International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:9, No:6, 2015

A Computational Study of Very High Turbulent Flow and Heat Transfer Characteristics in Circular Duct with Hemispherical Inline Baffles

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Abstract**—**This paper presents a computational study of steady state three dimensional very high turbulent flow and heat transfer characteristics in a constant temperature-surfaced circular duct fitted with 90^0 hemispherical inline baffles. The computations are based on realizable k-ε model with standard wall function considering the finite volume method, and the SIMPLE algorithm has been implemented. Computational Study are carried out for Reynolds number, Re ranging from 80000 to 120000, Prandtl Number, Pr of 0.73, Pitch Ratios, PR of 1,2,3,4,5 based on the hydraulic diameter of the channel, hydrodynamic entry length, thermal entry length and the test section. Ansys Fluent 15.0 software has been used to solve the flow field. Study reveals that circular pipe having baffles has a higher Nusselt number and friction factor compared to the smooth circular pipe without baffles. Maximum Nusselt number and friction factor are obtained for the PR=5 and PR=1 respectively. Nusselt number increases while pitch ratio increases in the range of study; however, friction factor also decreases up to PR 3 and after which it becomes almost constant up to PR 5. Thermal enhancement factor increases with increasing pitch ratio but with slightly decreasing Reynolds number in the range of study and becomes almost constant at higher Reynolds number. The computational results reveal that optimum thermal enhancement factor of 90^0 inline hemispherical baffle is about 1.23 for pitch ratio 5 at Reynolds number 120000.It also shows that the optimum pitch ratio for which the baffles can be installed in such very high turbulent flows should be 5. Results show that pitch ratio and Reynolds number play an important role on both fluid flow and heat transfer characteristics.

Keywords**—**Friction factor, heat transfer, turbulent flow, circular duct, baffle, pitch ratio.

I. Introduction

HEAT transfer rate from any surface can be enhanced by enhancing convection heat transfer coefficient or increasing heat transfer surface area. To increase the heat transfer coefficient, a pump or blower or fan is used, or existing one is replaced with a larger one. And heat transfer surface area may be increased using baffles or fins on the surfaces of the ducts [1]. Turbulent flows in circular pipes are encountered in many engineering applications such as nuclear reactors, shell and tube heat exchangers, cooling of gas turbines and combustion chambers, and cooling of electronic devices.

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Turbulent flow and heat transfer characteristics in circular pipes have been studied by a number of innovators. Al-Arabi [2] studied the effect of entrance condition on turbulence forced convection heat transfer in a tube for constant wall temperature and heat flux conditions. El-Sayed et al. [3] experimentally investigated to determine the detailed module-by-module pressure drop characteristics of turbulent flow inside circular tubes with longitudinal internal fins. Akansu [4] has numerically investigated the heat transfer and pressure drop for porous rings inserted in a pipe for turbulent flow. Tijing et al. [5] investigated the effect of straight and twisted internal fins on heat transfer enhancement. Nieckele and Saboya [6] performed an experimental investigation to determine average friction factor and heat transfer coefficient for turbulent flow in pinned annular region of a double-pipe heat exchanger. Yucel and Dinler [7] numerically investigated two-dimensional turbulent flow through a pipe with fins attached circumferentially. Dinler and Yucel [8] studied numerically two-dimensional turbulent flow and heat transfer in a pipe with a fin attached to the inner wall. Abraham et al. [9] numerically investigated heat transfer in pipe for laminar, transitional, and turbulent regimes Raj and Ganne [10] numerically studied the effect of shell-side baffle inclination angle on turbulent flow and heat transfer. Selvanaj et al. [11] carried out a numerical study at constant wall heat flux condition to study turbulent flow and heat transfer in a circular pipe for three types of grooved tubes (trapezoidal, square, and circular). Higher hydraulic performance was obtained for trapezoidal grooved tube. Turgut and Kizilirmak [12] numerically and experimentally investigated the effects of Reynolds number, baffle angle and baffle distance on three dimensional turbulent flow and heat transfer in a circular pipe for 3000<Re<5000. P. Promovonge and S. Sripattanipipat [13] had numerically investigated laminar flow heat transfer in square duct with V-inline baffles. Promovonge et al. [14] recently studied the thermal characteristics in a circular tube with fitted horseshoe baffles, Reynolds number ranging from 5300<Re<24000.

The literature survey reveals that there is a lack of information on steady state three-dimensional forced convection very high turbulent heat transfer and pressure drop for circular pipes with baffle, Reynolds number ranging from 80000-120000 which is very frequent in nuclear reactors and high performance heat exchangers. In this study, the characteristics of a steady state three-dimensional turbulent flow and heat transfer in a circular pipe with two baffles are investigated numerically using commercial flow solver Ansys Fluent 15.0 under constant wall temperature boundary condition. Realizable *k*-ε turbulence model with standard wall function is used to simulate the flow field. The effects of Reynolds number and higher pitch ratio (PR) on fluid flow and heat transfer characteristics are examined numerically. Numerical study has been carried out for PR ranging from 1-5 and baffle orientation angle at 90^0. Air at 300K with Prandtl number 0.73 is taken as the working fluid.

II. Baffle Geometry and Arrangement

The system of interest is a circular duct of diameter, d=100mm with two hemispherical baffles inserted into it in staggered configuration. It is assumed from the literature that the hydrodynamic entrance length for fully developed flow for such turbulent flows to be 64.5 times the hydraulic diameter of test section [12] and here, since the section is uniform and circular, so the entrance length is taken as 64.5d, i.e 6.45m and test section is taken as 2.55m in length i.e total length of the duct to be 9m.The air flows through the inlet section becomes fully developed and then surpasses the baffles inside it.

The pitch ratio is gradually changed while performing numerical computations ranging from 1 to 5. The flow is assumed to be steady, incompressible and turbulent. The properties of working fluid air are assumed to be constant. And body forces, radiation heat transfer, and viscous dissipation are ignored.

Fig. 1 Flow geometry

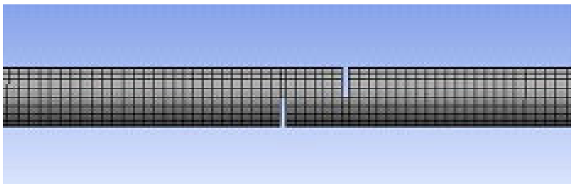


Fig. 2 Hexdominant mesh of flow geometry

III. Governing Equations

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

Continuity: $\nabla \cdot u = 0$

Momentum: $\nabla \cdot (\rho u u) = -\nabla P + \nabla \cdot (\tau)$

Energy: $\nabla \cdot (u \rho h) = \nabla (k_{eff} \nabla T)$

where P is the pressure, h the specific enthalpy, is the stress tensor, k_{eff} the effective thermal conductivity.

IV. Boundary Conditions

A fully developed turbulent flow is considered. The boundary conditions are listed below:

1. The fluid enters the entrance section with uniform velocity at z=0 ; i.e., "Velocity Inlet" is considered.

$$u=0,\ v=0,\ w=W_{in}$$

The constant temperature of the inlet section is maintained at 300 K.

2. At the walls: "No Slip" at the walls is considered.

$$u = v = 0$$

The constant temperature of the channel wall is maintained at 400 K.

3. At the exit: "Pressure Outlet" condition is considered.

$$P = 0 \text{ Pa}$$

The constant temperature of the outlet section is maintained at 380 K.

The non-dimensional parameter, Reynolds number is expressed as:

$$Re=(\rho.w.d_h/\mu)$$

where, ρ is the density of air=1.184kg/m^3 , μ is the dynamic viscosity of air = 1.849 x 10^{-5}kg/m-s, d_h is the hydraulic diameter of the circular duct which is equal to the diameter of the circular duct = 100mm.

The flow and heat transfer results are presented in terms of the dimensionless Darcy friction factor and Nusselt number.

The expression for friction factor in smooth pipe has been calculated from Petukhov equation:

$$f= (0.790 \ln Re - 1.64)^{-2} \text{ for } 3x10^3<Re<5x10^6$$

The expression for friction factor is given by:

$$f=(4 \times \tau_w/0.5 \times \rho \times w^2)$$

where τ_w is the wall shear stress and w is the magnitude of velocity throughout the flow field.

The expression for Nusselt number for the smooth pipe has been calculated from Dittus-Boelter equation:

$$Nu_0= 0.023\ Re^{0.8}Pr^{0.4} \text{ for } Re>10000$$

The average Nusselt number for the augmented pipe has been calculated numerically from FLUENT solver as:

$$Nu=(h.\ d_h /k)$$

where h and k stands for wall function heat transfer coefficient and thermal conductivity at flow conditions for the augmented pipe, respectively.

The thermal enhancement factor (η) is defined as the ratio of the heat transfer coefficient of an augmented surface, h to that of a smooth surface, h_0, at an equal pumping power and 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 pipe, respectively.

V. Numerical Procedure

Realizable k-ε model with standard wall function is treated while computations. Realizable k-ε is preferred over standard k-ε since at higher Reynolds number, the vorticity and helicity of flow tends to increase and so to achieve higher accuracy in results Realizable model is more effective. SIMPLE Algorithm is implemented and the governing equations were discretized with power law scheme and solved for finite volume method. The solutions were considered to be converged when the normalized residual values fall below 10^{-6} for all the variables as well as for energy equation.

To test the grid independency,6 different grid sizes ranging from 22864 to 27314 is taken for Re=80000 at pitch ratio 1 and it is found that from mesh size of order 25822 with orthogonal mesh quality 0.9 the value of friction factor remains almost constant with approximately 0% error upto 3-decimal places. So grid size of order 25822 with orthogonal mesh quality 0.75-0.85 is considered for current simulation of further input variables. Similar test is performed for Nusselt number as well and for all the flow geometry.

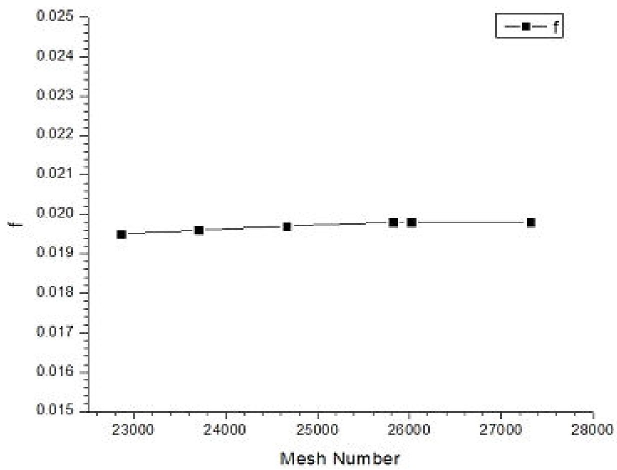


Fig. 3 Variation of friction factor with mesh number

VI. Validation Of Numerical Model

To validate the turbulence model and computational method of this present work, fully developed Nusselt numbers and friction factors for smooth pipe without baffles are obtained and compared with the values given in the literature. The Nusselt number obtained in the present work in smooth pipe is validated with Dittus-Boelter equation while the friction factor is validated with the Petukhov equation for smooth pipe.

Figs. 4 & 5 show that the Nu found in the present work are in good agreement with that of Dittus-Boelter equation for heating of fluid through the pipe within ±6% and f found in the present work shows an excellent agreement with Petukhov equation within 0.5% accuracy.

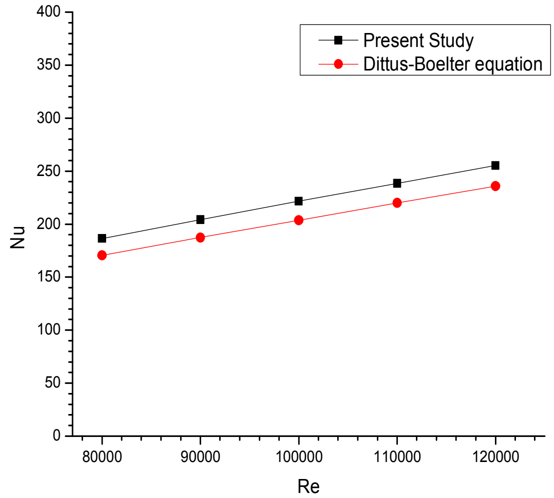


Fig. 4 Nu comparison between computational method and Dittus-Boelter equation

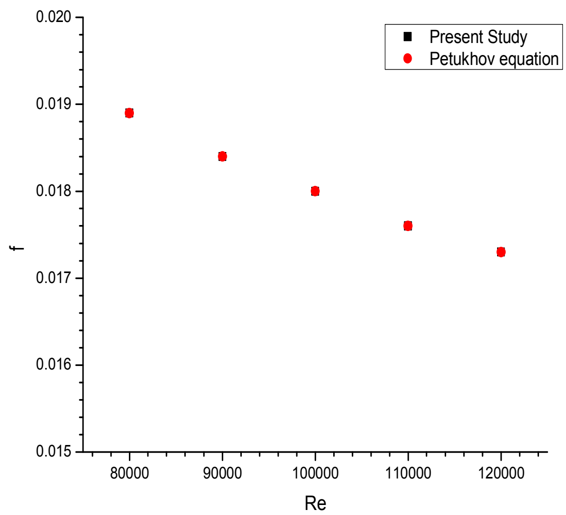


Fig. 5 f comparison between computational method and Petukhov equation

VII. Results and Discussions

A. Effect of Hemispherical-Baffles on Nusselt Number

The variation of average Nu has been depicted in the plot Nu/Nu_0 vs Re in Fig. 6. It has been revealed that the ratio Nu/Nu_0 tends to increase slightly while most of the time remains constant at higher Re number. This is due to the fact of vortex flow that is generated resulting due to fluid mixing provided by the baffles and higher turbulent intensity at such high Re number that causes destruction of thermal boundary layer and due to which the rate of heat transfer increases.

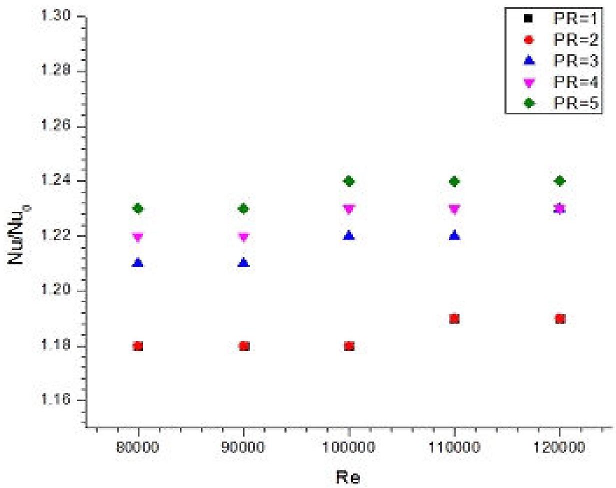
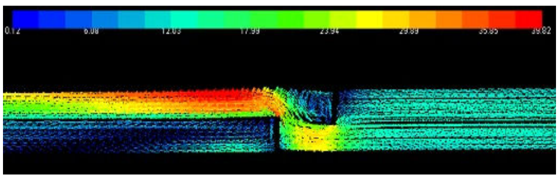


Fig. 6 Variation of Nu/Nu_0 with Re with hemispherical inline baffles

The use of two hemispherical baffles studied with the PR range 1-5 yields heat transfer rate of about 1-1.24 times higher than the smooth channel with no baffles.



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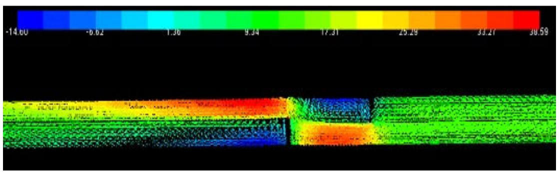

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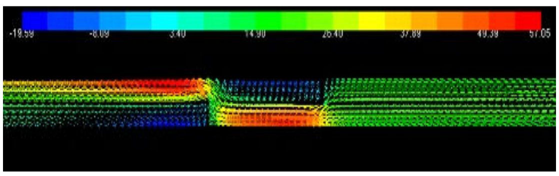

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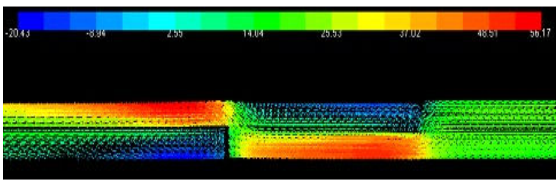

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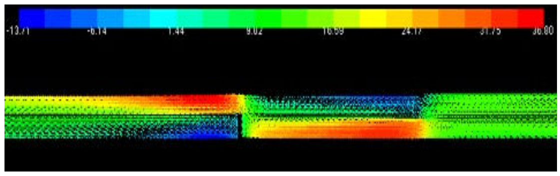

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Fig. 7 Variation of swirl flow (Re=80000) at (A) PR=1 (B) PR=2 (C) PR=3 (D) PR=4 and (E) PR=5

B. Effect of Hemispherical-Baffles on Friction Factor

Fig. 8 depicts the variation of f/f_0 with Re. And the plot reveals that there is a sharp increase of friction factor, f when baffles are used as compared to f in smooth pipe without baffles. This is due to the fact that, higher surface area causes higher dissipation of dynamic pressure of the fluid and swirl flow (Fig. 7) also augment to the cause. It has also revealed from the plot that the ratio f/f_0 tends to be same at higher pitch ratios like in PR 3, 4, 5. This is due to the approximately same value of friction factor at higher pitch ratios at higher Reynolds number. The use of two hemispherical baffles studied with the PR range 1-5 yields friction factor of about 1.05-1.06 times than the smooth channel with no baffles.

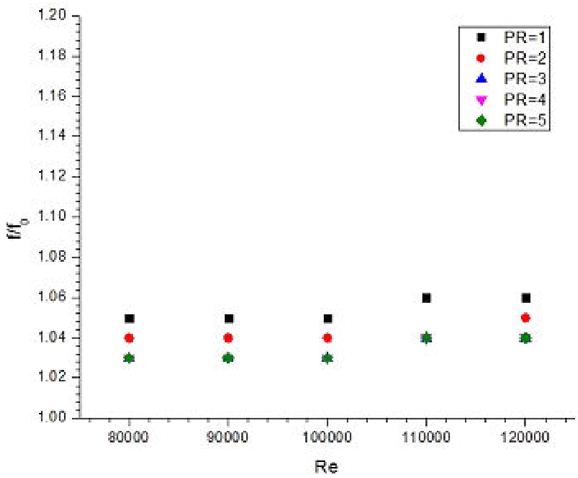
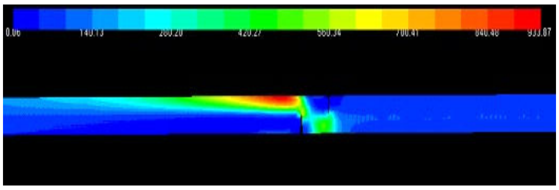


Fig. 8 Variation of f/f_0 with Re with hemispherical inline baffles

C. Effect of Pitch Ratio (PR)

The effect of baffle distance or pitch ratio has a major influence on heat transfer enhancement and friction factor reduction in pipes. Fig. 12 shows that heat transfer tends to enhance with the increasing pitch ratio, PR. This is also due to the higher turbulent intensity between the individual baffles. The baffles tends to increase the heat transfer by 1.18-1.19,1.18-1.19,1.21-1.23,1.22-1.23,1.23-1.24 times higher than that of smooth tube without baffles for PR 1,2,3,4 and 5 respectively. Fig. 11 shows that friction factor tends to decrease as baffle distance increases which lead to the reduction of dynamic pressure drop and thus it results in lower head loss. The friction factor tends to decrease from 1.05-1.06 times in case of PR=1 to 1.03-1.04 times in case of PR=5 compared to the smooth pipe without baffles.



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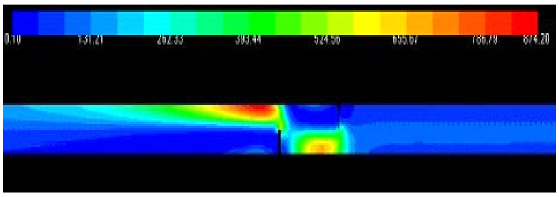

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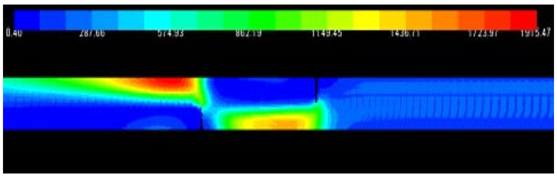

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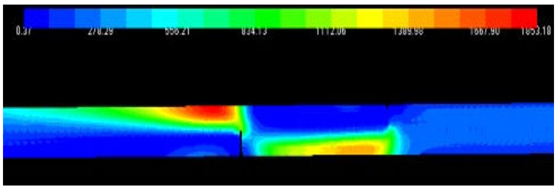

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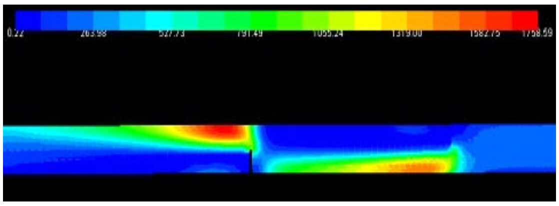

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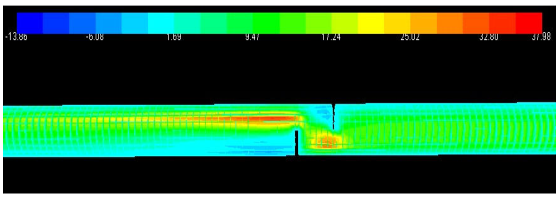
Fig. 9 Variation of dynamic pressure drop (Re=80000) at (A) PR=1 (B) PR=2 (C) PR=3 (D) PR=4 and (E) PR=5



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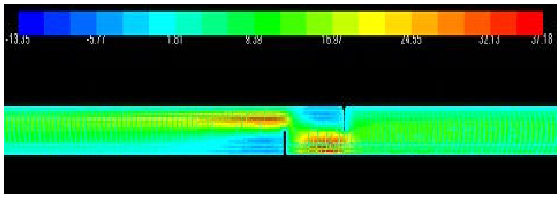

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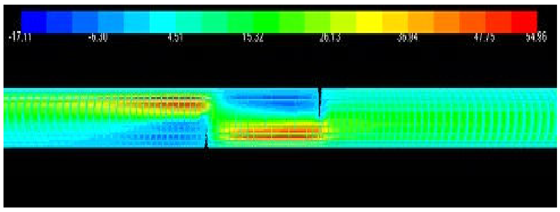

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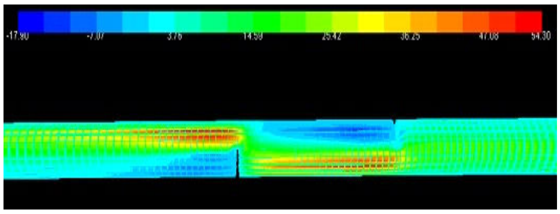

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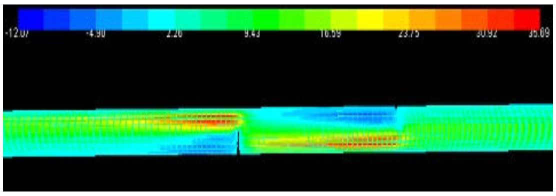

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Fig. 10 Variation of velocity contour (Re=80000) at (A) PR=1 (B) PR=2 (C) PR=3 (D) PR=4 and (E) PR=5

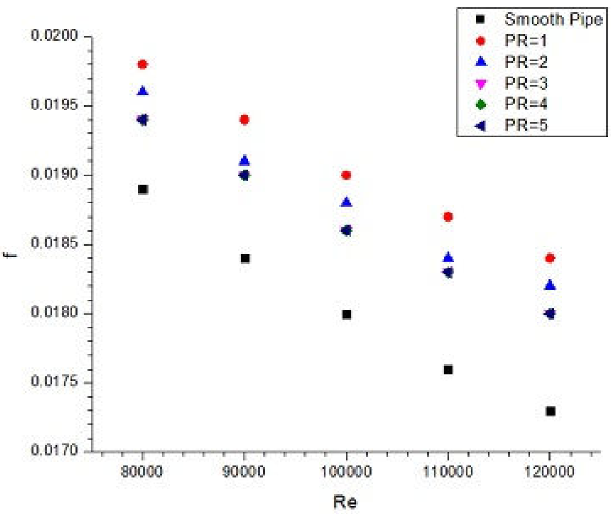

Fig. 11 Variation of friction factor (f) with Re (with and without baffles)

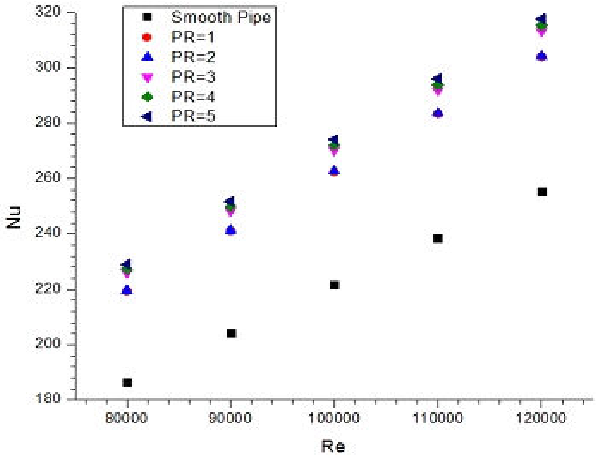

Fig. 12 Variation of Nusselt number (Nu) with Re (with and without baffles)

D. Effect of Hemispherical-Baffles on Thermal Performance

After solving the thermal enhancement factor, η, the thermal performance of the baffles is clearly estimated and from Fig. 13 it has been revealed that the thermal enhancement factor increases with PR at same Reynolds number while, it also tends to increase with Reynolds number. The value of η is found to be always above one which clearly predicts that hemispherical inline baffles in staggered configuration serves as a good insert at higher Re.

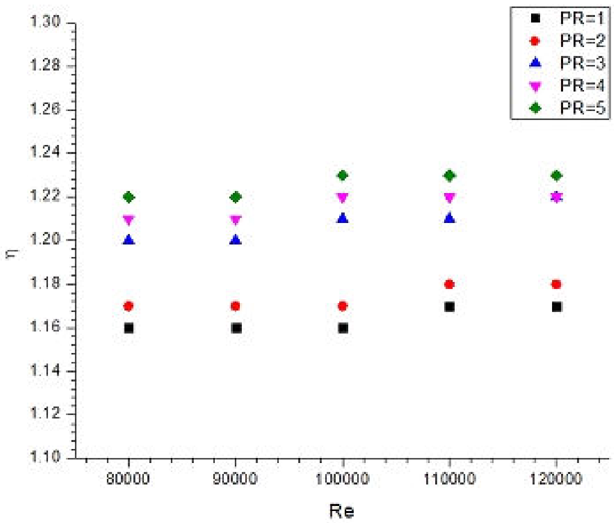

Fig. 13 Variation of η with Re at different PR

V. Conclusion

In the present study, computational study have been done to study the effect of hemispherical inline baffles on heat transfer and fluid flow characteristics in a pipe flow with the working fluid flowing through it at a very high Reynolds number ranging from 80000≤Re≤120000.The hemispherical baffle insert results in a considerable increase in heat transfer rate and friction. The net heat transfer increase after application of two baffles has been found to be 1.18-1.25 times higher than that of smooth pipe without baffles. Similarly, there is an substantial increase of friction factor (1.03-1.06 times) over the PR 1-5 and it tends to decrease with increasing PR. The heat transfer rate tends to increase at higher PR as well as with increasing Reynolds number. The thermal enhancement factor, η is in the range of 1.16-1.23.The maximum η is found to be 1.23 at Re=120000 at PR 5.Thus the application of hemispherical inline baffles at very high turbulent flows is found to be effective and promising insert to increase to reduce the frictional head loss as well as to increase the thermal performance of the heated tube.

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