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INFLUENCE OF DIFFERENT RIM WIDTHS ON LEAKAGE FLOW

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ABSTRACT

Three-dimensional simulations of squealer tip on GE-E3 blade with eight film cooling holes were numerically studied. The effect of the rim width and the blowing ratio on the blade tip flow was revealed. Numerical simulations were performed to predict the leakage flow and the tip heat transfer with the k-ε model. For the squealer tip, the depth of the cavity is the same and the width of the shoulder varies to form a narrow rim and a wide cavity, which can decrease the coolant momentum and the tip leakage flow velocity. This cavity contributes to the improvement of the cooling effect in the tip zone. To investigate the leakage flow influenced by the rim width, numerical simulations were made at four different models which have different rim widths of 0.58%, 0.89%, 1.16% and 1.74% axial chord (0.5mm, 0.77mm, 1 mm and 1.5mm, respectively) on both the pressure side rim and the suction side rim. From the simulated results, mathematical equations of mass flow rate of the leakage flow and the blowing ratio were proposed. With different rim widths, the effect of different global blowing ratios of M=0.5, 1.0 and 1.5 is investigated. In addition, calculation results of squealer tip and flat tip were compared. The simulation results are validated with some limited experimental data in the open literatures. [**Keywords:** film cooling; squealer tip; rim width; leakage flow]

INTRODUCTION

It is quite difficult to effectively cool the blade tip for gas turbines, because the tip is directly exposed to the leakage flow, which is driven by the pressure difference between the pressure and suction side of the blades. The leakage flow will accelerate in the tip gap and impinge on the blade tip, which is associated with high velocity, thin boundary layers and high temperatures, resulting in a high heat transfer region on the tip.

One common way is to extract some cooling air from the coolant passages by the film holes to protect the tip surface from hot leakage gas, which provides adequate cooling of the blade tips. Because film cooling is the most widely used and the most common technology in the cooling system of gas turbine blades, it is important to systematically study the discrete film holes associated blowing ratio and the tip structure which affect the film cooling effectiveness and heat transfer coefficient. The cooler air provides a protective film on the surface and prevents the blade surface from being exposed to the hot gas.

In order to reduce the tip leakage flow and the tip heat transfer, a squealer tip is a commonly used technique. The squealer tip as a labyrinth seal structure increases the flow resistance, reducing both the leakage flow and the tip heat transfer effectively. Because the tip clearance is very small, it is difficult to obtain the flow measurements in the tip region. Using three-dimensional numerical predictions to obtain a comprehensive understanding of the flow field and the associated heat transfer over the blade tip is convenient and cost-effective.

There have been numerous studies that described the heat transfer and flow with film cooling on the blade tip. Metzger et al. [1] and Chyu et al. [2] performed experiments and concluded that there is an optimum value of depth to width ratio for a given pressure difference across the gap, using cavities of varying depth to width and gap to width ratios. They found that although the heat transfer rate on the cavity floor is lower than that on a flat tip, the heat transfer reduction is offset by the high heat transfer on the downstream gap and by the additional heat transfer area of the side walls. They recommended that shallow cavities are preferred to get overall heat transfer reduction on the cavity wall.

For various chordwise sealing strips in a cascade, Bunker and Bailey [3, 4] experimentally investigated the heat transfer coefficient and leakage flow. To affect the main tip surface heat transfer, wooden strips were glued to the tip surface without the surface thermal response. Heyes et al. [5] studied the tip leakage flow field for a single suction side squealer and a single pressure side squealer. In an earlier study, Azad et al. [6] reported detailed heat transfer and leakage flow for both suction and pressure sides on the squealer tip, but the effects of different geometry arrangements were not in consideration. Dunn and Haldeman [7] investigated time-averaged heat flux data for a recessed tip with a platform of a transonic turbine blade. Ameri et al. [8] numerically studied the flow and heat transfer on the squealer tip which had a 2% and 3% cavity recess. Ameri et al.

[9, 10] numerically investigated the effect of tip clearance and casing (shroud) recess on heat transfer and stage efficiency for different recess depths. They also studied the blade tip heat transfer with a mean-camberline strip. Yang and Diller [11] reported local heat transfer coefficient on a squealer tip, using a single point measurement on the cavity bottom. Azad et al. [6, 12] and Kwak et al [13, 14] reported experimental results about both the flat and squealer tips of the GE-E3 first stage rotor blade, considering the effects of the turbulence intensity, the tip gap height and the tip geometry. To reduce the heat transfer and leakage flow, Acharya et al. [15, 16] numerically studied various leakage reduction strategies of blades tip, and flat and squealer tips with 7 film holes along the camber line. They found that the best configuration is the single suction side squealer tip. Yang et al. [17] systematically investigated different film hole arrangements on the flat and squealer tips, and they found the upstream and two-row film hole arrangements provided higher overall film cooling effectiveness than the camber line arrangement. They also found that the film cooling effectiveness was over-predicted, although it agreed with the experimental results very well.

Many geometric and aerodynamic parameters affect the blade tip heat transfer and flow. According to the literature review conducted above, we can find that there were few studies reporting the influence of the rim width on the squealer tip with the film cooling holes, although many researchers have investigated the GE-E3 blade tip both experimentally and numerically. Some studies on heat transfer on the squealer tip were done in Zhang's work [18] and previous work [19]. The similar blade tip model and boundary conditions are adopted to investigate the effects of the rim width on the tip leakage flow field. To get the relation between the blowing ratio and the mass flow rate of the leakage flow on squealer tip, different rim configurations under different blowing ratios of 0.5, 1 and 1.5 are used in this study. In addition, the effects on flat tip with different blowing ratios are in consideration. We attempt to explain the mechanism and composition of the leakage flow, to find a simplified formula to estimate decrease of mass flow rate of leakage flow with increasing blowing ratio.

NOMENCLATURE

h	Heat transfer coefficient, W/(m2K), $h=q/(T_w-T_\infty)$
M	Blowing ratio $=\rho_j V_j/\rho_\infty V_\infty$
Pt	Cascade inlet total pressure, Pa
P	Local static pressure, Pa
T	Temperature of flow, K
η	Film cooling effectiveness $=(T_{aw}-T_\infty)/(T_j-T_\infty)$
v	Cavity volume
d	Tip gap clearance
r	Rim width
Subscripts	
j	Coolant flow
∞	Mainstream flow
aw	Adiabatic wall
w	Wall
in	Inlet
s	Squealer tip
f	Flat tip
g	Flow blockage of the cavity

COMPUTATIONAL DETAILS

The models of squealer tip differ from those of flat tip, which have cavities (yellow zone) on the blade tips in Fig.1.

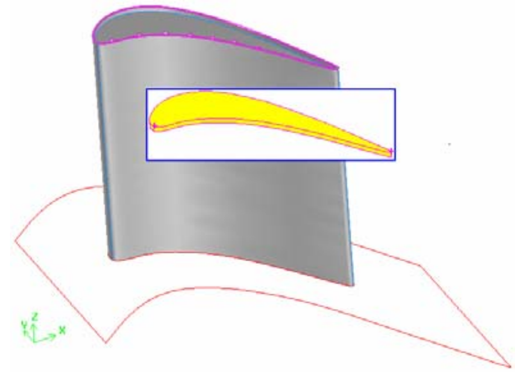


Fig. 1 Squealer tip structure

Acharya et al. [16] performed numerical simulation of film cooling on GE-E3 squealer tips with 8 film holes located in the vicinity of the pressure-side of the blade tip in stationary state. The blade is scaled up three times, which has an axial chord length of 8.61 cm, the aspect ratio of the span to the chord of 1.4, the blade leading edge pitch of 9.15 cm, and the tip clearance of 1.97 mm (1.5% of the blade span). The actual span height of the airfoils in the rig scale is 40.45 mm. The 8 film holes have the same diameter of 2.5 mm and a coolant delivery tube's length of 20 times film hole diameter. Figure 2 shows the computational domain.

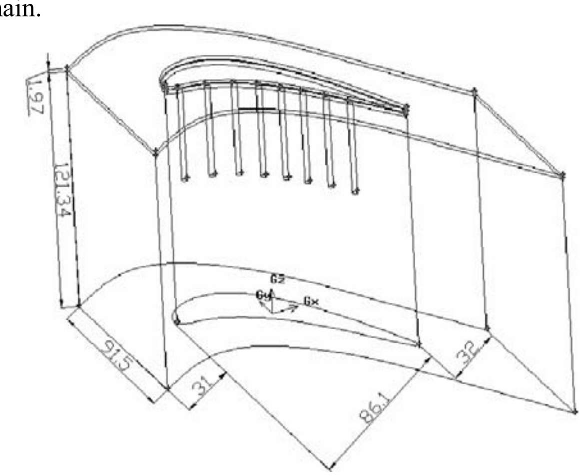


Fig. 2 Computational domain of squealer tip

In literature [20], compared to tip shoulder on pressure side or mid camber line, a squealer on suction side provides a better benefit. The leakage flow and heat transfer coefficient to the blade tip is mainly affected by the rim configuration in suction side, film cooling holes approach the pressure side. Moreover, the rim width in pressure side has a smaller variation range than that in suction side. The rim width in pressure side cannot increase after we have a rim width of 1.67mm, because it is near the film cooling holes. To investigate the flow and heat transfer influenced by the rim width, we change the rim width from 0.58% to 1.74% axial chord (from 0.5mm to 1.5mm), which is identical and uniform on both the pressure side rim and the suction side rim. In the present study, the models of both squealer tips and flat tips are investigated, and the four configurations are different in the rim width. For the squealer tip, the depth of the cavity is same (2% of the blade span) and the rim width varies to form a narrow rim and a wide cavity, which can decrease the coolant momentum and the tip leakage flow velocity. Different numerical model configurations are shown in Table 1, and the rim widths were axial chord of 0.58%, 0.89%, 1.16% and 1.74% (0.5mm, 0.77mm, 1 mm and 1.5mm, respectively). More information about the configuration of STb has been described in previous work [19], and we don't mention it in this paper redundantly. The other three models are shown in Fig. 3. Flat tip with film cooling holes were also shown. With different rim widths, the effect of different global blowing ratios of M=0.5, 1.0 and 1.5 is investigated.

Table 1. Configuration information

Name for short	Rim width (mm)	Tip structure
STa	0.5	squealer tip
STb	0.77	squealer tip
STc	1	squealer tip
STd	1.5	squealer tip
FT	X	flat tip

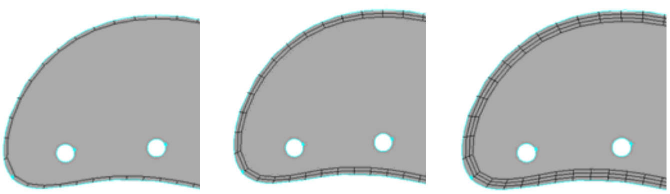

(a) (b) (c)

Fig. 3 Three rim widths of (a) STa, (b) STc and (c) STd

In this study, the simulations reported are performed using a commercial software called FLUENT 6.3.26. A finite volume method is used to discretize the compressible Reynolds-averaged Navier-Stokes equations, and every grid is generated using GAMBIT. The grid feature of the configurations is drawn in Fig. 4. The non-uniform structured grid is adopted on the pressure and suction surface. To acquire the accurate flow information, the grids are clustered in the near tip region (Fig. 4a). The unstructured grid is used on the tip gap region. The boundary layer mesh and the unstructured grid are used on the shoulder to generate the fine grid in the narrow region for the squealer tip. In the squealer cavity, 16 layer meshes are laid along the cavity depth direction (Fig. 4b).

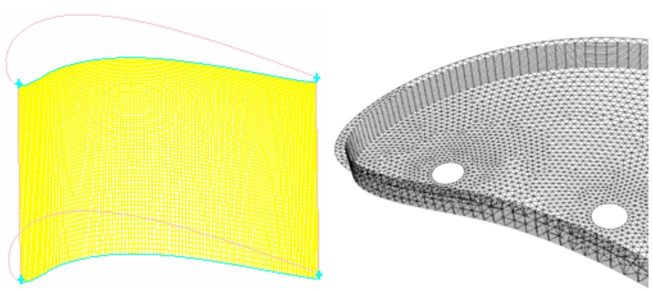

(a) (b)

Fig. 4 Grid feature of (a) pressure surface, (b) tip groove surface

Following is the boundary conditions (shown in Table 2): At the cascade inlet, the total temperature of 300 K, total pressure of 129.96 kPa, static pressure of 124.43 kPa, and an inlet flow angle of 32° are specified. The inlet turbulence intensity is taken as 9.7%. The static pressure is specified as 108.3 kPa at the exit. The inlet Mach number is 0.25. Three averaged film coolant exiting velocities are 42.5 m/s, 85 m/s, and 127.5 m/s which correspond to global blowing ratios of M=0.5, 1.0 and 1.5, respectively. When the film cooling effectiveness is calculated, the coolant temperature and the cascade inlet total temperature are set as 350 K and 300K, respectively. The adiabatic boundary condition is used on the blade, shroud surfaces, and the circular passage of the film hole. To calculate the heat transfer, the only difference is wall thermal boundary condition used to the blade surface with the wall temperature of 350 K, and the fluid temperature is the same as those of the cooling effectiveness calculation [21]. We must keep heat transfer in unidirectional transmission, so we give it a conveniently temperature of 350K.

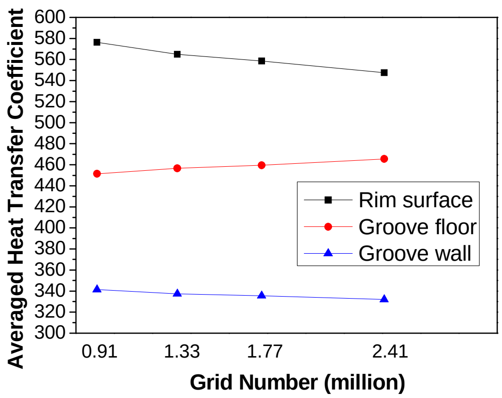
The coolant delivery tubes are assumed to be straight-through circular passages, and the influence of internal flow channel of the blade on the tip film cooling is neglected for simplification.

Table 2. Main boundary conditions

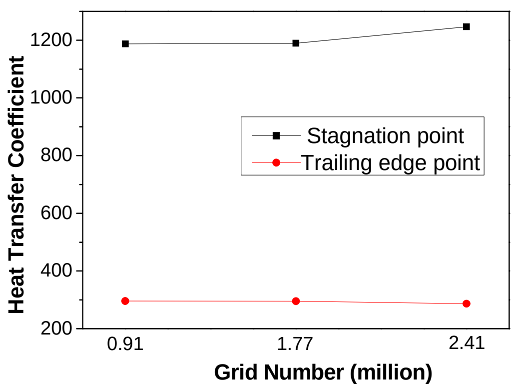
Zone	Type	Main Conditions
Mainstream Flow Inlet	pressure-inlet	Total Pressure: 129.96 kPa Static Pressure:124.43 kPa Temperature: 300 K Flow angle = 32 deg. Tu = 9.7% Ma = 0.25
Coolant Inlet	mass-flow-inlet	Mass Flow-Rate for a Single Hole (kg/s): 0.0002117505 (M=0.5) 0.0004235010 (M=1.0) 0.0006352515 (M=1.5) Temperature: 350 K
Mainstream Flow Outlet	pressure-outlet	Static Pressure:108.3 kPa Temperature: 350 K

In the present calculation, the mesh number for a film hole is about 10000. It has been validation that typical calculations are performed with about 1.5 million grid nodes for the flat tip blade and 1.8 million grid nodes for the squealer tip blade in literature [22]. We have studied the grid independence of STb (rim width of 0.77mm) on the averaged heat transfer coefficients of the rim surface, groove wall and groove floor in Fig. 5. We find the averaged heat transfer coefficient of the rim surface in a grid number of 1.77 million differs by 2% from the value corresponding to the finer grid number of 2.41 million. The averaged heat transfer coefficients of the groove floor and the groove wall in a grid number of 1.77 million differ by less than 1.5% from the values corresponding to the finer grid number of 2.41 million. So the final mesh selected is about 1.8 million to save time and resource.

Since the standard κ-ε turbulence model exhibits good agreement with the experimental result in predicting the blade heat transfer, the standard κ-ε turbulence model and the standard wall function method are adopted, and the density-based solver in FLUENT is selected in the present calculation [16]. All the cases presented are almost converged to residual levels of the order of 10^{-5} in Fig. 6a, and to less than 0.1% error in the mass flow rate between the inlet and the exit of the computational domain. For example, the mass flow rate of the leakage flow with the blowing ratio of 0.5 also has an insignificant error which is shown in Fig. 6b. We have a plot for the walls in terms of y+ (M=0.5) in Fig. 7.

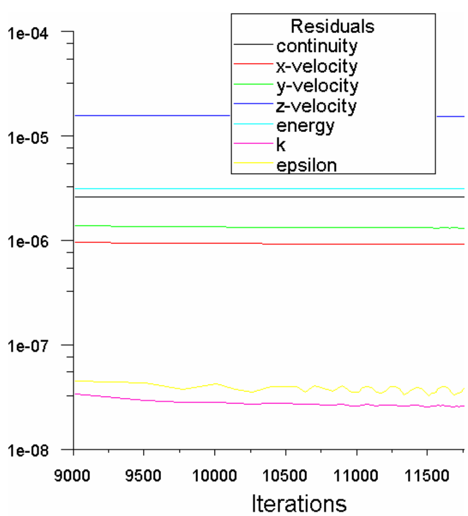


(a)



(b)

Fig. 5 Grid independence of (a) STb and (b) STc



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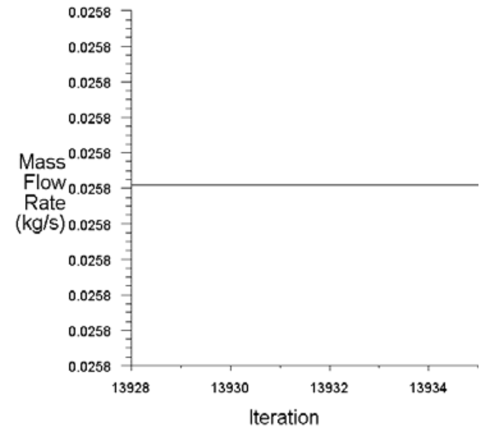


(b)

Fig. 6 Residual levels of (a) all zone, (b) the mass flow rate of the leakage flow (M=0.5)

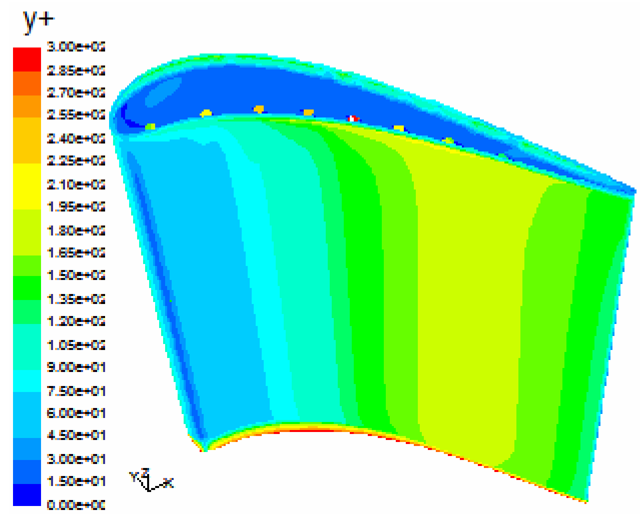
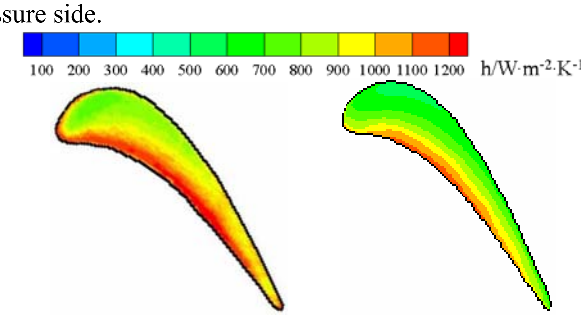


Fig. 7 Walls y+ (M=0.5)

RESULTS AND DISCUSSION

We use the experimental result of flat tip model in literature [14] to validate current calculating results in Fig. 8. It can be seen that the numerical results match with the experimental data. Because the sudden contraction of the flow path makes the leakage flow velocity suddenly increase, the high heat transfer coefficients value appears at the tip near the pressure side.



(a) (b)

Fig. 8 Comparisons of h value on flat tip (M=1.0) (a) experimental result [14], without film holes, (b) present simulation, without film holes

The tip leakage flow is driven by the pressure gradients from the pressure side to the suction side. A higher pressure ratio corresponds to a lower static pressure and a higher velocity, so the heat transfer coefficient is higher. Figure 9 shows calculating result of different configurations (STa, STb, STc and STd) on shroud, and the blowing ratio is 1. Compared with the numerical solution for the camber arrangement of film holes in Yang et al. [17], the present study exhibit a similar behavior. However, as a result of film holes located near the pressure side, the flow from film holes is mixed earlier, so there is little difference after it flows by the rim of the suction side. A zone of the low pressure ratio which is 1.16 emerges in the leading edge, and all of the models outside the suction side have a higher pressure ratio. There are some zones of high pressure ratio of STa near the holes which is larger (pressure ratio is 1.2) than those of STc and STd. We can find the pressure ratio above the blade tip gets smaller with increasing the rim width. Although the flow field of STc is similar to that of STd, the pressure ratio zone of 1.18 of STd is slightly larger than that of STc. It seems that the effect on the pressure ratio is not observable with the rim widening from 1mm to 1.5mm.

For the squealer tip, the leakage flow is complex inside the squealer cavity. The computed leakage flow path lines is shown in Fig. 10a. The gas flow into tip gap from the pressure side will reattach on the blade tip and passing by the suction side. The leakage flow passes this vortex and impinges onto the groove bottom. Fig. 10b shows path lines of the coolant. Some cooling air flow toward the suction side, rebound to the pressure side and the trailing edge; or else the coolant flow by the squealer rim of the suction side straightly. Some of them near the trailing edge roll up to leave the cavity to mix with the leakage flow. A leakage vortex occurs around the tip suction side, and it interacts with the passage vortex, changing the location and dimension of the passage vortex [19].

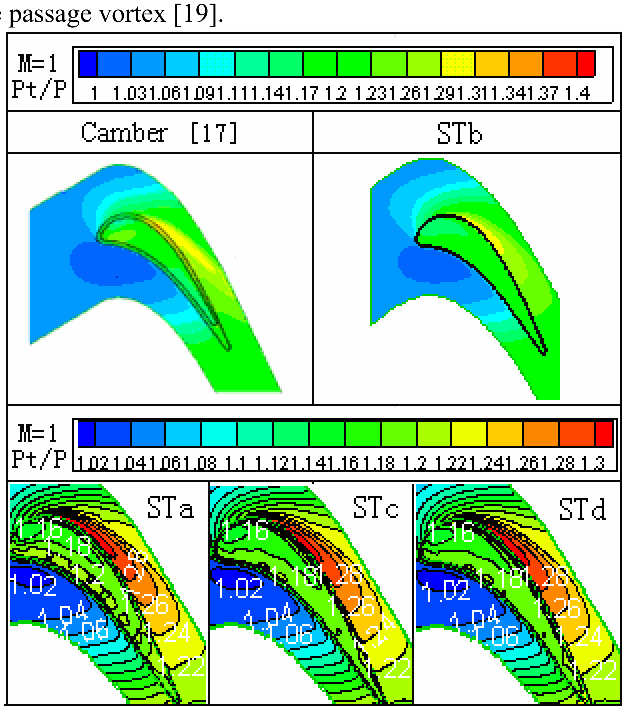
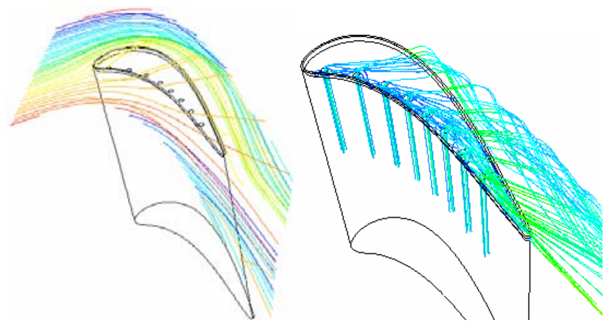


Fig. 9 Comparisons of Pt/P value on shroud with Yang et al. [17], different configurations(STa, STb, STc and STd), M=1



(a) (b)

Fig. 10 Path lines of (a) tip leakage flow on squealer tip and (b) coolant

It is a measure of the leakage flow that computing the mass flow rate through the tip gap along the mean camber line of the blade[23]. We compute the leakage flow through the tip gap along pressure side between the shroud and the blade tip to get the mass flow rate easily, because the two inner faces are closed in Fig. 11. The cavity on the squealer tip offers resistance to the leakage flow. The structure of the squealer tip blade (shown in Fig. 12a) consists of the flat tip blade (shown in Fig. 12b) on which there is a cavity (shown in Fig. 12c). From the leading edge, the cavity shape with yellow grid is further shown in Fig. 12b. We propose that the squealer tip blade is the flat tip blade with a cavity, and the squealer tip structure includes the flat tip structure. We can analyze the leakage flow with this theory. The mass flow rates of the leakage flow in different blade tip models (STa, STc, STd and FT) with different blowing ratios (0.5, 1 and 1.5) are shown in Fig. 13. The relation is presented with the following equation:

$$q_s = q_f - q_g \tag{1}$$

The flow in tip clearance driven by the pressure gradients from the pressure side to the suction side is called the tip leakage flow. q_s is the leakage flow rate of the squealer tip, and q_f is that of the flat tip. From Fig. 13, q_g is the difference between q_s and q_f , which is caused by the flow blockage of the cavity. We think the squealer tip leakage flow rate is equal to flat tip Leakage flow rate subtracts groove blockage flow. When the rim gets enough width, the squealer tip structure is changed into the flat tip structure.

Yang[23] did research on the leakage flow of squealer tip with the blowing ratio of 1 in Fig. 14a. In Yang's results, three different tip gap clearances of 1%, 1.5%, and 2.5% were investigated, and the configuration with 1.5% gap clearance is similar with the configuration in this paper. We use the ratio (q_s/q_{in}) of tip leakage mass flow to inlet mass flow to neglect inlet influence. We find the mass flow ratios of current simulated results are close to Yang's results in Fig. 14b. We keep the same geometrical structure to compare the simulation results with Yang's, and the results of current model can match well with the Yang's results.

The effect of groove depth for a high-pressure turbine blade was investigated in Bunker and Bailey's work [24]. They found that a deeper tip groove results in reduced heat transfer to the tip through changing tip clearance-to-groove depth ratios (groove depths of 1.02, 1.78, 2.54, and 3.05mm). Yang D. L. [22] also investigated several cases with different groove depth-blade span ratios of 1%, 2%, 3%, and 4%. The cavity volume changes with the rim width and the cavity depth. Different groove depths imply the cavity has different volumes in fact. In this study, we do research on different rim widths. It also changed the cavity volume. So we think the mass flow rate of leakage flow must concern the cavity volume.

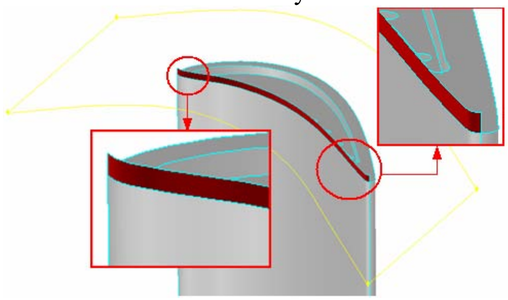


Fig. 11 Computing inner face of the mass flow rate (red surface)

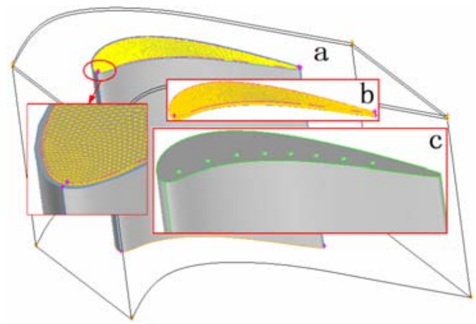


Fig. 12 Structure of (a) squealer tip blade, (b) cavity, (c) flat tip blade

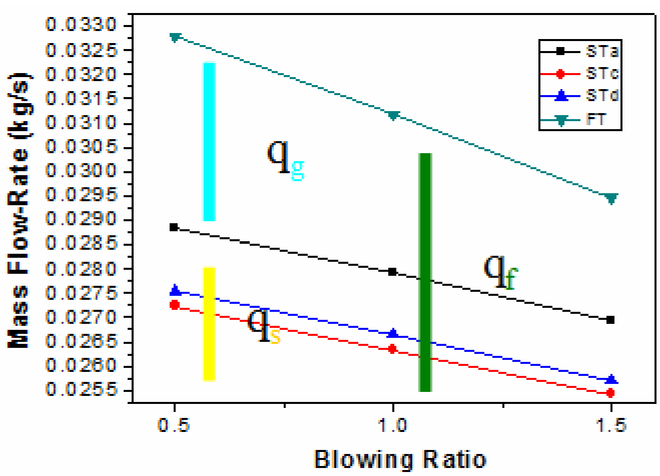
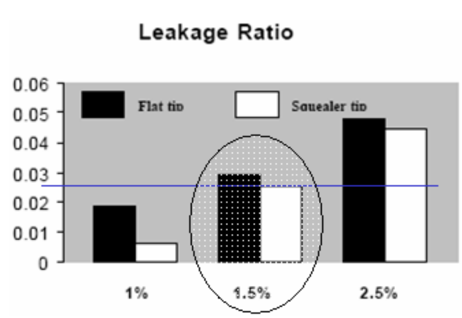
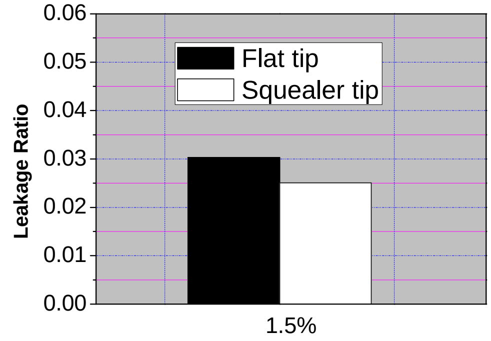


Fig. 13 The mass flow rates of the leakage flow in different blade tip models (STa, STc, STd and FT) with different blowing ratios (0.5, 1 and 1.5)



(a)



(b)

Fig. 14 The mass flow ratios of the leakage flow in (a) Yang's work [23] and (b) Current simulation

From the above, the leakage flow of squealer tip is affected by the blowing ratio, the tip gap clearance and the cavity volume in Fig. 15. The cavity volume (v) is affected by the rim width (r) and the cavity depth (h). There is an equation as follows:

$$q_s / q_{in} = S(M, v, d) \tag{2}$$

From their physical significance, the effects of the blowing ratio the cavity volume and the tip clearance on the leakage flow are independent. We just want to analyze the influence of the blowing ratio, so the cavity volume and the tip gap clearance are fixed. And then $S(M,v,d)$ is a function of the blowing ratio:

$$q_s = S(M, v, d) * q_{in} = G(M) \tag{3}$$

A simple observation of current figures would yield the system law as linear in Fig. 16. So the equation is deformed as follows:

$$k = G(M) / M \tag{4}$$

$$k = \tan X \tag{5}$$

k parameter acts as a proportionality factor and gradient, while X parameter acts as angle of inclination. Whatever the k parameter or the X parameter, we want to substitute a constant value for it. We get the k and X values from simulated results in Table 3. k parameter is average value from the blowing ratio of 0.5 to 1.5.

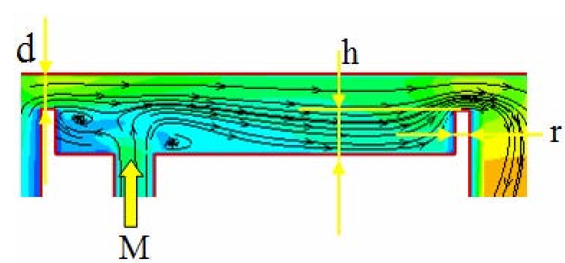
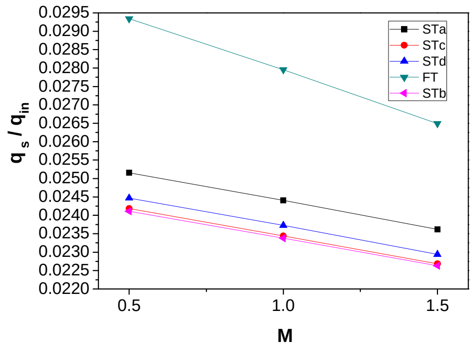
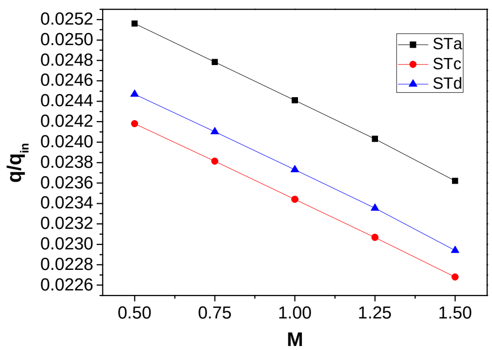


Fig. 15 Influence factors of leakage flow

The model of STb presents the least mass flow ratio of the leakage flow in Fig. 16a, and X parameter has the least value in the meanwhile. To obtain the least leakage flow, we may find which rim width is the best through the model with the least X value. From the number axis analysis in Fig. 17, we can find the optimal model with the least leakage flow should be the model with the rim width of 0.77~1 mm.



(a)



(b)

Fig. 16 The mass flow ratios of the leakage flow of (a) flat tip and squealer tip, and (b) squealer tip with different blowing ratios (0.5, 0.75, 1, 1.25 and 1.5)

Table 3. k value analysis

Model	k (M=0.5~1.5)	$X=\tan^{-1}k$
STa (r =0.5mm)	-0.001903297	-0.109050753
STb (r =0.77mm)	-0.001774575	-0.101675551
STc (r =1mm)	-0.001804948	-0.103415790
STd (r =1.5mm)	-0.001839763	-0.105410536

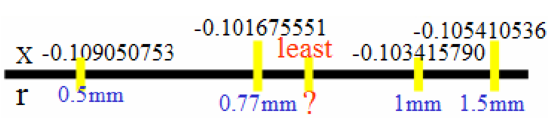


Fig. 17 Number axis analysis

Previous results [19] also support the above assumptions. In previous work, we used the asymmetric structure (WHb and WHc) in Fig. 18. The rim width on pressure side in the tip-gap region is different from the one on suction side ($r_1 \neq r_2$), which results in different and irregular volume. However, the mass flow ratios of the leakage flow have the similar gradient as the model of STb and changing cavity volume does not make the k parameter change obviously in Fig. 19. So we have proved that the effect of the blowing ratio on the leakage flow is irrelevant to the effect of the cavity volume, and the mass flow ratio of the leakage flow which is caused by increasing the blowing ratio may be also seen as a constant in different rim width structures.

From Table 3, X value is close to -0.1, so we can use equation as follows to simplify the computing of the mass flow ratio of the leakage flow:

$$k = -\tan 0.1 \qquad (6)$$

When the blowing ratio increase ΔM under the volume and the tip gap clearance is decided, the mass flow rate of the leakage flow can be estimated by combining and solving (3), (4) and (6). It is the resulting equation as follows:

$$\Delta q_s = k\Delta M = -\tan 0.1 * \Delta M \qquad (7)$$

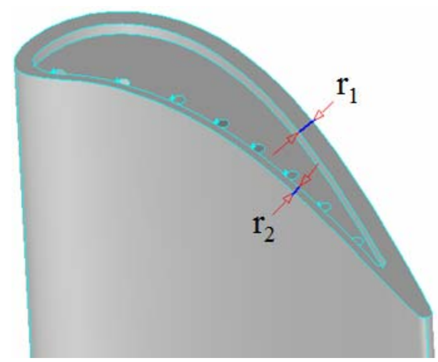


Fig. 18 The model of previous work [19]

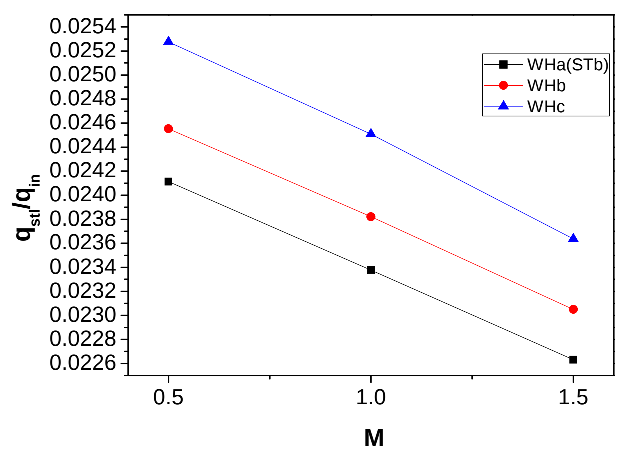


Fig. 19 The mass flow ratios of the leakage flow in different rim widths (STb, WHb and WHc) and different blowing ratios (0.5, 1 and 1.5) [19]

We can use the equation (7) to estimate the increase of mass flow ratio of the leakage flow. All of the results and equations presented are valid only for the E3 blade tip and still have to be confirmed for other airfoil geometries.

CONCLUSIONS

To investigate the leakage flow influenced by the rim width, numerical simulations were made at four different models which have different rim widths of 0.58%, 0.89%, 1.16% and 1.74% axial chord (0.5mm, 0.77mm, 1 mm and 1.5mm, respectively). All of the conclusions are applicable to current research range of geometries evaluated, and several major conclusions are following:

1) We can find the pressure ratio above the blade tip gets smaller with increasing the rim width. It seems that the effect on

the pressure ratio is not observable with the rim widening from 1mm to 1.5mm.

2) We propose that the squealer tip blade is the flat tip blade with a cavity, and the squealer tip structure includes the flat tip structure. The relation is presented with the following equation: $q_s = q_f - q_g$.

3) It seems that the leakage flow of squealer tip is affected by the blowing ratio, the tip gap clearance and the cavity volume. So we make current supposition into a equation as follows: $q_s / q_{in} = S(M, v, d)$

4) It was predicted the model of the smallest mass flow rate of leakage flow is the model with rim thickness of 0.77-1mm in current research.

5) The effect of the blowing ratio is irrelevant to the effect of the cavity volume, and it seems that the mass flow ratio of the leakage flow which is caused by increasing the blowing ratio is linear.

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