Influence of blade tip rounding on tip leakage vortex cavitation of axial flow pump

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Abstract. Tip leakage flow in axial flow pumps is mainly caused by the tip clearance, which is the main cause of tip leakage vortex cavitation and blade tip cavitation erosion. In order to improve tip clearance flow and reduce TLV cavitation, four schemes were adopted to the round blade tip. These are: no tip rounding, one time tip clearance tip rounding, two times tip clearance tip rounding, four times tip clearance tip rounding. Using SST k-ω turbulence model and Zwart cavitation model in CFX software, this simulation obtained four kinds of inner flow field results. The numerical results indicated that with the increase of $r^*$, NPSHc gradually increased and the cavitation performance reduced. However, corner vortex was eliminated so that cavitation in gap was restrained. But TLV vorticity increased and cavitation’s range here had a little expansion. Combined with the research of this paper and the different analyses of four schemes, we recommend adopting the two times of the tip clearance rounding.

1 Introduction

Leakage flow in axial-flow pump is the flow through the gap between the tip of a rotating blade and the fixed shroud. Leakage flow is caused by the pressure difference between the pressure and suction sides of the blade. As the leakage flow passes through the tip clearance, it interacts with the mainstream and secondary flows and forms a tip leakage vortex(TLV)\cite{Furukawa et al., Gerolymos et al., Gourdain et al.}. As a result, tip leakage cavitation caused by TLV erodes impeller chamber, blade tip and rotor flange. Research has been conducted in order to understand the flow phenomena near the blade tip region. Furukawa et al., Gerolymos et al., Gourdain et al.\cite{Furukawa et al., Gerolymos et al., Gourdain et al.} studied on the tip flow by numerical simulations, mostly performed using Reynolds average Navier–Stokes (RANS) approach. Masahiro Murayama\cite{Murayama} observed tip leakage vortex cavitations originating from the tip clearance of an oscillating hydro-foil experimentally, Huixuan Wu and Miorini R et al\cite{Wu, Miorini} measured flow structures in the tip region of a water-jet pump rotor, including the tip-clearance flow and the rollup process of a tip leakage vortex.

In the actual operation, by rounding the pressure surface and suction surface outer edge\cite{Wu}, cavitation can be lessened and eliminated when clearance cavitation erosion occurs. However, there are no reports focusing on the mechanism of this method to reduce the cavitation. The value of tip rounding should not be randomly selected because it brings both improvement of tip flow and more leakage loss. Numerical calculation has been adopted to investigate the clearance flow field of different tip rounding schemes to further understanding the mechanism of this method and tip
rounding selection principle. In addition, pump head and hydraulic efficiency in each scheme and different performances in different NPSH have been compared and analyzed thus appropriate tip rounding scheme has been selected in consideration of pump head, hydraulic efficiency and cavitation performance.

2 Pump geometry and simulation methods
In this paper, an axial flow pump model with specific speed 700 is shown in Fig.1. its design parameters are: flow rate $Q_{opt} = 392\text{m}^3/\text{h}$, Head $H=3.2\text{m}$, Rotation speed $n=1450 \text{ r/min}$, impeller diameter $D_2 = 200\text{mm}$, tip clearance $(TC)$ is $1\text{mm}$, hub ratio is 0.4681, impeller blade number is 4, its guide vane number is 7.

![Figure 1. Model pump and test-bed.](image)

Structural mesh was generated around the whole flow passage with 15 layers in the clearance region and the area near wheel flange was refined locally so as to accurately capture the clearance flow and leakage vortex structure. In this study, numerical calculation was based on Reynold-Average N-S (RANS) and SST $k-\omega$ closed equations for turbulence models. The $k-\omega$ based Shear Stress Transport model was designed to give a highly accurate prediction of the onset and the amount of flow separation under adverse pressure gradients by the inclusion of transport effects into the formulation of the eddy-viscosity. The superior performance of this model has been demonstrated in a large number of validation studies (Bardina et al. [10]). Cavitation simulation is based on Zwart-Gerber-Belamri cavitation model, whose the total interphase mass transfer rate per unit volume is:

$$m_{fg} = F \frac{3r_{nuc}(1 - r_g)\rho_f}{R_b} \sqrt{\frac{2}{3} \frac{|p_v - p|}{\rho_f}} \text{sgn}(p_v - p)$$ (1)

where $F$ is an empirical factor which may differ for condensation and vaporization, and $r_{nuc}$, whose value is $5 \times 10^{-4}$, is the volume fraction of the nucleation sites, $R_b$, whose value is $1 \times 10^{-6}$, is represents the bubble radius, $p$, is the pressure in bubble, $p$ is the pressure in the liquid surrounding the bubble, $\rho_f$ is liquid density.

The simulation in this study is based on ANSYS CFX software, and the setting and boundary condition is showed in Tab. 1.
Table 1. Simulation setting.

| Inlet           | Total pressure |
|-----------------|----------------|
| Outlet          | Mass flow rate |
| Wall            | No-slip wall   |
| Medium          | 25°C water and 25°C vapour |
| Interface       | Stage          |
| Turbulence model| SST k-ω        |

Figure 2. Structural mesh of domains.

3 Simulation of full flow field and performance testing

By calculating the full flow field, different pump heads and hydraulic efficiencies were obtained in different flow rate of the some model pump as well as the heads with different device NPSH by changing the inlet pressure.

The performance test was conducted on the model pump and thus curves of $Q-H$ and $Q-\eta$ were received. Starting the vacuum pump to extract the air in the cavitation tank which aim to lower the device NPSH, we obtained the $NPSH-H$ curve by measuring the head with different device NPSH. Comparing the performance and cavitation curve of comparative simulation calculation and the test, we knew that two curves show a good agreement in the operating condition and big operating condition while the error was lager in small one. What's more, the cavitation curves were well agreed with in the operating conditions with slightly better performance by simulation than in the fact. It proved the accuracy and rationality in this simulation calculation.

![Figure 4](image)

Figure 4. Comparison between experiment result and simulation result

4 Simulation study with different tip rounding scheme

4.1 Rounding blades model

On the basis of the last section, rounding processing on the model pump impeller pressure side flange and defining the rounding coefficient:

$$r^* = \frac{R}{TC}$$

Where $R$ is rounding radius, and $TC$ denote to tip clearance size, which is 1mm in this study. We know that it increases the gap size as well as the volume loss and leads to the increased leakage while it improves the flow conditions in the clearance area and reduces the local losses, therefore, we adopt three kinds of $r^*$ that equals to 1, 2, 4 respectively based on the analysis and experience in order to choose the right blade tip circle size.
This paper developed simulations and analyses only for the impeller fluid because the blade tip region was the key to the research and analysis. Assuming that the impeller flow condition formed cyclical symmetry, we only calculated the single flow passage of the impellers and created structural mesh for the computational domains with 2 million meshes for four kinds of schemes. The blade tip region mesh distribution was extremely important for this research, so 10 layer meshes arranged in the clearance area. The tip area of four plans is shown in figure 5.

4.2 The influence of different tip rounding schemes on the external characteristics

Applying the same setting to calculate the external characteristic for four schemes in computational domains, we obtained $Q-H$ curve and $Q-\eta$ curve of the four schemes respectively by statistics.

Figure 6 shows that the head and efficiency have little change after the impeller rims being rounded as $r^*=1$, $r^*=2$ and $r^*=4$. Compared with the original scheme, change of the head was not more than 1.4% and the efficiency was less than 0.7%. In different flow conditions, the trends of different tip rounding schemes were not the same. As shown in table 2 where $r^*$ varying from 0 to 1, there was obvious decrease in the head and efficiency in the operating conditions ($Q/Q_{opt}=1$, $n=1450$ r/min. This phenomenon was due to the large volume loss after the tip rounding. However, the head had the obvious rise and the efficiency remained level when $r^*$ changes from 1 to 2 with the flow in tip area improved. When $r^*$ changed from 2 to 4, the head fell and the efficiency changed little. Therefore, we suggest that the $r^*$ is 2 or so considering both the head and efficiency on the operating conditions.

4.3 The effect of different blade tip rounding schemes on the blade tip leakage cavitation

4.3.1 Compare of cavitation curves. In order to further receive the effects on the blade tip leakage cavitation by $r^*$, we changed the inlet pressure to change the device NPSH (NPSH) on the operating conditions. And the cavitation curves $NPSH-H$ of four schemes were obtained as shown in figure 8. When the head fell by 3%, we recognized the NPSH value as the critical NPSH (NPSHc) and
calculated the NPSH in the four kinds of schemes as shown in table 2.

By analyzing, we find that after tip rounding the cavitation performance presented worse than original, that is to say the NPSHc became larger. Further study showed that the NPSHc changed slowly when $r^*$ dropped to 2, while the NPSHc change continued to rise when $r^*$ varying from 1 to 2.

| $r^*$ | Head/m | Efficiency/% | NPSHc/m |
|-------|--------|--------------|---------|
| 0     | 3.28941| 89.30424     | 3.01546 |
| 1     | 3.24526| 88.80676     | 3.0869  |
| 2     | 3.28527| 88.75597     | 3.1007  |
| 4     | 3.27736| 88.78419     | 3.1734  |

Table 2. Performance indexes under different $r^*$.

4.3.2 In-plane vorticity and cavitation bubble fraction with different $r^*$. In order to analyze the flow characteristics of blade tip region, we compared the four schemes of in-plane vorticity (that is two dimension vorticity in one plane) and bubble distribution for the middle section of tip chord length.

As shown in figure 7, in original scheme where $r^* = 0$ in the A region, the vorticity is about $-3600 s^{-1}$ as the counterclockwise corner vortex forming for the narrow gap by the leakage flow. However, it could be found that the corner vortex significantly cut until it disappeared after tip rounding. Meanwhile, we also found that the leakage flow increased with the $r^*$ and thus the leaking fluid flowed into the gap with faster speed and entered into the back area. Therefore, separated vortexes enhanced and showed the larger counterclockwise vorticity in B region. Leakage vortexes which were formed by the entrainment between the leakage flow and mainstream was the most obvious phenomenon in the figure, that was C region. The scope of leakage vortexes widened with the increase of $r^*$, and in-plan vorticity became larger, but it was not obvious.

Coupled with the distribution of cavitation on the plane as shown in figure 8, it could be seen that there were more bubbles distributed in the tip clearance area of the original scheme, but tip rounding had greatly reduced separated vortexes in area A, so it eliminated low pressure area in this field and...
removed the position where cavitation occurred backward. When it reached \( r^* = 2 \), the clearance cavitation had largely disappeared. With the increase of \( r^* \), the bubbles induced by leakage vortexes extended to the leaking direction. Its area continued to expand. And when it reached \( r^* = 2 \), the leakage vortex cavitation had been out of the bubbles induced by fixed cavitation on the back. We could find that the position where the leakage vortexes occurred as well as the influence area had removed to the leaking direction after tip rounding.

5 Conclusions
The numerical calculation method adopted in the paper accurately calculated the external characteristic curve as well as the cavitation curve of the model pump. The calculation results fitted the experiment data quite well.

The head and efficiency slightly changed after the impeller rims being rounded as \( r^* = 1 \), \( r^* = 2 \) and \( r^* = 4 \). Compared with the original scheme, change of the head was not more than 1.4% and the efficiency was less than 0.7%. Tip rounding schemes had different variation trend on different flow conditions. Then we obtained the NPSHc of the four schemes by numerical calculation of cavitation. It could be found that with the increase of \( r^* \), NPSHc gradually increased and the cavitation performance became worse. However, the variation trend had a slowdown tendency when \( r^* \) reached 2. And then, tip rounding can eliminates the corner vortexes in the A region by analysis of in-plane vorticity contours and cavitation fraction on the middle chord, so the occurrence of cavitation was restrained and the bubble distribution in the gap area was greatly reduced thus the position where cavitation occurred was backward removed. With the increase of \( r^* \), the leakage flow velocity increased so the TLV vorticity increased and cavitation’s range here had a little expansion. What's more, the leakage flow shifted toward the leaking direction. Finally, combined with the research of this paper and the different analyses of four schemes, we recommend adopting the scheme of \( r^* = 2 \)(that is 2 times of the tip clearance) for dealing with tip rounding.

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