Numerical investigation of the grooves effect on the inlet flow characteristics and hydraulic performance of an axial-flow pump

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Abstract. In this paper, the axial position of the grooves effect on the inlet flow characteristics and hydraulic performance of an axial-flow pump is studied by numerical method. The computational results show that setting the grooves on the pump inlet pipe could significantly improve the inflow characteristics of impeller and the saddle zone would be effectively suppressed. Under the deep stall condition, the grooves could improve the axial velocity of inflow and effectively reducing its streamline bending degree, and it also can block and destroy the vortex structures and then dissipate them rapidly. Reducing the axial distance between grooves and impeller can improve the hydraulic performance and the inflow characteristics of axial-flow pump.

1. Introduction

The high specific speed axial-flow pump features in the large discharge and the low head, which is widely implemented for flood control and drainage, farmland irrigation, large-scale water diversion projects and water jet propulsion of ships. However, the axial-flow pump is always forced to operate at off-design condition. The hydraulic performance of the pump significantly dropped and often accompanied by violent vibration and noise, seriously endanger the safety of the pump unit under part-load condition \([1]\). Some researchers have studied the flow mechanism in the saddle region of axial-flow pump. Toyokura \([2]\) measured the flow field in the pump earlier and found that the radial velocity at the impeller inlet would increase suddenly when the water head of axial-flow pump had a sudden drop. Goltz \([3]\) used high-speed camera technology and oil flow technique to observe the flow pattern inside the impeller, and found that there were separated flows at the tip of the leading edge and the hub of the trailing edge of the blade suction surface under the deep stall condition. Miorini \([4,5]\) used PIV technology to study the influence of operating conditions on tip clearance flow of axial-flow pump, and found that tip leakage vortex under part-load conditions was significantly enhanced compared with optimal conditions and its influence area would be extended to the entire impeller passage. Zhang \([6]\) found that there was a significant swirling flow at the impeller inlet near the blade tip of the axial-flow pump when the flow was small, and discovered that this flow was an important factor causing the saddle area of the axial-flow pump. In order to improve the hydraulic instability characteristics of the axial-flow pump under the condition of small flow and avoid the formation of saddle region, Zhang \([7]\) also preliminarily studied the influence of groove control technology on the hydraulic
characteristics of axial-flow pump, and found that groove control technology could effectively suppress the formation of saddle region.

In order to deeply analyze the mechanism of the groove control technology to improve the internal flow characteristics and hydraulic performances of the axial-flow pump, this paper investigated the influence of three different axial positions of the grooves on hydraulic characteristics, impeller inflow characteristics, vortex flow characteristics and vortex structure under deep stall condition. This paper could lay a foundation for the establishment of the theoretical design method of groove control technology.

2. Computational methodology

2.1. Computational domain and grid generation

The computational domain of axial-flow pump included inlet extension section, inlet cone pipe, impeller chamber, guide vane zone, outlet pipe and outlet extension section, as shown in figure 1. Main structural parameters and design operating parameters of axial-flow pump were listed in table 1.

Grooves were arranged on the wall of the inlet pipe before impeller inlet, as shown in figure 2. The parameter $S_{ji}$ was defined as the distance between the groove tail (near the impeller side) and the center of the impeller in the axial direction, and the parameters $L_j$, $D_j$ and $\alpha_j$ represented the length, thickness and angle of the grooves respectively. The number of grooves was represented by $N_j$, and the specific values of each parameter were: $L_j=200\text{mm}$, $D_j=5\text{mm}$, $\alpha_j=10^\circ$, and $N_j=18$. In order to study the influence of the position change of the grooves on the hydraulic and internal flow characteristics of the axial-flow pump, three kinds of grooves in the axial position were designed. Three different axial distance of $S_{ji}$ were set as $S_{j1}=0.2D$ (60mm), $S_{j2}=0.5D$ (150mm), $S_{j3}=1.0D$ (300mm).

To effectively control the mesh distribution in axial-flow pump, the whole structured hexahedral grids were used in the present study. The O-type grid technology and local refinement method were adopted to control the near wall mesh, as shown in figure 3, which ensured the requirements of corresponding turbulence model at the near wall treatment. In order to establish grid-convergent results, a criterion based on the Richardson Extrapolation Method was used to improve grid resolution and systematically evaluate truncation error and accuracy. Ultimately, the total cell number of the non-groove model and three sets of groove model are all about 3.8 million.

Table1. Axial-flow pump geometric parameters and design performance parameters.

| Parameters                  | Symbol | Value      |
|-----------------------------|--------|------------|
| Impeller blades number      | $Z_1$  | 4          |
| Guide blades number         | $Z_2$  | 6          |
| Impeller diameter           | $D$    | 300 mm     |
| Tip clearance depth         | $\delta$ | 0.25 mm   |
| Impeller rotating speed     | $n$    | 1450 r/min |
| Volume flow                 | $Q_{\text{BEP}}$ | 374.05 L/s |
| Water head                  | $H_{\text{BEP}}$ | 6.219 m    |

2.2. Numerical method and boundary condition

In this paper, the three-dimensional viscous flow was modeled by the URANS equations which were discretized on the element-based finite volume method. Considering the local rotation characteristics and curvature effects of the flow in axial-flow pump, the rotation-curvature corrected Shear-Stress-Transport (SST-CC) turbulence model was used to calculate the Reynolds stresses. In order to decrease the numerical dissipation in simulation, the second-order high resolution scheme and the second-order backward Euler scheme was used separately for the advection term and transient term.

For boundary conditions, the mass flow rate was given at the inlet and the constant static pressure was imposed at the outlet of computational domain. The rotational and stationary domains were
connected by the General Grid Interface and the Transient Rotor-Stator interface model was utilized in the unsteady simulation. The non-slip wall, considering the effect of wall roughness, was implemented for the wall boundary condition. The residual convergence criterion for the RMS variables was set to $1 \times 10^{-5}$ and the mass-momentum imbalance target was less than 1%. In order to make a balance between the temporal accuracy and computing resource consumption, the time step size was set to $1.149 \times 10^{-5}$ s (about rotating 1 degree each step) and the total time was 15 circles of the impeller rotating. Moreover, the last circle results were averaged for post-processing.

3. Results and Discussion
Figure 4 described the influence of different axial positions of grooves on the water head of axial-flow pump. Under the conditions near design flow condition ($Q_{BEP}$), the groove control technology had little effect on the water head, but for the saddle area flow operating mode area ($Q < 0.53 Q_{BEP}$), the water head of three groove schemes all had different degrees of improvement, especially for the working condition of deep stall ($Q = 0.13 Q_{BEP}$). With the decreasing of the $S_h$, the improvement effect of the water head of axial-flow pump was more obvious.

$H_n$ is the regularized helicity [8], which can describe the vortex structure well. The specific expression is:

$$H_n = \frac{u \cdot \Omega}{|u| \cdot |\Omega|}$$  \hspace{1cm} (1)

Where, $u$ is the velocity vector; $\Omega$ is vorticity vector for the curl of the velocity vector. In the mathematical sense, the regularized helicity $H_n$ is the cosine value of the angle between the velocity vector and the vorticity vector (the direction is determined by the right-handed helical rule), and the range of its value is -1~1, $|u| \neq 0$ and $|\Omega| \neq 0$. 

![Computational domain](image1.png)

- **Figure 1.** Computational domain.

![The structure size of groove](image2.png)

- **Figure 2.** The structure size of groove.

(a) Impeller overall mesh  
(b) O-type grid

![Impeller grid distribution](image3.png)

- **Figure 3.** Impeller grid distribution.
Figure 5 showed the $H_n$ distribution of the normalized helicity of impeller inlet section at different axial position of grooves under the condition of deep stall ($0.13 Q_{BEP}$). Through comparative analysis, it could be seen that the impeller inlet sections of the three groove schemes all had $H_n=-1.0$ regions, indicating that the vortex vector and velocity vector in this region tend to be collinear. Compared with the original model, the vortex became stable and the stream-line bending degree decreased. For the area near the wheel hub where the $H_n$ value is close to 0 in the original model, the $H_n$ value of the three groove schemes was larger, indicating that the streamline bending degree in this area could be effectively reduced and the inflow condition could be improved to some extent after adding grooves.

A cross section M0 was token through the rotation axis in the inlet passage, as shown in figure 6. In order to appropriately describe the velocity distribution on the section M0, a dimensionless variable $VV$ was defined, and the expression was:

$$VV = \frac{\overline{V}_m}{|\overline{V}|}$$

$\overline{V}_m$ and $\overline{V}$ were the time-averaged axial velocity and time-averaged resultant velocity of a point on the section respectively, as shown in figure 7, $\overline{V}_a$ was the other time-averaged velocity. For the convenience of narration, the word "time-averaged" was omitted in the following references to time-averaged speed. $VV$ could reflect the proportion of $\overline{V}_m$. When $|VV|>0.707$ (while $\overline{V}_m = \overline{V}_a$ , the corresponding $VV$ value is 0.707), $\overline{V}_m > \overline{V}_a$, this indicated that axial velocity dominated the whole velocity. When $|VV|=1$, the axial velocity was the combined velocity. $VV$ value also reflected the extent of the recirculation zone, when $VV<0$ in a flow area, $\overline{V}_m <0$, indicated this area is a recirculation zone.

**Figure 4.** Comparison of the water head calculation results at different axial positions of grooves

**Figure 5.** The $H_n$ distribution of standardized helicity of impeller inlet section at different grooves under $0.13 Q_{BEP}$ conditions
Figure 8 showed the velocity distribution of section M0 at different axial positions of grooves under the $0.13Q_{\text{BEP}}$ condition. For the near-wall region, the $VV$ value of the upstream of the grooves in the three grooves schemes all changed from $-0.20$ to about $-0.10$, indicating that the proportion of the axial component of backflow decreased. As a result, the grooves had a certain blocking effect on the continuous spiral backflow developing upstream near the outer wall of the original model. For central area, three groove schemes solutions in the region of flow had significant improvement effects. The maximum $VV$ of forward flow increased from $0.30$ to above $0.60$, illustrated that the grooves can effectively reduce the rotation of the screw into the flow of central area. In particular, $VV$ value of impeller near the hub before import on $S_{\text{J1}}=0.2D$ scheme increased from $0.15$–$0.30$ to about $0.70$–$0.80$. This indicated that the axial flow component of the inlet near the wheel hub accounted for a larger proportion, the degree of streamline bending was greatly reduced, and the inlet condition was obviously improved under the grooves scheme. Grooves schemes could improve the internal region flow pattern most obviously. Except that there was still a certain range of backflow area in the area ahead of the water guide cone, other recirculation zones have been eliminated. At the same time, the $VV$ value of these areas has changed from about $-0.50$ (deep blue area) to more than $0.90$ (deep red area), showing a positive inflow dominated by axial velocity. In addition, it can be found from the figure that the range of these deep red areas increased as the grooves set closed to the impeller inlet. When $S_{\text{J1}}=0.2D$, it had taken up most of the areas, and the $VV$ value of other positive areas also increases from smaller than $0.30$ to larger than $0.60$. 

![Figure 6. Section diagram of M0](image)

![Figure 7. Diagram of time average velocity at a certain point in section M0](image)
Figure 8. The velocity distribution of section M0 at different positions of grooves under 0.13Q_{BEP} condition

Three sections M2~M4 perpendicular to the axis were taken in the inlet passage, and the axial distance h from the center of the impeller was 0.4933D, 0.9667D and 1.6667D respectively. The inlet section of the impeller (h= 0.1967D) was section M1. The position of section M1~M4 were shown as figure 9. Figure 10 reflected the structural characteristics of the recirculation vortex in the inlet passage of the original scheme and the grooves schemes at different positions under the 0.13Q_{BEP} operating condition. Through comparative analysis with the original model, it could be found that for the scheme $S_{J1}=1.0D$, under the impact of the grooves, the velocity field in the inlet cone was adjusted, the relatively complete flaky vortex structure between M1 and M2 was split into multiple flaky structures, and the fracture degree of the vortex structure between M2 and M3 was slightly increased. The strip vortex structure of the groove segment was broken by the grooves, and a small amount was attached to the groove wall surface. Only a few vortexes existed in the upstream region of the grooves. For the $S_{J2}=0.5D$ scheme, the vortex structure characteristics between M1 and M2 were similar to those of the $S_{J3}=1.0D$ scheme, showing multiple flaky structures. The continuous recirculation vortexes in the region near the grooves were also broken into small vortexes due to the barrier of the grooves and attached to the grooves wall surface. The strip vortex structure in the upstream region of the groove was broken earlier and disappeared, and the axial position extending upstream did not exceed M4. For $S_{J3}=0.2D$ scheme, the strongly continuous recirculation vortex near the outer wall of original model had been cut off by the grooves in front of the impeller imports. Distributed after breakage, in addition to the part attached to the grooves on the wall, the formation of other multiple strips vortex structure, which continued moving to the upstream disappeared after a certain
distance. This was because the inlet cone tube flow vortex had a large turbulent kinetic energy, and it still had the tendency to develop upstream after being affected by the grooves. With rapid dissipation of turbulent kinetic energy after the grooves, the vortex structure fractured and disappeared accordingly. Based on the above analysis, it could be seen that the continuous recirculation vortex structure around the outer wall could be broken after being blocked by the grooves, and then dissipate rapidly. Moreover, the closer the grooves were to the impeller inlet, the earlier the vortex structure was blocked by the grooves, which greatly shorten the distance for the circulation vortex to move upstream.

Figure 9. The position of section M1~M4

Figure 10. The structure characteristics of the reflux vortex in the inlet passage of the original model and the groove schemes under 0.13Q_{BEP} conditions (λ_{2}=-8.0 \times 10^{3})

4. Conclusions

Based on the numerical simulation method, the influence of grooves axial position on the hydraulic performance of axial-flow pump and the flow characteristics of impeller inlet was studied.

(1) Setting proper grooves on the pump inlet pipe can effectively improve the hydraulic performance of axial-flow pump under the part-load condition, and reducing the distance between the grooves and impeller can make the improvement more significant.

(2) In the case of deep stall, adding grooves can improve the axial velocity near the hub and make the axial component of the inflow account for a larger proportion, effectively reducing the curvature of the streamline in this area and improving the inlet condition of the pump. Moreover, reducing the distance between the grooves and impeller could make the effect better.

(3) Under the operation condition of deep stall, the grooves can make continuous flow vortex structures near the outer wall crushing after blocking occurs, then dissipating quickly. The closer the grooves are to the inlet of the impeller, the sooner the vortex structure is blocked by the grooves,
which greatly reduces the distance of the recirculation vortex moving upstream. In addition, the arrangement of the grooves can effectively reduce the degree of rotation of the positive spiral inflow in the central region of the flow passage, and can also eliminate most of the recirculation zones in the internal region of the original model, so that the flow pattern presents a positive inflow dominated by axial velocity.

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