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Conjugate Heat Transfer Analysis of Flat Plate Subjected to Impingement and Film Cooling

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Abstract: The present computational study involves a flat plate subjected to combined effect of jet impingement and film cooling. A conjugate heat transfer model in conjunction with k-ω SST turbulence model is employed to study the turbulence effects. The effect of Reynolds number varying from 389 to 2140 on static temperature, Nusselt number and film cooling effectiveness has be discussed for the blowing ratios of 0.6, 0.8, 1.0. The variation in the size of vortices formed on the impinging surface with Reynolds number is studied. It has been observed that the local Nusselt number shows a rising trend with the increase in Reynolds number, while the static temperatures follow the downfall in its values. As a result, an enhancement in the effectiveness is observed, which is credited to the capabilities of combined impingement and film cooling. At Reynolds number of 972, the coolant jet is found to be attached to the surface, for this condition the heat transfer phenomena for blowing ratios of 0.6, 0.8, 1.0, 1.2, 1.6, 2.0, 2.4, 2.6 are studied to understand the flow distribution on the plate surface.
Keywords: Jet impingement, film cooling, effectiveness, conjugate heat transfer

I. INTRODUCTION

Ever since the rise in Turbine inlet Temperature (TET), it has neither proved good for the turbine blades nor for our researchers and has landed them in the challenging grounds. High TET is always a threat to the blades as it degrades them in a short span of time or could melt them in no time leading to catastrophic failures of the aircraft engine. Also low TET means low thrust, which is again hard to accept and definitely not compromised with. The only solution perhaps is the employment of blade cooling techniques which allows these blades to take upon high temperature gases coming out of combustion chamber and still keep them working. Jet impingement with film cooling are two of the prominent methods of cooling, former being a primary and the latter a secondary mode of blade cooling. Both modes have their own positives and negatives, jet impingement offers the highest heat transfer co-efficients but fails to provide any protection to the outer surface of the blade whereas, the film cooling protects the outer surface of the blade but very poor heat transfer co-efficients at the inner surfaces. The newest innovation in the recent years is the combination of these two cooling methods & making it one known as IFC or combined impingement with film cooling. Both the modes complement each other by covering up for each other's weakness & together prove to be a better & improved cooling technique. Now that it is new, much needs to be studied and analysed, also there are various studies explaining various facets of just either jet impingement or film cooling but combined cooling of these two are witnessed by a lack of sufficient studies which can help us to understand it better as well as to improve it. Though, the studies are scarce, but there are enough to setup a base for further research. Researchers Miao *et al.*[5] have taken up combined IFC and compared it with just Film cooling(FC) & no impingement condition. They found out that effectiveness was higher for IFC than FC, also studied the effect of blowing ratio & film hole geometry on the effectiveness. The vortice formation because of the impinging jet has also been studied upon by Kurosaka *et al.*[2] & they have concluded that jet lift-off is influenced by these anti-clockwise vortices along with the entrainment of the cross flow of the fluid towards the plate surface. Indian researchers Chougle N K *et al.* [3] also have conducted CFD experiments on multi-jet impingement on a flat plate and reported that shear stress transport k-ω turbulence model is best suited as it captures the heat transfer and fluid flow phenomenon better than other models. Ashok kumar *et al.* [4] have computationally studied staggered arrangement of multiple circular air jets impinging on a flat plate with effusion holes and predicted the formation of a stagnation zone by secondary vortices and wash region by the primary vortices formed due to interaction of the jets. Rajesh kumar *et al.*[1] have carried out both experimental and computational studies of conjugate heat transfer from a flat plate under the influence of combined cooling of jet impingement and film cooling. They focused upon the effect of blowing ratio on film cooling effectiveness and noticed that the effectiveness improves with the increase in blowing ratio because of the enhancement in both jet impingement heat transfer and film cooling.

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Nomenclature

D	Diameter of film hole, m.	ρ	Density, kg/m^3
D_i	Impingement hole diameter, m.	k	Turbulent kinetic energy m^2/s
C	Specific Heat	h	Heat transfer coefficient, W/m^2K
ω	Specific dissipation rate, 1/sec	H	Height of orifice from the surface, m
ε	Effectiveness	p	Pressure, N/m^2
K	Thermal conductivity (W/m-K)	M	Blowing ratio,
Nu	Nusselt Number, (h x D_i)/K	Re	Reynolds number
T	Temperature, K	Y^+	Dimensionless wall distance
μ	Dynamic viscosity, m^2/s	IFC	Impingement and film cooling
FC	Film cooling	CHT	Conjugate heat transfer
TET	Turbine Entry Temperature	CFD	Computational Fluid Dynamics
SST	Shear Stress Transport	Pr	Prandtl number

II. DETAILS OF NUMERICAL SIMULATION

A. Computational Domain

The two dimensional computational domain for the present study is shown in Figure. 1. with the details shown in Table. 1. The coolant jet hits the impingement plate first, creates a cross flow which then flows through the film holes angled at 35^o and comes out on top of the hot target surface. There are two domains namely mainstream flow & coolant flow, both the domains are closed from all sides. The film holes are straight in nature.

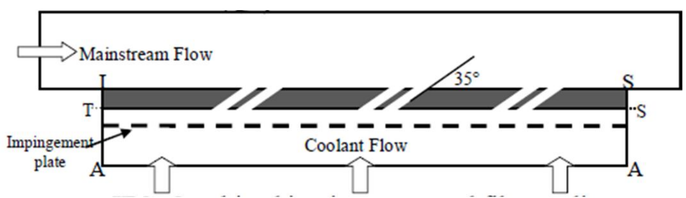

Figure. 1 Two dimensional computational domain

Table-1 Geometry details and parameters

Parameter	Value
Jet orifice (Impingement hole) diameter(D_i)	5.25 mm
Cylindrical Film hole diameter (D)	5 mm
Impinging plate thickness	4 mm
Target plate thickness	12.3 mm
Length to diameter ratio of the film hole	4
Height of the jet orifice from the target surface, (H/D)	1.26

Structural mesh shown in Figure. 2 was generated for the domain using ANSYS® ICEM CFD. A Fine mesh near the wall boundary regions have been modelled, especially at the interaction zones of mainstream flow and the coolant flow. The prerequisite of maintaining a Y^+ value less than 2 is achieved to fulfil the need of turbulence models. Grid independence study was carried out to finalise the mesh density, with its details tabulated in table 2.

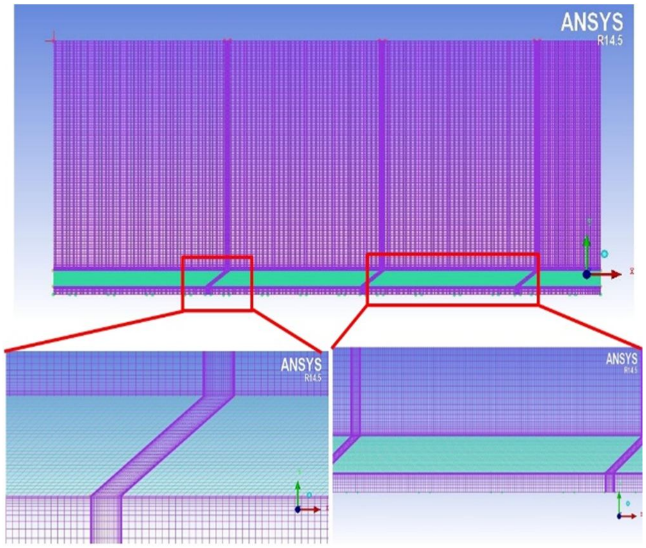

Figure. 2 Two dimensional Computational mesh

B. *Governing Equations*

The mass conservation equation for the fluid domain is

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}(\rho u_j) = 0 \qquad j = 1,2,3 \tag{1}$$

The conservation of momentum equation is

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_k}(\rho u_i u_k) = \frac{-\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu_{eff}\left(\frac{\partial u_i}{\partial x_k} + \frac{\partial u_i}{\partial x_i}\right) - \frac{2}{3}\mu_{eff}\frac{\partial u_i}{\partial x_j}\delta \;\; ij\right) i, j, k = 1,2,3 \tag{2}$$

The total energy equation is

$$\frac{\partial}{\partial t}(\rho H) - \frac{\partial p}{\partial t} + \frac{\partial}{\partial x_j}(\rho u_j H) = \left(q_j + \frac{\mu_t}{Pr}\frac{\partial h}{\partial x_j}\right) + \frac{\partial}{\partial x_j}\left\{u_i\left[\mu_{eff}\left(\frac{\partial u_i}{\partial x_k} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}\mu eff\frac{\partial u_i}{\partial x_i}\delta_{ij}\right] + \mu\frac{\partial k}{\partial x_j}\right\} \tag{3}$$

where u_i, $\mu_{eff}=\mu_l + \mu_t$, H are the velocity, total viscosity and total energy respectively.

The turbulent kinetic energy(k) equation is:

$$\frac{\partial k}{\partial t} + u_j\frac{\partial k}{\partial x_j} = P_k - \beta k\omega + \frac{\partial}{\partial x_j}\left[(\nu + \sigma_k \nu_T)\frac{\partial k}{\partial x_j}\right] \tag{4}$$

Rate of dissipation of turbulent kinetic energy (ω) equation is:

$$\frac{\partial \omega}{\partial t} + U_j\frac{\partial \omega}{\partial x_j} = \alpha S^2 - \beta\omega^2 + \frac{\partial}{\partial x_j}\left[(\nu + \sigma_\omega \nu_T)\frac{\partial \omega}{\partial x_j}\right] + 2(1 - F_1)\sigma_{\omega 2}\frac{1}{\omega}\frac{\partial k}{\partial x_i}\frac{\partial \omega}{\partial x_i} \tag{5}$$

The heat flux q_{ij} is calculated as

$$q_j = -K_f\frac{\partial T}{\partial x_j} \tag{6}$$

where K_f is the thermal conductivity of the fluid

For the CHT model, the energy equation reduces to

$$\frac{\partial}{\partial t}(\rho C T) = \frac{\partial}{\partial x_j}\left(K_s\frac{\partial T}{\partial x_j}\right) \tag{7}$$

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C. Boundary Conditions

The brief description of the boundary conditions are given below-

1) No slip boundary condition

$$V_w = 0, \text{ at wall} \quad (8)$$

2) Flow inlet condition,

$$V_{mainstream} = V_x, \quad (9)$$

$$V_{coolant} = V_z \quad (10)$$

3) Entrainment condition,

$$P = P_{ambient}, \text{ at Outlet}$$

4) Conjugate boundary condition,

$$T_s = T_f$$

$$K_s \frac{\partial T_s}{\partial x} = K_f \frac{\partial T_f}{\partial x} \quad (11)$$

5) Thermal conductivity of stainless steel plate 14.9 W/m-K.
6) The turbulence intensity and turbulent length scale at the mainstream inlet boundary are 1% and 10% of the hydrodynamic diameter and at the inlet boundary of coolant are 1% and diameter of the film hole respectively.
7) Inlet velocities for different blowing ratios of 0.6, 0.8 and 1.0 is given in Table 3. The temperature of both hot air stream and coolant flow are fixed at 318 K & 303 K respectively for all the cases of Reynolds number. Reynolds number is calculated by the formula $\rho V_{oe} D_i/\mu$,which is based upon the impingement hole diameter.

Table.2 Details of the input flow velocities

Blowing ratio	Hot air velocity V_{ha} (m/s)	Coolant velocity V_c (m/s)	Coolant Reynolds number (Re)
0.6, 0.8, 1.0	2	1.2	389
	3	1.8	583
	4	2.4	778
	5	3.0	972
	6	3.6	1167
	7	4.2	1361
	8	4.8	1556
	9	5.4	1750
	10	6.0	1945
	11	6.6	2140

D. Analysis of Grid Independence

The structured two dimensional computational domain was meshed. As shown in table 2, a coarse grid was generated, which was refined to a mesh density of 82120 with a Y^+ value of 0.912 that offered reasonably good results with less computational time.

Table 3- Mesh density details for grid independence study

Type	No. of elements
Coarse	32142
Medium	64566
Fine	82120

To validate the model, the computed static temperature on the hot target surface is taken into consideration and compared with the experimental results taken from [1] as shown in the table 4.

Table-4 Comparison of computed results with experimental results[1]

Study	Static temperature on hot target surface for M-0.8		
	First hole	Second hole	Third hole
Experimental work from [1]	307.5	306.8	306.3
Present work	307	305	303.8
% Deviation	0.16	0.586	0.88

The use of k-ω-SST turbulence model & good quality mesh have rendered values very close to the experimental work done in [1] and is clearly evident from the above table and as shown in the Fig. 3. The obtained computational results are in very close proximity, within ±1% with the experimental values.

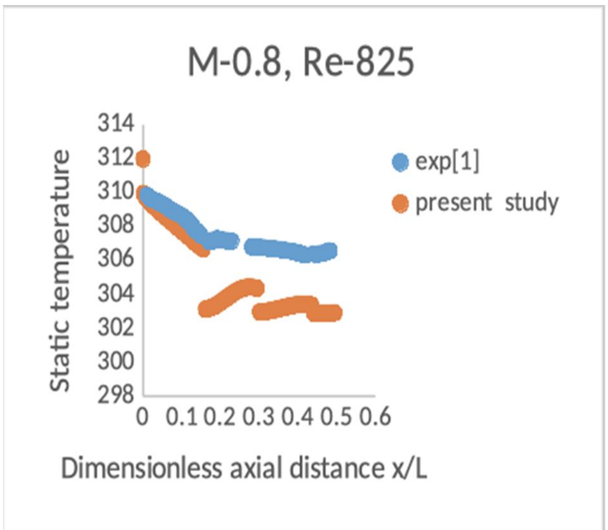


Figure. 3 Model validation of k-ω Turbulence model

III. RESULTS AND DISCUSSION

A. *Part-1: Effect Of Varying Reynolds Number For A Constant Blowing Ratio*

The present study is aimed is to investigate the effects of varying Reynolds number for a constant blowing ratio on static temperature, Nusselt number distribution and film cooling effectiveness on the hot target surface.

1) *Static Temperature Distribution:* Figure. 4 shows the static temperature curves when the coolant Reynolds number is varied from 389 to 2140 for a blowing ratios of 0.6, 0.8 and 1.0 respectively along the dimensionless axial distance of the hot target plate. It has been observed that cooling of the plate is evident at the jet impingement region along the length of the plate. A similar trend is followed for all the blowing rations. It is noticed that, the static temperature gradually drops with the increase in Reynolds number. Also, there is a rise in the local temperature at the extreme left section i.e. prior to first film hole. It is due to the fact, there is no coolant film layer to cover up the surface and the only cooling assistance is the jet impingement, because of which there is a local drop in temperature at locations near to the first film hole with the increase in Reynolds number. Soon after the first hole, the temperature keeps on dropping because of the addition of coolant from the subsequent film holes. Now, with the increase in Reynolds number the heat transfer at the impinging surface increases leading to drop in temperature of the hot target surface.

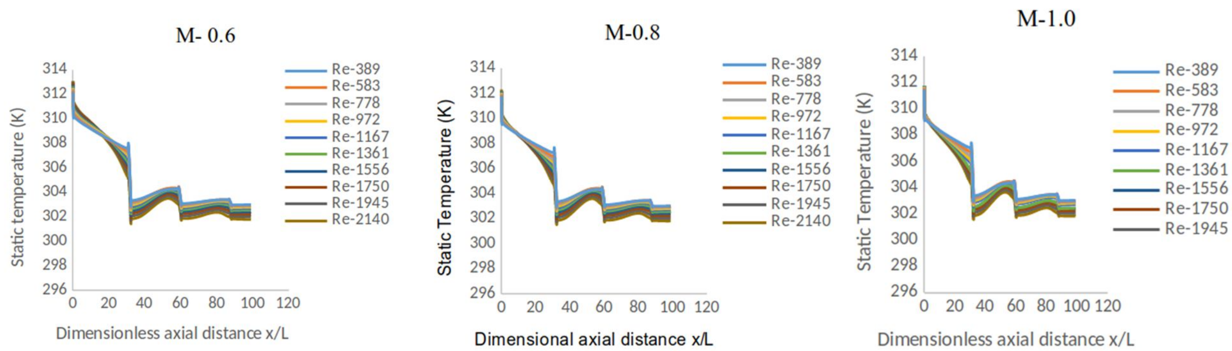


Figure. 4 Static Temperature distribution for blowing ratios 0.6, 0.8 and 1.0

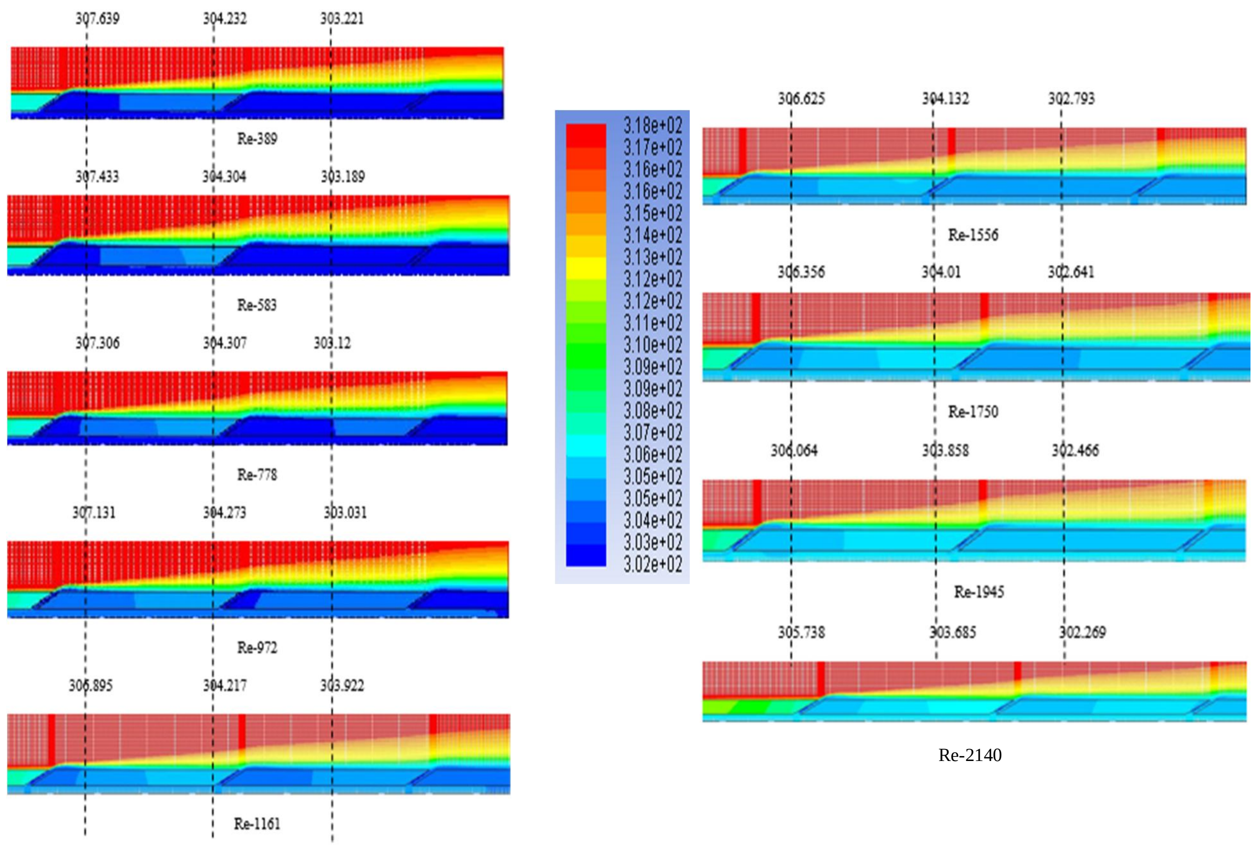


Figure. 5 Temperature contours for different Reynolds numbers

Figure 5 shows the temperature contours at three different locations (140, 260, 350 mm from the leading edge) for a blowing ratio of 0.6 at different Reynolds number varying from 389 to 2140. It can be seen that there is convection heat transfer in the plate, the hot air is heating up the top surface, but due to the formation of thin film on the surface of the place, the plate is getting cooled due to combined effect of impingement and film cooling process. From the plots it can be seen there is development of shear boundary layer owing to the temperature difference between the mainstream flow and coolant flowing over the hot target surface. The width of the shear boundary layer remains the same even with the increase in Reynolds number, because of the fact that blowing ratio is not showing any effect on it.

2) Nusselt Number Distribution

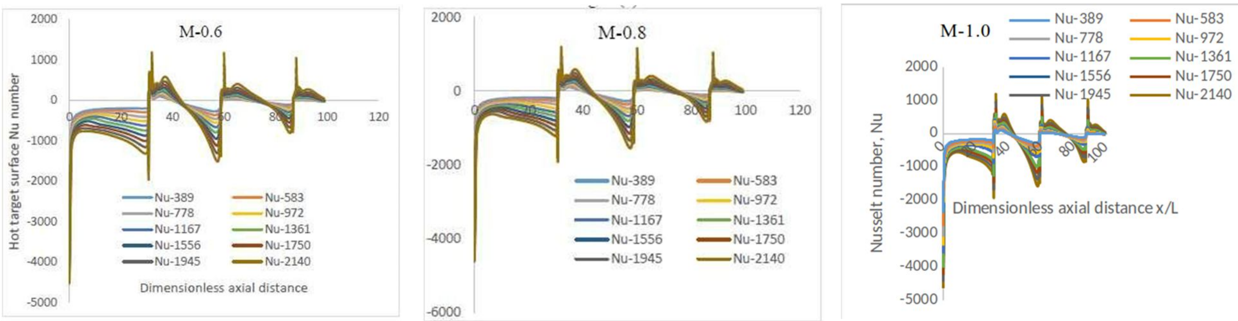


Figure. 6 Nusselt number plots for blowing ratio 0.6, 0.8 and 1.0

Figure 6 shows the Nusselt number distribution on the hot target surface for blowing ratio of 0.6, 0.8 and 1.0 respectively. The trend followed is opposite to what was seen for static temperature, as the Nusselt number increases with the increase in coolant Reynolds number. With the increase in coolant Reynolds number the convection heat transfer dominates the conduction and result in the rise in Nusselt number.

The negative Nusselt number indicates heat transfer from the hot fluid to the structure. At the film hole exit, the Nusselt number changes it's sign to positive indicating a fluid-fluid interaction between the hot gases and coolant. At these locations there is a heat transfer from the hot gases to the coolant entering. Also, the drop in static temperature of the hot target surface is attributed to the rise in the surface Nusselt number.

3) Impact of Film cooling Effectiveness

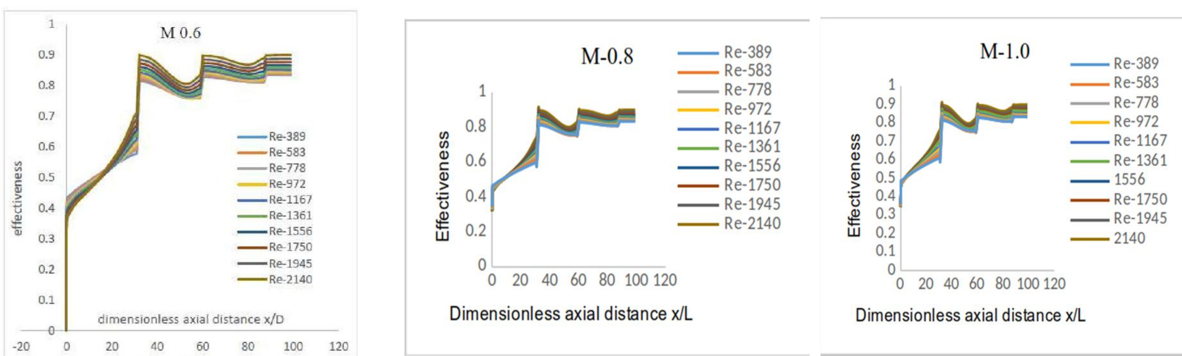
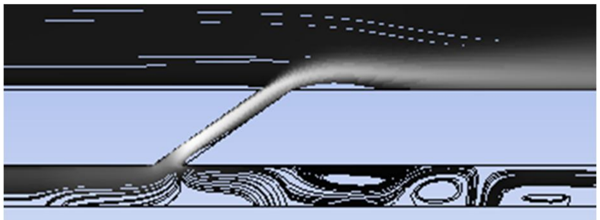


Figure. 7 Film cooling effectiveness plots for flowing ratio 0.6,0.8 and 1.0

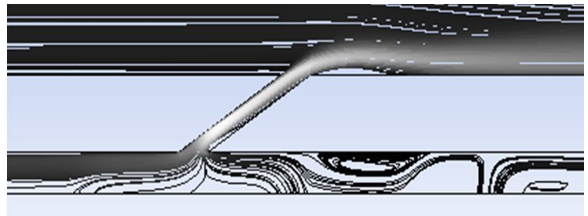
Figure 7 presents the film cooling effectiveness variation with Reynolds number for the blowing ratios of 0.6, 0.8 and 1.0 respectively. It is observed that effectiveness follows an increasing trend, again it can be seen that at locations to the of first film hole the effectiveness keeps on dropping, towards the first film hole exit the effectiveness rises steeply and drops again. It follows the same increase and decrease pattern at the location of second film hole and third set of film hole.

The peak values attained at the locations of film hole exit increase with the coolant Reynolds number. Also, the addition of coolant fluid through second and third film hole along the coolant flow path increases the surface area that comes in contact with the coolant thereby increasing the value of effectiveness along the length of the hot target plate. It was noticed that surface Nusselt number increases with coolant Reynolds number signifying the dominance of convective heat transfer from the coolant fluid to the hot surface and hence this justifies the increasing trend of the film cooling effectiveness.

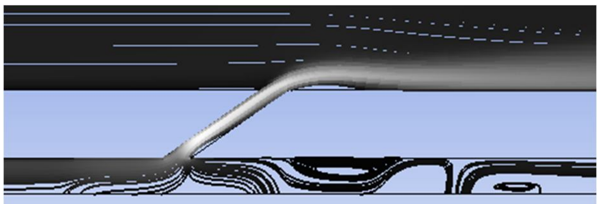
4) Coolant Main Stream Interactions



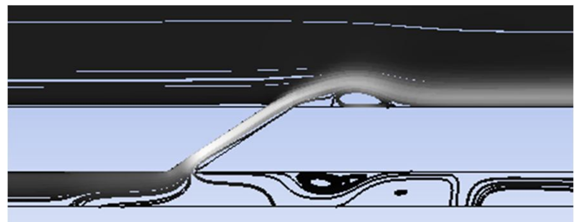
Re-389



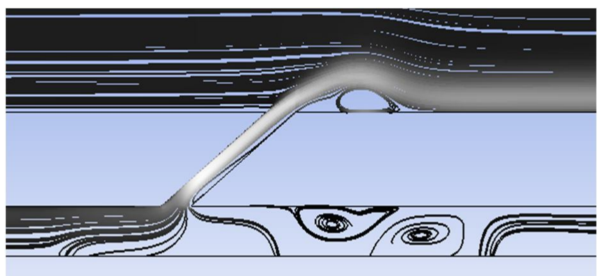
Re-586



Re-972



Re-1560



Re-2140

Figure. 8 Stream line plots for blowing ratio of 0.6

The above streamline plots shows the coolant flow path starting from the exit of impingement jet orifices till it mixes with the mainstream flow over the hot target surface. The streamline flow of the coolant is highly influenced by the fluid velocity. At first, when the impinging jet strikes the cold target surface, a recirculation zone is created characterised by a vortex formation. The size of this vortex changes with the velocity of the impinging jet. At low Reynolds number, there is sufficient time for the spent flow to get recirculated and grow into a big vortex, but with increasing impinging flow velocity there is little time left for the spent jet to form a bigger sized vortex as it is pushed towards the film hole inlet by the incoming high velocity coolant jet. A vortex can be seen just to the right of first film hole inlet, while the same vortex can be seen shrinking at high Reynolds number. At the mixing of the coolant and mainstream flow near the film hole exit, there is a phenomenon called "jet lift-off" because of which the coolant flow coming out of the film hole exit lifts-off from the target surface instead of sticking to it and forming a thin layer. The jet-liftoff should be avoided, as it can affect the film cooling effectiveness. From the above stream line plots, it can be seen that for higher Reynolds number like 1560 and 2140, there is a vortex formation at the exit of the film hole which creates a region of recirculation causing the coolant to lift-off from the surface. It can be noticed that for the Reynolds number of 972 the coolant jet is sticking on to the surface without any jet lift-off. A further study on heat transfer is carried out, by keeping the Reynolds number of 972 as constant and varying the blowing ratios.

B. Part-2. Effect of varying Blowing ratio for a constant Reynolds number

The part of present study's aim was also to investigate the effects of varying blowing ratio for a constant Reynolds number. The main heat transfer parameters like static temperature, Nusselt number and film cooling effectiveness is discussed in the following section.

1) *Static Temperature Distribution:* Figure 9 shows the static temperature curves when blowing ratio is varied from 0.6 to 2.6 for a constant Reynolds number of 972 respectively along the dimensionless axial distance of the hot target plate. It has been observed from the curves that for a higher blowing ratios the left portion of the first film can be cooled, while the portion of the plate to the right of the film hole does not see much change in temperature difference with the increase in blowing ratios. As per the temperature contours showing in Figure 10 the only area that needs to be focused is the extreme left portion of the plate before 1st film hole.

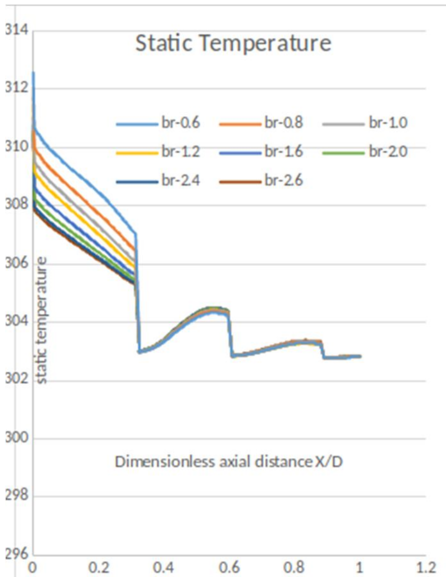
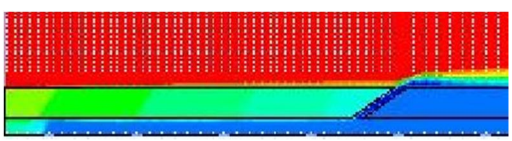
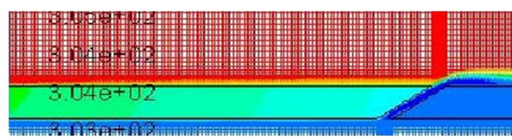
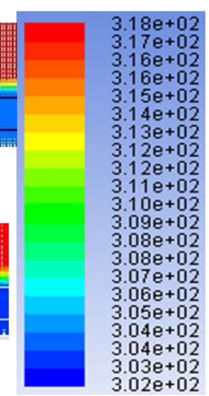


Figure. 9 Static Temperature contous for different blowing ratios

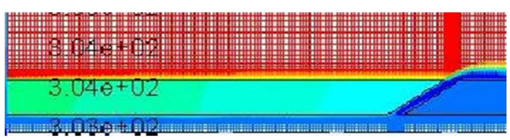


M-0.6

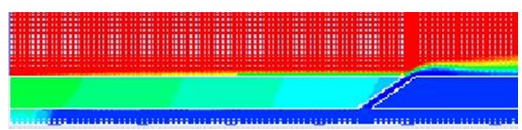




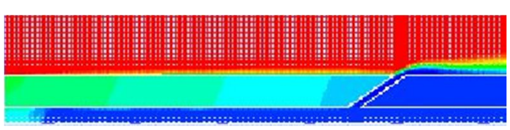
M-0.8



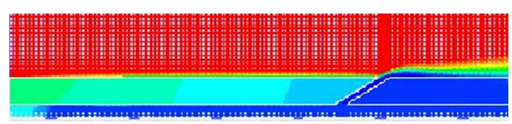
M-1.0



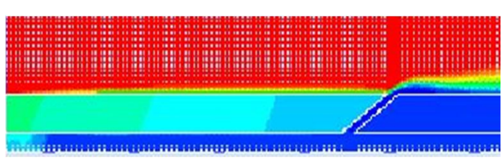
M-1.2



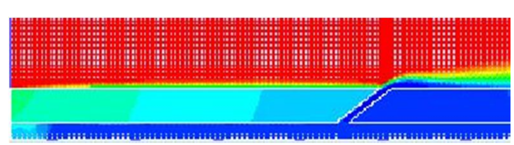
M-1.6



M-2.0



M-2.4



M-2.6

Figure. 10 Static temperature contours or various blowing ratios

As the blowing ratio is increased there is a drop in temperature indicating the cooling of the plate by jet-impingement and the cross flow by the spent jets. At the section of the plate prior to first film hole we can see the decreasing trend of the temperature which is pretty drastic with increasing blowing ratios. With the impinging jet, at a higher velocity than the hot air stream, takes away the heat from the hot target surface, which leads to drastic temperature drop due to the introduction of low temperature coolant air into the main flow. Apart from the drastic temperature drop prior to first film hole exit, no major temperature changes are noted with the increasing blowing ratios.

2) Nusselt Number distribution: Figure 11 shows the Nusselt number distribution on the hot target surface. The negative sign of the Nu indicates that there is a heat transfer from the hot air stream to the hot target plate whereas the positive values soon after first film hole indicates the interaction of the coolant layer with the hot air stream keeping the hot target plate safe without much heat build-up. Also it can be noticed that at the film hole exit, the Nu changes its sign which is due to the fluid-fluid interaction.

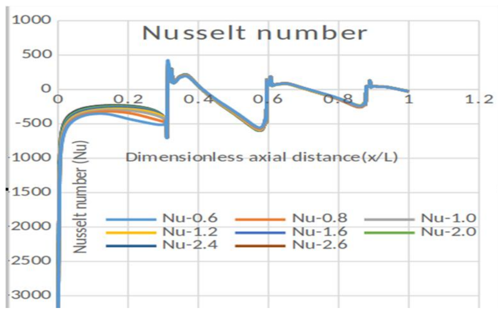


Figure 11. Nusselt Number plot for various blowing ratios at Re 972

The second observation is that Nu curves are moving upwards in the flow direction before 1st film hole exit. This is mainly because of the fact with the increasing in the blowing ratio the incoming hot air stream is slowing down with respect to the impinging coolant, thus making the impinging coolant jet more dominant in carrying away the heat produced. The peak Nu at the location of 1st film hole is the highest followed by the values at 2nd and 3rd film holes respectively. Since, the section between the first two film hole exit receives the least amount of coolant flow, when compared to the other two holes there is sufficient heating up of the surface which can be seen by the negative values in this section. The section between 2nd and 3rd film hole the amount of coolant flowing over the hot target surface increases with the drop in Nu number. The temperature rapidly drops because of the coolant interaction with the hot air stream and temperature of the target plate approaches close to the coolant temperature when moved axially from 1st film hole towards the 3rd film hole exit and that justifies the decreasing Nu peak values.

3.2.3 Impact of film cooling effectiveness: Figure 13 presents the film cooling effectiveness variation with blowing ratios of 0.6, 0.8, 1.0, 1.2, 1.6, 2.0, 2.4 and 2.6 respectively. The effectiveness curves shows an increasing trend with the increase in blowing ratio.

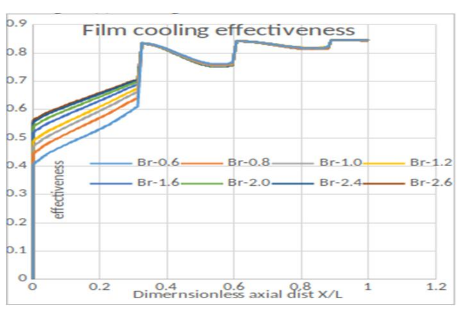


Figure 12. Film cooling effectiveness curves for M=0.6 – 2.6

The rise in effectiveness for the first part of the curve is justified, because of the change in the values of Nu number as it changes from negative to positive followed by decreasing static temperature. After the 1st film hole there is not any significant differences between the curves except for the fact that effectiveness remains flat as the flow move towards 2nd and 3rd film hole exit with the increase in blowing ratio.

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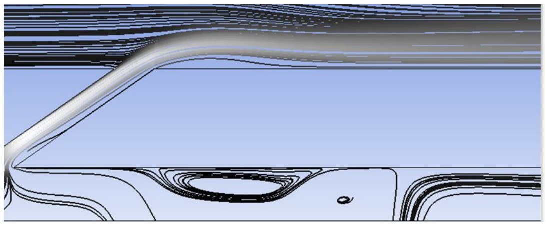
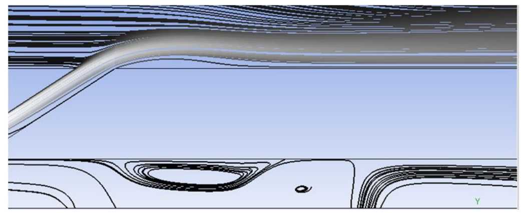
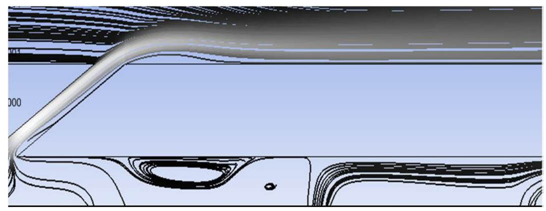
3) *Coolant Mainstream Interactions:* The study of coolant-mainstream interaction is of prime importance when the intention is to focus on film cooling effectiveness. The above streamline plots is needed to understand the flow distribution of coolant on the hot target surface. As the Reynolds number is kept constant with the blowing ratio is increased the hot air stream slows down. Since, the vortex formation by jet impingement on the inner surface of the plate is dependent only on coolant flow velocity. Under these conditions the least that can be is expected is the changes to the inner surface of the plate. It is pretty evident from the above figures which shows the flow distribution and recirculation near the 1st film hole.


M-0.6

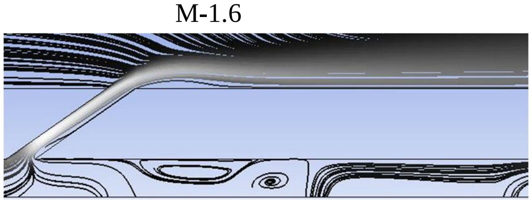

M-0.8


M-1.0


M-1.2


M-1.6


M-2.0


M-2.4


M-2.6

There are two vorteces that can be seen in almost all the figures and out of the two vortexes the bigger one is almost remains same without any noticeable change in its size whereas the smaller vortex varies in size. The other point to be noted is that because of the hot air stream is slowing down with increase in blowing ratio, it affects the mixing of the mainstream flow with the coolant flow at the top of the hot target surface.

During mixing, the energy is lost by both the flow, the flow with a higher velocity accounts for little losses when compared to the flow with less velocity. With a slowing mainstream velocity the energy lost during mixing with a high velocity coolant flow is high and not much is left to mix with the coolant layer and heats up the plate. From the above figures it can be seen that the hot air stream is flowing dominating at the blowing ratio of 0.6 and as the blowing ratio is increased to 0.8 and 1.0 the mainstream flow is sort of converging to the fast moving coolant flow, the trend further grows when the blowing ratio is raised from to 1.2, 1.6 and upto 2.6. With the increase in blowing ratio it becomes hard for the hot air stream to pierce through the dominant coolant flow and get in contact with the hot target plate. The best thing with keeping the coolant flow velocity constant is that there are minimal chances of facing jet-lift-off. But with a slowing mainstream velocity, there is always a concern for this, because high velocity main stream velocity helps in bending down the coolant flow and aiding it in sticking on to the top of the hot target surface. In the present scenario, there is no jet-lift-off that can be seen from any of the cases for the blowing ratios from 0.6 to 2.6.

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IV. CONCLUSIONS

The present study investigates the effect of varying Reynolds number for constant blowing ratio and varying blowing ratio for a constant Reynold number on a flat plate conjugate model with IFC cooling. For studies involving film cooling the primary parameter to be focused upon is the film cooling effectiveness. Because of the conjugate modelling of heat transfer the computational results for the static temperature distribution on the hot target surface agrees to an accuracy of ±1% with the experimental results from [1]. Conjugate models have again proved their ability to produce results very close to the experimental values and contributing to it in the selection of k-ω-SST as the turbulence model.

For the constant blowing ratio condition, there is a clear enhancement in the values of effectiveness with the increase in Reynolds number. Because of the increasing convective heat transfer coefficients or the Nusselt number there is a drop in static temperature of the hot target surface. The coolant exit temperature at the film hole exit is decreasing along the axial distance and continues the same trend with increase in Reynolds number.

Flow with a Reynolds number-972 and blowing ratio of 0.6 offers the best coolant-hot target surface sticking enhancing the effectiveness. If the blowing ratio is varied with a constant Reynolds number, there is a drastic enhancement in the values of effectiveness for the locations before 1st film hole exit. The dominance of coolant velocity over mainstream velocity results in Nu variation to shift towards the positive x-axis indicating better cooling on the top surface. Especially, the cooling for the first section of the plate without film cooling is provided is efficiently cooled by jet-impingement under the influence of high velocity coolant flow and slow main stream flow.

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