Effect of rotation on film cooling with a single row of shaped holes on blade pressure side

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Abstract
A numerical investigation is performed to illustrate the mutual interaction between coolant jet issuing from shaped film cooling hole and cascade primary flow as well as the resulting film cooling performance under rotational condition. Four various film-hole geometries are utilized for comparison, including the conventional cylindrical hole, fan-shaped hole, converging slot-hole and diffused slot-hole. Results show that a strong radial flow is induced toward blade tip on the pressure side due to the rotational effect, thus affecting the interaction mechanism between the coolant jet and primary flow. In general, rotational effects on film cooling are behaved as two aspects. On one hand, it makes the coolant jet deflect toward blade tip, resulting in lateral film coverage improvement in the region adjacent to the film holes for the cylindrical hole and fan-shaped hole relative to the stationary condition. On the other hand, it weakens the flow momentum of coolant jet along the streamwise direction, causing degradation of local film cooling effectiveness far from the hole-exit except for the zone near blade tip. The shaped-hole performs favorable film cooling enhancement, especially under higher blowing ratio. Relative to the stationary case, film cooling improvement by the film-hole exit shaping is degraded a little under the rotational condition. Among the presented shaped-holes, the converging slot-hole achieves the highest film cooling effectiveness and the diffused slot-hole is the next under the same blowing ratio.

Key words: Film cooling, Shaped-holes, Rotational effect, Blade pressure side, Adiabatic film cooling effectiveness

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$C$</td>
<td>chord length of blade (mm)</td>
</tr>
<tr>
<td>$C_s$</td>
<td>axial chord length of blade (mm)</td>
</tr>
<tr>
<td>$d$</td>
<td>diameter of film hole (mm)</td>
</tr>
<tr>
<td>$D_h$</td>
<td>corresponding hub diameter (mm)</td>
</tr>
<tr>
<td>$D_t$</td>
<td>corresponding tip diameter (mm)</td>
</tr>
<tr>
<td>$H$</td>
<td>blade spanwise height (mm)</td>
</tr>
<tr>
<td>$l$</td>
<td>hole-exit slot length (mm)</td>
</tr>
<tr>
<td>$L$</td>
<td>film-hole length (mm)</td>
</tr>
<tr>
<td>$m$</td>
<td>mass flow rate (kg/s)</td>
</tr>
<tr>
<td>$M$</td>
<td>blowing ratio</td>
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<tr>
<td>$y_n$</td>
<td>normal distance from blade surface (mm)</td>
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<tr>
<td>$z$</td>
<td>$z$-direction or radial direction</td>
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<tr>
<td>$z_n$</td>
<td>radial distance originated at blade hub (mm)</td>
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<tr>
<td>$\alpha$</td>
<td>oriented angle of turbine blade (°)</td>
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<tr>
<td>$\beta$</td>
<td>relative inlet angle of primary flow (°)</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>laterally diffused angle of fan-shaped hole (°)</td>
</tr>
<tr>
<td>$\eta$</td>
<td>film cooling effectiveness</td>
</tr>
<tr>
<td>$\mu$</td>
<td>dynamic viscosity (kg/(m·s))</td>
</tr>
<tr>
<td>$\theta$</td>
<td>inclination angle of film-hole (°)</td>
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1. Introduction

The development of advanced gas turbine engines results in continual increase of the operating temperature, which necessitates more innovative cooling techniques to protect the turbine hot section components from overheating. For a modern gas turbine, the effective cooling configuration is mainly composed of the internal convective cooling and external film cooling. Among the cooling technologies, film cooling is regarded as an important issue in the turbine cooling designs. Since the film cooling air is extracted from the compressor of the engine, vast amount use of the coolant for hot component cooling will be otherwise detrimental to the engine overall efficiency. Therefore, how to improve the cooling capacity with limited coolant utilization is a tremendous challenge for the researchers.

Over the past decades, a considerable amount of investigations have been performed to reveal the fundamental physics involved in film cooling. It is commonly concluded that the film cooling effectiveness is very dependent on the interaction mechanism between the coolant jet issuing from discrete film holes and the primary flow. For the conventional cylindrical hole, the mutual interaction between coolant jet and mainstream induces a large-scale pair of counter rotating kidney vortices (Eriksen and Goldstein, 1974;Andreopoulos and Rodi, 1984). This vortices pair is regarded to be detrimental to film cooling as it tends to induce coolant jet lift-off and hence hot mainstream entrainment beneath the coolant jet. To improve film cooling potential, a number of efforts are paid to explore effective means for mitigating the effect of kidney vortices. The most inspiring and remarkable achievement is the innovative shaped-hole for film cooling improvement (Bell et al., 2000; Bunker, 2005). Some typical shaped-holes include fan-shaped (Gritsch et al., 2005; Gao et al., 2008; Lee and Kim, 2010), arrowhead-shaped (Okita and Nishiura, 2007), tabbed-shaped (Nasir et al., 2003; Yang and Zhang, 2012), converging slot-hole (Console) (Sargison et al., 2005; Liu et al., 2010; Yao et al., 2013), and combined-hole (Gao et al., 2009; Han et al., 2012), etc. The shaped-hole generates a distinct anti-kidney vortex structure which leads to less shear mixing of the coolant jet with the mainstream, weak coolant jet penetration into mainstream and greater jet spreading in the spanwise direction. For the particular console geometry, the interaction between the coolant jet and the primary flow results in a pair of anti-kidney vortices originated from the exit-slot edge in the opposite sense to the kidney vortices.
To our knowledge, previous investigations on shaped-hole film cooling were conducted mainly on the flat plate or stationary cascades. Relatively few researches were devoted to reveal the rotational effects on the shaped-hole film cooling although some studies were devoted to the rotating blade film cooling performance. Dring et al. (1980) made an experimental investigation of film cooling on a large-scale rotor blade model at a low speed of about 400 rpm. Takeishi et al. (1992) measured film cooling effectiveness on a rotating blade using a heat-mass transfer analogy. It was reported that rotation has a little influence on film cooling over the suction side (SS) while a strong radial displacement is observed on the pressure side (PS). Blair (1994) experimentally studied the pressure and suction sides as well as on the hub platform surface for a large-scale rotating turbine model. It was illustrated that the convective heat transfer is enhanced due to the secondary flow effects. Ahn et al. (2006, 2007) used the pressure sensitive paint technique to measure the leading edge film cooling with two and three rows of radially-angled holes on a rotor blade. In their experiments, the primary Reynolds number was fixed at 200,000 based on the axial chord length and the exit velocity and the rotational speed was selected as 2400, 2550, and 3000 rpm, respectively. Their results show that the rotational speed alters the film cooling flow path direction. In general, the overall film cooling effectiveness levels decrease with the increase of rotational speed. Tao et al. (2008) made an experiment to study the rotational effect on film cooling over the flat wall which was modelled as blade pressure or suction side with different rotation orientations. Five rotational speeds were considered ranging from 0 to 1000 rpm. Li et al. (2013) performed an experimental investigation of rotating film cooling effectiveness issuing from a single cylindrical hole on the pressure side or suction side in a low speed 1.5-stage turbine. Measurements were made at three different speeds of 600 rpm, 667 rpm and 702 rpm with the blowing ratio ranging from 0.3 to 3.0. It was found that the film coverage and cooling effectiveness increases monotonically on the pressure side while the trend is parabolic on the suction side as the blowing ratio increases. More recently, Rezasoltani et al. (2015) conducted a combined experimental and numerical investigation of film cooling on the turbine blade tip with four different blade tip ejection configurations. Three blowing ratios (0.75, 1.25 and 1.75) and three rotational speeds (2000 rpm, 2550 rpm and 3000 rpm) were carried out for the first 1.5 turbine stage.

With regards to the numerical investigations, Garg (1997, 1999, and 2000) performed a series of numerical investigations on film cooling over a rotating turbine blade. Different turbulence models were applied to acquire the film cooling effectiveness and heat transfer coefficient. In the simulations, only a rotor cascade domain was adopted without considering the stator-rotor interaction. At the inflow boundary, the total temperature, total pressure, whirl and flow angle were specified. It was suggested that the entire span needs to be discretized in order to correctly model the rotational body forces for a rotating blade. Yuan et al. (2007) made a numerical simulation to study the effect of rotation on turbine blade film cooling effectiveness. The results showed that the increase of rotational speed decreases the film cooling effectiveness on the pressure surface. However, the film cooling effectiveness in the region near blade tip is increased with the rotational speed. Yang et al. (2008) used the Reynolds stress model to calculate the pressure, heat transfer coefficient and film cooling effectiveness distributions in a 1.5 turbine stage. Tao et al. (2009) performed a numerical simulation on a rotating blade with a flat test surface. The computational model only included the simplified rotor cascade flow domain. Zhang and Hassan (2012) numerically simulated the film cooling performance for a louver cooling scheme at the leading edge of a rotating turbine blade and only one period was selected in the span-wise direction. Their results revealed that the louver cooling scheme indicates a nearly monotonic variation with the blowing ratio and the rotational speed while an adverse variation trend was obtained for the conventional cylindrical hole depending on different levels of these two factors. More recently, Acharya and Moreaux (2014) made a numerical study of the flow past a turbine blade tip to reveal the effect of relative motion between blade and shroud. In their computational domain, only the rotor cascade was adopted. Stationary, sliding, and rotational cases were taken into considerations to delineate the effects of relative motion and rotational forces. Alzurf and Turan (2016) investigated the effects of rotation on film cooling effectiveness and heat transfer coefficient distributions on the suction and pressure surfaces of a gas turbine blade using LES method. In their simulations, the computational domain was also chosen as a rotor cascade in the rotational reference frame.

As film cooling is tightly dependent on the mutual interaction mechanism between the coolant jet issuing from discrete film holes and the primary flow, film cooling on a rotating turbine blade becomes more complex and differs from the stationary condition. For film cooling on the rotating blade, an interesting issue is the rotational effect on film cooling performance of various shaped holes. The present numerical investigation aims at the illustration of mutual interaction between the coolant jet issuing from shaped-holes and the cascade primary flow as well as the resulting film cooling performance under rotational conditions. Four various film-holes, including the cylindrical hole, fan-shaped
hole, converging slot-hole and diffused slot-hole, are utilized for comparison under two typical blowing ratios.

2. Computational Procedures
2.1 Physical model

In the real situation of a gas turbine, the mutual interaction between the coolant jet issuing from discrete film holes and the mainstream flow is very complex. For simplicity, some simplifications are made for the current simulations. Firstly, a simplified blade model is adopted without considering the complex twist of a real rotor blade. Secondly, the film jet issuing from each film hole is directly supplied from the film-hole inlet. The internal cross flow as well as the non-uniform distribution of mass flow rate between the individual film jets is not taken into account. Thirdly, the blade tip leakage is also not taken into account. These simplifications ignore the effects of blade contour, internal cross flow and tip leakage flow on the cascade primary flow as well as the coolant jet ejecting from the film cooling hole. Although these simplifications cause some distinctions relative to the real operating situation, the essential influence mechanisms of rotational effect on shaped-hole film cooling are acquired and distinguished and therefore the above simplifications are practically adopted in wide investigations. Another major simplification in the establishment of computational model is that only a rotor cascade domain is adopted in the current study without considering the stator-rotor interaction. In our opinion, the use of a complete turbine stage or 1-1/2 stages as the computational model is often suitable for the numerical simulation when the blade rotates at a specific high speed or in the vicinity of this specific speed. However, given the high rotational speed of the real turbine, a large incidence angle is likely to be induced under the stationary condition compared with that under the design rotational condition. This absolutely alters the normal flow of hot mainstream and leads to very serious separation along the blade surface which is meaningless and undoubtedly avoided. Consequently, in order to illustrate the effect of shaped-hole on the film cooling performance under rotational conditions, particularly its different actions under the stationary and rotational condition, a rotor cascade domain seems more suitable (Garg, 1997, 1999, 2000; Yuan et al., 2007; Tao et al., 2009; Acharya and Moreaux, 2014; Alzurfl and Turan, 2016). As this computational domain is built on the rotational reference frame, the relative inlet angle of cascade mainstream is kept the same for different rotational speeds. In addition, the relative inlet flow conditions (such as total pressure, total temperature, Mach number, relative velocity, etc.) are also kept the same for different rotational speeds and uniform. In doing so, part of complicated inlet flow conditions (such as non-uniform inlet flow angle and aerodynamic parameters) due to the stator-rotor interaction is lost. However, the major factors such as body forces are still well reflected on the shaped-hole film cooling under rotational conditions. Besides, the rotational effect on the shaped-hole film cooling may be investigated ‘in relative isolation’ by separating the influence of non-uniform inlet flow condition approaching the rotor blade.

The computational domain is schematically shown in Fig. 1(a), which is composed of two sub-zones, namely, the primary flow zone inside the cascade and the coolant flow zone inside the film hole. One cascade pitch is chosen as the computational domain according to the periodic flow feature inside the cascades. The x-axis is the rotational axis of the simulated rotor blade. The y-axis and z-axis are along circumferential and radial directions, respectively. The whole blade is simulated with the spanwise height \( H \) of 50 mm and the corresponding tip diameter \( D_t \) and hub diameter \( D_h \) are 740 mm and 640 mm, respectively. The blade geometric parameters are schematically shown in Fig. 1(b). The blade model has a chord length \( C \) of 41.8 mm and is oriented with an angle \( \alpha \) of 61.7° so that an axial chord length \( C_x \) of 36.8 mm is obtained. The cascade pitch \( P \) is set as 29.7 mm. The rotational speed \( n \) ranges from 0 rpm to 15,000 rpm. In all rotational speeds, the relative inlet angle \( \beta \) is specified as 38.3°. A single row of film holes is arranged on the pressure side with 30.4% axial chord length apart from the blade leading edge.
Four film-hole geometries are utilized for comparison, including the conventional cylindrical hole, fan-shaped hole, converging slot-hole and diffused slot-hole, as shown in Fig. 2. All the holes have the same inclination angle ($\theta$) of 30° and the same inlet diameter ($d$). The film-hole length ($L$) is $4d$. Here, the cylindrical film hole is selected as a baseline case, as seen in Fig. 2(a). For the fan-shaped hole, as shown in Fig. 2(b), the diffused portion is oriented at the middle of film-hole or $2d$ from the hole-inlet and the laterally diffused angle ($\gamma$) is set as 10°, resulting in an exit-inlet area ratio of 1.91. The geometric configurations of converging slot-hole and diffused slot-hole are shown in the Fig. 2(c) and Fig. 2(d), respectively. The cross-section of either converging slot-hole or diffused slot-hole transitions from a round shape at the inlet to a slot at the exit. The slot length ($l$) and width ($w$) of the converging slot-hole exit are $2.5d$ and $0.2d$, respectively so that an exit-inlet area ratio of 0.637 is obtained. As a result, the cross-sectional area decreases from the inlet to the exit and the cooling air accelerates inside the film hole. The diffused slot-hole has the same slot length with the converging slot-hole while the slot width is $0.6d$ so that the diffused slot-hole gives an approximately equal exit-inlet area ratio to that of the fan-shaped hole. The hole-pitch ($P_{\text{hole}}$) between adjacent holes is set as $3d$ and a single row of 26 holes are involved in the present computational domain.

2.2 Parameter definitions

Since the flow inside the rotating domain can be treated as steady flow when viewed from a rotational coordinate system which rotates steadily with an angular velocity $\omega$, the absolute flow velocity can be acceptably transformed to the relative velocity based on the rotating reference according to the following relationship

\[ \vec{v}_r = \vec{v} - \vec{u}_{\text{rot}} \]

where $\vec{v}_r$ and $\vec{v}$ are the relative and absolute velocity vector respectively, and $\vec{u}_{\text{rot}}$ ($\vec{u}_{\text{rot}} = \omega \times \vec{r}$, $\vec{r}$ is the rotational radial vector) is the rotational linear velocity vector. For the rotational cases, the steady-state governing equations can be expressed using the relative velocity instead of the absolute velocity. The additional effects of the centrifugal ($-\omega \times \omega \times \vec{r}$) and Coriolis ($-2\omega \times \vec{v}_r$) forces are involved in the momentum equation due to rotation.

The mainstream Reynolds number ($\text{Re}_\infty$) is defined based on the blade chord length and mainstream inlet velocity

\[ \text{Re}_\infty = \frac{\rho_r u_{\infty} C}{\mu_r} \]

where $\rho_r$ and $u_{\infty}$ are density and velocity of the primary flow at the cascade entrance, respectively; $\mu_r$ is dynamic viscosity of the primary flow. In the current study, the inlet mainstream Reynolds number is about $5 \times 10^5$. 

For the shaped-hole, it is noted that the cross-sectional area or the coolant flow velocity varies from the hole-inlet to the hole-exit. To guarantee the same coolant mass flow for the various film-hole shapes, the coolant parameters are uniformly taken from the hole-inlet. In the present study, the coolant-mainstream blowing ratio is defined in two ways

\[
M = \frac{(\rho_c u_{c,in})_{inlet}}{(\rho_c u_{c,in})_{exit}} \tag{3}
\]

\[
M_{local} = \frac{(\rho_c u_{c,in})_{inlet}}{(\rho_c u_{c,in})_{local}} \tag{4}
\]

where \( \rho_c \) and \( u_{c,in} \) are density and relative velocity of the coolant flow or secondary flow at the film-hole inlet, respectively. The primary flow parameters used in the definition of blowing ratio are taken from the mainstream entrance in Eq. (3) while from the cascade position right at the hole-exit in Eq. (4). In the current study, two typical blowing ratios (\( M=0.5 \) and 1.5 or \( M_{local}=0.8 \) and 2.4) are considered.

The adiabatic film cooling effectiveness (\( \eta \)) is defined to characterize the film cooling performance

\[
\eta = \frac{T_{aw} - T_m}{T_{aw} - T_c} \tag{5}
\]

where \( T_m \) and \( T_c \) are temperatures of mainstream and coolant flow, respectively; \( T_{aw} \) is adiabatic wall temperature on the blade surface with film cooling.

2.3 Computational methods

The boundary conditions of computational domain are specified as the following.

The mainstream passage: the mainstream inlet is defined as pressure-inlet that constant total pressure and total temperature are given. The total pressure of the primary flow at the inlet is 1.2 MPa and the total temperature is 1790 K. In this case, the inlet mass flow rate of the present computational domain is 0.65 kg/s and the corresponding cascade inlet Mach number is 0.41. A turbulence intensity of 5% and a turbulence length scale of 3% of the blade chord length are used. The flow outlet condition is set as pressure-outlet with a static pressure of 0.85 MPa and the cascade exit Mach number is around 0.72. Periodic boundaries are applied to the cascade passage except for the blade surfaces, as shown in Fig. 1(a). All the passage walls are simulated as adiabatic walls with non-slip velocity conditions. The cascade aerodynamic parameters in the current study are summarized in Table 1.

Film holes: adiabatic no-slip condition is also applied to the film-hole wall. The cooling air is directly introduced from the film-hole inlet by adopting mass flow inlet condition with constant mass flow rate and total temperature. The total temperature is set as 775K and the inlet mass flow rate is determined according to the blowing ratio. Two typical blowing ratios, \( M=0.5 \) and 1.5 or \( M_{local}=0.8 \) and 2.4, are given for comparison.

Three-dimensional numerical simulation is performed using Fluent-CFD software. The compressibility is taken into account since the mainstream Mach number is larger than 0.3. Realizable \( k-\varepsilon \) turbulence model with enhanced wall treatment is used to model turbulence as it has been successfully used by some researchers (Yao et al., 2014; Zhang and Hassan, 2012; Harrison and Bogard, 2008; Silieti et al., 2009; Ely and Jubran, 2009; Liu et al., 2012). Convergence is achieved when both of the following criteria have been met: (a) reduction in all residuals to five orders of magnitude, and (b) no observable change in local pressure surface temperature prediction with additional 50 iterations.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tr>
<td>Cascade entrance total pressure (( P_{e,entrance} ))</td>
<td>1.2 MPa</td>
</tr>
<tr>
<td>Cascade entrance total temperature (( T_{e,entrance} ))</td>
<td>1790 K</td>
</tr>
<tr>
<td>Cascade mass flow (( m ))</td>
<td>0.65 kg/s</td>
</tr>
<tr>
<td>Cascade flow Reynolds number (( Re_c ))</td>
<td>( 5 \times 10^5 )</td>
</tr>
<tr>
<td>Cascade entrance Mach number (( M_{e,entrance} ))</td>
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<tr>
<td>Cascade exit static pressure (( P_{exit} ))</td>
<td>0.85MPa</td>
</tr>
<tr>
<td>Cascade exit Mach number (( M_{e,exit} ))</td>
<td>0.72</td>
</tr>
<tr>
<td>Blowing ratio (( M ))</td>
<td>0.5, 1.5</td>
</tr>
<tr>
<td>Local blowing ratio (( M_{local} ))</td>
<td>0.8, 2.4</td>
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</table>
Multi-block structured meshes are generated for the whole computational domain except for the local zone in the vicinity of film holes, as shown in Fig. 3(a). The meshes are non-uniform with fine grids in the regions where the complicated flow occurs, especially near the viscous walls. Intensive grids are achieved at the blade surfaces through viscous clustering and the grids are generated from the surface with a stretching ratio less than 1.2, as shown in Fig. 3(b). The local $y^+$ on the blade surface is ensured to approach the magnitude of 1. Considering the irregularity of film-hole shapes, unstructured grids are adopted in the vicinity of film holes, as shown in Fig. 3(b).

A series of grid sensitivity tests are conducted before the computations. Taking the conventional cylindrical film-hole or baseline case as an example, the effect of grid number on film cooling effectiveness along the film-hole centreline is shown in Fig. 4(a). Here, the $s$-direction is defined along the pressure surface profile and originated from the centre of the hole-exit. It seems that the centreline effectiveness only gets a negligible change as the grid number is increased from 7.43 million to 10.82 million. As a consequence, 7.43 million is finally selected as the reasonable grid number since the grid-independent result is successfully obtained. In view of the different film-hole geometries, the acceptable total grid numbers are generated between 7.43 to 7.97 million with the similar grid density.

To validate the numerical method, the experimental model of blade film cooling by Li et al. (2013) which is similar to the present computational model is adopted to carry out a numerical simulation. A single film cooling hole is positioned at 28° on the blade pressure side. Figure 4(b) presents a comparison of adiabatic film cooling effectiveness at the centreline of the film trajectory between the present computational result and the experimental result presented by Li et al. (2013) under a blowing ratio of 1.0 and a rotational speed of 600 rpm. It is shown that the computed film effectiveness is a little lower than the experimental values within the range of $1<s/d<6$ and a little higher near the hole-exit and $s/d>6$ downstream. In general, the numerical values agree with the experimental result by Li et al. (2013).

### 3. Results and Analysis

#### 3.1 Rotational effect on flow fields
Figure 5 shows the effect of rotation on the velocity vector at the cross-section positioned 4d upstream from the hole colored by the contour of normal velocity. When the blade is stationary, as seen in Fig. 5(a), the primary flow field seems to be basically symmetrical to the middle plane of the blade. A passage cross flow originated from suction side is induced toward the pressure side with almost no change along the radial direction except for two local zones near the blade hub and tip. Both in the vicinity of blade hub and blade tip, corner vortexes are formed. When the blade rotates, the symmetric distribution of relative velocity vector under stationary condition is destroyed due to the rotational effect. Strong radial flow is induced toward the blade tip on the pressure side, as seen in Fig. 5(b). This computational result illustrated that rotation will have no significant impact on the film cooling over suction side. However, a strong radial displacement of the primary flow is observed on the pressure side, which will have an important influence on the interaction mechanism between coolant jets from discrete film holes and the primary flow.

![Figure 5 Static pressure and streamline at cascade cross-section upstream from film hole](image)

Figure 6 shows the dimensionless temperature (defined as $\Theta = \frac{T_{\infty} - T}{T_{\infty} - T_c}$) and relative velocity vector at the normal section located 2d downstream from the film hole under stationary condition and blowing ratio of $M_{local} = 2.4$. Here the selected film hole is located at the middle plane of blade. In this figure, the transverse coordinate $z_n$ is defined as a radial distance with its origin located at the blade hub. The longitudinal coordinate $y_n$ is a normal distance to the blade pressure side. For the cylindrical hole, the interaction between the coolant jet and the mainstream flow results in strong kidney vortices, as seen in Fig. 6(a). The mutual interaction between the vortex pair tends to lift the coolant jet off the surface and the hot mainstream air is forced to enter beneath the coolant jet. While for the shaped holes, the kidney vortices are effectively suppressed due to their anti-vortex mechanisms. The cooling air shows good attachment on the wall at this downstream station, as seen in Fig. 6(b)-6(d). It is interesting that the converging slot-hole shows a different behavior of secondary vortices from that in baseline case and the other shaped holes. When the cooling air is ejected from the converging slot-hole, the coolant flow inside the hole forms strong lateral flow toward the narrow sides of slot, resulting in a pair of counter rotating vortices originated from the side-edges of slot with a sense of rotation opposite to the kidney vortex pair as seen in the other film-holes. In addition, the primary flow could be ingressive to the gap between adjacent converging slot-holes, resulting in additional small vortices. With regard to the diffused slot-hole, as the internal flow inside the film-hole decelerates from the inlet to the outlet, the lateral flow toward the narrow sides of slot will be weakened relative to the converging slot-hole. In some sense, the diffused slot-hole is most likely to be a fan-shaped hole with a larger laterally-diffused angle. By comparing the fan-shaped hole and the diffused slot-hole with the same exit area, the latter one shows better coolant coverage in the lateral direction.

![Figure 6 Dimensionless temperature and relative velocity vector](image)
Figure 6 shows the dimensionless temperature and velocity vector at the normal section of s/d = 2 under n=0 rpm and M_{local}=2.4. Due to a strong radial primary flow toward the blade tip induced by the rotational effect, the film jets are pressed to deflect toward the blade tip. Therefore, the interaction between the coolant jet and the mainstream flow behaves obviously different from the stationary case. By comparison, it is found that the distortion of vortex configuration is especially serious for the cylindrical hole, as seen in Fig. 7(a). Because the coolant jet issuing from the cylindrical hole is easier to be lifted off the surface, thus the radial primary flow due to the rotational effect plays more significant action on the interaction between the coolant jet and the mainstream flow. The radial primary flow also makes the vortex distortion for the shaped holes, as seen in Fig. 7(b)-7(d). However, as the coolant jet attaches to the surface well relative to the cylindrical hole, therefore, the radial primary flow plays relatively weaker action on the coolant flow from shaped holes. Among these shaped holes, it seems that the radial primary flow has the weakest effect on the coolant flow from the converging slot-hole, as seen in Fig. 7(c).

3.2 Rotational effect on local film cooling

Figure 8 presents the adiabatic film cooling effectiveness distributions on the pressure side downstream of the film cooling hole, under rotational speed of n=0 rpm and blowing ratio of M_{local}=0.8. It is obvious that the region between adjacent film-holes is not well protected by the coolant jets from the cylindrical holes, as seen in Fig. 8(a). The fan-shaped hole produces much higher film cooling effectiveness in the region adjacent to the film holes than the baseline or cylindrical hole, however, the improvement of film coverage on the region between adjacent film-holes is not well demonstrated, as seen in Fig. 8(b). With regard to the slot-holes, either in converging slot or in diffused slot, the advantage of near joining of the slot-exits is that the coolant film becomes more continuous in the lateral direction, resulting in higher and more laterally uniform film cooling effectiveness, as seen in Fig. 8(c) and Fig. 8(d).
Figure 9 presents the adiabatic film cooling effectiveness distributions on the pressure side downstream of the film cooling hole, under rotational speed of $n=15,000$ rpm and blowing ratio of $M_{\text{local}}=0.8$. Under the rotational case, the adiabatic film cooling effectiveness distribution is most similar to that produced by a row of discrete compound holes with deflection angles under stationary condition (Zhang et al. 2009). As the coolant jet is forced to deflect toward blade tip due to the rotational effect, the lateral movement of the coolant jet is enhanced, thus, resulting in larger lateral film coverage in the region adjacent to the film holes relative to the stationary case for the cylindrical hole and fan-shaped hole, as seen in Fig. 9(a) and Fig. 9(b). While for both slot-holes, film jet deflection seems to have little influence on improving the film cooling effectiveness immediately downstream of the holes, as seen in Fig. 9(c) and Fig. 9(d). On the other hand, the film jet deflection weakens its flow momentum along the streamwise direction, thus, the film cooling effectiveness far from the holes will be less than the corresponding stationary case. Besides, the film cooling distribution shows very significant difference along $z$-direction. The film coverage along the streamwise direction is more superior in the region near blade tip to that near blade hub.

Figure 10 presents the adiabatic film cooling effectiveness distributions on the pressure side downstream of the film cooling hole, under rotational speed of $n=0$ rpm and blowing ratio of $M_{\text{local}}=2.4$. By comparing this figure with Fig. 8, it is confirmed that the effect of film-hole shape on film cooling shows a similar tendency with that under small blowing ratio case. Again, the film cooling enhancement relative to the baseline case is more significant. Under a high blowing ratio, the film jet issuing from a conventional cylindrical hole will experience detached then reattached processes due to its strong normal penetration into the primary flow, leading to serious deterioration of film coverage just downstream of the film hole. While for the shaped-holes, the coolant jet normal penetration is effectively suppressed so that the jet lift-off from the surface can be avoided under a high blowing ratio. At the same time, the flow
momentum of the coolant jet along the streamwise direction is enhanced with the increasing of blowing ratio, thus, resulting in much larger and more uniform film coverage compared with the baseline case.

Figure 10 Adiabatic film cooling effectiveness distributions under $n=0$ rpm and $M_{local}=2.4$

Figure 11 presents the adiabatic film cooling effectiveness distributions on the pressure side downstream of the film cooling hole, under rotational speed of $n=15,000$ rpm and blowing ratio of $M_{local}=2.4$. It is observed that coolant jet deflection toward blade tip due to the rotational effect is somewhat weakened relative to that under a small blowing ratio as shown in Fig. 9. This result agrees with the finding of Li et al. (2013) who pointed out that the coolant jet deflection behaves more strongly with the decrease of blowing ratio.

In order to reveal the rotational effect on the film cooling performance at different lateral locations on the pressure side of turbine blade, the local laterally-averaged film cooling effectiveness within a single hole pitch following the centerline trace of the film cooling fluid from one hole is derived along the streamwise direction, as shown in Fig. 12. Three typical lateral positions are adopted, including the second hole adjacent to blade hub, the central hole and the second hole adjacent to blade tip.

Under a small blowing ratio of 0.8, the film cooling effectiveness is obviously improved due to the rotational effect downstream of the film hole-exit within a region of $s/d<5$ for the cylindrical hole relative to the stationary case regardless of the location. For the shaped holes, the influence of rotational effect on film cooling down the film-hole exit is weakened, especially for the slot-holes. In addition, either under stationary or rotational conditions, all the shaped holes show great advantage over the cylindrical hole. The rotational effect on film cooling far from the film hole is more complicated. Near the blade hub, the rotational effect makes the local laterally-averaged film cooling
effectiveness reduce a lot relative to the stationary case, as seen in Fig. 12(a) left. This influence is more serious for the slot-holes. At the mid-span, the rotational effect seems to have a little negative influence on the film cooling, as seen in Fig. 12(b) left. While near the blade tip, the local laterally-averaged film cooling effectiveness is a little increased by the rotational effect, as seen in Fig. 12(c) left.

The same affecting tendency of rotational effect on the local laterally-averaged film cooling effectiveness is also demonstrated under a high blowing ratio of 2.4. By comparing with $M_{local}=0.8$ case, it is found that the positive affection of rotational effect on improving the film cooling downstream of the film hole is extended to $s/d\leq15$ for the cylindrical hole in the region either near the blade hub or mid-span, as seen in Fig. 12(a) right and Fig. 12(b) right. At the mid-span, the reduction of local laterally-averaged film cooling effectiveness due to the rotational effect relative to the stationary case for the converging slot-hole seems the weakest among three shaped holes, as seen in Fig. 12(b) right. However, near the blade tip, the local laterally-averaged film cooling effectiveness is improved obviously by the rotational effect for the shaped-holes, as seen in Fig. 12(c) right.

![Graphs showing film cooling effectiveness distributions](image)

**Fig. 12 Local laterally-averaged film cooling effectiveness distributions within one hole-pitch**

### 3.3 Rotational effect on total laterally-averaged film cooling effectiveness

Figure 13 presents the effect of rotational speed on the total laterally-averaged film cooling effectiveness along the
streamwise direction. For the cylindrical hole, it is evident that the film cooling just downstream of the film-hole is higher under a relatively lower blowing ratio, as seen in Fig. 13(a). The coolant jet has weak penetration capacity under low blowing ratio, which facilitates the coolant jet covering on the surface. However, the spreading capacity of the coolant jet along the streamwise direction is also weak under low blowing ratio, leading to a greater decay of film cooling effectiveness far from the hole. Under the blowing ratio of 2.4, the coolant jet is easy to be lifted off the protected surface and then reattaches on the downstream surface, producing worse film cooling closed to the film hole. As the rotational speed increases, the total laterally-averaged film cooling effectiveness immediately downstream of the film hole is improved due to the lateral movement of the coolant jet. The film cooling improvement of cylindrical hole due to the rotational effect behaves more obviously under high blowing ratio and rotational speed. The rotational effect on the total laterally-averaged film cooling effectiveness of the fan-shaped hole is somewhat similar to that of cylindrical hole as seen in Fig. 13(b). That is, the coolant jet deflection due to the rotational effect improves the total laterally-averaged film cooling effectiveness right downstream of the film hole. As the fan-shaped hole is capable of suppressing the coolant jet normal penetration and producing better film coverage in the region adjacent to the film hole relative to the cylindrical hole, there is only a slight improvement on the total laterally-averaged film cooling effectiveness. Under the blowing ratio of 2.4, it is also notable that the total laterally-averaged film cooling effectiveness gradually decreases with the increase of rotational speed in the region beyond 10d away from the film hole (s/d > 10). With regard to both slot-holes, as seen Fig. 13(c) and Fig. 13(d), the total laterally-averaged film cooling effectiveness decreases gradually with the increasing rotational speed despite of the blowing ratio. In addition, the rotational effect seems have no influence on the total laterally-averaged film cooling effectiveness in the region immediately downstream of the film hole relative to the stationary case. In general, the total laterally-averaged film cooling effectiveness is more sensitive to the rotational speed under a large blowing ratio.

Figure 14 shows the comparison of total laterally-averaged film cooling effectiveness distribution along the streamwise direction for the four film-hole geometries. It is evident that the shaped-hole performs favorable film cooling enhancement, either for stationary or rotational conditions, especially under a higher blowing ratio. It is also illustrated that the converging slot-hole achieves the highest film cooling effectiveness and the diffused slot-hole is the next. Relative to the stationary case, the improvement of the film cooling by the film-hole exit shaping seems to be
degraded a little under the rotational condition. For example, the total laterally-averaged film cooling effectiveness of fan-shaped hole is approximately 100% higher than that of the cylindrical hole right at the hole exit under $M_{local}=0.8$ and $n=0$ rpm. When the blade rotates at 15,000 rpm, the laterally-averaged film cooling effectiveness of fan-shaped hole is around 70% higher than that of the cylindrical hole.

![Fig. 14 Comparison of total laterally-averaged film cooling effectiveness for four film-hole configurations](image)

Looking at the above results, it seems that rotation is a second order effect when compared with the influence of blowing ratio. This conclusion is well consistent with the previous finding presented by Tao et al. (2011) who suggested that changing blowing ratios induces relatively larger variations of the film cooling effectiveness compared with the rotational speeds.

4 Conclusions

Film cooling issuing from shaped film-holes is numerically investigated on pressure side of a rotating turbine blade. Four various film-hole geometries are utilized for comparison, including the conventional cylindrical hole, fan-shaped hole, converging slot-hole and diffused slot-hole. Particular attention is paid to explore the rotational effect on film cooling behaviors under two typical blowing ratios ($M=0.5$ and 1.5 or $M_{local}=0.8$ and 2.4). The results are summarized as the following:

(1) Due to the rotational effect, the symmetric distribution of streamline and static pressure under stationary condition is destroyed by a strong radial flow toward the blade tip, especially on the pressure side. This additional radial flow plays an important influence on the interaction mechanism between the coolant jet and the mainstream flow.

(2) The rotational effect makes the coolant jet deflect toward blade tip and also weakens the flow momentum of coolant jet along the streamwise direction, thus playing very complex roles on the local film cooling performance over the pressure side of a blade. In general, the rotational effect significantly improves the film cooling performance of the cylindrical hole in the region adjacent to the film holes. On the other hand, the rotational effect causes a degradation of the local film cooling effectiveness far from the hole except for the zone near blade tip. The rotation is a second order effect when compared with the influence of blowing ratio.

(3) The shaped-hole performs favorable film cooling enhancement, either for stationary or rotational conditions, especially under a higher blowing ratio. Among the presented shaped-holes, the converging slot-hole achieves the highest film cooling effectiveness and the diffused slot-hole is the next. Relative to the stationary case, the improvement of the film cooling by the film-hole exit shaping seems to be degraded a little under the rotational condition.

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