Effect of trench geometry on Film Cooling Performance

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Abstract: The main objective of this experimental investigation is to determine both film cooling effectiveness and heat transfer coefficient in an optimized film hole embedded in trench geometry with different blowing ratio. Experimental investigations were done on a flat plate by using a single test transient IR thermograph technique. Evaluation of the cooling performance is obtained by estimated both film cooling effectiveness and heat flux ratios. Two models of coolant jet holes are investigated, each model consists of two rows of holes arranged in a staggered way, Model (1) downstream row with acute angle and the upstream row with obtuse angle; model (2) has the same configuration of reference model 1 but the holes of downstream row are embedded in(3D width and 0.75D depth) trench geometry. This arrangement will allow this work to focus on the effect of trench without the influence of other parameters. The holes diameter is 4mm, the longitudinal distance between the upstream and downstream rows (X/D) is 4D, and the span distance between two neighboring holes (S/D) is 3D. Three blowing ratios of (BR= 0.5, 1.0, and 1.5) were used in the investigation program. . The results showed that when introducing trench within the downstream hole row provides significant improvement to the average cooling performance by (40.3%, 33.4%, 55%) over the rang of the tested blowing ratio. The evaluation of cooling performance did detect well by using heat flux ratio technique rather than effectiveness technique whereas the later did not detects the effects of resistance of heat transfer created by film coolant.

Index Terms— Film cooling, Blowing ratio, Effectiveness, trench.

1 Introduction

Turbine blades require better cooling technique to cope with the increase of the operating temperature with each new engine model. Film cooling is one of the most efficient cooling methods used to protect the gas turbine blades from the hot gases. Jet holes arrangement offers reliable technique help to improve the coolant effectiveness of the film cooling.

Film cooling primarily depends on the coolant-to-mainstream pressure ratio or can be related to the blowing ratio, temperature ratio (Tc/Tm), the film cooling hole location, configuration, and distribution on a turbine elements film cooling. In a typical gas turbine blade, the range of the blowing ratios is of about 0.5 to 2.0, while the (Tc/Tm) values vary between 0.5 and 0.85 Han and Ekkad[1].

Injecting behavior of two rows of film cooling holes with opposite lateral orientation angles have been investigated by Ahn et al. [2] in which four film cooling hole arrangements were considered including inline and staggered ones. Detailed adiabatic film cooling effectiveness distributions were measured using thermochromic Liquid Crystal to investigate how well the injecting covers the film cooled surface. They found that staggered opposite lateral arrangement shows best cooling performance.

Dhungel et al. [3] obtained simultaneously detailed heat transfer coefficient and film effectiveness measurements using a single test transient IR thermography technique for a row of cylindrical film cooling holes, shaped holes. A number of anti-vortex film cooling designs that incorporate side holes. They found that the presence of anti-vortex holes mitigates the effect of the pair of anti-vortices. Experimental and numerical investigations were done by Lu et al. [4 and 5] to measure and to predict the film cooling performance for a row of cylindrical holes. They used adiabatic film effectiveness and heat transfer coefficients were determined on a flat plate by using a single test transient thermograph technique at four blowing ratios of 0.5, 1.0, 1.5 and 2.0. Four test designs crescent and converging slot, trench and cratered hole exits, were tested. Results showed that both the crescent and slot exits reduce the jet momentum at exit and also provide significantly higher film effectiveness with some increases in heat transfer coefficients.

Dia and Lin [6] investigated numerically three film cooling configurations, (cylindrical hole, shaped hole, crescent hole). All holes were inclined at 35 on a flat plate. All simulations are conducted at blowing ratio of 0.6 and 1.25, length to diameter ratio of 4 and pitch-to-diameter ratio of 3. They use (RANS) equations, the energy equation, and two-layer $(k-\epsilon)$ turbulence models. For the numerical investigation the commercial CFD software FLUENT with standard $(k-\epsilon)$ turbulent models is applied. They found that the crescent hole exhibits the highest film cooling effectiveness among the three configurations both in spanwise and streamwise especially downstream of the interaction of the two holes.

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Lee and Kim [7] evaluated the effect of geometric variables of a laidback fan-shaped hole on the film cooling effectiveness using a Reynolds-averaged Navier-stokes analysis. The shape of the laidback fan-shaped hole is defined by four geometric design variables: the injection angle of the hole, the lateral expansion angle of the diffuser, the forward expansion angle of the hole, and the ratio of the length to diameter of the hole. They concluded that the increase of the forward expansion angle makes a reduction of film cooling effectiveness, and the lateral expansion angle has the biggest impact among the four geometric variables on the spatially averaged film cooling effectiveness.

Numerical prediction of Alwan. [8] shows that the flow field structure of injected holes present vortices such as counter pair kidney vortex and horseshoe vortex have major effects on cooling performance, in which the strength of the kidney vortex decreases and the horseshow vortex was lifting up, leading to an improvement in the coolant performance. Therefore numerical model was suitable to design holes arrangement futures of film cooling system by introducing oriented holes row over single jet holes row.

Most literature focuses on the study of the effective parameters of film cooling for one row film holes in the forward direction with mainstream. There is no information available for the row of holes in the backward direction with mainstream flow, also lack information available for two rows of film cooling on forward direction. However, at the present work experimental were done to evaluate the cooling performance (film cooling effectiveness and heat transfer coefficient) by using a single test transient IR thermograph technique for a trench effect and at different blowing ratio.

2 Experimental Facilities

Low speed open duct test rig is used at the present investigation to supply uniform hot air to the test section as shown in figure 1. The settling chamber of the test rig contains a series of electrical heaters and row of screen to ensure adequate hot air of uniform temperature throughout the test rig. The hot air routed through a convergent- divergent contraction having a rectangular cross-section before flowing through the test section. In order to allow the air to reach the desired temperature, the air is initially routed out away from the test section by using a by-bass gate passage. The temperature of the air is continuously monitored at the exit of the gate and when the desired temperature is reached, the gate is gradually fully opened and the hot air is passes into a test section through a rectangular duct. The operating velocity in the test section is controlled to run from 20 to 40m/s. The test section has 50mm width and 100mm height. The bottom plate of the test section is made of (234x123) mm Plexiglas of 10mm thickness and used as the test model.
A centrifugal air blower was used to supply the coolant air to the plenum. The plenum was located below the test model. The coolant air enters a plenum then ejected through holes into the test section. The coolant air pressure is measured at the inlet of the test section. Digital thermometers were used to measure the mainstream and coolant air temperature. Pretesting showed that all holes exists constant desired flow rate and temperature.

Two rows of staggered holes with opposite orientation angles are included in the present study. The orientation angles (γ) is defined as the hole orientation toward the cross-flow in the mainstream and the inclination angle (θ) is defined as the angle between the centerline of the hole and the surface of the test wall as shown in figure 2. Two models of coolant jet holes are investigated. Each model consists of two rows of holes arranged in a staggered arrangement; each of rows (upstream and downstream row) contains eight holes as shown in table (1). Model (1) downstream row with acute angle and the upstream row with obtuse angle; model (2) has the same configuration of reference model 1 but the holes of downstream row are embedded in(3D width and 0.75D depth) trench geometry. This arrangement will allow this work to focus on the effect of trench without the influence of other parameters. The holes diameter is 4mm, the longitudinal distance between the upstream and downstream rows (X/D) is 4D, and the span distance between two neighboring holes (S/D) is 3D. Data collected only for three middle holes for each row to reduce the effects of the side wall as shown in figure (3).

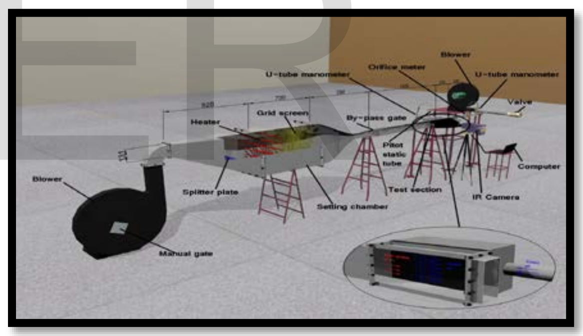

Figure 1 Schematic of the test rig

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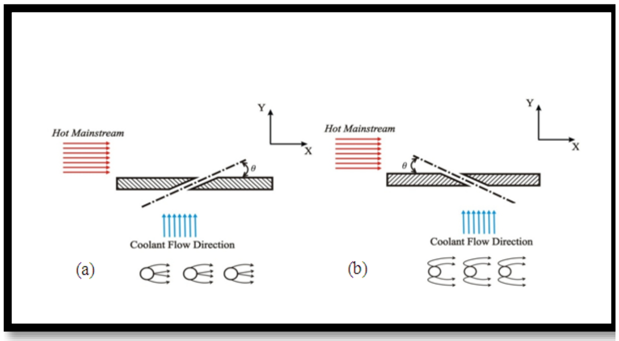


Figure (2) Illustrate diagram of the inclination and orientation angle : (a) $\theta = 30^{\circ}$ and $\gamma = o^{\circ}$, (b) $\theta = 30^{\circ}$ and $\gamma = 180^{\circ}$

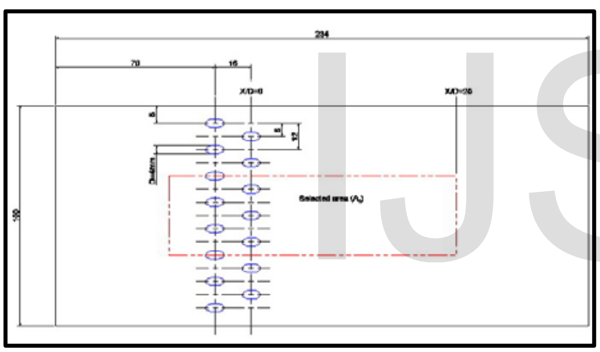
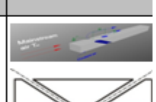
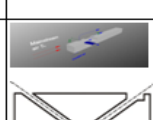


Figure 3 the middle selected area of the test section (A_o)

Table (1) illustrated geometry for the two models.

Model Number	Upstream Row		Downstream Row		Shape
	θ	γ	θ	γ	
Model 1 Stagger	30°	180°	30°	0°	
Model 2 Stagger Grove (depth=0.75D, width=3D)	30°	180°	30°	180°	

The surface temperature of test model was measured using an infrared thermographs technique. IR thermograph infrared camera type Fluke Ti32 is used at the present investigation. This camera is able to precisely record temperature variations. The IR system is greatly affected by both background temperature and local emissivity. The test surface is sprayed with mat black color to increase the emissivity like a perfect black body. The temperature measurement taken is not accurately recorded unless the IR system is calibrated.

THE SYSTEM WAS CALIBRATED BY MEASURING THE TEMPERATURE OF THE TEST SURFACE USING THERMOCOUPLE TYPE K AND THE READING OF IR CAMERA. THE TEST SURFACE IS HEATED BY MAINSTREAM HOT AIR. THE MEASURED OF TEMPERATURES OBTAINED BY BOTH WAYS ARE RECORDED AND STORED DURING THE HEATING PROCESS UNTIL ACHIEVING A STEADY STATE CONDITION. DUE TO THE EMISSIVITY OF THE TEST SURFACE THE TEMPERATURE OBTAINED BY IR CAMERA IS DIFFER FROM THE TEMPERATURE OBTAINED BY THE THERMOCOUPLE, THEREFORE IR CAMERA READING IS ADJUSTED UNTIL BOTH TEMPERATURES READING ARE MATCHED.

2-2 Film cooling effectiveness and heat transfer coefficient estimation

Consider the transient flow over a flat plate as shown in figure 4. In this case the test plate is initially at a uniform temperature Ti, and the convective boundary condition is suddenly applied on the plate at time t > 0. Now, heat assumed to be conducted only in the x-direction and perform an energy balance on the plate, therefore the one-dimensional transient conduction equation is

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha}\frac{\partial T}{\partial t} \quad (1)$$

The main approximation often applied to analyze transient conduction shown in Figure 4 is the semi-infinite approximation. The semi-infinite solid assumptions are valid for present investigation for two reasons. The test duration is small, usually less than 60 seconds. Secondly, the hot air flowing over the test surface made from Plexiglas of, low thermal conductivity, low thermal diffusivity, and low lateral conduction. Therefore the solution of equation (1) as given by Holman and Bhattacharyya [9] is as follows:

$$\frac{T_w - T_i}{T_m - T_i} = 1 - \exp\left[\frac{h^2 \alpha t}{k^2}\right] \operatorname{erfc}\left[\frac{h\sqrt{\alpha t}}{k}\right] \quad (2)$$

Where Tw measured by using IR camera, all the other variables in the equation (2) are either known variable or measured variable except the heat transfer coefficient (h).

In film cooling case, the film should be treated as a mixture of air mainstream and the coolant air, as shown in figure 5, the mainstream temperature (Tm) in equation(2) has to be replaced by the film temperature (Tf), therefore equation (2) become as:

$$\frac{T_w - T_i}{T_f - T_i} = 1 - \exp\left[\frac{h^2 \alpha t}{k^2}\right] \operatorname{erfc}\left[\frac{h\sqrt{\alpha t}}{k}\right] \quad (3)$$

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A non-dimensional temperature term is known as the film cooling effectiveness (η), and is defined as:

$$\eta = \frac{T_f - T_m}{T_c - T_m} \quad (4)$$

Equation (3) has two unknowns (h and Tf), to solve this equation, two sets of data points required to obtain the unknowns like:

$$\frac{T_{w1} - T_i}{T_f - T_i} = 1 - \exp\left[\frac{h^2 \alpha t_1}{k^2}\right] \operatorname{erfc}\left[\frac{h\sqrt{\alpha t_1}}{k}\right] (5)$$

$$\frac{T_{w2} - T_i}{T_f - T_i} = 1 - \exp\left[\frac{h^2 \alpha t_2}{k^2}\right] \operatorname{erfc}\left[\frac{h\sqrt{\alpha t_2}}{k}\right] (6)$$

In this case, a transient infrared thermograph technique will be used to obtain both h η from a single test as described by Ekkad et al. [10]. Thus, two images with surface temperature distributions are captured at two different times during the transient test.

A net heat flux ratio is the ratio of heat flux to the surface with film cooling to the heat flux without film cooling. This term is used to measure the combined effect of film effectiveness and heat transfer coefficient Ekkad and Zapata [11]:

$$\frac{q''}{q_o''} = \frac{h}{h_o}\left(1 - \frac{\eta}{\emptyset}\right) \quad (7)$$

The value for the overall cooling effectiveness (ø) ranges between 0.5 and 0.7.A typical value is Ø = 0.6 according to Albert et al. [12], and this in general assumed in the present experimental analysis.

The IR images for models surface at each investigated test was captured and stored by thermal camera. These images are transferred to PC. Smart View Software program supplied with Camera can be used to limit the selected area to avoid the effect of the test section walls. The IR images converted to corresponding temperature digital values and then saved as data in Excel sheet.

MATLAB programs Software are prepared using a semi-infinite solid assumption to introduce the film cooling effectiveness and heat transfer coefficient contours. Equations, (4), (5), (6), and (7) may be solved using MATLAB Software, Smart View Software, and Excel Software. The data were collected from the selected area denoted by (A_o); this area included only six staggered jet holes as shown in Figure (3).

The measurement uncertainty was determined by using the methodology given by Kline and McClintock [13].Error estimates for each variable are as follows: wall temperature$\Delta T_w = \pm 2$ °C, initial temperature$\Delta T_i = \pm 2$ °C, mainstream temperature$\Delta T_m = \pm 0.2$ °C, and coolant temperature $\Delta T_c = \pm 0.2$ °C. The camera frame rate is 60 Hz resulting in a time error $\Delta t = \pm 0.125$ sec and the test surface properties (α and k) uncertainty are taken from tabulated values, as a custom, 3% relative uncertainty is assumed for both variables. The resulting average uncertainty for heat transfer coefficient and film effectiveness is ±8.2% and ±11.0%, respectively.

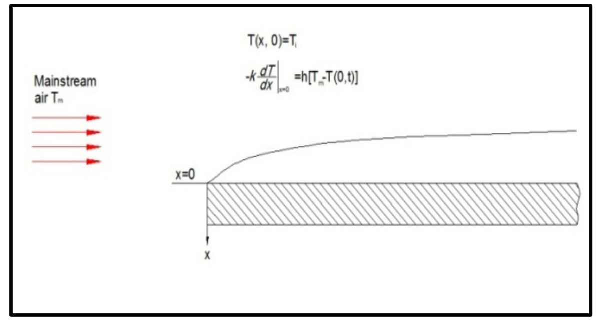


Figure 4 Flow over a flat plate

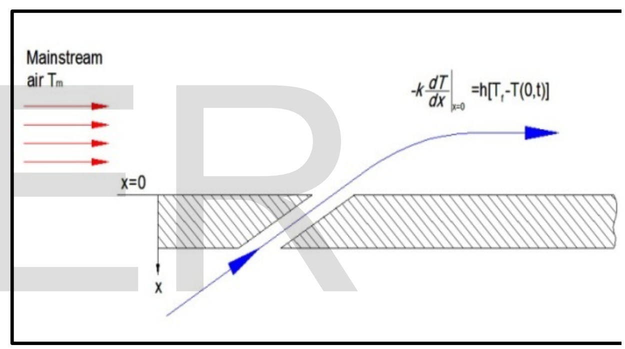


Figure 5 Flow over plate

3- Results and Discussion

Figure (6) show the distributions of the local film cooling effectiveness of models 1 and 2 for three BRs. It appears that the trench provides better performance in the region between any two neighboring holes especially in the near exit hole region. This figure also shows that the effectiveness has significant improvement with increasing blowing ratios particularly for the values in the region immediately downstream the trench. The downstream wall of the slot may provide an obstruction to the coolant jet and as a result, the coolant jet is forced to spread laterally into the trench and decrease the momentum of the jet and hence a reduction in the jet lift-off as well as ejecting out attached to the surface. The coolant jet thus flow closer to the surface rather than undesirable mixing with the hot air.

Figure (7) presents the distributions of the local heat transfer coefficient (h) for the same two cases. The values of heat transfer coefficient near the trench is significantly increased, this mainly due to the flow disturbance near the trench region coming from the impinging of the coolant jet flow at the trench edge. It appears that (h) increases as BR increases. Trench model with BR=1.5 gives a high values of heat transfer coefficient in comparison with the other models.

Figure (8 .a, b and c) shows the spanwise averaged film cooling effectiveness (ηsa) for the same two models. It appears that the trench model gives high (ηsa) than model 1 for all BRs especially in the region near the hole downstream. At X/D>12 and BR=0.5, the trench model gives lowest values of (ηsa) than model 1 as shown in figure (8 .a), but for BR=1.5, the trench model gives high values for all spanwise distance as shown in figure (8 .c).

Figure (9) shows the effect of blowing ratio on the averaged film cooling effectiveness (ηav) for the two models. This Figure shows clearly that the trench model gives high value of (ηav) than model 1. It appears that the averaged film cooling effectiveness increases with increasing BR for two models.

Figure (10) shows that the effect of BR on the heat transfer coefficient ratio (h/ho). It clearly shows that the trench model gives high (h/ho) especially in high BR.

Figure (11) represents the effect of blowing ratio on overall heat flux ratios (q/qo). It is clear from this figure that the two models have positive effects of cooling and (q/qo) decreases with increasing BR. The high value of (q/qo) is 0.4458 for model 1 with BR= 0.5, while the low value is 0.18 for the trench model at blowing ratio of 1.5. The results show that when introducing trench within the downstream hole row (model 2) provides significant improvement to the average cooling performance by (40.3%, 33.4%, 55%) at (BR= 0.5, 1.0, 1.5) respectively. Generally, it appears that the trench model provides better protection than that the model without trench. The only difficulties arise with the trench technique is the exhibition of poor blade mechanical properties.

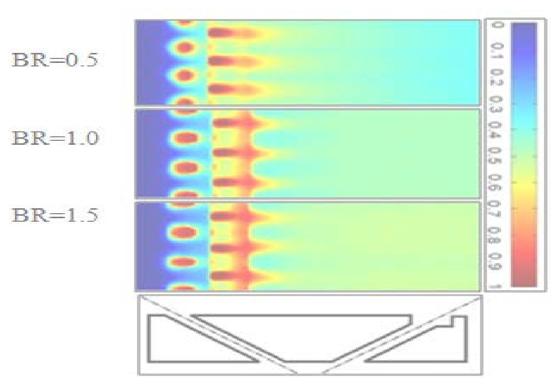
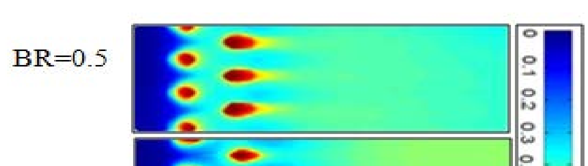


Figure (6) Contours of film cooling effectiveness for model 1(without trench) and model 2 (with trench) at different Blowing Ratio.



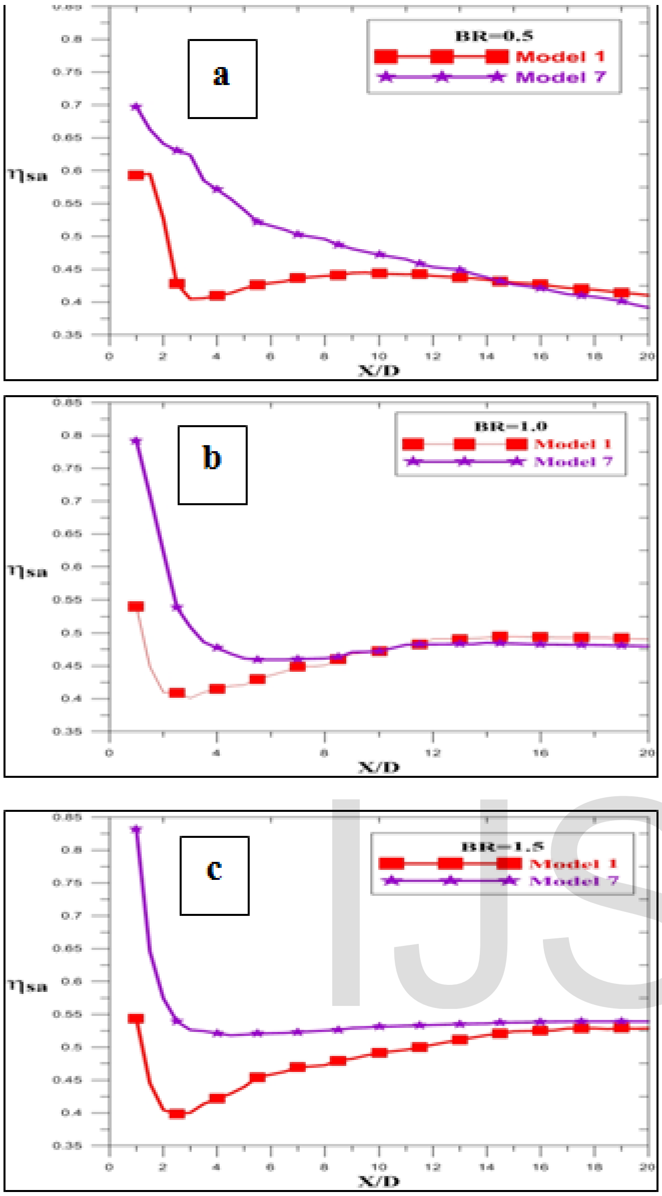


Figure (8) Effect of trench on span wise averaged film cooling effectiveness at: (a) BR=0.5, (b) BR=1.0, (c) BR=1.5.

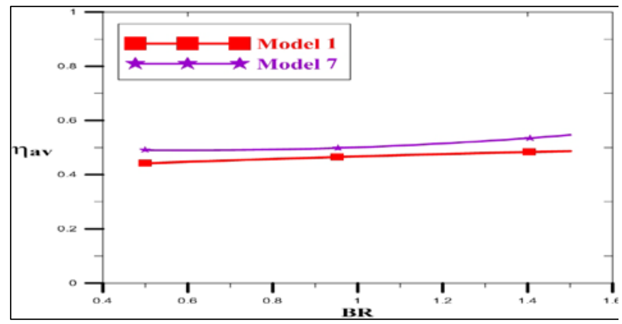


Figure (9) shows the effect of blowing ratio on the averaged film cooling effectiveness

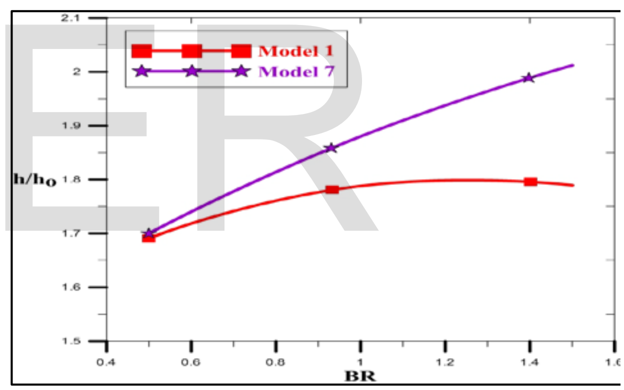


Figure (10) Effect of blowing ratio on averaged heat transfer coefficient.

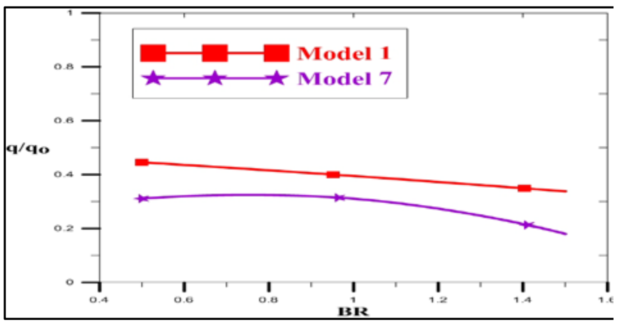


Figure (11) Effect of blowing ratio on overall heat flux ratio.

4 CONCLUSION

The present work has reached to the following conclusions:

1- The trench model provides significant improvement of film cooling effectiveness in the region immediately downstream the trench. Trench model at BR=1.5 shows better heat protection when compared with other model.

2- For low blowing ratio, the film cooling effectiveness is constructed at the holes exit region, while at high blowing ratio, the coolant jets developed downstream give better film cooling effectiveness.

3- The heat flux ratios are found to be less than unity for all holes arrangement, (i.e., good enhancement in film cooling effectiveness).

Nomenclatures

Ao selected area (m2).
BR blowing ratio
CFD Computational Fluid Dynamic
CVP Counter rotating vortex pair
(CVP)P Counter rotating vortex pair in parallel plan to main stream
D film hole diameter (m).
h heat transfer coefficient with coolant injection (W/m² ·K).
ho heat transfer coefficient without coolant injection (W/m² ·K).
K thermal conductivity of test surface (W/m² ·K).
S Span wise hole spacing (m).
T time when the IR image was captured (○C).
Tc coolant air temperature (○C).
Tf film temperature (○C).
Ti initial temperature (○C).
Tm mainstream temperature (○C).
Tw wall temperature (○C).
Uc coolant air velocity (m/s).
Um mainstream air velocity (m/s).
X Stream wise distance along the test surface (hole pitch) (m).
η film effectiveness.
ηsa spanwise average film cooling effectiveness.
ηav average film cooling effectiveness.
Ø overall cooling effectiveness.
α thermal diffusivity.
γ orientation angle (Degree).
θ inclination angle (Degree).

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