Enhancement of Adiabatic Film Cooling Effectiveness by Using Conical Shape Hole

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ABSTRACT.

Film cooling is one of the methods used to protect the surfaces exposed to high-temperature flows, such as those exist in gas turbines. It involves the injection of coolant fluid (at a lower temperature than that of the main flow) to cover the surface to be protected. This injection is through holes that can have various shapes; simple shapes, such as those with straight cylindrical or shaped holes (included many holes geometry, like conical holes). The computational results show that immediately downstream of the hole exit, a horseshoe vortex structure consisting of a pair of counter-rotating vortices is generated. This vortex generation affected the distribution of film coolant over the surface being protected. The fluid dynamics of these vortices are dependent upon the shape of the film cooling hole, and blowing ratio, therefore the film coolant coverage which determines the film cooling effectiveness distribution and also has an effect on the heat transfer coefficient distribution. Differences in horseshoe vortex structures and in resultant effectiveness distributions are shown for cylindrical and conical hole cases for blowing ratios of 0.5 and 1. The computational film cooling effectiveness values obtained are compared with the existing experimental results. The conical hole provides greater centerline film cooling effectiveness immediately at the hole exit, and better lateral film coolant coverage away of the hole exit. The conical jet hole enhanced the average streamwise adiabatic film cooling effectiveness by 11.11% and 123.2% at BR= 0.5 and 1.0, respectively, while in the averaged lateral adiabatic in the spanwise direction, the film cooling effectiveness enhanced by 61.75% and 192.6% at BR= 0.5 and 1.0, respectively .

Keywords: Film cooling, effectiveness, conical holes, Enhancement

1. INTRODUCTION.

The thermal efficiency and specific power of gas turbine can be improved by increasing the turbine inlet temperature, operation a high-temperature caused a thermal failure to prevent that risk film cooling is commonly used. Film cooling involved injection of coolant from film holes to form a thin thermal barrier layer to protect the blade surface from the hot gases flow. The objective of film cooling is to achieve low heat transfer from the surrounding hot mainstream to the turbine blades, and large effectiveness on the blade surface. In the recent years, several studies have focused on developing the holes shape to enhance film cooling effectiveness. Film cooling research on flat surface is common, flat surface models can be used to study the effects of individual parameters with relative ease and are less expensive. Studies have proved that the results obtained on simple flat surface models can be applied to real engine designs with slight corrections [1]. The definition of film cooling effectiveness is given in the dimensionless form, this effectiveness represents the efficiency of a cooling film; the maximum value of unity is achieved when the adiabatic wall temperature is the same as the coolant temperature. Studies on film cooling effectiveness in the hole streamwise direction on a flat are presented in **Fig. (1)**, for several existing researchers, such as, Eriksen and Goldstein [2], Goldstein and Yoshida [3], Goldstein et al. [4], Bergeles [5], Brown and Saluja [6], Sinha et al. [7], and Schmidt et al. [8]. From the literature, one can conclude that the optimum injection angle is found to be as (30º-35º) according to Bunker[9], in which the flow pattern of the 30^o and 35^o jet(s) can take different forms, depending on the blowing ratio, the jet(s) can remain attached, detach and reattach, or lift off completely. From the cited references above, it was suggested for a single jet and a row of jets of a given streamwise inclination, the effectiveness near the holes increased with increasing blowing ratio, until a certain value beyond which the jets started to lift off, and the effectiveness decreased, and this ratio is often referred as the optimum blowing ratio. The optimum blowing ratio for a single 30^o to 35^o jet is usually between a blowing ratio of 0.4 and 0.6, and slightly less for a single vertical jet around 0.5, and relatively less for a row of 60^o jets. It is well known that significant improvement can be achieved in cooling characteristics of the film by using cooling holes with appropriately designed expanded exits. Goldstein et al. [10] were among the first to pioneer the use of shaped film holes for improved film cooling performance. The performance of inclined holes with 10° laterally flared exit was compared with the performance of streamwise inclined cylindrical film holes. Effectiveness data showed that the shaped film hole provides better lateral coverage and better centerline effectiveness. Hyung et al. [11] investigated experimentally and numerically the film cooling performance around a conical-shaped film cooling hole with compound angle orientations. The result shows that the shaped holes reduced the penetration of jet, and more uniform cooling performance is obtained even at relatively high blowing rates, because the conically expanded hole exit reduces the momentum of the lateral spreading. The better cooling performance is obtained with shaped holes expands 4^o in all direction from the hole middle to the exit. Schmidt et al [8] and Sen et al. [12] presented two companion papers in which the effect of adding a 15° forward diffusion exit to a streamwise oriented hole was investigated. They found that the exit diffused film hole demonstrated better spread of adiabatic effectiveness than the cylindrical counterpart. From the heat transfer coefficient standpoint, the forward expanded hole performed poorly, presumably because of the increased interaction between the jet and mainstream. Kohli and Bogard [13] examined the film cooling effectiveness of the shaped holes on a flat plate using 35^o and 55^o injection angles, the result shows shaped holes with large injection angle has better cooling performance than cylindrical holes. They also report on the thermal and velocity fields in the region around the injection holes. Sargison et al. [14] Studied a converging slot-hole geometry (console) in which the hole transitions from circular to slot with convergence in the axial direction and divergence laterally. The attempt was to make the three-dimensional nature of the jet into a two-dimensional slot film. The results were aimed at improving effectiveness. Yuen and Martinez-Botas [15] studied the film cooling effectiveness using a cylindrical hole at an angle of 30^o, 60^o, and 90°. A hole length of L=4D was used, the free-stream Reynolds number based on the free-stream velocity and hole diameter was 8563, and the blowing ratio was varied from 0.33-2. For a single 30° hole, in the region immediately downstream of the hole, the maximum effectiveness occurred for a blowing ratio less than 0.5. Downstream of this immediate region, centerline effectiveness and lateral spread increased up to a blowing ratio of 0.5, then decreased with increasing blowing ratio due to jet penetration into the free stream. Also, the region with effectiveness greater than 0.2 did not extend beyond X/D=13. Baheri et al. [16] presented a comparative-numerical investigation on film cooling from a row of simple and compound angle holes injected at 35º on a flat plate with four film cooling configurations: (a) cylindrical film hole; (b) 15º forward diffused film hole; (c) trenched cylindrical film hole; and (d) trenched 15º forward diffused film hole. All simulations are at fixed blowing ratio of 1.25 and pitch to diameter ratio of 3. The mathematical film cooling model consists of the RANS, the energy equation and the standard (k-ε) model using a finite volume method. They found that the trenched compound angle injection shaped hole produces much higher film cooling protection than the other configurations. Fayyaz and Muhammad [17] compared computationally using CFD Fluent the film cooling effectiveness of cylindrical, square and two types of equilateral triangular holes with an inclination of 30^o with streamwise direction, theoretical results of the cylindrical hole compared with experimental results by Yuen and Martinez-Botas [15] showing well in agreement even for high blowing ratio. They also observed that triangular hole having lateral straight edge on leeward side shows much higher effectiveness values than circular film cooling hole case in the near hole region and almost similar coolant jet height as that in case of circular film cooling. Also, it is observed that triangular hole having lateral straight edge on windward side and converging corner on leeward side shows lesser coolant jet height and higher film cooling effectiveness in the region X/D>10, especially at blowing ratios greater than 1.0.

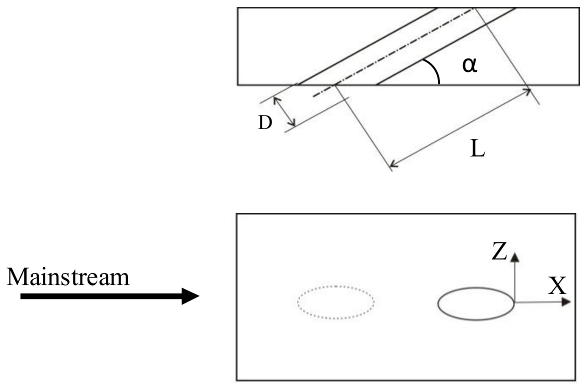

Figure (1): Hole at a streamwise angle, α.

2. PROBLEM DESCRIPTION.

The film cooling performance of conical hole geometry is compared with cylindrical hole geometry numerically by using a commercial CFD package (FLUENT from ANSYS). The computational results of cylindrical holes are compared with the existing experimental results given by Durham [18], Kohli [13], and Sinha [7]. The problem is treated as steady state, three dimensions, incompressible, and turbulent flow. Schematic diagram (with dimensions in millimeters) of the side view of the computational domain and both holes of the film cooling problem is shown in **Fig.(2)**, the cooling jet emerged from a plenum through one row of cylindrical or conical holes, each cylindrical hole has a diameter D of 5 mm, a length of 3.5D and an inclination angle of 35^o relative to the plane tangent to the flat plate. The spacing between the centers of the film-cooling holes in the spanwise direction is 3D.The conical hole has a diameter of 5 mm at plenum edge diverged by 4^o to the flat plate exit and an inclination angle(α) of 35^o between hole centerline and plane tangent to the flat plate. The exact dimensions and parameters have been chosen, so that the results can be compared to those discussed in references.

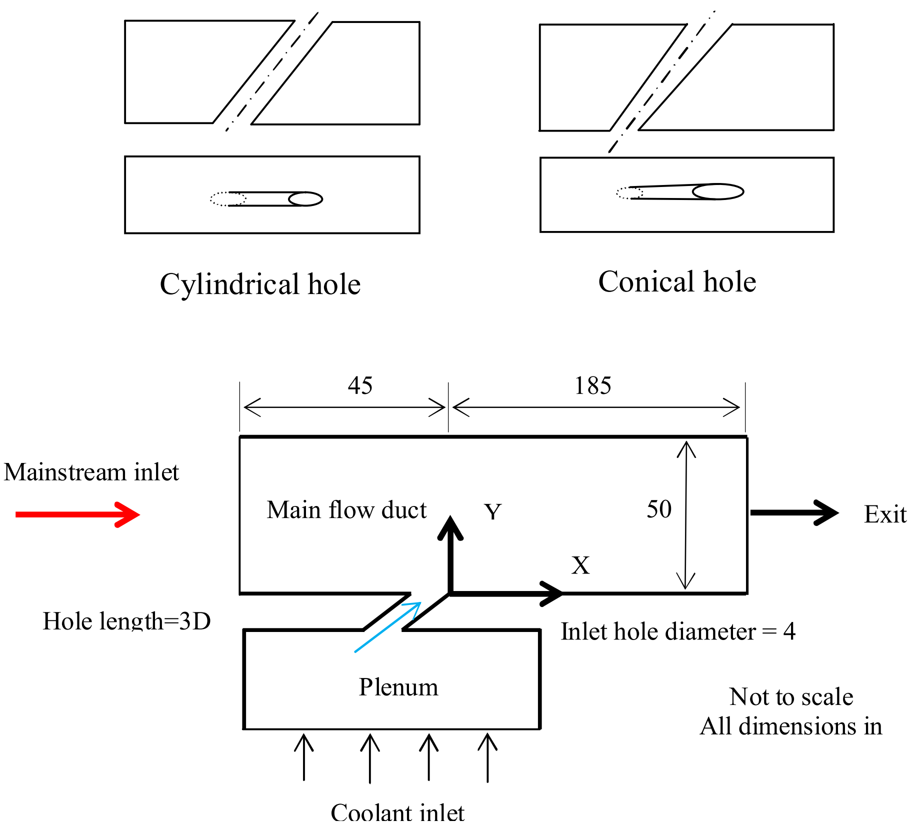

Figure (2): Schematic of the film-cooling configuration studied.

3. GOVERNING EQUATIONS.

The time-averaged, steady state Navier-Stokes equations as well as equations for mass and energy are solved. The governing equations for conservation of mass, momentum, and energy are given as:

$$\frac{\partial}{\partial x}(\rho u_i) = S_m \tag{1}$$

$$\frac{\partial}{\partial x}(\rho u_i u_j) = \rho \overline{g_j} - \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i}\left(\tau_{ij} - \rho\overline{\acute{u}_i \acute{u}_j}\right) + F_j \tag{2}$$

$$\frac{\partial}{\partial x}(\rho u c_p u_i T) = \frac{\partial}{\partial x_i}\left(\lambda \frac{\partial T}{\partial x_i} - \rho c_p \overline{\acute{u}_i \acute{T}}\right) + \mu\phi + S_h \tag{3}$$

Where τ_{ij} is the symmetric stress tensor defined as:

$$\tau_{ij} = \mu\left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j} - \frac{2}{3}\delta_{ij}\frac{\partial u_k}{\partial x_k}\right) \tag{4}$$

$(\mu\phi)$ Is the viscous dissipation, and λ is the thermal conductivity. The terms of $\rho\overline{\acute{u}_i\acute{u}_j}$ and $\rho c_p\overline{\acute{u}_i\acute{T}}$ represent the Reynolds stresses and turbulent heat fluxes, respectively which should be modeled properly for a turbulent flow.

Two additional transport equations for the turbulence kinetic energy (k) and the turbulence dissipation rate (ε) are solved. The standard (k –ε) model [19] is a semi-empirical model based on model transport equations for the turbulent kinetic energy (k) and its dissipation rate (ε). Assumed that the flow is fully turbulent, and the effects of molecular viscosity are negligible, the turbulence kinetic energy (k) and its dissipation rate (ε) are obtained from the following equations [20]:

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

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

Turbulent viscosity (μ_t) is computed as a function of (k) and (ε).

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

The generation of turbulence kinetic energy due to the mean velocity gradients (G_k) is computed by:

$$G_k = -\rho\acute{u}_i\acute{u}_j\frac{\partial u_j}{\partial x_i} \tag{8}$$

The generation of turbulence kinetic energy due to the buoyancy (G_b) can be neglected. Y_M represents the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate, ($C_{1\varepsilon}$), ($C_{2\varepsilon}$), and (C_μ) are taken as the default values ($C_{1\varepsilon}$ =1.44, $C_{2\varepsilon}$=1.92, and C_μ=0.09) in FLUENT [21]. (σ_k) and (σ_ε) are the turbulent Prandtl numbers for (k) and (ε) taken as 1 & 1.3, respectively. (S_k) and (S_ε) are user-defined source terms [21].

The term (λ) in the energy equation (3) is the effective thermal conductivity which is given by:

$$\lambda = k + \frac{C_p u_t}{Pr_t} \tag{9}$$

The cooling effectiveness is defined as:

$$\eta = \frac{T_m - T_w}{T_m - T_c} \tag{10}$$

4. BOUNDRY CODITIONS.

At all boundaries except those denoted as "main inlet", "coolant inlet", "exit", and symmetry boundary conditions shown in **Fig.(2)**, an adiabatic wall boundary condition is used. At the "main inlet," a velocity-inlet boundary condition is specified with x-velocity equal to 20 m/s, and all other components equal to zero. The temperature is given as 323K at the main inlet. The turbulence intensity and hydraulic diameter (which is used to determine the turbulence length scales) are specified as 2% and 24 mm, respectively. The plenum inlet mass flow rate was adjusted to produce the blowing ratio desired and the inlet temperature of the coolant is 298K to match the coolant to freestream density ratio (DR) of 1.09 with experiments. The turbulence intensity and hydraulic diameter are specified as 3% and 23mm, respectively. At the "exit", a pressure-outlet boundary condition is specified with a gage pressure equal to zero (giving an absolute pressure of 101.325 Pa).

5. SOLVER.

A 3D segregated, steady state solver is used for linearization of governing equations, implicit method is used. For turbulence modeling, (k-ε) model with standard wall functions is used [19&20].To avoid use of enhanced wall treatment the mesh was kept fine enough to have wall Y+ in the range 0-10. Discretization scheme used is 2nd order upwind for momentum as given by [22], turbulence kinetic energy, turbulence dissipation rate and energy, whereas for pressure standard, discretization scheme is used as given by [23]. For pressure-velocity coupling, a simple algorithm is used as given by [24].

6. RESULTS AND DISCUSSION.

For two geometries, two values of the blowing ratio which is defined as $BR=\rho_c U_c / \rho_m U_m$ are considered, namely BR=0.5 and BR=1.0. For comparison purposes, the averaged centerline (streamwise) adiabatic film cooling effectiveness and the averaged lateral (spanwise) adiabatic film cooling effectiveness are used as the main comparison parameters. In the first part of present work, the experimental data given by [7, 18 &13] are used to confine the present CFD results for the case of cylindrical hole shape. Results presented in **Figs. (3, 4&5)** are for averaged centerline film cooling effectiveness and averaged lateral film cooling effectiveness. These results reveal that the centerline effectiveness is in excellent agreement with the experimental results, **Fig. (3)**, in the entire region except in the near hole region when (X/D < 5). Immediate decrease of centerline effectiveness is due to either the mainstream penetration in to coolant jet or due to coolant jet lift-off from the adiabatic test surface, For low blowing ratios (0.5), as the coolant velocities are smaller as compared to mainstream velocity, so jet lift-off is low as clear from higher centerline effectiveness in the near hole region, the immediate decrease of effectiveness for these low blowing ratios in near hole region is due to the penetration of mainstream fluid into the coolant jet. For blowing ratios greater than 0.5, centerline effectiveness decreases to very low values in the near hole region, this is due to the jet lifting-off from the surface, as shown in **Fig.(4)**. CFD results for averaged lateral film cooling effectiveness are in a well agreement with the experimental results for low blowing ratios of 0.5, as shown in **Fig.(5)**, while **Fig.(6)** illustrates the main flow features of a single (cylindrical and conical) hole injected air in the mainstream direction at two blowing ratios of 0.5 and 1. The amount of central spreading of coolant far downstream of the hole is better for conical holes, especially at high blowing ratio. Comparisons of cylindrical and conical holes cases of effectiveness on the downstream test surface of the hole for blowing ratios of 0.5 and 1.0 are presented in **Fig. (7)**. Immediately downstream the holes exit, a so-called horseshoe vortex structure consisting of a pair of counter-rotating vortices is generated. This vortex generation has an effect on the distribution of film coolant over the surface being protected. The fluid dynamics of these vortices is dependent upon the shape of the film cooling holes and blowing ratio, therefore so is the film coolant coverage which determines the film cooling effectiveness distribution. In general, multiple vortex structures are produced where two large vortex structures have been detected, the pair of counter rotating vortices, and horseshoe vortices. The counter rotating vortex are the largest vortex appeared in the flow structure, which occurs at the interaction of coolant jet and hot mainstream, as shown in **Fig. (8)**. The vortex main structures are originated from the holes rims as shown in a front view direction, which agree with most literature [25, 26]. As the vortex propagates downstream the hole, the vortex grows continually downstream, as shown in **Fig. (8)**.

Kidney-vortices have sense of rotation that acts as the coolant lifter. The jet lift-off phenomenon typically occurs at a high momentum ratio (i.e., BR>0.5). For low blowing ratio (i.e., BR=0.5) the momentum ratio is also very low, the mainstream flow departs upward flow pushing the coolant jet towards the surface. This occurs in the cylindrical and conical hole producing low jet vortices levels, and the coolant stays attached to the surface, providing good film coolant effectiveness. But, at high blowing ratio (i.e., BR=1), the jet has high jet vortices levels in the cylindrical hole then the coolant lifted-off from the surface providing bad film cooling effectiveness, but in the conical hole and due to the divergent passage, the coolant jet has no high vortices levels, so it's providing a good film coolant effectiveness, as shown in **Fig.(7)**. The comparison was made between the numerical results of cylindrical and conical holes, at blowing ratio of 0.5 and 1are made in **Figs. (9 - 12)**, the streamwise and spanwise film cooling effectiveness provided by the conical hole is better than that of the cylindrical hole, particularly at blowing ratio of 1. The cylindrical hole provides an averaged centerline film cooling effectiveness of 0.3653 and area-averaged film cooling effectiveness for whole surface of 0.1145, while the conical hole provides 0.4059 for the averaged centerline film cooling effectiveness and 0.1852 for the area-averaged one. The enhancement for averaged centerline cooling effectiveness is 11.11% and 61.75% for the area-averaged film cooling effectiveness. At blowing ratio of 1, the cylindrical hole provides a centerline film cooling effectiveness of 0.2309 and area-averaged film cooling effectiveness for whole surface of 0.0787, while the conical hole provides 0.5153 for the centerline film cooling effectiveness and 0.2303 for the area-averaged one. The enhancement for centerline film cooling effectiveness is 123.32% and 192.63% for the area-averaged film cooling effectiveness.

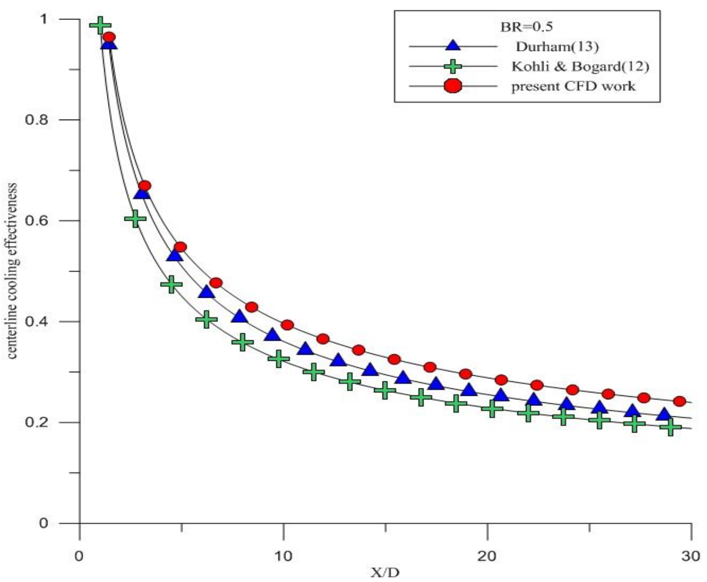

Figure (3): comparison of CFD centerline effectiveness with experiment (BR=0.5).

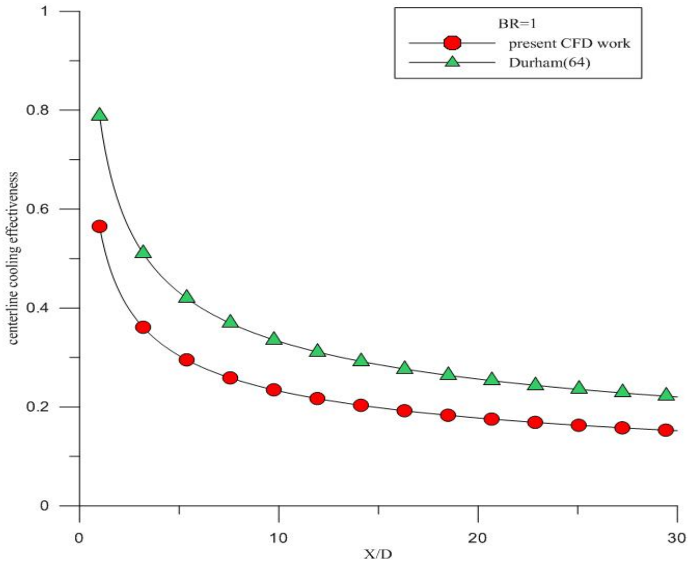

Figure (4): comparison of CFD centerline effectiveness with experiment (BR=1).

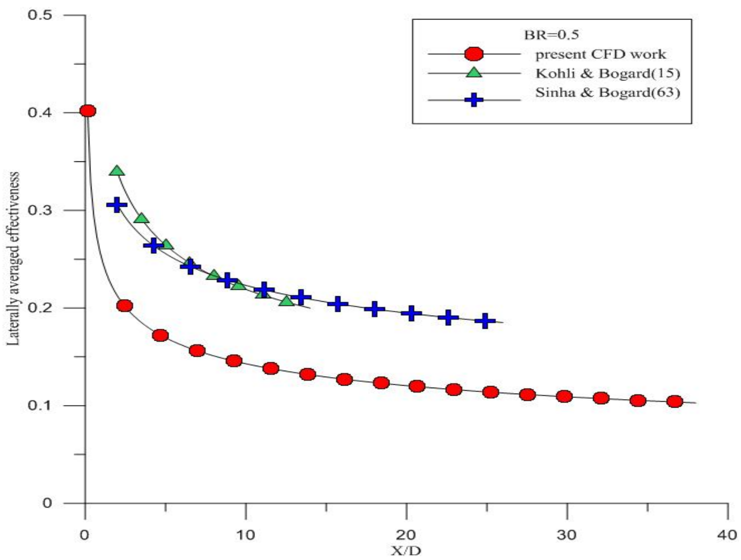

Figure (5): comparison of CFD laterally averaged effectiveness with experiment (BR=0.5).

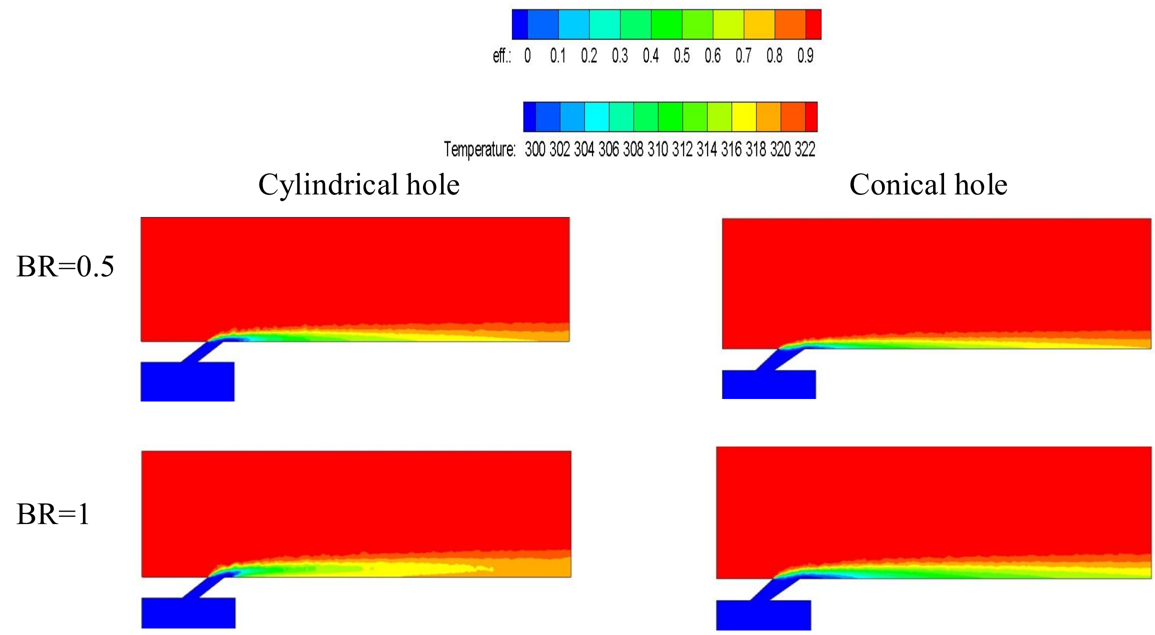

Figure (6): Mainstream and coolant jet interaction contour distribution for cylindrical & conical holes at different BRs.

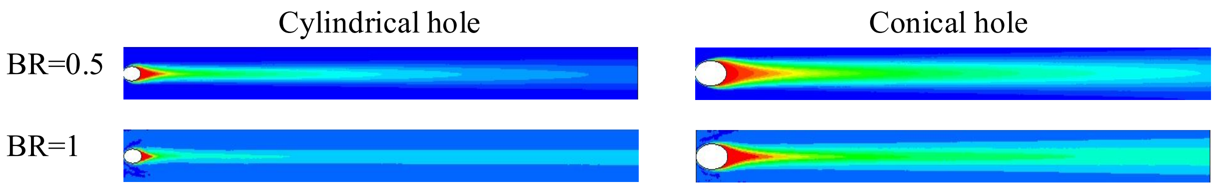

Figure (7): Effectiveness distribution for cylindrical & conical holes at different BRs.

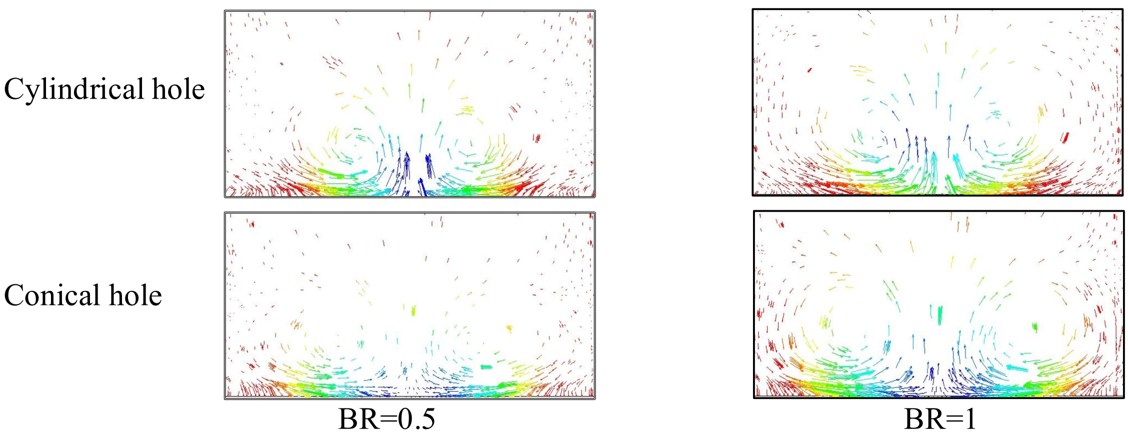

Figure (8): Flow vectors colored by temperature at X/D=3 for single hole forward injection at different BRs.

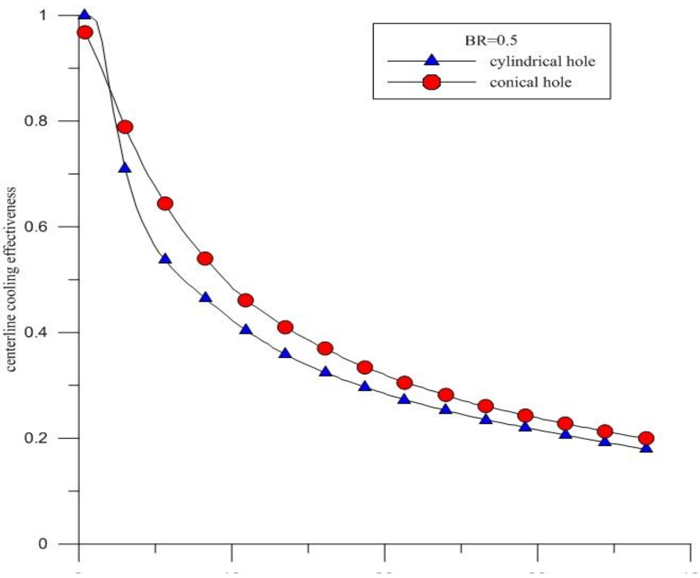

Figure (9): comparison of centerline effectiveness versus nondimensional position between cylindrical & conical hole (BR=0.5).

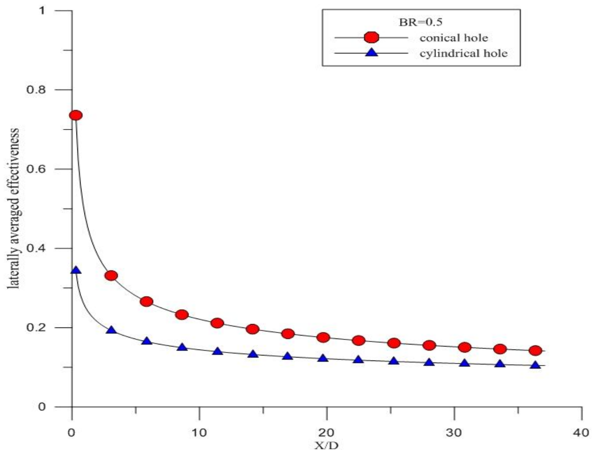

Figure (10): Comparison of laterally averaged effectiveness versus nondimensional position between cylindrical & conical hole (BR=0.5).

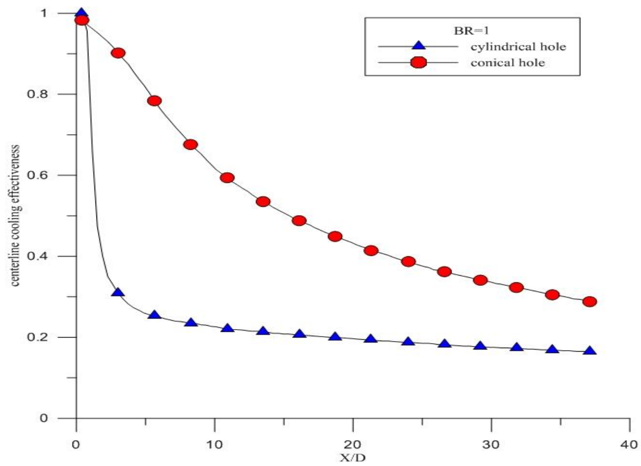

Figure (11): Comparison of centerline effectiveness versus nondimensional position between cylindrical & conical hole (BR=1).

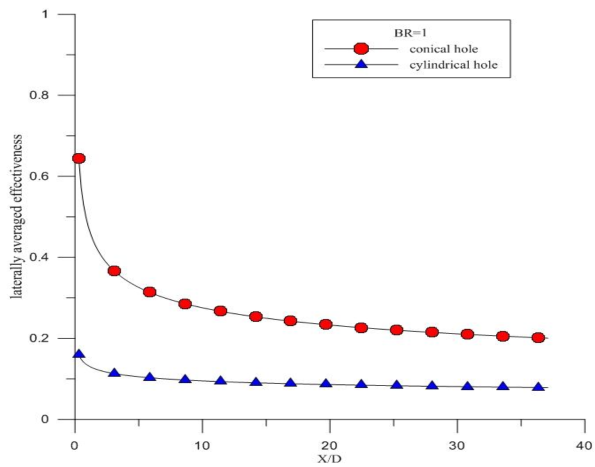

Figure (12): Comparison of laterally averaged effectiveness versus nondimensional position between cylindrical & conical hole (BR=1).

8. CONCLUSION.

1- Bench mark cylindrical film cooling hole case is successfully modeled, the CFD results of present work show a good agreement with the experimental work.
2- It was found that the conical hole gave a greater lateral separation of the kidney vortices immediately downstream of the hole that resulted in increased film cooling effectiveness immediately downstream of the hole and improved lateral distribution of coolant far downstream of the hole.
3- The film cooling effectiveness of conical hole is better than that of the cylindrical hole particularly at high blowing ratio.

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Nomenclature

BR Blowing ratio [$BR=\rho_c U_c / \rho_m U_m$]
C_p Specific heat
D Film hole diameter
DR Density ratios [$DR= \rho_c/\rho_m$]
k Turbulence kinetic energy
h_d Hydraulic diameter
Re Free stream Reynolds number
T Temperature
U Velocity
x Streamwise distance along the test plate
y Coordinate normal to surface

Greek Symbols

η Film effectiveness,
μ Viscosity of air
ε Turbulent kinetic energy dissipation rate
ρ Density
α Injection angle

Subscripts

c coolant
m mainstream
w wall

تحسين فعالية تبريد الغشاء باستخدام فتحات مخروطية الشكل

أ.د.عاصم حميد يوسف
قسم هندسة المكائن والمعدات
الجامعة التكنولوجية

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الخلاصة.

يعتبر تبريد الغشاء احد الطرق المستخدمة لحماية السطوح التي تتعرض لدرجات حرارة عالية كتلك الموجودة في التوربينات الغازية. هذه الطريقة عبارة عن نفث مائع التبريد ليكون غطاء يحمي السطح المراد حمايته تتم عملية نفث مائع التبريد عن طريق ثقوب باشكال متعددة . نتائج استخدام الحل العددي بينت انه نتيجة لتداخل المائعين البارد والساخن تتكون دوامة تعرف بدوامة حدوة الحصان horseshoe vortex تحوي على زوج من الدوامات المتعاكسة counter-rotating vortices تؤثر على توزيع غشاء التبريد فوق السطح المراد حمايته .ان عملية تكون هذه الدوامات تعتمد على شكل الثقب المستخدم لنفث المائع البارد وهذه بدورها تؤثر على توزيع فعالية التبريد ومعامل انتقال الحرارة للسطح. تمت مقارنة النتائج التحليلة لفعالية التبريد وشكل الدوامات لثقب اسطواني عند نسب نفخ ٠.٥ و ١ مع النتائج العملية الموثقة سابقا، اذ تم مقارنة اداء شكل الثقب المخروطي مع الشكل الاسطواني ووجد ان الثقب المخروطي يعطي فعالية اكبر على امتداد مركز الثقب وعرضيا باتجاه الباع. ان استخدام الثقوب المخروطية يؤدي الى زيادة فعالية التبريد الادياباتي عند منطقة اسفل الثقب بنسبة ١١.١١% و 123.2% لنسب نفخ ٠.٥ و ١ على التوالي، بينما كانت الزيادة في معدل فعالية التبريد الادياباتي على امتداد الباع بنسبة ٦١.٧٥% و ١٩٢.٦% لنسب نفخ ٠.٥ و ١ على التوالي.

الكلمات الرئيسية: تبريد الغشاء ، الفعالية ،الفتحات المخروطية ، تحسين الفعالية