Study on the influence of back blade shape on the wear characteristics of centrifugal slurry pump

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Abstract. CFX particle inhomogeneous model was introduced for the mechanism analysis of a centrifugal slurry pump which is equipped with back blades on impeller shrouds. Combining with the total efficiency correction, the simulation showed good prediction accuracy of external characteristics results compared with the experimental values. Vorticity and Q-Criterion were chosen as the variables to illustrate the abrasion morphology and wear mechanism by contrasting simulation result with worn impeller in engineering. The analysis showed that the large vorticity intensity areas are distributed at the edge of impeller shroud and intensively behind the back blades. Moreover, the vorticity scattered on suction surface of back blade shows the largest intensity. The contour of Q-Criterion demonstrated that the swirl scale in front cavity is obviously larger than that in back cavity. The distribution of vorticity on both front and back shrouds can reasonably explain the impeller wear characteristics. Finally, the forward curved back blade proved to be excellence performance in vorticity distribution.

1. Introduction
Centrifugal slurry pump is one of the most important fluid machines, which has been widely used in industries such as mine, coal and power [1-2]. Generally, the shroud of centrifugal slurry pump impeller is equipped with the back blades that rotate synchronously with the impeller, and the back blades can prevent the medium of solid and liquid being leaked through the cavity clearance in pump body for the high pressure around the outlet of impeller. Moreover, the back blades also adjust the axial force by increasing the movement speed of mediums in the clearances. However, in the engineering application, the impeller blades and the back blades trend to be worn even invalid due to the influence of hard solid particles with complex multiple flows. Therefore, it is of great importance to investigate the effect of back blade shape on the wear characteristics of centrifugal slurry pump. In the past few years, it has been studied by simulation and experiment research. Wang [3] investigated the wear characteristics of centrifugal slurry pump by analyzing the solid volume fraction, solid phase slip velocity and the motion laws of different diameters of solid particles. Wu [4] adopted the discrete phase model to study the effect of rotation speed, the diameters of solid particles and the parameters of impeller on the features of the particle erosion behavior. Tao [5] used the multiphase model to simulate the abrasion conditions of the impeller in a semi-open centrifugal pump, and the simulation results were also confirmed by the rapid wear test.

All the representative researches mentioned above utilized the conventional variables to
characterize the abrasion on impeller blades and passages, but there are not many articles that discuss the back blade on shrouds. Therefore, in this paper, numerical simulation for the slurry pump with the radial back blade is conducted by ANSYS CFX 14.0. The medium is the mixture of water and solid particles, whose volume fraction is 10%. This study analyzes the wear characteristics on impeller back blades and the improvement project is also put forward to ameliorate the abrasion.

2. Physical model

2.1 Slurry pump model and the main parameters

The main parameters of slurry pump model are presented in Table 1.

| Parameters | \( Q_d \) | \( H_d \) | \( \eta \) | \( \text{Shaft power } P_e \) | \( \text{Rotate speed } n \) | \( \text{Specific speed} \) |
|------------|-----------|-----------|-------|-----------------|-----------------|------------------|
|            | 500 m³/h  | 72 m      | 65.2% | 150.2 kW        | 1450 r/min       | 81.5             |

Table 1. Design parameters of slurry pump.

Outlet diameter \( D_2 \) | Outlet width \( b_2 \) | Number of impeller blade | Number of back blades on front shroud | Number of back blades on back shroud | Back blade width
|-----------------|-----------------|-------------------------|-------------------------------|-----------------------------------|-----------------|
| 515 mm          | 28 mm           | 5                       | 16                           | 8                                | 12 mm           |

The structure diagram of centrifugal slurry pump channel is shown in Figure 1, the clearance distance between the top of back blade and the pump body is 2mm. The impeller inlet channel is sealed up by angular contact seal, and labyrinth seal is designed in the place closed to the impeller inlet to prevent the mediums from being leaked out. The back blade of the original model is radial type \( (\beta_2' = 90°, \ \text{and } \beta_2' \text{ represents the outlet angle of back blade}) \), as shown in Figure 2.

Figure 1. Structure diagram of the flow channel.

2.2 Calculation model

All the pump components are modeled by Solidworks, which are based on the 2D engineering drawings. The 3D models include: inlet section, inlet liner, impeller, volute and shroud liner. The whole domain is built by the process of assembling and Boolean operation. To set the boundary conditions explicitly, the whole domain is divided into four parts: inlet, cavity, impeller and volute. All the full flow channels of the slurry pump are shown in Figure 3.
2.3 Grid division
As the twisted spatial structure of computational domains are rather complex, the unstructured tetrahedron grids for all the models are generated by ICEM, then the verification for grid independence is also carried out. When the total number of the grids is more than 2812520, the performance parameters are stabilized, and the relative errors of head and efficiency are respectively 0.68% and 0.78% when the total number is 3655673. Therefore, conclusion can be drawn that the quality of grid and the precision of simulation can satisfy the requirements when the total number of divisor grids is about 2.8 millions. The grid models are shown in figure 4.

3. Numerical simulation method
The volume fraction of solid particle in the mixture is 10% and another phase is water. The solid particles are set as the spherical pulverized coal which assumed to be spherical, whose diameter \( d = 0.2 \) mm, density \( \rho = 1350 \) kg/m\(^3\). Considering the complex flow conditions such as streamline curvature and whirlpool in the pump, the continuous phase adopts the RNG \( k-\varepsilon \) turbulence model, while the discrete phase adopts the zero equation model. Particle model is defined in phase transport model, and drag coefficient adopts the Schiller Naumann model which is suitable for the spherical particles. The interphase turbulent diffusion is Favre Averaged Drag model and turbulent diffusion coefficient is 0.9.

Velocity inlet is used as the boundary condition for the inlet. The Opening Pressure and Direction is chosen as the boundary condition for the volute outlet, where the static pressure is given by the theoretical head. The gradient of solid volume fraction on the outlet is 0. The interfaces between the
static and motive areas are set by “Transient Rotor Stator” model in transient calculation. The initial value of transient calculation comes from the result of steady calculation. The physical timescale $\Delta t = 1/n = 0.000689655s$ (n represents the rotate speed of impeller), and impeller rotates for 3 circles totally. High resolution is chosen for the transient solution model, which ensures the accurate calculate process. Monitor points set on the inlet and outlet surfaces are used to judge the level of computation convergence. The max residual of transient calculation is 0.00001.

4. Simulation results and analysis

4.1 Model validation

Test stand was set up in certain fluid machinery Co., Ltd. in Shanghai to perform the pure water performance experiment for the prototype model. Test standard is referred to the standard of centrifugal pump, francis pump, axial pump and vortex pump (GB3216-1989). Simulation and experimental results are shown in figure 5 and figure 6.

![Figure 5. H-Q curve.](image)

![Figure 6. $\eta$-Q curve.](image)

The formula of efficiency:

$$\eta = \frac{1}{\eta_h + \frac{\Delta P_d}{P_v} + 0.03}$$

(1)

Where, $\Delta P_d$ represents the disk friction power loss, Kw. The calculation method of $\Delta P_d$ can be referred by related references [6]. $\eta, \eta_h$ and $P_v$ can be calculated by simulation.

4.2 Wear analysis of impeller back blades

CFD-Post is utilized to analyze the calculation results. Vorticity distribution contour of solid phase is illustrated to analyze the wear situation on shroud and back blade. Vorticity is one of the most important parameters to describe the feature of eddy, which can directly characterize the swirl strength in the flow space. Solid particle is a kind of high hardness, strong abrasive medium. When the solid particles flow fast over the surface of flow components, especially for the impeller shrouds and back blades, they will lead to severe erosion and abrasion. According to the vorticity contour showed in figure 7, it can be observed that most of the vorticity intensity emerge around the edge of shroud and they are comparatively concentrated behind the back blades, which are closed to the suction surface of back blades. Although the vorticity intensity are slightly different in their distribution, location and
magnitude, obvious laws are still visible on the whole. The impeller dismantled from centrifugal pump unit in overhaul is shown in figure 8. From overview of the wear appearances on shroud, we find that long spiral-fluted and cured eyebrow grooves are consistently observed on each back blade passage, and they just look like the hooks along the peripheral direction approach to outlet edge, but carved into the shroud. Along the radical direction, the back blades are worn severely with the increase of impeller radius and the height of back blade becomes flat gradually, so we can deduce that the particle moment velocity is raised with the increase of radial distance, and wear extent is positive correlated with the particle velocity. It is obvious that the worn areas display bright metallic luster, which is resulted from the cutting action of the high speed particle on impeller shroud.

![Figure 7. Vorticity distribution on front shroud.](image1)

![Figure 8. Wear morphology on front shroud.](image2)

It can also be observed from figure 9 that there are some inconsistent areas of strong vorticity intensity on the suction surface of back blades (small range of yellow area), while continuous vorticity intensity are unlikely to be emerged on the pressure surface of back blades. The severe worn back blade in engineering is showed in figure 10. The outlet circumference edge is skived to be flake. The actual worn morphology in different back blades passages seems different and there are two or three grooves on most of the back blades. Furthermore, Q-Criterion is also defined as a variable to analyze the instantaneous vortex structure in three dimensional spaces, as is showed in figure 11. The closed envelopes in front cavity gap are obvious larger than that in back cavity, which indicates that the vortex scale is larger in the front cavity gap. Based on the worn morphology combined with simulation results, it can be deduced that the abrasion happened around the shrouds and back blades are caused by the strong vortex flow with high hardness solid particle medium.

![Figure 9. Vorticity distribution on back blade (front shroud).](image3)

![Figure 10. Wear morphology on back blade (front shroud).](image4)

From figure 12 to 15, it is obvious that the back shroud and its back blades are seriously worn than that of the front shroud. The number of blade on back shroud is less than that of the front shroud, and this means that heavier load must be burdened for each back blade on the back shroud, which will increases the probability for the solid particle colliding to back blade surfaces. The actual abrasion
circumstances show that the back blade passages have been linked into the impeller passages. However, any impeller structure out of work must be rejected in centrifugal pump. Differ from the bright metallic luster displayed on the front shroud, corrugated or scaly morphology are the main surface features on the back shroud. This comes from the reason that the swirl scale on back shroud is smaller and more dispersed. In general, the abrasion characteristics on back shroud are similar to the front shroud. The analysis above can reflect and illustrate the impeller abrasion. Moreover, the structure integrity of impeller will also greatly affect the imbalance and vibration of the rotor. Therefore, structure optimization and anti-wear performance are urgently studied in the next.

![Front cavity gap](image1.png) ![Back cavity gap](image2.png)

**Figure 11.** Contour of Q-Criterion.

![Vorticity distribution on back shroud](image3.png)

**Figure 12.** Vorticity distribution on back shroud.

![Wear morphology on back shroud](image4.png)

**Figure 13.** Wear morphology on back shroud.

![Vorticity distribution on back blade (back shroud)](image5.png)

**Figure 14.** Vorticity distribution on back blade (back shroud).

![Wear morphology on back blade (back shroud)](image6.png)

**Figure 15.** Wear morphology on back blade (back shroud).

4.3 The optimization of back blade shape

The analysis result shows that the back sides of back blades suffer the most severe abrasion. In order to reduce the interference effect of back blade on the vortex, forward curved back blade is optimized
to compare with the radial blade. Number, width and length along radial direction of all the back blades on front and back shroud remain unchanged. Then the same grid meshing and numerical boundary conditions are consistent with the original model.

As we can see in figure 16 and 17, the number of back blade passage seized by large vorticity intensity has been decreased in quantity. There nearly no continue distributed vorticity intensity around the edge of shroud, and distribution range of vorticity is also markedly shrunk on the suction surface of back blade. The figure 18 and figure 19 show that the value of particle vorticity on suction surface (such as range of the yellow area) also has been reduced. The integral improvement of vorticity intensity is acceptable compared with the situation before. In the meanwhile, the influence of back blade optimized on the performance should be considered to expound the feasibility of the optimization. For this reason, 0.2Qₘ, 0.4Qₘ, 0.6Qₘ, 0.8Qₘ, 1.0Qₘ and 1.2Qₘ flow rate are calculated for the new model, and then the performances, like head and efficiency curves, are contrasted with the original model, as showed in figure 20.

Figure 16. Vorticity distribution on front shroud.  
Figure 17. Vorticity distribution on back shroud.  
Figure 18. Vorticity distribution on back blade (front shroud)  
Figure 19. Vorticity distribution on back blade (back shroud)

The average values of head and efficiency in six different operation conditions are taken into account. The H-Q curve and η-Q curve of two models both show the similar variation law. Compared with the head of radial back blade model, the forward curved back blade model decreases by 3.25%. In the same way, the efficiency of forward curved back blade model decreases by 0.86% compared with the radial back blade model. So the effect of change of the model on head and efficiency are limited, and the forward curved back blade can effectively improve the distribution of vorticity, which is directly related to the back blade abrasion. Therefore, sacrificing few amounts of head and efficiency to prolong service life of impeller is a feasible way in engineering.
5. Conclusion

(1) The transient numerical simulation used in the research can get consistent results with experimental values. In the calculated flow rate, the average relative errors of head and efficiency between calculation and simulation are 1.28% and 1.93%. This comparison indicates the accuracy of calculation model and applicability of efficiency correction.

(2) The vorticity distribution contours and envelopes of swirl scale based on Q-Criterion can reasonably explain the wear mechanism. It can be speculated from the abandoned impeller that long spiral-fluted and cured eyebrow grooves are caused by strong vortex flow behind the suction surface of back blade. The simulation results of vorticity distribution have a good agreement with the severe abrasion morphology on worn impeller.

(3) According to the shape of worn grooves on shroud, forward back blade model is designed to verify the effectiveness on the improvement of vorticity distribution. The optimization result shows that the range of vorticity intensity on shroud has been shrunk largely. In the meanwhile, the influences brought by geometry change on external performances of centrifugal pump have also been considered. It is worthy of sacrificing few amount of head and efficiency to improve the abrasion on shrouds and back blades.

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