Influence of bionic structure on hydraulic performance and drag reduction effect of a centrifugal pump

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Abstract. Based on bionic theory, miniature dimple-type structures are constructed on the blades of a single-stage centrifugal pump to investigate the influence of the bionic structures on the hydraulic performance and drag reduction effect of the pump. The results show that dimples with shallower depth and which are located near the front section of the blade suction surfaces are more effective in improving the efficiency and reducing the drag of the pump. At small flow rate, due to the mismatch between the incoming flow angle and the blade placement angle, serious flow separation occurs in several flow channels, accompanied by the formation of the vortices, which would deteriorate the performance of the pump. With the arrangement of the dimples on blade suction surfaces, some low-speed vortices formed in the dimples, which is equivalent to increasing the effective thickness of the viscous substrate layer and decreasing the velocity gradient of the boundary layer. Therefore, the velocity distribution in the flow channels gets more uniform, and the turbulence kinetic energy and wall shear stress are thereby reduced.

1. Introduction

Energy conservation and environmental preservation have progressively become major topics of worldwide concern as a result of rising energy consumption, and low-carbon energy development has emerged as a key pathway[1]. According to statistics, the annual power consumption of pumps accounts for about 17% of China's total electricity consumption[2], and surface friction drag is one of the important sources of energy dissipation. For large-scale industrial equipment or systems that operate constantly, reducing the drag of fluid movement inside pumps not only saves energy but also reduces operation and maintenance expenses significantly. Therefore, it is crucial to conduct research on pump energy conservation and drag reduction.

The bionic non-smooth surface drag reduction method is a technique for reducing drag by mimicking biological epidermal structures[3], which is originally derived from observation of the epidermis of marine species. The remarkable swimming skills of large marine organisms like sharks have long attracted the interest of scientists. Researchers have found that shark skin is covered with non-smooth shield scale structures with microscopic grooves aligned in the direction parallel to the direction of fluid flow[4]. These grooves can alter the structure of the boundary layer near the shark's body surface, delaying the formation of turbulent eddies and thereby reducing swimming drag. In the 1980s, Walsh et al.[5-6] conducted the first experimental study on the drag reduction performance of groove structure
on shark skin. The results confirmed that the groove structure can significantly reduce the wall friction drag, and the drag reduction effect is closely related to the shape of the grooves. It is found that the symmetric V-shaped grooves can reduce friction drag by up to 8% at high-speed flow. This achievement provides insights for research in bionics. Various non-smooth structural features have since been extracted from the surfaces of other organisms, such as the existence of some elevated blocky nodules at the leading edge of the humpback whale's flipper[7]. When water flows through these nodules, vortex structures form and create perturbations in the flow field, thus effectively reducing the flow drag and noise. In addition, micro-dimple structures have been observed on the surface of fish such as the Chinese sturgeon, and studies have shown that dimple-type structures induce the turning of the laminar boundary layer to the turbulent boundary layer in advance, thereby suppressing flow separation in the channel[8].

In recent years, with the development of bionic theory, some progress has been made in the application of bionic microstructures in fluid machinery. Wang et al.[9] investigated the effect of triangular bionic microgrooves on the internal flow of a centrifugal pump. They discovered that the bionic microgrooves could reduce the energy loss caused by the impact of the flow and improve the hydraulic performance of the pump under various operating conditions. At the design flow rate, the head and efficiency of the pump increased by 3.7% and 0.8%, respectively. Dai et al.[10] constructed the V-groove structure on the blades of a centrifugal pump that resembled shark skin. It is found that the V-groove structure could improve the efficiency, the drag reduction rate, and the total sound pressure level noise reduction rate of the internal sound field at the rated operating condition, which further verifies the drag reduction and noise reduction function of the bionic structure. Zhao et al.[11] arranged the bionic nodular structure at the leading edge of the blades in a centrifugal pump. The results showed that the bionic model effectively suppressed the development of vacuoles, and the best inhibition performance was achieved at the cavitation incipient stage, with a 99.72% reduction in the average vacuole volume fraction. Qian et al.[12] studied the impact of the bionic convex domes on the erosion drag effect of a double suction pump. They found that the bionic blades exhibited much better erosion drag than the smooth blades. The high erosion-rate area was significantly reduced, and the erosion region became more spread across the entire bionic blade surface. Zhang et al.[13] optimized a vertical pipeline pump with reference to the sawtooth structure at the end of the owl feather. The results showed that the bionic sawtooth blade can significantly reduce the pressure pulsation and noise in the flow field, with the noise reduction effect being more noticeable under the design and high-flow conditions.

In summary, non-smooth surfaces have achieved some promising results in hydraulic machinery, however, the shapes of which usually adopt grooves, and the drag reduction effect of other-shaped surfaces need to be further studied. In the current paper, the bionic dimple-type structures are introduced on the blade suction surfaces. Multiple schemes are put up to examine the impact of the size and arrangement location of the dimple-type structures on the hydraulic performance and drag reduction effect of the pump, which would provide theoretical guidance for the optimization of the centrifugal pump.

2. Numerical calculation method

2.1. Calculation model

2.1.1. Original model. This paper takes a single-stage centrifugal pump as the research object, the design parameters of the model pump are as follows: flow rate $Q_n=500\text{m}^3/\text{h}$, head $H=48.2\text{m}$, rotating speed $n=1450$ rpm. The main geometrical parameters of the model pump are as follows: impeller inlet diameter $D_1=200\text{mm}$, impeller outlet diameter $D_2=420\text{mm}$, impeller outlet width $b_2=36\text{mm}$, blade outlet angle $\beta_2=20^\circ$, blade number $Z=6$, volute inlet diameter $D_3=450\text{mm}$, volute inlet width $b_1=72\text{mm}$. Figure 1 shows the assembly diagram of the model pump, the calculation domains mainly include the inlet and outlet pipes, impeller, and volute.
2.1.2. Bionic model. The blade flow direction is defined as U, and the blade spreading direction is defined as V. Four rows of dimple-type structures are arranged along the direction U in the form of matrix on the blade suction surfaces. Taking depth and the arrangement location of the dimples as design variables, 9 bionic schemes are put forward, which are denoted by SS_{ij} (1 ≤ i ≤ 3, 1 ≤ j ≤ 3), respectively. The first subscript i represents the location of the dimples, i=1, 2, and 3 indicate that the dimples are located at the leading section, middle section, and trailing section of the blades, respectively. The second subscript j represents the depth of the dimples, j=1, 2, and 3 indicate that the depth of the dimples are h=0.6mm, h=1mm, and h=1.5mm, respectively. Moreover, the depth-to-diameter ratio of the dimples is h/d=0.25[14], the distance between the dimples in direction U is s_U=1.7d, and the distance between the dimples in direction V is s_V=1.4d. Table 1 presents the detailed parameters of different bionic schemes. Figure 2 shows the schematic diagram of the dimple-type structures in the flow cross-section, and figure 3 exhibits the structural comparison of the original blade and the bionic blades.

<table>
<thead>
<tr>
<th>Scheme</th>
<th>Location</th>
<th>h /mm</th>
<th>d /mm</th>
<th>s_u /mm</th>
<th>s_v /mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>SS_{11}</td>
<td>Leading section</td>
<td>0.6</td>
<td>2.4</td>
<td>4.08</td>
<td>3.36</td>
</tr>
<tr>
<td>SS_{12}</td>
<td>Leading section</td>
<td>1</td>
<td>4</td>
<td>6.8</td>
<td>5.6</td>
</tr>
<tr>
<td>SS_{13}</td>
<td>Leading section</td>
<td>1.5</td>
<td>6</td>
<td>10.2</td>
<td>8.4</td>
</tr>
<tr>
<td>SS_{21}</td>
<td>Middle section</td>
<td>0.6</td>
<td>2.4</td>
<td>4.08</td>
<td>3.36</td>
</tr>
<tr>
<td>SS_{22}</td>
<td>Middle section</td>
<td>1</td>
<td>4</td>
<td>6.8</td>
<td>5.6</td>
</tr>
<tr>
<td>SS_{23}</td>
<td>Middle section</td>
<td>1.5</td>
<td>6</td>
<td>10.2</td>
<td>8.4</td>
</tr>
<tr>
<td>SS_{31}</td>
<td>Trailing section</td>
<td>0.6</td>
<td>2.4</td>
<td>4.08</td>
<td>3.36</td>
</tr>
<tr>
<td>SS_{32}</td>
<td>Trailing section</td>
<td>1</td>
<td>4</td>
<td>6.8</td>
<td>5.6</td>
</tr>
<tr>
<td>SS_{33}</td>
<td>Trailing section</td>
<td>1.5</td>
<td>6</td>
<td>10.2</td>
<td>8.4</td>
</tr>
</tbody>
</table>
2.2. Grid generation

Since the surface of the bionic blades is relatively complicated, the unstructured grid with strong adaptive is applied for the impeller domain and the structured grids with high accuracy are adopted for other domains by using the ICEM CFD software. The grid at the bionic structure is locally encrypted, and the mesh division of the impeller is shown in figure 4. Meanwhile, the grid independence of the model pump was checked to verify the accuracy of the simulation, and the results are displayed in table 2. When the number of meshes reaches 5.43 million, the head of the pump basically remains unchanged, hence the mesh number of $5.43 \times 10^6$ is chosen for the following calculations.
Figure 4. Grid of bionic impeller.

Table 2. Validation of mesh independence.

<table>
<thead>
<tr>
<th>Mesh scheme</th>
<th>Grid number/(×10⁶)</th>
<th>Simulated head/m</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.715</td>
<td>53.43</td>
</tr>
<tr>
<td>2</td>
<td>3.670</td>
<td>53.22</td>
</tr>
<tr>
<td>3</td>
<td>4.579</td>
<td>53.16</td>
</tr>
<tr>
<td>4</td>
<td>5.433</td>
<td>53.03</td>
</tr>
<tr>
<td>5</td>
<td>6.275</td>
<td>53.02</td>
</tr>
</tbody>
</table>

2.3. Turbulence model and boundary conditions
The internal flow in the impeller is characterized by large curvature, strong rotation, and high counterpressure gradient, therefore, the selection of the turbulence model is important to obtain ideal simulation results. Since the RNG k-ε model has good accuracy in simulating the boundary layer flow[15], flow separation, and secondary flow under strong counterpressure gradient, the current paper adopted the RNG k-ε model and used ANSYS CFX software to carry out the steady numerical simulation of different models. The multi-coordinate reference system was employed, with the impeller domain designated as the rotating domain, and the other domains designated as stationary domains. The dynamic and static component interfaces were set as frozen rotor interfaces, and all physical surfaces of the pump were designated as no-slip walls. Furthermore, the inlet boundary condition was set as total pressure, the outlet boundary condition was set as mass flow rate, and the convergence residual RMS was specified as 10⁻⁴.

3. Results and discussions

3.1. Performance characteristics curves
The comparison between the test and simulated values of the external characteristic parameters is shown in figure 5. It is demonstrated that the simulated head and efficiency of the model pump are basically consistent with the test values. At the design point, the calculation errors for head and efficiency are 0.28% and 2.1%, respectively. The maximum calculation error is 4.2% when the flow rate decreases to 0.4Qn, the deviations are within the acceptable range, so further analysis would be made based on the numerical simulation method.
3.2. Analysis of flow drag reduction effect of the bionic structure
The flow in the pump is usually subject to various drags during operation, among which fluid drag and mechanical drag are the main factors. In order to overcome the drag to rotate the impeller, sufficient torque must be exerted on the pump shaft. The more drag there is, the more torque is necessary to turn the impeller. As a result, the drag reduction effect of the bionic model can be measured using the torque of the impeller. Gu et al.[16] provided the formula for the drag reduction rate.

\[
C = \frac{N_s - N_r}{N_s} \times 100\%
\]  

(1)

Where \(N_s\) is the torque of the smooth-surface impeller and \(N_r\) is the torque of the bionic-surface impeller. A positive C indicates that the dimple-type structures could reduce drag, and vice versa. The greater absolute value of C indicates that the dimples are more effective in reducing or increasing drag.

Fluid drag, such as friction drag and differential pressure drag during the operation of the pump, is primarily determined by the properties of the fluid and the flow conditions[17]. Among these, the friction drag has a significant impact on the energy consumption of the pump and is the object to be optimized in the structure design of the pump. Since the drag reduction rate C can only give quantitative analysis of the drag in the pump, it is not possible to obtain the specific location where the drag appears. Therefore, a new parameter needs to be introduced to further analyze the local drag in the pump.

According to Newton's law of internal friction, the friction drag is directly proportional to the wall shear stress. Therefore, the magnitude of the wall shear stress can be applied to measure the drag reduction effect of the bionic model. The reduction rate D of wall shear stress is defined by equation (2):

\[
D = \frac{W_s - W_r}{W_s} \times 100\%
\]  

(2)

Where \(W_s\) is the average wall shear stress of the smooth-surface impeller and \(W_r\) is the average wall shear stress of the bionic-surface impeller.

3.2.1. Effect of depth of the dimples on the performance of the centrifugal pump. Table 3 compares the external characteristic parameters and drag reduction rate of the original model and different bionic models under rated working condition. It can be seen that the head of all bionic models is elevated compared to the original model, while the efficiency of the bionic model is related to the depth of the dimples. The efficiency of scheme SSII all increased relative to the original model, in general, the
hydraulic performance of scheme SS1 is better than other bionic schemes, and scheme SS11 increases the head and efficiency of the original pump by about 1.1% and 0.6%, respectively. The hydraulic performance of the model pump gradually decreases as the depth of dimples increases, therefore, a shallower depth of the dimple is recommended to improve the performance of the centrifugal pump. In addition, the drag reduction rates of all bionic schemes are negative, indicating that the torque of the bionic impeller under the design condition is increased with respect to the original model, but the increment is quite small, with a maximum drag increment of 0.7846%.

<table>
<thead>
<tr>
<th>Scheme</th>
<th>η/%</th>
<th>H/m</th>
<th>C/%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Origin</td>
<td>80.50</td>
<td>53.04</td>
<td>-</td>
</tr>
<tr>
<td>SS11</td>
<td>80.97</td>
<td>53.60</td>
<td>-0.6729</td>
</tr>
<tr>
<td>SS12</td>
<td>80.69</td>
<td>53.34</td>
<td>-0.4411</td>
</tr>
<tr>
<td>SS13</td>
<td>80.44</td>
<td>53.12</td>
<td>-0.3137</td>
</tr>
<tr>
<td>SS21</td>
<td>80.87</td>
<td>53.32</td>
<td>-0.1522</td>
</tr>
<tr>
<td>SS22</td>
<td>80.52</td>
<td>53.06</td>
<td>-0.09849</td>
</tr>
<tr>
<td>SS23</td>
<td>80.38</td>
<td>53.12</td>
<td>-0.3831</td>
</tr>
<tr>
<td>SS31</td>
<td>80.61</td>
<td>53.30</td>
<td>-0.4610</td>
</tr>
<tr>
<td>SS32</td>
<td>80.45</td>
<td>53.22</td>
<td>-0.4374</td>
</tr>
<tr>
<td>SS33</td>
<td>80.26</td>
<td>53.27</td>
<td>-0.7846</td>
</tr>
</tbody>
</table>

3.2.2. **Effect of location of the dimples on the performance of the centrifugal pump.** According to the previous section, a smaller depth of the dimple is beneficial to improve the performance of the model pump. Therefore, schemes SS11, SS21, and SS31 with h=0.6mm are selected to further examine the effect of the location of dimples on the performance of the centrifugal pump.

Figure 6 shows the hydraulic performance comparison between the original model and three bionic models. It can be seen from figure 6(a) that the efficiency of all bionic models is lower than that of the original pump at 0.4 Qn, among which scheme SS11 has the smallest reduction amplitude. With the position of the dimples away from the front section of the blade, the efficiency of the bionic pump gradually decreases, and the efficiency of scheme SS31 dropped by 5.8% compared with the original model. When the flow rate is greater than 0.4 Qn, the efficiency of all bionic models exceeds that of the original model. In general, scheme SS11 has a better effect on improving the efficiency, and the efficiency is increased by up to 1.7% at 0.6 Qn. According to figure 6(b), the heads of schemes SS31 and SS21 differ very little from that of the original model at various flow rates. Scheme SS11, on the other hand, exhibits a different pattern. When the flow rate is less than 0.6 Qn, the head of scheme SS11 is lower than that of the original model. As the flow rate approaches the design flow rate, the head of Scheme SS11 gradually increases and reaches the maximum value at the design point.

Figure 7(a) depicts the torque of the impeller of different schemes, where the vertical axis denotes the absolute value of torque. It is demonstrated that the torque of the four schemes has similar changing trends, that is, the torque increases gradually as the flow rate increases. At 0.4 Qn, the torque of scheme SS11 is slightly lower than the original model, while the torque of the other two schemes is higher than the original model. Table 4 shows the comparison of the drag reduction rates of three bionic impellers. It can be seen that the torque of schemes SS21 and SS31 raised by nearly 2.9% and 6.5% compared with the original model. It means that the arrangement of the dimple-type structures in the middle and trailing
section of the blades has a negative influence on the drag reduction of the pump at lower flow rates. With the increase in flow rate, the torque of the bionic models reduces in comparison to the original model, and the magnitude of the decrement reaches the largest at 0.6 $Q_n$. According to table 4, the torque of scheme SS$_{11}$ is reduced by 2.7% compared with the original model at this time. When the flow rate raises to the design point and above, the dimple-type structures show little effect on the torque of the model pump.

Figure 7(b) presents the comparison of the average wall shear stress on the impeller blade of different schemes. It is obvious that as the flow rate increases, the average wall shear stress on the blade exhibits a tendency of decreasing first and then increasing, and it reaches the minimum value at 0.6 $Q_n$. The wall shear stress of scheme SS$_{11}$ is lower than the original model at all working conditions. Table 5 displays the reduction rates of wall shear stress of three bionic impellers, it is evident that the arrangement of the dimples significantly reduces the average shear stress on the blade surface, of which the reduction of scheme SS$_{11}$ has the largest magnitude, and the wall shear stress of the impeller is reduced by 13.54% compared with that of the original model at the rated condition. Overall, scheme SS$_{11}$ outperforms the other two schemes, so the internal flow field of scheme SS$_{11}$ is further analyzed to investigate the drag reduction mechanism of the bionic structure.

![Figure 6](image.png)

(a) Comparison of efficiency of different schemes.

(b) Comparison of head of different schemes.

**Figure 6.** External characteristic parameters of different models under various working conditions.

**Table 4.** The drag reduction rates of three bionic impellers.

<table>
<thead>
<tr>
<th>$Q/Q_n$</th>
<th>SS$_{11}$</th>
<th>SS$_{21}$</th>
<th>SS$_{31}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.4</td>
<td>0.8164</td>
<td>-2.870</td>
<td>-6.484</td>
</tr>
<tr>
<td>0.6</td>
<td>2.744</td>
<td>1.093</td>
<td>0.5115</td>
</tr>
<tr>
<td>0.8</td>
<td>-0.7826</td>
<td>0.9302</td>
<td>0.2605</td>
</tr>
<tr>
<td>1.0</td>
<td>-0.6729</td>
<td>-0.1522</td>
<td>-0.4610</td>
</tr>
<tr>
<td>1.2</td>
<td>0.1039</td>
<td>0.5614</td>
<td>0.1331</td>
</tr>
</tbody>
</table>
(a) Comparison of torque of different schemes.  
(b) Comparison of wall shear stress of different schemes.

**Figure 7.** Torque and wall shear stress of different models under various working conditions.

**Table 5.** The reduction rates of wall shear stress of three bionic impellers.

<table>
<thead>
<tr>
<th>(Q/Q_n)</th>
<th>SS(_{11})</th>
<th>SS(_{21})</th>
<th>SS(_{31})</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.4</td>
<td>21.03</td>
<td>5.700</td>
<td>8.521</td>
</tr>
<tr>
<td>0.6</td>
<td>17.33</td>
<td>-4.268</td>
<td>0.6051</td>
</tr>
<tr>
<td>0.8</td>
<td>17.95</td>
<td>9.597</td>
<td>7.659</td>
</tr>
<tr>
<td>1.0</td>
<td>13.54</td>
<td>9.536</td>
<td>8.691</td>
</tr>
<tr>
<td>1.2</td>
<td>8.463</td>
<td>5.624</td>
<td>4.436</td>
</tr>
</tbody>
</table>

3.3. *The internal flow field of the original model and the bionic model*

To describe the typical flow structure in the model pump, the relative velocity is dimensionless by equation (3):

\[
W' = \frac{W}{u_2}
\]  

(3)

Figure 8 presents the distribution of velocity and streamlines at the mid-span of the two models. It shows that the original model and the bionic model have similar flow patterns under different operation conditions. At small flow rate, the velocity distribution inside the pump is extremely uneven, and a wide range of low-velocity regions appear in the flow channels close to the tongue, accompanied by flow separation, vortices, and other unstable flow structures. With the increase of the flow rate, the distribution of velocity and streamlines in the pump becomes more uniform, and the unsteady structures gradually disappear.

As can be seen from figure 8(d)–(f), the bionic structure performs admirably in improving the internal flow of the impeller. At 0.6 \(Q_n\), the bionic structure effectively controls the flow separation on the blade suction surfaces and reduces the number of vortices in the flow channels, thereby improving the
performance of the pump to a certain extent. At both rated and over-load operating conditions, the bionic structure stabilizes the streamlines inside the impeller, the vortex in the flow channel basically disappears, and the fluid moves steadily along the profile of the blade. In addition, the velocity distribution at the outlet of the impeller becomes more uniform, and the area of the high-speed zone is reduced. At 1.2 \( Q_n \), it is clear that the high-speed region at the outlet of the flow channel facing the tongue essentially vanishes.

![Figure 8. Relative velocity distributions and streamlines at the mid-span of the pump.](image)

Turbulence kinetic energy is a parameter that reflects the intensity of fluid turbulence, the flow field would grow more unstable as the turbulence kinetic energy increases. Figure 9 depicts the distribution of turbulence kinetic energy at the mid-span of the pump, it is clear that the turbulence kinetic energy is quite strong at the impeller exit and near the tongue. At 0.6 \( Q_n \), it can be observed in the streamline diagram that multiple large-scale vortices appear in the flow channel near the tongue at this time, and the vortices continuously fall off and hit the volute, thus causing strong turbulence kinetic energy in the corresponding region near the tongue. As evident in figure 9(d), the bionic structure improves the internal flow of the pump, and the turbulence kinetic energy at the impeller outlet close to the tongue is significantly reduced. In addition, the regions with moderate intensity of the turbulence kinetic energy can also be observed as marked by the red circle in figure 9(a), which is caused by the flow separation on the blade suction surface. The turbulence kinetic energy in this region is also substantially decreased for the bionic structure successfully suppresses the flow separation on the suction surface.
As the flow rate increases, the flow in the centrifugal pump grows more uniform, at the design flow rate, the internal flow of the pump is considered to be the most stable since the turbulence kinetic energy inside the pump reaches the lowest. It is glaring from figure 9(b) and (e) that the distribution of turbulence kinetic energy in the original model and the bionic model is similar, yet the bionic structure reduces the turbulence kinetic energy of the regions in the left-hand side and the diffusion section of the volute. When the flow rate raises to 1.2 \( Q_n \), the impacting effect of the fluid increases, and the high-speed fluid from the impeller exit has an intensive interaction with the tongue, causing significant turbulence to form close to the tongue. As seen in figure 8(f), the bionic structure improves the uniformity of the velocity in the flow field, and the local high-speed region at the impeller exit vanishes, which weakens the impact of the fluid on the tongue and reduces the turbulence kinetic energy near the tongue.

![Turbulence Kinetic Energy Distribution](image)

(a) Original model, 0.6\( Q_n \)  
(b) Original model, 1.0\( Q_n \)  
(c) Original model, 1.2\( Q_n \)

(d) SS\(_{11}\) model, 0.6\( Q_n \)  
(e) SS\(_{11}\) model, 1.0\( Q_n \)  
(f) SS\(_{11}\) model, 1.2\( Q_n \)

Figure 9. Turbulence kinetic energy distributions at the mid-span of the pump.
Figure 10 shows the distribution of wall shear stress of different models. It can be seen that the regions with higher wall shear stress are mainly concentrated on the leading section of blade suction surfaces. It could be explained by the fact that the axially moving fluid acquires a peripheral velocity at the inlet of the impeller and is thrown towards the blade suction surfaces under the action of centrifugal force, thus resulting in higher wall shear stress at the corresponding location. With increase in flow rate, the velocity of inflow raises, which increases the wall shear stress on the suction surfaces. Moreover, the distribution range of wall shear stress gradually spreads from the leading edge to the entire suction surface.

From figure 10(d)–(f), it is evident that bionic structures effectively reduce the wall shear stress on the whole blade surfaces. And as shown in table 5, the reduction rate of wall shear stress decreases with the increase in flow rate. This is because the velocity gradient on the blade surface is large and severe flow separation could occur under small flow conditions. The bionic structure, on the other hand, can inhibit the flow separation, thus apparently reducing the wall shear stress. At large flow rate, the flow in the impeller is relatively stable, and only slight flow separation occurs, so the dimple-type structure has less influence on it.

As shown in figure 11, two circumferential positions R1=76mm and R2=210mm are selected to analyze the relative velocity distribution at the inlet and outlet of the impeller.

Figure 12 shows the relative velocity distribution of two models on the circumferential position R1. The parameter θ, as shown in figure 11, is introduced to represent the angle of the point from the vertical axis. It can be seen that the relative velocity of the original model and the bionic model have similar trends at different flow rates. In general, the velocity distribution at the inlet of the impeller is relatively uniform at the design and large flow conditions. The relative velocity gradually increases from the blade pressure surface to the suction surface. It is evident that the relative velocity of the bionic model is slightly lower than that of the original model, as the velocity at the impeller inlet is reduced, the impact of the inflow on the wall is weakened accordingly. As a result, the wall shear stress at the inlet of the impeller is reduced. The relative velocity within different channels appeared to be noticeably distinct at 0.6 Q_n. Overall, the relative velocity in channel 1 and channel 6 is lower than that of other channels, and the sharp decline of relative velocity is generated at the suction side of channel 1 and the pressure side.
of channel 6. This is because the velocity of inflow is comparatively lower at small flow rate, and the incoming flow angle is smaller than the blade placement angle, resulting in apparent flow separation. From Figure 12(a), it is obvious that the bionic structure increases the relative velocity at the inlet of the flow channel near the tongue, and eliminates the steep drop in velocity of channel 6, thereby inhibiting the flow separation on the blade surface to some degree.

**Figure 11.** Definition of the circumferential parameters of the impeller.
Figure 12. The relative velocity distribution on the circumferential position R1.

According to Zhang et al.[19-20], the internal flow field of the volute is significantly influenced by the jet-wake structure at the impeller exit. The average velocity in the jet region is usually high, whereas the average velocity in the wake region is relatively low. As a result, the inflection point of the velocity can be regarded as the demarcation point of the jet wake structure.

Figure 13 presents the relative velocity distribution of two models on the circumferential position R2. At both rated and over-load operating conditions, the variation of the relative velocity at the impeller exit of different flow channels is relatively similar. From the blade pressure surface to the suction surface, the relative velocity follows an ascending and subsequently descending trend. Obviously, the relative velocity of the flow channel 1 is comparatively large, especially at 1.2 $Q_n$. As can be seen from figure 11, channel 1 faces the tongue, where the cross-sectional area of the volute is the smallest. Therefore, the convergence area of the jet-wake structure at the channel exit is small, resulting in higher velocity here. Strong energy loss would be generated under the interaction between the high-velocity fluid and the tongue. From Figure 13(b) and (c), it is evident that the bionic structure effectively reduces the velocity in the jet region of channel 1, and makes the velocity distribution in each channel more uniform, thereby weakening the interaction between the jet-wake structure and the tongue.

When the flow rate decreased to 0.6 $Q_n$, the relative velocity in the original model begins to show an irregular trend. The sharp decline of relative velocity is generated at the pressure surface of channel 1, channel 5, and channel 6, indicating that unstable flow structures may occur at these regions, which is consistent with the distribution of streamlines as shown in figure 8. It can be seen from figure 13(a) that the bionic structures effectively suppress the low-velocity region near the pressure surface of the aforementioned flow channels. In addition, it is worth noting that the bionic structure increases the overall velocity of the jet-wake structures in channels 1 and 6, yet makes the distribution of velocity more uniform, thus improving the flow filed at the impeller outlet. Overall, the velocity fluctuation at the impeller exit of the bionic model is small and the velocity distribution in each channel is relatively uniform, which indicates that the bionic structure reduces the velocity gradient of the jet-wake structure, thus reducing the shear effect between the high-velocity fluid and the low-velocity fluid, which further stabilizes the flow field inside the impeller and reduces the energy loss.
To further reveal the drag reduction mechanism of the bionic structure, figure 14 displays the distribution of the velocity vector and streamlines on the blades at 0.6 $Q_n$. As can be seen from figure 14(a), at small flow rate, the flow inside the impeller is characterized by low velocity and high counterpressure gradient, the momentum of the moving fluid itself cannot resist the effect of viscosity and pressure difference. As a result, backflow occurs on the blade surfaces, which would squeeze the fluid particles in the boundary layer away from the surface, and the boundary layer gradually thickens and eventually detaches from the wall, causing severe flow separation to occur in the flow channel. With the arrangement of the dimples on the suction surface, vortices with lower velocity are formed in the dimples as presented in figure 14(b), which is equivalent to increasing the effective thickness of the viscous substrate layer and decreasing the velocity gradient of the boundary layer[21], thus effectively reducing the friction drag on the wall.
Figure 14. The vector distributions and streamlines on the blade of the original and bionic models at 0.6 $Q_n$.

4. Conclusions
In this paper, the dimple-type structures are constructed on the blade surfaces of a centrifugal pump based on bionic theory. The influence of the depth and location of the dimples on the hydraulic performance and drag reduction effect of the model pump is investigated by numerical simulation. The main conclusions are as follows:

(1) The hydraulic performance of the model pump gradually decreases as the depth of dimples increases, therefore, a shallower depth of the dimple is recommended to improve the performance of the centrifugal pump. Moreover, the arrangement of the dimples near the front section of the blade suction surfaces is more effective in improving efficiency and reducing the drag of the pump. At 0.6 $Q_n$, the best bionic scheme $SS_{11}$ increases the efficiency and drag reduction rate of the model pump by up to 1.7% and 2.7%, respectively.

(2) At the design and large flow conditions, the velocity distribution in the impeller channel is relatively uniform, and the relative velocity at the inlet and outlet of the bionic model is lower than that of the original model, which weakens the impacting effect of the fluid on the wall, thereby reducing the
wall shear stress of the blade surfaces. At small flow rate, the sharp decline of relative velocity is generated at the blade surfaces of several channels, indicating that unstable flow structures may occur in these regions. The bionic structures effectively suppress the low-velocity region of the aforementioned flow channels, which makes the velocity distribution more uniform, thus stabilizing the flow field and reducing the energy loss.

(3) With the arrangement of the dimples on the suction surface, vortices with lower velocity are formed in the dimples, which would change the velocity distribution and decrease the velocity gradient of the boundary layer. Therefore, the friction drag on the blade surface is reduced.

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