Analysis of Pressure Fluctuation and Stress Deformation in Reverse Generation of Large Vertical Axial Flow Pump

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Abstract: In order to study the stability of the reverse generation of a pumping station in the eastern route of the South to North Water Transfer Project, a full numerical simulation of the flow passage was carried out to study the pressure fluctuation and the stress distribution law in the reverse generating condition. At the inlet of the guide blade, the monitoring points are set up in three sections of the inlet and outlet of the runner. The calculation results show that the pressure pulsation at the outlet of the wheel is the largest, the pressure pulsation between the guide vane and the wheel is the second, and the pressure pulsation at the inlet of the guide blade is the smallest. At the inlet and outlet of the impeller, the blade is dominated by the frequency. The flow state is uneven, and the pressure fluctuating from the hub to the rim gradually increases. The results of fluid solid coupling show that the stress mainly concentrates on the blade pressure surface and the root of the suction surface, and the maximum equivalent stress appears at the impeller root of the suction surface of the blade. The maximum equivalent stress value is in the safety range of the blade material, and does not affect the life and failure of the turbine running unit.

1. Introduction
In the South to North Water Transfer Project, a pumping station can reserve power generation to prevent floods, solve the water issues for industrial and agriculture, make full use of the discharged water energy and bring considerable benefits. When these pumps run in reverse power, they will generate pressure pulsation, shorten the service life of these pumps, or even cause large vibration in some serious cases, which will damage the structure of the plant.

There are many studies on the pressure pulsation and reserve operation of the axial flow pump. For example, Dai Qifan[1] studied the performance of the reversible axial flow pump devices of Huaiyin pumping, by analyzing the performance of the model factors such as blade airfoil, blade number and hub ratio on the performance of the pump, to determine the optimal hydraulic model. Tang Fangping carried out a three-dimensional unsteady constant value simulation of the axial flow pump to analyze the pressure pulsation at several different monitoring points of the internal flow of the axial flow pump[2]. Zhao Haoru analyzed the inlet and outlet water flow components of a vertical axial flow pump devices in the case of different flow rates[3]. Zheng Yuan analyzed the variation law and
characteristics of the internal pressure pulsation of the axial flow pump under different blade placement angles and lifts\cite{4}. Dong Xinghua carried out a numerical simulation of the full flow channel of the axial flow pump, and analyzed the distribution law of the internal and external characteristics of the axial flow pump\cite{5}.

There are many previous studies on the stability of axial pump pumping, but the stability analysis of pump station reverse operation is rare. In this article, a full-scale simulation of the pumping station flow path is carried out, and the time-domain and frequency-domain diagrams generated by different pressure pulsation monitoring surfaces when the pumping station is reversed are analyzed. Through the two-way fluid solid coupling analysis of the pressure distribution during the reversal of the pump, the stability of the pump in reverse power generation is studied.

2. Mathematical model and boundary conditions

2.1. Computational model and meshing

As shown in Fig.1, this article combines a pumping station on the east route of the South to North Water Transfer Project to simulate the reverse power generation. The basic parameters of the pumping station are as follows: the diameter of the runner is 2700mm, the running speed is 150r/min, the design head is 4.70m, the design water level is 23.00m upstream, and the downstream is 18.30m, the impeller blades have 3 pieces, and the setting angle is 0 degrees. The pump section is shown in Fig.2. The upstream of the design water level is 23.00m, and the pressure inlet is used, the pressure value is 225630Pa. The downstream of the design water level is 18.30m, and the pressure outlet is used, the pressure value is 179523Pa. The solid wall of the axial flow pump device adopts the no-slip condition, when closing the solid boundary region, adopts the wall function, and the flow term discrete format adopts the second-order upwind style (High Resolution) with a convergence precision of 0.00001.

Due to the axial flow pump structure is very complex, this article use the unstructured mesh to mesh the calculation area, and the runner part and the guide vane part are encrypted. A boundary layer grid is selected at the near wall to reflect the flow effects in the near wall area. Firstly, given four different meshing schemes of each component, the model is verified that it is not relevant to the mesh, as shown in Table 2-1, the efficiency calculation results of different schemes are shown in Fig. 3. The number of meshing units of the final determination scheme is: inlet flow passage 270540, runner body 914034, guide vane body 981778, and outlet flow passage 229853 totaling 2396205 grid units, and the grid quality is controlled above 0.6.

| Scheme | Inlet flow passage (10k) | Guided vane (10k) | Runner (10k) | Outlet flow passage (10k) |
|--------|-------------------------|------------------|-------------|------------------------|
| 1      | 18                      | 53               | 56          | 16                     |
| 2      | 22                      | 78               | 82          | 20                     |
| 3      | 25                      | 85               | 88          | 22                     |
| 4      | 27                      | 91               | 98          | 23                     |
2.2. Basic control equations and turbulence models
The three-dimensional Reynolds time-average equation is used in this simulation to describe the turbulent flow of the fluid inside the axial flow pump. Compared with the standard, the turbulence model adds a formula that resolves the flow resilience of low Reynolds numbers, accordingly improving the accuracy of calculating turbulent eddies\(^6\).

When operating the simulation calculation of the unsteady value of the axial pump reversal condition, the runner hub area uses sliding grid technology. The rotating speed is 150 \(r/min\) and a cycle lasts \(T = \frac{1}{n} = 0.4s\). In order to ensure the stability of the analysis, the set time step is \(T/360\), the sampling time is controlled in 10 cycles for 4s, and the pressure pulsation is analyzed in the last two cycles of the calculation.

3. Monitoring point and simulation settings

3.1. Pressure pulsation monitoring point layout
At the inlet of the vane, the monitoring surface 1, 2, 3 is set at the inlet of the runner and at the exit of the runner. The distance between the monitoring surface 1 and the monitoring surface 3 is 3932 mm, and the distance between the monitoring surface 2 and the monitoring surface 3 is 2240 mm, as shown in Fig.4. The monitoring points of the three monitoring surfaces are arranged as shown in Fig.5.
3.2. Bidirectional fluid solid coupling settings

When the pump is in reverse power generation, the blades work in the water, the main load of the blades during operation is the water pressure on the surface of the blade, the gravity and the centrifugal force generated by the rotation. The impeller parts used the impeller parts of the Sanbao model pump 3600ZXM50-3.65. Because the surface of the blade is a distorted spatial surface, the pressure distribution on the surface of the blade during the operation of the unit is complex. Use the structure field and fluid field module in the workbench to transfer data between the CFX and the structural analysis module on the interface. The flow field pressure distribution is solved by CFX, and the surface pressure information of the blade is transmitted to the structural analysis module. The structural analysis module solves the displacement deformation and other information of the blade and then transmits the data to CFX, thereby realizing the reverse power generation of the pump based on fluid solid coupling’s working condition analysis.

3.2.1. Meshing and structure field constraints

The impeller structure is divided by a hybrid mesh, and the blade partial mesh has been encrypted. Before the meshing, the solid field mesh was first verified for independence, and the mesh size of the blade was 30mm, 20mm and 10mm, and the steady-state deformation was calculated, and the results are shown in Table 2. The maximum displacement error of the blade under the three schemes is within 0.5 mm. The mesh results of the blade structure division are shown in Fig.6.

| Grid partitioning scheme | Number of element | Number of nodes | Maximum stress /MPa | Maximum displacement /mm |
|--------------------------|------------------|----------------|----------------------|-------------------------|
| 30mm                     | 5833             | 11246          | 51.93                | 1.373                   |
| 20mm                     | 15663            | 27646          | 63.68                | 1.409                   |
| 10mm                     | 128274           | 202141         | 76.13                | 1.419                   |

Fig.7 shows the filed structure constraint diagram of the runner. The finite element static stress calculation needs to impose sufficient constraints to constrain the motion of the structure and prevent the structure from generating rigid body displacement. Since the cylindrical surface of the blade shaft in contact with the hub can only prevent the radial and axial movement of the blade, and the tangential direction can be rotated, a cylindrical constraint is imposed on the cylindrical surface here to
constrain the radial and axial movement of the surface. The fluid solid coupling interface is the entire blade portion.

![Diagram of fluid solid coupling interface]

**3.2.2. Flow field setting**

The inlet and outlet are all set to a pressure distribution and distributed along the water depth with a water head of 4.7 m. The interface between the runner and the leading vane is set to "transient rotor-static model" to simulate the change of the flow field during the transient relative rotation of the impeller and the front and rear fixed vanes\(^9\). The runner blade is set to Ansys Multifield, which transmits pressure to the solid part coupling surface and receives the solid part coupling surface deformation.

### 4. Result analysis

**4.1. Pressure pulsation time domain analysis**

Three sections were tested by unsteady flow. Each detection surface is mainly divided into two pressure pulsation time domain diagrams in the circumferential direction and the radial direction. In the pressure pulsation time domain diagram, the abscissa takes two periods of 0.8 s, and the ordinate is the area-weighted average static pressure at each point, and the unit is Pa.

![Graph of pressure pulsation time domain analysis](image)

(a)Circumferential monitoring point at inlet guide vane(A2-A4-A5)
In order to study the circumferential uniformity of the inlet water flow, A2, A4, A5 are circumferentially arranged at the inlet section of the guide vane. As shown in Fig. 8(a), the guide vane has a certain distance from the runner area, and the water flow is less affected by the rotation of the runner, so that the pressure pulsation law of the three points is not obvious. As shown in Fig. 9, due to the geometric design of the inlet flow passage and the turbulence of the main shaft above the runner, the water flow does not uniformly enter the vanes, so that the pressure values of the circumferential water flow are different.

A3, A2, A1 are arranged radially along the center of the runner to the edge of the runner. As shown in Fig. 8(b) the water flow is less affected by the rotation of the runner, and the pressure pulsation law of the three points is not obvious. The pressure distribution of the water flow at the outlet of the guide vane is different and the flow state is not uniform. A3, A2 is located in the center of the runner, the water flow changes drastically, and the pressure value is large; A1 is at the edge of the vane, and the value is small\textsuperscript{10}.

Fig. 8 The time domain characteristics of pressure pulsation at the inlet of the guide blade

Fig. 9 Streamline graph
The circumferential monitoring points B2, B4, B5 are arranged on the middle section of the runner and the vane. As seen from (a) of Fig.10, since the vanes do not rotate, the water flow is somewhat smooth. Before the water flows through the runner, it is still affected by the rotation of the runner. The three-point pulsation is obvious, and there are regular peaks and troughs. When the water flows from the vane into the runner, the runner performs work and causes great disturbance to the water flow. Some vortex and back flow occur in the severe interference area, and the water flow in the circumferential direction does not completely enter the runner, so that the pressure value at B5 is significantly smaller than the other two points.

The radial direction of the water flow is set at three points B1, B2, and B3 along the edge to the center of the runner. (b) in Fig.10 shows that the pressure pulsation law at three points is good [11]. Three peaks and valleys appear in one cycle. Since the center of the water flow is not affected by the rotation of the runner, the pressure pulsation amplitude and pressure value are small. The water flow impacts the blades to do work, so that the water flow in the middle and the edge of the runner changes drastically. The amplitude of the pulsation in the middle and the edge of the runner is larger, which is 4.76 times that at the edge.
In order to study whether the circumferential direction of the runner is uniform, the C2, C4, and C5 are circumferential distributed at the impeller inlet section. As shown in (a) of Fig.11, after the water flows out of the runner, it is still greatly affected by the main frequency of the rotating wheel. The pulsation law of the three-point pressure pulsation point is obvious, and there are three peaks and troughs in one cycle. Since the circumferential flow state of the water flow after the exit wheel is relatively uniform, the pulsation amplitude of the three points is close to each other\cite{12}.

In the radial direction of the water flow, three points C1, C2, and C3 are arranged along the edge to the center of the runner. As shown in Fig.11(b), the three points have obvious peaks and troughs due to the rotation of the wheel. After the water flow enters the runner, the middle water flow is greatly affected by the rotation of the runner, and the pulsation amplitude is the largest, which is 2.37 times of the hub center and 5.13 times of the edge. At the edge of the impeller, the water partially flows back due to the presence of the gap, resulting in a poor flow regime with less pressure pulsations. The center of the water flow is not affected by the runner, and the pressure at k is gentle.

4.2. Pressure pulsation frequency domain analysis

The time domain information of the internal pressure fluctuations of each monitoring point when the axial flow pump is in the reverse power generation condition of the pump is obtained by numerical calculation\cite{13}. In order to eliminate the influence of the static pressure of the monitoring point itself on
the pressure pulsation of the point, a dimensionless pressure pulsation coefficient $C_p$ is introduced into the analysis, and its expression is: $C_p = \frac{P_i - P_{\text{ave}}}{P_{\text{ave}}}$. In the formula, $C_p$ is a dimensionless pressure coefficient; $P_i$ is the static pressure value of the monitoring point at a certain moment, and the unit is $Pa$, $P_{\text{ave}}$ is the average value of the static pressure in a rotation period, and the unit is $Pa$.

In order to analyze the pressure pulsation frequency component, the relative pulsation coefficient value of the monitoring point is introduced into Origin to perform Fourier transform to obtain the pressure pulsation frequency domain diagram of each monitoring surface.

Fig. 12 Frequency domain characteristics of pressure pulsation at the inlet of the impeller

(a) Circumferential monitoring point at the inlet of the impeller (B2-B4-B5)

(b) Radial monitoring point at the inlet of the impeller (B3-B2-B1)

Fig. 12 Frequency domain characteristics of pressure pulsation at the inlet of the impeller

(a) and (b) in Fig. 12 are pressure pulsation frequency characteristic diagrams between the runner and the vane. The main frequency of each monitoring point in the circumferential direction and the radial direction is concentrated in the low frequency region. Due to the mutual interference of the rotation of the runner and the guide vane, and the diffusing effect of the flow channel, the high frequency region also has a wide pressure pulsation coefficient. Due to the rotation of the runner, the water flow is uneven at this time, and the three-point pulsation coefficient of the circumferential direction B2, B4, and B5 is different. The pressure pulsation coefficient at two points B1 and B2 is large and close, and B3 is less disturbed at the center of the runner exit, and the pulsation is not obvious. The flow pattern of the impeller gap is poor and the pressure pulsation amplitude at the edge and the impeller is increased. The pressure pulsation amplitude at the rim is 3.55 times of the h1 hub.
Fig. 13(a) and (b) are circumferential and radial detection patterns of the pressure pulsation frequency domain characteristics at the impeller exit. On the circumferential side, the pressure pulsations at three points of C2, C4, and C5 are mainly concentrated in the low frequency range of 0 to 50 Hz, and the remaining frequencies are not obvious. The distribution of pressure pulsation amplitude is consistent, the frequency of the main frequency is the same, and the water flow is evenly distributed in the circumferential direction.

In the radial direction, C2 is located in the middle of the runner exit, and the pressure pulsation coefficient is larger. Compared with the runner inlet, the pressure pulsation coefficient is 2.23 times of the previous one. The pulsation frequency at the center of the runner and the edge of the runner is not obvious, mainly concentrated in the low frequency region. Affected by the runner, the intermediate water flow changes drastically. The pressure pulsation amplitude at C2 is the largest, which is about 4.14 times of the edge of the C1 impeller and 3.87 times of the center of the C3 runner. There is still a large pressure pulsation in the middle portion of the impeller outlet. The water flow is seriously affected by the rotation of the runner, the water flow is unstable, and eddy currents are generated.

4.3. Reverse power generation blade stress analysis
Fig.14 is a graph of blade stress magnitude. It can be seen from the figure that the equivalent stress is about 50 MPa and fluctuates up and down. Since the water flow periodically acts on the solid structure of the blade, different degrees of stress distribution are formed on the blade. In this section, the hydraulic turbine operating conditions based on fluid solid coupling are calculated. The equivalent stress of the blade at different moments is compared with the surface pressure of the blade in the fluid domain. It is observed that the leaf surface pressure distribution and the blade solid equivalent stress distribution are similar at different times, but there is a difference in the value. Therefore, the results were analyzed when 0.8s was selected\textsuperscript{14}. 
The surface pressure distribution of the blade is loaded into the structure for finite element calculation, and the corresponding stress distribution of the blade under hydraulic excitation is obtained, as shown in Fig. 15. For the surface pressure distribution of the blade, the high pressure zone appears on the inlet side of the blade pressure surface, and the low pressure zone appears on the inlet side of the suction side of the blade; the pressure surface pressure gradually decreases from the inlet side to the outlet side, the suction side. The pressure gradually increases from the side of the inlet to the side of the outlet. The change of pressure gradient is larger near the inlet side, which reflects the fact that the impeller enters the water side and does more work on the fluid. It can be seen from the figure that stress concentration occurs at the pressure surface of the blade and the root of the suction surface, and the maximum equivalent stress appears at the root of the impeller of the blade suction surface[15].

4.4. Reverse power generation blade deformation analysis

Fig. 16 is a graph showing the total blade deformation with time in the reverse power generation condition of the unit. It can be seen from the figure that the amplitude is 1.3mm to oscillate up and down.

The total deformation distribution of the blade at each time step can be obtained by two-way fluid solid coupling calculation. Fig. 17 is a diagram showing the total deformation of the solid portion of the blade when the unit water pump is in reverse power generation. It can be seen from the figure that...
during the rotation of the impeller, the total deformation distribution of the blade mainly occurs on the inlet side of the blade, the deformation of the outlet side is small, and the displacement of the inlet side is along the hub to the rim. It is gradually getting bigger\(^{[16]}\).

![Fig.16 Total blade deformation](image)

![Fig.17 Total deformed cloud picture of blade](image)

5. Conclusion
The numerical simulation of the unit axial flow pump under the reverse power generation condition of a pump station in the South to North Water Transfer Project was carried out, and the stability of the unit's reverse power generation was studied.

(1) The stability of the internal operation of the axial flow pump was simulated by setting different monitoring points before and after the vane and the runner. Under the working condition of the turbine, the pressure pulsation is obviously affected by the rotating frequency of the rotating wheel. The impeller outlet pressure pulsation is greater than the pressure pulsation at the impeller inlet, and the pressure pulsation amplitude is about 2.27 times of the inlet. The main frequency of the pressure pulsation is mainly the blade passing frequency, and the pressure pulsation value is gradually increased from the hub to the rim.

(2) Through the bidirectional fluid solid coupling calculation, the blade deformation distribution and stress distribution can be obtained. The total deformation distribution of the blade mainly occurs on the inlet side of the blade, and the deformation at the outlet side is small, and the displacement of the inlet side unit gradually increases along the hub to the rim. The impeller stress is mainly concentrated in the blade root and changes periodically. During blade machining, the blades can be angled or thickened to extend service life.

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