International Journal of Enhanced Research in Science Technology & Engineering, ISSN: 2319-7463
Vol. 3 Issue 7, July-2014, pp: (267-276), Impact Factor: 1.252, Available online at: www.erpublications.com

CFD analysis of laminar heat transfer in a channel provided with baffles: comparative study between two models of baffles: diamond-shaped baffles of different angle and rectangle

Amit Garg[1], Sunil Dhingra[2], Gurjeet Singh[3]
[1]M.Tech. Scholar, Department of Mechanical Engineering, U.I.E.T, Kurukshetra, India
[2]Asstt. Prof., Department of Mechanical Engineering, U.I.E.T, Kurukshetra, India
[3]Asstt. Prof. Department of Mechanical Engineering PEC University, Chandigarh, India

---

Abstract: In this research work heat transfer and fluid flow characteristics in a channel in the presence of a diamond-shaped baffles has been numerically investigated in the laminar flow regime. The computations are based on the finite volume method, the Navier Stokes equations along with the energy equation have been solved by using SIMPLE Technique. The unstructured triangular mesh is used for the computational domain. The fluid flow and heat transfer characteristics are presented for Reynolds numbers based on the hydraulic diameter of the channel ranging from 100 to 600. Effects of different baffle tip angles on heat transfer and pressure loss in the channel are studied and the results of the diamond baffle are also compared with those of the flat baffle. The velocity profiles were obtained for all the geometry considered and selected for different sections, namely, downstream and between the two baffles and the friction coefficients were obtained for different sections and for different Reynolds numbers. It is observed that apart from the rise of Reynolds number, the reduction of the baffle angle leads to an increase in the Nusselt number and friction factor.

Keywords: Periodic channel flow, Baffle, Laminar flow, Heat transfer, Diamond baffle, Fluent, Finite volume method.

---

1. Introduction

One of the most effective methods of enhancing the heat transfer rate in a smooth channel is the use of baffles placing on the channel walls with in line or staggered arrays. This is because the baffles help to interrupt the hydrodynamic and thermal boundary layers and to induce a recirculation zone or vortices flow behind the baffle. The flow reattachment downstream of the baffle gives rise to the washing up or down the channel walls leading to the increase in the heat transfer rate. This method of heat transfer enhancement is commonly used in engineering applications such as compact heat exchangers, air cooled solar collectors and electronic packages, so the literature on this topic is widely apparent. Although heat transfer is increased through the baffle arrangement, the pressure drop of the channel flow is also increased due to the decreased flow area effects. Therefore, baffle spacing and height are among the most important parameters used in the design of channel heat exchangers. Closer spacing or larger baffle height causes higher heat transfer rate but resulting in poor stream distribution and higher pressure drop. On the other hand, higher baffle spacing or smaller baffle height causes the reduction of the pressure drop but provides more longitudinal flow leading to the decrease of heat transfer. It is, thus, difficult to realize the advantage of baffle arrangements or geometry. The effects of baffle spacing on heat transfer and pressure drop have been studied by various researchers. However, there is no precise criterion to determine optimum baffle shapes in the literature. The use of baffle height and pitch spacing of 0.5 and 2 times the channel height respectively is introduced in most of previous work.

Nomenclature

A	convection heat transfer area of channel, m^2	U	mean velocity, m/s
Cp	specific heat capacity of air, J/kg K	V	volumetric flow rate, m^3/s
D	hydraulic diameter, m(2H)		
f	friction factor	Greek letters	
H	channel height, m	θ	half angle of baffle tip, degree
h	average heat transfer coefficient, W/m^2 K	ρ	density of air, kg/m^3
k	thermal conductivity of air, W/m K	Γ	thermal diffusion
L	length of tested channel, m		
Nu	Nusselt number (hD/k)	Subscripts	
P	pitch (axial length of rib cycle), m	o	smooth channel
ΔP	pressure drop, Pa	ave	average
Pr	Prandtl number	in	inlet
Re	Reynolds number (UD/ν)	w	wall
Q	heat transfer, W		
T_s	Average temperature of heated wall, K		
T_o	Average temperature of outlet		
T_i	Inlet temperature		

The first work on the numerical investigation of flow and heat transfer characteristics in a duct with the concept of periodically fully developed flow was conducted by Patankar et al. [1]. Berner et al. [2] suggested that a laminar behavior for a channel with baffles is found at a Reynolds number below 600 and for such conditions the flow is free of vortex shedding. The numerical investigation of fluid flow and heat transfer characteristics in a smooth channel with staggered baffles, based on the periodically fully developed flow conditions of Patankar et al. [1], was reported by Webb and Ramadhyani [3]. Kellar and Patankar [4] computed the heat transfer in channels with staggered baffles and found that the heat transfer increases with the rise in baffle height and with the decrease in baffle spacing. Their results showed the same behavior as Webb and Ramadhyani [3] results. Cheng and Huang [5] investigated the case of asymmetrical baffles and indicated that the friction factor shows a great dependence on baffle location, especially for a large height of baffle. Cheng and Huang [6] again presented laminar forced convection in the entrance region of a horizontal channel with one or two pairs of baffles placed on the walls.

Habib et al. [7] reported the characteristics of turbulent flow and heat transfer inside the periodic cell formed between segmented baffles staggered in a rectangular duct and pointed out that the pressure drop increases with the baffle height. Amiri et al. [8] investigated both experimentally and numerically the laminar flow and heat transfer in two-dimensional channels with packed bed porous media using a two-phase equation model for the transport. A numerical investigation of laminar forced convection in a three-dimensional channel with baffles for periodically fully developed flow and with a uniform heat flux in the top and bottom walls was conducted by Lopez et al. [9].

Guo and Anand [10] studied the three dimensional heat transfers in a channel with a single baffle in the entrance region. Numerical studies for both solid and porous baffles in a two dimensional channel for the turbulent flow [11] and for the laminar flow regimes [12,13] were conducted and similar thermal performance results for both the solid and porous cases were reported. Ko and Anand [14] carried out an experiment for turbulent channel flow with porous baffles and found that the porous baffles present a flow behavior as good as the one with solid baffles. Mousavi and Hooman [15] numerically studied the heat transfer behavior in the entrance region of a channel with staggered baffles for Reynolds numbers ranging from50 to 500 and baffle heights between 0 and 0.75 and reported that the Prandtl number affects the precise location of the periodically fully developed region. Tsay et al. [16] investigated numerically by using baffles for enhancement of heat transfer in laminar channel flow over two heated blocks mounted on the lower plate.

Most of the investigations on laminar flow, cited above, have considered only the heat transfer characteristics for blockage ratio and spacing ratio values for porous or solid flat baffles. In the present work, the numerical computations for two dimensional laminar periodic channel flows over a pair of staggered diamond baffles mounted on the channel walls are performed with the main aim being to study the changes in the flow pattern and heat transfer performance.

2. Flow configuration and mathematical foundation

2.1 Physical Model.

The system of interest is a horizontal plane channel with a baffle pair placed in a staggered array on the upper and lower channel walls as shown in Fig 2.1. The flow under consideration is expected to attain a periodic flow condition in which the velocity field repeats itself from one cell to another. The concept of periodically fully developed flow and its solution procedure has been described in Ref. [1]. The fluid enters the channel at an inlet temperature, Tin, and flows over a staggered baffle pair where b is the baffle height, H set to 0.02 m, is the channel height and b/H is known as the blockage ratio, BR. The distance between the baffles is set to s in which s/H is defined as the spacing ratio, S. To investigate a geometry effect of the interaction between both baffles, the baffle half tip angle, θ is varied as of θ=7.5° and 17.5° in the present investigation. Also, a typical flat baffle pair with thickness of 0.02H is introduced for comparison.

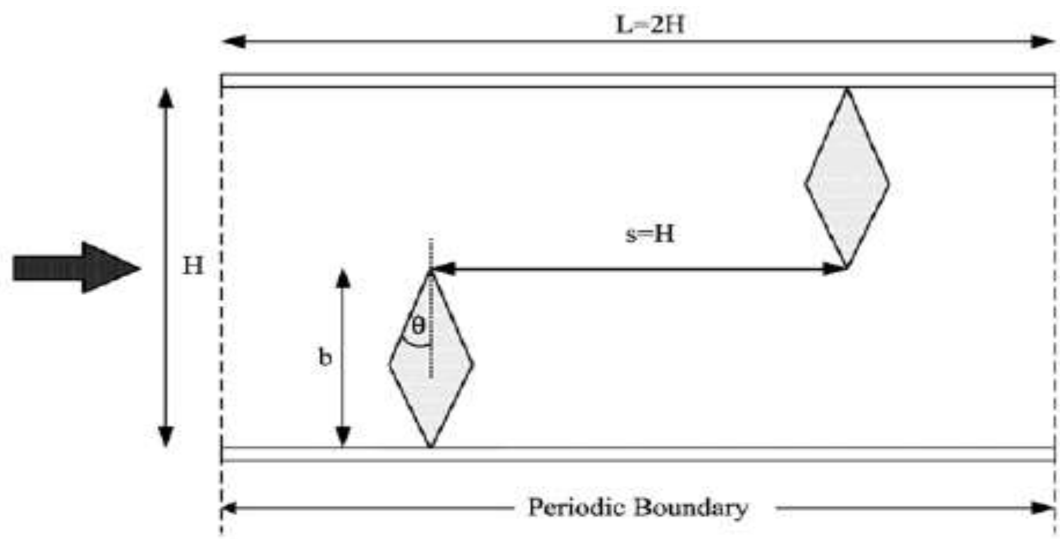


Fig 2.1 Channel geometry and computational domain of periodic flow.

2.2 Numerical Method.

The numerical simulations were carried out using ANSYS-14.0 CFD Software package Fluent-6 version that uses the finite-volume method to solve the governing equations. Geometry was created for air flowing in an electrically heated copper channel. Meshing has been created in ANSYS model with quard/tri shapes. In this study Reynolds number varies between 100 - 600.

The numerical model for fluid flow and heat transfer in the channel was developed under the following assumptions:
• Steady two-dimensional fluid flow and heat transfer.
• The flow is laminar and incompressible.
• Constant fluid properties.
• Body forces and viscous dissipation are ignored.
• Negligible radiation heat transfer.

Based on the above assumptions, the channel flow is governed by the continuity, the Navier- Stokes equations and the energy equation. In the Cartesian tensor system these equations can be written as follows:

Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0.$$

Momentum equation:

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right].$$

Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_j}\left(\Gamma\frac{\partial T}{\partial x_j}\right)$$

Where Γ is the thermal diffusivity and is given by

$$\Gamma = \frac{\mu}{\mathrm{Pr}}.$$

The governing equations were discretized by the second order upwind scheme, decoupling with the SIMPLE algorithm and solved using a finite volume approach [17]. The solutions were considered to be converged when the normalized residual values were less than 10^{-6} for energy and 10^{-3} momentum variables.

Table 1.1: Properties of air at 25^{0}C

Properties	Value
Density, ρ	1.225 kg/m^3
Specific heat capacity C_p	1006 J/kg K
Thermal conductivity, k	0.0242 W/m K
Viscosity, μ	1.7894 x10^{-5} kg/m s

2.3 Data Reduction

Five parameters of interest in the present work are the Reynolds number, friction coefficient, friction factor, Nusselt number and thermal enhancement factor. The Reynolds number is defined as

$$\mathrm{Re} = \rho\bar{u}D_h/\mu.$$

The skin friction coefficient, Cf is given by

$$C_f = \frac{\tau_w}{\frac{1}{2}\rho\bar{u}^2}.$$

The friction factor, f is computed by pressure drop, Δp across the length of the periodic channel, L as

$$f = \frac{(\Delta p/L)D_h}{\frac{1}{2}\rho\bar{u}^2}.$$

The heat transfer is measured by average Nusselt number which can be written as

$$Nu = \frac{h_x D_h}{k}.$$

The thermal enhancement factor (η) is given by

$$\eta = (Nu/Nu_0)/(f/f_0)^{1/3}$$

where Nu_0 and f_0 stand for Nusselt number and friction factor for the smooth channel, respectively.

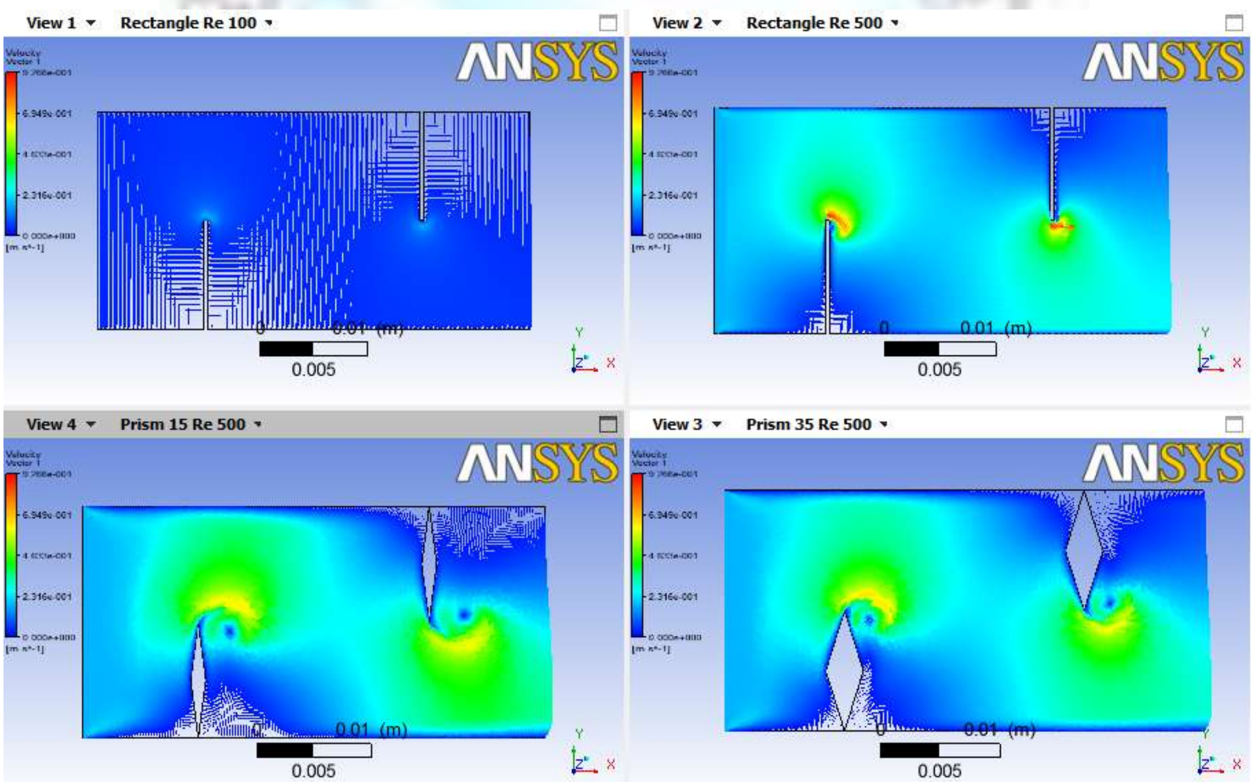
International Journal of Enhanced Research in Science Technology & Engineering, ISSN: 2319-7463
Vol. 3 Issue 7, July-2014, pp: (267-276), Impact Factor: 1.252, Available online at: www.erpublications.com

2.4. Boundary conditions

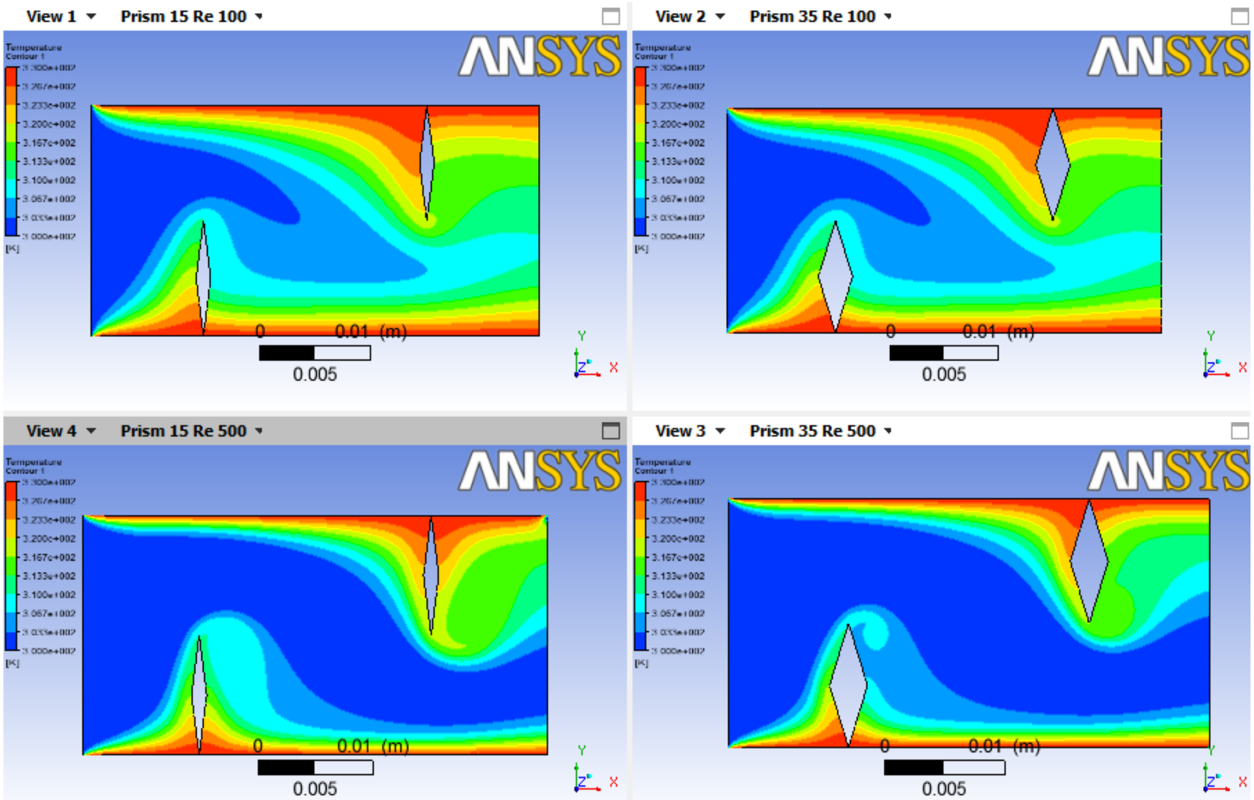
Periodic boundaries are used for the inlet and outlet of the flow domain. Constant mass flow rate of air with 300 K (Pr=0.7) is assumed in the flow direction rather than constant pressure drop due to periodic flow conditions. The inlet and outlet profiles for the velocities must be identical. The physical properties of the air have been assumed to remain constant at average bulk temperature. Impermeable boundary and no-slip wall conditions have been implemented over the channel wall as well as the baffle. The constant temperature of the bottom and upper plates is maintained at 330 K while the baffle is assumed at adiabatic wall conditions. It should be noted that the dimensionless temperature must be identical between the inlet and outlet.

3. RESULTS AND DISCUSSION

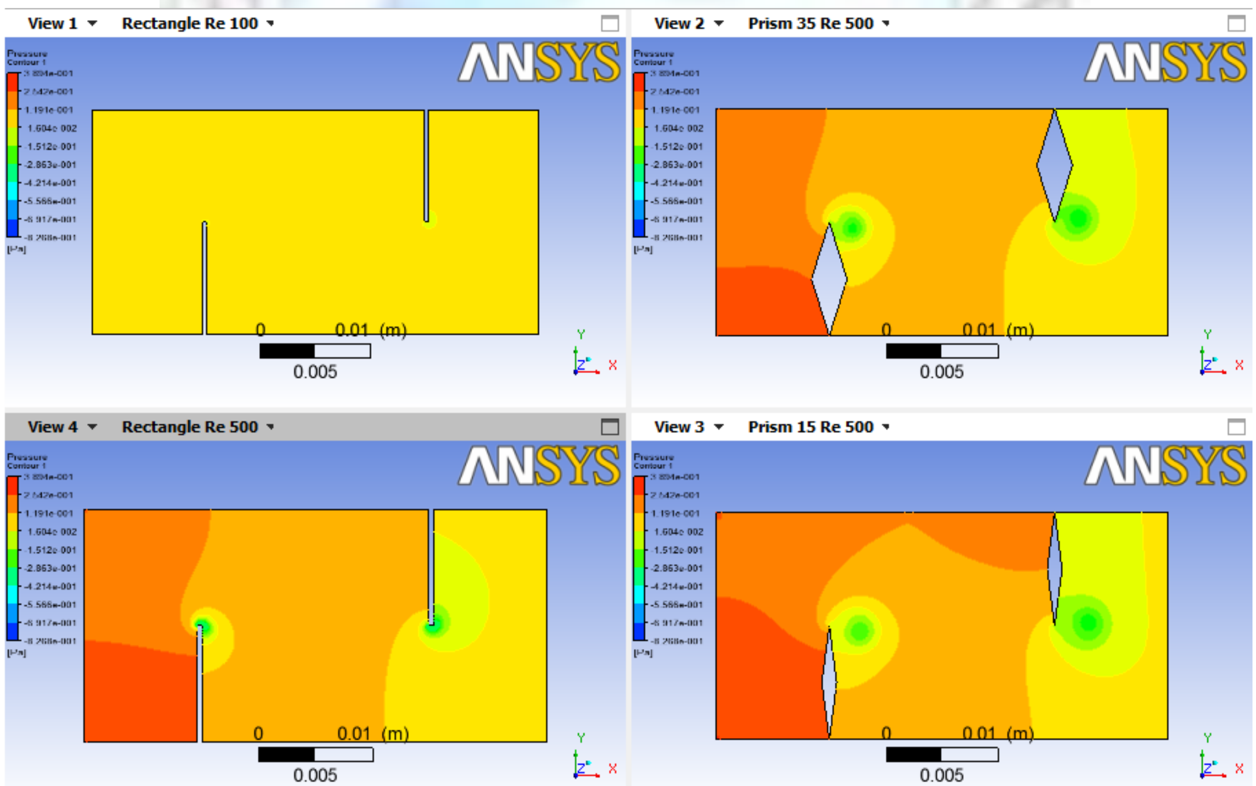
The flow structure in presence of diamond shaped baffle can be discerned by looking at velocity vector plots. The velocity vector plots for both the orientations ($2\theta=15^0,35^0$) and rectangle baffle for Reynolds number 500 are shown below. The flow stream attach the baffle tip cause the circulation zone, the strength of which strong is in case of rectangle baffle which cause the large pressure difference as compared to diamond shaped baffles. The flow passage decreases as the flow moves towards the baffle and the flow passage increases as the flow moves away from the baffle. The figures 3.1, 3.2 and 3.3 show the velocity vector, temperature contour and pressure contour of the computation domain of the plane channel for both the orientations of diamond shaped and of rectangle which shows the difference between different orientations. The figures below show the temperature contours of the computation domain of the plane channel for both the orientations of baffles. The presence of the obstacle causes the formation of counter rotating vortices which cause the mixing of fluid and hence and increase in the heat transfer coefficient of the fluid and hence the temperature of the fluid increases.



Fig; 3.1 Velocity vector for Prism baffle ($2\theta=15^0,35^0$) and rectangle baffle



Fig; 3.2 Temperature contour for Prism baffle (2θ=15⁰,35⁰) at Reynolds no. 100 &500



Fig; 3.3 Pressure contour for Prism baffle (2θ=15⁰,35⁰) and rectangle baffle

International Journal of Enhanced Research in Science Technology & Engineering, ISSN: 2319-7463
Vol. 3 Issue 7, July-2014, pp: (267-276), Impact Factor: 1.252, Available online at: www.erpublications.com

Figure 3.4 shows the axial velocity profiles at x = 0.02m (Series1) and x = 0.035m (Series2) respectively for the two cases treated (A) rectangular baffles, (B) diamond shaped baffles at Re = 100 and 500. It is indicates clearly that the values of velocity are very low in the vicinity of the two baffles especially in the areas located downstream. This is due to the presence of the zones of recirculation. One notices also the increase velocity in space between the top of each baffle and the walls of the channel. This increase is generated first of all by the singularity represented by the obstacles, also by the presence of a recycling which then results an abrupt change from the direction of the flow. It is also noticed that the most values velocity appear close the top of the channel with a process of acceleration which starts just after the second baffle. The variation of velocity for the two cases appears clearly on contours and their scales which present positive and negative values.

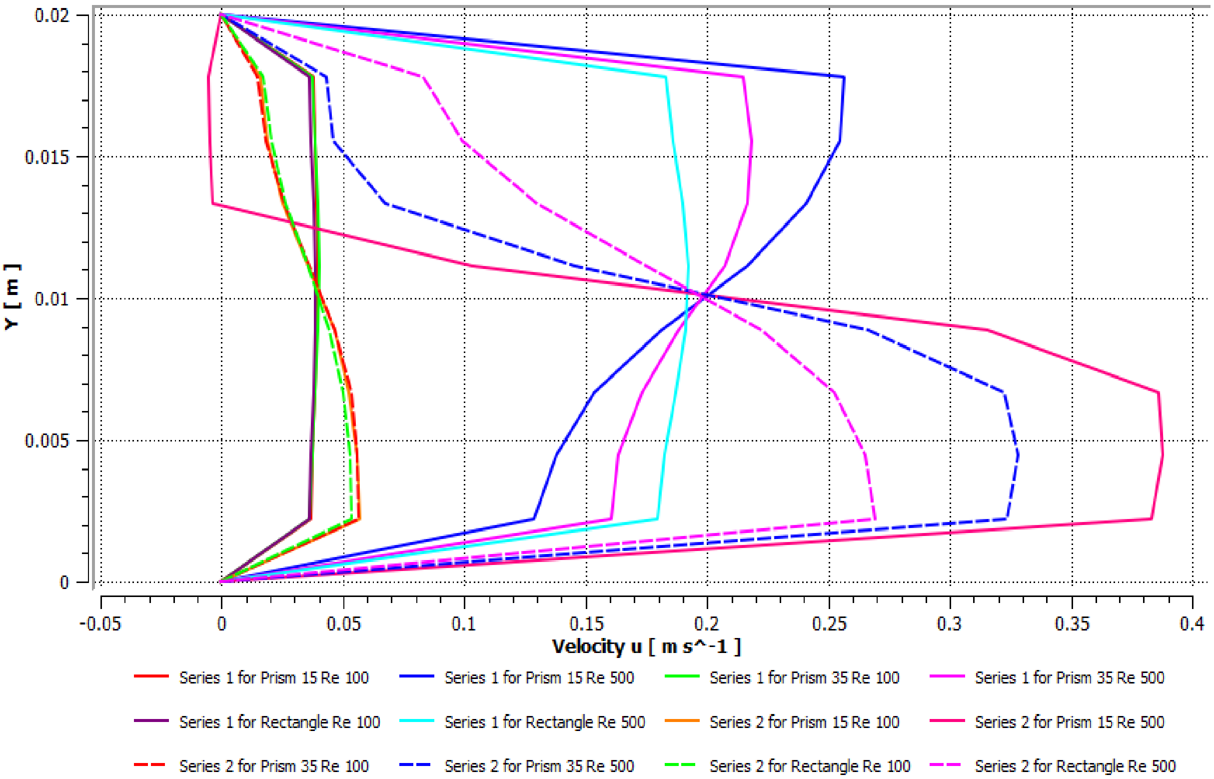


Fig. 3.4: Axial velocity profiles at x=0.02(Series=1) and x=0.035(Series=2) for re= 100 & 500

Fig 3.5 shows the variation of surface Nusselts number with Reynolds number for diamond shaped baffles and rectangle baffle for comparison. As we increase the inlet flow velocity there is increase in the surface Nusselts number and at all Reynolds number the values is less for diamond shaped baffles than that to rectangle baffle except at Reynolds number 100.

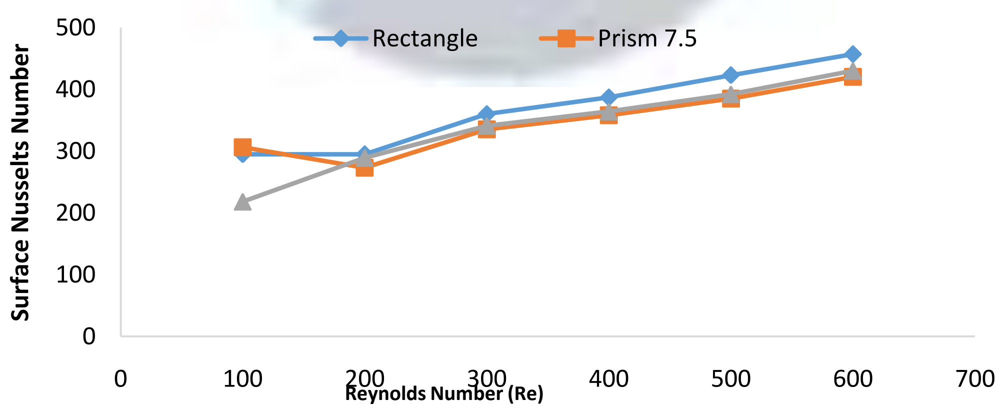


Fig 3.5: Surface Nusselts number Vs Reynolds number

International Journal of Enhanced Research in Science Technology & Engineering, ISSN: 2319-7463
Vol. 3 Issue 7, July-2014, pp: (267-276), Impact Factor: 1.252, Available online at: www.erpublications.com

In general, the augmentation in heat transfer is concerned with penalty in terms of increased skin friction coefficient leading to higher pressure drop. Fig 3.6 shows the skin friction coefficient, C_f along the lower channel wall for different Reynolds number values in the cases of different baffles. In the figure, the increase of skin friction coefficients is found to be larger than that of the heat transfer coefficients caused by the temperature field. This may imply that the flow field develops more rapidly than the temperature field.

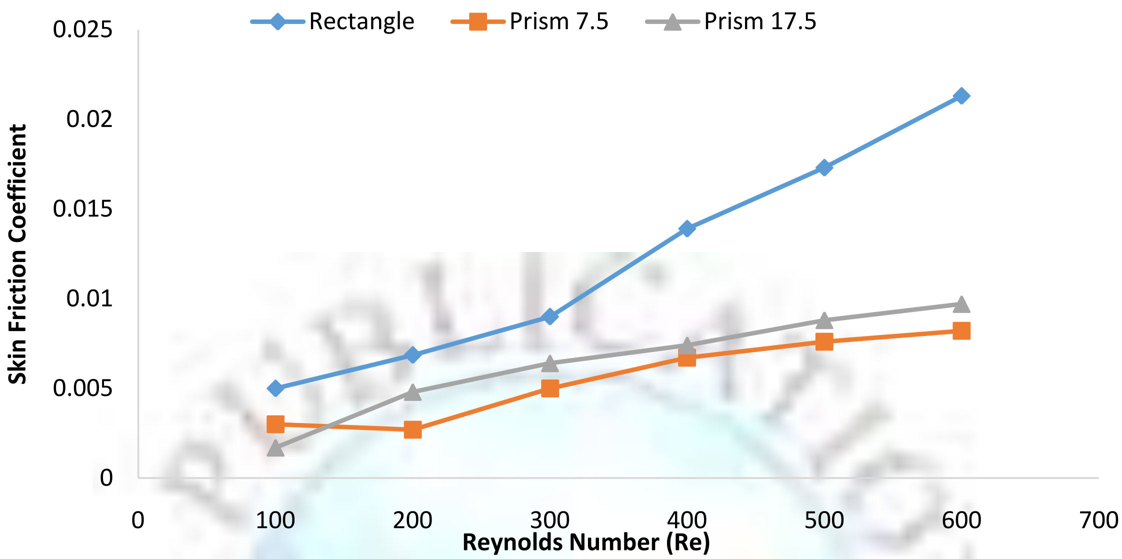


Fig 3.6: Skin friction coefficient Vs Reynolds number

Effect of the baffles geometry on the heat transfer rate is presented in the form of average Nusselt number and friction factor with Reynolds number as depicted in Figure-3.7 & 3.8. In the figure, the baffles provide considerable heat transfer enhancements in comparison with the smooth channel and the average Nusselt number values for using baffles increase with the rise of Reynolds number. This is because the baffles interrupt the development of the boundary layer of the fluid flow and create the reverse/recirculating flow on the tip of baffles. At low Reynolds number below 500 the average Nusselts number for diamond shaped baffles is low as compared to rectangle but at Reynolds number 600 the values for diamond shaped baffles is high than that of rectangle baffles.

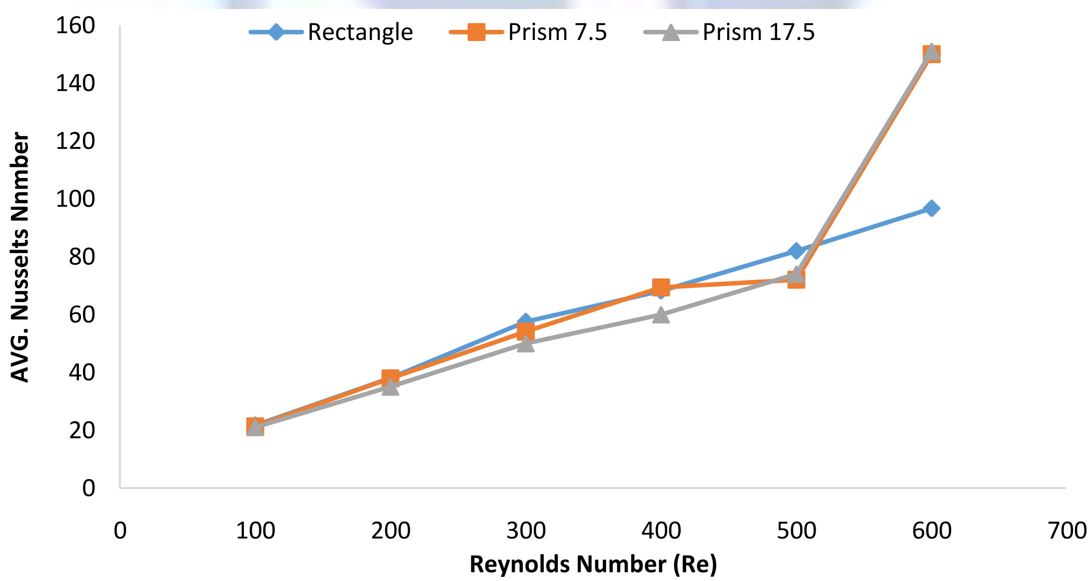


Fig 3.7: Average Nusselts number Vs Reynolds number

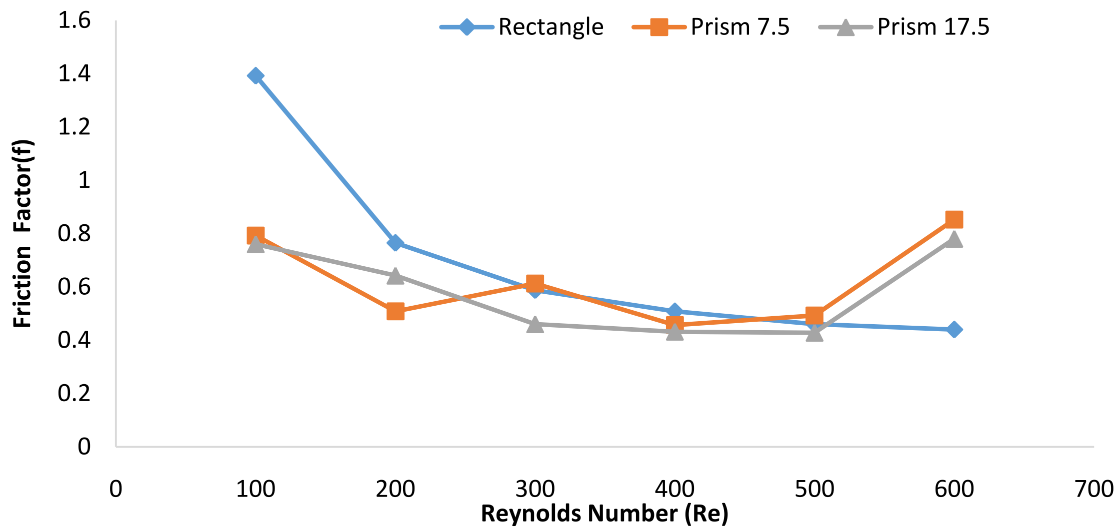


Fig 3.8: Friction factor Vs Reynolds number

4. CONCLUSION

In the present problem the numerical simulation of laminar flow in a parallel plate channel with a built-in diamond shaped baffle is performed. The flow structure and heat transfer characteristics are studied in detail. On the basis of the results obtained the following conclusions are made:

- Due to insertion of diamond shaped baffle a re-circulation zone is built to tip of baffle with wash up the fluid particles which remain in contact with baffles.
- The presence of diamond shaped baffle significantly improves the heat transfer performance. The percentage increase in average Nusselt number with the use of diamond shaped baffle at low Reynolds number 100 is 2-5% but at high Reynolds number (600) the percentage increase is 20-30.
- Heat transfer increases as the Reynolds number is increased. The percentage increase in average Nusselt number at Reynolds number of 200 is 60-70 % more as compared to average Nusselt number at Reynolds number of 100 but as the Reynolds number is increased to 600, the percentage incrcredement is up to 90%.
- The friction factor is lowest for the diamond shaped baffle ($2\theta = 35^0$). The percentage decrease in friction factor at Reynolds number 300 is 20% and 10% for diamond shaped baffle ($2\theta = 35^0$) and ($2\theta = 15^0$) respectively as compared to rectangle baffle.
- The friction factor is decreased as we increased the Reynolds number. But at Reynolds number above 500, the friction factor is increased considerably for the case of diamond shaped baffle.

5. SCOPE FOR FUTURE WORK

The results of this work reveal that the diamond shaped baffle as a vortex generator is a useful device for improving heat transfer in a parallel plate channel. Here the computations have been done assuming flow regime to be laminar. The present problem can be extended in future in the following ways:

1. The present study can be extended for the turbulent flow. Using an appropriate turbulent model, the performance of the purposed design can be computed for higher Reynolds number.
2. The present problem can be extended for more than two baffles. The simulation can be performed for various angles of diamond shaped baffles.
3. The present study is extended to different blockage ratio.
4. The study can be performed for other different geometries and can be compared for the best geometry.

ACKNOWLEDGEMENT

The authors gratefully acknowledge the financial support by the Kurukshetra University Research Found.

REFERENCES

[1]. S.V. Patankar, C.H. Liu, E.M. Sparrow, Fully developed flow and heat transfer in ducts having streamwise-periodic variations of cross-sectional area, ASME J. Heat Transfer 99 (1977) 180–186.

[2]. C. Berner, F. Durst, D.M. McEligot, Flow around baffles, Trans. ASME J. Heat Transfer 106 (1984) 743–749.

[3]. B.W. Webb, S. Ramadhyani, Conjugate heat transfer in a channel with staggered ribs, Int. J. Heat Mass Transfer 28 (1985) 1679–1687.

[4]. K.M. Kelkar, S.V. Patankar, Numerical prediction of flow and heat transfer in a parallel plate channel with staggered fins, ASME J. Heat Transfer 109 (1987) 25–30.

[5]. C.H. Cheng, W.H. Huang, Laminar forced convection flows in horizontal channels with transverse fins placed in entrance regions, Int. J. Heat Mass Transfer 20 (1991) 1315–1324.

[6]. C.H. Cheng, W.H. Huang, Numerical prediction for laminar forced convection in parallel-plate channels with transverse fin arrays, Int. J. Heat Mass Transfer 34 (11) (1991) 2739–2749.

[7]. M.A. Habib, A.M. Mobarak, M.A. Sallak, E.A. Abdel Hadi, R.I. Affify, Experimental investigation of heat transfer and flow over baffles of different heights, Trans. ASME J. Heat Transfer 116 (1994) 363–368.

[8]. A. Amiri, K. Vafai, T.M. Kuzay, Effects of boundary conditions on non-Darcian heat transfer through porous media and experimental comparison, Numer. Heat Transf. Appl. 27 (1995) 651–664.

[9]. J.R. Lopez, N.K. Anand, L.S. Fletcher, Heat transfer in a three-dimensional channel with baffles, Numer. Heat Transf., A Appl. 30 (1996) 189–205.

[10]. Z. Guo, N.K. Anand, Three-dimensional heat transfer in a channel with a baffle in the entrance region, Numer. Heat Transf., A Appl. 31 (1) (1997) 21–35.

[11]. Y.T. Yang, C.Z. Hwang, Calculation of turbulent flow and heat transfer in a porousbaffled channel, Int. J. Heat Mass Transfer 46 (2003) 771–780.

[12]. B.M. Da Silva Miranda, N.K. Anand, Convective heat transfer in a channel with porous baffles, Numer. Heat Transf., A Appl. 46 (5) (2004) 425–452.

[13]. N.B. Santos, M.J.S. De Lemos, Flow and heat transfer in a parallel-plate channel with porous and solid baffles, Numer. Heat Transf., A Appl. 49 (5) (2006) 471–494.

[14]. K.H. Ko, N.K. Anand, Use of porous baffles to enhance heat transfer in a rectangular channel, Int. J. Heat Mass Transfer 46 (2003) 4191–4199.

[15]. S.S. Mousavi, K. Hooman, Heat and fluid flow in entrance region of a channel with staggered baffles, Energy Convers. Manag. 47 (2006) 2011–2019.

[16]. Y.L. Tsay, J.C. Cheng, T.S. Chang, Enhancement of heat transfer from surfacemounted block heat sources in a duct with baffles, Numer. Heat Transf., A Appl. 43 (8) (2003) 827–841.

[17]. S.V. Patankar, Numerical Heat Transfer and Fluid Flow, McGraw-Hill, New York, 1980.

[18]. F. Incropera, P.D. Dewitt, Introduction to heat transfer, 3rd editionJohn Wiley & Sons Inc, 1996.