Study on dynamic stress of pump turbine runner during power failure in pump mode

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Abstract. The power failure process of pumped storage unit is one of the most dangerous transient processes. The unit enters the pumping braking mode and then generation mode after power failure. In this process, the rotating speed changes from the negative direction to the positive direction, and the hydraulic torque of the blade also changes rapidly, resulting in the great change of runner stress. In this paper, the amplitude of dynamic stress and the location of stress concentration of runner were calculated and analysed to evaluate the safety of runner in the process of power failure in pump mode. The coupling method of one-dimensional pressure wave calculation and three-dimensional flow simulation is used to predict the change of hydraulic loads and the fluid-structure-interaction method is used to calculate the dynamic characteristics of runner. It is found that the position of stress concentration changes in the process of rapid decrease of rotating speed to zero.

1. Introduction
Pumped-storage power station is important for energy storage and regulation of the power grid. The pump-turbine is the core component of modern pumped storage power stations with larger diameter and flatter structure compared with the conventional water turbine runner. Due to the long running time and pressure fluctuation of runner under off-design conditions, the runner is prone to problems such as strong vibration and fatigue crack during actual operation [1-4]. In addition, the pump-turbine suffers transient processes frequently and may encounter accidents such as load rejection under power generation conditions or power failure under pumping conditions. Under such extreme conditions, the fatigue life of the runner will be greatly reduced.

There are many hydraulic factors that affect the stress of the runner, such as rotor-stator-interaction [5-7], the Karman vortex and the draft tube vortex.

The research methods of dynamic stress of runner mainly include experimental measurement and numerical calculation. Huang X et al. [8] conducted on-site tests on a large axial flow turbine and measured the natural frequency of the runner blades, the static stress under full load conditions, and the dynamic stress during startup and shutdown. The maturity of numerical simulation methods makes the research on the dynamic stress more in-depth. He L et al. [9] used the fluid-solid coupling method to
analyze the modal and dynamic stress of the runner. The resonance phenomenon and resonance curve were also obtained and agreed well with test results.

At present, there are few researches on the dynamic stress characteristics of runners in the transient process, thus the related research work needs to be improved. This paper adopts the calculation method of coupling the one-dimensional piping system with the three-dimensional internal flow of unit to study the flow characteristics, and further uses the fluid-solid coupling method to study the dynamic characteristics of the runner. This paper perfects the understanding of dynamic stress characteristics and influencing factors under extreme conditions.

2. Pump-turbine unit
The prototype pump turbine is used in this research. The rated speed is 500 r/min under pumping conditions and the diameter is 4.3 m on the high-pressure side. The unit has 16 guide vanes and 10 blades. The three-dimensional fluid domain of the pump-turbine unit is shown in Fig. 1.

The power station piping system consists of the upstream and downstream pipelines, upstream and downstream reservoirs, surge tanks and other components. All the components except pump-turbine unit are simplified into one-dimensional models to participate in the calculation.

Fig. 2 shows the model of pump-turbine runner. In order to eliminate the stress concentration phenomenon at the root of the blade, there is fillet between blades and crown/band and the radius is 50 mm.

3. Numerical model and boundary conditions

3.1. Three-dimensional flow calculation
The three-dimensional flow calculation of the unit needs to mesh the calculation domain. This paper divides structural grids into the computational domains with regular structures such as stay vanes, guide vanes, clearances and draft tube, and divides irregularly shaped regions such as volutes and runners into unstructured grids to capture three-dimensional structural features well. The total number of nodes in the 3D computing domain is 4.73 million after grid independence check.

When calculating the flow of pump mode, the boundary conditions are as follows: the discharge boundary condition is set at draft tube inlet and static pressure at volute outlet. The runner and clearances are set as rotating domain.

3.2. One-dimensional flow calculation
It’s assumed that the upstream and downstream water levels are 762.1m and 98m respectively. The initial condition of the one-dimensional calculation is that the pump-turbine is operating normally at the rated flow, and power failure occurs at time 0 s. During this process the guide vanes refuse to move.
3.3. **Fluid-Structure Coupling Calculation**

In this study, solid 185 elements are used to discretize the structure. The total number of nodes is 330,000 after checking the grid independence.

It is assumed that the connecting area between runner and main shaft is fixed support; the load of runner includes gravity, centrifugal force and hydraulic loads on the surface of runner, which are obtained by three-dimensional flow calculation.

Based on the one-dimensional hydraulic calculation, several time points are selected. The transient flow characteristics and the dynamic stress characteristics of the runner were calculated at time points. The following results are based on this step-by-step loading method.

4. **Results and discussion**

4.1. **Energy characteristics**

Fig. 3 and Fig. 4 show the relative discharge and relative speed changes with time during power failure process in pump mode, which are the ratio of discharge Q to the rated value \( Q_n \) and the ratio of rotating rate n to the rated value N. For the sake of simplicity, the word "relative" is omitted below. The process can be divided into 3 stages. In the first stage, the unit is operating in pump mode. Due to the occurrence of power failure, the discharge drops rapidly, and reached zero at 4.6 s. During this process, the reverse rotation rate decreased from -1 to -0.722. In the second stage, the unit enters the braking mode. The water flows back to the runner and the positive discharge increases rapidly. Under the impact of the positive flow, the rotating rate of the runner continued to decrease, and dropped to zero at 13.58 s. The positive discharge was 0.973 at this time. In the third stage, the unit enters the power generation mode. The discharge gradually decreases, and the rotating rate increases. Finally, it reaches runaway at 27.8 s. At this time, the runaway rotating rate reaches 1.31 times the rated value. Since the guide vanes cannot be normally closed during this process, the unit will reach the runaway state many times, causing strong vibration and endangering the safety of the unit. This research focuses on the process before the unit reaches runaway for the first time.

![Figure 3](image1.png) ![Figure 4](image2.png)

**Figure 3.** Variation of discharge with time.  **Figure 4.** Variation of rotating rate with time.

Fig. 5 shows the change of torque on the blades during this process. The black line in the figure is the torque variation curve obtained by one-dimensional calculation. Since the one-dimensional calculation is carried out on the basis of the model test, the calculation result is consistent with the test. At the same time, the three-dimensional flow calculation can also predict the torque, and the results are also plotted in Fig. 5. The comparison of the two shows that the calculation results of the three-dimensional flow in this study are reliable.
4.2. Dynamic stress characteristics of runner

On the basis of the three-dimensional flow calculation in the previous section, a fluid-solid coupling calculation is carried out to analyze the dynamic stress of the runner. Fig. 6 shows the typical stress concentration locations. Among them, the stress recording points A and B locate at the leading edge, and the stress recording points C and D locate at the trailing edge.

Fig. 8 shows the stress changes at stress recording points A, B and C. In order to illustrate the changing trend of stress, the relative torque and rotating rate are also plotted in Fig. 8. It can be seen that the change trends at the three points are basically the same. The maximum stress occurs in the third stage, and the maximum is 123.7 MPa, 93.6 MPa and 123.9 MPa respectively. In the first stage, the stress at the three points is mainly affected by the rotating rate. As the rotating rate decreases, the stress value gradually decreases. In the second stage, as the speed decreases further, the minimum stress appears, and then the hydraulic torque gradually becomes the main factor affecting the stress. When the hydraulic torque reaches its peak, the stress also peaks. After entering the third stage, the hydraulic torque gradually decreases, and the stress also decreases. But with the increase of rotating rate, its influence on stress becomes significant again. With the increase of rotating rate, the effect of centrifugal force becomes more and more obvious, and the stress increases greatly.

Fig. 9 shows the stress change at point D. Point D is mainly affected by the hydraulic torque, especially in the second and third stages, the change trend of the stress at point D and the hydraulic torque is basically the same. Under the action of torque, the stress at point D can reach 302.6 MPa.
Different from the three points A, B and C, the influence of rotating rate on the stress at point D is very limited. The maximum rotation rate does not cause greater stress at runaway.

In order to clarify the reasons, the stress distribution of runner under the action of centrifugal force at runaway (27.8 s) is calculated. Fig. 10 is the equivalent stress distribution near the 4 stress recording points. The runner deforms outward under the action of centrifugal force, and the band deforms upward. Therefore, the three points A, B, and C are all subjected to tensile stress, showing a higher equivalent stress, while the point D is subjected to compressive stress, and the equivalent stress is lower. This indicates that compared to other positions, the effect of high rotating rate on point D is not obvious.

In addition, the stress distribution under the conditions of only the centrifugal force and the hydraulic loads are compared. In the case of only centrifugal force, point D is in a compressed state. The pressure difference between the pressure side and suction side of blade is basically zero at runaway. At this time, the stress in most of the runner is relatively low, and it is the compressive stress under the action of hydraulic pressure. But at D, the runner is subjected to tensile stress. Therefore, in the runaway state, the tensile stress caused by the water pressure and the compressive stress caused by the centrifugal force cancel each other out, and the stress value at point D is smaller.
5. Conclusion
In this paper, the method of coupling the one-dimensional pipeline system and the three-dimensional internal flow calculation of the unit is used to analyze the pressure fluctuation in the pipeline and the three-dimensional flow characteristics of the unit during power failure in pump mode. On this basis, the fluid-structure coupling method is used to calculate the dynamic stress of the runner. The main conclusions obtained are as follows:

1. This paper proposes a one-dimensional and three-dimensional coupling calculation method for calculating transient processes such as power failure and a step-by-step loading fluid-structure coupling analysis method, which effectively predicts the dynamic stress of runner;

2. In the process of power failure, the runner has experienced three stages: pump mode, braking mode and turbine mode. The maximum hydraulic torque on the blades can reach 1.3 times the rated value. The rotating rate can reach 1.31 times of the rated value at runaway. In the transient process, the instantaneously increased load is likely to aggravate the fatigue damage of the blade and reduce the service life of the blade;

3. The roots of the blade's leading edge and trilling edge suffer stress concentration. The stresses at points A, B, C are affected alternately by the rotation rate and the hydraulic torque. When the unit goes through the pump mode and the rotating rate is high, the stress is mainly affected by centrifugal force. After the unit enters the braking mode, the rotating rate gradually drops to zero, and the blade hydraulic torque increases. At this time, the stress is mainly affected by the hydraulic torque. When the rotating rate drops to 0, the blade hydraulic torque reaches its maximum value, and the stress also peaks at this time. After the unit enters the generation mode, the rotating rate gradually increases and the torque decreases, and the rotating rate gradually becomes the main factor affecting the stress. When the unit reaches runaway, the stress peaks;

4. The stress of point D is mainly affected by the blade hydraulic torque. When the hydraulic torque reaches its maximum value, the stress at point D also reaches its maximum value. The influence of rotating rate on point D is not obvious. In the design process of the runner, the strength near point D should be emphasized to ensure the fatigue life of the runner.

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