Effect of the Bionic Circular Groove Non-Smooth Structure on the Anti-Wear Performance of the Two-Vane Pump

Yunqing Gu 1,2, Muhan Yan 1,2, Jiayun Yu 3, Ke Xia 4, Longbiao Ma 1,2, Jiegang Mou 1,2,*, Denghao Wu 1,2 and Jianxing Tang 5

Abstract: The characteristics of the material transported by the two-vane pump can cause the impeller to wear out, leading to a deterioration in hydraulic efficiency. Appropriately, the research goal of this paper is to consolidate the anti-wear performance of the two-vane pump conveying a solid-liquid two-phase flow. Based on the bionic principle and the anti-wear structure of blood clams, the circular non-smooth structure adapted from blood clams is arranged in the wear-prone area. Through numerical simulation, we compare the main indexes of the pump: the head, the pressure distribution, the vortex pressures, and the average wear rate, to reveal the wear resistance mechanism of circular non-smooth structures. The results illustrate that the use of a circular non-smooth structure does not modify the external characteristics of the pump; the pressure distribution inside the impeller is similarly consistent, and the vortex pressures are all approximately the same. The average wear rate is higher when the diameter of the circular non-smooth structure is either 0.25 mm or 0.30 mm, and the simulation results are poor. At a diameter of 0.20 mm, the average wear rate of circular non-smooth blades is at its lowest point. The circular non-smooth surface structure causes impurities to be “caught” by the vortex zone and not freely struck against the wall, resulting in the particles migrating away from the blade.

Keywords: two-vane pump; circular non-smooth structure; numerical simulation; flow field characteristics; resistance to wear and tear

1. Introduction

The two-vane self-priming pump is a sewage centrifugal pump. It has a simple hydraulic structure: two symmetrical blades. This structure makes it not easy to clog [1], and compared with other blade pumps, two-vane pumps have symmetrical flow channels, balanced operation, and excellent performance, so they are widely used in slurry situations containing solid-phase impurities [2]. The physical properties of the slurry depend on the type, size, shape, and number of particles; the distribution of solid particles in the liquid; the concentration of solid particles in the liquid; the size of the conduit; the turbulence level; the temperature of the carrier [3]; and the absolute viscosity. As the core hydraulic element of the two-blade pump, the anti-wear performance of the impeller also represents the anti-wear performance of the sewage pump [4]. Research has identified that the world’s energy consumption is anticipated to swell by 28% from 2020 to 2040 [5]. According to a study conducted by the International Energy Agency [6], currently, electric motors consume 46% of the world’s electricity, and centrifugal pumps consume 25% to 60% of factory motor energy, depending on the industry [7]. Therefore, improving the energy consumption of centrifugal pumps could reduce global energy consumption. With the
expansion of centrifugal pump applications [8], pump wear is an important consideration as it affects the initial cost and the component life of the pump. The role of the double-blade pump is to transport sewage, which mainly contains sand, dirt, and other solid impurities. These impurities will continue to impact and cut the surface of the flow parts in the process of transportation and wear is inevitable, leading to the failure of the impeller blade through burst fracture. This seriously affects the performance and life of the pump, resulting in additional energy consumption. Therefore, blade wear of two-blade pumps is one of the main reasons leading to the failure of self-priming pumps and increased energy consumption [9]. Figure 1 shows a picture of impeller wear [10].

![Figure 1. Physical picture of impeller wear](image.png)

At present, the traditional method to upgrade the wear resistance of portable parts is the assigned material method: coating the flow surface of the impeller with a layer of material with exceptional wear resistance (such as polyurethane) [11]. When the coating layer is worn, it needs to be re-packaged. This method has comparatively large defects, a high cost, and it is difficult to discern the replacement time [12]. Another method is to use a spray coating and choose wear-resistant materials, such as using nano-alumina ceramic-composite polymer materials for coating, construction, and curing at room temperature. The surface can reach a 0.1 \( \mu m \) roughness and by polishing the stainless steel surface smooth to over 10 times higher than normal [13], the coating has the effect of an anti-corrosion protection [14]. After spraying the coating, the roughness of the pump’s flowing parts can be reduced. However, the overall spraying of the flowing parts will lead to an increase in manufacturing cost; and the coating is also severely limited by temperature, poor strength performance, and other shortcomings. There are also surface texture methods, found in the 1940s by Gachot, et al. [15], for solving the problem of engines which are stuck under hot working conditions. High load, high temperature, and high speed conditions caused by the lack of lubricating oil and serious wear, ultimately lead to shock and failure. To solve this problem, cross-hatch slots have been fabricated on the piston bushings to provide a reservoir of lubricant in the field to avoid sticking. However, this can cause the opening groove to be continuously worn after a short time of use and is it is a high cost. The compound lubrication method, the application of the new multi-layer Ti3C2TX-Carrier (MXenes) as a solid lubricants method [16], uses Ti3C2T nanoparticles which have excellent tribological properties, and reduce the friction coefficient substantially by about 300%; and can also significantly diminish adhesion wear, abrasive wear, and tropical wear. However, the maintenance needs of this method require a continuous supplement of lubricant, the consumption of raw materials, and last for only a short duration resulting in high costs and energy wastage.

In recent years, bionic tribology [17,18] research and practice have demonstrated [19] that bionic surfaces can gain wear resistance, and non-smooth surfaces have better wear resistance and will decrease wear [20,21]. For example, studies have uncovered that shark
skin can decrease the surface resistance along the flow direction because grooves exist on the skin to essentially regulate the flow boundary layer [22]. The body surface structure of many creatures is special, presenting different types of non-smooth surfaces, which provide new ideas for the study of anti-wear, drag-reduction, and other technologies [23–25]. The mollusk shell is a typical natural biomineralization material with excellent wear resistance [26]. In particular, blood clams have no obvious surface wear under the impact of sand and stone in the water flow. This is due to a geometric network structure of longitudinal and transverse grooves, which makes them resistant to erosion and wear. At the same time, the blood clam’s bionic structure tool can effectively improve the heat dissipation efficiency of the friction interface. Therefore, on this basis, the marine anti-wear organism, the blood clam, was used as a prototype to extract the characteristics of its circular non-smooth surface structure. These characteristics were applied to the most easily worn area inside the two-vane pump, through the numerical simulation method, under different concentrations of particles, and the analysis of the bionic non-smooth structure affecting the performance of a centrifugal pump impeller anti-wear, revealed the bionic non-smooth anti-wear mechanism of the circular structure. The analyses were performed under the conditions of the same concentration of particles and the changing structure of the non-smooth diameter; and simulation was carried out on lift and efficiency of the two-blade pump. This study provides theoretical guidance for improving the anti-wear performance of two-blade pumps.

2. Numerical Model Establishment and Numerical Model Calculation

2.1. Pump Characteristic Parameters

The 80ZW40-30 two-vane self-priming pump (Wenzhou Self-priming Pump Industry, Wenzhou, China) was selected as the model pump in the research process, as shown in Figure 2. Its main parameters were: rated flow \( Q_e = 40 \text{ m}^3/\text{h} \), rated head \( H_e = 30 \text{ m} \), speed \( n = 2800 \text{ r/min} \), and power \( p = 4 \text{ kW} \). The impeller inlet inner diameter was \( D_1 = 80 \text{ mm} \) and the impeller inlet outer diameter was \( D_2 = 164 \text{ mm} \).

![Figure 2. 80ZW40-30 two-vane self-priming pump.](image)

2.2. Wear Model

In the process of conveying the solid-liquid two-phase flow medium, the solid particles will continuously impact the surface of the flow parts inside the two-blade pump, leading to serious wear loss. Therefore, it is necessary to choose an appropriate erosion–wear model to study the solid-liquid two-phase flow of the two-blade pump when calculating and analyzing the erosion–wear characteristics of the flow-through components.

The Finne erosion model is about the micro-cutting erosion of metal materials, which mainly considers the avoidance of materials. As with the environment of the two-blade pump, the impact velocity, the angle of the interaction between particles, and the wall in
the flow field, are three parameters, without considering other parameters and factors. The expression of this is shown in Equations (1) and (2) [27]:

$$ ER = kv_p^n f(\gamma), $$ (1)

$$ f(\gamma) = \begin{cases} 
\frac{1}{2} \cos^2 \gamma & \text{if } \tan \gamma > \frac{1}{2} \\
\sin(2\gamma) - 3\sin^2 \gamma & \text{if } \tan \gamma \leq \frac{1}{2}
\end{cases} $$ (2)

where, $k$ is a constant related to the material; $ER$ is wear rate, kg/s²·m; $v_p$ is solid phase velocity, m/s; $\gamma$ is particle impact angle; $f(\gamma)$ is the particle impact angle function; and $n$ is the parameter determined according to the wall base material, with the value of 2.3~2.5.

2.3. Biological Characteristics Analysis

In the long evolutionary process, to satisfy their survival and development needs, indigenous organisms have progressively evolved a variety of biological postures to adapt to the climate. These structural forms have formed the maximal adaptability and coordination ability within the living environment through optimization and coupling. Human beings can use their observation, thinking, and design ability to imitate biological systems or artificial technology to optimize the functions of the biological itself, so that it can be used to upgrade the flow field of centrifugal pumps and improve the abrasion resistance of the wall [28].

The main habitats of the blood clam are largely in the inner bay, where fresh water flows in. They are primarily distributed in the sea with a water depth of 5–7 m, though a water depth of more than 14 m is uncommon, where the impact of shifting sand is powerful. Nonetheless, the surface of the blood clam is not glossy. The reason for this is shown in Figure 3. Figure 3 demonstrates the surface characteristics of the blood clam [29]. In order to resist the erosion of silt in its long-term evolution, the radial ribs on the shell surface are prominent and dense, with 30–34 radial ribs. These ribs not only enhance the strength of the shell, but also improve its resistance to erosion and wear. In addition, the shell usually has a complex surface topography and has good wear resistance. Due to its remarkable mechanical strength and outstanding toughness, the shell case has long been considered the best organic armor. For example, some shells are well preserved after a long time of erosion, and the wear resistance of some shells is even analogous to that of a diamond-like carbon coating [30].

![Figure 3. Surface characteristics of the blood clam [29]. “l” is the distance between the peak curvature point (PCP) and the sampling center.](image)

The main structural morphology of the surface of the blood clam is a grooved feature. Noting the groove-type characteristic structure that conforms to the streamline-type characteristic structure, a circular structure can be constructed. At the same time considering that the suction side of the centrifugal pump impeller is the most prone to wear and tear, in the successive process of numerical simulation, it was concluded that the blood clam surface structure optimization for the structure of a circular non-smooth surface, could
be added to the suction side of the impeller where it is the most prone to wear. Figure 4 shows a schematic diagram for the bionic non-smooth surface map, where \( m \) is the groove depth and \( h \) is the distance of groove. Considering the actual size of the blood clam, its processing difficulty, the influence of the flow field, and the blade structure, the value of \( h = 3 \) mm was used. In two-vane pump suction side of the blade wear the most serious layout circular structure, bionic non-smooth surface with different groove size \( m \) round of non-smooth surface structure affect the performance of the two-vane pump abrasion, in circular non-smooth surface structure, groove section semicircle shape, the size of the groove \( m \) depends on the size of the circular diameter \( d \), combined with the structure size of the biomimetic prototype, the diameter of the circular non-smooth surface structure \( d = 0.15 \) mm, \( d = 0.20 \) mm, \( d = 0.25 \) mm, \( d = 0.30 \) mm were set and compared with the smooth blade \( d = 0 \) mm. The fluid domain of a circular non-smooth surface structure with four diameters are shown in Figure 5.

![Figure 4. Schematic diagram of bionic non-smooth surface, “m” is the groove depth and “h” is the distance of groove.](image)

![Figure 5. Impeller fluid field of a two-vane pump with circular non-smooth blades of different diameters: (a) \( d = 0.15 \) mm; (b) \( d = 0.20 \) mm; (c) \( d = 0.25 \) mm; and (d) \( d = 0.30 \) mm.](image)

2.4. Grid Division

An unstructured grid is an effective way to deal with complex calculation domains. The unstructured grid has very flexible adaptability in exhibiting the irregular, no-fixed structure characteristics of the irregular areas [31]. So, it is a division of the volute-tongue blade-insulation terminal, the non-smooth surface structure in areas such as the reason for the computational domain by unstructured grid partitioning. At the same time, to precisely capture the flow state of the blade boundary layer, the grid at the blade wall surface was encrypted. In addition, considering that the accuracy of the numerical simulation and the quantity and quality of grids have a close correlation, the more the number of grids then the more the requirement for high-performance computer configuration; and a smaller number...
of the grids would directly alter the precision of the numerical simulation. In order to take into account the speed of numerical simulation and calculation accuracy, grid correlation analysis was carried out on the model. Finally, so that the calculation error of the head was minor and within the allowable error range, the impeller grid number was given at 1.278 million and the volute grid number was given at 1.174 million.

2.5. Parameter Settings

Because of the special working environment of the two-vane pump, the slurry medium set in the two-vane pump was sand flow, and the solid particles in the sand flow were predominantly SiO$_2$, with a density of 2650 kg/m$^3$ and a particle size of 0.5 mm. The solid-liquid two-phase flow media with particle mass concentrations of $\rho_p = 0$ kg/m$^3$, 10 kg/m$^3$, 30 kg/m$^3$, 50 kg/m$^3$, 70 kg/m$^3$, and 90 kg/m$^3$ were set to simulate different research conditions. The corresponding volume fractions of gravel particles in the conveying medium under varied working conditions were: $\varphi_p = 0, 0.377\%, 1.132\%, 1.887\%, 2.642\%$, and $3.396\%$.

In general, the SIMPLE algorithm was commonly utilized, but due to the complex nature of the fluid, the SIMPLEC pressure-velocity coupling algorithm with good convergence was used [32]: the boundary conditions of the inlet and outlet were velocity inlet and free outflow, respectively. The convergence accuracy was $10^{-4}$. The impeller speed of 2800 r/min was set to match the selected rated speed of the two-vane pump. The inlet of the discrete phase was the inlet of the water inlet, and the outlet of the water outlet was the only outlet of the discrete phase. The other walls were set as reflective walls, and the contact between the particles in the slurry and the wall was consistent with the actual situation. When the discrete phase contacts the wall, it will reflect, and the reflection law conforms to the reflection coefficient of the discrete phase. The normal function is expressed in Equation (3), and the tangential function is expressed in Equation (4) [33].

$$y_n = 0.993 - 0.0307\theta + 4.75 \times 10^{-4}\theta^2 - 2.61 \times 10^{-6}\theta^3, \quad (3)$$

$$y_t = 0.988 - 0.029\theta + 6.43 \times 10^{-4}\theta^2 - 3.56 \times 10^{-6}\theta^3, \quad (4)$$

where, $y_n$ is the normal discrete phase reflection coefficient, dimensionless number; $y_t$ is the tangent discrete phase reflection coefficient, dimensionless number; and $\theta$ is the particle incidence angle, deg.

The standard $k-\varepsilon$ model is the most widely used model and does not need much calculation and memory. Thus, because of the simple internal structure of the two-vane pump, and that only the internal flow field was to be considered, the standard $k-\varepsilon$ model was used as the turbulence model for a comparatively optimal solution [18]. The discrete particle model, which can track the movement information of each particle, and consider particle–particle, particle–wall collision, and particle–fluid interaction, was adopted in the solid–liquid two-phase flow model. The erosion model is as follows [34]:

$$WR = 2.17 \times 10^{-7} \times F_p r_p^2 \cdot 41 F(\theta), \quad (5)$$

$$ER = \sum \frac{m_p WR}{t A_{cell}}, \quad (6)$$

where, $WR$ is wear weight loss; $F_p$ is particle shape coefficient, $F_p = 1$ for spherical particles; $m_p$ is single solid particle mass, kg; $t$ is total impact time, s; $A_{cell}$ is the grid area of cell wall is calculated, m$^2$; and $F(\theta)$ is impact Angle function, as shown in Table 1.

Three parameters, such as the incident velocity of solid particles, the number of particle collisions, and the angle, can alter the rate of wall erosion. Although there are as many as 100 variables affecting the erosion wear rate, the erosion model selected in this study can obtain high-reliability prediction results within the calculation range of the discrete phase [17].
Table 1. Impact angle function [23].

| Point | $\theta$ (deg) | Value |
|-------|----------------|-------|
| 1     | 0              | 0     |
| 2     | 20             | 0.8   |
| 3     | 30             | 1     |
| 4     | 45             | 0.5   |
| 5     | 90             | 0.4   |

3. Influence Analysis of the Circular Non-Smooth Structure on the Rotating Field Characteristics of Blades

3.1. Analysis of the External Characteristics of the Two-Vane Pump

Figure 6 shows the external characteristic curves of non-smooth two-vane pumps with varied circular diameters. Figure 6a illustrates that with the increase in solid particle concentration, the head of the two-vane pumps with five different blades declined. When the particle concentration, $\rho_p$, was from 0 to 10 kg/m$^3$, the head decreased the most: within 2.2%. This is because the addition of solid particles causes the collision between particles and particles and between particles and the wall surface of the flow component, to consume the total energy in the flow field. Under the conditions of the same particle concentration, the larger the diameter, $d$, then the smaller the head of the two-vane pump. With diameters of $d = 0.15$ mm and $d = 0.20$ mm, the head of the two-vane pump with circular non-smooth blades was close to that of smooth blades, and the decrease was within 1.4%. When the diameters were $d = 0.25$ mm and $d = 0.30$ mm, the head of the two-vane pump decreased greatly. In particular, when $d = 0.30$ mm of the round non-smooth blade, the pump head of two-vanes decreased by 1.3 m, indicating that the smoother the blade surface is, the more it conforms to the fluid flow law. The existence of the circular non-smooth structure can make the water flowing through the concave easily form a low-speed backflow zone. When the water rotates, it will carry sand and gravel to impact the wall of the circular non-smooth structure, absorbing the energy of the fluid medium and resulting in a head drop. The larger the diameter of the circular non-smooth blade surface, the more prone it is to induce the loss of flowing energy.

Figure 6. The external characteristics curve of the two-vane pump with non-smooth blades of different circular diameters: (a) Particle concentration-head curve; and (b) Particle concentration-efficiency curve.
According to the efficiency curve in Figure 6b, when particle concentration $\rho_p = 0$ kg/m$^3$, the efficiency of the five two-vane pumps reaches its maximum. With the increase in particle concentration, the efficiency of the two-vane pumps decreases. Under the same particle concentration conditions, the efficiency of the two-vane pumps also decreases with the increase in the diameter of the circular non-smooth structure. The efficiency of the two-vane pumps with a circular non-smooth blade where $d = 0.15$ mm and $d = 0.20$ mm, are relatively close, while the efficiency of the two-vane pumps with a circular non-smooth blade where $d = 0.30$ mm is at the minimum. Compared with smooth blades, the maximum decrease is about 2%. This is due to the existence of a circular non-smooth structure on the surface of the blade, the phenomenon of fluid momentum consumption such as rotation when flowing through the fluid, particle collision, and so on, which affects the transformation of the pressure energy to the speed energy of the two-vane pump, resulting in the reduction of head and efficiency. When the diameter of the circular non-smooth structure is small, the shape of the blade surface can better fit the fluid flow curve without excessive bending the angle, and the efficiency and head of the blade are close to that of a smooth blade. When the diameter of the circular non-smooth blades increases, the blade surface form, the internal form low backflow area, and the blade mainstream interference of low-speed reflux area is less; and the low-speed recirculation region of the internal toroidal vortex will form a large area of dead zones. The larger toroidal vortex will consume more energy, which in turn will affect the lift and efficiency. At the same time, the impact of solid particles and a collision will cause energy loss, which will exist within the circle of the non-smooth structure with the low backflow zone, known as the “air cushion” effect which occurs when the diameter is small and the vortex is formed by the small effect of solid-phased particle trajectory. With the larger diameter, the solid-phase particles can be directly channeled into the internal circular non-smooth structure, which is equivalent to increasing the contact area between the solid particles and blades.

There are two main reasons for the close efficiency of the two-vane pump with circular non-smooth blades $d = 0.15$ mm and $d = 0.20$ mm: the smaller the diameter, the smaller will be the vortex formed by the fluid, resulting in less energy loss. However, the larger the vortex, the greater the reduction in the frequency of solid particles hitting the blade surface, to some extent. Therefore, under the comprehensive action of these two reasons, the efficiency of the two dual-blade pumps with different diameters is relatively close.

To sum up, when the diameter of the circular non-smooth structure is set reasonably, such as when $d = 0.2$ mm, the gripping control of the mainstream field of the impeller on the liquid flow in the bionic pit layer becomes weaker, and will not make the solid phase rotating with the water flow on the impeller friction and wear. Therefore, the efficiency and the head of the non-smooth two-vane pump with a smaller diameter are closer to that of the smooth blade two-vane pump.

### 3.2. Analysis of Pressure Distribution of Two-Vane Pump

Figure 7 is the cloud diagram of pressure distribution at a cross-section of the two-vane pump with non-smooth blades of different circular diameters when particle concentration $\rho_p = 0$ kg/m$^3$. Figure 7 shows the two-vane pump overall uniform distribution, including that: the blade inlet side has less pressure in the center area, due to the blade rotation of fluid power; from the blade inlet to the outlet ends, pressure rises by gradient; the outlet pressure of the spiral case is the largest; and the role of the centrifugal pump is to pressure fluid. Thus, the simulation results conform to the requirements of the performance of the centrifugal pump. In Figure 7a,b, the low-pressure area in the central area of the blade is slightly larger than that in Figure 7c,d, indicating that with the increase in the diameter of the circular non-smooth surface structure, the low-pressure area at the inlet end of the blade will decrease. Meanwhile, it can be observed that the pressure gradient from the inlet end to the outlet end in Figure 7a,b is more uniform, while the pressure gradient fluctuation in Figure 7c,d is relatively large, which is also related to the non-smooth surface structure changing the flow-field boundary layer distribution inside the impeller. The increase in
the diameter of the circular non-smooth surface structure has a greater influence on the change in the distribution of the internal flow field. When the diameter reaches a certain value, the turbulent kinetic energy in the flow field will increase, forming a rotating water mass, consuming fluid energy, and affecting the pressure gradient distribution. All four figures show that the different internal pressure of the two-vane pump impeller blade distribution is relatively uniform and symmetrical, that the volute pressure distribution is roughly similar, only close to the low-pressure area around the diffuser, and that every tongue is different. Figure 7c, d in the low-pressure area was significantly greater than in Figure 7a, b in the This is due to that near the tongue, it is inevitable that the special structure of the tongue and the fluid at the outlet end of the rotating blade produce strong rotor-stator interference.

Figure 7. Cross section pressure distribution of the two-vane pump with circular non-smooth blades of different diameters: (a) \( d = 0.15 \) mm; (b) \( d = 0.20 \) mm; (c) \( d = 0.25 \) mm; and (d) \( d = 0.30 \) mm when \( \rho_p = 0 \) kg/m\(^3\).

4. Influence Analysis of Circular Non-Smooth Structure on Wear Characteristics of Impeller

Figure 8 shows the curves of the average wear rate and particle concentration of non-smooth blades with different circular diameters. It shows that with the increase in particle concentration, the mean wear rate of circular non-smooth blades increases linearly with a proportional trend. Figure 9 shows the average wear rate diameter histogram of circular non-smooth blades with different particle concentrations. It can be intuitively observed from the bar chart that under different particle concentrations, when \( d = 0.20 \) mm, the average wear rate of circular non-smooth blades is at the lowest point, which is the same as the trend of head and efficiency. With the increase or decrease in the diameter \( d \), the average wear rate of blades increases. At the same time, it can be seen from Figure 8 that when \( d = 0.25 \) mm and \( d = 0.30 \) mm, the average wear rate is significantly higher than the other three blades, which is related to the particle size of the sand. The particle size of the sand is 0.5 mm, which can enter directly into the groove of a non-smooth bionic structure and contact and collide with the groove wall. When the diameter of the circular non-smooth surface structure is too large, fluid can form a low-speed backflow area. The water inside the groove can not only make the head and the efficiency loss more serious,
but at the same time carry sand that will enter inside groove. This is due to the existence of the vortex in the centrifugal force being bigger and the physical parameters of sand and water being different, which means that the sand that rotates with the water flow is more likely to be “thrown” out of the vortex area. As the centrifugal force constantly works on the sand particles, the impact force of the sand particles on the groove wall is greater than that on the blade surface directly, thus causing a more severe damage and impact on the groove wall.

![Figure 8. Average wear rate-particle concentration curve of non-smooth blades with different circular diameters.](image)

![Figure 9. Average wear rate-diameter bar chart of circular non-smooth blades with different particle concentrations.](image)

Figure 10 is the cloud diagram of the wall wear rate distribution of non-smooth blades with different circular diameters. Figure 10 shows that with the increase in particle concentration, the severely worn area is mainly concentrated on the water inlet end and the suction surface of the blade near the front cover and the wear areas on the blade also show asymmetry. When the particle concentration $\rho_p = 10 \text{ kg/m}^3$, random pitting on the blade suction surface appears. When $\rho_p > 10 \text{ kg/m}^3$, the pitting area presents a changing trend of point-block-strip, which shows that when the particle concentration reaches a certain extent, under the actual working condition of the two-vane pump, at the suction side of
the blade the appearance of a wear area of radial uneven bars will change the hydraulic parameters and affect pump performance. This is serious and can be the cause of blade fracture failure which in turn will cause an accident. When particle concentration is the same, combined with the comprehensive comparison in Figure 9, the results show that only when the diameter of the circular non-smooth surface structure reaches a certain level, the anti-wear performance of the blade can be improved. When particle concentration is $\rho_p = 90 \text{ kg/m}^3$, the increase in diameter means that the wear area on the suction surface moves downward from near the front cover. The larger that the wear area is, the more serious the wear degree.

To sum up, we can find that the wear degree of the three two-vane pumps with a non-smooth structure is very different from that of those with smooth blades, under the same particle concentration. Figure 8 shows the average wear rate–particle concentration curve of non-smooth blades with different circular diameters. When the particle concentration is $90 \text{ kg/m}^3$ and the diameter is 0.2 mm, the wear degree of the blade with the round, non-smooth structure is small. This is because the existence of the circular non-smooth surface structure leading to the sand will be “caught” by the vortex area, and it is not easy to follow the water flow to impact the wall, so the particles will stay away from the blade. However, when the diameter of the circular non-smooth blade increases, larger pits will be formed on the surface of the blade. Although a low-speed backflow area will be formed inside the non-smooth circular structure, when the diameter of the non-smooth circular structure is large, the solid particles can directly enter the inner part of the circular non-smooth structure, which is equivalent to increasing the contact area between the solid particles and the blade.

Figure 10. Cont.
Figure 10. Cont.
To sum up, we can find that the wear degree of the three two-vane pumps with a non-smooth structure is very different from that of those with smooth blades, under the same particle concentration. Figure 8 shows the average wear rate–particle concentration curve of non-smooth blades with different circular diameters. When the particle concentration is 90 kg/m$^3$ and the diameter is 0.2 mm, the wear degree of the blade with the round, non-smooth structure is small. This is because the existence of the circular non-smooth surface structure is leading to the sand will be “caught” by the vortex area, and it is not easy to follow the water flow to impact the wall, so the particles will stay away from the blade. However, when the diameter of the circular non-smooth blade increases, larger pits will be formed on the surface of the blade. Although a low-speed backflow area will be formed inside the non-smooth circular structure, when the diameter of the non-smooth circular structure is large, the solid particles can directly enter the inner part of the circular non-smooth structure, which is equivalent to increasing the contact area between the solid particles and the blade.

5. Conclusions

The circular characteristic structure of the blood clam surface was extracted, and the circular non-smooth structure was applied to the most easily worn part of the suction surface of the two-vane pump impeller, by using its anti-wear property in sand and stone, and the circular non-smooth surface blade structure model was established. The circular non-smooth structure and blade arrangement has stronger wear resistance, compared with the impeller blade of the standard pump. In this paper, the internal erosion–wear characteristics of two-vane pumps with different circulating structures and non-smooth structures under different working conditions were studied, and the external characteristics, flow field characteristics, and impeller wear characteristics of two-vane pumps were analyzed. The results show that the circular non-smooth structure had little effect on the hydraulic performance of the two-vane pump and changing the blade surface structure did not affect its hydraulic performance. With the increase in particle concentration, the wear degree of circular non-smooth blades with a diameter of 0.20 mm was less than that of the standard two-vane pump with a diameter of 0 mm, while the wear degree of circular non-smooth blades with the other three diameters was greater than 0 mm. With the increase in the diameter of the circular non-smooth surface structure, the wear area also increased. This is because the fluid in the circular non-smooth structure formed a low-speed reflux zone. This study shows that with a larger diameter there was a stronger continuous centrifugal force on the sand and stone; therefore, the stronger was the damage to, and the impact on, the pump wall. Under six particle concentrations, the head and efficiency of the two-vane pump with a circular non-smooth structure of 0.20 mm in diameter were better. In other words, the circular non-smooth surface structure can result in effective wear resistance. Therefore, it can be concluded that this structure can also be used in places other than in two-vane centrifugal pumps.
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