Influence of impeller and guide vane hub size on hydraulic performance of mixed-flow pump

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Abstract. Because of the beneficial properties of the mixed flow pump, it is widely used in the fields of irrigation and drainage. Our research team designed a mixed-flow pump, which with maximum efficiency more than 87%. However, its maximum operating efficiency range is biased towards large flow areas. In order to find the reasons for deviation from the design head, we have a three-dimensional modeling of the pump system and numerical simulation to analyze the internal flow. The numerical simulation results agree well with the experimental results and the experimental results verified the accuracy of the numerical simulation. When Q = 288 L/s, 360 L/s and 432 L/s, poor flow conditions appear near the hub. We refer to change the impeller and guide vane hub ratio method which only change the impeller and guide vane hub radius as blade shape remains unchanged. The result of this approach shows that it is possible to shift the highest efficiency point to the design point, but this can result in poor impeller performance. So we try to only change the guide vane hub radius and numerical simulation results show that it is effective. Finally, in the design head point, experimental test proved that when the guide vane hub radius is increased by 11 mm, pump system efficiency is increased by about 2.3%. This article can provide some references for the design of a mixed-flow pump.

1. Introduction
The specific speed of the mixed-flow pump is usually between 300-600, and the usual range of head is 10-20 m[1-3]. Mixed-flow pumps have many advantages, such as it can provide a higher head, it is present with wide ranges of high efficiency area and is not easy to cavitation but easy to start[4-7]. Because of the beneficial properties of the mixed-flow pump, it is widely used in the fields of irrigation and drainage. Impeller and guide vane is the core parts of the whole unit, which directly relates to the efficiency, safety, credibility and cost of the equipment. The rational matching of the impeller and the guide vane is the most important thing for the designers[8-10].

The hub-tip ratio is one of the most important parameters for the impeller. The hub is used to fix the blades. It ensures the requirements of the structure and strength, it is also can install and adjust blades. In terms of hydraulic loss performance, reducing the hub-tip ratio can reduce the loss of hydraulic friction[11,12]. When the flow area is increased, it is favorable to improve the cavitation performance of the impeller. But overly reducing the hub-tip ratio will increase the twist of the blade. When deviating from the design of the pump it causes the disorder of the liquid flow such as inverse flow at the outlet of the impeller. It will also decrease the efficiency of the pump and narrow the range of high efficiency. Therefore, rational selection of the hub
radius is extremely important to the performance of the pump. Our research team designed a mixed-flow pump, which has a maximum efficiency of more than 87%. However, its maximum operating efficiency range is biased towards large flow areas. Pump performance as showed in Figure 1.

2. Numerical simulation

Figure 2 shows geometric model of mixed-flow pump device, which is used for measurement of the entire flow field. The impeller radius is 300 mm, the number of blades is 5, the design flow rate is 360 L/s, the design head is 14 m, the specific speed as 438, and the rotation speed of impeller set at 1450 r/min. The guide vane matched with the impeller is selected and its number of blades is 11.

This paper used the Turbo-Grid software to build the model and carry out the mesh division of the guide vane and the impeller. The tip clearance of the impeller has a great effect on the efficiency of the device when creating the impeller model, the effect of tip clearance is taken into account, and this paper sets the tip clearance at 0.2 mm. Used Pro/E software to build models of the inlet straight pipe with water-guiding cone and the standard 60 outlet pipe, then used the ICEM software to carry out the structured-grid division. After inspection, the quality of the grid is reached more than 0.4 which is satisfied the requirements of the calculation. And then, we did the mesh independence of the total model at the design flow rate, specific data shown in Figure 3. When the total number of pump device mesh is more than 3 million, calculating efficiency will be unchanged while the mesh number increases. In the end, we chose the total number of mesh as 3226655, at this time, the number of impeller mesh is 1132745, the $y^+$ of the impeller tip clearance is about 65, the number of guide vane mesh is 1160005. According to Document [13], the number of mesh meets the calculation requirements. The meshes of the impeller are presented in the Figure 4.

3. Test verification

The external characteristic test of mixed-flow pump device is made to verify the reliability of numerical simulation results, by using high precision hydraulic machine test-bed in Yangzhou University of China. The test-bed is vertical and closed circulation system. Select the mixed-flow pump device which is consistent with the numerical simulation and set the speed at 1450 r/min.
Figure 2. Numerical calculation domain.

Figure 3. The mesh independence of the whole model.

Figure 4. The meshes of the impeller.

Figure 5. The impeller and guide vane for experiment.
The impeller and guide vane for the experiment are shown in Figure 5. The flow rate is measured by an electromagnetic flowmeter (DN400 mm), the head is measured by a differential pressure transmitter (EJA110A), the shaft power is measured by a rotary torque meter (ZJ 500 N•m), and calculated by the Equation 1. The efficiency of the device is determined by Equation 2.

\[ N = \frac{\pi}{30} n(M - M') \]  
\[ \eta = \frac{\rho g Q H}{N} \times 100\% \]

\( N \): the shaft power, W; \( n \): the rotating speed, r/min; \( M \): the input torque, N•m; \( M' \): the loss of torque, N•m.
\( \eta \): the test efficiency, %; \( \rho \): the density of water, kg/m\(^3\); \( g \): gravity acceleration, m/s\(^2\); \( Q \): the flow rate, m\(^3\)/s; \( H \): the head, m; \( N \): the shaft power, W.

The energy result comparison of numerical simulation and test is shown in Figure 6. Prediction (CFD) shows agreement well with the experimental efficiency (EXP) and the variation trend is the same. At the design flow rate point \( Q=360 \) L/s, the results of experiments and numerical simulations are essentially the same. Compared with the test results, prediction head curves are little bit deviated, large flow rates is higher while small flow rates are on the opposite. Although the two curves do not correspond completely, the data error is within 5%. Our main consideration is to improve the design point performance, so this result meets our analytical requirements.

4. Numerical Simulation Results Analysis

According to the above calculation results, take out three flow rates (\( Q=288 \) L/s, 360 L/s and 432 L/s) to analyze the distribution of internal pressure in the plane where after the blade of guide vane. The pressure distribution is shown in Figure 7. In general, the greater the flow, the overall pressure is lower. As \( Q=288 \) L/s, the pressure is divided into four parts by the near two blades. As \( Q=360 \) L/s, the pressure distribution is uneven in the right and left direction, the low pressure area is roughly distributed between the hub and the shroud. As \( Q=432 \) L/s, the pressure is roughly distributed around the circumference, from the shroud to the hub, the pressure becomes lower.

Similarly, the velocity distribution is shown in Figure 8. The three operating conditions have obvious high-velocity and low-velocity areas. With the increase of the flow rate, the range of high-velocity areas also becomes larger. Similar to the situation of pressure distribution, as the flow rate increases, the position of the low velocity area also approach to the hub.

From the pressure profile and the velocity profile, the pressure and flow velocity distributions are not uniform. The rectification effect of guide vane is not well, inevitably leads to increased hydraulic loss. If used in the pumping station, will result in low efficiency and even unstable operation. It is necessary to analyze the flow state inside the impeller and the guide vane. As the span of the hub to shroud is 0.1. Take out the velocity vector inside the impeller and guide vane, the result shown as Figure 9. The internal flow in the impeller is smooth and there is no vortex, but in the guide vane is terrible. With the flow rate increased, the swirl area from the leading edge of the blade move to near the trailing edge inside the guide vane, the swile area is smaller also. This also shows that there is enough room inside the guide vane to allow water on the design condition through. In other words, the guide vane inner space is too large for small flow rate.
Figure 6. The energy result comparison of numerical simulation and test.

Figure 7. Different conditions of pressure distribution.

Figure 8. Different conditions of velocity distribution.

5. Optimization measures
Several early investigations of rotor-stator interaction were carried out. We have taken many traditional methods to change the guide vane also, such as changed the number of blade or varied the distance of impeller and guide vane. In this article, we want to introduce a new simple and effective method.

In order to get the highest efficiency point back to the design point, we consider minimizing the impact on the impeller performance, therefore does not alter the impeller hub ratio to prevent excessive distortion of the blade, while the hub radius of the impeller and the guide vane is enlarged by 5.5mm and 11mm at the same time. The performance curves after changing the hub radius is like the Figures 10-11.

Increasing the radius of the hub will reduce the head of the pump, the greater the flow rate,
the more obvious the head is reduced. As the radius of the impeller and guide vane increases by 11mm, the head of pump drops to about 60% and therefore no further increase in the hub radius. The pump head decreases, indicating that the loss of the pump increases or apply work of the pump decreases.

However, the flow rate-efficiency curves have a different change from the flow rate-head curves. When the flow rates are less than 396 L/s, the bigger the hub radius is, the higher the efficiency is. When flow rates exceed 396 L/s, the efficiency decreases as the hub becomes larger, and the greater the flow rate, the more obvious this trend. We look at the performance of guide vane and impeller separately, hoping to be able to find out the reasons for this change.

Calculate the hydraulic loss from the guide vane inlet to the calculation domain outlet, hydraulic loss curves for different hub radius are shown as Figure 12. The hydraulic loss of the guide vane of different schemes is varied distinctly, increasing the radius of the hub can shift the position with the least hydraulic loss to a small flow rate. As Q=360 L/s, the hub radius increases by 11mm and the hydraulic loss decreases by approximately 50%. This is also the main reason for the increase in the efficiency of small flow rates. At large flow rates, hydraulic losses increase with radius, which is an important reason for the efficiency decline.

Due to the changes of the hub radius, the performance of the pump system has been greatly

Figure 9. Velocity vector inside the impeller and guide vane.
changed, the efficiency of the small flow rates has been improved, but at the expense of large flow rates performance. Therefore, it is required further analysis of the flow conditions in the impeller and guide vane.

In order to understand the influence of the hub radius on the internal flow state, as Q=360 L/s, we analyzed the flow field inside the impeller. The blade pressure distribution as shown in Figures 13 and 14. Under the same operating conditions, increasing the hub radius of the impeller will cause the internal flow area of the impeller to decrease and increase the flow velocity in the channel. On the suction side of the blade, there is an obvious low pressure area near the middle of the blade surface, as the radius of the hub increases, the area of the low pressure area also increases, the increase of the low pressure area will lead to the deterioration of blade cavitation performance.

As the hub radius increases, the whole pressure on the blade pressure surface decreases, the working capacity of the blade is weakened leading to head of the impeller reduction. Owing to increase the flow velocity in the flow channel, the friction loss between the water flow and the flow-through components increases, original pump design condition has also changed, which also results in the decrease of the pump performance. From the above analysis about the design point operation, increasing the hub radius will lead to a drop in the impeller performance and reduce the hydraulic loss of the guide vane. From the flow rate-efficiency curves, reveal that the improvement of guide vane performance compensates for the reduction in impeller performance. Although this can increase the efficiency of the pump at small flow rates, the unstable operation of the impeller may cause some new problems. In order to keep the impeller performance unchanged, we sought to only change the hub radius. The initial impeller matches calculation with guide vanes whose hub radius increases by 5.5 mm and 11 mm respectively. Similarly, take the plane after the outlet of the guide vane blade, as Q=360L/s to analyze the distribution of internal pressure and velocity. The pressure distribution as shown in Figure 15.
Figure 12. Flow rate-hydraulic loss curves.

(a) $\Delta r_1 = 0$ mm  
(b) $\Delta r_1 = 5.5$ mm  
(c) $\Delta r_1 = 11$ mm

Figure 13. The pressure distribution of the blade suction surface.

(a) $\Delta r_1 = 0$ mm  
(b) $\Delta r_1 = 5.5$ mm  
(c) $\Delta r_1 = 11$ mm

Figure 14. The pressure distribution of the blade pressure surface.

This method is only changing the hub radius of the guide vane by making it simple and very effective. The flow state inside the guide vane is enhanced, basically no effect on the flow in the impeller, and the overall performance of the pump is improved. Inside the guide vane, there are still unfavorable flow patterns. Next step is to consider take comprehensive measures to eliminate.

6. Test comparison
Based on our calculations, we processed the guide vane which showed an 11mm increase in hub radius. In the original test bench, only the new guide vane was replaced, a comparative test was made. The Figure 18 shows the performance of the comparative test.

Test results are respond to our expectations. The performance of the pump system has greatly
improved. In the small and design flow rates, both the head and efficiency increase obviously. When the flow rate exceeds 420 L/s, the efficiency decrease is due to the flow area reduced and caused the increase in hydraulic loss. Increase the hub radius of guide vane can shift the highest efficiency point to smaller flow rate and widen the range of high efficiency zones. As Q=360 L/s, increased efficiency by approximately 2.3%.

7. Conclusion
We are combined with numerical simulation and model experiment to optimize the design of the mixed-flow pump. Tested performance of the original pump system and verified the reliability of the numerical simulation. Numerical results show that the low pressure area seriously affects the internal flow state of the original guide vane, the pressure and flow velocity distributions are not uniform. The excessive flow area inside the guide vane is an important reason for the low performance of the design point. After improving the hub radius of the impeller and guide vane, small flow rates performance improves but sacrifices high flow rates performance. Increasing the
Figure 18. Comparison of test results under different hub radius of guide vane.

hub radius of impeller and guide vane, will lead to a drop in the impeller performance at the design point and reduce the hydraulic loss of the guide vane. The reduction in the performance of the impeller will cause the instability of the pump operation. Only enhance the hub radius of the guide vane by 11mm will improve the overall pump performance. Comparing two guide vanes through model test, increasing the hub radius by 11 mm can improve the design point efficiency by approximately 2.3 %.

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