Large eddy simulation of film cooling on turbine vane

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Abstract
Large eddy simulations were performed for film cooling on scaled-up C3X turbine vane at the nominal blowing ratio of \(M=0.5\text{−}1.5\), and the Reynolds number, \(Re=3000\), based on the mainstream inlet velocity and hole diameter. On the pressure surface, large-scale coherent structures including hairpin vortexes and horseshoe vortexes are generated in film-cooling flow fields. Hairpin vortexes promote the mixture between hot mainstream and coolant jet and degrade cooling performance. The anti-entrainment of horseshoe vortexes improves the lateral-covering capability of coolant jet in the near-field region and results in the formation of a pair of low-temperature strips wrapped around the hole at high blowing ratio. On the suction surface, the transition of the boundary layer takes place in the downstream of cascade throat but in the upstream of the discharged hole, plenty of broken vortexes dominate film-cooling flow fields. The distribution of turbulent kinetic energy also indicates that the coolant jet from the suction surface generates higher turbulent intensity than that from the pressure surface. For pressure signals for film cooling on the suction surface, small-scale and random fluctuation takes the dominant role. For pressure signals for film cooling on the pressure surface, a dominant frequency corresponding to Strouhal number \(St\approx2.1\) both exists at low and high blowing ratios. The film-cooling system on the suction surface exhibits a higher random degree than that on the pressure surface.

Keywords: Film cooling, Gas turbine, Boundary layer, Large eddy simulation, Coherent structures

1. Introduction

Due to exceptionally high turbine inlet temperatures, high-efficiency gas turbine engines must employ active cooling schemes in order to keep metal components within allowable operational temperature ranges. Film cooling is one of the most efficient cooling schemes for turbine blades. In film cooling, cooler air is ejected through the blade surface into the external boundary layer and forms a relatively cool insulating film on the blade surface, which protects the blade surface from coming in contact with high gas temperature effectively. Film-cooling flows belong to the jet-in-crossflow (JICF) class. It has been widely recognized that the flow in the vicinity of the discharged holes is particularly complex due to the interaction between the coolant jet and the surrounding laminar or turbulent boundary layer. A detailed understanding of the flow physics is a must to improve existing film cooling techniques (Bogard et al., 2006).

The flow patterns for film cooling are characterized by the development of a horseshoe-like vortex wrapped around the jet exit, a counter-rotating vortex pair (CRVP) dominating the far field, roller vortexes existing in the shear layers between mainstream and coolant jet, and small-scale wake vortexes. In LES results from Ziefle and Kleiser (2008), Tyagi and Acharya (2003), Peet (2008), Sarkar and Babu (2015), Sakai et al. (2014), Zhong et al. (2016) and Wang et al. (2018a, 2018b), hairpin vortexes are the dominant 3-dimensional coherent structures in the downstream of the round hole, and the head, horizontal legs and vertical legs of hairpin vortexes are related to roller vortexes, CRVP and upright wake vortexes respectively. Sarkar and Babu (2015) reported that a low value of effectiveness is observed below the head of the hairpin and immediate upstream due to jet lift-off; furthermore, the surface between two consecutive hairpins appears to have relatively high effectiveness as the horizontal legs of hairpin come closer to the wall. Zhong et al. (2016) indicated that a steady core with low-temperature variance appears near the exit of the round hole, and high-temperature variance appears around this steady core. However, in large eddy simulation (LES) results from
Renze et al. (2008), Guo et al. (2006) and Konopka et al. (2013), hairpin vortexes are replaced by a large number of broken vortexes, and horseshoe vortexes also become unobvious. It might be because that, in these cases, the mainstream turbulent boundary layer is fully-developed before reaching the discharged hole. Yuan et al. (1999) indicated that CRVP originates from a pair of hanging vortexes in the skewed mixing layer between the jet and the crossflow. Guo et al. (2006) reported that CRVP is caused by the shearing effect of the jet-crossflow interaction and the streamwise oriented vortexes contained in the near-wall layer of the jet and the crossflow. Foroutan et al. (2015) reported that CRVP forms due to the roll-up of the shear layers on the lateral sides of the jet, and grows in size in its evolution process. Peet (2008) reported CRVP brings hot crossflow fluid underneath the jet by strong recirculating motion and enhances mixing between the jet and crossflow. Li et al. (2017a) found that CRVP origination varies with varied length-to-diameter ratio. The origination is mainly spiral vortexes and the Kelvin–Helmholtz structures for holes with small length-to-diameter ratio. For a moderate length-to-diameter ratio, the CRVP originates from the emerging of in-tube vortex pair and Kelvin–Helmholtz vortex. However, for a high length-to-diameter ratio, CRVP mainly comes from Kelvin–Helmholtz vortex. Li et al. (2017b) also indicated that, for internal crossflow case, hole exit velocity rather than turbulence intensity determines the film cooling performance, and the uniform hole exit velocity distribution provides favorable influence on the film effectiveness.

Film cooling on the turbine blade is much more complicated than that on a flat plate. Besides the flow detachment/reattachment caused by blade curvature, turbulence also plays a major role on the heat transfer, and unsteady heat transfer characteristics are strongly correlated with the coherent vortex dynamics (Kim, 2008). The early experimental studies by Drost et al. (1998) showed that mainstream turbulence had a weak influence on the suction surface cooling, but higher effectiveness was noted on the pressure surface at high turbulence. The transition behavior of the boundary layer and the high unsteadiness of the separation process tend to limit the predictive capability of traditional Reynolds Average Navier-Stokes (RANS) simulations. York and Leylek (2003), Laskowski et al. (2008), Fachini et al. (2004), and Luo and Razinsky (2007) evaluated the C3X vane of Hylton et al. (1983) using different computational fluid dynamics (CFD) codes. Significant differences between the predicted and measured surface temperatures were found in all four computational studies, particularly on the suction side of the vane where the transition to turbulence occurs. To solve this problem, LES has been used for film cooling on the turbine blade. AI-Zurfi and Turan (2016) investigated the rotation effect on cooling performance in the turbine blade with a row of air film injection by LES. They reported that film cooling effectiveness increases with the rise of rotating speed, and the rotation promotes an earlier boundary layer transition and increases the transition length on the suction surface. Shi et al. (2013) investigated the unsteadiness behavior of film cooling on MT1 turbine vane at the transonic condition. In their LES results, without film cooling, coherent vortexes of shock-induced transition exist on blade suction surface near the trailing edge; however, with film cooling, the surface boundary layer on the jet exit directly transits to turbulence, and the vortexes are stretched by the reverse transition.

In this paper, LES with the Smagorinsky-Lilly model was introduced firstly; and then LES results for film cooling on a scaled-up C3X turbine vane were tested by previous open-published experimental data; finally, coherent structures and their influences on film cooling were analyzed in detail.

2. Computational procedures
2.1 LES Model

Governing equations are the grid-filtered, conservative equation, compressible N-S equations, energy equation:

\[
\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0
\]

\[
\frac{\partial (\rho u_i)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = \frac{\partial}{\partial x_j} (\mu \frac{\partial u_i}{\partial x_j}) - \frac{\partial p}{\partial x_i} - \frac{\partial \tau_{ij}}{\partial x_j}
\]

\[
\frac{\partial (\rho \tilde{h})}{\partial t} + \frac{\partial (\rho \tilde{h} \tilde{h})}{\partial x_j} (\rho u_i \frac{\partial \tilde{h}}{\partial x_j} - \tau_{h,i,j}) = 0
\]

where, a line over variable indicates a grid-filtered quantity. The variables \( \rho, u, p \) and \( h \) denote the density, velocity,
pressure and enthalpy of gas respectively. \( t \) denotes time. \( \tau \) is the shear stress tensor and expressed by:

\[
\tau_{ij} = -\frac{1}{3} \tau_{kk} \delta_{ij} = -2 \mu_S \tilde{S}_{ij}
\]  

(4)

where, \( \tilde{S}_{ij} \) is the rate-of-strain tensor and can be expressed by:

\[
\tilde{S}_{ij} = \frac{1}{2} \left( \frac{\partial \tilde{u}_i}{\partial x_j} + \frac{\partial \tilde{u}_j}{\partial x_i} \right)
\]  

(5)

Subgrid enthalpy flux, \( \tau_{h,j} \), in Eq.3 and subgrid-scale viscosity, \( \mu_t \), in Eq. 4 and are determined by the Smagorinsky-Lilly mode (Smagorinsky, 1963):

\[
\mu_t = \rho (C_s \Delta_s)^2 \left| \tilde{S} \right|
\]  

(6)

\[
\tau_{h,j} = \frac{\mu_t}{Pr_t} \left( \frac{\partial \tilde{h}}{\partial x_j} \right)
\]  

(7)

where, \( \Delta_s \) is the characteristic length. \( C_s \) is the Smagorinsky constant, and \( C_s=0.1 \) in the present simulation. Turbulent Prandtl number, \( Pr_t \), is set to be 0.9 in the present study.

2.2 Computation domain and boundary condition

The vane model is shown in Fig. 1. The airfoil geometry used was a 3.88 times scale geometry the NASA C3X vane geometry of Hylton et al. (1983). The true chord of the scaled vane was \( C=531 \) mm, the cascade had a pitch between vanes of 457 mm and the setting angle of 34°. The film cooling holes locate at a distance downstream of the stagnation line of 0.67 \( C_x \) on the suction surface and 0.35 \( C_x \) on the pressure surface. The hole has a diameter of 6.35mm, a pitch to diameter ratio of 4 and an inclined angle of 25°. The origin of the axes is the center of the hole exit, \( x \), \( y \) and \( z \) axis are taken to be the streamwise, vertical and spanwise direction to the blade surface, and \( s \) denotes the streamwise distance downstream of the hole center. The periodic boundary is specified on the spanwise planes, and the computation domain is from \( z/D=-2 \) to 2.

![Fig. 1 The vane model in the present study](image_url)
set identically to the mainstream inlet. Incompressible ideal gas whose density is only determined by temperature is used, and the density ratio is 1.1 in the present study. The simulation is performed at two nominal blowing ratios, $M = \rho_c u_c / \rho_\infty u_\infty$ of 0.5 and 1.5. The surface of the turbine blade is set as an adiabatic wall, and the heat flux on the blade surface is 0 W/m$^2$. The adiabatic film cooling effectiveness is defined as:

$$\eta_{loc}(x, z) = \frac{T_\infty - T_{ad,c}}{T_\infty - T_c}$$  \hspace{1cm} (8)$$

$$\eta_{lat}(x) = \frac{1}{\Delta z} \int_{z_1}^{z_2} \eta_{loc}(x, z) \, dz$$  \hspace{1cm} (9)$$

where, $T_\infty$ is the mainstream temperature, $T_c$ is the coolant temperature, and $T_{ad,c}$ denotes the temperature of the adiabatic wall. $\eta_{loc}$ and $\eta_{lat}$ denote local and lateral-averaged adiabatic film cooling effectiveness respectively.

ANSYS Fluent 14.0 is used to perform LES studies. The least squares cell-based scheme is used for the gradient interpolation. The second-order scheme is used for the pressure interpolation. For the momentum equations, the bounded central differencing scheme is used. To prevent unphysical oscillations in the solution field, the bounded central differencing scheme uses a pure central differencing scheme blended with the first and second-order upwind schemes. For the energy equation, the second-order upwind scheme is used. The pressure-based implicit splitting of operators (PISO) algorithm is used for pressure-velocity coupling. The transient formulation is done by the second-order implicit scheme. The time step is determined by $\Delta t = 0.02 D/u_\infty$.

Take for example the blade with a discharged hole on the suction surface. The boundary condition and structural meshes are shown in Fig. 2. The grid points are cluster near the walls of the vane surface and the pipe. The maximum wall-normal distance of the control volume is 0.001D, which corresponds to $y^+ \approx 1.0$ (wall normal direction). In the upstream and downstream of the hole, the maximum value of $x^+$ is about 50 (streamwise direction), and the maximum value of $z^+$ (spanwise direction) is about 35. The stretching factor in the wall-normal direction is about 1.05. Grid-independent tests were performed to determine the optimal grid number. Fig. 3 shows the variation of time-averaged $\eta_{lat}$ with the grid number. Overall, as the grid number exceeds 13,280,232, the values of indices do not change obviously with the grid number. Accordingly, the optimal grid number for the case is 13,280,232 (the grid number in the hole is 119,234). Similarly, for the case with a hole on the pressure surface, the grid number in the hole is 105,655, and the total number of grids is 112,982,513.

![Fig. 2 Boundary conditions and computational meshes](image-url)
2.3 Model validation

Validation by experimental data from Dees et al. (2012) The airfoil geometry is a 3.9 times scale geometry the NASA C3X vane geometry of Hylton et al. (1983). The total grid number in the computational domain is 8,393,252. The maximum $y^+$ (wall normal direction) is about 1.0, the maximum value of $x^+$ (streamwise direction) is about 60, and the maximum value of $z^+$ (spanwise direction) is about 40. The stretching factor in the wall-normal direction is about 1.1. The inlet velocity is 5.8 m/s, and the turbulent intensity of $Tu=0.5\%$ with the turbulent length scale $L_s=30$ mm is specified for the inlet condition. Fig. 4 shows the distribution of the pressure coefficient along the vane surface. On the pressure surface, the pressure coefficient decreases along the streamwise direction. On the suction surface, along the streamwise direction, the pressure coefficient decreases firstly and then starts to increases after the cascade throat. Overall, LES predicts the pressure coefficient well.

Fig. 4 LES results vs. Experimental results from Dees et al. (2012)

Validation by experimental data from Williams et al. (2014) The airfoil geometry is a 3.88 times scale geometry the NASA C3X vane geometry of Hylton et al. (1983). The grid distribution is the same as the primary calculation. The
experimental results from Drost et al. (1998) show that, due to the reformation of the boundary layer after leading edge, the influence of turbulent intensity on film cooling performance on the suction surface is weak. However, to be consistent with the experimental condition from Williams et al., the turbulent intensity of $\text{Turbulent Intensity} = 20\%$ with the turbulent length scale $L_s=30\text{mm}$ is specified on the mainstream in the present case. The film cooling hole is located at a distance downstream of the stagnation line of 0.512 $C$. The hole has a diameter of 6.35mm, a pitch to diameter ratio of 4, and a surface angle of 25°. Typical values for $T_\infty$ and $T_c$ are 305K and 254K. Fig. 5 shows the distribution of film cooling effectiveness in streamwise and spanwise directions. Overall, the simulation results agree well with the experimental results.

![Graphs showing film cooling effectiveness](image)

(a) $\eta_{lat}$ on streamwise direction  (b) $\eta_{loc}$ on spanwise direction ($s/D=5.0$)

Fig.5 LES results vs. Experimental results from Williams et al. (2014).

3. Analysis of LES results

3.1 Low blowing ratio

The Q-criterion method is used for identifying coherent structures. Fig.6 shows coherent structures for film cooling on the turbine blade at low blowing ratio. On the suction surface, the mainstream transits from laminar to turbulent flow immediately downstream of the cascade throat, and plenty of hairpin vortexes are generated. Before reaching the film cooling hole, these hairpin vortexes are broken into smaller-scale vortexes, and a fully-developed turbulent boundary layer is formed. The jet vortex ring interacts with the fully-developed boundary layer, and plenty of broken vortexes dominate film-cooling flow fields. On the pressure surface, the mainstream keeps the laminar flow and starts to transit from laminar to turbulent flow after interacting with coolant jet. From Fig.6 (b), it can be seen that a line of several hairpin vortexes distributes regularly downstream of the hole on the pressure surface. The morphological details of the hairpin coherent structure and its association with the 2-dimensional vortex structures were shown in Fig. 7. Tyagi et al. (2003) indicated that the head, vertical legs and horizontal legs of hairpin vortexes correspond to roller vortexes, wake vortexes, and CRVP respectively. The entrainment effect of hairpin vortexes promotes the mainstream-coolant mixture and deteriorates cooling performance. The temperature on the outer surfaces of hairpin vortexes is much higher than on inner surfaces. During the evolution process of hairpin vortexes, the hairpins grow in size and break into smaller scale vortexes in the far-field region. Wang et al. (2018a, 2018b) reported that the formation of hairpin vortexes is mainly attributed to the evolutions of roller vortexes (or called shear layer vortexes). Because of the shear effect in the boundary layer, the roller vortexes gradually grow up horizontal legs and eventually evolve into hairpin vortexes. Another important coherent structure in the near-field region of film cooling is a horseshoe vortex pair. In Fig. 6(b), a horseshoe vortex pair is clearly generated immediately upstream of the jet hole and wrapped around the hole. The formation of the horseshoe vortex pair is due to the pressure gradients immediately upstream of the hole exit. The rotation direction of the horseshoe vortex pair is contrary to that of CRVP, and the anti-entrainment effect of the horseshoe vortex pair improves film cooling performance in the near-field region (Zhong et al., 2016).

Figure 8 shows the distribution of film cooling effectiveness on the turbine blade at low blowing ratio. On the suction surface, the mainstream velocity is higher than the inlet velocity, and the actual blowing ratio is much lower.
than the nominal value. So, the coolant is diluted by the mainstream quickly after issued from the hole, and the region of $s/D>8.0$ on the suction surface cannot be protected effectively by coolant. On the pressure surface, the mainstream velocity is lower than the inlet velocity, and the actual blowing ratio is higher than the nominal value. So, the cooling performance on the pressure surface is much better than that on the suction surface. Furthermore, in the near-field region, the lateral-covering capability of coolant jet from the pressure surface is better than that from the suction surface. This can be contributed to the anti-entrainment effect of horseshoe vortexes. The distribution profile of instantaneous cooling effectiveness on the pressure surface is much smoother than that on the suction surface. It is because that the boundary layer downstream of the cooling hole on the suction surface is fully-developed, but the boundary layer on the pressure surface just starts to transit after interacting with the coolant jet.

![Fig. 6 Coherent structures for film cooling on turbine vane at low blowing ratio](image)

![Fig. 7 Details of flow fields in the vicinity of a hairpin vortex (Tyagi et al., 2003)](image)

![Fig. 8 Distribution of film cooling effectiveness on the turbine blade at low blowing ratio](image)

Figure 9 shows the distribution of time-averaged and instantaneous flow fields on the cross-sections downstream of the hole, and background color corresponds to non-dimensional temperature. At low blowing ratio, the coolant jet
exhibits good attachment performance both on the suction and pressure surfaces. However, compared with the suction surface, the cross-section of coolant jet from the pressure surface is of a lower temperature. In previous RANS simulation results, CRVP is the dominant vortex structure downstream of the hole, and the entrainment of CRVP deteriorates cooling performance (Mahesh, 2013). In the present LES results, besides of CRVP, the cross-sections of horseshoe vortexes also exist on the lateral sides of coolant jet from the pressure surface. The streamline direction of horseshoe vortexes is contrary to that of CRVP. CRVP promotes the detachment of coolant jet from the wall, while horseshoe vortexes make coolant jet attach to the wall. For film cooling on the suction surface, because of high turbulent intensity, CRVP is buried by plenty of small-scale vortexes in instantaneous results. Overall, the streamlines on the cross-sections downstream of the hole on the pressure surface are smoother than that on the suction surface.

![Distribution of flow fields on the cross-sections downstream of the hole at low blowing ratio](image)

Figure 9 Distribution of flow fields on the cross-sections downstream of the hole at low blowing ratio

Figure 10 (a) shows the distribution of TKE (turbulent kinetic energy) on the cross-sections downstream of the hole. Three major zones of high turbulence levels can be observed on the suction surface. The first zone occurs in the region downstream of the cascade throat. The adverse pressure gradient after cascade throat results in the flow transition, accordingly TKE shows a significant increase. The second zone occurs in the near-field region of film cooling. It is because that the interaction between the mainstream boundary layer and the jet vortex ring leads to high shear stress and the production of the highest turbulence level. In the wake region of the blade, the fluid along the suction and pressure surfaces mix rapidly, and strong unsteady characteristics result in high turbulent intensity. For jet issued from the suction surface, TKE on the jet center is the highest and decreases with the increase of $s/D$. As $s/D$ exceeds 10, TKE distributes uniformly in the boundary layer. For jet issued from the pressure surface, TKE in the jet core is much higher than that on the lateral sides; however, as $s/D$ increases, the difference of TKE becomes small. Fig. 10(b) shows distributions of RMS of wall-normal fluctuating velocity ($u'_{y,rms}$) along the centerline. As wall-normal distance increases, $u'_{y,rms}$ increases firstly and then decreases. Moreover, as $s/D$ increases, the profile of $u'_{y,rms}$ becomes smooth, and the effect of coolant jet on the boundary layer becomes weak.
(a) TKE distribution on the cross-sections  
(b) Distribution of $u'_{y,\text{rms}}$ along the centerline  

Fig. 10 Distributions of TKE and $u'_{y,\text{rms}}$ at low blowing ratio

(a) $s/D=2.0$, hole on the suction surface  
(b) $s/D=5.0$, hole on the suction surface  

(c) $s/D=2.0$, hole on the pressure surface  
(d) $s/D=5.0$, hole on the pressure surface

Fig. 11 Time-frequency analysis of pressure signals at low blowing ratio
3.2 High blowing ratio

Figure 13 shows coherent structures for film cooling on the turbine blade at high blowing ratio. On the suction surface, the mainstream flow transits from laminar to turbulent after reaching cascade throat, and plenty of small-scale broken vortexes dominate the film-cooling flow fields. There are some differences for coherent structures on the pressure surface at low and high blowing ratios. As mentioned above, at low blowing ratio, because of the shear effect in the boundary layer, the roller vortexes gradually grow horizontal legs and eventually evolve into hairpin vortexes. However, at high blowing ratio, the coolant jet detaches from the suction surface, and roller vortexes cannot grow horizontal legs. So, roller vortexes become the dominant coherent structures at high blowing ratio. As shown in Fig.13 (b), a line of several roller vortexes distributes regularly downstream of the hole. During the evolution process of roller vortexes, the vortexes grow in size and break up in the far-field regions. Besides roller vortexes, a pair of horseshoe vortexes wraps around the hole and extends downstream. The scale of horseshoe vortexes does not change obviously with the blowing ratio.

Figure 14 shows the distributions of film cooling effectiveness on the turbine blade at high blowing ratio. Previous researches show that coolant jet will detach from the flat wall at $M=1.5$, which results in poor cooling performance (Bunker, 2005). On the suction surface, the mainstream velocity is much larger than the nominal inlet velocity, and the actual blowing ratio is lower than 1.5. So, cooling effectiveness on the suction surface is high even at high nominal blowing ratio. However, for film cooling on the pressure surface at high blowing ratio, the coolant jet detaches from the wall immediately downstream of the hole and slightly reattaches on the wall after $s/D>4$. One interesting phenomenon for instantaneous film cooling effectiveness on the pressure surface is that a pair of low-temperature strips are wrapped around the hole. The existence of the low-temperature pair is due to the anti-entrainment effect of horseshoe vortexes. The distribution profile of instantaneous cooling effectiveness on the pressure surface is much smoother than that on the suction surface; however, film cooling performance on the pressure surface is poorer than that on the suction surface.

Figure 15 shows distributions of flow fields on the cross-sections downstream of the hole, and background color corresponds to non-dimensional gas temperature. Because the actual blowing ratio is low, the coolant jet from the suction surface attach closely to the wall. However, the jet from the pressure surface exhibits strong detachment effect from the wall, and the CRVP scale is much larger than that at low blowing ratio. For film cooling on the suction surface, the turbulent boundary layer is fully-developed, and plenty of small-scale vortexes are observed downstream of the hole. Accordingly, instantaneous CRVP is buried by these small-scale vortexes and cannot be observed clearly. For film cooling on the pressure surface, the differences between instantaneous CRVP and time-averaged CRVP are not distinct, but the instantaneous CRVP is somewhat asymmetric. Overall, the streamlines on the cross-sections downstream of the hole on the pressure surface are much smoother than that on the suction surface.
Fig. 13 Coherent structures for film cooling on the turbine blade at high blowing ratio

(a) Hole on the suction surface
(b) Hole on the pressure surface

Fig. 14 Distribution of film cooling effectiveness on the turbine blade at high blowing ratio

(a) Instantaneous value, hole on the suction surface
(b) Time-averaged value, hole on the suction surface
(c) Instantaneous value, hole on the pressure surface
(d) Time-averaged value, hole on the pressure surface

Fig. 15 Distribution of flow fields on the cross-sections downstream of the hole at high blowing ratio

(a) s/D=3.0, hole on the suction surface
(b) s/D=3.0, hole on the pressure surface
Figure 16(a) shows the distribution of TKE on the cross-sections downstream of the hole. The rise of the blowing ratio promotes the interaction between coolant jet and mainstream. So, adding the blowing ratio increases TKE downstream the hole both on the suction surface and pressure surface. The highest TKE locates on the jet center, and the value of TKE decreases with the increase of spanwise distance. For film cooling on the suction surface, as s/D increases, TKE increases firstly and then decreases. However, for film cooling on the pressure surface, as s/D increases, TKE shows a continuous decrease. It is contributed to the serious detachment of the jet from the pressure surface at high blowing ratio. Overall, TKE on the cross-section downstream of the hole on suction surface distributes more uniformly than that on the pressure surface. Fig. 16(b) shows distributions of RMS of wall-normal fluctuating velocity ($u'_{y,rms}$) along the centerline. As wall-normal distance increases, $u'_{y,rms}$ increases firstly and then decreases. Furthermore, the profile of $u'_{y,rms}$ becomes smooth as s/D increases. Overall, the distribution of $u'_{y,rms}$ at high blowing ratio is similar to that at low blowing ratio.

Figure 17 shows the frequency spectrum of the pressure fluctuation signals. For film cooling on the pressure surface, the dominant frequency locates at $St\approx 2.1$. At high blowing ratio, roller vortexes replace hairpin vortexes as the dominant coherent structures, and this dominant frequency is related to the evolution frequency of roller vortexes. For film cooling on the suction surface, the frequency profile exhibits strong oscillation and the dominant frequency is not obvious. Overall, the frequency spectrum does not change obviously with the blowing ratio. The dominant frequency is determined by the combined effect of mainstream and coolant jet. However, in the present study, compared with the mainstream, the coolant jet has a weak effect on the frequency spectrum. It illustrates that the mainstream plays a dominant role in the vortex evolution in the present model.
4. Conclusions

Large eddy simulations were performed for film cooling on scaled-up C3X turbine vane at the nominal blowing ratio of $M=0.5$–1.5, and the Reynolds number, $Re=3000$, based on the mainstream inlet velocity and hole diameter. Some useful conclusions are as follows:

1) Mainstream boundary layer on the suction surface transits from laminar to turbulent flow in the downstream of cascade throat but in the upstream of the discharged hole, and plenty of broken vortexes dominate the film-cooling fields. On the pressure surface, large-scale coherent structures including hairpin vortexes and horseshoe vortexes are generated in film-cooling flow fields. However, at high blowing ratio, hairpin vortexes disappear, and roller vortexes become the dominant coherent structure.

2) At low blowing ratio, film cooling performance on the pressure surface is better than that on the suction surface. As the blowing ratio increases, the film cooling effectiveness on the pressure surface drops sharply due to flow separation, and film cooling performance on the suction surface is better at $M=1.5$. The entrainment of hairpin vortexes degrades film cooling performance, while the effect of horseshoe vortex is contrary. In the instantaneous flow fields on the cross-sections downstream of the hole on the pressure surface, CRVP and horseshoe vortex pair take the dominant role. However, for film cooling on the suction surface, because of high turbulent intensity, CRVP is buried by plenty of small-scale vortexes.

3) For film cooling on the pressure surface, TKE in the jet core region is much higher than that on the lateral sides. For film cooling on the suction surface, because the boundary layer is fully-developed, the distribution of TKE on cross-sections downstream of the hole is relatively uniform except for the jet core region. Overall, coolant jet from the suction surface generates higher turbulent intensity than that from the pressure surface.

4) For pressure signals for film cooling on the suction surface, small-scale and random fluctuation takes the dominant role, and the frequency spectrum exhibits strong oscillation. For film-cooling signals on the pressure surface, dominant frequency exists at $St=2.1$ both at low and high blowing ratios, which corresponds to the evolution of
large-scale coherent structures. The film-cooling system on the suction surface exhibits a higher random degree than that on the pressure surface.

Due to the lack of unsteady experimental data for the turbine blade, a quantitative comparison of the simulation results to the experiment is only possible for the time-averaged data. To provide detailed data for model validation, more unsteady measurements on film cooling on the turbine blade should be performed in the future.

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Nomenclature

\[ C \] true chord length blade
\[ C_p \] pressure coefficient
\[ C_s \] smagorinsky constant
\[ C_x \] axial chord length of the blade
\[ D \] diameter of the film cooling hole
\[ f \] frequency
\[ h \] gas enthalpy
\[ I \] momentum flux ratio, \( I = \frac{\rho_c U_c^2}{\rho_\infty U_\infty^2} \)
\[ L_s \] the turbulent length scale
\[ l_{char} \] characteristic length
\[ M \] blowing ratio
\[ p \] blade pitch
\[ P \] pressure
\[ p_{bc} \] total pressure of the coolant
\[ p_{in} \] static pressure of the mainstream
\[ Pr_t \] turbulent Prandtl number
\[ s \] arc length along the blade surface
\[ S \] rate-of-strain tensor
\[ St \] stouhal number, \( St = f D / u_\infty \)
\[ t \] time
\[ T_i \] turbulent intensity
\[ u \] velocity

Greek symbols

\[ \rho \] gas density
\[ \tau \] shear stress tensor
\[ \tau_h \] subgrid enthalpy flux
\[ \mu_s \] subgrid-scale viscosity
\[ \Delta x \] characteristic length
\[ \delta_{99} \] thickness of the boundary layer
\[ \eta \] film cooling effectiveness, \( \eta = (T_\infty - T_{ad})/(T_{bc} - T_c) \)
\[ \theta \] non-dimensional temperature, \( \theta = (T - T_c)/(T_{bc} - T_c) \)
\[ \omega \] vorticity

subscript

\( \infty \) mainstream
\( c \) coolant
\( x, y, z \) streamwise, wall normal and spanwise direction
\( \text{loc} \) local value
lat  laterally averaged value
w  wall
ad  at an adiabatic condition
rms  root mean square

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