Influence of Blade Tip Clearance on Performance of Axial-Flow Waterjet Pump

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Abstract. In this paper, the fine/turbo software of NUMECA was used. Based on the three-dimensional steady Reynolds averaged N-S equations, the S-A turbulence model was used to calculate the relationship between tip clearance and diameter of the impeller, which were δ=0.1%, 0.3%, 0.5% and 0.7% respectively. The clearance flows of four water jet axial flow pumps with different tip clearances were numerically studied at different flow rates. The results indicate that the size of the clearance affects the cavitation performance of the blades, the efficiency and head of the pump, and has a great impact on the performance of the pump. It provides a useful exploration for the design improvement of axial-flow waterjet pump.

Keywords. Axial-flow waterjet pump, blade tip clearance, hydraulic performance, numerical study

1. Introduction


Based on the assumption of rapid design of waterjet pump, this paper adopts the traditional design method to design a waterjet axial flow pump with good performance, and studies the influence of tip clearance variation on hydraulic performance and flow field of the waterjet pump under various working conditions, so as to provide a useful exploration for the design improvement of axial-flow waterjet pump.

2.1. Physical Model

The impeller diameter of the three-dimensional model of the waterjet pump used in the numerical simulation study is 300 mm, the impeller number is \( z = 5 \), the hub ratio is 0.45, the number of stators is \( z = 9 \), the blade airfoil is NACA airfoil, the rotating speed of the impeller is 1450 r/min, the flow under the best working condition is 484 kg/s, the lift is 9.5 m, the axial clearance between the rotors and stators is 30 mm, the impeller are set with tip clearance, and the stationary blades have no clearance, as shown in Figure 1. The computational domain of axial-flow water jet pump is shown in Figure 2. The boundary conditions to be dealt with in the numerical simulation process include inlet, outlet, periodic and wall.

![Figure 1. The model of pump.](image1)

![Figure 2. Computational domain.](image2)

2.2. Grid and Numerical Method

To control the quality of grid, the impeller areas and the stator areas are adopted with the O-H grid. O-shaped grids are used for encryption near the impeller, and butterfly grids are used at the tip clearance of the impeller, as shown in Figure 3. The total number of grids in the pump is about 1.46 million. The calculated \( y^+ \) of all walls is less than 20.

![Figure 3. Calculation grid.](image3)

The fine/turbo software package of NUMECA is used to simulate the designed axial-flow pump stage of waterjet pump. The control equation is the three-dimensional steady Reynolds averaged N-S equations, the turbulence model is the S-A turbulence model. The equation difference adopts the spatial central difference format and the time term is solved iteratively by the multi-step Runge Kutta method, and the implicit
residual smoothing method and multi grid technology are used to speed up the calculation.

2.3. Grid Independence Verification

To ensure the independence check of the grid, four grid models with different grid nodes are chosen for performance analysis, as shown in Figure 4 and Figure 5. It can be found that the efficiency and head of the waterjet pump are not sensitive to the change of the number of grids. To ensure the accuracy of calculation results, the number of grids selected is 1.66 million.

<table>
<thead>
<tr>
<th>Grid Number</th>
<th>Efficiency (%)</th>
<th>Head (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>120</td>
<td>84</td>
<td>7.5</td>
</tr>
<tr>
<td>140</td>
<td>85</td>
<td>8.0</td>
</tr>
<tr>
<td>160</td>
<td>86</td>
<td>8.5</td>
</tr>
<tr>
<td>180</td>
<td>87</td>
<td>9.0</td>
</tr>
<tr>
<td>200</td>
<td>88</td>
<td>9.5</td>
</tr>
</tbody>
</table>

![Figure 4. Efficiency vs grid number.](image1)

![Figure 5. Head vs grid number.](image2)

3. Results

The performance of the waterjet pump is analyzed with the tip clearance relative to the diameter of the impeller $\delta$ taken as 0.1%, 0.3%, 0.5% and 0.7%, and the flow rate is changed from 400 kg/s to 560 kg/s.

3.1. Performance Analysis of the Pump

The hydraulic performance of the pump is shown in Fig. 6. When the rate of flow is 484 kg/s (the design condition), the efficiency of the pump is close to 87.5% at $\delta = 0.1\%$ and 0.3%, both of which are high. At the same flow rate, with the decrease of the tip clearance the efficiency of the pump increases. But when $\delta$ is 0.3%, the efficiency increases rapidly under the low condition (400 kg/s-484 kg/s), which is lower than that under the same flow rate at $\delta = 0.1\%$. Under the same flow rate, the larger the tip clearance, the lower the head. The increase of the tip clearance will produce a larger clearance leakage, leading to an increase in the clearance leakage loss, and a reduction in the head. When the tip clearance increases to a certain extent, the performance degradation accelerates under low flow conditions. For example, when the flow rate is 400 kg/s at $\delta = 0.7\%$, the efficiency and head of the pump decrease the fastest. Compared with the flow rate of 440 kg/s, the efficiency decreases by 6.12% and the head cuts down by about 6.9%, which is significantly lower than the other three size types. This is because when the tip clearance increases, the clearance leakage loss plays
a major role in the flow loss. At the same time, under the same tip clearance, the head also cut down with the increase of flow.

3.2. Analysis of Flow Field

The static pressure distribution of pressure and suction part of impellers under design conditions are investigated, as shown in Figures 7 and 8. According to Figures 7 and 8, the narrow high-pressure area of impeller inlet (elliptical area on the right side of Figure 7) at \( \delta = 0.1\% \) is large, and the low-pressure area (elliptical area on the left side of Figure 8) appears, which causes large disturbance to the liquid, especially near the blade tip. With the increase of tip clearance, the high-pressure area at the impeller inlet and the low-pressure area on the suction surface are continuously improved. When \( \delta \) is 0.7\%, the pressure distribution of the impeller has been relatively uniform. Since blade cavitation usually occurs at low pressure, the blade inlet is more prone to cavitation when the tip clearance is small. The tip clearance has a certain effect on the low-pressure area of the pressure part of the impeller, and the high-pressure area near the outlet tip (the elliptical area on the left side of Figure 7) becomes smaller. This is due to the liquid flow from the pressure side to the suction side through the tip clearance increases, which expands the low-pressure area in the middle of the blade and reduces the high-pressure area near the outlet tip.

The entropy distribution of impeller under different tip clearances under design conditions is analyzed, as shown in Figure 9. At \( \delta = 0.1\% \), the entropy loss at the suction surface of the impeller tip is small. With the increase of the tip clearance, the entropy loss near the blade tip begins to increase. When \( \delta \) is 0.7\%, the entropy loss caused by tip clearance leakage at the suction surface has been very large, and the clearance leakage is more serious. This is because when the tip clearance is small, the friction resistance of the impeller wall is relatively large, the clearance leakage is small, and the kinetic energy of the clearance leakage is relatively small compared with the mainstream. When the tip clearance increases, the clearance leakage flow increases continuously, its disturbance to the mainstream increases at the same time, and the flow loss on the suction surface side of the impeller also increases gradually. This is the reason why while there is the weak corner separation at \( \delta = 0.7\% \), its efficiency and lift are lower than that at \( \delta = 0.1\% \) under design conditions.
Figure 7. Static pressure distribution of rotor pressure surface under different tip clearance.

Figure 8. Static pressure distribution of rotor suction surface under different tip clearance.
(a) δ=0.1%  (b) δ=0.3%
(c) δ=0.5%  (d) δ=0.7%

Figure 9. Entropy distribution of rotor under the design condition.

(a) δ=0.1%  (b) δ=0.3%
(c) δ=0.5%  (d) δ=0.7%

Figure 10. Stator suction surface streamline under the design condition.
Finally, Figure 10 analyzed the limit streamline of the suction part of the blade under different tip clearances under design conditions. With the increase of tip clearance, the angular separation of the suction part gradually decreases, which is serious especially at $\delta = 0.1\%$. And the separation area is large and there are separation spiral points, which is more likely to produce cavitation in practical application, so it is not suitable for this pump at $\delta = 0.1\%$. At the same time, the angular separation area of the stator of the other three tip clearances is not much different, which is small. Combined with the analysis of the performance characteristics of the pump, the relative tip clearance is selected as 0.3% for the pump.

4. Conclusion

Tip clearance has different effects on hydraulic performance and flow field of waterjet pump. In this paper, the hydraulic performance of four different tip clearances at different flow rates is studied to provide a basis for selecting reasonable tip clearances. It can be found that the tip clearance has some effect on the pressure distribution at the impeller inlet, and the angular separation of the suction surfaces is reduced. When the relative tip clearance increases to 0.7%, the leakage loss further increases. At this time, the clearance leakage loss plays a major role. Therefore, the hydraulic performance of the pump decreases when the tip clearance increase. Since the angular separation of the blade is serious when the relative tip clearance is 0.1%, and the flow loss caused by the leakage when the relative tip clearance is 0.7%, the relative tip clearance selected for the pump is 0.3%.

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References

