Performance analysis of axial flow pump on gap changing between impeller and guide vane

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Abstract. In order to study the influence on gap changing of the static and dynamic components in axial flow pump, the axial flow pump model (TJ04-ZL-06) that used in the eastern of south-to-north water diversion project was selected. Steady turbulence field with different gaps was simulated by standard $\kappa-\varepsilon$ turbulence model and double-time stepping methods. Information on the pressure distribution and velocity distribution of impeller surface were obtained. Then, calculated results were compared with the test results and analyzed. The results show that the performance of pump is not sensitive with the axial gap width under design conditions and the large flow rate condition. With increasing gap width, it will be improved in low flow rate condition. The attack angle of impeller inlet in small flow rate condition become small and the flow separation phenomenon can be observed in this condition. The axial velocity distribution of impeller outlet is nonlinear and to increase the axial gap is to improve the flow pattern near the hub effectively. The trend of calculating results is identical with test. It will play a guiding role to the axial pump operation and design in south-to-north water diversion project.

1. Introduction
Axial flow pump is a low head pump and is widely applied in irrigation, municipal water supply and drainage, water diversion works, power plant circulating water projects. In recent years, it have also been applied in nuclear power, jet propulsion, etc. Flow components include inlet, impellers, vanes and outlet. The performance of the various hydraulic components affect hydraulic performance of the pump. A lot of research have been done by domestic and foreign scholars. Kochevsky AN etc.[1] analyzed internal axial flow of axial flow pump with back guide vane by using CFD software. Zierke WC[2] studied flow characteristics and testing techniques of the tip clearance in high Reynolds axial flow pump. In order to study the variation and relationships of internal pressure pulsation and noise in axial flow pump in different conditions ,Wang Hongguang etc.[3] simulated the distribution of flow field and sound field in axial flow pump by using numerical simulation, computational fluid dynamics software and acoustic software, and analysis the time domain and frequency domain by setting the unsteady pulse power of vanes as sound source, and comparatively analyzed the difference of acoustic field distribution between the pumps with and without...
pump casing vibration by using the boundary element method. Huang Huanming etc.[4] conducted numerical simulation for axial impeller flow field on the commercial software CFX platform, and compared relative speed and streamline distribution of the 90% impeller outlet height and root position with PIV measurement results obtained relative speed and streamline the n and are compared, and verified the validity of the numerical simulation. Li Zhong, etc.[5] measured the outlet flow field of three conditions of a axial flow pump by using a spherical five-hole probe and obtained the distribution of circumferential, radial velocity component of impeller outlet absolute velocity and velocity circulation.

Currently, the study on the gap between impeller outlet and guide vanes of axial flow pump is not enough, the velocity field, pressure field are not been studied well, especially movement coupling between the impeller and guide vane. Thus, this paper carried a full flow passage steady turbulent flow calculation for axial flow pump and compared the external characteristics of unsteady calculation for different gaps with experimental results, the velocity field within different axial clearance was analyzed on this basis.

2. Unsteady calculations

2.1. Steady turbulent equations

Liquid flow patterns follow three conservation laws. Namely the conservation of mass, momentum and energy. Unsteady flow is solved by second-order implicit time stepping method, and standard $\kappa$-$\varepsilon$ turbulence model was chosen from turbulence model[6, 7], and the full three-dimensional flow field of 0.6Qd, 0.8 Qd, 1.0 Qd and 1.2 Qd was calculated.

Under the relative coordinate system of constant rotating angular velocity $\omega$, and the internal flow of axial flow pump is three-dimensional, steady, incompressible flow, the basic time-averaged movement equation is as follow:

$$\frac{\partial u_i}{\partial x_i} = 0$$  

(1)

$$\frac{\partial (u_i u_j)}{\partial x_j} = -\frac{\partial P}{\partial x_j} + \left(\mu + \mu_t\right) \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) + S_i$$  

(2)

where: $u_i, u_j$ - the average velocity component, $x_i, x_j$ - coordinate component, $P$ – pressure, $\mu, \mu_t$ - fluid viscosity and turbulent viscosity, $S_i$ - generalized source term.

Standard $k$-$\varepsilon$ model is the most commonly used assumptions which based on eddy viscosity turbulence model. Turbulent kinetic energy equation ($k$ equation) and the turbulent dissipation rate equation ($\varepsilon$ equation) are as follows:

$$\frac{\partial (\rho k u_i)}{\partial x_i} = -\frac{\partial}{\partial x_j} \left[\left(\mu + \mu_t\right) \frac{\partial k}{\partial x_j} \right] + P_i - \rho \varepsilon$$  

(3)

$$\frac{\partial (\rho \varepsilon u_i)}{\partial x_i} = -\frac{\partial}{\partial x_j} \left[\left(\mu + \mu_t\right) \frac{\partial \varepsilon}{\partial x_j} \right] + \frac{\varepsilon}{k} \left(\rho P_i - c_1 \rho \varepsilon\right)$$  

(4)

where: turbulent kinetic energy and turbulent dissipation rate are as follows:

$$k = \frac{1}{2} \mu u_i$$, \quad $$\varepsilon = \frac{\mu_t}{\rho} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)$$  

(5)

Turbulent viscosity is:
\[ \mu_i = \frac{\rho e_{ij} k^2}{\varepsilon} \]  

(6)

\( P_k \) is the pressure generated item caused by velocity gradient:

\[ P_k = \mu_i \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_j} \]  

(7)

Model parameters are: \( \sigma_i = 1.0, \sigma_\varepsilon = 1.3, c_1 = 1.44, c_2 = 1.92, c_\mu = 0.09 \).

2.2. Computational domain of axial flow pump

The excellent axial flow pump model (TJ04-ZL-06) of Tianjin same table test of South-to-North Water Diversion Project presided by Ministry of Water Resources was chosen as the research object. This model was applied in the Liushandatao Pumping Station of Eastern Route of South-to-North Water Transfer Project, the nominal specific speed is 1000. The main design parameters are as follows: the flow rate \( Q_d = 350 \text{L/s} \), the head \( H_d = 4.6 \text{m} \), rotational speed \( n = 1450 \text{rpm} \). The main geometric parameters are as follows: impeller diameter \( D_2 = 300 \text{mm} \), hub diameter \( d_h = 120 \text{mm} \), impeller blades \( z_r = 3 \), the guide vane blades \( z_s = 5 \). Impeller and guide vane shapes are shown in figure 1 (a), (b), the water body of whole computational domain is shown in figure 1 (c). Axial clearance is shown in figure 1 (d), 10, 20, 30 and 40mm was chosen as the axial spacing \( s \) between impeller outlet and guide vane.

![Figure 1. 3D model of impeller and guide vane](image)

2.3. Boundary conditions and mesh processing

In order to obtain more stable flow, the length of water taken straight pipe is chosen as 3 times of the outer diameter of impeller. Liquid has been fully developed into a turbulent state when entering in the impeller. The inlet boundary conditions of computational domain was chosen as velocity inlet. The velocity of inlet is axial and symmetric. The size is determined by the flow rate of working conditions. The standard atmospheric pressure was chosen as the reference pressure of inlet. Outlet was chosen as the boundary condition of outlet. The calculation illustrate that backflow phenomenon might occur at run-time of axial flow pump at outlet. In order to improve the stability of computation [8-10], standard wall function was chosen at the near-wall region and the appropriate wall roughness should be set depending on the circumstances. Accuracy of calculation results was set as 10-5.

Structured grid was chosen for computational grid, which with the characteristics of orderly arranging,
clear structure and excellent mesh quality. For the flow field of axial spacing for axial flow pump movement component, the pitch is small when comparing with the whole machine, the results by numerical simulation with the structured grid can reflect the true flow pattern of axial spacing better. The flow components structured grid shown in figure 2. With the improvement of grid quality, performance prediction errors will reduce gradually. grid-independent inspection was done when meshing in this paper. Taking into the computational precision and economy[13-14], 1,105,046 was chosen as the number of computational grids.

3. Calculation results
Under the four typical axial flow pump conditions of high flow rate conditions (1.2 $Q_d$), design conditions (1.0 $Q_d$) and low flow conditions (0.6 $Q_d$, 0.8 $Q_d$), axial spacing was changed (10,20,30, 40mm) to study the inside flow of pump, the working fluid is water (density $\rho$ take 1000kg/m$^3$).

3.1. The effects axial spacing values have on the outer characteristics of model pump
The curve of flow capacity-head and curve of flow capacity-head that under different axial spacing is shown in figure 3. As is shown in figure 3, the axial clearance sensitivity from head and efficiency is low in the large flow rate condition. The head increase with the decrease of axial clearance at the low flow rate condition that around the hump, the where $s$ is 10mm, the head is 6.95m, this is the smallest difference with the experimental values. When $s$ is 40mm, the head is only 6.32mm. The prediction of efficiency is not accurate at the small flow rate working condition, the difference can be as much as 20%, especially when $s$ is 40mm. This is because the secondary flow of the impeller surface at low flow rate working condition and inlet attack angel deviate from optimum operating condition[11]-[12]. The efficiency increase when the axial clearance reduce. Calculation results share the same tendency with the experimental outer characteristic curve. The numerical simulation results predicted the performance axial flow pump accurately.
3.2. Analysis of physical quantities at the mid-section

To analyze the change law of physical quantities in different axial gap, the physical parameters of the intermediate section s/2 was analyzed. The cloud pictures of pressure and velocity vector of the axial gap intermediate section s/2 was shown in figure 4 and figure 5. As is shown in figure 4, the minimum and maximum of pressure was alternating at the middle section. There are 5 maximum pressure area, the same as the number of guide vines. These area is affected by guide vines, the maximum pressure area decreased with the increase of axial gap. Thus, the increase can decrease the inlet pressure of guide vines. As is shown in figure 5, the velocity gradient of intermediate section decreased with the increase of axial spacing. As is in figure 4, the minimum and maximum of pressure was alternating at the middle section. The maximum of velocity shown at the edge. The velocity of hub is small, the circular motion is obvious.

3.3. The effects axial spacing have on impeller

The relative velocity of the near inlet area is high and the pressure is low. The energy loss caused by the vortex that occurred at the gap by the interaction of guide vanes and blades. Considering the movement component spacing is very important to the relative velocity distribution of the working face of vines when designing an axial flow pump. The velocity vector distribution of vines working face from different working conditions is shown in figure 6, figure 7 and figure 8. As is shown in figure 6, velocity gradient variation is obvious in certain part of head and tail of vines under low flow rate conditions. The velocity gradient reduced with the increase of the axial gap. The maximum of the impeller surface speed is 24.92m/s. The maximum velocity gradient area of head and tail is also shown in figure 7 and figure 8. With the increase of flow rate, the flow around the hub strengthened and reached the maximum velocity, the minimum velocity shown around the hub. The minimum velocity area of all working conditions increased with the increase of axial gap.

![Figure 4. Pressure vector contour of z=s/2 surfaces under 1.0Q_d condition](image1)

![Figure 5. Velocity vector contour of z=s/2 surfaces under 1.0Q_d condition](image2)

![Figure 6. Velocity vector contour of impeller surfaces under 0.6Q_d condition](image3)
3.4 The effects variation of axial spacing have on axial velocity of impeller outlet

The axial velocity distributions of impeller outlet of different working condition and different axial spacing are shown in figure 9. $R$ is the impeller radial dimension, the $R$ of hub is 60mm and the $R$ of shroud is 150mm. As is shown in figure 9, the tendency of the axial velocity distributions of impeller outlet of different working condition and different axial spacing are similar. The axial velocity of impeller outlet change monotonously from the hub to the shroud under design condition and large flow rate condition. The axial velocity distributed linearly. The effects variation of axial spacing have on axial velocity distribution is small. The effects variation of axial spacing have on axial velocity distribution of impeller outlet is obvious under low flow rate condition. It distributed nonlinearly.

The axial spacing of impeller outlet and vines inlet are important parameters in axial flow pump design. It affect the pump performance and flow field characteristics. The increase of axial spacing can improve the flow in impeller and gaps.

4. Conclusions

The commercial CFD software ANSYS CFX was used in this paper. The axial flow pump model (TJ-ZL-06) with the namely specific speed of 1000 was chosen. The test data of from the Tianjin same table test of South-to-North Water Diversion Project presided by Ministry of Water Resources was selected. The effects variation of axial spacing of different movement component have on performance of axial flow pump have calculated. The calculations are as follows:

1) The effects axial spacing of movement component have on performance of axial flow pump is
small under design condition and large flow rate condition. The effect of small flow rate condition is obvious. The appropriate increase of the gap of impeller outlet and guide vines inlet can improve the flow of movement component under low flow rate condition effectively.

2) Five maximum and minimum areas of pressure was alternating at the middle section of axial spacing. It is equal to the number of guide vines. The rule can also apply to velocity. The flow in this area was interfered mightily by the guide vines.

3) The impeller outlet axial velocity distribution is nonlinear under low flow rate conditions. The impeller outlet axial velocity distribution is approximate linear under design and large flow rate conditions. The appropriate increase of axial spacing can improve the flow around the hub effectively.

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