Evolutions of guide vane moment of a pump-turbine during runaway transient process after pump trip

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Abstract. The collision of guide vanes with runner of a pump-turbine during transient processes is rarely seen in pumped-storage power stations, but the consequence is serious when happens. To investigate the reason of a case of such an accident, we analyse the evolution rules of guide vane moment during the runaway transient process after a pump trip of a prototype pumped-storage system by using 1D-3D method. First the histories of macro parameters such as flow rate, rotational speed and runner moment and trajectory of working condition point are given. Secondly, the difference of guide vane moment under different working conditions is analysed, and the fluctuation components are decomposed by frequency domain analysis. Finally, the main factors affecting the fluctuation of guide vane moment are obtained by comparing pulsation and flow patterns. The results show that, due to different flow structures in different working conditions, there is a significant difference in the characteristics of guide vane moment, which affects the closing and opening trend of guide vanes in different positions. In contrast, the pulsation amplitudes of guide vane moment in no-load mode are the largest, followed by the pump braking mode, and those in the turbine and pump mode are smallest. The maximum guide vane moment is smaller than the design moment of guide vane fastening device during the transient, but it reaches 85.7% of the design value.

1. Introduction

In the operation of a hydropower station, there is a strong rotor-stator interaction (RSI) between guide vanes and runner blades, also the disturbance of the internal flow in the unit during the transient process leading to great fluctuations of guide vane moment, which affect the operation of the power station [1]. At present, the researches on the guide vane moment of the pump-turbine are not perfect. Ji et al. [2] carried out numerical simulations by changing different blade rotation angles for Zhuzhou tubular turbine. It is concluded that the flow rate has great influence on the hydraulic moment of the guide vanes, and the calculated moment of the asynchronous guide vanes is 2.4 times than that of the synchronous guide vanes. Shao [3] tested the guide vane moment of the Francis turbine, expounding the test method and process in detail. Devals [4] used two-dimensional (2D) and three-dimensional (3D) methods to predict and test the Francis Turbine's guide vane moment. It is found that when the action of guide vanes is not synchronous, its adjacent guide vane moment will increase. Botero [5] was used to carry out non-invasive detection test on the vane-less space of a pump-turbine, and the two
reflux and stall vortices in the internal flow field were visualized. Li [6] used 2D dynamic mesh method to simulate the start-up process of a pump-turbine by using a pair of prearranged guide vanes. The 2D flow field characteristics under the increasing speed and opening of the guide vanes were obtained. Huang [7] simulated the dynamic flow field of a Francis turbine during the transient process with the closure of guide vanes, and obtained the coherent structure of vortices around the guide vanes. Li [8] carried out numerical simulation of the whole flow passage of a pump-turbine, and concluded that the flow separation at the inlet of the blade would cause vortices and runner blockages in the runner. Roth [9] simulated and tested the guide vane vibrations of a model Francis turbine by using fluid-structure interaction method. It is found that the vibrations of the guide vanes are strongly coupled with the complex flow around them.

However, most of these studies are aimed at the steady operating conditions of the model units. Based on 1D3D coupling method, the author simulated the runaway transient process in pump mode of a prototype pumped-storage power station. The changes of a series of macro parameters such as flow rate, rotational speed and guide vane moment are analysed.

2. 3D CFD setups

Software for simulation: 3D numerical simulations were carried out by commercial software ANSYS FLUENT 15.0.

Computational domain: The object of this numerical simulation is a prototype pumped-storage power station with the layout of double units per diversion tunnel, including upstream reservoir, diversion tunnel, penstock, pump-turbine unit, downstream conduit and downstream reservoir. Only one of the units was selected to simulate in this paper. For the too long length of diversion tunnel, penstock and downstream conduit, they were simulated by one-dimensional (1D) method, while the pump-turbine unit was simulated by 3D method [10]. The profile of computational domain is shown in figure 1. Some parameters of the prototype unit are shown in table 1. $D_1$ donates the runner outlet diameter (pump mode), $D_2$ donates the runner inlet diameter (pump mode), $GD^2$ donates the rotational inertia, $Z_b$ donates the number of runner blades, $n_{gv}$ donates the number of guide vanes, $n_{sv}$ donates the number of stay vanes, $n_s$ donates the rated rotational speed, $n_r$ donates the specific speed (pump mode).

![Figure 1. Profile of computational domain](image)

| Parameter | Value   | Parameter | Value |
|-----------|---------|-----------|-------|
| $D_1$ (m) | 5.2265  | $Z_b$     | 7     |
| $D_2$ (m) | 4.132   | $n_{gv}$  | 20    |
| $GD^2$ ($10^9$kg·m$^2$) | 11.66 | $n_{sv}$  | 20    |
| $n_s$ (m·m$^3$/s) | 59.9  | $n_r$ (rpm) | 200   |

Turbulence model: SAS-SST turbulence model was adopted due to the scale adaptive characteristic. The model adds the $Q_{SAS}$ term to the generation term of the omega transport equation in
the standard SST turbulence model. This term introduces the von Karman scale into the turbulence scale equation and has large eddy simulation characteristics in the flow separation region.

**Mesh generation:** Structured mesh was adopted in all parts. The total number of grids is about 6.9 million. The boundary layers are set up for the region of stay vanes, guide vanes and runner.

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**Figure 2.** Schematic diagram of mesh and guide vanes number

**Numerical scheme:** After many comparisons, the timestep was selected for 0.00125s, corresponding to the time needed for the runner to rotate 1.5 degrees. The convergence criteria of residuals at each timestep were set to 1.0E-4, and the maximum number of iterations per timestep was set to 40. For both steady and unsteady simulations, the SIMPLEC algorithm was chosen to achieve the coupling solution for the velocity and pressure equations. The second order discretization in time and in space was used.

**Boundary conditions:** The upstream outlet and downstream inlet were set as the pressure boundary conditions, according to the upstream and downstream water levels, and the friction head loss and local head loss was considered for each pipe lines, respectively. The total pressure was defined at the downstream inlet, and the static pressure was defined at the upstream outlet. The directions of rotational speed and flow rate in pump mode are defined as positive directions. The rated speed of the runner is 200r/min and the initial discharge is 139.5 m³/s (corresponding to the guide vane opening 23.6 degrees), which is near the best performance working condition in pump mode.

### 3. Results

#### 3.1. Verification of the numerical method

Before the transient process, the whole flow passage is verified under steady working condition. The results of the simulation and experiment are shown in table 2. In addition, the load rejection transient process with the closure of guide vanes was experimented and simulated under this working condition. The results shown in figure 3. are in good agreement.

**Table 2.** Comparison between the calculated and experimental results

|          | Torque (MW) | Discharge (m³/s) |
|----------|-------------|------------------|
| Experiment | 161.94      | 139.5            |
| CFD      | 166.11      | 142.4            |
| Error    | 2.57%       | 2.08%            |

**Figure 3.** The changes of rotational speed after pump trip with the closure of guide vanes
3.2. Calculation results of runaway transient process after pump trip

3.2.1 Transient properties during the runaway process

From figure 4., it can be seen that runaway transient process passes through 6 modes, including T1 (pump mode), T2 (pump braking mode), T3 (turbine mode), T4 (turbine braking mode), T5 (turbine mode), T6 (no-load mode). Due to the low head of the pump-turbine, the speed and flow rate tends to be steady when it gets into the T6 mode, which is obviously different from the dynamic performance of the high head pump-turbine.

Figure 4. History of macro parameters during transient process

The moments of guide vanes 5#, 10#, 15# and 20# are selected for time domain analysis. From figure 5., we can see that when the unit operates in steady condition, namely at zero time, the guide vane moment direction is consistent with the torque direction of the runner, which is opposite to the
rotational speed direction and along the guide vane opening direction, and the amplitudes of the pulsation are small. With the operation condition passes through the T1-T6 region in turn, the dynamic variation of the guide vane moment presents a regional change. In the period of T1, the pump abruptly breaks down, so the power moment is insufficient. Then the torque of the runner is affected by the resistance moment and the rotational speed begins to decrease. At this time the flow rate also begins to decrease, leading to a decrease in the impact of flow on guide vanes, so the guide vane moment begins to decrease. However, due to the rotation of the runner and inertia of water, the operation condition is still in the pump mode, so the guide vane moment will gradually resume to the initial value. When it reaches the T2 period, namely in the pump braking mode, the water begins to reverse with the rotating stall vortices producing inside the runner, leading to the guide vane moment direction wandering back and forth between the opening and closing direction. As a result, the amplitudes of the pulsation increase gradually, and the high amplitudes keep existence in the whole pump braking mode. The maximum guide vane moment is about 40000 N \cdot m, which reaches 57.1% of design value. After arriving at the T3 stage, the unit enters the turbine mode. There are high pulsation amplitudes while the flow rate is small, but much smaller than that in pump mode. As the flow rate increases, the working condition starts to move to the optimal operation point, and the amplitudes of the pulsation decrease. But when exceeding it, the amplitudes of pulsation begin to increase again. During this period, the directions of guide vane moment are all the closing directions. Once it passes through the runaway point and gets into the T4 stage, the maximum guide vane moment is about 60000 N \cdot m, which reaches 85.7% of design value. It has been found that the dynamic characteristics of the pump-turbine have oscillatory ring characteristics on the four quadrant curve. Because the head of this pump-turbine is low, it can only form a convergent ring, so the final speed and flow rate tend to be steady, which is different from the high head pump-turbine. During this process, the working condition goes through T5 and T6 stages.

3.2.2 Frequency spectrum and flow evolution
A time-frequency analysis is performed on the transient guide vane moment fluctuations by Short Time Fourier Transform (STFT), and the result is shown in figure 6.. At the beginning of the transient process, the characteristics of guide vane moment are mainly influenced by the runner. The dominant frequency in the spectrogram is blade passing frequency, and the rest of high frequency is an integer multiple of the blade passing frequency. Due to the decrease of rotational speed and flow rate, stall vortices generate in the region of guide vanes, stay vanes and runner, enforcing the guide vane moment producing low frequency signals with high amplitudes (t=4-7s), as shown in figure 7. (a, b). In the process of conversion from pump mode to pump braking mode, it is obvious that, due to the change of the flow direction, the high frequency signals produces on the basis of the low frequency signals, and the change of the frequency is in accordance with the change of the rotational speed (t=7-13s). The reason is that reverse flow impacts the runner blades to increase the amplitudes of pulsation during the process of working condition transformation (figure 7. (c)). Meanwhile, stall vortices in the guide vanes, stay vanes and runner regions begin to disappear, but they will still keep existence for a long time until the unit enters the turbine mode with relatively big flow rate (figure 7. (d)). When the unit goes into the turbine mode with better operation region, the amplitudes of guide vane pulsation decrease greatly (t=18-21s, figure 7. (e)). However, once passing through the optimal operating point, the amplitudes of the pulsation increase gradually because of the violent RSI (t=21-28s). After t=28s, the unit goes to T4, T5 and T6 in turn. All the low and high frequency signals with high amplitudes generate again. The flow patterns at t=40.0s in guide vane region are selected (figure 7. (f)). Many vortices appear in flow channels of the runner, which is the reason of low frequency signals, while the unsteady flow structures in vane-less space cause the high frequency signals. At the same time, the amplitudes at blade passing frequency and its harmonics are enhanced. For the asymmetry of the spatial position, the above two kinds of flow blockages lead to obvious differences in the moment of different guide vanes.


**Figure 6.** Spectrogram of 5# guide vane moment

![Spectrogram of 5# guide vane moment](image)

**Figure 7.** Flow patterns in the region of stay vanes, guide vanes and runner

![Flow patterns in the region of stay vanes, guide vanes and runner](image)

### 4. Conclusions

The runaway transient process after pump trip passes through various operating conditions, and dynamic characteristics of guide vane moment also show regional variability. Due to the asymmetry of flow structures, the differences between all guide vane moment are obvious.

1) The pulsation amplitudes of guide vane moment in no-load mode are the largest, followed by the pump braking mode, and amplitudes in the turbine and pump mode are smaller. The maximum guide vane moment is smaller than the design moment of guide vane fastening device during the transient, but it reaches 85.7% of the design value.

2) Under different working conditions, the reason for moment pulsations is different. The stall vortices in the regions of guide vanes, stay vanes and the runner mainly cause the low frequency signals with high amplitudes, while the unsteady flow structures in the vane-less space lead to the high frequency signals with high amplitudes.
Acknowledgments

This work was supported by the National Natural Science Foundation of China (Grant No. 51579187), Natural Science Foundation of Hubei Province (Grant No. 2018CFA010) and Science and Technology Program of State Grid Corporation of China (Grant No. SGBXSJJS1700007)

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