Research Article

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Effect of forward expansion angle on film cooling characteristics of shaped holes

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Abstract: According to the design requirements of high-temperature combustion chamber, an advanced shaped hole structure was designed for film cooling. Numerical method was applied in this study to investigate the flow and heat transfer characteristics of shaped holes and compared with those of cylindrical holes. The influence of the forward expansion angle of shaped holes on the flow and heat transfer was studied. The results show that compared to cylindrical holes, the diffused structure of shaped holes decreases the momentum of jet flow, improves the adhesion characteristics of the cooling air film, increases the diffusion of the coolant air outflow and improves the cooling efficiency between adjacent columns of holes in the lateral direction. When the forward expansion angle increases, the expansion section induced the flow vortex, which reduces the radial velocity of coolant flow and enhances the diffusion of cooling air film both in streamwise and spanwise directions. However, as the forward expansion angle increases further, the scale of vortex inside the shaped hole grows. Too large vortex inside the shaped hole increases the coolant eject angle, which weakens the film covering effect. Additionally, the shaped hole results in an increase in lateral spreading and enhances the cooling effect between adjacent columns of the film hole. The enhancement of the film cooling characteristics is due to the change in the shape of the film hole, resulting in the enhancement of the flow vortex, which induces complicated secondary flow.

Keywords: film cooling, shaped hole, expansion angle, lateral diffusion

1 Introduction

With an increase in performance of modern aeroengine, the inlet temperature of modern gas turbine engines has reached 2,000°C [1]. In order to maintain the operational reliability of components such as turbine blades and combustion chambers in high-temperature environments, effective cooling measures must be taken, and film cooling has been regarded as one of the main cooling technologies used in combustion chambers. The performance of film cooling is affected by many factors, such as aerothermal conditions, hole size, number of cooling holes, hole arrangement, injection angle and hole shape [2–6]. The shape of the film hole and the angle of coolant injection are currently being researched by many scholars.

Two shapes of holes are used for film cooling: cylindrical holes and shaped holes [7]. In the study of shaped holes, Goldstein [8] found that compared with the cylindrical hole, the cooling effect of the shaped hole in the downstream is significantly improved, especially on the downstream surface of the film hole on both sides. Crawford et al. [9] conducted a series of experiments for full coverage film cooled surfaces with three geometries: normal, 30° and 30° × 45° compound angle and demonstrated that the structure with a spanwise expansion angle significantly improved cooling effectiveness. Yuen et al. [10,11] carried out experimental study on film cooling with cylindrical, compound angle and fan-shaped holes, and the results indicate that compound angle holes gave the higher values of effectiveness when compared with the non-compound case at blowing ratio (BR) > 1, but at BR < 0.67, the conclusion was on the contrary. The fan-shaped holes gave the highest cooling effectiveness at given blow ratios and flow rates. Gritsch and Thole et al. [12–16] compared the cooling efficiency and the heat transfer performance of two fan-shaped holes. The results show that the structure with both lateral and forward expansions has better heat transfer performance than the structure with only lateral expansion. Under high blow ratio, this advantage is even more obvious. Dittmar et al. [17] compared the cooling effect of double-row slots, double-row cylindrical holes, single-row fan-shaped holes and fan-shaped holes with

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that of compound angles by experimental methods. The results show that the film cooling effectiveness distribution of all models at low BRs is similar, and when the blowing is relatively high, the jet from cylindrical hole detaches from the penetrate wall so that the cooling performance is poorer, and the fan-shaped hole maintains great cooling performance. Colban and Thole [18] compared the adiabatic and cooling performance of cylindrical and fan shaped holes, and the results showed that the performance of the fan-shaped holes was better. Bai et al. [19] investigated the cooling performance of a row of cylindrical and double-fan-shaped holes by experimental and numerical methods and found that the two-fan-shaped hole demonstrated better film cooling performance and superior film expansion in the lateral direction than that of the cylindrical hole. Bohn and Moritz [20] conducted a series of numerical simulations for a plate with seven rows of cylindrical and shaped holes and reported that a lower inclined angle gave higher cooling effectiveness and forward and lateral expansion angles reduced the exit radial velocity of cooling gas, which led to an improvement in film coverage. Kanani et al. [21] studied film cooling effectiveness on a flat plate with cylindrical and laterally diffused holes with a streamwise angle of 30° and a spanwise angle of 0° . They found that a better lateral coverage and a higher cooling effectiveness value were obtained by laterally diffused hole since the laterally diffused structure decreased the momentum of coolant jet flow at the exit of the hole. This is consistent with the experimental results of Yuen and Martinez-Botas [22].

All in all, many studies have been conducted to investigate the effects of hole shape and hole inclination

angle on the film cooling performance [23–25]. However, few efforts were devoted to reveal the influence of the forward expansion angle of the shaped hole on the flow and cooling characteristics. In this study, numerical simulation method was used to compare the flow and heat transfer characteristics of cylindrical and shaped holes. Particular attention was paid to explore the effect of forward expansion angle on flow field and film cooling performance of shaped hole in which the lateral expansion angle was fixed.

2 Computation scheme

2.1 Physical model

Figure 1(a and b) is a schematic diagram of the structure of inclined cylindrical hole and shaped hole. The cylindrical hole inclination angle $\alpha = 45^\circ$, the film hole diameter $d = 0.7$ mm and the parameters of the shaped hole are shown in Figure 1(b). In the figure, L_T is the length of the unexpanded section, L is the total length of the hole, α is the inclination angle of the hole, β is the lateral expansion angle and δ is the forward expansion angle. The thickness of the perforated plate $t = 2$ mm; the diameter of the film hole $d = 0.7$ mm; the spacing of the hole rows $S = 4.24$ mm; the adjacent hole-to-hole pitch $P = 4.9$ mm; the inclination angle of the holes $\alpha = 60^\circ$; the expansion ratio $(L - L_T)/L = 1/2$ was the same; three models of forward expansion angle δ of 8° , 15° and 20° were established; and the geometric parameters of each model are shown in Table 1.

Figure 1: Schematic of cylindrical (a) and shaped holes (b).

Table 1: Geometrical parameters

Shape of hole	Model	α	B	δ	$(L - L_1)/L$
Cylindrical	Model 0	45°	0°	0°	1
Shaped hole	Model 1	60°	8°	8°	1/2
	Model 2			15°	
	Model 3			20°	

2.2 Computation domain and boundary conditions

Figure 2 shows the structure of computational domain. The length of the calculation domain in the flow direction (in the x direction) is 200 mm, the height of the mainstream channel (in the z direction) is 25 mm, the height of the secondary channel is 15 mm, the distance between the first row hole and the last row hole is set to be 160 mm. Considering the full development of the flow field, the upstream and downstream of the perforated plate contain a 20 mm non-porous area, and the width of the domain (in the y direction) is chosen as one spanwise hole-to-hole pitch, which is equal to 4.9 mm.

Periodic boundary conditions are applied on both sides, and the perforated wall surface is set as a fluid–solid coupling surface. The solid domain material used was steel. Since the pressure of the primary and secondary flows does not change much, the mainstream and secondary flows are set as ideal gases. The blow ratio is defined as

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

where ρ_c and u_c represent the mean density and the velocity of coolant flow, respectively, and ρ_{∞} and u_{∞} represent the mean density and the velocity of the mainstream, respectively. In this study, $\rho_c = 11.48 \text{ kg/m}^3$,

$u_c = 100.08 \text{ m/s}$, $\rho_{\infty} = 5.66 \text{ kg/m}^3$ and $u_{\infty} = 97.8 \text{ m/s}$, so $BR \approx 2$.

The primary and secondary flow inlets are all set as pressure inlets, and the outlets are all set as pressure outlets. The mainstream (hot flow) inlet total pressure is 3.1 MPa, the total temperature is 1,960 K, the total outlet pressure is 3.05 MPa, the total temperature is 1,700 K, the secondary flow (cold flow) inlet total pressure is 3.3 MPa, the total temperature is 860 K, the total outlet pressure is 3.28 MPa and the total temperature is 860 K.

The film cooling effectiveness η is defined as:

$$\eta = \frac{T_g - T_w}{T_g - T_c}, \quad (2)$$

where T_g is the mainstream temperature, T_w is the perforated wall temperature and T_c is the coolant flow inlet temperature.

2.3 Computation approach

The software ANSYS ICEM-CFD was applied to generate the computational grids. Figure 3 shows the grids at the central plane of the hole [26–28]. In order to improve the calculation accuracy, structured hexahedral meshes were generated for the whole computational domain and fine grids were generated in the vicinity of the wall surface suffering hot flow where the complicated flows occur. Figure 4 shows the grids of the perforated wall surface, and Figure 5 shows the grids of the entrance, exit and wall of the shaped hole.

Figure 6 shows the effect of grid number on the average temperature of the perforated wall surface. The grid independence tests showed that when the number of model grids is greater than 2 million, the temperature

Figure 2: Computational domain.

Figure 3: Grids at the central plane of the hole.

fluctuation is less than 1%, and it can be considered that the grid independence requirement is met. Therefore, in the subsequent calculations, different geometry models adopt a similar grid division method, and the number of whole computational domain grids is about 2 million, where the height of the first layer of the boundary layer is 0.01 mm and the grid spacing ratio is 1:1.1.

It has been proved that the standard $k-\varepsilon$ model was good in predicting the trend of film cooling effectiveness and has been successfully used by some other authors [29–31]. Therefore, the standard $k-\varepsilon$ turbulence model with the standard wall function was finally selected for the calculation. Second-order upwind is selected for discreteness of the flow equation, the turbulent kinetic energy k and the dissipation rate ε . The sub-relaxation is employed to control the computation speed. Convergence is regarded to be achieved when the residual is less than 1×10^6 .

Figure 4: Grids of the protected wall.

Figure 5: Grids of the shaped hole. (a) Grids of hole entrance, (b) grids of hole side, (c) grids of hole exit.

Figure 6: Effect of grid number on the average temperature of the perforated wall surface.

3 Results and analysis

3.1 Comparison between cylindrical and shaped holes

3.1.1 Flow field and temperature field downstream of film hole

Figure 7 shows the contours of wall temperature of Model 0 and Model 1, respectively. The temperature contour of the shaped hole (Model 1) is similar to the

Figure 7: Contours of wall temperature.

Figure 8: Two positions of cooling effectiveness selection.

Figure 9: The cooling effectiveness of Model 0 and Model 1 on line I and line II.

cylindrical hole (Model 0) in the spanwise direction, but the wall temperature is lowered at the same axial position, that is, the cooling efficiency is increased at the same axial position. At the axial position of $x = 80$ mm, the wall temperature of Model 1 has dropped below 1,000 K, while the wall temperature of Model 0 is above 1,000 K.

Figure 8 shows two positions on the perforated wall surface, marked as line I and line II. The cooling efficiency analysis is performed on the position of these two lines.

Figure 9 shows the cooling effectiveness distribution of Model 0 and Model 1 in line I and line II. The cooling effectiveness trend of Model 0 and Model 1 is similar. Line I is the symmetry line of film holes, so the cooling effectiveness increased with fluctuation along line I. The cooling efficiency of Model 1 is always higher than Model 0, at $x/L = 80$. The cooling efficiency of Model 1 reaches 0.9 and continues to increase, while the cooling effectiveness of Model 0 is lower than 0.9 at any position; the cooling efficiency of Model 1 along line II is still higher than that of Model 0, which means the shaped holes provide a better thermal protection to the wall between two columns of holes.

The cross sections in the three different streamwise positions downstream of the third row and the eighth row and the fifteenth row of film hole are selected as shown in Figure 10 and are, respectively, marked as I, II and III cross sections.

Figure 11 shows the temperature distribution and streamline diagram of Model 0 and Model 1 in cross sections I, II and III. The kidney vortices of Model 1 present wider distance in spanwise than that of Model 0

Figure 10: Different cross sections of downstream.

Figure 11: Temperature contours and streamlines in cross sections I, II and III.

in section I, which offers strong support for expanding the lateral coverage of coolant film. In cross section II, the film of Model 1 has merged together between two adjacent holes in the lateral direction, while that of Model 0 does not. The cooling air film of Model 1 has fully developed in cross section III, forming a lower temperature and more uniform film covering on the wall surface which is benefit for wall cooling. While for Model 0, the kidney vortices are further lifted off the wall surface, causing strong mixing of hot flow and coolant flow. Therefore, the film cooling effect of Model 0 is worse than that of Model 1.

expansion section of Model 1 causing flow separation, so that the momentum of coolant outflow is reduced, the second flow is closer to the wall surface, and the adhesion property and the ductility of the gas film are improved. The cold flow keeps a high velocity from the inlet to the outlet, so that the momentum of film outflow is high, and the penetration ability to the mainstream is strong, which is not conducive to the formation of a stable film coverage.

3.2 Influence of forward expansion angle

3.1.2 Flow inside holes

Figure 12 shows the velocity field inside holes of Model 0 and Model 1. The coolant flow is decelerated in the

3.2.1 Analysis of cooling effectiveness

Figures 13 and 14 present the cooling effectiveness in line I and line II of film hole, respectively. According to

Figure 12: Velocity fields inside holes of Model 0 (a) and Model 1 (b).

Figure 13: The cooling effectiveness on line I.

Figure 14: The cooling effectiveness on line II.

Figures 13 and 14, the trend of cooling effectiveness of the three models is consistent. In addition, the cooling effectiveness of Model 2 is always the highest both in line I and line II after $x/L = 60$, while Model 3 is basically the same as Model 1.

3.2.2 Flow field and temperature field downstream of film hole

Figures 15–17 show the temperature field and velocity vector in sections I, II and III, respectively. As shown in Figure 15, the kidney vortex of Model 2 is closer to the

wall surface than that of Model 1. This is because the forward expansion angle is increased, which reduces the momentum of the cooling gas outflow, that is, the penetration capacity of coolant flow is reduced. As the expansion angle is further increased, the kidney vortex of Model 3 is as close to the wall surface as Model 2, but the lateral width and the core low-temperature area of the kidney vortex are smaller than Model 2.

As the cooling air film develops downstream, the cooling air film expanded in both the lateral direction and the height direction, and the kidney vortices merge

Figure 15: Temperature contours and velocity vectors on cross section I (the third row). (a) Model 1, (b) Model 2, (c) Model 3.

Figure 16: Temperature contours and velocity vector on cross section II (the eighth row). (a) Model 1, (b) Model 2, (c) Model 3.

together in the lateral direction in section II (as shown in Figure 16). In section III, the cooling film of Model 2 is the flattest in spanwise, followed by Model 3 and Model 1. Flat cooling air film demonstrates that the intensity of kidney vortices is rather weak, the vortices are limited beneath the film layer and that is helpful to cool the wall surface.

Figure 18 reports temperature distribution of protected wall in I, II and III sections. The temperature distribution of Model 1, Model 2 and Model 3 in section I

is the same, lower near the exit of the hole and increased on both sides of holes. Besides, the temperature of Model 2 is the lowest, while the temperature of Model 1 is the highest. In section II, the lateral temperature distribution of Model 1 is the same as in section I, but the wall temperature between holes of Model 2 and Model 3 significantly decreased, which is almost the same as the temperature downstream of the exit of the hole. This is because two kidney vortices merged together forming a cooling film between adjacent holes in the lateral

Figure 17: Temperature contours and velocity vector on cross section III (the fifteenth row). (a) Model 1, (b) Model 2, (c) Model 3.

Figure 18: Temperature along the spanwise direction in different cross sections.

direction. In section III, the cooling air film is fully developed, and the wall temperature between adjacent holes is further reduced in Model 2 and Model 3, which is about 150 K lower than that in section II.

3.2.3 Flow inside holes

Figure 19 shows the velocity field through the center plane of the hole. The coolant flow is accelerated in the inlet

section of the hole due to the deflection and convergence of the flow. Then, flow separation takes place in the diffused section of the hole. The low speed zone in Model 1 is smaller, the radial velocity of coolant flow is higher and the velocity in the flow direction is lower, which results in strong penetration into the mainstream. With an increase in forward expansion angle, the area of the low-speed zone in Models 2 and 3 increase in turn and even reverse flow occurs, generating vortices, which greatly reduce the momentum of the coolant jet, making the cooling film attach to the wall better.

Figure 20 shows the temperature contours and streamlines at the diffused section of holes. The streamlines show that the vortex scale increases with the forward expansion angle. In Model 3, a large-scale vortex is formed in the expansion section, which causes the coolant flow entrain the main flow into the hole, so the coolant flow and the main flow are premixed inside holes, resulting in an increase in the temperature of the coolant outflow. A similar result was found in the experiments conducted by Kohil and Bogard [32] for a shaped hole with an expansion angle of 15° and an injection angle of 55° . Since this vortex is away from the wall surface of hole, the surrounding coolant flow is lifted by the vortex, which increases the injection angle of coolant jet and weakens the effect of improving the adhesion of the cooling film that results from the increase in the expansion angle.

Figure 21 shows four cross sections inside the hole, marked as I', II', III' and IV' and presents maps of streamlines and turbulence kidney energy.

Figure 19: Velocity fields through the central plane of the hole. (a) Model 1, (b) Model 2, (c) Model 3.

Figure 20: Temperature contours and streamlines at the diffused section of holes. (a) Model 1, (b) Model 2, (c) Model 3.

Figure 21: Cross section inside the hole.

Figure 22 presents X vorticity contours and streamlines at different cross sections. The vorticity and velocity distributions of the three models in the undiffused section (section I') in the hole are basically the same, the left vortex in the hole rotates counterclockwise and the right vortex rotates clockwise. It is worth noting that the vortices of Model 1 interact and substantially

completely attenuated in section III', while the vortices of Model 2 show a trend of asymmetric development and a small vortex generated near the wall (as shown in section III'), which reduces the momentum of the coolant outflow. The vortices scale and intensity of Model 3 in section III' are larger than that of Model 2, which causes the coolant flow to be lifted off the wall and mixing with the hot flow inside the film hole.

Figure 23 presents the turbulent kinetic energy contours at different sections. The turbulent energy distribution of the three models is similar in the undiffused section. The turbulent energy of Model 1 decays rapidly along sections I' to IV', while the turbulent energy of Models 2 and 3 increases in section III', which is caused by the vortex generated in section III'. However, the vortex scale and strength of Model 3 are so large that the coolant flow entrains the mainstream in the boundary layer.

Figure 24 presents the temperature contours of the outlet plane of the hole. The outflow temperature

Figure 22: Distribution of eddy and streamlines.

distribution of Model 1 is relatively uniform and that is similar to Model 2, but in the case of Model 3, the temperature of coolant outflow is obviously increased. So the vortex induced in the diffused section can reduce the momentum of coolant outflow on one hand, allow the cooling film to adhere to the wall more easily, on the other hand, the excessive vortex scale causes the coolant flow to entrain the high temperature mainstream and reduce the quality of the cooling gas.

4 Conclusions

In this study, the flow and the cooling performance of cylindrical and shaped holes were compared by the numerical method, and the effect of the forward expansion angle on the flow field, temperature field and cooling effectiveness downstream of film hole is studied. The main conclusions are as follows:

- (1) Compared to cylindrical holes, the diffused structure of shaped holes reduces the radial velocity of coolant jet, increasing the velocity in flow and lateral directions, which enhances the adhesion of the cooling film and strengthens the spreading ability in the lateral direction and results in a better cooling effect.
- (2) Within a certain range, as the forward expansion angle increases, the momentum of cooling jets decreases and the forward and lateral coverage of cooling film improved further.
- (3) Too large forward expansion angle results in stronger vortex generation in the diffused section away from the hole wall, which increases the injection angle of coolant flow and causes poor film coverage. In addition, the vortex in the mainstream allows the cold air to be mixed with the mainstream, which enhances the convective heat transfer in the hole, increases the cooling outflow temperature and reduces the downstream cooling efficiency of the gas film.

Figure 23: Turbulence kidney energy contours.

Figure 24: Temperature contours of the outlet plane of the hole. (a) Model 1, (b) Model 2, (c) Model 3.

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