during the successive load rejection were emphatically analyzed. Several conclusions were obtained as follows.

During the successive load rejection process, the flow discharge, the speed and the
pressure change greatly, resulting in the axial hydraulic thrust of the unit will also vary dramatically.

The spiral casing inlet pressure and the flow discharge are dynamically unstable during the successive load rejection process. The pressure seriously pulsates at the moment when the unit enters into the reverse pump condition in the successive load rejection process. And there are a same variation trend between the spiral casing inlet pressure and rotational speed of the unit.

It can be obtained that axial hydraulic thrust during the successive load rejection process is mainly caused by the complex flow in the runner. The complex flow is affected both by hydraulic disturbance and the successive load rejection process. The magnitudes of axial hydraulic thrust have the same changing trend with the flow discharge of the unit.

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**References**

[1] Jian Zhang, Weihua Lu, Jianyong Hu. The influence of layout of water conveyance system on the hydraulic transients of pump-turbines load successive rejection in pumped storage station. Journal of Hydroelectric Engineering. 2008, 10, 27(5), 158-162.

[2] Chun Zhang. Study on transition process of load successive rejection of multiple units per penstock in umped storage hydropower power station. Water Resources and Hydropower Engineering. 2011, 12(42), 66-70.

[3] Rong Liu, Jiandong Yang. Analysis on successive load rejection by two pump-turbines sharing one main diversion tunnel. Journal of Hydroelectric Engineering. 2017, 36(4), 71-77

[4] Xiaodong Yu, Jian Zhang, Ling Zhou. Hydraulic transients in the long diversion-type hydropower station with a Complex Differential Surge Tank. The Scientific World Journal. 2014.

[5] Jintao Liu, Shuhong Liu, Yuekun Sun, Yulin Wu, Leqin Wang. Three dimensional flow simulation of load rejection of a prototype pump-turbine. Engineering with Computers. 2013, 29:417–426

[6] Quan-Zhong Liu, Wen-Tao Su, Xiao-Bin Li, Ya-Ning Zhang. Dynamic characteristics of load rejection process in a reversible pump-turbine. Renewable Energy. 2020, 146, 1922-1931.

[7] Xiaolong Fu, Deyou Li, Hongjie Wang, Guanghui Zhang, Zhenggui Li. Xianzhu Wei. Analysis of transient flow in a pump-turbine during the load rejection process. Journal of Mechanical Science and Technology. 2018, 32 (5), 2069-2078.

[8] Xiaolong Fu, Deyou Li, Hongjie Wang, Guanghui Zhang, Zhenggui Li, Xian Zhu Wei. Dynamic instability of a pump-turbine in load rejection transient process. Science China Technological Sciences. 2018, 11 (61), 1765-1775