International Refereed Journal of Engineering and Science (IRJES)
ISSN (Online) 2319-183X, (Print) 2319-1821
Volume 5, Issue 4 (April 2016), PP.24-31

Determining relation among Shape of Perforation and Convective Heat transfer from Lateral fin arrangement using Simulation by Computational Fluid Dynamics

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Abstract:- Heat transfer removal from surfaces has a great importance in many engineering application including high heat generation rate from heat source, excessive rubbing or high speed flow problem from solid surface. Fins are being used as an enhancer to facilitate the dissipation of heat from such surfaces in order to maximize the thermal applicability of material. Although size, shape and arrangement of fin can be utilized further to enhance the heat transfer phenomenon. In this article shape factor of perforation is studied for optimizing heat transfer and navel method are introduced to find out the optimum shape for which heat transfer can be maximize for given flow condition.

Keywords:- Ansys13.0, Finite volume method, Nusselt Number, Perforations, Reynolds Number

I. INTRODUCTION

Heat transfer rate is often increased by providing extended surfaces which increases the effective heat transfer area. This is done by adding fins, pins, or other extensions to the heat transferring surface. The cylinder and heads of air cooled engines are finned, the condenser tubes of most domestic refrigerators are attached to extra metal which extends the surface, and in air conditioning systems the outside surfaces of the cooling and dehumidifying coils are provided with fins. Heat transfer is by conduction along the fins which is dissipated to the surroundings by convection from the surface of the fins. The heat transfer rate is substantially increased by providing fins on the side having high heat transfer resistance.

Ankit Vyas, Sunil Gupta, Sandeep Gupta (2014) worked on perforated fins with varying contact ratio with base plate analyzed for the augmentation of heat transfer from the base plate. A Comparison of Nusselt number with Reynold number has been done for various configurations of fins.[1] Kavita H. Dhanawade, Vivek K. Sunnapwar and Hanamant S. Dhanawade (2014) experimentally investigated increase in heat transfer over horizontal flat surface with rectangular fin arrays , lateral square and circular perforation by forced convection. They compared the results obtained from rectangular fin with circular/square perforations with an equivalent rectangular solid fin and found that percentage improvement in temperature of square perforated fin arrays is more than fin arrays of circular perforated fin of same size [2,3]. B. V. S. S. S. Prasad & A. V. S. S. K. S. Gupta (1998) suggested a method for reducing the weight of a straight rectangular fin by providing a semicircular cut at its tip [4]. U. Akyol and K. Bilen (2006) studied heat transfer and friction loss characteristics in horizontal rectangular channel having attachments of hollow rectangular profile fins. Two arrangements in-line and staggered fin were studied for one fixed span wise ,four different stream wise distances and correlation equations for Nusselt number and friction factor are determined [5]. N. Souidi, A. Bontemps (2001) worked on counter current gas liquid flow in narrow rectangular channels simulated by plain fins and perforated fins [6]. Abdulla H.M, AlEssa (2008) worked on heat transfer dissipation from horizontal rectangular fin embedded with triangular perforations numerically using finite element technique [7]. Heat dissipation rate of solid fin and perforated fin are calculated and compared. D. Abdullah H. AlEssa (2012) studied increase in natural convection heat transfer from a rectangular fin with rectangular perforations. Results obtained shows that for different dimensions of rectangular perforations there is an increase in heat dissipation of perforated fin over that of identical solid fin with the increase in perforations [8]. A.H. AlEssa et al, (2008,2009) studied increase in natural convection heat transfer from a horizontal rectangular fin embedded with rectangular perforations of aspect ratio of two with the help of finite element technique. From results obtained it is found that heat transfer coefficient of fin surface can be increased by introducing surface roughness and therefore promoting turbulence [9,10,11]. Wadah Hussain and Abdul Razzaq Al-Doori (2011) conducted an experiment to study heat transfer by natural convection in a rectangular fin plate with circular perforations as heat sinks [12]. Mohamad I. Al-

Widyan and Amjad Al-Shaarawi (2012) studied numerically perforated fins under natural convection. Two geometric parameters, spacing between holes and hole diameter are taken for analysis and it is found that heat transfer from perforated fins increases with Grashof number and also by decreasing spacing between the holes [13]. S. Chamoli, R. Chauhan and N.S.Thakur (2011) worked on Computational Fluid Dynamics (CFD) to examine the heat transfer and friction loss characteristics in a horizontal rectangular channel which is having attachments of circular profile fins over one of its heated surface [14]. O. N. Sara, T.Pekdemir, S.Yapici, M.Yilmaz (2001) studied increase in heat transfer and corresponding pressure drop over a flat surface in a channel flow due to perforated rectangular cross-sectional blocks attached on flat surface [15]. Md. Ashiqur Rahman, Taifur Rahman, Md. Hasibul Mahmud and Md. Mahbubul Alam (2005) worked numerically as well as experimentally increased forced heat transfer characteristics from a horizontal flat plate provided by solid and drilled fins of rectangular profile. They found changes in Nusselt number and heat transfer rates due to use of number of equally spaced drilled fins placed at heater surface at different humid condition and a comparison is made with that of equivalent solid fins [16]. M.R. Shaeri, M.Yaghoubi and K Jafarpur (2009) studied numerically three dimensional incompressible laminar fluid flow and heat transfer from an array of solid and perforated fins mounted on a flat plate. Navier Stoke equations and RNG based k- ε turbulent model is used to predict turbulent flow parameters. SIMPLE algorithm of finite volume method is used to solve conjugate differential equations for gas and solid phase [17,18]. R. Sam Sukumar, G.Sriharsha, S. Bala Arun, P.Dilip kumar and Ch.Sanyasi Naidu (2013) studied fluid flow and heat transfer characteristics of standard continuous heat sinks of different designs through Computational Fluid Dynamics to determine heat sink which is most excellent for efficient cooling of electronic devices [19]. Bayram Sahin and Alparslan Demir (2008) studied increase in heat transfer and pressure drop over a flat surface provided with square and circular cross-sectional perforated pin fins in a rectangular channel. Results obtained shows that there is an increase in turbulence of flow in longer fins resulting in an increase in heat transfer and also there is an enhancement in heat transfer with perforated fins higher than that of solid fins [20,21]. R.B. Gurav, J.D. Patil, S.M. Gaikwad, Pritee Purohit and A.A. Ramgude (2013) studies heat transfer augmentation from a horizontal rectangular fin embedded with elliptical perforation under natural convection compared to non-perforated fin using finite volume method [22]. Rupali V. Dhanadhya, Abhay S. Nilawar and Yogesh L. Yenarkar (2013) studies augmentation of heat transfer from horizontal rectangular fin with circular perforations under natural convection compared with solid fin [23].

II. MATERIALS AND METHODOLOGY

In the present study fluid flow and heat transfer in perforated fin array with Rectangular, Circular and Elliptical channel using computational fluid dynamics has been carried out. The simulation set up consists of a duct of 220 mm x 220 mm cross sectional area with 280 mm test length. The test section consists of a base plate of 200 mm x 200 mm (width x height) having thickness 6 mm. The base plate was placed inside the duct in such a way that its lower base is at 15 mm from the bottom of the duct surface. Upper surface of base plate was exposed to the duct environment. Its position is horizontal and in suction mode. Setup geometry is created in Ansys 13.0 and shown in Figure 1.

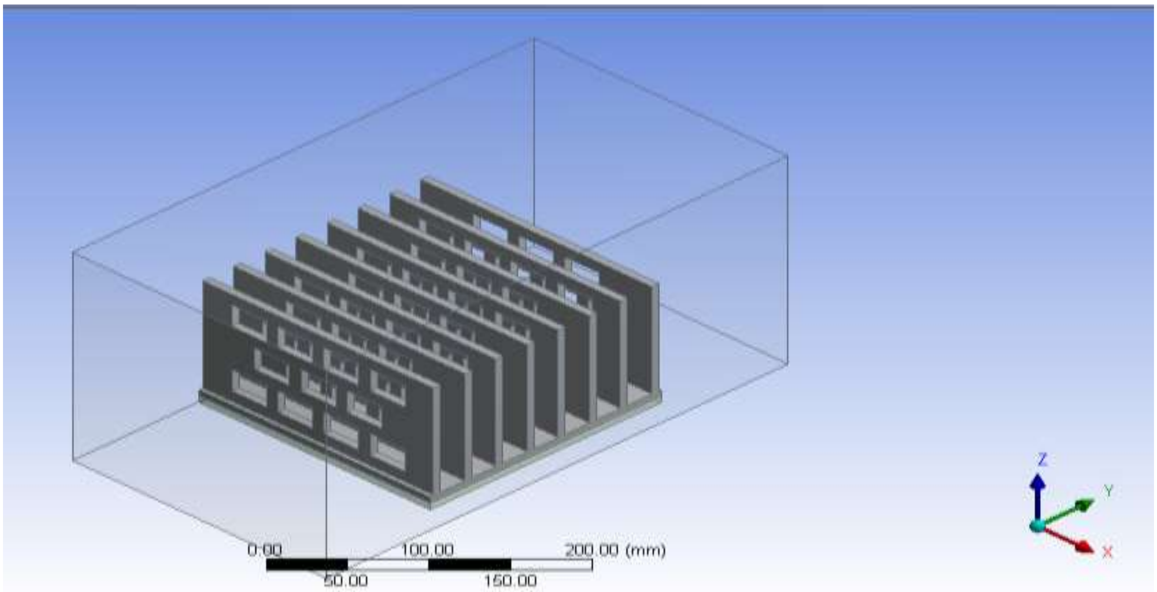

Fig. 1: Experimental setup created in Ansys 13.0 for simulation

A total of eight fin arrays are placed on base plate having cross-sectional area 200 mm x 100 mm (width x height) each having thickness 6 mm. The distance between each individual fin is 21 mm on base plate and distance of fins from left and right side is 3 mm and 2 mm. The simulation is conducted at 515 watts heat input. The air flow is considered to be steady and laminar with constant properties. The fin material is aluminum. Peroration position for Rectangular shape is shown in Fig. 2(a) and Fig. 2(b) for front and adjacent to the front plate.

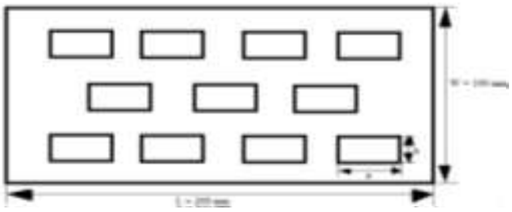

Fig. 2 (a): Front plate perforation arrangement

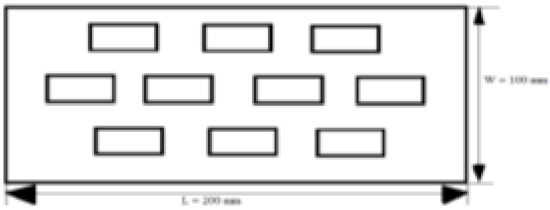

Fig. 2(b): Second plate perforation arrangement

III. DATA REDUCTION

At the steady state all the energy supplied to the base plate is convected by the upper surface of base plate (as lower base surface is insulated) and the fins provided on the base plate. Also resistance between contact of perforated fins and base plate is neglected i.e. contact resistance between fin surface and base plate is equated to zero. As heat input to base plate and perforated fin system is 3333.3 W/m^2 of the base plate. Total heat supplied to the system can be calculated using formula given by equation (1) as :-

$Q_N = q_N \times A_f$ (1)

Where q_N = 3333.3 W/m^2 and A_f = base plate area= 40000 – 48= 39952 mm^2

Q_N can also be calculated using equation (2) in terms of average heat transfer coeficient as:-

$$Q_N = h_{av}\, A_T\, [T_s - \frac{(T_{out} + T_{in})}{2}] \quad (2)$$

Where h_{av} = Average convective heat transfer coefficient ,
A_T = Total wetted Surface area = 100 x 200 + 2x8 + 200x200 – 8x6x200 = 50416 mm^2,
$T_s = T_{out} = T_{av}$= Outlet Temperature
and T_{in} = 22+273 K = Inlet Temperature.
Using equation (1) and equation (2) value of h_{av} can be obtained.
Dimensionless Nusselt number, Reynold number and Prandtl number can be derived as follows in equation 3(a), 3(b)and 3(c) by:-

$Nu = \frac{h_{av} L_c}{k}$ (3a)

$R_e = \rho v D / \mu$ 3(b)

$Pr = \frac{\mu\, Cp}{K}$ 3(c)

Where L_c(Characteristic Length) $= \frac{4A}{P}$ = 0.0744 m,
A = Cross Sectional Area, P = Perimeter and K = Thermal conductivity of air.
For air : $\mu = 1.98 \text{ X } 10^{-5}$, $C_p = 1 \text{ X } 10^3$, K = 0.024 in SI units.
Temperature at outlet is obtained using simulation in Ansys 13.0. For simulation k-ϵ turbulent model is performed in fin array arrangement which is based on conservation of mass, momentum and energy as described in equation 4(a), 4(b) and 4(c) as:-

Continuity equation:

$\frac{\partial \rho}{\partial t} + \nabla.(\rho\, U) = 0$ 4(a)

X- direction momentum equation:

$\rho \frac{Du}{Dt} = \frac{\partial \rho u}{\partial t} + \nabla(\rho \mathbf{U})$ 4 (b)

Energy equation:

$\rho \frac{DE}{Dt} = \frac{\partial \rho E}{\partial t} + \nabla(\rho E \mathbf{U})$ 4 (c)

For solving these equation SIMPLE solution algorithm with Tri Diagonal Matrix Algorithm solution procedure is applied for solving equation 4(a), 4(b) and 4(c).

a. GRID INDEPENDENCE TEST

Grid independence test is required to perform in order to make decision on optimum grid size so that convergense as well as possibility of error due to discretization of geometry can be minimised. To make results independent of size of grid is important because energy calculation must be independent of coordinate system as well as its resolution(for Eludiean space). Finer mesh quality can give the promising results but with the penalty of performance of the simulation. It becomes the point of concern for geometry with large geometry size in spatial space, very low residuals demand, large source term in Navier stokes equation etc. So grid independence test is required to perform, in order to ensure the independent results with high accuracy but at the same time not compromizing the performance of simulation as well as taking consideration of the System resource capability to perform optimum simulation.

To decide the optimum size of the grid for the calculation of the project, grid independence test is performed on simple geometry so that convergence can be obtained along with the steady reduction in residuals and reaching to asymptotic behavior at the end of iteration.

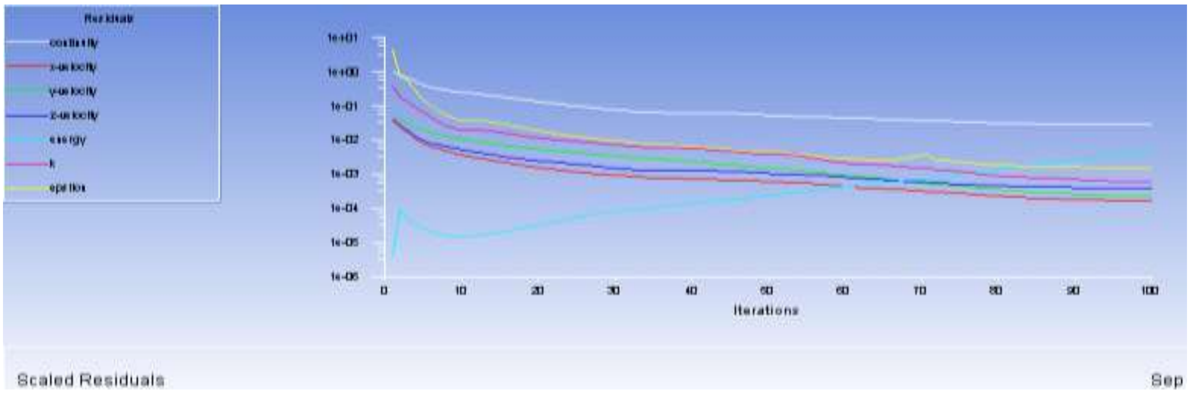

Fig. 3 Residuals for Grid Size 0.02 m

As shown in above figure 3 solution stops diverging from zig-zag manner but still some residual values are showing increasing behavior which leads to divergence again. As we keep on decreasing the size of mesh residual start dereasing and achiving the assymptotic behavior. The temperature at the outlet of the geometry also reaches almost same value at the end of the simulation. So finally, we acquired the the mesh size of 0.01m i.e. 10mm(face size as well as internal size)for all the simulation onward.

Graph is plotted(fig. 4) for temperature with grid size along with table 1 to show the behaviour of energy conservation at the outlet.

TABLE 1 Variation of Temperature with Grid Size

S.No.	Grid size(in m)	Temp. at Outlet(in K)
1.	0.100	300.9411
2.	0.080	190.9727
3.	0.060	300.0763
4.	0.040	152.5798
5.	0.020	301.1167
6.	0.010	301.2924
7.	0.009	301.2192

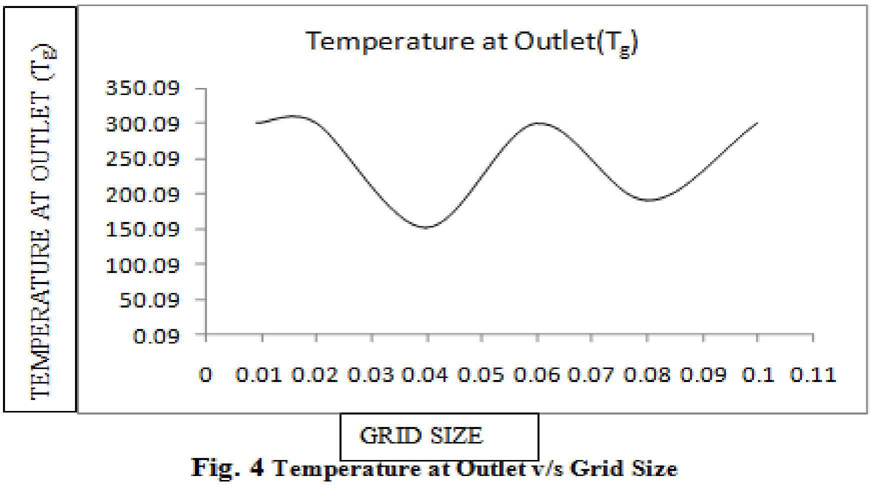

Fig. 4 Temperature at Outlet v/s Grid Size

IV. RESULTS AND DISCUSSIONS

Dittus-Boelter equation can be used in the cases where temperature difference is not too high. It provides an explicit form of equation which can be calculated easily using equation (5)

$$Nu = 0.023\ Re^{4/5} \times Pr^{0.4} \qquad (5)$$

Using equations as disscused above the values of Nu and h_{av} can be calculated for different values of temperature obtained by simulation and selected Re numbers. Results are shown in TABLE 1.

TABLE 1 Values of h_{av} and N_u numbers for different cases

Geometry Of Perforation	Temperature Diffrence (T_{out}-T_{in})	Reynold Number (Re)	Average Heat Transfer Coefficient (h_{av})	Nusselt Number (Nu)
Rectangular	6.05194	20000	18.93	55.448
	5.51926	40000	18.96	55.536
	5.33691	60000	18.9803	55.595
Elliptical	5.63324	20000	18.9601	55.536
	5.31223	40000	18.9820	55.600
	5.20923	60000	18.9890	55.621
Circular	5.61295	20000	18.9615	55.540
	5.30365	40000	18.9825	55.602
	5.20309	60000	18.9894	55.622

Using these values of Nusselt numbers and Reynolds numbers curve fiitng is obtained for rectangular case and same is formulated with R^2 value of 95.6% and represented in equation (6) as:-

$$Nu = 54.799 \times Re^{0.002} \qquad (6)$$

Graph below (Fig. 3) shows the relation between Nusselt number and Reynolds number for different cases.

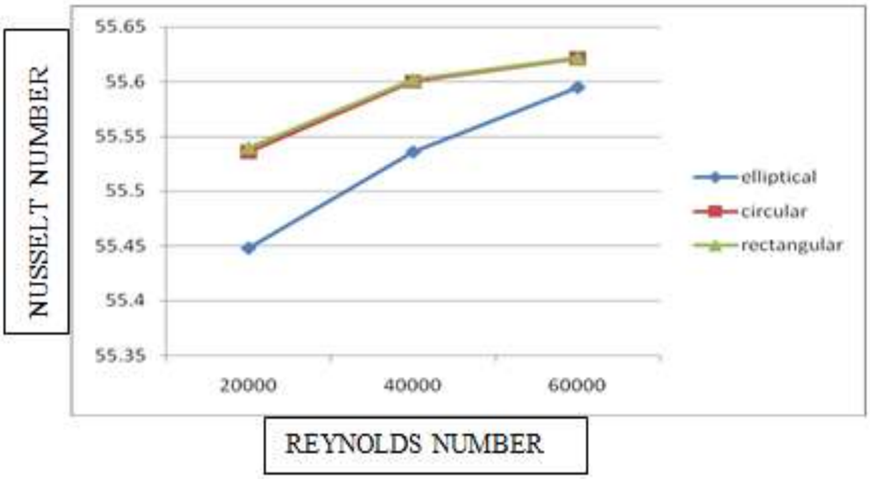

Fig. 5: Graph Showing Relation between Nusselt number and Reynolds number

Results obtained by equation (6) compared are with formulla given in equation (5) and results are shown in TABLE (2).

TABLE 3: Compared values of Nusselt number by Dittus-Boelter equation and simulation results for Rectangular perforations

Reynold Number (Re)	N_u(Dittus-Boelter)	N_u(Simulation)
20000	57.94864	54.80
40000	100.8944	54.87
60000	139.5533	54.92

From these values it can be shown that at lower Reynolds number (Re of around 20000) Dittus-Boelter eqution matches well with simulated results. Streamlines in rectangular perforation is shown in Fig 3. From Fig. 3 it is clear that due to the perforations a part of air moves the space between fins which causes more heat transfer through the base plate.

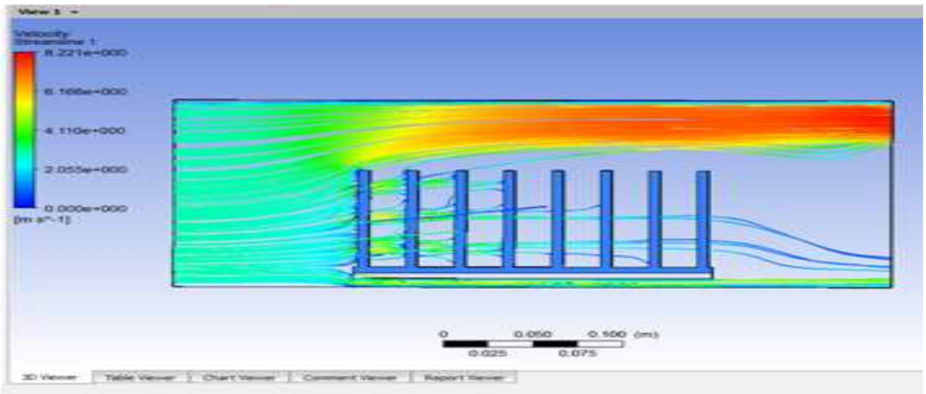

Fig. 6: Streamlines for rectangular cross section (Re=20000)

Temperature distribution at the outlet for Re=20000 is shown in Fig. 4. From these figures it can be deduced that rectangular shape of perforation is efficient as temperature at the center is high in case of rectangular shape which is a indication of more heat loss by fin area by the convection to the air.

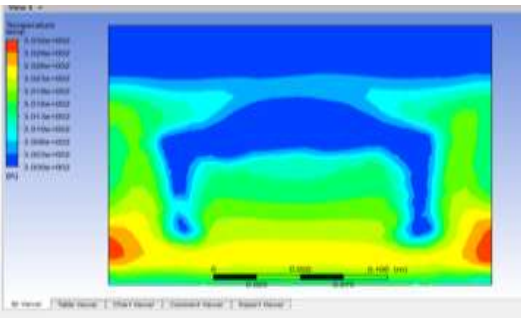

Fig. 7(a): Temperature distribution at outlet for Rectangular shape

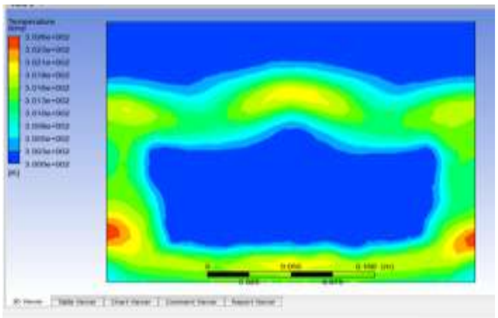

Fig. 7(b):Temperature distribution at outlet for Elliptical shape

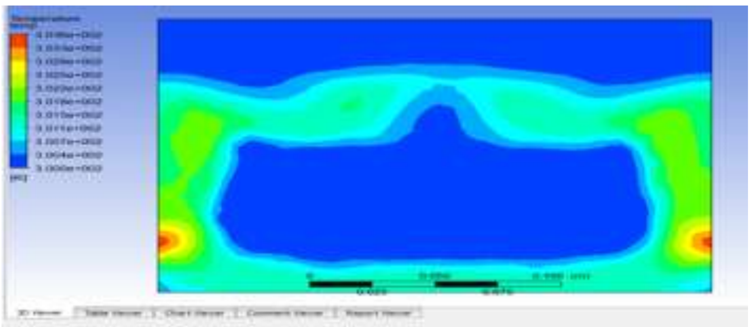

Figure 7(c): Temperature distribution at outlet for Circular shape

Temperature distribution along the base plate(at mid line along the flow direction) is shown in Fig 8(a) and Fig 8(b) for circular and rectangular cross section respectively.

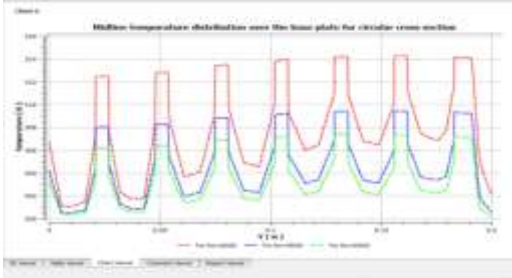

Figure 8(a): Temperature Distribution over Base Plate for Circular Cross Section

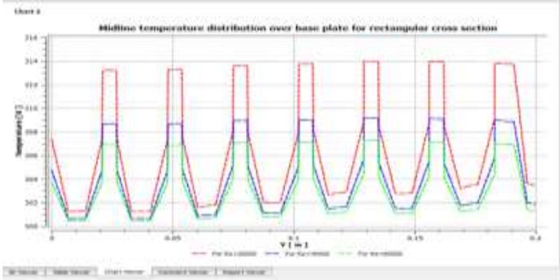

Figure 8(b): Temperature Distribution over Base Plate for Rectangular Cross Section

Using these Fig. 6(a) and 6(b) temperature distribution of base plate at fin bases position (along the flow direction) can be obtained. Equation (7) represents the relation between midline line temperature at fin bases position and temperature for rectangular cross section at Re= 20000.

$$T = 0.3249\ y + 301.2 \tag{7}$$

Where T is temperature in kelvin and y is distance in meter along flow direction.

V. CONCLUSION

Some useful results from these discussion can be deduced as follows-

1. Dittus-Boelter equation can be used at high Reynolds number but it does not give good agreement for very high Reynold number.
2. Heat transfer rate is high for Rectangular shape of perforation decreasing with elliptical and circular shape. It shows that shapes with lower ecentricity have good heat removal capacity. So ecentricity as low as possible should be manufactured when perforated fin are manufactured.
3. Heat transfer from first some fins are high as compared to later fins. So incresing number of fins in lateral direction has no much effect on heat transfer rate specially when space and weight are constraints.
4. At very high Reynold number heat transfer rate becomes almost constant irrespective of shape of perforation and flow rate of air.

ACKNOWLEDGEMENT

Authors are thankfull to the Jodhpur Institute of Engineering and Technology for providing lab facility and software access for the project.

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