Effects of casing motion on tip leakage flow and film-cooling effectiveness of squealer tips

Fengna CHENG*, Haiping CHANG*, Jingyang ZHANG** and Jingzhou ZHANG*
*College of Energy and Power Engineering, Nanjing University of Aeronautics and Astronautics
No. 29, Yudao Street, Nanjing, Jiangsu 210016, China
E-mail: cphppe@nuaa.edu.cn
**College of Astronautics, Nanjing University of Aeronautics and Astronautics
No. 29, Yudao Street, Nanjing, Jiangsu 210016, China

Received: 9 May 2017; Revised: 3 August 2017; Accepted: 12 September 2017

Abstract
A comparative investigation of leakage flow and film-cooling characteristics on different squealer tips is conducted under the conditions of stationary and moving casing. Firstly, the numerical method used in the calculations has been verified. Then, the influences of film-hole arrangement, squealer height and blowing ratio are studied in detail. Three different film-hole configurations are taken into consideration. For type-A and type-B, the film holes are arranged in one row, but located on the middle-camber line and near the suction side squealer, respectively. Type-C places two rows of 7 film holes at the leading edge and one row of 6 film holes at the middle chord region. The results show that the relative motion of the casing leads to the formation of a scraping vortex near the casing, which not only compresses the cavity vortex, but also acts as a pair of labyrinths with the cavity vortex to the leakage flow. The effect of the moving casing on the film-cooling effectiveness of squealer tip varies with the film-hole arrangements. When the casing is moving, the tip leakage loss increases with the decrease of the rim height because the blockage effect of the vortexes in the cavity decreases. Nevertheless, type-C obtains the best tip cooling performance in both cases of stationary and moving casing.

Keywords: Squealer tip, Tip leakage loss, Film-hole configuration, Film-cooling effectiveness, Casing motion

1. Introduction

To avoid the friction between the high pressure turbine blades and the casing, there is usually a small gap between them, which is called tip clearance gap. A part of the mainstream near the tip flows through the tip clearance gap under the pressure difference between the pressure side (PS) and the suction side (SS) of the blade, known as tip leakage flow. The leakage flow not only reduces the efficiency and output work of the turbine, but also makes the tip of blade expose to high temperature. The common method to reduce the tip leakage loss and protect the blade tip is to use the cooled squealer tip. Previously, most researches about the tip leakage flow and film-cooling characteristics of squealer tips are conducted with stationary casing, but there is relative motion between the blade and casing in the reality. So it is significant to investigate the effect of the relative motion on the tip leakage loss and cooling performance of squealer tips.

Many studies referring to the relative motion mainly focus on the flat tip (Srinivasan and Goldstein, 2003; Rhee and Cho, 2006; Zhang et al., 2010; Palafox et al., 2012). Compared to the flat tip, relatively fewer publications investigate the effects of relative motion on the squealer tip. Yang et al. (2006) performed the simulations for the nonrotating and rotating middle-camber line film-hole configuration under the engine conditions. It was found that the rotation decreases the film-cooling effectiveness on the plane tip, but only slightly affects the cooling characteristic of the squealer tip. However, the rotation significantly increases heat transfer coefficient on the shrouds. Yang et al. (2010) numerically analyzed the effect of the relative motion of the casing, the centrifugal force, and the Coriolis force on the blade tip leakage flow and heat transfer. They concluded that the main effect of the rotation on the tip leakage flow and
heat transfer results from the relative motion of the shroud, especially for the squealer tip blade. Acharya and Moreaux (2012) studied the influence of the blade rotation and relative motion between the blade tip and shroud numerically. Their results showed that the magnitude of the rotational forces is relatively small compared to the relative motion effects. Geometric effects are also of importance with the lower tip clearance reducing leakage flow and allowing the tip coolant to migrate towards the SS with relative motion. Zhou (2015) illustrated the scraping vortex caused by the motion of the casing for the first time and pointed out that the height of the squealer has a great influence on the vortex in the cavity. Zhang et al. (2015) compared the film-cooling effectiveness on squealer tip with the trailing rim wall cut between the stationary and moving shroud cases. It was revealed that blade movement poses significant negative impacts on film-cooling effectiveness.

To date, the film cooling on the squealer tip have been investigated in a number of publications. Acharya et al. (2002) denoted that the flow inside the squealer cavity exhibits streamwise directed flow, which alters the trajectory of the coolant jets and reduces their effectiveness. Yang et al. (2006) explored the effect of three film-hole arrangements on the squealer tip by numerical method. It was indicated that the upstream and two-row arrangements provide a better tip cooling performance than the middle-camber line arrangement, especially at high blowing ratios. Wang et al. (2012) emphatically studied the film-cooling characteristic on the squealer tip with different rim widths under various blowing ratios. They concluded that as the rim width increase, the low film-cooling effectiveness region at the leading edge becomes smaller, and the higher film-cooling effectiveness emerges from the trailing edge. Park et al. (2014) put emphasis on the cooling characteristic of the inner rim surfaces of a blade with a squealer rim, and suggested a proper cooling system to enhance the film-cooling effectiveness on the squealer tip. More recently, the detailed numerical investigations of film-cooling effectiveness of squealer tips with three different film-hole configurations were reported by Zhang et al. (2015). The results showed that the film-hole arrangement with three dust purging holes along the middle-camber line and six film holes near the suction side rim obtains the best cooling performance with the same coolant mass flow.

In summary, although there are some researches addressing on the influence of relative motion of the casing on the squealer tip, they mainly concern with the heat transfer characteristic rather than the aerodynamic and cooling performance. In addition, the film-hole arrangements are a pivotal issue for the cooling performance of the blade tip. Therefore, the aim of this paper is to explore the effect of casing relative motion on the tip leakage loss and cooling characteristic of the squealer tip with three film-hole arrangements. The influences of squealer height and blowing ratio are also investigated in detail. The findings of the current study can aid engineers to design more efficient turbine blades.

**Nomenclature**

\( C \): Chord length (mm)

\( C_p \): Pressure coefficient

\( C_{p-\text{profile}} \): Profile loss coefficient

\( C_{p-\text{tip}} \): Tip leakage loss coefficient

\( C_{p-\text{total}} \): Total loss coefficient

\( C_x \): Axial chord length (mm)

\( d \): Diameter of film-cooling hole (mm)

\( H \): Height of the squealer (mm)

\( M \): Blowing ratio

\( m \): Mass flow rate (kg/s)

\( P \): Pressure (Pa)

\( S \): Cascade pitch (mm)

\( T \): Temperature (K)

\( t \): Tip clearance gap (mm)

\( V \): Velocity (m/s)
2. Calculation Details

2.1 Computational model

The main object of this paper is the first stage blade of GE-E3 high pressure turbine (Timko, 1984). The blade is straight, which is formed by top profile of the real blade, as shown in Figure 1. The span and axial chord of the blade are 41mm and 28.7mm, respectively. The squealer has a width of 0.7mm (1.7% span) and two different heights of $H =$1.7mm(4.1% span) and 1mm(2.4% span). The tip clearance gap is chosen as $t = 1mm(2.4\% \text{ span})$ in all simulations. The diameter and length of the film holes are 0.423mm and 1mm, respectively. The parameters are listed in Table 1.

![Fig.1 Blade parameters and shroud tip definition.](image)

| Parameter                        | Value  |
|----------------------------------|--------|
| Chord length, $C$                | 38.2mm |
| Axial chord length, $C_x$        | 28.7mm |
Cascade pitch, $S$ & 30.5mm \\
Height of the squealer, $H$ & 1.7 mm/1mm \\
Width of the squealer, $W$ & 0.7mm \\
Aspect ratio & 1.4 \\
Inlet flow angle, $\alpha_1$ & 32.01° \\
Blade exit angle, $\alpha_2$ & 65.7° \\

The computational domain is shown in Figure 2. It is a fluid domain formed by a single blade according to the periodic feature of the turbine blade. The entrance and exit length of the domain are $1C_x$ and $1.5C_x$, respectively. Since the leakage flow is confined near the blade tip (Lee et al. 2010), only 57% span is contained in the domain (Zhou, 2011). The back and front surfaces of the domain are set as periodic, the left and right surfaces are set as inlet and outlet, and the bottom surface is specified as symmetry. The coolant plenum is installed under all the film holes. It has an external profile as the cavity and a height of 3mm.

To compare the film-cooling performance on the blade tip, 3 different film-hole configurations are investigated in the current study, as presented in Figure 3. For all the cases, the total number of film holes is the same, i.e. 13. Type-A is the most common configuration in the real turbine, which arranges all the film holes along the middle-camber line and has a hole-to-hole pitch of $5d$, as shown in Figure 3(a). For type-B, the film holes are located close to the SS squealer. Specifically, all the film holes have a distance of $2.5d$ to the suction side squealer (SS squealer) and a distance of $5d$ to the adjacent hole, as seen in Figure 3(b). The last configuration of type-C places two rows of 7 film holes (3 film holes in the first row and 4 film holes in the other row) at the leading edge and one row of 6 film holes in the middle chord region, as shown in Figure 3(c). It is proposed according to pre-simulate of the tip leakage flow field of squealer tip without film-cooling under the condition of stationary casing. And our study (Cheng et al. 2017) has described the structural parameter of type-C in detail. Figure 4 presents the definition of the separation line. Table 2 is the summation of the structural parameter of the film-hole arrangements.

![Fig.2 Computational domain.](image)

![Fig.3 Different film-hole arrangements.](image)
Fig. 4 Limiting lines on the squealer tip (Cheng et al., 2017).

Table 2 Main differences between film-hole arrangements.

| Type | Feature                        | Hole-to-hole pitch | Distance between hole and SS squealer | Distance between hole and separation line |
|------|--------------------------------|--------------------|--------------------------------------|------------------------------------------|
| A    | Single row at middle-camber line | $5d$               | —                                   | —                                          |
| B    | Single row close to SS squealer  | $5d$               | $2.5d$                              | —                                          |
| C    | Leading edge: two rows          | $2.5d$             | —                                   | $0.5d$                                   |
|      | Middle chord: single row        | $5d$               | $2.5d$                              | —                                          |

2.2 Computational grids and method

The grids used in the simulations are all generated by software Gambit. Before drawing the mesh, the computational domain is divided into five parts, namely, the mainstream, the tip clearance, the cavity, the film holes and the coolant plenum. On account of the boundary layer, the mesh near all the surfaces (the casing, the blade, the film holes and the coolant plenum) are refined. Figure 5(a) shows the details of mesh for a specific blade tip configuration.

A grid independence test has been carried out to eliminate the influence of the grids on the computational result, in which four mesh numbers (986,125, 1,839,513, 3,317,417 and 4,214,796) are studied, as shown in Figure 5(b). It is observed that the averaged film-cooling effectiveness on the blade tip almost does not change when the number reaches 3,317,417. So the total number of about 3.3 million is chosen as the total grid for all the calculations. There are 35 mesh layers generated in the spanwise direction within the tip clearance gap, and the averaged wall $y^+$ is 2.16.

In this paper, the CFD software ANASYS FLUENT is used to solve the compressible Reynolds averaged N-S equations in all simulations. Considering the turbulence model, the standard $k-\varepsilon$ model and enhanced wall treatment stand out due to the good prediction and convergence. This model has been adopted in many researches (Wang et al., 2012; Yang et al., 2002). The SIMPLIC algorithm is selected as the pressure-velocity coupling scheme and the second order upwind is chosen as the spatial discretization method. The ideal gas is selected as the working fluid. The properties of the air, i.e., specific heat, thermal conductivity and dynamic viscosity, are set as piecewise-polynomial, kinetic-theory and Sutherland, respectively. The convergence criterion of the continuity, momentum and heat transfer equations are all set to be $10^{-6}$.

The accuracy of the numerical method is validated by applying it to a simulation of an experimental case (Kwak and Han, 2003). The computational model and boundary conditions are the same as the experiment. The model is formed by a 3 times scaled GE-E3 blade. The blade has an axial chord length of 8.61 cm, a blade leading edge pitch of 9.15 cm, and a tip clearance of 1.97 mm (1.6% span). The computational domain includes only one single blade, and the periodic conditions are specified to the pitch direction surfaces. The inlet of the computational domain is set as a
pressure–inlet with a total pressure (126.9kPa), a total temperature (300K) and an inlet flow angle (32.01°). A static pressure of 102.7kPa is specified at the outlet of the computational domain. All the surfaces of the blade are set as no-slip adiabatic conditions. The inlet of the coolant plenum is set as a mass-flow-rate-inlet determined according to the global blowing ratio $M=1$ with a temperature of 328K. Here the blowing ratio is defined as $M = \frac{\rho_{c}V_{c}}{\rho_{\infty}V_{\infty}}$, which will be explained in subsection 2.4 in detail. Figure 6 (a) and (b) compare the pressure ratio distribution at 97% span on the blade surface and the tip film-cooling effectiveness of computational results with the experimental data. It can be seen from Figure 6(a) that the CFD can well predict the pressure parameters. In addition, the distribution and value of the film-cooling effectiveness on the squealer tip predicted by the calculation also agree well with the experimental data, as shown in Figure 6(b). The high film-cooling effectiveness region is located between the middle-camber line and the pressure side squealer (PS squealer) (marked by 'A') since the coolant is drawn to the PS of the blade by the cavity vortex. Besides, the remaining region on the tip has almost no film coverage (marked by 'B'). Thereby, it can be concluded that the computational method is feasible.

2.3 Boundary conditions

The boundary conditions of numerical simulations are listed in Table 3. To simulate the real engine condition, the
inlet of the mainstream is set as a total pressure of 303kPa and a total temperature of 683K, and the outlet of the computational domain is given a static pressure of 122.7kPa in accordance with the open data of Timko (1984). The back and front surfaces are periodic boundaries, as shown in Figure 2.

Considering tip cooling, the coolant with a temperature of 344K is applied to the inlet of the coolant plenum. Then it is injected into the cavity through the film holes. To obtain the desired blowing ratios (0.5, 1, 1.5 and 2), four mass flow rates are specified to the plenum inlet. On the other hand, each solid surface in the computational domain is set as no-slip adiabatic wall condition for the predicting of film-cooling effectiveness.

In terms of the effects of relative casing motion, a speed is given to the casing while the blade remains stationary. This treatment is based on the study of Zhou (2015). In the paper of Yang et al. (2010), three different cases are compared: (1) rotating blade and stationary casing; (2) stationary blade and rotating casing; (3) stationary blade and stationary casing. The results indicated that the relative motion plays the main role in the rotation effect, while the influence of the centrifugal force and the Coriolis force is limited. Therefore, this paper focuses on the influence of the relative motion of the casing.

| Parameter                             | Value    |
|--------------------------------------|----------|
| Inlet total pressure, $P_{in}^*$      | 303 kPa  |
| Inlet temperature, $T_{in}$          | 683 K    |
| Exit average static pressure, $P_{out}$ | 122.7 kPa|
| Overall blade pressure ratio          | 2.47     |
| Blowing ratios, $M$                   | 0.5, 1, 1.5 and 2 |
| Coolant temperature, $T_c$           | 344 K    |
| Rotation speed                       | 8450rpm  |

### 2.4 Parameter definitions

In the current study, the pressure coefficient is defined in the following way:

$$ C_p = \frac{(P_{in}^* - P)}{(P_{in}^* - P_{out})} $$  \hspace{1cm} (1)

where $P_{in}^*$ is the total pressure at the mainstream inlet, $P$ indicates the local static pressure, and $P_{out}$ presents the static pressure at the mainstream outlet.

The total-pressure loss coefficient is defined as:

$$ C_{pt-total} = \frac{(P_{in-ref}^* - P^*)}{(P_{in-ref}^* - P_{in-ref}^*)} $$ \hspace{1cm} (2)

where $P_{in-ref}^*$ is a reference total pressure, and $P^*$ is the local total pressure.

The $P_{in-ref}^*$ presents a combination total pressure of the mainstream and coolant, defined as:

$$ P_{in-ref}^* = (m_{in} \cdot P_{in}^* + m_c \cdot P_{in,c}^*) / (m_{in} + m_c) $$ \hspace{1cm} (3)

where $m_{in}$ and $m_c$ are the mass flow rate at the inlet of mainstream and coolant, respectively; $P_{in,c}^*$ is the inlet total pressure of the coolant.

According to the definition of tip leakage loss in the paper of Zhou (2011), the profile loss must be subtracted from the total loss near the tip. So it can be presented as:

$$ C_{pt-tip} = C_{pt-total} - C_{pt-profile} $$ \hspace{1cm} (4)

where $C_{pt-profile}$ is the profile loss coefficient, which is calculated at the midspan of the mainstream outlet.

The blowing ratio is defined as:
where \( \rho_c \) and \( V_c \) are the density and velocity of coolant at the inlet of the film hole, respectively; \( \rho_\infty \) and \( V_\infty \) indicate the density and velocity of the mainstream inlet, respectively. For the overall blowing ratio, the values of density and velocity are averaged at the corresponding position.

The film-cooling effectiveness is defined as:

\[
\eta = \frac{T_\infty - T_{aw}}{T_\infty - T_c}
\]

where \( T_\infty \) is temperature at the mainstream inlet, \( T_{aw} \) represents the adiabatic wall temperature, and \( T_c \) is the temperature at the coolant inlet.

3. Results and discussions

3.1 Tip leakage flow field

Figure 7 shows the tip leakage flow field on the uncooled squealer tip with a squealer height of \( H=1.7\text{mm} \) (4.1\%span) for (a) stationary casing and (b) moving casing. According to the study of Cheng et al. (2017), the film-cooling characteristic on the squealer tip is closely related to the leakage flow in the cavity, which is formed by the leakage flow entering the tip clearance from the leading edge of SS (red streamlines) and leading edge of PS (blue streamlines). In order to show the effect of the rotation more clearly, the leakage characteristic within the cavity is analyzed in detail. Above all, the impingement area of leakage flow on the cavity bottom is marked by ‘C’. Furthermore, the leakage flow is divided into 4 parts. Firstly, a part of the red streamlines forms a vortex near the PS squealer on account of the back flow at the leading edge, as marked by ‘s1’. Secondly, the remaining red streamlines leave the cavity directly from the following SS squealer after the impingement, as marked by ‘s2’. The third part also refers to the leakage flow leaving the cavity directly, but it enters the tip clearance from the leading edge of PS, as marked by ‘s3’. The last part of the leakage flow forms a vortex in the cavity, so the streamlines flow back to the PS, as marked by ‘s4’. All the leakage flow mixes with the mainstream outside the SS of the blade, giving a birth to the tip leakage vortex.

Compared to the stationary casing (Figure 7 (a)), the leakage flow field changes a lot when the casing is provided with a relative motion (Figure 7 (b)). Obviously, the impingement area ‘C’ moves towards the middle chord and the PS of the blade. Specifically speaking, the vortex formed by ‘s1’ becomes larger, the location that ‘s2’ and ‘s3’ leaving the tip clearance moves backward, and the capture zone of ‘s4’ at the leading edge becomes smaller. In addition, the relative motion of the casing has an inhibitory effect on the leakage vortex outside the SS of the blade.
Figure 8 presents the pressure coefficient distribution and streamlines at 30% $C_x$ of the uncooled squealer tip with (a) stationary casing and (b) moving casing. By comparing the two figures, it can be seen that the motion of the casing makes the pressure coefficient within the cavity become uniform. Actually, a scraping vortex forms near the casing because of the shearing force (Figure 8 (b)). As a consequence, the vortex in the cavity is compressed.

![Figure 8](image)

**Fig.8 Pressure coefficient distribution and streamlines at 30% $C_x$ of the uncooled squealer tip with (a) stationary casing and (b) moving casing.**

Figure 9 is the pressure coefficient distribution on the uncooled squealer tip with (a) stationary casing and (b) moving casing. The pressure coefficient at the impingement area of the leakage flow is low, as marked by 'D'. It can be seen that the pressure coefficient at the leading edge region is small but the pressure gradient is large, while the pressure coefficient at the trailing edge region is large but changes little for both cases of stationary and moving casing. With the moving casing, the low pressure coefficient area resulting from the impingement effect near the SS squealer becomes bigger and moves backward. The pressure coefficient at the blade tip decides the outlet pressure of the coolant, and then affects the tip cooling performance.

![Figure 9](image)

**Fig.9 Pressure coefficient distribution on the uncooled squealer tip with (a) stationary casing and (b) moving casing.**

### 3.2 Effect of relative motion

Figure 10 shows the tip leakage loss of different squealer tips under the overall blowing ratio $M=1$ with $H=1.7$mm (4.1%span). The tip leakage loss decreases greatly when the relative motion is added to the casing, and the maximum reduction is up to 11%. When the casing is moving, the scraping vortex near the casing acts as a pair of labyrinths with the cavity vortex, which increases the blockage effect of the squealer tip to the leakage flow. Besides, the relative motion has an inhibitory effect on the formation of the tip leakage vortex. The contraction area over the SS squealer is also an evidence of this effect, as seen in Figure 13. Whether the casing is stationary or moving, the tip leakage loss
from small to large order is type-A, type-B and type-C. The back pressures of the film holes in type-C are larger than those in the other two types, so type-C produces a larger coolant loss.

Figure 10 Tip leakage loss of different squealer tips under $M=1$ ($H=1.7\text{mm}$).

Figure 11 is the film-cooling effectiveness on the squealer tip under the overall blowing ratio $M=1$ with $H=1.7\text{mm (4.1\%\text{span})}$. Figure 12 presents the film-cooling effectiveness on the inner rim surfaces. It is clear that the distribution of the film-cooling effectiveness is closely related to the film-hole arrangement.

For type-A, the high film-cooling effectiveness is distributed between the middle-camber line and the PS squealer with both the stationary and moving casing since the coolant is mainly affected by the 's4'. Actually, the film-cooling effectiveness on the wake of the film holes is weaker with the moving casing in comparison to the stationary casing. It is caused by the stronger cavity vortex in the case of moving casing, as shown in Figure 7(b). Besides, the first film hole has no coolant inject owing to the movement of the impingement area under the moving casing condition. Withal, the film-cooling effectiveness on the inner rim surfaces is almost unchanged, and the high film-cooling effectiveness area is located at the trailing edge of the PS squealer for both cases of stationary and moving casing.

For type-B, the long strip-shaped film coverage is formed near the SS squealer with both the stationary and moving casing since the coolant is mainly driven by 's2' and 's3' and 's4'. In the stationary casing condition, the first film hole has no coolant inject; the coolant injection from the second film hole leaves the cavity with 's2' and 's3'; the remaining coolant is pushed towards the PS and trailing edge of the blade by 's4'. By contrast, in the case of moving casing, the film-cooling effectiveness on the second half of the cavity is much better as the vortex formed by 's4' has a greater tendency towards the PS squealer. Regarding the inner rim surfaces, the casing motion causes a large reduction in the film-cooling effectiveness on the rear half of the SS squealer in that the leaving location of the coolant moves towards the trailing edge on the effect of the scraping vortex.

For type-C, the tip film-cooling effectiveness is quite different under the condition of stationary and moving casing. When the casing is stationary, the film covers almost the entire tip owing to the reasonable arrangement of film holes. Specifically speaking, the coolant injecting from the first row of film holes at the leading edge is mainly affected by 's1' and 's2' and 's3', and the coolant injecting from the second row of film holes at the leading edge and the film holes at the middle chord region is mainly affected by 's4'. For the moving casing case, the domain region of the 's4' becomes smaller at the leading edge, so the film coverage in the downstream of the leading edge film holes disappears. On the other hand, the film-cooling effectiveness at leading edge of the inner rim surfaces of the squealers increases sharply as more coolant leaves the cavity directly. Notably, the high film-cooling effectiveness area on the inner rim surfaces is just the region with the largest heat load according to the study of Park et al. (2014).

Additionally, the local blowing ratios in the three different film-hole arrangements are summarized in Table 4. The values of the local blowing ratios of type-A and type-B are listed from the leading edge to the trailing edge, and those of type-C are shown in the turn of the first row, the second row and the middle chord row. Generally, the local blowing ratios of the film holes of type-A and type-B increase from the leading edge to the trailing edge. For type-C, the local blowing ratios of the film holes are more uniform and higher than those of type-A and type-B, especially at the leading edge. As a result, a well cooling performance is obtained by type-C. When a relative motion is added, the local blowing ratios at the leading edge of type-A and type-B increase, while those at the middle chord and trailing edge decrease. For type-C, the change of the local blowing ratios are relatively small.
Fig. 11 Film-cooling effectiveness on the squealer tip under $M=1$ with (a) stationary casing and (b) moving casing ($H=1.7\,\text{mm}$).

Fig. 12 Film-cooling effectiveness on the inner rim surfaces under $M=1$ with (a) stationary casing and (b) moving casing ($H=1.7\,\text{mm}$).

Table 4 Local blowing ratios in the three different film-hole arrangements.

| Type   | Film-hole number |
|--------|------------------|
|        | 1    | 2    | 3    | 4    | 5    | 6    | 7    | 8    | 9    | 10   | 11   | 12   | 13   |
| A, 0rpm| 0.03 | 0.57 | 0.84 | 0.95 | 1.03 | 1.08 | 1.12 | 1.15 | 1.18 | 1.20  | 1.23  | 1.25  | 1.26  |
| A, 8450rpm | 0    | 0.24 | 0.56 | 0.83 | 1.05 | 1.15 | 1.21 | 1.24 | 1.28 | 1.33  | 1.37  | 1.39  | 1.41  |
| B, 0rpm | 0    | 0.14 | 0.25 | 0.74 | 1.15 | 1.27 | 1.29 | 1.30 | 1.32 | 1.32  | 1.41  | 1.46  | 1.47  |
| B, 8450rpm | 0.36 | 0.25 | 0.33 | 0.61 | 0.81 | 0.99 | 1.15 | 1.26 | 1.33 | 1.38  | 1.49  | 1.54  | 1.56  |
| C, 0rpm | 1.14 | 0.51 | 0.39 | 1.41 | 0.76 | 0.52 | 0.59 | 0.80 | 1.05 | 1.33  | 1.44  | 1.49  | 1.53  |
| C, 8450rpm | 0.99 | 0.76 | 0.80 | 1.24 | 0.60 | 0.64 | 0.83 | 0.85 | 1.04 | 1.11  | 1.27  | 1.39  | 1.50  |
Figure 13 illustrates the pressure coefficient distribution and streamlines of type-A at 30% $C_x$ (the center of the third film hole) under the overall blowing ratio $M=1$ with $H=1.7$mm (4.1% span). It is clear that the pressure coefficient at the outlet of the film hole decreases when a speed is given to the casing. Therefore, the back pressure of the film hole increases greatly, which leads to the decrease of the discharge coefficient and local blowing ratio of the film hole. Besides, the stronger vortex makes the coolant leave the bottom of the cavity earlier.

Figure 14 shows the averaged film-cooling effectiveness on squealer tips under the overall blowing ratio $M=1$ with $H=1.7$mm (4.1% span). It can be seen from the figure that the averaged tip film-cooling effectiveness of type-C is much larger than those of type-A and type-B, and the difference between the latter two is small. The effect of casing motion on the averaged film-cooling effectiveness of squealer tip is closely related to the film-hole arrangement. Specifically, the averaged film-cooling effectiveness of type-A and type-C decreases, while that of type-B increases.

3.3 Effect of squealer height

Figure 15 presents the tip leakage loss of squealer tips with two rim heights under the overall blowing ratio $M=1$. With the decreasing of the squealer height, the tip leakage loss decreases in the case of stationary casing, while it increases in the case of moving casing. When the casing is stationary, the leakage flow entering the cavity decreases as the squealer height decreases. So the pressure loss within the cavity decreases. Besides, the outlet pressure of film holes decreases with the decrease of the squealer height, so the coolant loss also decreases. When the casing is moving, the scraping vortex and cavity vortex increase with the increasing of the squealer height, which poses a stronger blockage effect to the leakage flow. Besides, the inhibitory effect of the relative motion on the leakage vortex also increases with the increasing of the squealer height. As a result, the tip leakage loss decreases when the squealer height increases.
Figure 16 shows the film-cooling effectiveness on the squealer tip under the overall blowing ratio $M=1$ with $H=1\text{mm}$ (2.4\% span). Comparing to Figure 11, the tip cooling performance varies with the film-hole configuration when the rim height decreases. For type-A, the film coverage is no longer limited to the region between middle-camber line and PS squealer. Namely, the film covers both sides of the film holes and the whole trailing edge owing to the weakening of the cavity vortex and the enhancement of the coolant accumulation effect. For type-B, the film-cooling effectiveness at the trailing edge becomes better for stationary casing and worst for moving casing. In fact, the cavity vortex in the case of moving casing is stronger than that in the case of stationary casing, which makes the coolant cover the whole trailing edge, especially when the squealer is higher. For type-C, the film coverage at the leading edge and middle chord region near the PS squealer disappears. The reason is that the size of the cavity vortex (‘s4’) decreases, and more leakage flow spills out the cavity directly (‘s2’ and ‘s3’) after the impingement at the leading edge. In addition, the film-cooling effectiveness on the top of the SS squealer is increased due to the increase of the leakage flow that leaves the tip clearance directly, especially under the moving casing condition.

Fig.16 Film-cooling effectiveness on the squealer tip under $M=1$ with (a) stationary casing and (b) moving casing ($H=1\text{mm}$).
Figure 17 presents the film-cooling effectiveness on the inner rim surfaces under the overall blowing ratio $M=1$ with $H=1\text{mm}$ (2.4%span). In comparison to Figure 12, the film coverage on the inner rim surfaces increases for type-A and type-B. However, for type-C, the film-cooling effectiveness decreases on the PS inner rim surface, and increases on the SS inner rim surface. Besides, the casing motion provides a weaker influence on the film coverage of the inner rim surfaces when the squealer height is small due to the weakening of the vortexes within the tip region.

![Type A, B, and C diagrams for stationary and moving casings](image)

Fig. 17 Film-cooling effectiveness on the inner rim surfaces under $M=1$ with (a) stationary casing and (b) moving casing ($H=1\text{mm}$).

Figure 18 indicates the pressure coefficient distribution and streamlines at 30% $C_t$ of type-A under the overall blowing ratio $M=1$ with $H=1\text{mm}$. In comparison to Figure 13, the cavity vortex becomes weaker and more leakage flow passes through the tip clearance directly as the squealer height decreases for stationary casing. Under the condition of moving casing, the cavity vortex and craping vortex both decrease with the decrease of squealer height. In addition, more leakage flow impinges on the cavity bottom under the craping vortex near the SS squealer and then spills out the tip clearance over the SS squealer.

![Pressure coefficient distribution and streamlines at 30% $C_t$](image)

(a) Stationary casing                          (b) Moving casing

Fig. 18 Pressure coefficient distribution and streamlines at 30% $C_t$ of type-A under $M=1$ with (a) stationary casing and (b) moving casing ($H=1\text{mm}$).

Figure 19 shows the averaged film-cooling effectiveness on squealer tips with different rim heights under the overall blowing ratio $M=1$. When the squealer height decreases, the averaged value of type-A increases, while that of type-C decreases in the cases of both stationary and moving casing. It is worthy to notice that the averaged value of type-B increases in the case of stationary casing and decreases in the case of moving casing. On the whole, type-C obtains the best tip cooling performance under the conditions of both stationary and moving casing.

Figure 19 shows the averaged film-cooling effectiveness on squealer tips with different rim heights under the overall blowing ratio $M=1$. When the squealer height decreases, the averaged value of type-A increases, while that of type-C decreases in the cases of both stationary and moving casing. It is worthy to notice that the averaged value of type-B increases in the case of stationary casing and decreases in the case of moving casing. On the whole, type-C obtains the best tip cooling performance under the conditions of both stationary and moving casing.
| squealer type       | H=4.1% span, stationary casing | H=2.4% span, stationary casing | H=4.1% span, moving casing | H=2.4% span, moving casing |
|---------------------|--------------------------------|--------------------------------|---------------------------|---------------------------|
| type-A stationary   | 0.25                           | 0.25                           | 0.30                      | 0.30                      |
| type-A moving       | 0.20                           | 0.20                           | 0.25                      | 0.25                      |
| type-B stationary   | 0.15                           | 0.15                           | 0.10                      | 0.10                      |
| type-B moving       | 0.10                           | 0.10                           | 0.05                      | 0.05                      |
| type-C stationary   | 0.05                           | 0.05                           | 0.00                      | 0.00                      |
| type-C moving       | 0.00                           | 0.00                           | 0.05                      | 0.05                      |

![Graph](image1.png)

Fig.19 Averaged film-cooling effectiveness on squealer tips with different rim heights ($M=1$).

### 3.4 Effect of blowing ratio

Figure 20 presents the averaged film-cooling effectiveness of squealer tip with $H=1.7\text{mm}$ (4.1% span) under four overall blow ratios. In the case of stationary casing, the film-cooling effectiveness on the squealer tips with different film-hole arrangements increases as the blowing ratio increases. Concretely, type-C has a much bigger growth rate than the other two types. The averaged film-cooling effectiveness of type-A and type-B is almost the same at the same overall blowing ratio.

In addition, the influence of the relative motion of the casing on the averaged film-cooling effectiveness of the squealer tip is closely related to the location of the film holes and the blowing ratio. When the blowing ratio is low ($M=0.5$), the averaged film-cooling effectiveness of type-A and type-B decreases obviously, while that of type-C increases slightly. Under $M=1$, the averaged film-cooling effectiveness of type-A and type-C decreases obviously, while that of type-B increases slightly. For $M=1.5$, the averaged film-cooling effectiveness of type-A and type-C decreases, and the mutative extent of type-C is much larger than that of type-A, while the averaged film-cooling effectiveness of type-B changes a little. At $M=2$, the averaged film-cooling effectiveness of type-C decreases greatly, while those of type-A and type-B are almost unchanged.

![Graph](image2.png)

Fig.20 Average film-cooling effectiveness on squealer tips with different overall blowing ratios ($H=1.7\text{mm}$).

### 4 Conclusions

In this paper, the influence of casing relative motion on the leakage flow and film-cooling effectiveness of the squealer tip is studied numerically. Three film-hole configurations, two rim heights and four blowing ratios are taken into consideration. The main contributions are summarized in the following:

1. The relative movement of the casing leads to the formation of a scraping vortex near the SS squealer, which not only compresses the cavity vortex, but also acts as a pair of labyrinths with the cavity vortex to the leakage flow. So the tip leakage flow and the domain area of the cavity vortex decrease compared to the stationary casing.

2. In comparison to the stationary casing, the tip leakage loss decreases greatly when a speed is applied to the
casing. The effect of it on the film-cooling effectiveness of squealer tip is closely related to the film-hole arrangement. Specifically, the film coverage area of type-A and type-C decreases, while that of type-B increases. Type-C obtains the best tip cooling performance in both cases of stationary and moving casing on the surfaces including the cavity bottom and the inner rim surfaces.

3. Under the condition of stationary casing, the decreasing of the squealer height makes the cavity vortex become weaker, and more leakage flow passes through the tip clearance directly without entering the cavity. Similarly, in the case of moving casing, both the cavity vortex and scraping vortex decrease, so more leakage flow impinges on the cavity bottom under the scraping vortex near the SS squealer and then pills out the tip clearance over the SS squealer.

4. In the case of stationary casing, the tip leakage loss decrease slightly for all the three film-hole configurations when the rim height decreases due to the decrease of the pressure loss within the cavity and the coolant loss. In addition, the film-cooling effectiveness of type-A and type-B increases, while that of type-C decreases due to the change of the film coverage area. When the casing is moving, the tip leakage loss increases with the decrease of the rim height because the blockage effect of the vortexes in the cavity to the leakage flow decreases. Besides, the film-cooling effectiveness of type-A increases, while those of type-B and type-C decrease. Notably, type-C is still the best film-hole configuration from the tip cooling point of view when the squealer height decreases, especially for the high heat transfer coefficient area.

5. With the increasing of blowing ratio, the tip film-cooling effectiveness increases under both stationary and moving casing conditions. The relative motion has the greatest decreasing effect on the cooling performance of type-C at the relative high blowing ratios. Actually, type-C still obtains better tip cooling performance under the conditions of both stationary and moving casing at the same blowing ratio compared to type-A and type-B.

Acknowledgments

The authors gratefully acknowledge the financial support for this project from the National Natural Science Foundations of China (grant No: U1508212, 51406085).

References

Acharya, S., Yang, H. T., and Ekkad, S. V., Numerical simulation of film cooling on the tip of a gas turbine blade, ASME Paper GT-2002-30553 (2002).
Acharya, S., and Moreaux, L., Numerical study of the flow past a turbine blade tip: effect of relative motion between blade and shroud, Journal of Turbomachinery, Vol.36, Issue3 (2012), pp. 471-480.
Cheng, F. N., Chang, H. P. Zhang, J. Y. and Tian, X. J., Effect of film-hole configuration on film-cooling effectiveness of squealer tips, Journal of Thermal Science and Technology, Vol.12, Issue 1 (2017), pp. 1-15.
Kwak, J. S. and Han, J. C., Heat transfer coefficients and film-cooling effectiveness on the squealer tip of a gas turbine blade, ASME Journal of Turbomachinery, Vol.125, Issue 4 (2003), pp. 648-656.
Lee, S. W. and Kim, S. U., Tip gap height effects on the aerodynamic performance of a cavity squealer tip in a turbine cascade in comparison with plane tip results part 1-tip gap flow structure, Experiments in Fluids, Vol.49, Issue3 (2010), pp. 1039-1051.
Palafox, P., Oldfield, M. L. G., Ireland, P. T., Jones, T. V. and LaGraff, J. E., Blade tip heat transfer and aerodynamics in a large scale turbine cascade with moving endwall, Journal of Turbomachinery, Vol.134, Issue2 (2012), pp. 469-482.
Park, J. S., Lee, D. H., Rhee, D. H., Kang, S. H. and Cho, H. H., Heat transfer and film cooling effectiveness on the squealer tip of a turbine blade, Energy, Vol.72, Issue7 (2014), pp. 331-343.
Rhee, D. H. and Cho, H. H., Local heat/mass transfer characteristics on a rotating blade with flat tip in a low-speed annular cascade – part 2: tip and shroud, Journal of Turbomachinery, Vol.128, Issue1 (2006), pp. 96–109.
Srinivasan, V. and Goldstein, R. J., Effect of endwall motion on blade tip heat transfer, Journal of Turbomachinery, Vol.125, Issue2 (2003), pp. 267–273.
Timko, L. P., Energy efficient engine high pressure turbine component test performance report, NASA CR-168289 (1984).
Wang, J., Sunden, B., Zeng, M. and Wang, Q. W., Influence of different rim widths and blowing ratios on film-cooling
characteristics for a blade tip, Journal of Heat Transfer, Vol.134, Issue6 (2012), pp. 3885-3890.
Yang, H.T., Acharya, S. and Ekkad, S.V., Numerical simulation of flow and heat transfer past a turbine blade with a squealer tip, ASME Paper GT-2002-30193 (2002).
Yang, D. L., Yu, X. B. and Feng, Z. P., Investigation of leakage flow and heat transfer in a gas turbine blade tip with emphasis on the effect of rotation, Journal of Turbomachinery, Vol.132 Issue 4 (2010), 041010.
Yang, H. T., Chen, H. C. and Han, J. C., Film-cooling prediction on turbine blade tip with various film-hole configurations, Journal of Thermophysics and Heat Transfer, Vol.20, Issue3 (2006), pp. 558-568.
Zhang, Q., O’Dowd, O., He, L., Oldfield, M. and Ligrani, P., Transonic turbine blade tip aero-thermal performance with different tip gaps: part 1 – tip heat transfer, Journal of Turbomachinery, Vol.133, Issue4 (2010), pp.335-346.
Zhang, X., Yang, Z. and Tian, S. Q., Numerical study of the film cooling with discrete-hole arrangement on a cut back squealer blade tip, ASME Paper GT2015-43877 (2015).
Zhou, C. and Hodson, H., The tip leakage flow of an unshrouded high pressure turbine blade with tip cooling, ASME Journal of Turbomachinery, Vol.131, Issue4 (2011), pp. 929-942.
Zhou, C., Effects of endwall motion on thermal performance of cavity tips with different squealer width and height, International Journal of Heat and Mass Transfer, Vol.91, Issue3 (2015), pp.1248–1258.