The Effects of Different Splitter Blades Number on Characteristics of Miniature Super-Low Specific Speed Centrifugal Pump

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Abstract. Splitter blades are often used to improve the performance parameters of super-low specific speed centrifugal pumps, while the number of splitter blades is the most important influencing factor of head and efficiency. In order to study the effect of different number of splitter blades between long blades on the external characteristics and internal flow field of centrifugal pumps, the numerical simulation of four impeller models has been carried out by Fluent. The results show that splitter blades can raise head and efficiency of the centrifugal pump, so that the performance curve moves toward large flow rate area. The low speed area on the pressure surface of the long blade reduces with the slip and separation inhibited. When the number of splitter blades increase gradually, the head of the centrifugal pump has not much increase while the efficiency decreases. Most of the medium flow out through the runner on the pressure surface leaving the others blockage due to the backflow and vortexes, resulting in large hydraulic loss.

1. Introduction

Super-low specific speed centrifugal pumps generally refer to the centrifugal pumps with the design flow rate less than 6 m³/h and specific speed less than 30. The blade outlet width and impeller inlet diameter are both small. The flow channel is narrow. The inlet is severely squeezed and the efficiency is partial low. In order to improve the head, efficiency and internal flow, the design of splitter blades method is generally adopted [1].

With the development of computational fluid dynamics, using numerical simulation to predict performance and optimal design provides an important tool for the study of miniature super-low specific speed centrifugal pumps [2]. The number of splitter blades is the most important factor on the head and efficiency [3]. At present, there are many studies on the number of splitter blades. Chu discussed the performance of the centrifugal impeller of the micro-turbine by the number of pairs of blades with numerical simulation [4]. Li discussed the number of blades on the performance and internal flow field of the centrifugal pump with numerical simulation. The reasonable arrangement of the splitter blades can improve the head and efficiency [5]. Korkmaz E explored the effect of the number of blades alternating with long and short blades on the performance parameters of the deep well pump through experiments, while the best efficiency is obtained when the number of blades is 6.

The increase of the number of splitter blades can improve the theoretical head [6], but the actual heading effect is limited by the layout of the splitter blades and the flow characteristics in the impeller.
under specific working conditions. Most of the literatures at home and abroad are concerned about the arrangement of single splitter blades between long blades. There are still few studies on multiple splitter blades between long blades. Therefore, the study of impact on super-low specific speed centrifugal pumps with different number of splitter blades between long blades is of great significance.

In this paper, a miniature super-low specific speed centrifugal impeller under a high-pressure circulation system is selected as the research object. Multiple splitter blades are arranged between the long blades. The effect of the number of splitter blades on the external characteristics and internal flow field of the centrifugal pump is carried out by numerical simulation, providing a reference of design and optimization to super-low specific speed centrifugal pumps.

2. The layout scheme

Basic parameters of the miniature super-low specific speed centrifugal pump are as follow. The design flow rate is 0.36 m$^3$/h, the design head is 9 m, the speed is 3000 r/min, the specific speed is 21, the closed impeller inlet diameter is 19 mm, the outlet diameter is 70 mm, the blade outlet width is 1.5 mm, the long blade number is 5. The structure of the pump is shown in Figure 1.

![Figure 1. The pump structure.](image)

The hydraulic model is optimized by splitter blades and the blades are added to the original model M0. The profile is obtained from the long blade’s profile intercepted, adjusted and modified. The splitter blade are evenly arranged along the circumferential direction. The number of long blades is unchanged, the number of splitter blades between the long blades increase gradually. Three composite impellers M1 to M3 are obtained. The total number of splitter blades is 5, 10, 15 respectively. The layout scheme of the four blades is shown in Figure 2.

![Figure 2. The layout scheme of blades.](image)

3. Numerical simulation

3.1. Computational domain

In order to reduce the simulation error and the influence of the gap between the pump cavity [7], the numerical model of the whole flow field is conducted to predict the performance and internal flow distribution. Creo is used to perform three-dimensional modeling of the calculation domain including the inlet and outlet extension, suction chamber, front pump chamber, impeller flow path, vortex chamber, and rear pump chamber as shown in Figure 3.
3.2. Governing equations and turbulence model

The continuity equation and RANS equation used are as follows.

\[
\frac{\partial \rho}{\partial t} + \frac{\partial \rho u_i}{\partial x_i} = 0
\]  

(1)

\[
\frac{\partial}{\partial t} (\rho u_i) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \mu \frac{\partial u_i}{\partial x_j} - \rho u_i u_j' \right) + S_i
\]  

(2)

The Reynolds stress tensor is represented by the average velocity gradient [8].

\[
-\rho u_i u_j' = \mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \left( \rho k + \mu_t \frac{\partial u_i}{\partial x_j} \right) \delta_{ij}
\]  

(3)

The turbulent viscosity \( \mu_t \) is determined by the turbulent model. The SST \( k-\omega \) turbulence model in this paper uses the \( k-\omega \) model to predict the low Reynolds number flow in the near-wall region [9] and applies it to the boundary layer. The \( k-\epsilon \) model is not sensitive to the incoming flow conditions so it is applied to the turbulent core area of the far field. The SST \( k-\omega \) model can predict a wider range of flow and solve the flow separation problem better.

3.3. Grid generation and boundary conditions

ICEM is used to mesh the computing area. To improve the accuracy of the solution, hexahedral structured mesh is adopted. The independence validation has been carried out on the four groups of grids. As the number of grids increased, the relative changes in head and efficiency were all less than 0.5%. The total number of grids was finally determined to be 4 million. Taking M1 as an example, the grids of the impeller and the whole flow field are shown in Figures 4 and 5.

Figure 3. The whole flow field computational domain.

Figure 4. The grids of impeller M1.

Figure 5. The grids of whole flow field.

The numerical calculation is carried out by fluent with multiple reference frame model and steady solution. The inlet boundary condition is set to pressure inlet and the outlet boundary condition is set to mass flow outlet.
4. Results and discussions

4.1. Impact on performance

According to the simulation results, the $H$-$Q$, $P$-$Q$, $\eta$-$Q$ comparison curves of the four schemes are shown in Figure 6, Figure 7 and Figure 8.

It can be seen intuitively from Figure 6 that the arrangement of splitter blades can improve the head of the centrifugal pump. The head curve of M1 moves upward compared with M0 and the head in the design condition point is relatively increased by 10%, but the splitter blades can not eliminate the "hump" of the head curve. When the number of splitter blades in a single channel is increased to 2, the head of M2 is improved compared with the single splitter blade scheme M1, but the degree of improvement is not greater. When the number of splitter blades increases to 3, the head curve of M3 is basically the same as that of M2. It shows that the head can no longer continue to increase when the splitter blade increases to a certain number.

It can be seen from the power comparison diagram in Figure 7 that the power increases when the splitter blades increase, indicating that the splitter blades can improve the performance of the centrifugal impeller. As the number of splitter blades continues to increase, the power gradually increases. The greater the power increase, the more obvious the law is.

It can be seen from the efficiency comparison chart in Figure 8 that the best working point of the ordinary impeller M0 is $1.8Q_d$. The best working point of the impeller M1 with a splitter blade is $2.0Q_d$, indicating that the splitter blade makes the best working point move to the large flow area and improve the efficiency of the large flow operating area. The changing speed of efficiency in M1 at the optimal point slows down and the high efficiency area of the centrifugal pump is widened. The efficiency of M2 is basically the same as that of M1, but the efficiency of M3 has decreased to a certain extent. The reason is that when the number of splitter blades increases, the increase of power is higher than that of head, resulting in a decrease in efficiency.

4.2. Impact on internal flow field

Figures 9 are comparative graphs of the relative velocity distribution of M0, M1, M2 and M3 under the design conditions. It can be seen that the splitter blades play a diversion role which drain part of the high-speed medium on the suction surface of the long blade to the pressure surface in M1, so the area of the low-speed area on the pressure surface is greatly reduced. The boundary layer separation is suppressed. However, the splitter blade also brought a certain disturbance. The boundary layer separation occurs in the low-speed region on the long blade suction surface near the outlet of the splitter blade. With the gradual increase of the number of splitter blades, the low speed area of the long blade pressure surface gradually decreases with the slip and separation inhibited. In M2 and M3, a high-speed zone forms in the splitter channel on the pressure surface of the long blade. The greater the number of splitter blades, the smaller the overflow area is. That is, most of the medium flows out through the splitter channel. In the diffusion-type flow channel, the local high-speed area means that the flow distribution is not equal indicating that overmuch short splitter blades may interfere with each other and cannot achieve the purpose of drainage. In other splitter channels, more serious backflow and vortexes are generated which cause the blockage of the outlet channel. The blocking effect restricts the positive effect of the splitter blades, reducing the efficiency of the centrifugal pump limited with a certain extent.
It can be seen from the comparison of the internal flow field that the splitter blades mainly improve the relative velocity distribution, with the slip and separation of the main flow region suppressed and the performance of the impeller improving. When the number of splitter blades is overmuch, it can not improve the internal flow distribution on the contrary. The disturbance effect to the flow field will aggravate, resulting in the blockage to some flow channels and large hydraulic loss.

5. Conclusion

Four types of impellers with different numbers of splitter blades were simulated to study external characteristics and internal flow field of the super-low specific speed centrifugal pump.

(1) With the splitter blades arranged, the head of M1 is relatively increased by 10%. The power and efficiency are significantly improved. The performance curve moves toward the large flow operating area. When the number of splitter blades come to M2, the head slightly increases and the power increases. The efficiency is basically unchanged. When the number of splitter blades is to M3, the head is basically unchanged. The power increases and the efficiency decreases instead.

(2) The splitter blades can reduce the adverse effects of the tongue. They can also reduce the low-speed area on the pressure surface of the long blade greatly and suppress slip and separation. When the number of splitter blades is overmuch in M2 and M3, the disturbance of the flow field exacerbates. Most of the medium flows out through the splitter channel near the pressure surface of the long blades. The other channels have serious backflows and vortexes, which cause blockages and greater hydraulic loss.

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