Numerical Investigation on Film Cooling Characteristics from a Row of Holes with Ridge-Shaped Tabs

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Abstract

Three-dimensional numerical simulation was conducted to investigate the enhanced cooling performance caused by ridge-shaped tabs located along the upstream edge of the film cooling holes. Three covering ratios of ridge-shaped tab on film hole and four blowing ratios were considered in the present. The results show that the presence of ridge-shaped tabs in the nearby region of the primary film cooling holes mitigates the primary vortices due to mainstream-coolant jet interaction and transfers the higher coolant jet momentum flux to streamwise direction mainly. The coolant jet penetration along vertical direction is suppressed and the peak velocity along streamwise direction is augmented under the action of ridge-shaped tabs, providing an increment in the film cooling effectiveness and enhancement of heat transfer coefficient over the baseline case. The ridge-shaped tabs provide enhancements in cooling effectiveness, but this is at the expense of larger pressure drop. It is suggested that the ridge-shaped tab with middle covering ratio should be the best choice in the present study.

Key words: Film cooling; Ridge-shaped tab; Cooling effectiveness; Heat transfer coefficient; Discharge coefficient

1. Introduction

The demand of advanced propulsion systems with a higher thrust to weight ratio has resulted in continual increase of the operating temperature, which leads to a greater thermal stress imposing on the combustor liner and turbine blade. Thus highly active cooling system is essential to protect the turbine components from overheating. Film cooling, as an efficient technique, has been employed to protect the external surface of gas turbine blades from the hot mainstream gas by ejecting the internal coolant air through discrete holes or slots at several locations on the blade exterior surface. Since the film coolant is extracted from the compressor of the engine, film cooling configuration has to be properly designed and optimized to minimize the thermal efficiency loss.

Over the past thirty years, a considerable amount of investigations have been performed in order to understand the fundamental physics involved in film cooling so that the design of film cooling systems can be optimized to produce the most effective film cooling with a minimum amount of coolant. The studies (Andreopoulous and Rodi, 1984; Gogineni et al., 1996) on the field of interaction between coolant jet from discrete film holes and the mainstream flow revealed that the interaction results in the formation of kidney vortices, i.e. a pair of counter rotating vortices. These vortices are detrimental to film cooling because they bring about two undesirable effects. Firstly, the hot mainstream air is forced to enter beneath the coolant jet, thus heating the plate wall. Secondly, the mutual interaction between the vortex pair tends to lift the coolant jet off the plate surface which diminishes the film cooling. To mitigate the effect of kidney vortices that causes the coolant jet to lift off, some innovative "shaped" film cooling holes (such as fan shaped hole, cratered film hole, converging slot-hole, upstream tab orientation) were present and have been explored, some in great depth and others to a lesser extent. A recent review of shaped film cooling and its effects may be found in Bunker (2005). Haven et al. (1997) found that there is an anti-kidney vortices effect in shaped film holes. Shaped holes control the mainstream-coolant jet interactions by reducing the jet momentum with keeping the mass flow rate constant, thus, producing higher film effectiveness. Gritsch et al. (1998; 2005) have presented adiabatic effectiveness and heat transfer measurements, and Thole et al. (1996) have presented flow field measurements on fan shaped holes that expand...
laterally and forward near the hole exit. The expansion of the fan shaped holes increases the lateral spread of the coolant film downstream of the holes and minimizes the penetration of the coolant flow into the mainstream. Fric and Campbell (2002) presented a cratered film hole in which the circular hole exits into a shallow right circular surface depression. The flow actually impinges on the edge of this depression causing it to deflect and fill the depression prior to issuing onto the external surface. Bunker (2002) also reported the similar idea of trenched film hole geometry. Sargison et al. (2002a, b) demonstrated a so-called console film cooling geometry (Converging Slot-Hole). In this geometry, the hole transitions from circular to slot with convergence in the axial direction and divergence laterally. This accelerated flow is speculated to have lower jet turbulence and more stability. Detailed flow structures of console film cooling flow field were also investigated by Azzi and Jubran (2007), Yao and Zhang (2011), and Liu et al. (2009). These shaped holes have been proved to provide higher film cooling effectiveness than conventional cylindrical holes in the region downstream of the coolant jet.

An effective tabbed hole for improving film effectiveness was demonstrated by Ekkad et al. (2000). They placed discrete tabs on injection holes angled at 35° along the streamwise direction and examined the effect of tab locations (the tabs were located along either the upstream edge or the downstream edge of the hole or symmetrically along the spanwise edges) on the film-cooling behavior. It was observed that placing the tabs along the upstream edge of the hole had the best performance. Nasir et al. (2003) made an experimental test to explore the effect of upstream tab orientations (parallel to the surface, oriented downwards at 45° and oriented upwards at 45°, respectively, as seen in Figure 1) on the film-cooling behavior. Results showed that the tabs oriented downwards provide the highest effectiveness at a blowing ratio of 0.56 while the tabs oriented horizontally provides the highest film effectiveness at blowing ratios of 1.13 and 1.7. Li et al. (2006) made a numerical computation on cylindrical film holes with the additional of triangular tabs covering the upstream edge of the holes. The tabs performed in the above studies may be viewed as sheet-shaped tabs. Such tabs are very thin, so that they are impossible to be adopted in the practical applications.

Ridge-shaped tabs embedded inside film holes were present by Yang and Zhang (2012). Experimental study was conducted by the authors to investigate the cooling performance of cylindrical film holes with ridge-shaped tabs covering the upstream edge of the holes. The results showed that the presence of ridge-shaped tabs provided an increment in the film cooling effectiveness and also enhanced heat transfer coefficient over the cylindrical film holes. To further understand the mechanism on improvement of film cooling effectiveness using ridge-shaped tabs and explore the effect of ridge-shaped tab on the coolant jet flow features, a three-dimensional numerical simulation was conducted in the present paper.

Figure 1 Sheet-shaped tab configurations studied in reference (Nasir H et.al, 2003)

2. Computation Scheme
2.1 Physical model

The physical model for a flat plate with a row of cylindrical inclined holes with ridge-shaped tabs is shown in Figure 2, and the baseline case is a row of cylindrical inclined holes. The computational domain is composed of three sub-zones, such as primary flow zone, perforated solid zone, and secondary flow zone. The film holes are located 50mm downstream of the edge of the primary passage inlet. The diameter of each hole (d) is 6mm and the spacing between adjacent holes is 2d. The holes are inclined 35° in the streamwise direction (x-direction). The height of primary flow passage (in y-direction) is chosen as 60mm, the width of the domain (in z-direction) is chosen as one spanwise hole-to-hole pitch and two symmetry plane are applied to the domain boundary. The coordinate origin is located at the
The ridge-shaped tab is a special tetrahedron. Its horizontal delta-alike top plane covers partly the film hole outlet and the bottom apex originates at the front edge of film hole inlet. These ridge-shaped tabs are placed on the upstream side of a row of film holes, which make the cross-sectional area of film hole decrease and the coolant stream flow accelerate from inlet to outlet. Three ridge-shaped tab configurations are presented (Table 1). Each side length \(a\) of the triangular top planes is ranged from 3mm to 5mm. Correspondingly the covering ratio \(B\) of the tab on film hole is about 0.1 to 0.4. The baseline case is a row of cylindrical inclined holes without tabs, marked as Case0.

### Table 1 Ridge-shaped tab configurations

<table>
<thead>
<tr>
<th></th>
<th>side length ((a))</th>
<th>covering ratio ((B))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case0</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Case1</td>
<td>3</td>
<td>11%</td>
</tr>
<tr>
<td>Case2</td>
<td>4</td>
<td>21%</td>
</tr>
<tr>
<td>Case3</td>
<td>5</td>
<td>39%</td>
</tr>
</tbody>
</table>

#### 2.2 Parameters definition

To study the effect of various amount of coolant flow for a fixed mainstream flow, a parameter known as the blowing ratio \(M\) is defined.

\[
M = \frac{(\rho_c u_c)_{inlet}}{\rho_c u_c} \tag{1}
\]

Where \(\rho_c\) and \(u_c\) are density and velocity of the secondary flow or coolant flow respectively at inlet of film hole, and \(u_c\) are density and velocity of the primary flow respectively.

The average adiabatic film cooling effectiveness \(\eta_{ad}\) is defined as:

\[
\eta_{ad} = \frac{T_c - T_{aw}}{T_c - T_{\infty}} \tag{2}
\]

where \(T_c\) is coolant flow temperature; \(T_\infty\) is primary flow temperature; \(T_{aw}\) is adiabatic wall temperature at the plate surface suffering the primary flow.

Heat transfer coefficient ratio \(E_g\) is defined as:
\[ E_g = \frac{h}{h_0} \]  
(3)

Where \( h \) is the local heat transfer coefficient with film cooling; \( h_0 \) is the heat transfer coefficient without film holes on a flat surface.

\[ h = \frac{q}{T_w - T_{aw}} \]  
(4)

where \( T_w \) is the wall temperature when the wall being heated, \( q \) is the heat flux imposing on the heated wall.

The discharge coefficient \( (C_d) \), which is inversely proportional to the pressure drop across the coolant hole, is defined as:

\[ C_d = \frac{m_c}{A_c \sqrt{2\rho_c (P_c^* - P_c)}} \]  
(5)

where \( P_c \) is the coolant flow total pressure at film hole inlet; \( P_c^* \) is the coolant flow static pressure at film hole outlet; \( A_c \) is the inlet area of film hole; \( m_c \) is coolant massflow.

### 2.3 Computational procedure

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

The primary flow passage: the primary flow inlet is defined as velocity-inlet with \( u_\infty = 25 \text{m/s} \), and inlet temperature of the primary flow \( T_\infty \) is 353K. A turbulence intensity of 0.5% and a turbulence length scale of 3% of the inlet height are used. And flow outlet condition is set as pre-sure-outlet with static pressure \( p_{out} = 101325 \text{Pa} \).

Film holes: the coolant jet is directly introduced into film holes from the bottom of the perforated plate by adopting velocity-inlet. The temperature of secondary flow \( T_c \) is set as 300K, and the inlet velocity can be calculated according to the blowing ratio ranged from 0.5 to 2.0.

Perforated plate: adiabatic no-slip condition is applied for the solid wall boundaries except for the interface between the primary flow and perforated plate. The conductivity of the perforated plate is set as 0.01W/(mK), so that the contributions of conduction and internal cooling inside the solid wall to the film cooling effectiveness could be negligible.

Symmetry planes: the symmetry planes are assumed to be adiabatic modeled with zero heat flux. The plane is given zero velocity in the z direction \((w=0)\) thus preventing the flow from crossing the boundary but yet allowing a velocity profile to develop.

The thermal boundary on the plate surface is set as adiabatic or constant heat flux \( q=5000\text{W/m}^2 \). The former is used to determine the adiabatic wall cooling effectiveness, and the latter is used to determine the heat transfer coefficient ratio.

Three-dimensional numerical simulation is employed by using Fluent-CFD software, and the Realize k-\( \varepsilon \) turbulence model is applied. Convergence is considered achieved when both of the following criteria have been met: (a) reduction in all residuals of five orders of magnitude, and (b) no observable change in surface temperature prediction for an additional 30 iterations.

### 3. Results and Analysis

To illustrate the mechanism on improvement of film cooling effectiveness using ridge-shaped tabs, a comparison between Case0 and Case2 is presented here. Figure 3 presents the secondary flow vectors and isotherm contours on the streamwise planes for a representative blowing ratio (such as \( M=2.0 \)). As the coolant is discharged with a certain angle, the coolant flow velocity components from cylindrical film holes can be mainly divided into two parts: the tangent velocity (in x-direction) and normal velocity (in y-direction). It is known that the interaction between coolant jet from inclined film holes and the mainstream flow results in the formation of kidney vortices. These vortices are detrimental to film cooling because they bring about two undesirable effects as mentioned in Section 1. With the ridge-shaped tabs located upstream of the hole center, although the kidney vortices are also remained, it is interest to find that the coolant jet from the holes embedded ridge-shaped tabs could be pressed to the sides downstream of the film hole, which is benefit to suppressing the coolant penetration at center of film hole and enlarging the film lateral flow.
Figure 3 Velocity vectors and isotherm contours on the normal plane ($M=2.0$)

Figure 4 presents the velocity profiles downstream of the film hole $x/d=1$ for the Case0 and Case2. Figure 5 shows the location of peak velocity at $x/d=1$ for different blowing ratios. For the cylindrical inclined holes without ridge-shaped tabs (Case0), velocity profile shows an inflection due to the coolant jet just downstream of injection at $x/d=1$ with the peak velocity around $y/d=0.75$, and the location of peak velocity is closer to the wall ($y/d=0.55$) and the peak velocity is bigger for the ridge-shaped tabs (Case2), indicating that the coolant jet penetration along $y$-direction is reduced and spread along $x$-direction is augmented in the presence of the ridge-shaped tabs. It is noted that although the coolant jet momentum flux ratio ($L=(\rho_{\infty}u_{c,\text{jet}}^2)/(\rho_{\infty}u_{\infty}^2)$, here $u_{c,\text{jet}}$ is velocity of the coolant flow at outlet of film hole) for the Case2 is greater than that for Case0 at the same blowing ratio, owing to the outlet is partial covered by the ridge-shaped tab, the higher coolant jet momentum flux for Case2 does not result in greater coolant jet penetration in vertical direction ($y$-direction).

From the above, the mechanism on improvement of film cooling effectiveness using ridge-shaped tabs could be contributed two aspects. Firstly, greater tangent velocity is expected to maintain wall jet momentum to flow over the surface of the film plate. Secondly, the coolant flow penetrates the primary flow in the normal direction is effectively
suppressed. As expected, the lower coolant jet penetration along y-direction and higher spread along x-direction will be of benefit to film cooling effectiveness.

Figure 6 presents the detailed film cooling adiabatic temperature distributions for the Case0 and Case2 under $M=0.5$ and 1.5. For Case0 (no ridge-shaped tabs), the adiabatic temperature distributions are lower along the coolant jet centerlines, with almost no cooling between the holes for most of the plate length. When the blowing ratio is higher, the coolant jet appears to lift off at the downstream edge and reattach farther downstream, creating a higher temperature region just downstream of the holes under $M=1.5$. After reattachment, the coolant jets spread laterally and full spanwise coverage is observed around $x/d=8$. While under littler blowing ratio, the coolant jet penetration normally to the primary flow is weaker, the phenomenon of coolant jet lift off and reattach the surface is not appeared. The present temperature distribution feature for no-tab case is consistent with results from earlier studies (Gritsch et al., 1998; Nasir et al., 2003). In contrast to the no-tab case, ridge-shaped tabs show much higher film cooling effectiveness than Case0 over the entire region, especially under higher blowing ratio, complete coverage of film outflow is achieved within 3~4 times hole diameter under $M=1.5$.

![Figure 6 Film cooling adiabatic temperature distributions](image)

Figure 7 presents the laterally averaged film cooling effectiveness distributions along streamwise direction for the Case0 and Case2. In this figure, the experimental data are taken from Yang and Zhang (2012). By comparison, the computational results are consistent to the experimental results. For the no-tab case, the higher cooling effectiveness is achieved under relative lower blowing ratio, and the highest effectiveness occurs just downstream of the holes. When $M=0.5$, the coolant jet has lower penetration capacity, which is helpful to made the coolant jet covering the surface just downstream of the holes. But the maintain capacity of jet spread along streamwise direction is also lower under this lowest blowing ratio, leading to a greater decay of film cooling effectiveness along streamwise direction. Once the ridge-shaped tabs adopted, the varying trends of film cooling effectiveness versus blowing ratio are changed in contrast to the no-tab case, the highest cooling effectiveness occurs at higher blowing ratio. As mentioned earlier in section 3.1, the ridge-shaped tab is capable of reducing the self-induction of vorticity and transferring the higher coolant jet momentum flux to streamwise direction mainly. The coolant jet is closer to the surface and is therefore responsible for the higher film cooling effectiveness. The rapid lateral coverage of film is also indicative of enhanced film cooling effectiveness.
Figure 8(a) presents the effect of ridge-shaped tab covering ratio (ranged from 0.11 to 0.39) on the laterally averaged film cooling effectiveness distributions under different blowing ratios. Under $M=1.0$, the ridge-shaped tab covering ratio has greater effect on the laterally averaged film effectiveness distributions, the best result for effectiveness is obtained for Case3 with covering ratio of 0.39. Under $M=2.0$, the ridge-shaped tab covering ratio has weak effect on the laterally averaged film effectiveness distributions in the present study. Figure 8(b) presents the effect of ridge-shaped tab covering ratio on the laterally averaged heat transfer coefficient ratio distributions. Comparing the three cases with ridge-shaped tabs, it is evident that Case3 has the highest heat transfer coefficient ratio and is correlated to the highest turbulence intensity values obtained for this case.

Figure 9 presents the velocity profiles downstream of the film hole $x/d=2$ for the Case1, Case2 and Case3 under blowing ratio $M=1.5$. The peak velocity is located at $y/d=0.78$ with $u/u_\infty=1.6$ for Case1. While for Case3, the peak value
of \(u/\infty\) is reached to 1.8 and the location of peak velocity is closer to the wall \((y/d=0.54)\). As the ridge-shaped tab covering ratio increases, the effects of ridge-shaped tab on suppressing coolant jet penetration and transferring coolant jet momentum flux to streamwise direction are more obvious.

The discharge coefficients are plotted in Figure 10. The discharge coefficients is inversely proportional to the pressure drop across the coolant hole, so there is a larger pressure drop with the ridge-shaped tabs relative to the baseline case, especially for the ridge-shaped tabs with bigger covering ratio. The ridge-shaped tabs do provide enhancements in cooling effectiveness, as seen earlier, but this is at the expense of considerably greater pressure drop. It is important to note here that the ridge-shaped tabs with lower covering ratios provide similar levels of discharge coefficients as that shown by Gritsch et al. (1998) for their shaped holes.

Figure 9 Velocity profiles downstream of film hole at \(x/d=2\)

Figure 10 Effect of ridge-shaped tab on discharge coefficients

4. Conclusions

The influence of placing a ridge-shaped tab along the upstream edge of the film cooling hole on the film effectiveness, surface heat transfer coefficient and discharger coefficient has been numerically investigated. Three different ridge-shaped tab configurations have been tested. The results are summarized as follows:

1. The presence of ridge-shaped tabs in the nearby region of the primary film cooling holes mitigates the primary vortices due to mainstream-coolant jet interaction and transfers the higher coolant jet momentum flux to streamwise direction mainly. The peak velocity is bigger and the location of peak velocity is closer to the wall compared with baseline case. As the ridge-shaped tab covering ratio increases, the effect of ridge-shaped tab on suppressing coolant jet penetration along vertical direction is more obvious.

2. With the ridge-shaped tabs, the coolant jet is closer to the surface and is therefore responsible for the higher film cooling effectiveness. The rapid lateral coverage of film is also indicative of enhanced film cooling effectiveness. The ridge-shaped tab covering ratio has greater effect on the laterally averaged film effectiveness distributions under lower blowing ratio.

3. The heat transfer coefficient ratio is bigger for the ridge-shaped tabs, indicating that the mixing of the mainstream and coolant jet provides a better coverage film but increases local turbulence production enhancing the heat transfer coefficient.

4. There is a larger pressure drop with the ridge-shaped tabs relative to the baseline case, especially for ridge-shaped tabs with bigger covering ratio. As the coolant jet Reynolds number increases, the discharge coefficients for all the cases increase.

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