Effect of Cutbacks on Tip Leakage Flow and Film-Cooling Effectiveness of a Turbine Blade Tip under Relative Moving Condition*

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Numerical investigations are conducted to study the tip leakage flow and cooling effectiveness on various film-cooled squealer tips. The effect of partial squealer rims on the leakage flow structure, tip leakage loss and tip film-cooling effectiveness is illustrated under four blowing ratios by considering the relative motion between the turbine blade and the stationary casing. Six cutback configurations of partial squealer rims, including three pressure-side trailing-edge cutbacks and three suction-side trailing-edge cutbacks with different cutback lengths, are considered in total. Two types of film-hole arrangements are used, both having the same film holes. In the first type, 13 film holes are arranged uniformly along the middle camber line (referred to as CL mode). In the second type, seven film holes are located in the vicinity of the leading-edge and the remainder of the film holes are arranged near the suction side in the mid-chord region (referred to as LE mode). It is found that the partial squealer with cutback near the trailing-edge produces larger tip leakage mass flow ratios than the full squealer, especially for the suction-side squealer cutback. However, the blade tips with a suction-side squealer cutback are conducive to reducing the tip leakage loss, especially for the partial squealer with a moderate cutback length. Furthermore, the suction-side squealer cutbacks can improve the tip film-cooling effectiveness at the trailing-edge of the blade to a great extent. It is also confirmed that the LE mode produces a more uniform and higher tip film-cooling effectiveness than the CL mode.

Key Words: Squealer Tip, Cutbacks, Film-cooling Effectiveness, Leakage Flow, Tip Leakage Loss

Nomenclature

C_x : axial chord
 C_p : dimensionless pressure
 $C_{pt-total}$: total-pressure loss coefficient
 $C_{pt-profile}$: profile loss coefficient
 C_{pt-tip} : tip leakage loss
 d : diameter of film holes
 H : height of the squealer
 m : mass flow rate
 M : blowing ratio
 P : pressure
 P_{in-ref}^* : reference total pressure
 R_{mtip} : tip leakage mass flow ratio
 S : area
 t : tip clearance gap
 T : temperature
 W : width of the squealer
 u : velocity

Subscripts

c: coolant flow
in: inlet
out: outlet

av: area-averaged
aw: adiabatic wall condition
 ∞ : mainstream

Superscripts

*: stagnation parameter

1. Introduction

Tip leakage flow is a typical phenomenon in turbine cascades due to the tip clearance between the rotating turbine blade and the stationary casing. As a result of the pressure difference between the two sides of the blades, some high-temperature gases are driven across the tip clearance gap, which causes an inevitable loss of turbine efficiency and a high thermal load on the blade tip. A large number of studies have shown that the film-cooled squealer blade tip is the most effective scheme to reduce tip leakage loss and avoid blade thermal damage. On one hand, the squealer on the blade tip causes additional flow resistance to the leakage flow. On the other hand, the cooling airflow is used to protect the tip surface from the high-temperature mainstream. To achieve good aero-thermal performance, the optimization of the squealer type is a major issue for engineering designers.

Over the past few decades, many studies had been conducted to illustrate the effect of tip shape on the aero-thermal performance of blades. Azad et al.^{1,2)} explored the thermal characteristics of several different squealer tips using the

transient liquid-crystal technique. A squealer on the suction side was demonstrated to offer greater benefit than other types. Kwak et al.^{3,4)} further measured the heat transfer coefficients on several different tip surfaces. Their results indicated that the tip shape affects the flow trajectory of leakage flow and subsequently the heat transfer coefficient map on the tip. The tip heat transfer coefficient was the lowest when the squealer rim was located on the suction side. Nasir et al.⁵⁾ and Camci et al.⁶⁾ revealed that partial squealer rims are beneficial to weaken the strength of the leakage vortex and the heat transfer coefficient on the blade tip. Yang et al.⁷⁾ indicated that the tip heat load of the shallow squealer is the lowest. Moreover, rotation had little effect on the average heat transfer coefficient although it indeed affected the local heat transfer coefficient. Prakash et al.⁸⁾ found that the pressure-side inclined shelf leads to greater tip flow separation and higher turbine efficiency compared with the straight shelf. Krishnababu et al.^{9,10)} compared the effect of relative motion and stationary casing on different squealer tips from both aerodynamic and heat transfer perspectives. They found that the relative motion of the casing would reduce the leakage mass flow thanks to reducing the driving pressure difference of the leakage flow. As a result, the heat transfer coefficient of the blade tip was reduced. Researchers^{11–13)} obtained similar results about relative casing motion through experimental and numerical investigations. Recently, Lee et al.¹⁴⁾ illustrated the detailed flow field of a different cavity squealer tip using a naphthalene sublimation technique. Mischo et al.¹⁵⁾ compared several blade-tip shapes in a turbine stage. Their results showed that the heat transfer coefficient on the blade tip could be reduced by utilizing an appropriate tip shape. Nho et al.¹⁶⁾ experimentally studied the aero-thermal performance of 11 tip types in linear cascades with five blades: squealer tip, plane tip, chamfered tip, grooved tip, and dimpled tip. It was found that the squealer tip performs well in terms of both aerodynamic and heat transfer characteristics. Lomakin et al.¹⁷⁾ studied nine variants of squealer tips based on CFD simulation. The effects of partial squealer extension and rim inclination were the main focus. It was confirmed that suitable trailing-edge cutoff rims on the suction side are beneficial for controlling tip leakage flow. Jung et al.¹⁸⁾ applied a squealer tip to a transonic axial compressor. Their results showed that the appropriate depth and length of the squealer rim are beneficial to improve the stall margin. In addition, the tip leakage flow was very weak due to strong vorticity in the recess cavity. Da Silva et al.¹⁹⁾ numerically studied the influences of different rotor tip geometry configurations, including plane tip, winglet tip, squealer tip, and squealer-winglet tip, on turbine efficiency. It was indicated that the rotor equipped with the squealer-winglet tip configuration presents a better pressure ratio compromise.

It is common sense that the tip leakage flow and heat transfer on the turbine tip are tightly associated with the squealer configuration. To our knowledge, relatively little effort has been devoted to reveal the comprehensive performance of partial squealer tips on tip leakage flow and film cooling characteristics, especially when considering both the relative

casing motion and arrangement of the film holes at the same time. In this study, numerical simulations are conducted to illustrate the effect of partial squealer rims on the leakage flow structure, tip leakage loss and film-cooling effectiveness on the tip. Two types of film-hole arrangements are considered. For the first type, 13 film holes are arranged uniformly along the middle camber line (referred to as CL mode). For the second type, seven film holes are located in the vicinity of the leading-edge and the remaining six film holes are arranged near the suction-side squealer in the mid-chord region (referred to as LE mode). In previous studies,^{20,21)} we demonstrated the superiority of the LE mode for improving the film-cooling effect on the squealer tip. In this paper, we continue to explore the effect of trailing-edge cutbacks while considering the relative motion between the blade tip and the casing. Three relative cutback lengths and two cutback locations (i.e., either the pressure side or suction side) of squealer rims are taken into consideration. Four typical blowing ratios are selected in a practical range between 0.5 and 2.0.

2. Physical Model Description

Figure 1(a) shows a physical model of turbine blades that is formed by the top profile of the GE-E3 first-stage high-pressure turbine blade.²²⁾ The blade pitch is 30.5 mm and the axial chord length (C_x) is 28.7 mm. The inlet flow angle (α_1) and exit angle (α_2) of the blade are 32.01° and 65.7° , respectively. The span height of the blade is 41 mm. The height of the squealer rim (H) is 1.7 mm (4.1% blade span). The width of the squealer rim (W) is 0.7 mm. In this investigation, the blade tip clearance (t) is selected as 1 mm (2.4% blade span) considering that this is a typical gap value in turbine design.

Due to the periodicity of the cascade, one blade pitch is selected to establish a three-dimensional calculation domain, as shown in Fig. 1(b). The distance between the inlet plane and the blade leading-edge is $1C_x$ and the outlet plane is located $1.5C_x$ downstream of the blade trailing-edge. Lee and Kim²³⁾ indicated that the tip leakage flow is mainly influenced by the flow conditions close to the blade tip. Accordingly, the span size of the computational domain is selected as 57% blade height (25 mm) in order to reduce the number of computational meshes. Similar to the treatment of Zhou and Hodson,²⁴⁾ the bottom of the computational domain is set as a symmetrical surface. For simplicity, a simple coolant plenum-mode is adopted to provide the cooling air, as shown in Fig. 1(c).

In this study, two types of film-hole arrangements are taken into account, as shown in Fig. 2. The total number of film holes in both types is the same (i.e., 13 holes). All of the cooling holes have the same diameter (d) of 0.423 mm and a hole length of 1 mm. The CL mode is the basic case in which film holes are arranged uniformly along the middle camber line with a hole-to-hole pitch of 5 deg, as demonstrated in Fig. 2(a). Figure 2(b) displays a diagram of the LE mode. It has seven film holes located in the vicinity of

(a) Definition of the blade tip and casing

(b) Computational domain

(c) Coolant plenum

Fig. 1. Physical model and computational domain.

the leading-edge (i.e., corresponding to 12–26% C_x) and the remaining six film holes are arranged near the suction-side squealer in the mid-chord region (i.e., corresponding to 34–60% C_x).

Seven squealer rim configurations are designed: (a) one full squealer rim or baseline squealer rim (denoted as Baseline, as shown in Fig. 2), (b) three partial squealer rims with different cutbacks on the pressure side (denoted as PScut), and (c) three partial squealer rims with different cutbacks on the suction side (denoted as SScut). The typical configurations of partial squealer rims combined with the film-hole arrangements are schematically shown in Fig. 3, and the cutback locations are shown in Table 1. Consequently, a total of 14 different blade tips are investigated in this study. It is remarkable that the absolute length of PScut1 is the same as that of SScut1.

3. Computation Procedure

3.1. Computation scheme

Numerical calculations are conducted by solving three-

Fig. 2. Film-hole arrangements.

Fig. 3. Typical configurations of blade tip.

Table 1. Cutback locations.

Cutback	Beginning	Ending
Baseline	None	None
PScut1	75% C_x	Trailing-edge
PScut2	80% C_x	Trailing-edge
PScut3	85% C_x	Trailing-edge
SScut1	80% C_x	Trailing-edge
SScut2	85% C_x	Trailing-edge
SScut3	90% C_x	Trailing-edge

dimensional compressible steady Reynolds-Averaged Navier-Stokes (RANS) equations using the commercial solver FLUENT.²⁵⁾ The governing equations are discretized by the finite volume method and the standard two-equation turbulence model is chosen. According to previous studies,^{26,27)} the simulation results obtained using the k - ϵ turbulence model are in good agreement with the experimental results. Pressure and velocity are coupled using the SIMPLEC algorithm and the second-order upwind scheme is applied to spatial discretization. The convergence criterion is set to the order of the magnitude of residual less than 10^{-5} , and the difference in the mass flow rate between inlet and outlet is less than 0.1%.

The computational boundary conditions are derived from a full-scale turbine test,²²⁾ as summarized in Table 2. An adiabatic non-slip condition is adopted for all of the solid walls. The coolant mass flow rate is determined by the global blowing ratio, defined as,

$$M = \frac{\rho_c u_c}{\rho_\infty u_\infty} \quad (1)$$

where ρ_c and u_c are the coolant density and ejecting velocity, respectively, and ρ_∞ and u_∞ are the mainstream density and velocity at the cascade inlet, respectively.

The symmetry boundary condition is applied for the hub section, and the periodic boundary condition is applied for

Table 2. Main aero-thermal parameters.

Parameter	Value
Inlet total pressure, P_{in}^*	303 kPa
Inlet total temperature, T_{in}^*	683 K
Inlet turbulence intensity	9.7%
Outlet 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
Coolant inlet turbulence intensity	3%
Relative rotating speed	8,450 rpm

the cascade passage surfaces in the computational domain, as shown in Fig. 1(b). According to the work of Acharya and Moreaux,²⁶⁾ the magnitude of rotational force effect is relatively small compared to the relative motion effect. Thus, the linear velocity corresponding to a rotating speed of 8,450 rpm is forced to the casing to simulate the relative motion effect.

3.2. Grid generation and independence test

The commercial software GAMBIT is chosen to generate the computational mesh. Enhanced wall treatment is adopted for the near wall regions, so the mesh near the solid walls is refined to ensure that y^+ is less than 1.89. Additionally, the mesh within both the tip clearance and squealer cavity is also dense because the tip region is of the greatest interest. Thirty-five and 40 nodes are set within the tip clearance and cavity along the blade span direction, respectively. The growth factor between neighboring cells is less than 1.2. Figure 4 illustrates the grid information for some areas.

To verify that the calculation results are independent of the number of grids, we calculate the film cooling effect on the blade tip using four total grid numbers (i.e., 986,125, 1,839,513, 3,829,437 and 4,214,796). The film cooling effect is expressed by film-cooling effectiveness, which is defined as follows,

$$\eta = \frac{T_\infty - T_{aw}}{T_\infty - T_c} \quad (2)$$

where T_∞ and T_c are the mainstream temperature at the cascade inlet and coolant temperature, respectively. T_{aw} is the adiabatic wall temperature.

Figure 5 presents the influence of grid number on the tip pitch-averaged film-cooling effectiveness for the CL-Baseline case. Here the pitch-averaged film-cooling effectiveness is defined as the line-averaged film-cooling effectiveness of the cavity bottom surface along the pitch or y direction, and it changes with the dimensionless axial length (x/C_x). As seen in Fig. 5, the tip pitch-averaged film-cooling effectiveness varies little after the grid number reaches 3,829,437.

3.3. Validation of CFD method

To validate the calculation method, three validation cases are studied based on the experimental models used by Azad et al.¹⁾ and Kwak et al.,^{28,29)} respectively. In both validation cases, the relative motion effect is not taken into consideration, since there are few available data obtained under rotating conditions.

Fig. 4. Computational mesh.

Fig. 5. Grid independence test (CL-Baseline).

The simulation model is the same as that of the experiment, which was formed using a 3X scaled GE-E3 blade. The axial chord and pitch of the scaled-up blade are 8.61 and 9.15 cm, respectively. A squealer height of 4.6 mm and tip clearance of 1.97 mm are chosen in the calculations. Only one single blade is taken into consideration and the periodic conditions are applied to the pitch direction surfaces. The total temperature (300 K) and pressure (126.9 kPa) are specified at the inlet of the computational domain. The static pressure of 102.7 kPa is set at the outlet. The temperature of the coolant is set as 328 K and the coolant mass flow rate is determined in accordance with the global blowing ratio $M = 1$. No-slip adiabatic conditions are applied to the surfaces of the blade. Note that the detailed flow conditions are presented in Kwak et al.,⁴⁾ which is the latest description of the serial experiments.

Figure 6(a) is a comparison of the calculated and experimental (Azad et al.¹⁾) local pressure ratios at 97% blade height. Here, the pressure ratio is defined as P_{in}^*/P , where P_{in}^* and P denote the inlet total pressure and local static pressure, respectively. It can be seen that the calculated results are perfectly consistent with the experimental results. Figure 6(b) presents a comparison of the heat transfer coefficient distribution on the blade tip between the current calculation and the experimental results of Kwak et al.²⁸⁾ It is obvious that the present calculation provides a good prediction of heat transfer coefficient distribution. The high local heat transfer occurs in the leading-edge region of the cavity bottom. Toward the trailing-edge, the local heat transfer coefficient decreases gradually. Figure 6(c) presents a comparison

 (a) Distribution of P_{in}^*/P along the blade at 97% span

(b) Blade tip heat transfer coefficient distribution

 (c) Blade tip film-cooling effectiveness distribution ($M=1$)

Fig. 6. Validation of CFD method.

of film-cooling effectiveness distribution on the blade tip between the current calculation and the experimental results of Kwak and Han.²⁹⁾ It is also confirmed that the present calculation agrees well with the experimental results. The high local film-cooling effectiveness appears in the region between the middle line and pressure side (marked as 'A'). In the region between the middle line and the suction side (marked as 'B'), almost no film covering is formed.

4. Results and Discussions

4.1. Effect of film-hole arrangements under $M = 1$

Figure 7 presents a comparison of the flow streamlines between the CL-Baseline and LE-Baseline cases with a blowing ratio of $M = 1$. It is obvious that the position of the film

 Fig. 7. Flow streamlines on the baseline squealer tip with different film-hole arrangements at $M = 1$.

hole has a relatively slight influence on the tip flow-field of the mainstream, but a significant influence on the coolant flow pattern.

Leakage flow enters into the tip clearance from two locations; the suction-side leading-edge and pressure-side leading-edge, respectively. Its flow paths near the squealer are tightly associated with the original location. As shown in Fig. 7(a), leakage flow entering the tip gap from the leading-edge of the suction side (marked as 's1') impacts the bottom of the cavity and then flows out immediately from the front of the suction side. Differently, the leakage flow entering the tip gap from the leading-edge of the pressure side (marked as 's2') flows toward two directions. One part discharges the tip clearance from the front of the suction side after impinging on the bottom of the cavity. The other part flows back from the medial camber line to the pressure side after impinging on the cavity bottom and then leaves the clearance from the trailing-edge of the suction side. The mainstream entering from the middle chord and trailing-edge of the pressure side (marked as 's3') passes through the clearance directly without any interaction with the cavity bottom.

With regard to the coolant flow streamlines, the flow pattern in the case of the CL-Baseline is simpler compared to that of the LE-Baseline. As seen in Fig. 7(a), there is nearly no cooling air ejected from the first film hole. The reason is that the film hole is just located in the zone (marked as 'C') subjected to serious impingement by the leakage flow originating from the leading-edge of the pressure side (i.e., second part of 's2'). Accordingly, the coolant streamlines flow to the blade pressure side and then leave the tip clearance. However, the coolant jets and tip leakage flow produce a more complicated interaction in the case of the LE-Baseline, as illustrated in Fig. 7(b). Firstly, a part of the coolant is forced to

the leading-edge and then flows along the squealer rim to the suction-side squealer, marked as 'cs1'. Secondly, part of the coolant is forced to the suction-side squealer and then leaves the cavity from the front of the suction side, as labeled by 'cs2'. Additionally, the coolant exiting the holes located between $0.13C_x$ to $0.46C_x$ near the suction-side squealer is found to be divided into two parts due to the complex leakage flow. One part is forced to the suction side and then turns toward the pressure-side trailing-edge, as marked by 'cs3'. The other part flows to the trailing-edge of the pressure-side squealer directly, as marked by 'cs4'. Due to the more reasonable organization of coolant flow inside the tip cavity, the LE-Baseline configuration produces more uniform film coverage on the blade tip compared to the CL-Baseline configuration.

Figure 8 compares the adiabatic film-cooling effectiveness distributions between the CL-Baseline configuration and LE-Baseline configuration at $M = 1$. It can be seen that CL-Baseline configuration has a long strip of gas-film coverage between the middle arc and the pressure side of the blade. In addition, since the coolant jet is mainly discharged from the rear of the suction side, the corresponding squealer surface and inner rim walls are effectively protected. For the LE-Baseline configuration, as shown in Fig. 8(b), the film coverage on the cavity floor, top surface of the suction-side squealer and inner surfaces of the squealer rims are all greatly improved compared to the CL-Baseline configuration.

Figure 9 compares the pitch-averaged film-cooling effectiveness distributions at the bottom of the cavity between the CL-Baseline and LE-Baseline configurations at $M = 1$. The LE-Baseline configuration is demonstrated to provide higher film-cooling effectiveness than that of the CL-Baseline configuration over the entire chord direction, especially near the leading-edge. The most significant difference between them appears at $0.12C_x$.

Figure 10 and Fig. 11 present 2-D views of streamlines and dimensionless pressure distributions at different cross-sections. Here, the dimensionless pressure is defined as follows,

$$C_p = \frac{P_{in}^* - P}{P_{in}^* - P_{out}} \quad (3)$$

where P_{out} denotes the outlet static pressure.

It is obvious that the film-hole arrangement has a clear influence on the flow-field inside the cavity, but only has a slight effect on the tip leakage vortex outside of the suction-side squealer. The coolant of CL-Baseline configuration flows to the blade pressure side, as shown in Fig. 10(b). Differently, the coolant injected from the film holes at the front of the LE-Baseline configuration flows to both pressure and suction sides of the blade, as seen in Fig. 11(a), and the coolant injected from the rear film holes goes to the blade pressure side, as seen in Fig. 11(b). Additionally, the development process of the leakage vortex outside the suction side is discerned from the streamlines.

To charter the tip leakage flow, a tip leakage mass flow ratio (R_{mtip}) is introduced,

Fig. 8. Film-cooling effectiveness distributions on the blade tip with baseline squealer at $M = 1$.

Fig. 9. Pitch-averaged film-cooling effectiveness at the bottom of the cavity at $M = 1$.

$$R_{mtip} = \frac{m_{tip}}{m_{\infty}} \quad (4)$$

where m_{tip} is the mass flow rate of the mainstream entering into the tip clearance and m_{∞} is the mass flow rate of the full-span cascade.

According to the work of Zhou and Hodson,²⁴⁾ a parameter for evaluating the tip leakage loss (C_{ptip}) is used,

$$C_{ptip} = C_{pt-total} - C_{pt-profile} \quad (5)$$

where $C_{pt-total}$ is the total-pressure loss coefficient and $C_{pt-profile}$ is the profile loss coefficient for the constant area mixed-out midspan exit flow,

$$C_{pt-total} = \frac{P_{in-ref}^* - P^*}{P_{in-ref}^* - P} \quad (6)$$

where P_{in-ref}^* and P^* present the reference and local total pressure, respectively.

The reference total pressure is defined by taking the total pressure of the coolant into account for a film-cooled tip,

$$P_{in-ref}^* = \frac{m_{\infty} \cdot P_{in}^* + m_c \cdot P_{in-c}^*}{m_{\infty} + m_c} \quad (7)$$

where m_c is the mass flow rate of the coolant and P_{in-c}^* is the total pressure of the coolant inlet.

Fig. 10. Streamlines and dimensionless pressure distributions of the CL-Baseline configuration.

Fig. 11. Streamlines and dimensionless pressure distributions of the LE-Baseline configuration.

Fig. 12. Effect of squealer cutbacks on flow streamlines in the vicinity of the squealer tip at $M = 1$.

From the computational results, it is derived that both film-hole arrangements produce nearly the same tip leakage mass flow ratio of 4.38% and an approximate leakage loss coefficient of 0.142.

4.2. Effect of squealer cutbacks at $M = 1$

Figure 12 shows the flow streamlines of cutback squealer tips for two different film-hole configurations at $M = 1$. In comparison with the baseline configurations (Fig. 7), it is clearly seen that a partially cut squealer mainly affects the flow paths of the leakage flow and coolant near the cutbacks, for both CL and LE modes. When the cutbacks are located on the pressure side, more leakage flow enters the tip clearance from the cutbacks, and some coolant flowing towards the trailing-edge mixes with the leakage flow, and then discharges from the tip clearance along with the leakage flow, as seen in Fig. 12(a1) and (a2). For the squealer tip with cutbacks located on the suction side, the coolant is more inclined to leave the tip clearance with some leakage flow through the cutbacks, as seen in Fig. 12(b1) and (b2).

Figure 13 and Fig. 14 present the effects of squealer types on film-cooling effectiveness distributions at a blowing ratio of $M = 1$, for CL and LE modes, respectively. Due to the existence of the cutbacks, the interaction between the leakage flow and cooling jets is changed. Consequently, the film coverage on the blade tip is affected. It is found that the suction-side squealer cutback plays an obvious role in improving the film-cooling effectiveness at the trailing-edge of the blade, for both CL and LE modes. In fact, the cooling air is inclined to leave the tip clearance through the cutback of the suction-side squealer, so the adjacent areas are well protected. As squealer cutback increases on the suction side, the improvement in film-cooling effectiveness on the trailing-edge is enhanced. Differently, the tip film-cooling effectiveness seems to be slightly affected in comparison to the baseline squealer when the cutback is located on the pressure-side trailing-edge.

The effect of squealer cutbacks on pitch-averaged film-cooling effectiveness at the bottom of the cavity floor is shown in Fig. 15. In general, the suction-side squealer cutback is clearly demonstrated to be favorable for improving

Fig. 13. Effect of cutbacks on film-cooling effectiveness distributions for the squealer tip with CL mode at $M = 1$.

Fig. 14. Effect of cutbacks on film-cooling effectiveness distributions for the squealer tip with LE mode at $M = 1$.

the cooling effect near the trailing-edge of the blade, while the film-cooling effectiveness is influenced slightly by the pressure-side squealer cutback. Actually, the film-cooling effectiveness on the trailing-edge of the cavity floor increases as the suction-side squealer cutback length increases, but it decreases slightly when the pressure-side squealer cutback length increases. The reason for this effect is that the long suction-side squealer cutback increases the coverage area of the coolant on the trailing-edge, while the long cutback on the pressure-side squealer makes the coolant leave the cavity floor in a more forward position owing to the strong leakage flow above the cutback. In addition, it is obvious that the film-cooling effectiveness in the front and middle region of the blade tip is also affected by the cutbacks, but the impact is weak. Notably, it is also confirmed that the LE mode

Fig. 15. Effect of squealer types on the pitch-averaged film-cooling effectiveness at the bottom of the cavity floor at $M = 1$.

is superior to the CL mode from the perspective of improving tip film-cooling effectiveness.

Figure 16 shows 2-D views of streamlines and dimensionless pressure distributions at the specific cross-section (i.e., 90% chord). When the squealer cutback is located on the pressure side (CL-PScut1 configuration), the gap height in the slotted area increases. As a result, more leakage flow enters the tip gap from the cutback area when compared to the CL-Baseline configuration. Since the leakage flow near the pressure-side trailing-edge is no longer blocked by the squealer, the flow separation behind the squealer is weakened and the pressure coefficient increases, as seen in Fig. 16(a). It is also found that the leakage vortex shows nearly the same scale as that of the CL-Baseline configuration (Fig. 10(c)). With regard to the suction-side squealer cutback (i.e., CL-SScut1), some leakage flow leaves the tip clearance through the cutback instead of passing over the squealer rim. This part of the leakage flow is restricted to the blade suction side, which is not involved in the leakage vortex. Therefore, the tip leakage vortex is found to be weaker and the pressure coefficient inside the tip clearance is lower than the CL-Baseline configuration, as seen in Fig. 16(b).

As the partial squealers weaken the blockage effect of the squealer tip on leakage flow, the tip leakage mass flow ratio of the blade with squealer cutbacks is increased by making comparisons with the baseline cases. Figure 17 presents the effect of squealer types on the tip leakage mass flow ratio at $M = 1$. It is obvious that suction-side cutback has a more

Fig. 16. Two-dimensional views of streamlines and dimensionless pressure distributions at a specific cross-section (90% chord).

significant influence on the tip leakage mass flow ratio than the pressure-side cutback. For instance, the tip leakage mass flow ratio of the CL-PScut1 configuration increases about 5% as compared to the CL-Baseline configuration. The CL-SScut1 configuration makes the tip leakage mass flow ratio increase approximately 11% as compared to the CL-Baseline configuration. As the cutback length increases, the mass flow rate of the mainstream entering into the tip clearance increases. It should be noted that the position of the film holes has little effect on the tip leakage mass flow ratio for a blade with partial squealers.

Figure 18 presents the influence of squealer types on the tip leakage loss at $M = 1$. It is interesting to find that the suction-side squealer cutbacks are beneficial to reduce the tip leakage loss when compared to the baseline squealer, although they produce larger tip leakage mass flow ratios. Two reasons are suggested to be associated with this result. Firstly, the leakage vortex of the SScut configurations is weaker than that of the baseline squealer, as some leakage flow leaves the tip clearance through the cutback, which reduces the mixing loss of the leakage flow and the mainstream. Secondly, the mixing loss of the coolant and the flow in the cavity is also reduced since some coolant also leaves the tip clearance through the cutback. In the squealer types studied in this paper, SScut2 configurations are demonstrated to be the best from the perspective of reducing tip leakage loss. Compared to the baseline squealer, tip leakage loss is reduced by approximately 5.6% for SScut configurations. Combined with Fig. 17, it can be concluded that the SScut2 configurations provide a strong suppressing effect on the mixing between the mainstream and leakage flow when the increase in tip leakage mass flow rate is not too much. For the pressure-side squealer cutbacks, tip leakage losses are found to be larger than the baseline squealer. In these cases,

Fig. 17. Effect of squealer types on the tip leakage mass flow ratio at $M = 1$.

Fig. 18. Effect of squealer types on the tip leakage loss at $M = 1$.

although the leakage vortex is nearly the same as the baseline squealer (Fig. 10(c) and Fig. 16(a)), the increase in leakage flow rate near the pressure-side squealer cutback results in an increase in mixing loss. Among them, the tip leakage loss of PScut1 configurations increases approximately 3.1% compared to the baseline squealer. Regarding the film-hole arrangement, the tip leakage loss of the LE mode is marginally larger than that of the CL mode for the same squealer configurations. In fact, the film holes of the LE mode are located in the high-pressure region, which increases the loss of coolant.

4.3. Effect of blowing ratios

Figure 19 presents the area-averaged film-cooling effectiveness on the blade tip at four blowing ratios. The area-averaged film-cooling effectiveness is defined as follows,

$$\eta_{av} = \frac{\int_s \eta \cdot dS}{S} \quad (8)$$

where S is the area.

It is clear that the area-averaged film-cooling effectiveness increases monotonously when the blowing ratio increases. Significantly, the amount of increase for the LE mode is much larger than that for the CL mode. For both CL and LE modes, suction-side cutback squealers are illustrated to produce higher averaged film-cooling effectiveness as compared to other squealer forms at the same blowing ratio. Among the suction-side squealer cutbacks, the cutback with the largest cutback length produces the most significant film cooling effectiveness improvement at a low blowing ratio.

Fig. 19. Effect of blowing ratios on area-averaged film-cooling effectiveness.

When the blowing ratio increases from 0.5 to 2, the maximum increase in the area-averaged film-cooling effectiveness of the CL mode is 73%, while the maximum increase for the LE mode is 108%.

Figure 20 presents the effect of blowing ratios on the tip leakage mass flow ratio. As the blowing ratio increases, the blockage effect of the coolant jets on the leakage flow entering the tip clearance is enhanced. Consequently, in general, the tip leakage mass flow ratio takes on a slight decreasing trend as the blowing ratio increases.

Figure 21 presents the effect of blowing ratios on the tip leakage loss. As the blowing ratio increases, the exhausted coolant originating from film holes increases. However, the leakage flow is suppressed due to the blockage effect of the coolant jets. The overall result is that the tip leakage loss increases only slightly. It is also demonstrated that the suction-side squealer cutbacks are conducive to reducing the tip leakage loss as compared to the baseline squealer, especially for SScut2 configurations. In combination with previous results of the film cooling performance, this study concludes that LE-SScut2 is the optimal configuration.

5. Conclusions

Detailed numerical investigations were carried out to study the aerothermal performance of various cooled squealer tips by considering the relative motion between the turbine blade and the stationary casing. The effect of partial squealer rims on the leakage flow structure, tip leakage

Fig. 20. Effect of blowing ratios on the tip leakage mass flow ratio.

Fig. 21. Effect of blowing ratios on the tip leakage loss.

loss and tip film-cooling effectiveness are illustrated. The results are as follows:

- (1) The LE mode (i.e., some film holes arranged at the

leading-edge and the remaining film holes placed close to the suction side) produces more uniform and higher tip film-cooling effectiveness when compared to the traditional CL mode with film holes located in the medial camber line. However, the arrangement of film holes causes little impact on the tip leakage mass flow ratio and tip leakage loss.

(2) Compared to the baseline squealer, a partial squealer with cutbacks near the blade trailing-edge produces a larger tip leakage mass flow ratio, especially for the suction-side squealer cutback. The pressure-side squealer cutback increases the tip leakage loss, but the suction-side squealer cutback is conducive to reducing tip leakage loss, especially for the specific SScut2 configuration with a moderate cutback length.

(3) The suction-side squealer cutbacks are more favorable for improving the tip film-cooling performance in the trailing-edge region. Additionally, the improvement effect is enhanced as the cutback length increases. From the perspective of comprehensive influence of the squealer cutbacks on aerodynamics and film cooling, based on the results of this study, LE-SScut2 is suggested to be the optimal configuration.

(4) As the blowing ratio increases, the mass flow of the coolant increases, so the area-averaged film-cooling effectiveness on the blade tip increases monotonously. The increase in the area-averaged film-cooling effectiveness, together with the blowing ratio in the LE mode, is more significant than that in the CL mode. Additionally, in general, the tip leakage mass flow ratio decreases slightly while the tip leakage loss takes on a very slight increasing trend as the blowing ratio increases.

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