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INFLUENCES OF DEPTH AND RIM WIDTH OF SQUEALER TIP ON THE LEAKAGE FLOW AND HEAT TRANSFER

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ABSTRACT

Blade tip structure effects the strength of tip leakage flow and the intensity of heat transfer, which is caused by tip clearance between blade tip and shroud in turbine. To weaken the tip leakage flow, squealer tip which is mainly controlled by squealer rim width and height, has been studied in academia and industry. Several cases were calculated in this work, including four different squealer rim width and four different squealer rim height for each width. Means of solving Reynolds-averaged N-S equations in conjunction with SST turbulence model was used. The result shows that the increase of squealer rim height can reduce the tip leakage loss, while the increase of squealer rim width makes more tip leakage loss. Meanwhile, three high thermal transport areas on the blade tip surface need attention.

INTRODUCTION

In turbomachinery, in order to avoid rubbing between the blade tip and casing surface, there is a proper tip clearance between the rotating blade tip and the casing. Considering assembly allowance and thermal expension when turbine is operating, the clearance can not be too small. In addition, the pressure between the pressure side and suction side of the blade is different, that leads to a part of fluid flowing from the tip gap, and results in aerodynamic loss(Mischo et al, 2008).

There are various numerical and experimental studies about reducing the tip leakage flow. Generally, blade tip design is benefical to the turbine effectiveness by reducing the tip leakage flow, like pressure side squealer, suction side squealer and squealer tip on the blade. Ameri et al.(Ameri et al, 1998) conducted a numerical calculation on the GE E^3 first-stage, and found that there were at least two distinct vortices within the tip squealer, which offered additional blockage to the flow through the tip clearance. Nho et al.(Nho et al, 2012) dealt with an experimental investigation on effects of turbine blade tip shape on total pressure loss and showed that the double squealer tip and the grooved along pressure side tip performed best. An experimental study by Jung et al.(Jung et al, 2021) on various squealer tips with five-hole probe measurement system showed that with the squealer depth increasing, the mass-averaged loss for the squealer tip decreases, has a minimum value when squealer depth to span is 3.82%, and then increases but with the squealer width increasing, the loss tends to increase. Senel et al.(Senel et al, 2018) carried out a numerical investigation on the influence of squealer width and height near a high-pressure turbine blade and the conclusion is that the squealer tip can reduce mass of tip leakage flow rate and as the squealer depth increases and the width decreases, the tip leakage flow rate tends to decrease. Jeong and Sang(Jeong & Lee, 2020) obtained that the tip leakage flow became less with a squealer tip than a flat tip and when the tip clearance incearsed, the loss change with squealer depth became more considerable. In terms of thermal load, blade tip has a high thermal load regions because of the existance of tip leakage flow. Scholars carry out numerous studies on cooling blade tip, including using film cooling. Heat transfer prediction for turbine blade was dealt by Yang and Feng(Yang & Feng, 2007) using numerical calculation, and the conclusion was that the average heat transfer rate on the tip surface reduces with squealer depth increasing. Lu et al.(Lu et al, 2013) carried out a numerical investigation of film cooling effectiveness of the blade tip, and found that more area of the blade tip was covered by the film cooling with squealer tip than flat tip, but neither of them provided adequate protection to the leading and trailing edge. Lee et al.(Lee et al, 2009) experimentally studied the heat transfer with a high-turning turbine rotor blade, and found that high thermal load was observed along the tip gap vortices and the re-attachment line.

In the literature, there are limited number of studies considering collaborative effect of squealer depth and width on tip leakage flow. This paper researches four different squealer depths, each with four different squealer widths by numerical simulation.

NUMERICAL METHODS

The research blade is a straight blade based on a gas turbine large turning angle blade, showing in Fig.1. The inlet boundary of the simulation domain from the leading edge of the cascade is 1 time the axial chord length, and the outlet boundary of the simulation domain from the trailing edge of the cascade is 1.5 times the axial chord length. The design specifications of the cascade used in this paper are listed in Table 1. The mesh for calculating is unstructured and all the boundary layers is set as 0.005mm to keep the y+ value at a reasonable level.

Table 1 Design specifications of the cascade

Blade Span, s [mm]	119
Blade Axial Chord, C_a [mm]	105
Blade Pitch, p [mm]	90
Tip Gap Clearance, τ [mm]	1
Inlet Attack Angle [°]	0
Inlet flow velocity, v [m/s]	30
Inlet Mass Flow Rate, m_0 [kg/s]	0.2754
Outlet Pressure, p_0 [Pa]	101325

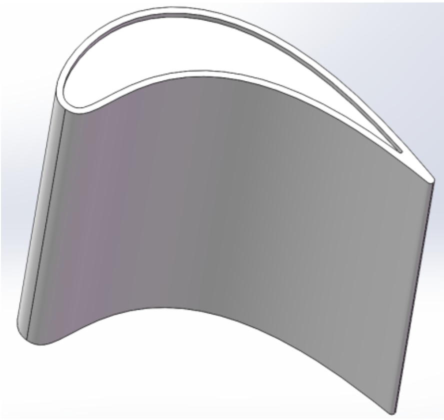


Figure 1 blade using in this paper

Numerical calculations are performed by solving the three dimensional, steady and turbulent form of the Reynolds-Averaged Navier-Stokes (RANS) equations using the ANSYS CFX 19.0. In literature(Senel et al, 2018; Kharati-Koopaee & Moallemi, 2020; Shao et al, 2020), the Shear Stress Transport (SST) turbulence model is used to simulate similar situations, and show good agreements with experimental results. Therefore, the SST turbulence model is used in this paper. The heat transfer mode is total energy and the advection scheme as well as the turbulence numerics mode is high resolution. The working gas is ideal gas, which temperature is 350K and the solid wall is adiabatic.

This paper considers 16 cases: four different squealer depths, 1, 2, 3, 4mm, and four different squealer widths, 1, 2, 3, 4mm, for each depth. For ease of description, in the following part of this paper, use w1d1 to represent the squealer width and the squealer depth are 1mm and so on.

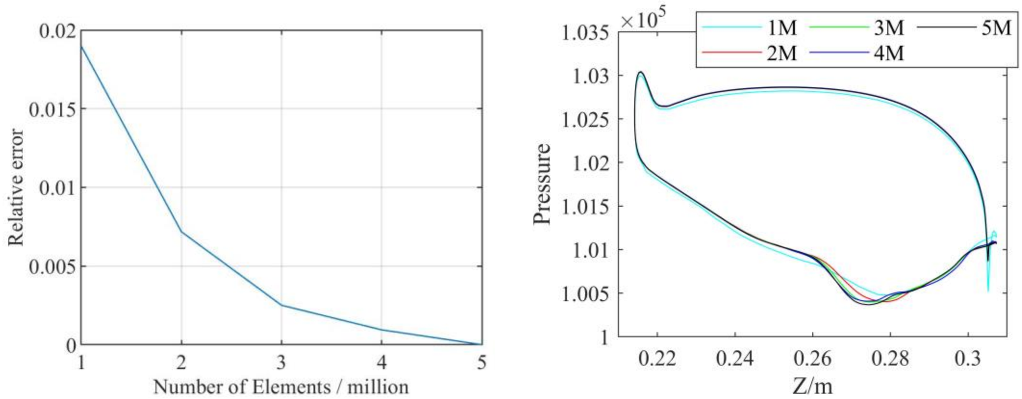


Figure 2 mesh independence test: relative deviation of leakage flow and pressure distribution

A mesh independence study was carried out for w1d1 using 1 million to 5 million elements respectively. The deviation of the leakage flow relative to 5 million elements and the pressure distribution at 95% of the blade height are shown in Fig.2. As can be seen from Fig.2 that the deviation between 4 million and 5 million elements is very small. Therefore, the mesh elements is based on 4 million for all cases in this paper.

RESULTS AND DISCUSSION

Aerodynamic analysis

The main purpose of this paper is to understand the effect of different squerler depth and width on the tip leakage flow so that the proportion of the mass of tip leakage flow in the total inlet mass flow acts as a target to evaluate effects of different squerler. Generally speaking, the tip leakage flow is formed because the pressure at the pressure side of the turbine blade is higher than that at the suction side, casuing a part of working gas flowing from the pressure side through the blade tip clearance to the suction side. While in the actual turbine, the situation is more complicated. A part of working gas may flow from the suction side through the blade tip clearance to the pressure side when the attack angle is zero or negative. To simplify these situations, the tip leakage flow contains flow from the pressure side, as well as flow from the suction side. When calculating the mass of tip leakage flow, all of the working gas flow into the tip gap will be included.

Numerical results of 16 different squealer tip cases are listed in Table 2 and Fig. 3 presents the proportions of tip leakage flow at various squealer widths and depths. The tip leakage flow tends to increase with the increase of squealer width at any fixed depth but when the squealer depth increase, the tip leakage flow will reduce at any fixed width. In another word, within the range of this research, effect of squealer depth and width on the tip leakage flow is obviously, deep and thin squealer leading to a small tip leakage flow. This conclusion is as same as the result of Senel et al.(Senel et al, 2018) and is similar as the result of Jeong and Sang(Jeong & Lee, 2020). In this paper, the squealer depth to the blade span does not reach to 3.75% as mentioned in the work of Jeong and Sang, so there is not turning point in this paper. Researches on the effect of squealer tip on the tip leakage flow, different research groups with different blades or methods may not be exactly the same, which is recognized by scholars(WANG & XUAN, 2020).

Table 2 Numerical result

Case	m/m_0	Case	m/m_0	Case	m/m_0	Case	m/m_0
w1d1	0.017042	w1d2	0.016264	w1d3	0.016070	w1d4	0.015622
w2d1	0.017757	w2d2	0.017067	w2d3	0.016698	w2d4	0.016371
w3d3	0.018215	w3d2	0.017473	w3d3	0.017011	w3d4	0.016854
w4d1	0.018330	w4d2	0.017672	w4d3	0.017300	w4d4	0.017162

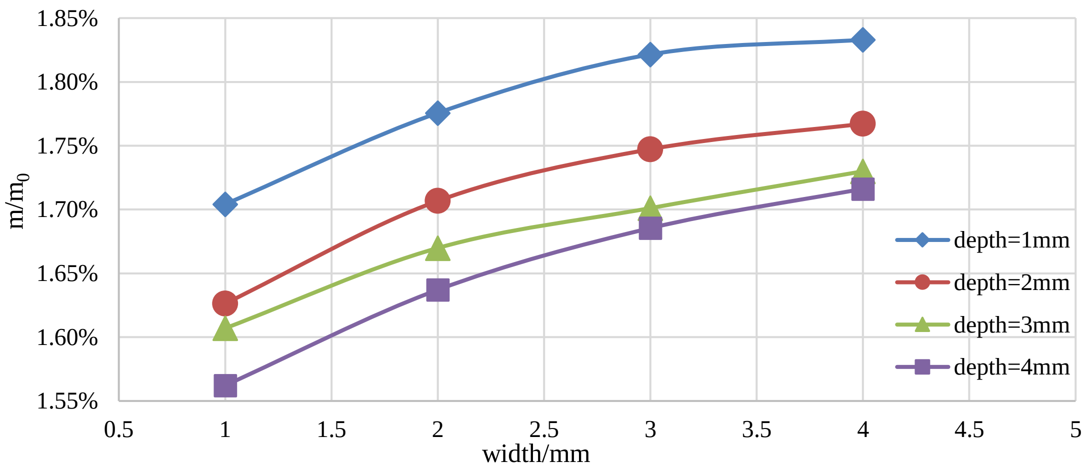


Figure 3 Numerical result

Using w/d represents squealer width divided by squealer depth in order to get a dimensionless parameter of squealer tip, shown as Fig. 4. The Y-axis in Fig. 4 still is the proportion of the mass of tip leakage flow in the total inlet mass flow, while the X-axis is the natural logarithm of w/d. What is important is that, the mass of tip leakage flows are very close when the w/d are equal, regardless the value of squealer width or depth. As shown in Fig. 4, these 16 different squealers can be descripted with one simple straight line.

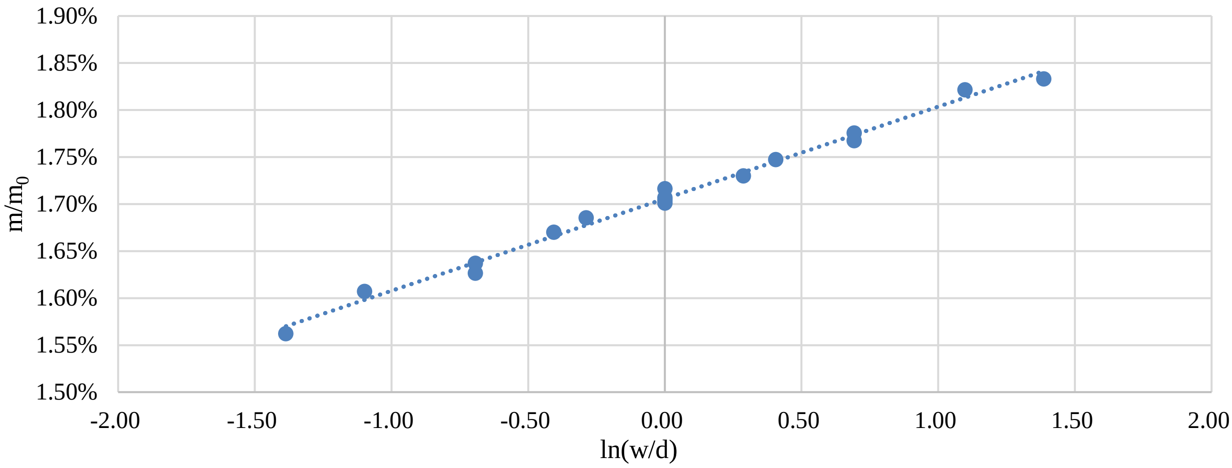


Figure 4 Numerical result with natural logarithm of w/d

The main flow structures observed of four different squealer depths with a fixed width equalling to 1mm are shown in Fig. 5 and 6 using average vorticity (Ws-ave) and streamlines to obtain visual aerodynamic effects. The Ws-ave is defined as follow:

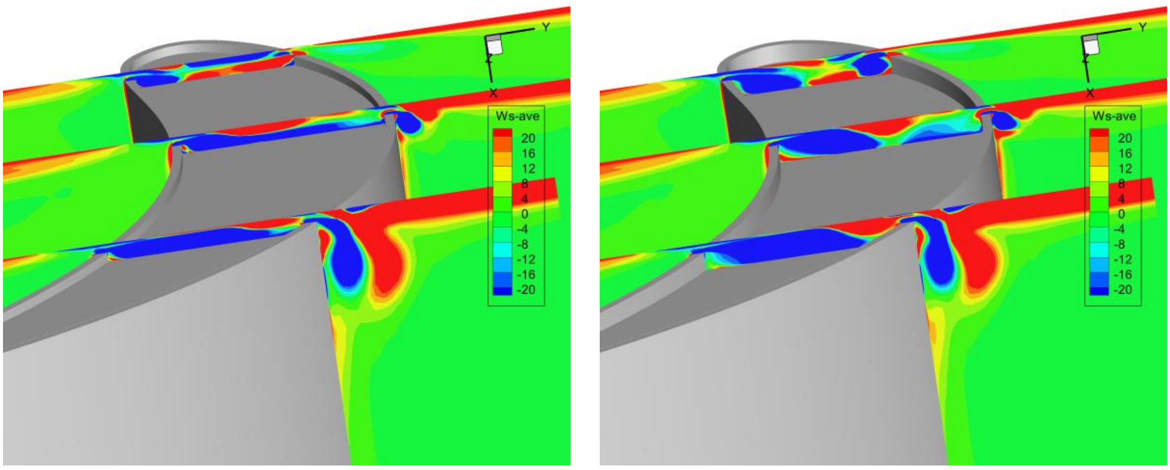
$$\text{Ws-ave}=\frac{\left[\left(\frac{\partial w}{\partial y}-\frac{\partial v}{\partial z}\right)*u+\left(\frac{\partial u}{\partial z}-\frac{\partial w}{\partial x}\right)*v+\left(\frac{\partial v}{\partial x}-\frac{\partial u}{\partial y}\right)*w\right]}{\sqrt{(u^2+v^2+w^2)}}*\frac{\text{L}}{\text{U}}$$

where the u, v, w are the velocity in the x, y, z directions, L is the characteristic length and U is the inlet velocity.

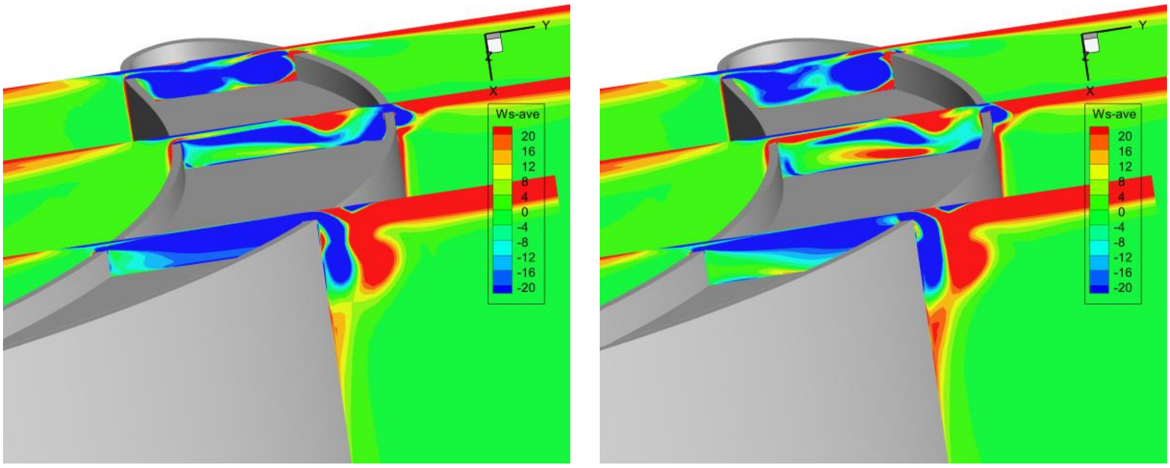
According to Fig.5 and 6, there are two vortex structures in the squealer near the leading edge. As the squealer depth increases, the vortex structures near the leading edge getting stronger and the vortex structure near the pressure side transfer to the suction side faster, leaving the tip clearance from the suction side closer to the leading edge.

Fig.7 and 8 represent the main flow structures observed of four different squealer width with a fixed depth equalling to 4mm, using average vorticity and streamlines to obtain visual aerodynamic effects. The two vortex structures near the leading edge become less with the squealer width increase when depth is fixed but the regions that the pressure side vortex structures leaving the tip clearance from the suction side keep same for all these four cases.

From the above analysis, it can be seen that among the range of this paper, whether the squealer depth increase or the squealer width reduce, leading to strengthen the vortex structures near the leading edge, which means, larger space of thesquealer tip making the vortex structures stronger. Combining the conclusion of the effect of squealer tip to the mass of tip leakage flow, a inference can be abstracted that: thinner and deeper squealer meaning bigger squealer tip space, which is beneficial to development of vortex structures near the leading edge, and then the stronger vortex structures can reduce the tip leakage flow, resulting in a less tip leakage flow mass rate.



depth=1mm **depth=2mm**



depth=3mm depth=4mm

Figure 5 average vorticity result of squealer width=1mm

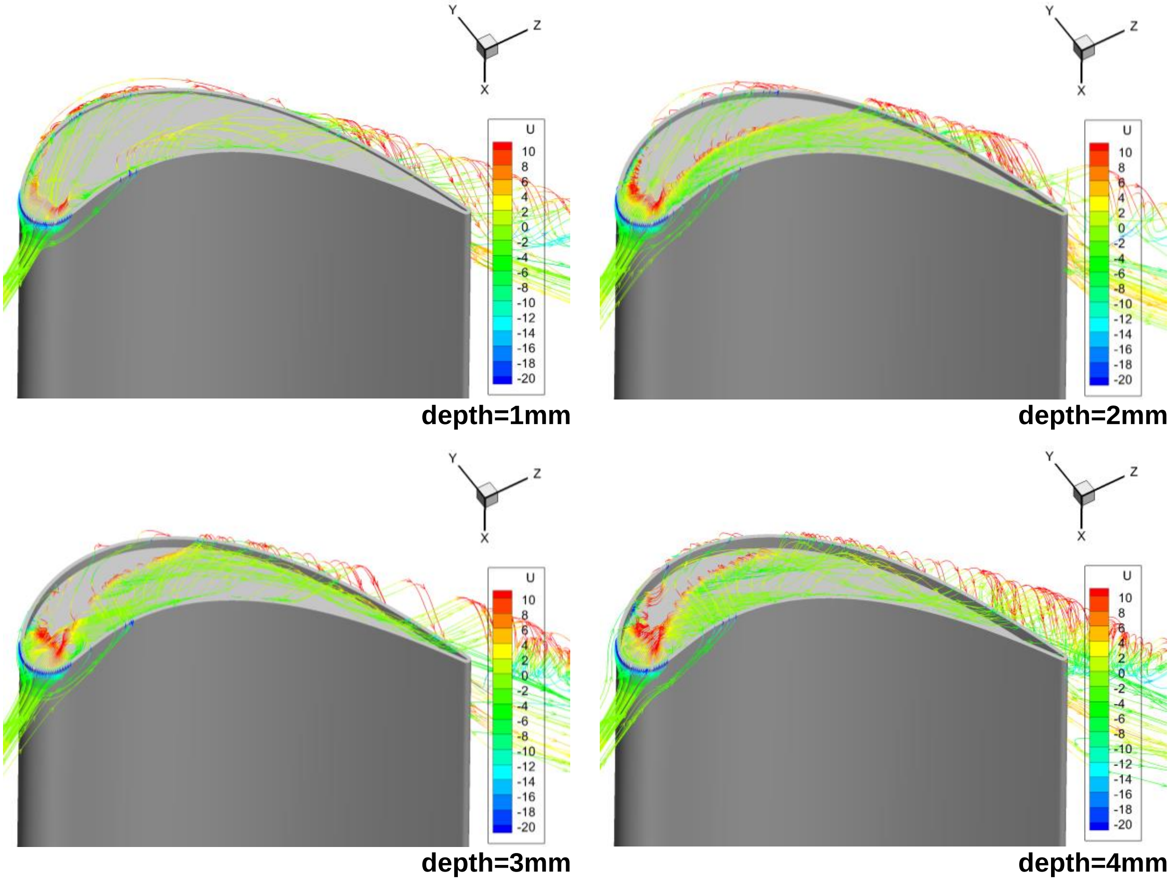
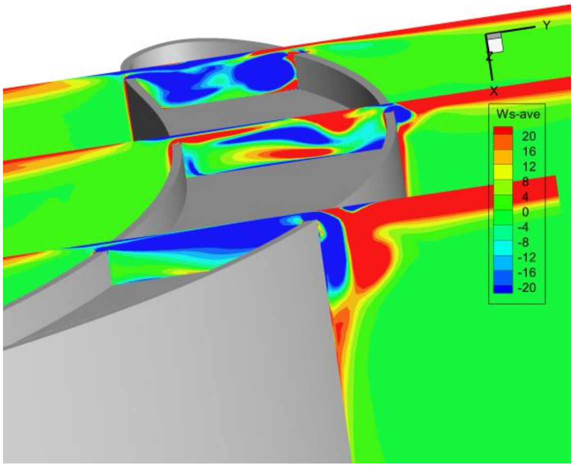
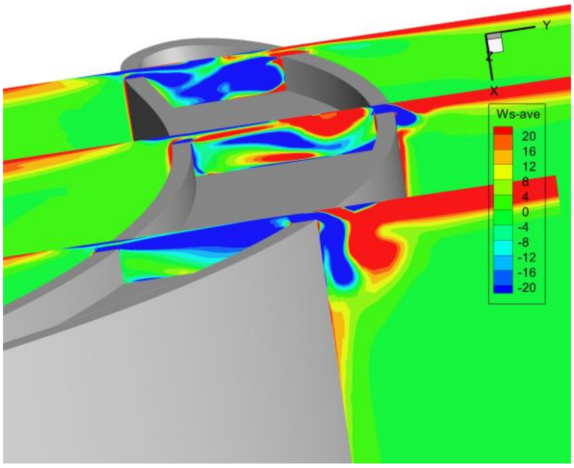


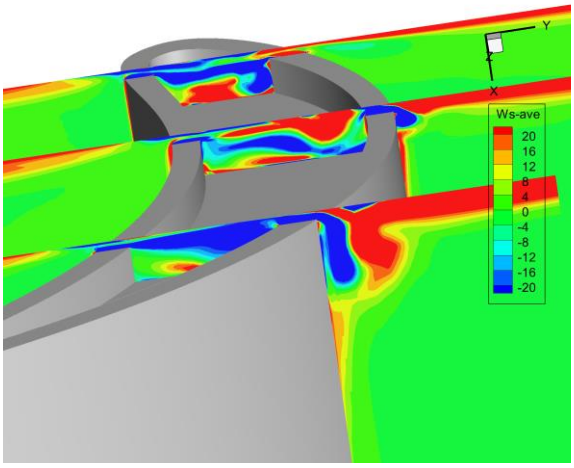
Figure 6 streamlines result of squealer width=1mm



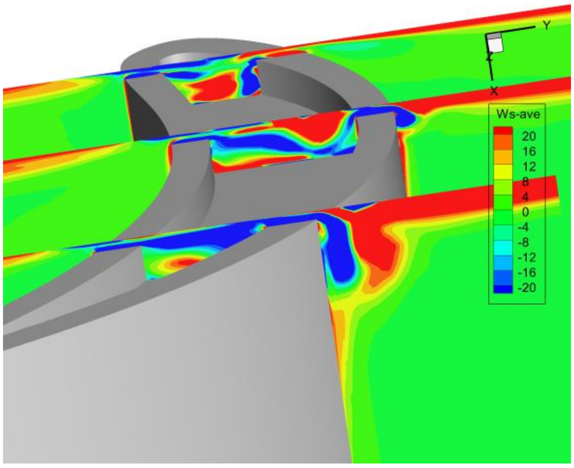
width=1mm



width=2mm

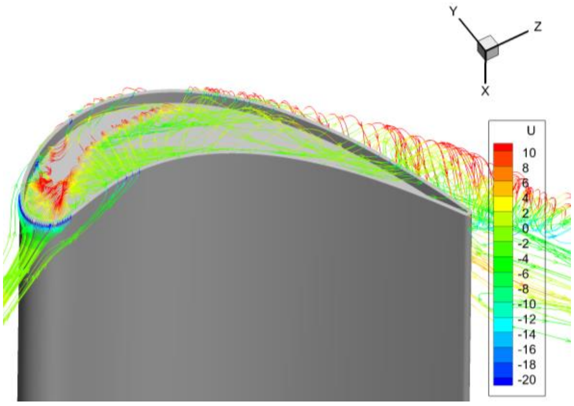


width=3mm

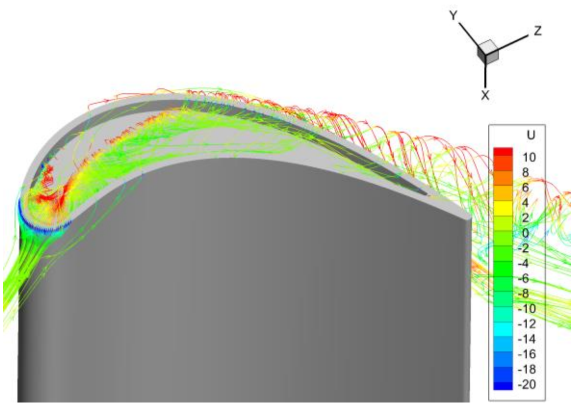


width=4mm

Figure 7 average vorticity result of squealer depth=4mm



width=1mm



width=2mm

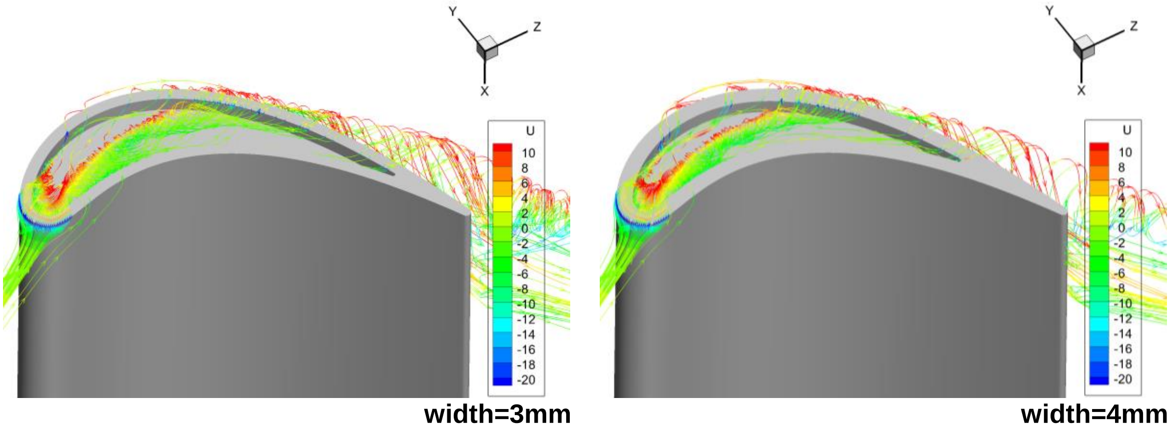


Figure 8 streamlines result of squealer depth=4mm

Heat transfer investigation:

These squealer tip cases including w1d1, w1d4, w4d4 are considered in this paper to understand the heat transfer on the blade tip with a squealer. The heat transfer coefficient is used to evaluate the thermal load on the blade tip. The heat transfer coefficient is defined as,

$$h=\frac{Q}{T_w-T_{aw}} \tag{1}$$

where the Q is wall heat flux, T_w is isothermal wall temperature, 300K and the T_{aw} is wall temperature.

When studying the effect of squealer depth and width on the tip leakage flow in the above ariticle, the thermal boundary condition of the blade is adiabatic. So in heat transfer investigation, a new numerical calculation case is added, in which the thermal boundary condition of the blade is 300K.

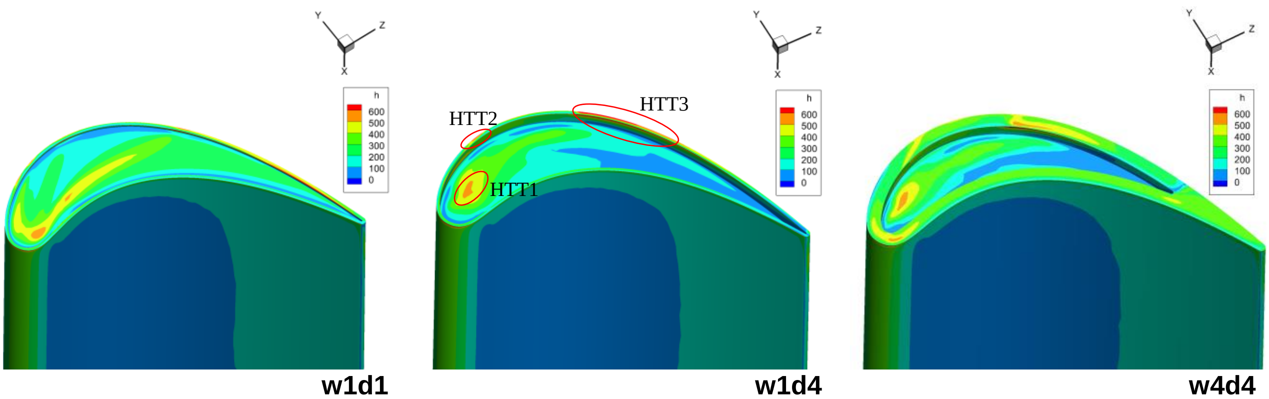


Figure 9 high thermal transport zones in w1d1, w1d4 and w4d4

The heat transfer coefficient of these three cases, w1d1, w1d4, w4d4, is shown in Fig. 9. There is a high thermal transport zone (HTT1) near the leading edge in the tip squealer and two high thermal transport zones (HTT2&HTT3) at the squealer rim on the suction side. According to analysis of aerodynamic, the reasons why these zones are high thermal transfer can be conjectured as follow:

Due to the existence of blade tip squealer, the tip leakage flow will generate two vortex structures in the tip squealer near the leading edge, evolving to downstream. So the flow situation is very complex at HTT1, resulting in HTT1 to a high thermal transport zone.

The vortex structures in the tip squealer mainly consists of two parts: one part is near the pressure side and another is near the suction side, and both of them flow out the tip clearance from the suction side. The regions that these two vortex structures leave the suction side and the regions of HTT2&HTT3 coincide basically, so it is very likely that these two vortex structures flow generate HTT2&HTT3.

CONCLUSION

The effect of squealer depth and width on the leakage flow and heat transfer are investigated by using numerical calculation with RANS in this work. Four squealer widths including 1, 2, 3, 4mm and four squealer depths including 1, 2, 3, 4mm for each width are considered, and the proportion of the tip leakage flow mass in the total inlet flow mass is obtained. The heat transfer of three tip squealers are analysised to find out high thermal transport zones. The conclusions are listed as follows:

Compared to the flat tip, turbine blade with a squealer tip can reduce the tip leakage flow mass, and then reduce the loss of the turbine. The reducing of the loss is different with different tip squealer. Through our research on the depth and width of the groove, it is found that as the depth of the groove increases, the tip leakage flow is effectively suppressed. However, as the width increases, the tip leakage flow increases instead. In general, bigger space of the squealer, less mass of the tip leakage flow.

There are two vortex structures in the tip squealer near the leading edge, which is enhanced as the space of the squealer increases. Therefore, the vortex structures in the squealer can reduce the tip leakage flow mass and the larger vortex structure has a better result in reducing the tip leakage flow mass than the smaller one can be conjectured.

In terms of heat transfer, there are three main high thermal transport areas on squealer tip. One is near the leading edge, caused by the complex flow situation, while the others are located at the suction side of the squealer rim, caused by the vortex structures leaving the tip clearance from the suction side.

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