Numerical simulation optimization of axial flow pump device for elbow inlet channel

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Abstract. Based on RNG $k$-$\varepsilon$ turbulence model, three-dimensional simulation of a prototype axial flow pump device for an elbow inlet channel. The optimization strategy of the first part and then the whole part is used to optimize the elbow inlet flow channel and then optimize the outlet flow channel through the integral pump device. The optimization results show that the elbow-type inflow channel recommends the optimization of the inflow channel optimization scheme II, hydraulic loss reduced from 3.4cm to 2.8cm, uniformity of flow rate increased from 96.0% to 96.6% according to the comparison and analysis of the inflow channel optimization; for the comparison and analysis of the outlet flow optimization, the siphon outlet flow channel is recommended to optimize the outlet flow optimization scheme VI, the backflow of the outlet flow channel was eliminated and the hydraulic loss was reduced from 31cm to 29cm.; both 1# impeller and 2# impeller meet the pump station operation requirements when the blade installation angle is -4 degrees. From the perspective of full operating conditions, the 2# impeller has the highest efficiency weight.

1. Introduction
The elbow inlet channel is characterized by a large height and a small width. It is the earliest and most widely used in large-scale pumping stations in China. [1] The elbow inlet channel can provide good water inlet conditions for the impeller of the pumping device, and its own hydraulic loss is also relatively small. It is widely used as a form of inlet channel with excellent hydraulic performance. In recent years, relevant scholars have undergone extensive research on elbow inlet flow channels. [2-15]

This paper optimizes the design of the axial flow pump device including the elbow inlet channel and the siphon outlet channel in combination with a new pump station. The elbow inlet channel and the siphon outlet channel suitable for the operation of the pump station are determined through optimization, and the axial pump impeller suitable for the operation of the pump station is determined through the comparison of the hydraulic model.

2. Project overview
The design flow of the pump station is 48.3m$^3$/s. The pumping station selects 5 sets of 1800ZLQ mechanically-adjusted vertical axial flow pump device, and its single-machine flow is about 9.66m$^3$/s, and the design net head is 4.6 m. At the same time supporting five sets of TL1000-28 high-voltage motor which its single power 1000kW, total installed capacity of 5000kW, rotational speed 214.3rpm. The
pumping station uses an elbow flow channel for inflow, siphon flow channel for outlet, and the vacuum break valve stops flow. The specific parameters are shown in Table 1.

| Diameter of impeller | Rotating speed | Flow | Head |
|----------------------|----------------|------|------|
| D/mm                 | n/r·min⁻¹      | Q/L·s⁻¹ | H/m |
| Prototype            | 1800           | 214.3 | 9660 | 4.6  |

3. Numerical simulation

3.1. Prototype pump device
The numerical calculation is selected for the more widely used 1# and 2# hydraulic models for optimization, and the blade placement angle is -4 degrees. As showed in Figure 1.

3.2. The choice of numerical calculations model
The RNG k-ε model was proposed by Yakhot and Orzag in 1986. [16] The RNG k-ε model is a model deduced from the mathematical method of using the renormalization group for transient N-S equations. It is similar in form to the standard k-ε model, but it has been improved in two aspects[17]: The coefficients in the standard k-ε model is the empirical constants obtained from the experimental results and typical examples, while the coefficients in the RNG k-ε model are calculated by the renormalization group theory; the RNG k-ε model modifies the transport equation for the dissipation rate ε, adding the R term to account for the effects of high turbulent strain rates, thereby improving the prediction accuracy for more complex turbulent flows such as swirling flows and large curvature flows. The transport equation for turbulent kinetic energy k and dissipation rate ε in RNG k-ε model can be expressed as:

\[
\frac{\partial k}{\partial t} + u_j \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \frac{\partial k}{\partial x_j} \right) + \frac{\partial}{\partial x_j} \left( \frac{\partial \varepsilon}{\partial x_j} \right) + \frac{\partial}{\partial x_j} \left( \frac{\partial \varepsilon}{\partial x_j} \right) + G_k - \varepsilon \tag{1}
\]

\[
\frac{\partial \varepsilon}{\partial t} + u_j \frac{\partial \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \frac{\partial \varepsilon}{\partial x_j} \right) + C_{\varepsilon_1} \varepsilon \frac{G_k}{\varepsilon} - C_{\varepsilon_2} \frac{\varepsilon^2}{k} - R \tag{2}
\]

PS: \( v_t = C_{\mu} \frac{k^2}{\varepsilon} \quad R = \eta \frac{1 - \frac{\eta}{1 + \beta \eta}}{\epsilon} G_k \quad \eta = \frac{\varepsilon^k}{\varepsilon} \quad S = \frac{2 S_{ij} S_{ij}}{2} \quad S_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \)

The constants in the transport equation are:

\( C_{\varepsilon_1} = 1.42, \quad C_{\varepsilon_2} = 1.68, \quad C_{\mu} = 0.0845, \quad \sigma_k = 0.7179, \quad \sigma_{\varepsilon} = 4.38, \quad \beta = 0.012. \)
3.3. Meshing

In this paper, SW (solid work) is used modeling of inlet, elbow and outlet channel. It's shown in Figures 2, 4 and 6. The built model can be meshed using the ANSYS ICEM software. Using this method, figures 3, 5 and 7 show the generated hexahedral-structured grid of the inlet channel, elbow and outlet channel. The grid number and grid nodes are shown in Table 2. In addition, the grid quality is greater than 0.35.

For numerical calculation, the number of blades in the hydraulic model of axial-flow pump is 4, and the number of guide vanes is 7. Solid modeling and meshing of impeller and guide vane were carried out using the ANSYS Turbo-Grid software. Figures 8(a)–(d) show the structured grids of impeller and guide vane. The grid number is provided in table 2. Based on a standard prototype pump, $D$ is equal to 1800mm during the modeling, and unilateral blade clearance is 1.2mm.
Figure 8(a). Impeller grid: vertical figure.  Figure 8(b). Guide vane grid: vertical figure.

Figure 8(c). Impeller grid: front figure.  Figure 8(d). Guide vane grid: front figure.

Figure 8. Impeller and guide vane grid.

| Calculation domain         | Node number | Grid number |
|----------------------------|-------------|-------------|
| Impeller domain            | 552796      | 511520      |
| Guide vane domain          | 817992      | 753480      |
| Outlet channel             | 1027975     | 997410      |
| Inlet channel              | 449306      | 432490      |
| Total                      | 2848069     | 2694900     |

Table 2. Grid node and grid number.

The requirements of grid independence are presented in Ref. [18]. In this study, by changing the grid number of overall pump and calculating its efficiency, it was found that when the grid number increases to a certain number, the efficiency no longer increases and remains stable. To satisfy the requirements of grid independence, when the grid number impeller was 511520, the grid number of vanes was 753480, and the grid number of the entire pump was 2694900.

3.4. Calculation boundary conditions and method

The grid model of each segment was imported into CFX-Pre, and a pump device was built. Using the CFD calculation method, the speed of impeller was set to 214.3 rpm, respectively. Outlet pressure of pump device was set to 1 atm. The inlet channel, outlet channel, impeller hub, shell, and guide vane of axial-flow pump were fixed as the stationary wall, and nonslip conditions were applied. The near-wall region was used the boundary condition of standard wall function.

The relative position of rotating impeller and guide vane has no impact on the dynamic and static interface. A “Stage” interface was utilized to control the transmission of parameters between the impeller and guide vane. The discretization of the control equation was based on the finite-volume method. The diffusion term and the pressure gradient were represented by finite-element function. The convective term uses a high-resolution scheme.

In the preprocessor, first the pressure increments and efficient expression of the pump inlet and outlet cross-sections were written. Then, as an auxiliary monitoring point, a real-time observation was taken up in connection with the calculation. The convergence condition was the residual value of $10^{-6}$.
5, and the pressure increment and efficiency of pump’s inlet and outlet section were monitored till they were stable.

3.5. Calculation formula

3.5.1. Uniformity of axial velocity distribution at outlet section. The inlet channel should to provide a uniform flow distribution and pressure distribution for the impeller. The uniformity ($V_u$) of the axial velocity distribution at the entrance reflects the design quality of the inlet channel. The closer the $V_u$ is to 100%, the more uniform the axial-flow velocity distribution at the exit of the inlet channel. The calculation formula is as follows:

$$V_u = \left(1 - \frac{1}{v_a} \sqrt{\sum_{i=1}^{n} (v_{ai} - v_a)^2} \right) \times 100\%$$

PS: $V_u$ is the uniformity of axial-flow velocity distribution at the exit section of the inlet channel (%), $v_a$ is the arithmetic mean value of the axial-flow velocity at the exit section of the inlet channel, $v_{ai}$ is the axial velocity of each calculating unit of the exit section of the inlet channel (m/s), $n$ is the number of units on the exit section.

3.5.2. Prediction of energy performance of pump. The pump head was calculated using the Bernoulli energy equation, and the obtained velocity field, pressure field, and torque acting on the impeller was used to predict the hydraulic performance. The total energy difference between the inlet and outlet of the pump is defined as the unit head. Device head is defined as the total energy difference between the inlet and outlet flow channel, as showed below:

$$H_{net} = \left( \int_{S_2} \frac{p_2 u_2 dS}{\rho g} + H_2 + \int_{S_2} \frac{u_2^2 dS}{2Qg} \right) - \left( \int_{S_1} \frac{p_1 u_1 dS}{\rho g} + H_1 + \int_{S_1} \frac{u_1^2 dS}{2Qg} \right)$$

PS: $H_1$ and $H_2$ are the inlet and outlet section elevations of pump, unit: m. $S_1$ and $S_2$ are the inlet and outlet section areas of pump, unit: m$^2$. $u_1$ and $u_2$ are the inlet and outlet flow rates of pump, unit: m/s. $u_{1i}$ and $u_{2i}$ are the inlet and outlet flow rates of a normal component of the pump, unit: m/s. The efficiency of the pump can be calculated as follows:

$$\eta = \frac{\rho gQH_{net}}{T_p \omega} \times 100\%$$

PS: $T_p$ is the torque, unit: N·m. $\omega$ is the rotational angular velocity of impeller, unit: rad/s.

4. Numerical simulation results analysis

4.1. Numerical analysis of elbow inlet flow

The original design scheme of the inlet flow channel is designated as scheme I, and the parametric modeling of the inlet flow channel profile is completed in ANSYS DM. The inlet flow channel model was introduced into ICEM to complete the meshing of the flow channels. The comparison of the inlet flow channels of the respective schemes is shown in Figures. 9 and 10, and the cross-sectional area of each scheme is linearly and uniformly varied. The principle of the inlet flow channel design is to provide better inlet conditions for the impeller, while ensuring its own hydraulic loss is minimal. The hydraulic
The scheme I is planned and designed after the original design plan has been checked for the necessary flow path installation dimensions and flow path cross-sections. The ratio of the channel width of the impeller diameter is 2.5, the ratio of the total length of the inlet channel to the impeller diameter is 4.6, and the ratio of the impeller center to the floor elevation to the impeller diameter is 1.583.

On the basis of the scheme I, the lowest end of the inflow channel is lowered by 20cm, while ensuring a uniform change in the area of the section, forming the scheme II. The ratio of the height of the impeller center to the floor and the diameter of the impeller is 1.694.

On the basis of scheme II, the lowest end of the inflow channel is lowered by 20cm, and at the same time, the area of the section is uniformly changed to form scheme III. The ratio of the height of the impeller center to the floor and the impeller diameter is 1.805.

4.1.1. Comparison and analysis of optimization of hydraulic loss and uniformity of flow rate of elbow inlet channels. The hydraulic loss of the elbow inlet channel and the uniformity of the impeller inlet flow velocity for the three options are shown in Figures 11 to 12.

The hydraulic loss and flow rate of the inlet flow channel is monotonously increasing, and the hydraulic loss under the calculated operating conditions is from 0.02 to 0.05 m. The hydraulic loss of scheme III is the smallest and the hydraulic loss of scheme I is the greatest. Scheme II and scheme IV have the same inlet flow channel. The hydraulic loss of the inlet flow channel is the same ones. The overall difference of the four schemes is not large, and the effect on the performance of the whole pump device is relatively small.

The uniformity of the flow velocity at the outlet of the elbow inlet flow channel is monotonously increasing, and the flow velocity uniformity is from 95.4% to 96.7% under the calculated conditions. Under the conditions of calculation, the flow velocity uniformity at the outlet of the inlet channel of the
scheme IV and the scheme II is the best, and the uniformity of flow velocity at the outlet of the inlet channel of the scheme I is the worst.

4.1.2. Comparison and analysis of flow field optimization of elbow inlet channel.

Figure 13. Scheme I elbow inlet flow channel $Q = Qd$ streamline diagram and static pressure distribution cloud diagram (Left).

Figure 14. Scheme II elbow inlet flow channel $Q = Qd$ streamline diagram and static pressure distribution cloud diagram (Right).

Figure 15. Scheme III elbow inlet flow channel $Q = Qd$ streamline diagram and static pressure distribution cloud diagram.

The simulation results demonstrate that the flow state of the elbow-type inlet flow channel is better. In the scheme I, there is a low-pressure area inside the elbow close to the exit, but the pressure value is relatively high. It will not generate cavitation at the impeller inlet, and at the same time, the flow uniformity of imports is as high as 96%, the higher flow uniformity indicating that the impeller inlet flow state is better. By comparing the internal streamline diagrams and the static pressure distribution cloud diagrams of scheme I and scheme II, it can be seen that the velocity distribution of the impeller inlet of scheme II is better than that of scheme I, and the area of the pressure region of scheme II is obviously smaller than that of Scheme I. At the same time, the minimum pressure of scheme II is larger than that of the scheme I, and the uniformity of flow velocity is also increased. The hydraulic loss of scheme II is smaller than that of scheme I, but the effect is not obvious. By comparing the internal streamline diagrams and the static pressure distribution cloud diagrams of scheme II and scheme III, it can be seen that the velocity distribution at the impeller inlet of scheme III is worse than that of scheme II, and that of scheme III and scheme II is similar. And the scheme III minimum pressure has no obvious improvement compared with the scheme II. The uniformity of the flow rate is better than that of the
scheme I. There is no observable change compared with the scheme II. The scheme III hydraulic loss is smaller than the scheme II, but the effect is not obvious.

4.2. Numerical analysis of siphon outlet flow channel
When the outlet flow channel model is established, the location of the vacuum break valve is taken into consideration, and a smooth transition of the water flow in the outlet flow channel is taken into account. The designed outlet flow channel is shown in Figures. 16 and 17.

The optimization scheme II of the outlet flow channel relative to the original scheme (optimization scheme I) is to change the flow rate at the top of the hump, from the initial scheme of 2.35 m/s to 2.0 m/s, while keeping other parameters unchanged. The height of the hump top section is unchanged. The width becomes wider and the turn radius at the hump increases slightly.

The optimization scheme III of the outlet flow channel optimizes the flow rate of the top section of the hump from 2.0 m/s to 2.2 m/s, and changes the height of the hump section from 0.76 times the diameter of the impeller to 0.67 times the diameter of the impeller. The top of the hump does not change and the shape of the bottom of the hump remains basically unchanged, and the turning radius of the hump at the top of the hump becomes smaller.

The optimization scheme IV is relative to optimization scheme II, keeping the hump position unchanged, shifting the outlet to the right by 2 m, the parts before the hump do not change, the descent angle after the hump becomes smaller, and the outlet flow becomes longer.

The optimization scheme V is relative to the optimization scheme IV and moves the top of the hump horizontally to the right by 1 m. The outlet of the outlet flow channel also moves 1 m to the right. At this time, the outlet flow channel changes greatly. The dip angle of the outlet flow channel becomes smaller, and the outlet flow channel becomes longer.

The outlet flow channel optimization scheme VI is relative to the optimization scheme III. The outlet flow rate is increased to 0.8 m/s, the top position and outlet width of the outlet section are kept constant, and the outlet bottom is correspondingly lifted. The cross-sectional area of the outlet is reduced.

Figure 16. Front view of different schemes for siphon outlet flow channels (Left).
Figure 17. Vertical view of different schemes for siphon outlet flow channels (Right).

4.2.1. Comparison and analysis of hydraulic loss optimization of siphon outlet flow channel. The hydraulic loss comparison of the siphon outlet flow channel design scenarios is shown in Figure 18.
Figure 18. Comparison of hydraulic losses in different schemes of siphon outlet flow channels. The comparison shows that the hydraulic loss is minimized in the siphon outlet flow optimization scheme IV, but there is backflow in the hump falling stage, and the backflow of the descending section of the hump is eliminated in the optimized scheme VI. At the same time, the hydraulic loss is only increased by 0.02m compared to the optimized scheme IV. Therefore, it is more reasonable to apply the optimized scheme VI.

4.2.2. Comparison and analysis of flow state optimization of siphon outlet flow channel.

Figure 19. Optimization scheme I \( Q_d \) flow outlet flow channel internal streamline diagram (Left).
Figure 20. Optimization scheme II \( Q_d \) flow outlet flow channel internal streamline diagram (Right).

Figure 21. Optimization scheme III \( Q_d \) flow outlet flow channel internal streamline diagram (Left).
Figure 22. Optimization scheme IV \( Q_d \) flow outlet flow channel internal streamline diagram (Right).
Figure 23. Optimization scheme V $Qd$ flow outlet flow channel internal streamline diagram (Left).
Figure 24. Optimization scheme VI $Qd$ flow outlet flow channel internal streamline diagram (Right).

Figure 25. Static pressure cloud diagram of intermediate section of $Qd$ optimization scheme I (Left).
Figure 26. Static pressure cloud diagram of intermediate section of $Qd$ optimization scheme II (Right).

Figure 27. Static pressure of intermediate section of $Qd$ optimization scheme III (Left).
Figure 28. Static pressure of intermediate section of $Qd$ optimization scheme IV (Right).

Figure 29. Static pressure of intermediate section of $Qd$ optimization scheme V (Left).
Figure 30. Static pressure of intermediate section of $Qd$ optimization scheme VI (Right).
Through comparison between the scheme I and the scheme II, it can be found that the streamline of the scheme I of the outflow channel is disordered, the backflow of the hump falls, the static pressure distribution on the wall of the flow channel is not uniform, the distribution of the static pressure on the middle section of the flow channel is nonuniform, and the outlet flow channel optimization scheme II has improved the flow regime, but the static pressure distribution on the wall of the flow channel is still nonuniform, the static pressure distribution in the middle section of the flow channel is also poor, the backflow has not been eliminated, and the hydraulic loss of the outlet flow channel has been reduced.

According to the comparison of the optimization scheme II and III, it can be found that the scheme III streamline of the outlet flow channel are disordered relative to the scheme II flow channel, and the flow wall static pressure distribution is slightly uniform, and the distribution of the static pressure distribution in the middle section of the flow channel is also relatively uniform, but the whole effect is
not obvious. The backflow of the hump fall is not eliminated, and the hydraulic loss of the outlet flow channel is increased relative to the scheme II.

By comparing the optimization scheme II and IV of the outflow flow channel, it can be found that the scheme IV streamline of the outflow flow channel are slightly more uniform than that of the optimization scheme II, and the static pressure distribution on the flow channel wall surface is slightly uniform, and the distribution of the static pressure distribution in the middle section of the flow channel is also relatively uniform, the overall effect is obvious, and the hydraulic loss of the outlet flow channel is reduced relative to scheme II, but the backflow of the descending stage of the hump still has not been eliminated.

By comparison of the optimization scheme IV and V, it can be found that the scheme V streamline of the flow channel are more nonuniform than those of the optimization scheme IV, and the gradient of the static pressure in the flow channel wall is also greater, and the static pressure in the middle section of the flow channel is greater. The distribution gradient is not obvious. As the length of the flow channel increases, the hydraulic loss along the flow channel increases, the flow state degrades, and the hydraulic loss of the outlet flow channel increases relative to scheme IV, and the backflow of the hump fall stage has not been eliminated.

By comparison of the optimization scheme VI and scheme III, it can be found that the scheme VI streamline of the flow channel are more uniform than those of the optimization scheme III, and the gradient of the static pressure in the flow channel wall is also smaller, and the flow channel is the static pressure in the middle section also decreases. The distribution is evenly distributed. Compared with the scheme III, the hydraulic loss of the outlet flow channel is reduced, and the backflow of the hump fall stage is eliminated.

4.3. Comparison of different options for axial flow pump devices

4.3.1. Selection of different flow channel schemes for axial flow pump devices. Extract the pressure from the result file obtained from the simulation calculation, calculate the external characteristic data of the vertical axial flow pump device according to the formula, and draw the corresponding curve.

Table 3. Scheme for simulating inflow and outflow of prototype pumps for vertical axial pumps.

| Scheme | Inlet channel | Outlet channel |
|--------|---------------|----------------|
| 1      | I             | I              |
| 2      | II            | II             |
| 3      | III           | III            |
| 4      | II            | VI             |

(Note: The whole pump device scheme uses No. 1, 2, 3, 4 and the flow channel optimization program uses No. I, II, III, IV, etc.)

Calculate the results of different flow channels of the same impeller flow ~ head and flow ~ efficiency as shown in Figures 37 and 38 below.
Comparing the calculation results of four different inlet and outlet flow channel combinations of the same impeller, the whole pump device scheme 1 has an efficiency of 79.929% near the design flow conditions, and the corresponding flow rate is 9.66 m$^3$/s and the head is 4.499 m.

The efficiency of the pump device near the design flow condition of the whole pump device scheme 2 is 80.268%, the corresponding flow rate is 9.66 m$^3$/s, and the head is 4.513 m.

The efficiency of the pump device near the design flow conditions of the whole pump device scheme 3 is 80.315%, the corresponding flow rate is 9.66 m$^3$/s, and the head is 4.523 m.

The efficiency of the pump device near the design flow conditions of the whole pump device scheme 4 is 80.340%, the corresponding flow rate is 9.66 m$^3$/s, and the head is 4.517 m.

In the case of the whole pump device scheme 2, scheme 3, and scheme 4, near the design condition, the head and efficiency are obviously better than that of the whole pump device scheme 1, and the whole pump device scheme 4 siphon outlet flow channel backflow is eliminated, considering from the viewpoint of safe operation. The whole pump device scheme 4 is preferred. At the same time, the whole pumping device scheme 4 is designed to meet the operational requirements of the blade at a design speed of -4 degrees.

4.3.2. Axial flow pump device different impeller scheme selection. Calculate the results of the flow rate of the different impellers in the same flow channel and the flow rate and efficiency are summarized in Figure 39.

Comparing with the calculation results of different impeller schemes with the same inlet and outlet flow channels, the efficiency of the pump device near the design flow condition of the 1# impeller is 80.340%, the corresponding flow rate is 9.66 m$^3$/s, and the head is 4.517m.
The 2# impeller efficiency is 80.00% near the designed flow conditions. The corresponding flow rate is 9.66 m$^3$/s, and the head is 4.820 m. The efficiency of the two schemes is equivalent when designing conditions.

The maximum efficiency point of the 1# impeller appears at 1.1 times the designed flow condition. At this time, the efficiency of the pump device is 81.088%, the corresponding flow rate is 10.626 m$^3$/s, and the head is 3.499 m.

The maximum efficiency point of the 2# impeller appears at 1.1 times the design flow conditions. The efficiency of the pump device is 83.40%, the corresponding flow rate is 10.626 m$^3$/s, and the head is 4.050 m.

The comparison of the highest efficiency points shows that 2# is better. However, both impellers can meet the operational requirements in the same design speed at the blade angle of the pump device is -4 degrees.

5. Conclusion
(1) The schemes II and III on the elbow-type inflow channel are obviously better than the scheme I. The scheme II needs to reduce the elevation of the floorboard by 20cm on the basis of the original scheme. The scheme III requires that the elevation of the floorboard should be reduced by 40cm on the basis of the original scheme. The siphon outlet flow optimization scheme VI eliminates the recirculation area that exists in the siphon descending section and is preferred.

(2) Axial flow pump device in the blade set angle of -4 degrees, 1# impeller pump device in the design flow $Q = 9.66$ m$^3$/s, head $H = 4.517$ m, device efficiency of 80.340%, this time the head is low. 2# impeller pump device in the design flow $Q = 9.66$ m$^3$/s, head $H = 4.820$ m, device efficiency of 80.00%, at this time, the head is high, the efficiency of the two schemes are quite under the design conditions, 1# impeller is slightly better. The maximum efficiency point of the 1# impeller occurs at the 1.1 times design flow rate. At this time, the efficiency of the pump device is 81.088 %, the corresponding flow rate is 10.626 m$^3$/s, and the head is 3.499 m. The highest efficiency point of the 2# impeller occurs at the 1.1 times design flow rate. The efficiency of the pump device is 83.40%, the corresponding flow rate is 10.626 m$^3$/s, and the head is 4.050 m. However, both impellers can meet the operational requirements in the same design speed as the blade angle of the pump device is -4 degrees.

(3) The CFD software was used to calculate the internal flow field of the pump device, and the external flow curve of the pump device under different working conditions was given. The calculation results show that the shaft power is not more than 700 kW when the blade is placed at the angle of -4 degrees and the highest device head, and the supporting motor power is 1000 kW with sufficient margin.

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