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Effect of Multi-hole Configuration on Film Cooling Effectiveness

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ABSTRACT: A numerical study is performed to investigate the effects of shaped multi-hole on film cooling effectiveness over a flat plate. Hence a single cylindrical film cooling hole with 11.1 mm diameter is replaced with the shaped multi-hole (14 holes with 2.97 mm diameter) while maintaining constant blowing ratio. Numerical simulations are performed at a fixed density ratio of 1.6, length-to-diameter of 4 and an inclined angle of 35°. Two configurations of hook and fan shapes are considered for multi-hole. The control-volume method with a semi-implicit method for pressure linked equations-consistent algorithm has been used to solve the steady-state Reynolds-averaged Navier–Stokes equations. The k-ε model is applied for modeling the turbulent flow and heat transfer field. It is found that replacing a single hole with the shaped multi-hole leads to a considerable increase in the film cooling effectiveness in both axial and lateral directions. Results of the present study show that for blowing ratio of 0.6, the hook shape and fan shape configurations of multi-hole, provide a higher area-averaged film cooling effectiveness by 48% and 58.2% more than the single hole respectively.

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1- Introduction

In advanced gas turbines, high inlet turbine temperature is considered for increasing the output power. The melting point of the materials in the combustor section is often exceeded and the components may fail due to thermal stresses. Thermal Barrier Coatings (TBCs), are advanced materials systems usually applied to metallic surfaces, such as on gas turbine parts, for operating at elevated temperature. In addition, relatively cooler air from the compressor section upstream can be extracted and used to cool the components in the turbine section [1].

Film cooling is an external cooling technique in which the cooling air is ejected through discrete holes in the desired surface creating a layer of insulation over the surface. The cool layer, protect the surface from the hot gas in the region directly downstream of the holes.

Several studies indicated that using cylindrical holes in film cooling had disadvantages in gas turbine applications due to the jet lift off from the surface, particularly at higher momentum flux ratios (~1 and above) leading to deterioration the film cooling performance. In film-cooling, the coolant separates from the wall just after the injection hole and create counter rotating vortices, which known as kidney vortices [2]. Design and Operating parameters affect the generation and growth of kidney vortices [3,4].

The mixing of secondary flow with the hot mainstream increases with the presence of kidney vortices. Hence, kidney vortices should be reduced to keep the surface with coolant film and consequently better film cooling. Miao and Wu [5], Leedom and Acharya [6], Baheri et al. [7] considered the use of a trench including cylindrical and forward-diffused holes to evaluate the applicability of this concept. They found that trenching diffused hole reducing the Counter-Rotating Vortex Pair (CRVP) at the hole exit and lead to a significantly improving lateral spreading and showed the best overall performance. Lu [8] studied the effect of hole configurations on film cooling from cylindrical inclined holes and show that the film cooling jet exiting the trenched hole is more two-dimensional than the typical cylindrical holes and crater holes. Shaped holes have proven to provide the highest adiabatic effectiveness among film cooling configurations as investigated by Laveau and Abhari [9] and Gao and Han [10] but the shaped holes are expensive and difficult to manufacture.

The research for new developments to optimize film cooling performance has been intensified in recent years. Instead of using holes with shaped exits, Zhang et al. [11] investigated the effects of placing upstream steps with unevenly spanwise distributed height on the film cooling effectiveness. An influence of novel upstream steps on film cooling performance has been considered by Abdala et al. [12]. Sister holes another technology investigated by Ely and Jubran [13] to increase cooling effectiveness by reducing pockets of reversed flow. Heidmann [14] considered an antivortex hole shape for mitigating the adverse effects of jet vorticity by adding two small branched holes. These holes create inverse vorticity against the kidney vortices which results in better cooling effectiveness. Repko et al. [15] considered the effect of the free stream turbulence on the effectiveness of the multi (anti-vortex) hole. The results show that as the free stream turbulence intensity is increased, the cooling flow will stay more attached to the wall, providing improved

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coolant coverage and higher cooling effectiveness. Singh et al. [16] concluded that film cooling effectiveness of a reverse injection hole is higher than the forward holes. Also they showed that kidney vortices are not presented in the case of reverse holes and hence better film cooling can be obtained.
Yuzhen et al. [17] presented the effectiveness of film cooling of three various multi-hole designs. They tested the effects of spacing between cooling row, span-wise hole pitch and the hole inclination angle. The results showed that the row spacing ratio affects the film cooling performance. Also, better adiabatic cooling performance is achieved using a smaller pitch, especially for the multi-hole patterns. Ai and Fletcher [18] studied the similar case and concluded that the effectiveness at places close to the exit of jets for wide hole spacing is slightly higher than for small hole spacing. By the way, the small hole spacing executed better than wide hole spacing at downstream locations due to the interaction of neighboring jets [19]. The influence of multi-hole arrangement on effusion film cooling is analyzed by Chengfenga and Jingzhou [20]. They concluded that the coolant jets from front rows of multi-holes merge rapidly and the strength of the kidney vortices due to mainstream coolant jet interaction in the downstream region are mitigated under super-long-diamond arrangement where the streamwise hole-to-hole pitch is bigger than spanwise hole-to-hole pitch.
The previous study concluded that the effectiveness of a single film hole is often less than that an array of holes. Roy [21] investigated the flow field emanating from an array of jets entraining on a hot cross-flow. Roy discovered that the CRVPs emanating from adjacent jets counteracts one another. This effectively keeps the cooling flow on the surface of the test plate and improves the film cooling performance.
As investigated in the previous researches the shaped film cooling holes, can produce significantly better cooling performance than the single cylindrical holes under wide range of blowing ratios, the manufacturing of the shaped holes is much expensive than a cylindrical hole. Therefore, it is profitable finding the novel film cooling designs based on cylindrical holes. One way to decrease the momentum of the coolant jet is applying a larger sectional area at the outlet of the hole. Therefore, the multi-hole configuration with multiple exits can be used for the film cooling by cylindrical holes to reduce the momentum of coolant jet [22].
In the present study, a single hole has been replaced with the multi-hole film cooling while maintaining the area of the injection holes constant. It is desired that the multi cylindrical holes achieve a favorable performance for cooling hole without any additional expenditure on manufacture. The other geometrical parameters for single hole and multi-hole, like the hole length to diameter ratio and inclined angle, were the same. The aim of the present numerical work is to investigate the effects of the multi cylindrical holes on the flow structure and adiabatic cooling effectiveness compared to that of one classical cylindrical hole.

2- Computational Domain

The computational domain of the present study is based on the experimental results of Schmidt et al. [23] and the results are used as a benchmark work for further analysis in multi cylindrical holes. The hole diameter of 11.1 mm used by Schmidt et al. [23] is used as the reference diameter for one cylindrical hole computational domain. For the computational domain of the multi-hole, the 11.1 mm hole has been divided to 14 holes with 2.97 mm diameter.
The computational domain of single cylindrical hole has been shown in Fig. 1. Hole geometry has an injection angle $\beta = 35^o$, inlet diameter D = 11.1 mm, and the discharge pipe length to diameter ratio of 4. For accurate modeling of the cross-flow interaction, it is essential to model the cross-flow, film hole regions, and plenum flow simultaneously [24]. Therefor plenum is also used to study the effect of the coolant velocity profile.

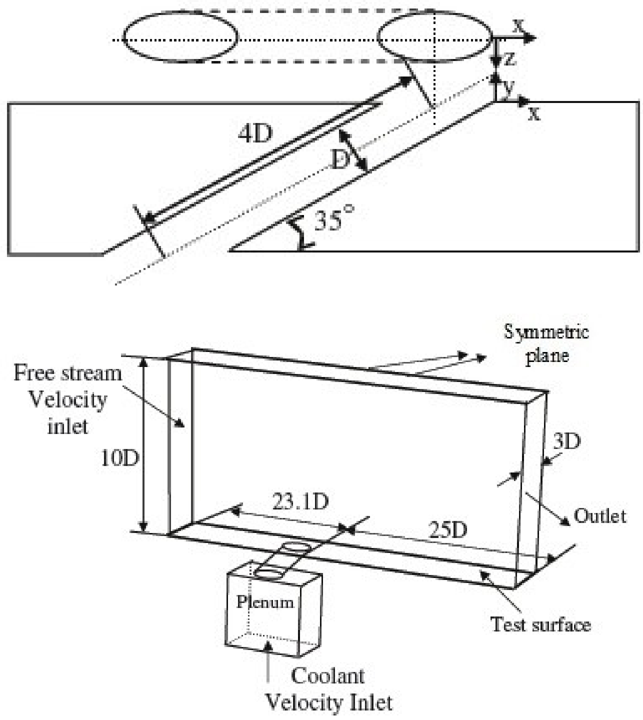


Fig. 1. Computational domain and boundary conditions for a single hole [7].

The plenum geometry is identical to geometry of Schmidt et al. [23]. This plenum is 50.8 mm high and 101.6 mm wide. The uniform inlet velocity of the plenum is set to achieve Re=18700 for every case and the main stream velocity is varied to achieve the desired blowing ratio conditions, in conform to the Schmidt et al. [23] cryogenic setup. The operating parameters used to calculate the performance of the film coolant was based on the experimental study of Schmidt et al. [23] and given in Table 1.

Table 1. operating parameters

Property	value
Freestream temperature	300 K
Blowing ratio	0.6 and 1.25
Density ratio	1.6
Coolant temperature	187.5 K

Turbulence intensity of freestream and coolant is assumed 0.2 and 0.1%, respectively. Symmetry boundary condition was used for the lateral planes in the first block. The top surface is also considered as the symmetry plane. The freestream and coolant inlet are defined as the velocity inlet, and the outflow

boundary condition are applied for the outlet condition, whereas the plate, coolant-pipe and plenum have been modeled as an adiabatic wall with a no-slip condition.
14 cylindrical holes with 2.97 mm diameter have been used for the multi-hole configuration. Multi-holes were arranged in hook and fan shapes configurations. These configurations have been shown in Fig. 2. The center-to-center spacing of adjacent holes in these configurations was set to 1.5 *D*.

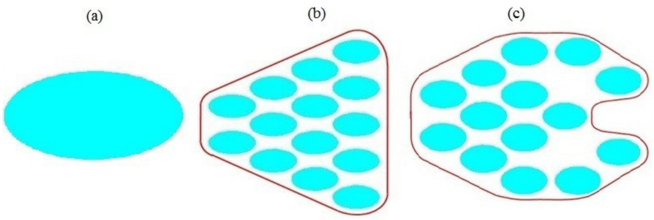


Fig. 2. Film cooling configurations, a) single hole, b) multi-hole with fan shape, c) multi-hole with hook shape.

3- Numerical method

The simulations were performed by using ANSYS FLUENT 16.0.0 [25]. The mathematical film cooling model consists of the steady state Reynolds-Averaged Navier-Stokes (RANS) equations, the energy equation, and the k–ε turbulence model with standard wall function. These equations are given as:

$$\frac{\partial U_i}{\partial x_i}=0 \tag{1}$$

$$\rho U_i\frac{\partial U_i}{\partial x_i}=-\frac{\partial p}{\partial x_j}+\mu\frac{\partial^2\Theta}{\partial x_i\partial x_i}-\frac{\partial}{\partial x_i}\left(\rho\overline{u_iu_j}\right) \tag{2}$$

$$\rho\overline{u_iu_j}=\mu_t\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)-\frac{2}{3}\left(\rho k+\mu_t\frac{\partial u_k}{\partial x_k}\right)\delta_{ij} \tag{3}$$

$$U_i\frac{\partial\Theta}{\partial x_i}=\alpha\frac{\partial^2\Theta}{\partial x_i\partial x_i}-\frac{\partial}{\partial x_i}\left(\overline{u_i\theta}\right) \tag{4}$$

where $\overline{u_iu_j}$ and $\overline{u_i\theta}$ are known as the Reynolds stress tensor and the turbulent heat flux vector respectively. In the case of the k–ε model, two additional transport equations (for the turbulence kinetic energy, k, and either the turbulence dissipation rate, ε) are solved:

$$\frac{\partial}{\partial x_i}\left(\rho k u_i\right)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] \tag{5}$$
$$+G_k+G_b-\rho\in-Y_M+S_k$$

$$\frac{\partial}{\partial x_i}\left(\rho\varepsilon u_i\right)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x_j}\right] \tag{6}$$
$$+C_{1\varepsilon}\frac{\varepsilon}{k}\left(G_k+C_{3\varepsilon}G_b\right)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}+S_\varepsilon$$

Details of the transport equations for the turbulence kinetic energy (k), and its dissipation rate (ε), and the model constants can be found in [24,25]. μ_t is computed as a function of k and ε as follows:

$$\mu_t=\rho C_\mu\frac{k^2}{\varepsilon} \tag{7}$$

where C_μ is a constant. The governing equations are solved by using a three-dimensional finite-volume method that allows the use of arbitrary nonorthogonal multi-block grids. The pressure-velocity coupling algorithm is achieved by using the Semi-Implicit Method for Pressure Linked Equations-Consistent (SIMPLEC) algorithm. The residual error for convergence is set to 10^{-4} for continuity, and 10^{-6} for momentum and energy equations.

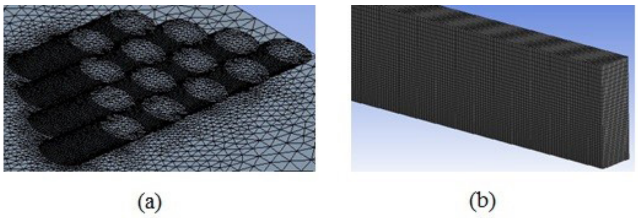


Fig. 3. Computational grid for fan shape multi-hole, a) second block, b) first block.

The quality of a computational solution is strongly linked to the quality of the grid generation. So a nonuniform, orthogonalized, multi-block fine grid was generated with grid nodes considerably refined in the near-wall region. Fig. 3 shows the grid of these two blocks used for fan shape holes. The grid sensitivity test for a cylindrical hole is shown in Fig. 4.

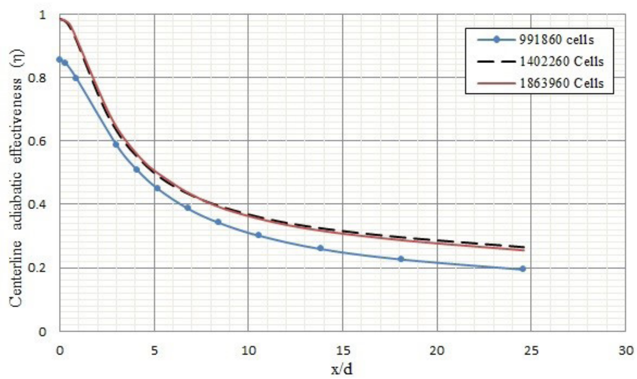


Fig. 4. Grid sensitivity for a single cylindrical hole, *M* = 0.6.

Comparisons between the predicted results of the film cooling effectiveness at the centerline of the test plate using the standard, Re-Normalization Group (RNG), realizable k-ε, and the experimental results of Schmidt et al. [23] at M=0.6 are shown in Fig. 5. As shown in this figure, there is no significant difference between these turbulence models in prediction the centerline adiabatic effectiveness. Hence the standard k-ε model has been applied for all cases in the present numerical study.
The predicted centerline effectiveness results are obtained directly using Eq. (8). A numerical integration is used to find the lateral averaged effectiveness values, as shown in Eq. (9):

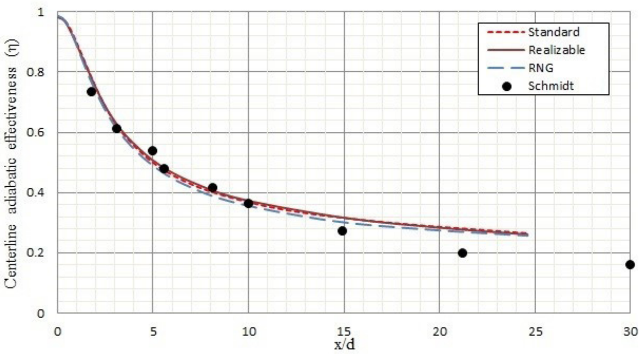


Fig. 5. Predicting of centerline effectiveness, M = 0.6.

$$\eta = \frac{T_{aw} - T_{\infty}}{T_C - T_{\infty}} \tag{8}$$

$$\overline{\eta} = \frac{1}{L}\int_0^L \eta dz \tag{9}$$

where L in Eq. (9) is the length of the lateral distance between the symmetric boundaries.

4- Results and discussion

Fig. 6 compares the centerline adiabatic effectiveness of a single cylindrical hole with the experimental data of Schmidt et al. [23]. The results of low blowing ratio (M = 0.6) show a good agreement with the experimental data. At high blowing ratio (M = 1.25), the lift-off of the jet from the surface creates secondary vortices, and consequently the standard k–ε model overpredicts the film cooling effectiveness near the hole region. Acharya [26] showed that the k–ε model overpredicts the penetration of cooling air near the hole region.

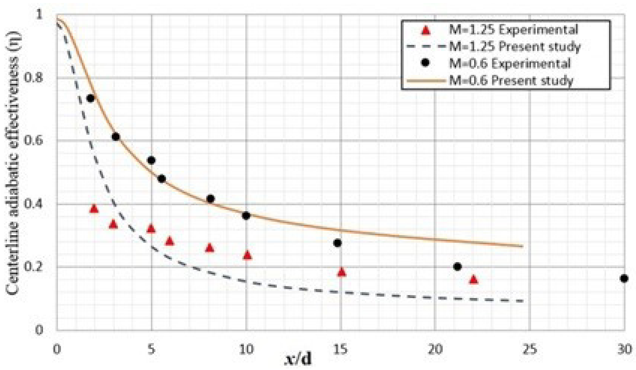


Fig. 6. Comparison of the centerline effectiveness with the experimental data [23].

Figs. 7 and 8 show the temperature distributions on the centerline plane under blowing ratios of M=0.6 and M=1.25, respectively. At a low blowing ratio (M=0.6), the penetration of cooling air to the mainstream flow is weaker than high blowing ratio (M=1.25), which consequences a higher film cooling effectiveness. It is obvious from the Fig. 8 (a) that at M=1.25, the hot stream has been pulled under the cooling air and reduced the effectiveness of the coolant and the jet tends to lift-off for single cylindrical hole case. As shown in Figs. 8 (b) and 8 (c) for multi-hole configurations the jet lift-off has been reduced, so the coolant remains near the

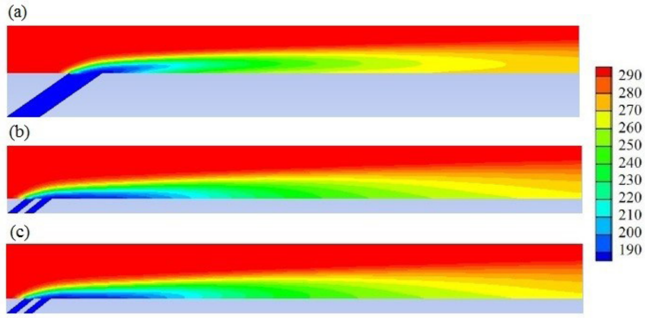


Fig. 7. Centerline plane temperature contours for M=0.6, a) single hole, b) fan shape holes, c) hook shape holes.

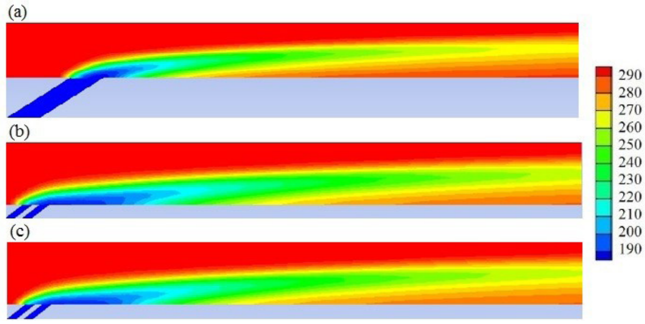


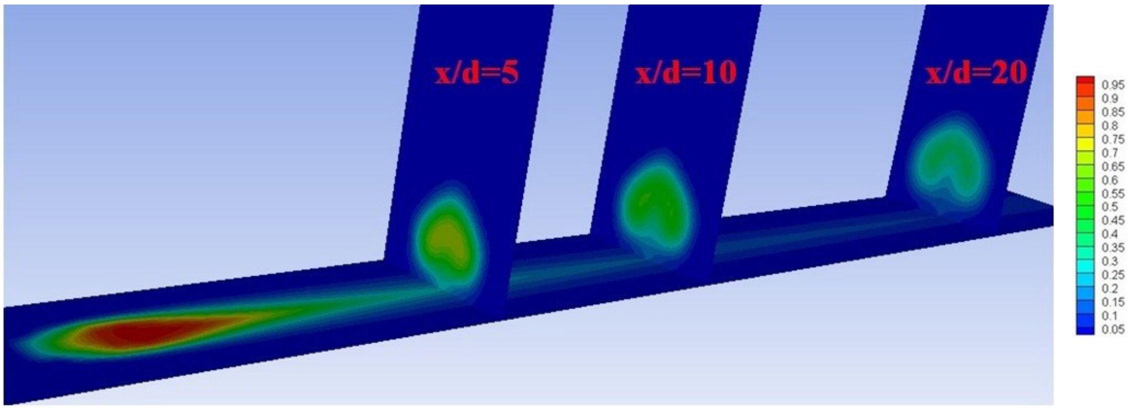
Fig. 8. Centerline plane temperature contours for M=1.25, a) single hole, b) fan shape holes, c) hook shape holes.

hot surface rather than an undesirable mixing with the hot mainstream gas. The cooling air and main stream interaction causes the formation of a counter-rotating vortices pair which consequences the coolant air to lift off from the surface.

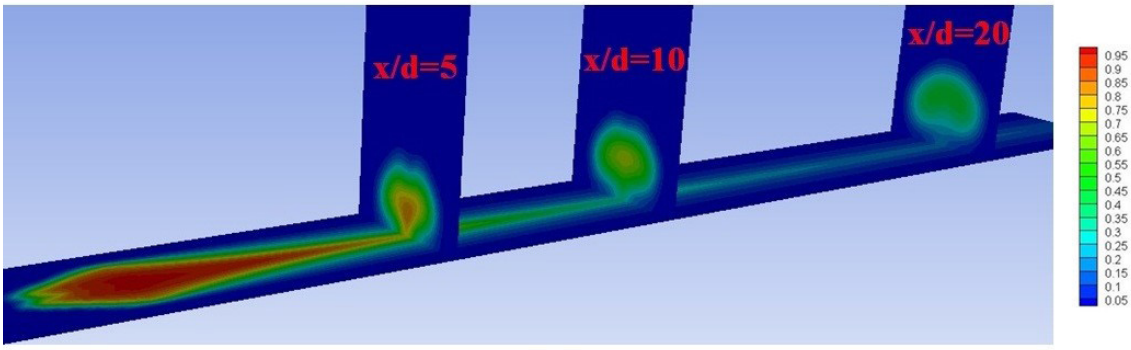
One of the primary focuses of the film cooling technique is to reduce the primary vortex pair. The counter-rotating of the vortices strongly reduces the effectiveness of the film cooling. The method of weakening the vortex pair has been explained by Walters and Leylek [27] who showed that the two important mechanisms that influence on the counter rotating vortex pair are the interaction between the jet and the free stream.

Fig. 9 shows contours of the adiabatic effectiveness change with non-dimensional temperature at different streamwise location at M=1.25 for the single hole and the fan shaped configuration. Walters and Leylek [27] concluded that as the coolant moves to downstream, the cooling flow is moved away from the surface by the well-known counter rotating vortex structure, as shown in Fig. 9. As shown in the figure the strength of the counter rotating motion increases with increasing blowing ratio.

Fig. 10 shows the cross plane streamline along with temperature contours at X/D = 3 under a blowing ratio of M=1.25. The counter rotating vortex pairs at the exit of the jet are apparent and strong for the single cylindrical hole, like those were reported by Leylek and Zerkle [28] for a cylindrical hole. As the single hole is replaced with multi-hole, these vortexes have been weakened and hence the cooling air becomes closer to the surface, (see Figs. 10 (b) and 10 (c)). It is clear from Fig. 10 that the center of counter rotating vortex pairs for the single cylindrical hole is upper than that for multi-hole. The weakest velocity vectors in the counter rotating vortex pairs are found for the fan shape, which is the main reason for higher film cooling effectiveness for this



(a)



(b)

Fig. 9. Contours of adiabatic effectiveness change with non-dimensional temperature at different streamwise locations at *M*=1.25, a) single hole, b) fan shape holes.

case. As proved by Haven and Kurosaka [29], by increasing the span-wise distance, the separating of kidney vortices reduces the induction caused by the vortex pair, which results in the jet tends to remain near the surface.

Fig. 11 show the detailed adiabatic wall film cooling effectiveness distributions for the single cylindrical hole, fan and hook multi-hole for M =1.25. According to Fig. 11 (a) the distribution of cooling air for the single cylindrical hole is very low in both streamwise and spanwise directions. This can be due to the jet lift-off further downstream of the film hole, which leads to a low cooling effectiveness. When the multi-hole film cooling injections are applied, the lateral and streamwise effectiveness increases.

Fig. 11 shows that the penetration of the hook shape holes is weaker than that for the fan shape holes. It is notable that, the configuration of multi-hole arrangement, can significantly affects the distribution of the adiabatic film cooling effectiveness. As shown in Fig. 11, the distribution of coolant for fan shape holes is better than hook shape holes in streamwise and spanwise directions. As mentioned by Roy [21] the cooling effectiveness of a single hole is often less than that an array of holes. As shown in Fig. 11 (b), this effectively forces are higher in fan shape holes which have led to an increase in the lateral spreading of the coolant.

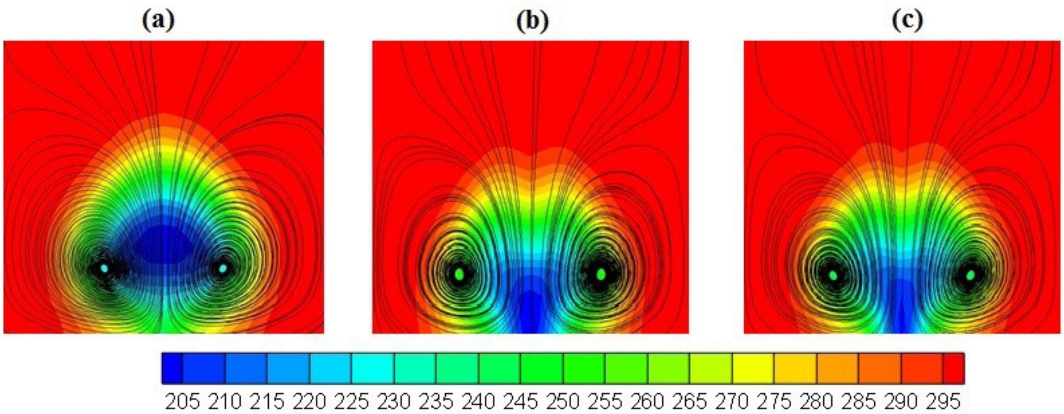


Fig. 10. Velocity contours along with streamlines change with Velocity contours along with temperature, *M* =1.25, *x/d* = 3, a) single hole, b) fan shape holes, c) hook shape holes.

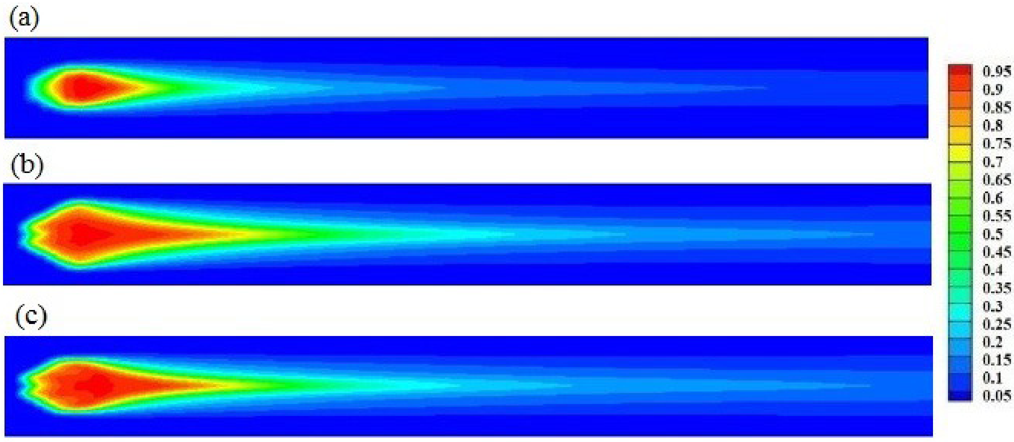


Fig. 11. Contours of film cooling effectiveness for *M* =1.25, a) single hole, b) fan shape holes, c) hook shape holes.

Fig. 12 shows the centerline effectiveness for the single hole, fan shape holes and hook shape holes for $M = 0.6$ and M=1.25. As mentioned by Walters and Leylek [27], as blowing ratio is increased the influence of the film hole flow becomes more significant. While, at low blowing ratios, the effect of the flow in the film hole is less important. Under a lower blowing ratio ($M = 0.6$) the single cylindrical hole shows a lower film cooling effectiveness in comparison with fan and hook shape holes. When the multi-hole injection is used, the decrease in strength of kidney vortices leads to a significant increase of the film cooling effectiveness.

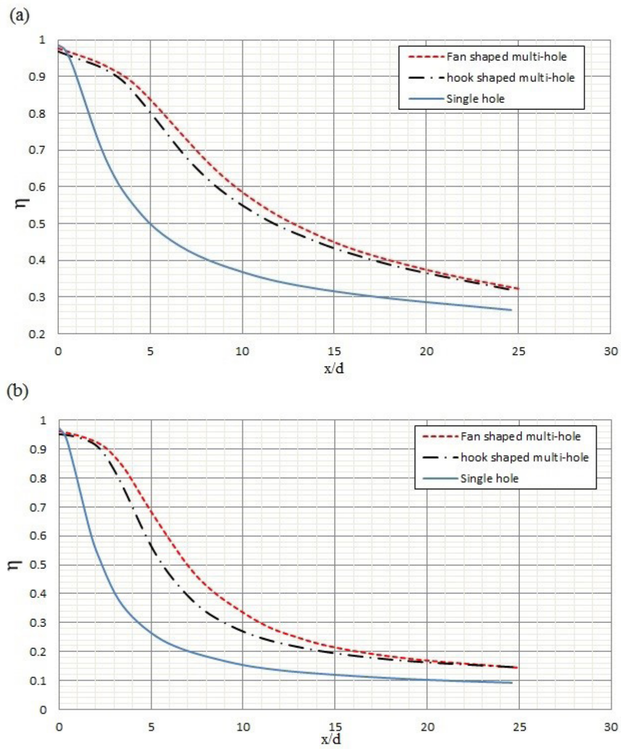


Fig. 12. Centerline film cooling effectiveness, a) *M* = 0.6 and b) *M*=1.25.

Under a higher blowing ratio, the decrease in the coolant jet tangent velocity weakened the streamwise jet momentum, which shortened the film coverage along the streamwise direction. Under a blowing ratio M =1.25, the effect of shaped holes on the improvement of film cooling effectiveness was very considerable, especially in fan shape holes. As shown in Fig. 12 (b), centerline adiabatic effectiveness of fan shape holes have been increased more than twice that obtained for the single hole in the most centerline areas of the test plate and this incremental process of effectiveness has been continued to the downstream of the injection holes. Adiabatic effectiveness of the multi-hole is higher than the single hole in the region near the injection region.

The advantage of multi-hole arrangement can also be observed at spanwise direction. The lateral effectiveness of a single cylindrical hole is compared with multi-hole arrangement in Fig. 13. It is evident from the figure that the fan shape holes tend to provide the highest lateral effectiveness for both blowing ratios considered in the present study. As the blowing ration increases to 1.25 more difference between lateral adiabatic effectiveness of fan shape and single hole can be seen.

Fig. 14 shows a comparison of the laterally averaged film cooling effectiveness between the cylindrical hole and shaped multi-holes for two blowing ratios of $M = 0.6$ and $M = 1.25$. The fan shape holes result in higher film cooling effectiveness than the cylindrical hole and the hook shape holes. By increasing the blowing ratio to 1.25, the fan shape holes produce a more obvious improvement in the film effectiveness. Fig 14 shows that the cooling fluid from the multi-hole penetrates deeper in the lateral direction than the cooling air from the single cylindrical hole. Also it can be inferred from the figures that the coolant from multi-hole spreads widely on the surface in the lateral direction and prevent the surface from the mainstream hot air.

The comparisons of the area-averaged film cooling effectiveness for blowing ratios of 0.6 and 1.25 are reported in Table 2. In the present study the area-averaged film cooling effectiveness can be calculated as:

$$\bar{\eta}(x,z) = \frac{1}{3D \times 25D} \int_{-1.5D}^{1.5D} \int_{0}^{25D} \eta(x,z)\,dx\,dz \tag{10}$$

According to the Table 2, an increase in area-averaged film cooling effectiveness is observed with decreasing the blowing ratio single, fan shape and hook shape holes. The fan shape hole provides a higher area-averaged film cooling effectiveness by 58.2% and 101.5% more than the single hole at blowing ratios of 0.6 and 1.25 respectively.

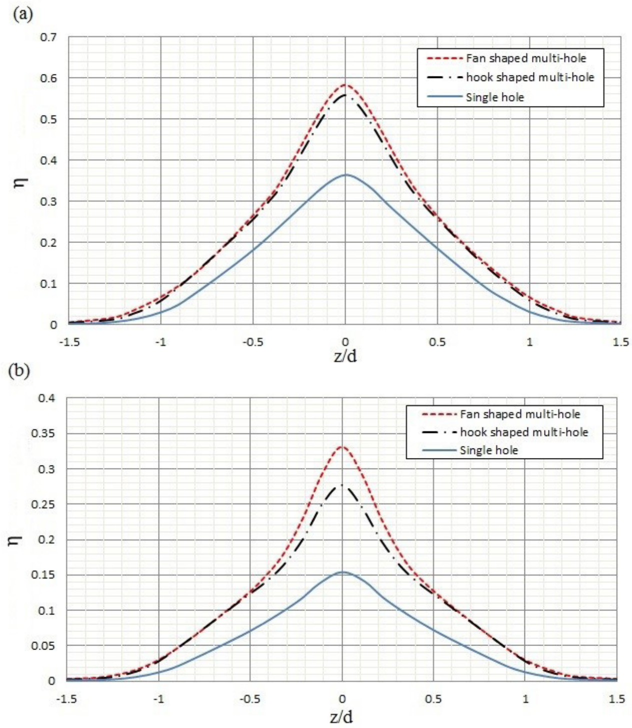


Fig. 13. Spanwise film cooling effectiveness at *x/D* = 10, a) *M* = 0.6, b) *M* = 1.25

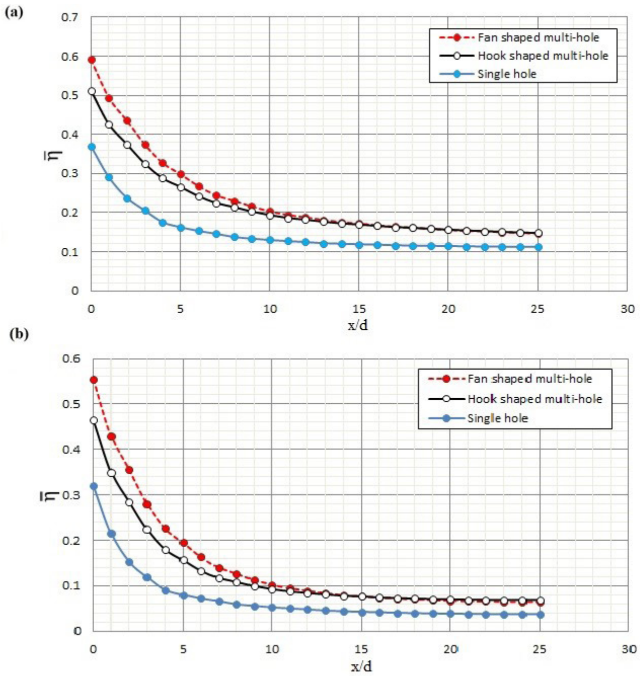


Fig. 14. Laterally averaged film cooling effectiveness, a) *M*=0.6, b) *M*=1.25.

Table 2. Comparisons of the area-averaged film cooling effectiveness

Blowing ratio	Area-averaged film cooling effectiveness ($\bar{\eta}_{ave}$)		
	Single hole	Hook shape	Fan shape
M=0.6	0.146	0.216	0.231
M=1.25	0.069	0.122	0.139

Effect of the number of holes on the centerline adiabatic effectiveness for the fan shaped multi-hole is shown in Fig. 15. The results show that, when 14 holes with 2.97 mm diameter is replaced with 9 holes with 3.7 mm diameter, centerline effectiveness is reduced, especially at near of the holes. When 9 holes are used, the decrease in the coolant jet tangent velocity is less than that obtained for 14 holes, and consequently the streamwise jet momentum is higher. So the CRVP(s) effects are increased and this leads to a significant decrease of the film cooling effectiveness at near of the holes.

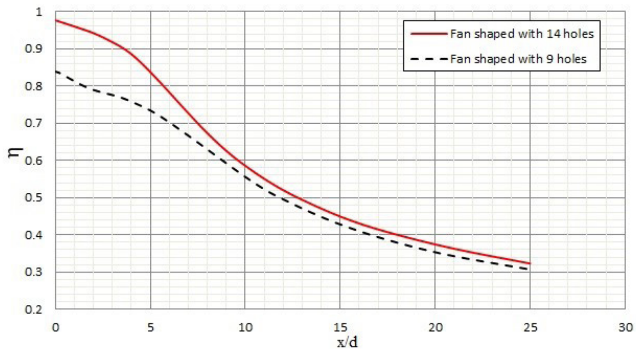


Fig. 15. Comparison of centerline adiabatic effectiveness of fan shaped multi-hole at *M* =0.6 with 9 and 14 holes.

5- Conclusions

Numerical investigation is performed to enhance the cooling effectiveness over a flat plate by applying shaped multi-hole. A single cylindrical film cooling hole with the diameter of 11.1 mm has been replaced with 14 small holes with 2.97 mm diameter while maintaining constant blowing ratio. The multi-hole (14 small holes) has been arranged in two fan shape and hook shape configurations. Numerical simulations are performed at length-to-diameter of 4, inclined angle of 35^o and two blowing ratios of 0.6 and 1.25. The control-volume method with a SIMPLEC algorithm has been used to solve the steady-state RANS equations. The k-ε turbulence model is applied for modeling the turbulent flow and heat transfer. The numerical predictions of film cooling effectiveness agree well with the available experimental data. The higher film effectiveness is achieved at low blowing ratio for single and multi-holes configurations due to jets lift-off. Results of the present study indicate that the arrangement of multi-hole has a significant effect on the film cooling effectiveness in both axial and lateral directions. The multi-hole generates weaker anti-vortices as compared to the single cylindrical hole which consequences a lower mixing between the main and the hot air stream. It has been observed that replacing a single hole with the shaped multi-hole leads to a considerable increase in film cooling effectiveness in both axial and lateral directions. The multi-hole with fan shape arrangement provides a better film protection in comparison with the single and hook multi-hole configuration. It is found that the multi-hole with fan shape arrangement provides a higher area-averaged film cooling effectiveness by 58.2% and 101.5% more than the single hole at blowing ratios of 0.6 and 1.25 respectively.

Nomenclature

DR	Coolant to free-stream density ratio = ρ_c/ρ_∞
D	Diameter of the hole
k	Turbulent kinetic energy
L	Length of the hole
M	Blowing ratio = $(\rho U)_c/(\rho U)_\infty$
Re	Reynolds number
T	Temperature
Tu	Turbulent intensity
x/D	Non-dimensional streamwise distance
Greek symbol	
β	Streamwise injection angle
ε	Dissipation rate of turbulent kinetic energy
η	Adiabatic film cooling effectiveness; $\eta=(T_{aw}-T_\infty)/(T_C-T_\infty)$
ρ	Density of the fluid
τ_w	Wall shear stress
μ	Dynamic viscosity
μ_t	Turbulent dynamic viscosity
Θ	Internal energy
θ	Heat flux
Subscript	
c	Coolant
aw	Adiabatic wall
∞	Free stream

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