Hydrodynamic Engineering Calculation of Hull Suction Volume in the Backflow of Cascade Angle Region

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Abstract. This article uses computational fluid dynamics (CFD) method to carry out a numerical study on a rectangular high-load diffuser cascade of a ship hull with a full-blade high suction trough unit at different chord positions, and analyses the chord position and other parameters of the suction trough. The influence on the distribution law of the suction volume; the effect of the local extended suction scheme is explored through the cascade experiment. The computational domain of the numerical simulation includes the vacuum cavity inside the suction blade, and the boundary conditions are set according to the experimental conditions. The distribution law of the suction volume along the span is affected by the combined effect of the cascade flow path and the flow field of the blade cavity. Therefore, the suction surface suction groove/hole should be refined according to the specific characteristics of the three-dimensional high-load diffuser cascade flow field design.

Key words: Hull suction, fluid mechanics engineering, cascade corner area, backflow.

1. Introduction
When a ship is navigating in sea waves, rolling is more likely to occur and has a large amplitude, which seriously affects the seaworthiness and living comfort of the ship. Compared with the motion of other degrees of freedom, the roll is affected by the viscosity and the nonlinear characteristics of the motion itself are more obvious. In the early stages of the development of engineering ships, most of them were non-self-propelled ships, and their application areas were mostly inland or offshore. With the rapid development of the marine industry and the continuous advancement of market globalization, while engineering ships are developing towards large-scale and professional development, their operating areas have become wider and wider, so there are more and more self-propelled engineering ships. Was introduced to the market [1]. On the other hand, with the increasingly severe global energy situation, how to reduce energy consumption and improve efficiency has become a common concern in various fields. Under such background conditions, the research on the resistance performance of engineering ships with large openings on the waterplane has begun to receive more and more attention.

The article applies the surface layer suction technology to the suction surface of the large turning angle diffuser cascade mainly to reduce the high-profile loss and wake mixing loss caused by excessive load, increase the stage pressure ratio, and reduce the size of the compressor quality. Existing studies on adsorption compressors often focus on the influence of the chordal position of the suction groove on the suction surface to determine the optimal suction position. However, as the load of the cascade increases,
the end wall effect of the cascade is significantly enhanced, and the flow field near the suction surface of the blade shows significant three-dimensional characteristics. Therefore, the three-dimensional high-load diffuser cascade cannot copy the design experience in outflow applications, and must be based on the characteristics of the secondary flow field realize the optimal configuration of the suction position and the suction volume. In addition, the suction may bring additional aerodynamic interference to the flow field of the cascade, and the full-blade high suction groove will also reduce the strength of the blade to a certain extent. The computational domain of the numerical simulation in this paper includes the vacuum cavity inside the suction blade, and the boundary conditions should be set as far as possible in accordance with the actual conditions of the relevant cascade experiment.

2. Establishing the friction loss model of turbine suction reflux

2.1. Model establishment

When the gas flows through the compressor, due to the effect of viscosity, the flow velocity is the smallest near the wall of the flow channel, and the flow velocity of the main flow in the middle is the largest [2]. In this way, the flow is divided into many layers, and the flow velocity between layers is different, resulting in friction loss.

\[ h_{f'} = h_{f}\left[1 + 0.075 \frac{Re^{0.25}}{d_{h}} \right] \]  

(1)

In the formula, \( h_{f'} \) is the friction loss when the hypothetical channel is a straight pipe.

\[ h_{f} = 2c_{f}\frac{L_{w}}{d_{h}}w_{av}^{2} \]  

(2)

Where \( c_{f} \) is the friction factor, \( c_{f} = 0.0412Re^{-0.0375} \); \( d_{h} \) is the average hydraulic diameter of the blade channel, m; \( L_{w} \) is the average length of the blade channel, m; \( w_{av} \) is the average velocity in the blade channel, m/s.

\[ L_{w} = \frac{\pi}{8} \left( 2D_{2} - \frac{3D_{1} + D_{a} + b_{1}}{2} \right) (\sin \beta_{1s} + \sin \beta_{2s})/2 + \sin \beta_{2s} \]  

(3)

\[ d_{h} = \frac{D_{2} \sin \beta_{2s} + \frac{Z_{2}}{\pi} + \frac{b_{2}}{b_{2} - \frac{D_{2} \sin \beta_{1s}}{D_{1} - D_{a}}} + \frac{Z_{2}}{\pi}}{2} \left( \sin \beta_{1s} + \sin \beta_{2s} \right) \]  

(4)

\[ w_{av} = \left( w_{i} + w_{o} + 2w_{2} \right)/4 \]  

(5)

\( w_{i}, w_{o}, w_{2} \) is the outer diameter of the impeller inlet, the relative speed of the inner diameter, and the relative speed of the outlet m/s; \( r_{c} \) is the average radius of curvature of the blade channel on the meridian plane, m.

\[ r_{c} = \frac{1}{2} \left( \frac{D_{1} \frac{D_{1} + D_{a} - b_{1}}{2}}{4} + L_{2} \right) \]  

(6)

Where \( D_{2} \) is the impeller outlet diameter, m; \( \beta_{2s} \) is the impeller outlet installation angle; \( D_{1}, D_{a} \) is the impeller inlet outer diameter and inner diameter, m; \( \beta_{1s}, \beta_{1a} \) is the impeller inlet outer diameter and inner diameter installation angle \( \beta_{1s} = \beta_{1a} + b_{2} \) is the blade outlet width, m; \( Z_{2} \) is the number of blades at the exit of the impeller, in pieces; \( L_{2} \) is the axial length of the impeller, in m. \( D_{1} \) is equal to the diameter of the impeller inlet section, \( D_{1} = D_{2}; D_{a} = D_{1} - 0.003 \).
2.2. Optimization analysis of structural parameters

The treatment of \( \frac{d_b}{2D_1} \) in formula (1) is as follows. Divide the numerator and denominator of this fraction by \( D_2 \) at the same time to get:

\[
\frac{d_b}{D_2} = \frac{\sin \beta_{l,t}}{\pi} + \frac{Z_2}{b_2 \pi} + \frac{(D_{b,t} + D_{a,t})(\sin \beta_{l,t} + \sin \beta_{a,t})}{2D_1} \frac{2D_2}{(D_{b,t} + D_{a,t})} \quad (7)
\]

\[
\frac{r_b}{D_2} = \frac{1}{2} \left( \frac{D_{b,t} + D_{a,t}}{4D_1} - \frac{b_2}{2D_1} + \frac{L_b}{D_1} \right) \quad (8)
\]

The same method to deal with the \( \frac{L_b}{d_b} \) term in formula (2):

\[
\frac{L_b}{D_2} = \frac{\pi}{8} \left( 2 \cdot \frac{3D_{b,t} + D_{a,t} + b_2}{2D_2} \right) \frac{\sin \beta_{l,t} + \sin \beta_{a,t}}{2} + \sin \beta_{l,t} \quad (9)
\]

Substitute simplification conditions for equations (7) - (9): \( D_{b,t} = D_1, \quad \frac{D_{b,t}}{D_1} = 0.03, \quad \beta_{l,t} = \beta_{a,t} = \beta_{l,t} \) and compare the magnitudes, simplifying to

\[
\frac{d_b}{D_2} = \frac{\sin \beta_{l,t}}{\pi} + \frac{Z_2}{b_2 \pi} + \frac{2D_2}{100\sin \beta_{l,t}} \quad (10)
\]

\[
\frac{r_b}{D_2} = \frac{1}{2} \left( \frac{D_{b,t} + D_{a,t}}{4D_1} - \frac{b_2}{2D_1} + \frac{L_b}{D_1} \right) \quad (11)
\]

\[
\frac{L_b}{D_2} = \frac{\pi}{8} \left( 2 \cdot \frac{D_{b,t} + b_2}{D_1} \right) \frac{\sin \beta_{l,t} + \sin \beta_{a,t}}{2} \quad (12)
\]

The meaning of each parameter in the above formula is detailed in formulas (1) - (6). Next, we will deal with the friction factor and the average velocity in the blade channel in equation (2).

3. Numerical simulation of aerodynamic performance of adsorption compressor cascade

Based on the comparison of the blade load with the expansion factor as the measurement index, the paper carried out a high-efficiency and high-load large turning angle compressor cascade blade type. This section numerically studies the effect of surface layer suction on the aerodynamic performance of a super-high load compressor cascade under low-speed conditions [3]. Three different suction volumes are adopted for six different suction positions on the suction surface, and the blades are analysed. The distribution of the total pressure loss at the outlet of the grid, the diffusion factor and the airflow angle along the height of the blade, and the limit streamline of the suction surface and the static pressure of the profile are given to discuss how to use the surface layer suction technology in a high-load compressor. Choosing the best suction position and suction volume, as well as the suction and removal effect at different suction positions and suction volumes, provide for the upcoming wind tunnel experiment and the future blade design for higher load and diffusion capacity. The necessary theoretical basis.

3.1. Numerical method and scheme design

The thesis uses CFX software to simulate, and the discrete grid of runner space is generated by the pre-processing module ICEM. Near the blade wall and the cavity surface are all surrounded by O-shaped grids, and the remaining positions are surrounded by H-shaped grids. Control the mesh densification near the wall [4]. The first layer of mesh on the blade wall and end wall meets \( y+ = 18 \), and the average
Total number of mesh nodes is about 650,000. The N-S equation solver uses the FoxServer module to numerically simulate the steady flow field of the blade cascade. The high-resolution format is used to solve the continuum equation, the momentum equation and the energy equation, and the turbulent flow energy and the turbulent energy dissipation rate equation are solved in the upside-down style. The turbulence model uses the K-Epsilon model. The aerodynamic boundary conditions used in the calculation are given as follows: the reference pressure is 0Pa; the total inlet temperature is 300K, and the total pressure is 104325Pa; given the design conditions, the inlet air attack angle is 0°, and the average static pressure of the cascade outlet section is 101325Pa. In the calculation process, this article firstly carried out an experimental check on the CFX calculation results, carried out multiple trial calculations according to the experimentally measured boundary conditions, and adjusted some of the software settings, so that the calculation results can be compared with the experimental results. Figure 1 shows a comparison chart of the CFX experiment of a certain straight blade cascade. The results show that the numerical simulation is credible according to the above-described settings.

Figure 1. Comparison of calculation and experimental results

The angle of the suction groove of the blade cascade (the angle between the feed direction of the experimental processing of the suction groove and the tangent to the blade surface at the corresponding position) is 90°. The centre position of the groove is 15%, 25%, 35%, 42%, 48%, 60% of the relative chord length l/B on the suction surface from the leading edge of the blade [5]. The suction groove is 2mm wide, 2.5mm deep, and high. 140mm, the geometric structure is symmetrical along the direction of the blade height, and the suction groove and suction position are selected according to the strength of the blade in the experiment. The geometric parameters of the cascade are shown in Table 1. In the calculation, there are three types of intake β, which are respectively 0.5%, 1.0%, and 1.5% of the inlet flow. When discussing the results, "or" represents the prototype, that is, no suction cascade. For example, in the example "40-15", the former represents the inhalation position, and the latter represents the percentage of the inhalation.

Table 1. Cascade parameters

| Turning angle θ(°) | 60 |
|-------------------|----|
| Chord length B (mm) | 120 |
| Consistency       | 1.33 |
| Leaf inlet angle β1P(°) | 50 |
| Aspect ratio h/b  | 1.33 |

3.2. Results and discussion

Figure 2 shows the distribution of the average total pressure loss coefficient of the pitch mass along the leaf height for different leaf turning angles. It can be seen from the figure that the five suction positions close to the leading edge have similar effects on the total pressure loss of the cascade outlet by using the
surface layer suction. In the middle of the cascade and the 20% relative blade height area from the end wall, the experimental selection Within the scope of the suction scheme, the total pressure loss at the outlet of the suction cascade is significantly lower than that of the non-suction cascade, and the greater the suction volume, the greater the reduction in loss. Among them, the distance from the leading edge of the axis is 15%. The reduction in total pressure loss is the lowest when the chord length position is absorbed by the surface layer; when the axial chord length is 60% from the leading edge, the total pressure loss at the outlet of the cascade is changed compared with the other five suction layers [6]. The air position is obviously larger when the surface layer is used for suction. After suction, the loss in almost the entire leaf height range is lower than that without suction. Among them, the loss in the middle area is lower than that of the other five suction positions. The suction time is similar, and in most other blade height ranges, the loss reduction of the suction cascade is greater than that of the other five suction positions, and the greater the suction volume, the greater the loss reduction., But the loss of the three types of inspiratory flow is not big; in addition, the absorption of the boundary layer has no significant effect on the total pressure loss in the proximal wall area. For the ultra-high load compressor cascade, the main part of the total loss comes from the end area, and the use of suction on the suction surface can effectively reduce the end wall loss, which is effective in reducing the total loss of the cascade. Further analysis can also find that the effect of inhalation is the best in the low-energy flow accumulation zone.

Figure 2. Pitch mass average total pressure loss coefficient distributed along leaf height

The expansion factor is an important parameter to measure the expansion capacity of the cascade and the load of the cascade. The blade profile used in this article is designed with a high expansion factor as the goal. Figure 3 shows the distribution of the average diffusion factor of the pitch mass along the leaf height. It can be seen from the figure that the diffusion of the blade cascade is in the middle of the blade cascade and the 20% relative blade height area from the end wall after the surface layer is sucked [7]. The factor is increased when not inhaling, and the further the inhalation position is, the greater the degree of influence of inhalation on the expansion factor of the cascade; the inhalation at the position closest to the trailing edge causes the expansion factor to be in full the increase in the leaf height range is significant, and the increase degree increases with the increase of the inhalation volume.
Figure 3. The average spreading factor of the pitch mass is distributed along the leaf height

The above analysis results show that the use of surface layer suction can significantly reduce the loss of the cascade, and improve the pressure expansion capacity of the cascade, and the closer the suction position is, the greater the performance improvement of the cascade, at a distance of 60% from the leading edge the suction effect of the axial chord is the best. Adopting different suction positions for suction has little effect on the performance of the middle part of the cascade, that is, when the suction effect in the end zone is the best, the overall suction effect of the cascade is also the best. The high-load compressor studied in this paper has a large loss in the end region of the cascade, and a large amount of low-energy fluid accumulates in the corner area. Adopting the surface layer to suck the low-energy fluid away, thereby reducing the cascade loss and improving the pressure diffusion of the cascade ability.

The profile static pressure is also one of the parameters reflecting the load and diffusion capacity of the blade cascade. Figure 4 shows the distribution of the profile static pressure of different blade heights along the blade profile. It can be seen from the figure that in various suction schemes, only the suction at 60% of the axial chord length from the leading edge has an effect on the static pressure distribution of the pressure surface at 50% of the leaf height, and the static pressure distribution of the suction surface at this time has a greater impact, while the static pressure distribution of other solutions does not change much. After taking inhalation at 60% of the axial chord length from the leading edge, the load of the cascade at the middle and end of the cascade increases significantly in most of the chord length, and the load on the trailing edge is slightly reduced. After the airflow passes through the leading edge the continuous expansion length of the cascade has also increased significantly, and the static pressure flat section of the suction surface (that is, the section that loses the expansion capability) is basically eliminated, especially the improvement of the end is more obvious, which greatly strengthens the expansion capability of the entire cascade [8]. The absorption of low-energy fluid at the end accelerates the flow velocity of the airflow in the end zone and weakens the flow separation in the end zone, so that the performance of the entire cascade is improved. In addition, the static pressure distribution before the inhalation position changes with the inhalation flow rate to be greater than that after the inhalation position, which indicates that the performance improvement before the inhalation position by using the boundary layer is significantly greater than that after the inhalation position. It can also be found in the figure that the best suction position is close to the position where the middle part of the non-aspirated cascade begins to lose its diffusion capacity, and this is also the beginning of the large separation of the cascade.
4. Conclusion

The distribution law of the suction volume along the leaf height is mainly affected by two factors. When the local space of the blade cavity adjacent to the suction groove is large, the aerodynamic characteristics in the cascade channel dominate the distribution of the suction volume, making it a C-shaped distribution with high ends and low medium diameter. Excessive stroke resistance will result in a large gap between the pitch diameter and the local suction volume of the end zone. When the suction volume is 1.5% of the inlet flow, the two-end suction and middle suction schemes studied in this paper respectively push the suction surface layer and thin the suction surface layer by reducing the corner zone separation. The flow field near the mid-diameter of the suction surface is improved, and the mid-diameter loss of the prototype cascade is reduced by 33% and 16%.

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