Double Slot Jet Impingement Cooling of a Round Block

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This study aims the investigation the effect of impinging of two slot jets on a round block positioned at various angles and aiming at the center of the block. The cumulative effect of the heat transfer along with the flow contours of double jets were studied numerically. RANS approximation is used for the discretized governing equations and the study considered Reynolds numbers of 6000 & 20000 which was defined on the diameter of the round block. The non-dimensional nozzle to block spacing (H/S) of 2 & 8 was analyzed. The fluid is considered as turbulent incompressible two-dimensional flow. The angular placing of the both jets were varied from $\phi=45^o$ to $\phi=180^o$. For the present numerical study. As a methodology, heat transfer data captured using various RANS turbulence prediction models were analyzed with available experimental data from the previous studies to prescribe the best model. In addition, the investigative understanding of control parameters such as distance from the nozzle to round blocks, Reynolds number, curvature parameter (D/S), and ϕ was discussed in detail. The maximum Nusselt number (Nu_D) was found to be the highest when slot jets were positioned at exactly opposite locations.

INTRODUCTION

Air Jets are common in use for many engineering and industrial applications [1-5] due to it's abundant availability, and easy handling properties. The application areas include cooling of very small microcircuits to large Turbine blades. The technology is also widely used in the iron (steel) and Glass industries. The objects cooled with the air jets have considerably amazing shapes including flat surfaces to curved ones. However, the literature available on public space highly emphasizes the flat surfaces and very few articles on curved surfaces are available. These objects includes TMT bars in industries to electronic board components. Sparrow et al. [6], Tawfek [7], Singh et al. [8] and Wang et al. [9] investigated cooling of a round heated object. They aimed to investigate parameters such as Reynolds number (Re_D), distance of nozzles to round block (H/D), and curvature of the round block with respect to heat transfer and fluid flow characteristics.

Gori and Bossi [10] observed that the mean Nussult number maximized when the distance of nozzle and block is 6 times the slot width. They [11] further examined the surface curvature (D/S) of the block with dimensionless curvature parameter (D/S) ranged from 1 to 4, to concluded that the optimal curvature parameter D/S = 2 is the best for cooling the round blocks with slot nozzles. Pachpute and Premachandran [12] study the effects of lower semicircular closure behind a cylindrical object to conclude that the fluid stream, after separation near the stagnation point, move inside the closed walls behind the round block and most of the liquid leaves the system from the confinement opening without any effect near the end of the round block. In the case of smaller confinements, the recirculation strength is higher than in unconfined flows cases or when the confinement is large. Henceforth, a higher cooling rate is observed on the lower surface of the heated round block with smaller confinements. McDaniel and Webb [13] concluded that sharp-edged slots provide better heat transfer than contoured slots. Brahma et al. [14] concluded that the pressure in the stagnation region decreases with higher value when normalized curvature (D/S) and increases with increasing Reynolds number (Re_D).

In a computational study, Olsson et al. [15] compared suggested that SST k-ω turbulence is a better turbulence model to study the jet impingement cooling of the slot jets. In a peculiar study, a heated round block kept on flat horizontal solid plane was investigated by Singh and Singh [16] using the PIV method. They conclude that the separation of the flow together with the round block and the flat surface occurs at smaller angles and lower normalized curvature (S/D) parameters. Gori and Peteracci experimentally [17] investigated the effect of slot widths for the above problem. Their observations conclude that smaller size slot nozzles are more efficient than the larger slot size. Ganatra and Singh [18] performed a computational and experimental study with semicircular closure orifice 0^{O}-120^{O} to find that the SST k-ω is a better turbulence models in predicting heat transfer results. They also mentioned that the Reynolds number does not have a significant effect on flow and heat transfer using bottom closure, and the magnitude of recirculation increases with increasing bottom opening angle.

All these studies performed have given society a significant insight into the energy transfer mechanism of slot jet impingement of a round block concerning heat transfer and fluid flow. However, there is an absence of study performed with two slot jets for the above-mentioned question which might increase or decrease the effectiveness of the whole process. Therefore, this study reports the double slot jet impingement cooling of a round block by numerical method. For this numerical modelling, five different two-equation turbulence models were analyzed, i.e., standard k-ε, realizable k–ε, RNG k-ε, standard k-ω, and SST k-ω models. The study then analyzed the best turbulence model for jet cooling of a round object. The turbulence test conditions were Re_D = 6000 & 20000, H/S = 2 & 8. Letter, the better turbulence model was used to study the double-jet impact of the cylindrical object for various slot angles at the block centre ($\phi = 45^o$ to 180^o).

PROBLEM STATEMENT

The double-jet cooling configuration of the round block is shown in Figure 1. The size of the slot given by S, and the height from the nozzle opening to the round block is H. The angle between the two slot jets is given by ϕ. The length 80S for the slot provided to ensure developed airflow at outlet.

This study investigates the dimensionless current output at around block surface distance (H / S) of 2 and 8. The Reynolds number is based on air properties, mean airflow rate in the slot channel, and the diameter of round block.

$$Re_D = \frac{\rho V D}{\mu}$$

The Nusselt number over the round block were analyzed to present the effects. Nusselt number formulation for the study is given by,

$$Nu_D = \frac{q'' D}{k_a(T_s - T_J)}$$

q" is constant heat flux on surface of the round block, *k* is thermal conductivity of air, T_s is the surface temperature and T_j is temperature of air at the jet exit.

NUMERICAL MODEL

A two-dimensional steady-state numerical study is conducted by considering air to be incompressible under operating conditions. To model turbulence during fluid and energy transfer, the conservation equation with Reynolds Averaged Naiver-Stokes (RANS) approximations was used. The equations of conservative quantities for the present numerical study is given as,

$$\frac{\partial u_i}{\partial x_i} = 0$$

$$\rho u_j \frac{\partial u_i}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \rho\overline{u_i' u_j'}\right]$$
$$\rho u_j \frac{\partial T}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\frac{\mu}{Pr}\left(\frac{\partial T}{\partial x_j}\right) - \rho\overline{T' u_j'}\right]$$

$\rho\overline{u_i' u_j'}$ and $\rho\overline{T' u_j'}$ are turbulent shear stress and heat flux. These variables are solved with the closure, $\rho\overline{u_i' u_j'} = \mu_T\left(\frac{\partial \overline{u_i}}{\partial x_j}\right)$ and $\rho\overline{T' u_j'} = \frac{\mu_T}{Pr_T}\left(\frac{\partial \overline{T}}{\partial x_j}\right)$. Where, Pr_T is turbulent Prandtl number and $Pr_T = 0.85$ has been used. For extra details on these models readers are suggested to go through the literature mentioned in Table 1.

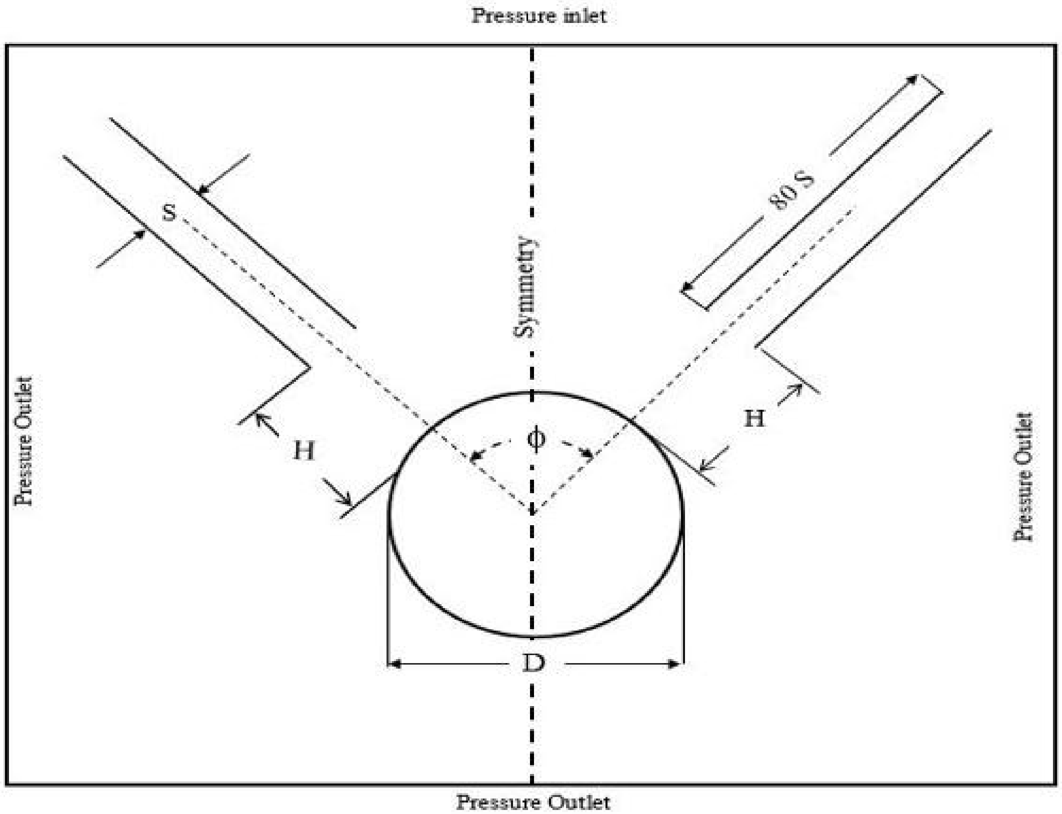


Figure 1 Configuration of double slot jet impingement cooling of the round block.

SOLUTION METHODOLOGY

The present numerical analysis considered the domain to be a two-dimensional symmetric space. The working medium properties were designated as density= 1.225 gr/lit, viscosity = 0.00001789 kg/ms, thermal heat capacity = 1006.43 j/kg K, and K (Thermal Cond.) = 2.42×10^{-2} W/mK, which is comparable as air jet with low Mach number.
FVM discretization with the SIMPLE method of pressure velocity coupling [24] is applied to the solution. The gradients were discretised based on the least square cell-based method. 4 cases with available data in open literature were compared for suitability of RANS turbulence models. For near-wall treatment, the Enhanced wall treatment model is chosen and solutions conversion criteria for continuity, velocity and turbulence quantities were set to10^{-5} and for energy equation, it was set to 10^{-8}.

Table 1. Details of Governing equations of various RANS models used in the present study.

Model	Governing Equation	Model Constants	Reference
Standard k- ω	$\frac{\partial}{\partial x_i}(ku_i) = \frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + \frac{1}{\rho}G_k - \varepsilon$ $\frac{\partial}{\partial x_i}(\varepsilon u_i) = \frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + \frac{1}{\rho}C_{1\varepsilon}\frac{\varepsilon}{k}G_k - C_{2\varepsilon}\frac{\varepsilon^2}{k}$	$C_{1\varepsilon} = 1.44, C_{2\varepsilon} = 1.92$ $C_\mu = 0.09, \sigma_k = 1$ $\sigma_\varepsilon = 1.3$	[19]
RNG k- ω	$\frac{\partial}{\partial x_i}(ku_i) = \frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + \frac{1}{\rho}G_k - \varepsilon$ $\frac{\partial}{\partial x_i}(\varepsilon u_i) = \frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + \frac{1}{\rho}C_{1\varepsilon}\frac{\varepsilon}{k}G_k - C_{2\varepsilon}^{*}\frac{\varepsilon^2}{k}$	$C_{1\varepsilon} = 1.42, C_{2\varepsilon} = 1.68$ $C_\mu = 0.09, \sigma_k = 0.7194$ $\sigma_\varepsilon = 0.7194, \gamma_0 = 4.38$ $\beta = 0.012$	[20]
Realizable K- ω	$\frac{\partial}{\partial x_i}(ku_i) = \frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + \frac{1}{\rho}G_k - \varepsilon$ $\frac{\partial}{\partial x_i}(\varepsilon u_i) = \frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + \frac{1}{\rho}C_{1\varepsilon}\frac{\varepsilon}{k}G_k - C_{2\varepsilon}\frac{\varepsilon^2}{k}$	$C_{1\varepsilon} = \max\left[0.43, \frac{\eta}{\eta + 5}\right]$ $C_{2\varepsilon} = 1.92$ $\eta = \sqrt{2S_{ij}S_{ij}}\frac{k}{\varepsilon}$	[21]
Standard k- ω	$\frac{\partial}{\partial x_i}(ku_i) = \beta^* k\omega + \frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]$ $\frac{\partial}{\partial x_i}(\omega u_i) = \alpha S^2 - \beta\omega^2 + \frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\omega}\right)\frac{\partial k}{\partial x_j}\right] + 2(1 - F_1)\sigma\omega_2\frac{1}{\omega}\frac{\partial k}{\partial x_i}\frac{\partial \omega}{\partial x_i}$	$\beta^* = 0.09, \sigma_{k1} = 0.85$ $\sigma_{k1} = 1, \sigma_{\omega 1} = 0.5$ $\sigma_{\omega 2} = 0.856, \alpha_1 = \frac{5}{9}$ $\alpha_2 = 0.44, C_\mu = 0.09$ $\beta_1 = \frac{3}{40}, \beta_2 = 0.0828$	[22]
SST k- ω	$\frac{\partial}{\partial x_i}(ku_i) = \frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] - \beta^* k\omega$ $\frac{\partial}{\partial x_i}(\omega u_i) = \alpha S^2 - \beta\omega^2 + \frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\omega}\right)\frac{\partial k}{\partial x_j}\right] + 2(1 - F_1)\sigma\omega_2\frac{1}{\omega}\frac{\partial k}{\partial x_i}\frac{\partial \omega}{\partial x_i}$	$\beta^* = 0.09, \sigma_k = 0.85$ $\sigma_{\omega 1} = 0.5, \sigma_{\omega 2} = 0.856$ $\alpha = \frac{5}{9}, C_\mu = 0.09$ $\beta = 0.075$	

DOMAIN AND BOUNDARY CONDITIONS

The computational domain was prepared such that it has an arrangement for 5 slot jets. The jets inlet boundaries were alternatively changed to pressure inlet and the slot walls were changed to the interior without altering the grid structure and mesh refinements. Provision for all five slots was provided in the same geometry at radially symmetrical positions with exactly similar grid refinements. Pressure inlet boundary condition is used at the inflow boundaries. The Air temperature at the domain entrance of pressure inlet is considered as 300 K. *q''* constant boundary wall is considered at the heated round block with a no-slip condition for fluid flow. At the inlet boundary the air velocity is considered with zero gradient with the fluid temperature of 300 K and the length of the slot was made 80 times the slot width. The Nozzle wall is considered as an adiabatic wall. The temperature of 300 K is specified on the Nozzle wall. The mesh size was restructured at the no slip boundaries to achieve a y+ value less than 1 near the heated cylindrical surface.

RESULTS AND DISCUSSION

Grid Independence Study

Structured quadrilateral mesh grids were prepared and the Nusselt number on the cylindrical surface was compared. The grid test was performed taking a high Reynolds number ($Re_D = 20000$) to capture the higher magnification in the grid test results. A total of 5 mesh were studied for curvature ratio (D/S) = 8.5 and nozzle to plate spacing (H/S) = 2 with cell counts of 19808, 24096, 29992, 34544 and 38464. Fig 2 (a) depicts the Nusselt number on the round block from these grids. It was observed that the grids with cell counts of 34544 and 38464 were almost identical in their trends along with the circumferential locations. Similarly four mesh were studied for curvature ratio (D/S) = 8.5 and nozzle to plate spacing (H/S) = 8 with cell counts of 19336, 28632, 42568, 56504. The Nusselt number obtained from these grids are shown in Fig 2 (b). Here, the grids with cell counts of 42568 and 56504 were almost identical in their trends along with the circumferential locations. Therefore, in this study of twin jet cooling of a round block, mesh with a cell count of 34544 and 42568 were used for curvature ratio (D/S) = 8.5 and nozzle to plate spacing (H/S) = 2 and curvature ratio (D/S) = 8.5 and nozzle to plate spacing (H/S) = 8 respectively.

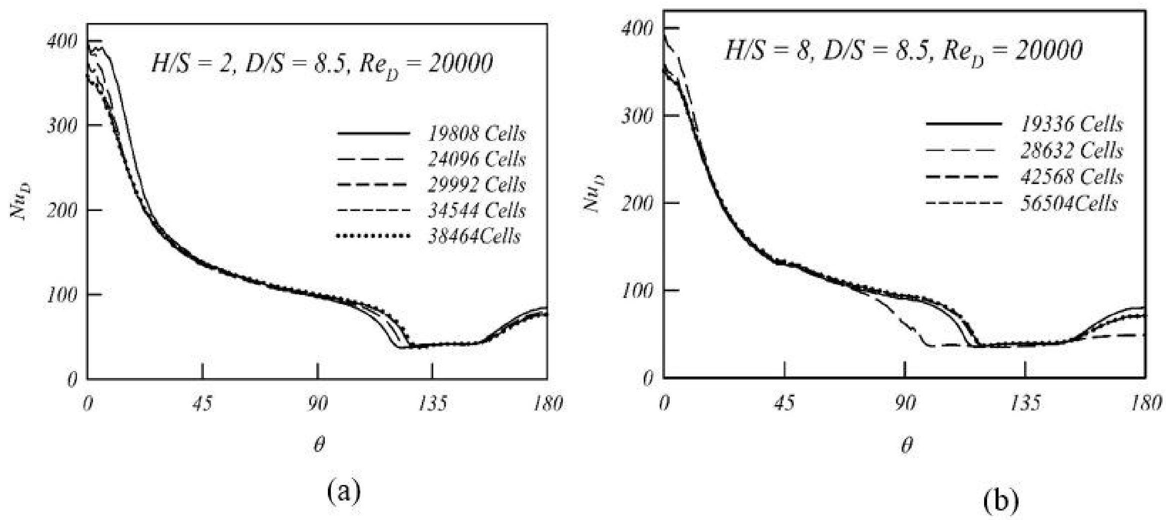


Figure 2 Grid independence test for (a) D/S = 8.5 and H/S = 2, (b) D/S = 8.5 and H/S = 8

Effect of Turbulence Model

Figure 3 depicts the variation of various turbulence formulations on twin air slot jet cooling of the round block for curvature ratio (D/S) = 8.5 and nozzle to plate spacing (H/S) = 2 in Fig 3 (a & b), and nozzle to plate spacing (H/S) = 8 in Fig 3 (c & d), $Re_D = 6000$ in Fig 3 (a & c) and $Re_D = 20000$ in Fig 3 (b & d). Five different 2-equation turbulence formulations namely Std. k-ε, Real. k-ε, RNG k-ε, Std. k-ω, and SST k-ω models were studied to identify the best among these models. In Figure 3 (a), it is seen, that both k-ε models and k-ω models have predicted higher data sets for the Nusselt number at impingement location when matched for the experimental data [12]. However, in figure 3 (b-d) the results of the SST k-ω turbulence model shows better predictions than others at the stagnation point and Henceforth, std. k-ω model is used for the rest of the study on twin air jet impingement study of the round block.

Distribution of Nusselt number

As the problem was symmetric, the locations θ = 0 to θ = 180 was observed in the plots for the effect of changing the angles of slot jets (α) on the Nusselt number on the periphery of the round block in Figure 4 for curvature ratio (D/S) = 8.5 and nozzle to plate spacing (H/S) = 2 for $Re_D = 6000$ and $Re_D = 20000$ in Fig 4(a) and 4(b) respectively. It is observed in Fig 4(a) that the peak nusselt number was observed around 170 to 180 and the peak value location was found to be shifting at θ = α/2 on both sides of the symmetry locations. A secondary peak was seen at θ = 0 and a tertiary peak at θ = 180 due to the formation of recirculation and fluid reattachment zones which will be discussed in detail in the next section. Similar trends were also observed in Figure 4(b) with the peak value of nusselt number at around 310 to 320 at similar locations. It is to be noted that in cases of lower value α, the secondary peak was higher than the tertiary peak which gradually decreases when α increases. This will continue and reverse at α = 180° will further reverse due to the symmetry of the round block and the jets.

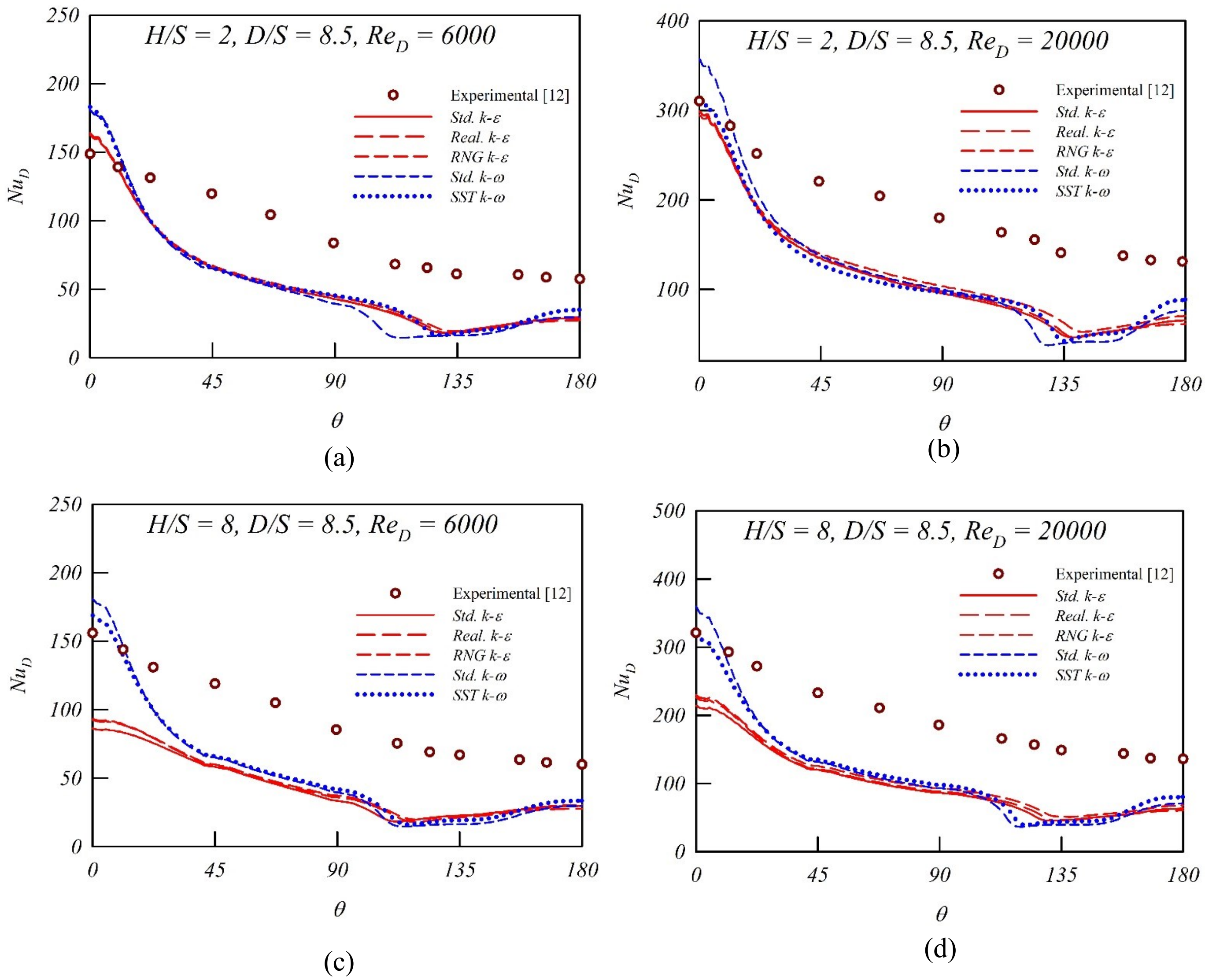
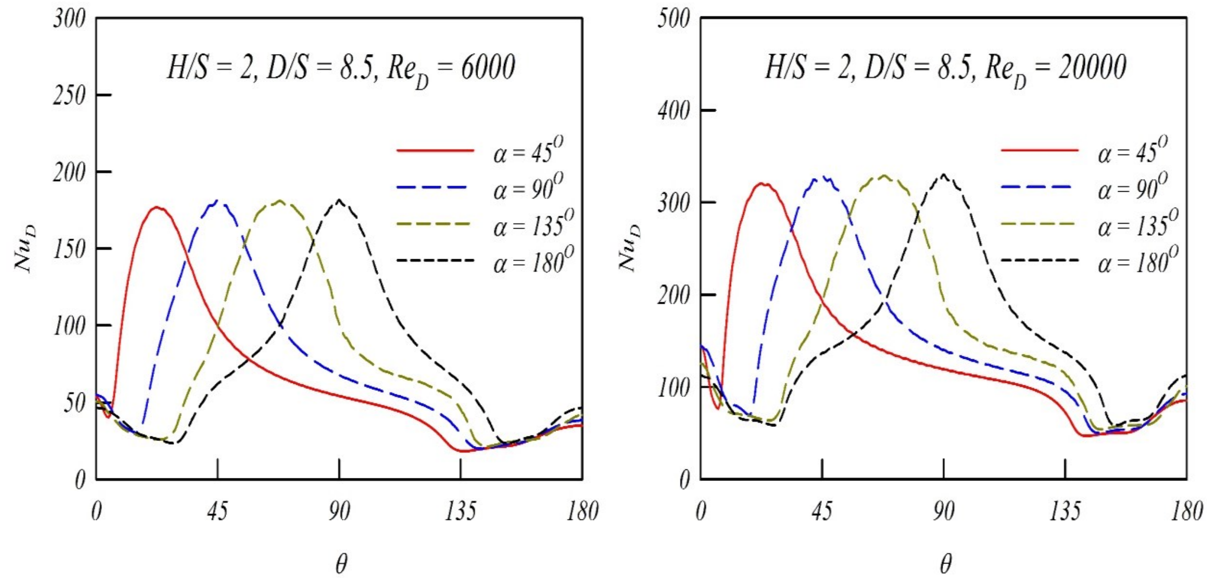
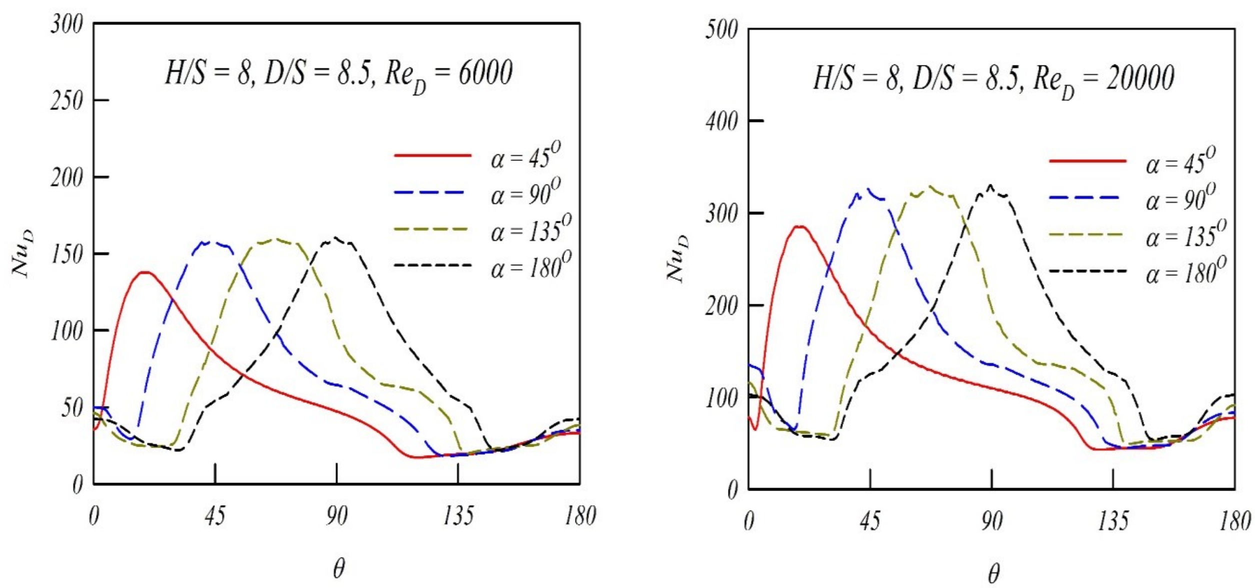


Figure 3 Analysis of Nusselt number distribution for comparison of different turbulence models for (a) D/S = 8.5, H/S = 2 and Re_D = 6000 (b) D/S = 8.5, H/S = 2 and Re_D = 2000 (c) D/S = 8.5, H/S = 8 and Re_D = 6000 (d) D/S = 8.5, H/S = 8 and Re_D = 20000.



(a) (b)

Figure 4. Comparisons of Nusselt number distribution for double slot placements at (a) D/S = 8.5, H/S = 2 and Re_D = 6000 (b) D/S = 8.5, H/S = 2 and Re_D = 2000



(a) (b)

Figure 5 Comparisons of Nusselt number distribution for double slot placements for at (a) D/S = 8.5, H/S = 8 and Re_D = 6000 (b) D/S = 8.5, H/S = 8 and Re_D = 20000.

Figure 5 the effect of changing the angles of slot jets (α) on the Nusselt number on the periphery of the round block for H/S = 8 for Re_D of 6000 & 20000 in Fig 5(a) and 5 (b) respectively. Similar to earlier cases, the three different peaks were observed however, this time, unlike for H/S = 2, the Nusselt number peak was seen to increase as the angles of slot jets (α) increases from 0° to 180°. The highest nusselt number was seen for α = 180° and then reversal is expected due to the symmetry of the problem. Otherwise, the other trends are exactly similar to Figure 4.

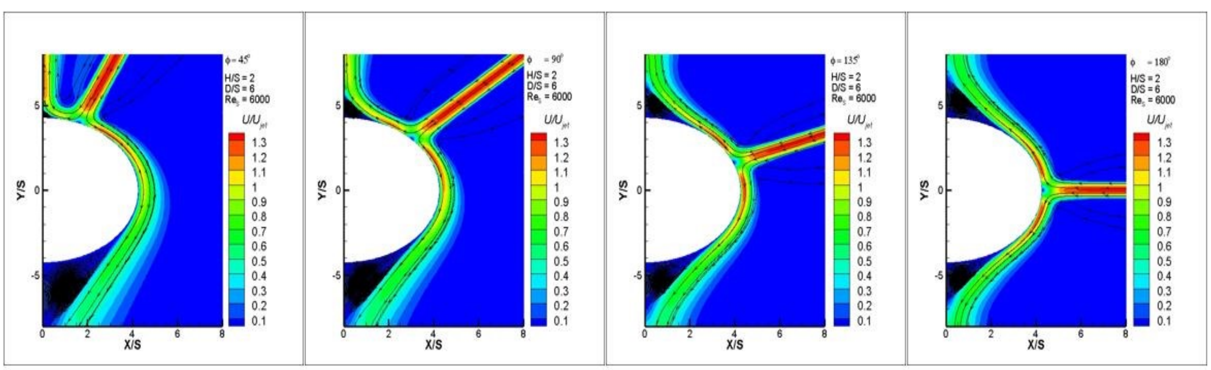
Table 2: Averaged Nusselt number over the circumferential locations of the round block due to double slot jet cooling at an angle of α.

H/S	***2***		***8***	
Re_D	***6000***	***20000***	***6000***	***20000***
$\alpha = 45^o$	*65.992*	*136.08*	*55.775*	*121.08*
$\alpha = 90^o$	*71.139*	*145.33*	*64.886*	*137.65*
$\alpha = 135^o$	*73.545*	*149.89*	*67.5*	*142.52*
$\alpha = 180^o$	*74.423*	*151.64*	*68.233*	*144.01*

The surface averaged nusselt number of twin slot jet cooling of round block is presented in table 2. It can be found that when the H/S ratio was increased from 2 to 8, the averaged nusselt number reduces for all cases whereas, the nusselt number observed enhance due to increasing of the Reynolds number (Re_D). It was also interesting to note that as α increases from 0° to 180°, the averaged nusselt number increases and peaks. This suggests that maximum cooling can be achieved when both the jets are impinging exactly opposite on the round block.

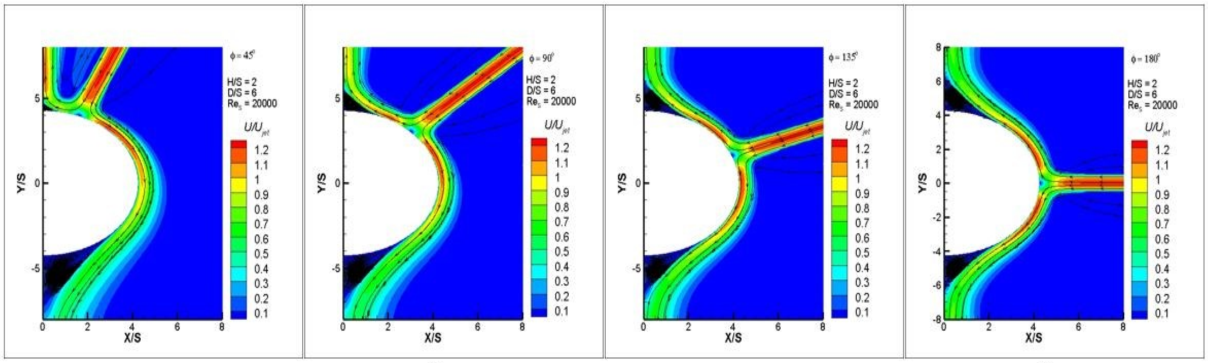
Fluid flow, Separation and reattachment

Figure 6 shows the contours of velocity magnitude during slot jet impingement cooling on the round block for curvature ratio (D/S) = 8.5 and nozzle to plate spacing (H/S) = 2 and Re_D = 6000 including the streamlines. In Fig 6(a) for for ϕ = 45°, It can be seen that after the impingement of the air stream the separation of jet fluid takes place at the θ = α(stagnation point). After the impingement, the one jet steam moves towards θ = 0 and reattaches with the jet mirror jet from the other end. The other jet stream reattached at θ = 180°. A higher amount of fluid moved towards the θ = 0° and lower mass of jet fluid went towards θ = 180° , which is why the second peak value was higher than the tertiary peak. As the slot angles were changed to ϕ = 90°, it was observed that the amount of fluid going towards θ = 0° has reduced whereas the mass of the fluid moving towards θ = 180°, has subsequently increased as depicted in figure 6(b). This trend continues until ϕ = 180° where the mass flow rate of both separated streams was noted to be similar.
The similar observations were also noted for other cases as depicted in Figure 7 for nozzle to plate spacing (H/S) = 2 5 and Re_D = 20000, in Figure 8 for nozzle to plate spacing (H/S) = 8 and Re_D = 6000 and in Figure 9 for nozzle to plate spacing (H/S) = 2 and Re_D = 20000.



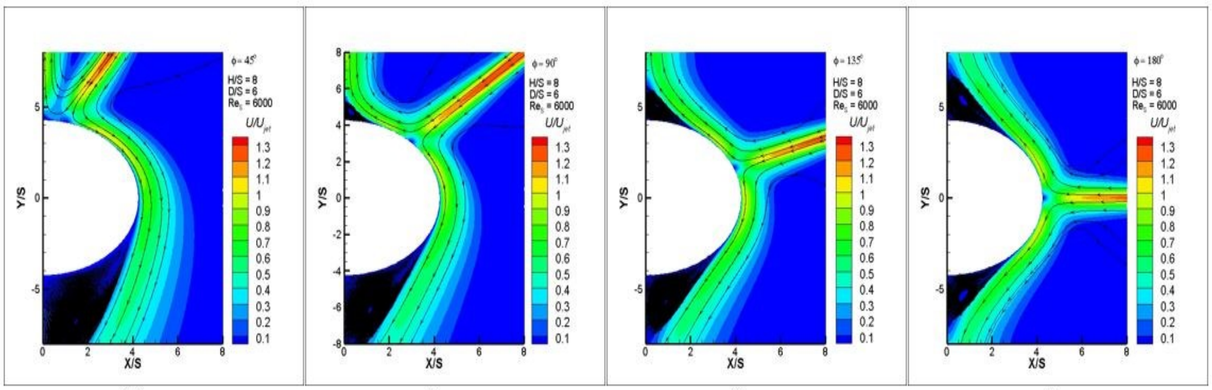
(a) (b) (c) (d)

Figure 6 Effect of ϕ on impinging jets on the Contours of velocity magnitude and the streamlines for Nozzle to plate spacing (H/S) = 2 and Reynolds Number (Re_D) = 6000 at (a) ϕ = 45°, (b) ϕ = 90°, (c) ϕ = 135°, (d) ϕ = 180°.



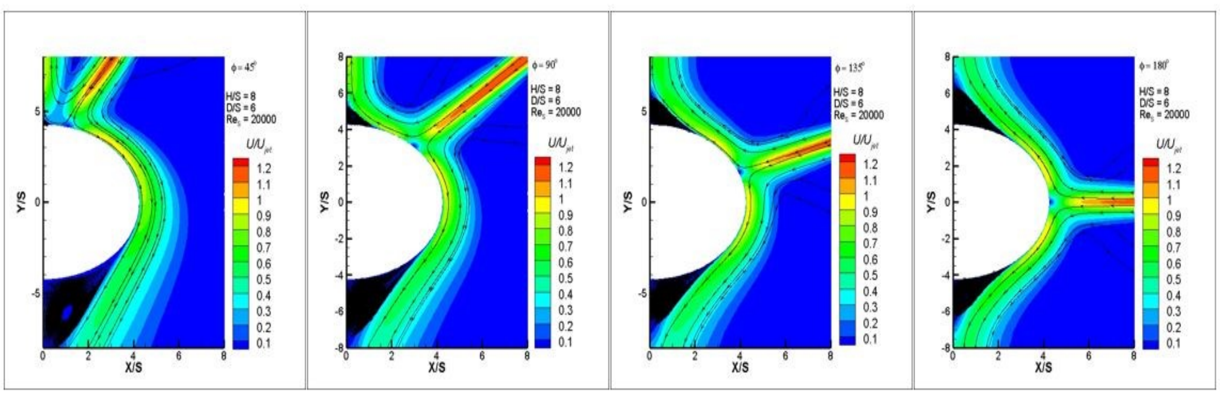
(a) (b) (c) (d)

Figure 7 Effect of ϕ on impinging jets on the Contours of velocity magnitude and the streamlines for Nozzle to plate spacing (H/S) = 2 and Reynolds Number (Re_D) = 20000 at (a) ϕ = 45º, (b) ϕ = 90º, (c) ϕ = 135º, (d) ϕ = 180º.



(a) (b) (c) (d)

Figure 8 Effect of ϕ on impinging jets on the Contours of velocity magnitude and the streamlines for Nozzle to plate spacing (H/S) = 8 and Reynolds Number (Re_D) = 6000 at (a) ϕ = 45º, (b) ϕ = 90º, (c) ϕ = 135º, (d) ϕ = 180º.



(a) (b) (c) (d)

Figure 9 Effect of ϕ on impinging jets on the Contours of velocity magnitude and the streamlines for Nozzle to plate spacing (H/S) = 8 and Reynolds Number (Re_D) = 20000 at (a) ϕ = 45º, (b) ϕ = 90º, (c) ϕ = 135º, (d) ϕ = 180º.

Recirculation zones were observed at $\theta = 0^o$ than at $\theta = 180^o$ and in case of $\alpha = 45^o$, which continuously weakens as α was increased to 90^o and so on. Similarly, the recirculation zone at $\theta = 180^o$ was observed to strengthen as α was increased. At $\alpha = 180^o$, both the recirculation zones were found to be of similar magnitude.

CONCLUSION

A computational study of double slot jet cooling over a round block has been carried out to analyze the heat transfer and fluid flow structure of the entire process to understand the underlying physics involved. The results obtained from the study at $Re_D = 6000 - 20000$, $H/S = 2 - 8$ and $D/S = 8.5$ for $\phi = 45^o$ to 180^o are discussed in detail for heat transfer and fluid flow. The following points are concluded from the above study.

1. During the double slot impingement cooling, to separation zone and two reattachment zones on the round block are observed. At these designated places, the heat transfer rates are slightly higher than at the other locations.
2. The highest average heat transfer from the round block was attained at $\phi = 180^o$.
3. Two regions of recirculation zones were seen on both symmetries near the jet reattachment area. The size of the recirculation areas was larger at the downstream segment.
4. The size of the recirculation zone, secondary and tertiary peaks and the magnitude of the reattached jets were all equal for $\phi = 180^o$.

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