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Study on improvement of hump characteristic of an axial flow pump by grooving inlet wall

K F Yang¹, J J Feng, G J Zhu, J L Lu and X Q Luo

¹Institute of Water Resources and Hydro-electric Engineering, Xi’an University of Technology, Xi’an 710048, China

E-mail: 18709210664@163.com

Abstract: When operating at part load conditions, flow separations may occur in the impeller of an axial flow pump due to the increase of the incidence angle at the impeller leading edge, probably resulting in instability in head curve. In this paper, the hump characteristic in head curve of an axial flow pump has been examined by CFD method with the help of ANSYS CFX 16.0. The results show that the head of the axial flow pump in the hump area decreased by 40% at 61% of the design flow rate, caused by stalls in the pump impeller. Furthermore, the improvement of head curve has been conducted by applying axial grooves at the wall of the pump inlet section. It is found that under the condition of small flow rates, the axial groove can effectively reduce the inlet circulation and the attack angle at the leading edge of the impeller. As a result, the back flow on the suction side of the impeller has been reduced. Consequently, the hump phenomenon in the head curve of the axial flow pump has been effectively eliminated.

1. Introduction

The pump is the power equipment that transforms the mechanical energy of the prime mover into the pressure and kinetic energy of the fluid so as to realize the directional transportation of the fluid [1]. Axial flow pump which is a kind of high specific speed pump with large flow and low head characteristics, is widely used in agricultural irrigation, flood control and drainage, municipal water supply, water environment, water cycle and water jet propulsion ship power plant etc [2-3]. With the development of computational fluid dynamics and blade design theory (computational fluid dynamics, CFD) is becoming more and more mature, the efficiency of axial flow pump has reached a high level, but the axial flow pump hump phenomenon still restricts the development and operation of axial flow pump[4]. Hump is a common non steady state phenomenon in hydraulic machinery, and the positive slope characteristic of pump head appears in water pump [5]. The hump is easy to produce in the small flow condition of axial flow pump, the hump means operation condition is very unstable, affecting the unit efficiency and causing a lot of energy consumption, affecting the running safety of the unit more serious. The establishment of effective methods and predict the performance of axial flow pump of axial flow pump internal flow mechanism and the exploration of control means to curb the adverse flow, which is conducive to the efficient operation of expanding the scope of the axial flow pump and improve the stability of its operation.

Many researches have been done on the research and improvement methods of axial flow pump hump. Toyokura[6] studied the velocity field around the axial flow pump impeller and the pressure distribution near the flange, He found that the impeller flow is relatively smooth and no obvious return in the optimum operating point, but with lower flow, blade stall, radial velocity at the impeller of the
sudden increase of the hydraulic performance of axial flow pump decreased sharply. Through the experiment of hydraulic characteristics of axial flow pump, Goltz[7] discovered the critical stall condition, close to the rim to capture the structure at the top of reflux and trailing edge on the suction surface front leaves respectively; in depth at stall condition, they observed the vertical vortex flow in the blade surface. Fay[8] study showed that the stall is due to the flow separation on the surface of leaves, the number of axial flow pump blade 3-4, stall shift in Impeller channel, when the Impeller vane number is more than 6 pieces, the stall cell will transfer in Impeller channel. Chou [9] found that additional fixed guide vane, selection of water cone with proper length, reduce the inlet bend before the throat height, increasing the width and length of straight channel export, optimizing the shape line, can inhibit and weaken the blade inlet section two times flow, improve the axial velocity uniformity, thereby improving the operation efficiency and service life of the water pump. Qian [10] found that the impact angle and tail off flow are two main factors that lead to the guide blade hydraulic loss; adjustable guide vane can be significantly improved by adjusting the angle of axial flow pump flow leaves, reduce water loss, improve the pump lift and efficiency. Study on flow characteristics of the axial flow pump with small flow condition, through numerical simulation and theoretical analysis reveals the unstable condition of axial flow pump internal flow characteristics and mechanism of the use of axial slot technology, we explore the methods to improve the inhibition of bad flow, has important academic value and application value of the research work.

2. Numerical method

2.1. control equations

Axial flow pump internal flow belongs to the three-dimensional steady (relative steady [11]) incompressible turbulent flow, regardless of gravity, in constant angular velocity rotating around Z axis relative to the Cartesian coordinate system, the internal flow law of conservation of mass and momentum conservation law, the governing equations are as follows:

\[
\frac{\partial (\rho u_i)}{\partial x_i} + \frac{\partial (\rho u_j u_i)}{\partial y_j} + \frac{\partial (\rho u_k u_i)}{\partial z_k} = 0 \tag{1}
\]

\[
u \frac{\partial (\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\mu}{\partial x_i} \frac{\partial^2 u_i}{\partial x_i} + s_i \tag{2}
\]

2.2. Geometry model

This paper selected a small axial flow pump as the research object, we studied the effects of imported wall grooving on the performance of axial flow pump, the axial flow pump blade number 6, the guide vane 11, impeller diameter 300mm, impeller speed of 1450 r/min. As shown in figure 1 (a), the three-dimensional hydraulic model is divided into four parts: inlet section, impeller, diffuser, and outlet section.
Figure 1. Physical model

Figure 1 (b) is a groove type inlet cone pipe model. Axial grooves are arranged on the wall of an inlet pipe of an axial flow pump, and the number of axial grooves is 60, and uniformly distributed with an angle, when the angle is $3^\circ$. The groove length $L=200$ mm [12] is used to study the effect of different groove depth on the hump phenomenon of axial flow pump.

2.3. Mesh generation

Using CFD commercial software CFX16.0 for numerical simulation of axial flow pump with high solving speed, high accuracy, good deformation of hexahedral mesh [13], the computational domain grid independent verification, when the grid number is 7 million 60 thousand, the external characteristics of axial flow pump has stabilized, so choose to carry out numerical grid number 7, 060, 000 for grid computing. In order not to affect the data transfer of the interface, the distance between the axial groove and the Impeller inlet is 1mm.

Figure 2. Computational domain and mesh of flow passage components

2.4. Boundary condition setting

The inlet boundary condition is set as the total pressure inlet, the value is one atmospheric pressure (101kPa), and the exit boundary condition is set to the mass flow. The boundary with no slip boundary conditions, using High Resolution Scheme discrete scheme for high order discrete scheme has good numerical diffusion binding, calculation can be higher on the accuracy of the calculation problem of convergence. The turbulence model selects the SST $k-\omega$ turbulence model with high prediction accuracy both in the boundary and in the whole turbulent region [14]

3. Results and discussion

Figure 3. Comparison of simulation results and experimental results
In order to verify the reliability of CFD calculation, the performance of axial flow pump under different operating conditions is simulated and compared with the model test results. Figure 3 is a comparison of the head curve between the numerical simulation results and the model test results. We can see from figure 3, obtained by the numerical simulation model and experimental results of hydraulic lift shaft flow pump in good agreement, without considering the influence factors of wall friction, hydraulic leak and gap flow calculation, there is always a certain value error calculation value and test. From the result, the calculated value has the same change rule as the experimental value, and the error control is in the effective range, and the reliability of the CFD numerical simulation is verified.

3.1. Characteristics of axial flow pump
In order to study the effect of slotting depth on hump phenomenon, numerical simulation of 4 different groove depth is carried out. Among them, the groove depth is expressed by K, and K is the ratio of groove depth to impeller diameter.

3.2. Internal flow analysis
The above analysis shows that the slotting technique of the inlet wall of axial flow pump is very effective in restraining the formation of hump zone under small flow condition. In order to study the mechanism of groove to improve the hump phenomenon of axial flow pump, figure 4 shows the axial velocity distribution and circumferential velocity distribution of impeller inlet under two flow conditions. Figure 5 shows that in the same condition, the axial velocity distribution of impeller inlet of the two programs are basically the same, especially near the axial flow pump under the optimal working condition, the axial velocity of the impeller inlet is more evenly distributed, circumferential velocity tends to zero. Under the same working condition, the axial velocity of the slotted axial flow pump is obviously larger than that of the axial pump without slotting, but the circumferential velocity is just the opposite. The analysis found that the speed distribution of different flow conditions, the
axial flow pump at small flow rate, axial velocity of impeller inlet radius distribution is not uniform, with the increase of the radius, the axial velocity increases first and then decreases, flow appears near the wall, circumferential velocity first decreased, then increased, reached the maximum value in the near the shroud. It can be seen that the axial velocity of axial pump decreases and the circumferential velocity increases under the condition of small flow rate, which results in the increase of thrust angle of axial pump inlet. At this point, the separation of flow is severe at the suction surface of the blade, and the incoming flow is reduced, and the recirculation is further transferred to the blade inlet and the vortex formed with the incoming shock. Impeller blade channel vortex can not immediately be broken up or out of the impeller channel, resulting in further accumulation and development, eventually blocked the channel.

![Graph](image)

(a) Axial velocity  
(b) Circumferential velocity

**Figure 5. Impeller inlet velocity distribution**

![Streamlines](image)

(a) Span=0.8, without groove  
(b) Span=0.8, with grooves  
(c) Span=0.85, without groove  
(d) Span=0.85, with grooves  
(e) Span=0.9, without groove  
(f) Span=0.9, with grooves  
(g) Span=0.95, without groove  
(h) Span=0.95, with grooves

**Figure 6. Streamlines under different working conditions**

Figure 6 shows the streamlined diagram of the different sections at a stall condition (Q/Q_{des}=0.55). As shown in figure 6(a), at Span=0.8, the double vortex appears in the Impeller path, the vortex near
the impeller inlet rotates clockwise, and the vortex near the outlet rotates counterclockwise. At Span=0.85, the vortex at the outlet of the impeller disappears, instead of the outlet of the impeller and the boundary of the return flow. At this point, the impeller inlet vortex begins to shift from the center of the channel to the inlet pressure surface of the blade. When the cross section is further extended to Span=0.9, the inlet vortex of the impeller decreases. The backflow is further increased and flows out from outlet of the Impeller. The backflow changes the flow direction and flows downward to the Impeller passage due to the impact of the incoming flow. The flow condition of the lower Impeller passage is similar to that of the Impeller in the upper Impeller. The flow in the lower impeller passage will flow further to the Impeller path. Therefore, similar flow characteristics are found in all Impeller channels. At Span=0.95, the inlet vortex of the impeller almost disappears, and the backflow reaches the maximum value at this time. After slotting, observation map (b), (d), (f), (H) show that the impeller inlet wall slot increases the impeller inlet velocity increases the flow pressure and can effectively eliminate the passage vortex in the impeller.

The calculation results of deep stall condition (Q/Qdes=0.45) are selected. Relative pressure on the impeller blade surface has been compared at three span locations in figure 7. L represents the percentage of length along the height of the blade. It can be observed that in the deep stall condition, except in the change of pressure at the back of the blade near the hub at the outlet is large, the other position of blade pressure distribution is uniform. When grooving, the pressure of the front and back of the Impeller blade, especially near the blade outlet, is very obvious. The pressure difference in the middle part of the blade is further enlarged, and the load in the middle part of the blade is increased, and the head of the axial flow pump is improved. There is no large pressure drop or even negative pressure on the blade head, which prevents the formation of the flow off the back of the blade and inhibits the occurrence of cavitation.

![Figure 7](image-url)

Figure 7. Pressure on impeller blade surface, Q/Qdes=0.45

4. Conclusion
In this paper, based on CFX numerical simulation technology, the steady flow numerical simulation of axial flow pump internal flow is carried out by using SST k-ω turbulence model. The improvement of axial flow pump hump phenomenon is studied by adopting axial grooves at the pump inlet. The following conclusions can be drawn:

The axial flow pump of the original model has a hump phenomenon starting at Q/Qdes=0.61, where the head of the pump has been decreased abruptly by 40%, caused by stalls in the pump impeller.

By applying axial grooves at the inlet section, the hump in the head curve has been removed, and the best groove depth is K=0.02, based on the impeller diameter.

Under the condition of small flow rates, the axial groove can effectively reduce the inlet circulation and the attack angle at the leading edge of the impeller. As a result, the back flow on the suction side of the impeller has been reduced.

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