



Research Article

NUMERICAL HEAT TRANSFER INVESTIGATION IN A TUBE WITH INCLINED FLOWER BAFFLES

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ABSTRACT:

A numerical investigation has been conducted to examine turbulent periodic flow and heat transfer behaviors in a three-dimensional isothermal-fluxed tube with flower baffle inserts. The computations based on the finite volume method were made by using the SIMPLE algorithm to handle the pressure-velocity coupling. The fluid flow and heat transfer characteristics are presented for Reynolds numbers ranging from 4000 to 20,000. All flower baffles are mounted periodically in the tube with a single inclination angle, $\alpha = 30^\circ$. Effects of different pitch ratios, $(PR = p/D)$ in the range of 0.75-2.0 with two flower leaf closure angles, $\theta = 25^\circ$ and 35° on heat transfer and pressure loss in the tube are studied. It is apparent that counter-rotating vortex flows created by the flower baffles exist and help induce impinging flows on the tube wall leading to drastic increase in heat transfer rate over the smooth tube. In addition, the decrease in the PR results in the rise in the heat transfer and friction factor. The computational results reveal that the optimum thermal performance for using the flower baffles is about 1.6 at $PR = 0.75$, $\theta = 25^\circ$.

Keywords: *Periodic flow, round tube, turbulent flow, heat transfer, flower baffles*

1. INTRODUCTION

The applications of vortex generators in the cooling duct heat exchanger such as ribs, dimples, grooves or baffles are commonly known in enhancement of the convective heat transfer rate leading to the compact heat exchanger and increasing the efficiency. The cooling or heating fluid is supplied into the ducts mostly mounted with several ribs/baffles to increase the degree of cooling or heating levels and this configuration is often used in the design of heat exchangers. Therefore, baffle spacing, inclination angle and height are among the most important parameters in the design of heat exchanger tubes.

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]. Then, Guo et al. [2] presented CFD simulation for the heat transfer enhancement in a circular tube fitted with helical screw-tape inserts from the viewpoint of field synergy principle, they found that the helical inserts with alternate right- and left-twists have significantly better performance than the helical inserts with uniformly right twists.

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Zhang et al. [3] presented a numerical study on heat transfer and friction factor characteristics of a tube fitted with four different widths of helical screw-tape without core-rod inserts; they found that the thermal performance of the helical screw-tape inserts of different width varies between 1.58 and 2.35. Promvonge et al. [4-7] examined numerically heat transfer and flow characteristics in a square channel with various angled continuous baffles placed on one/two walls and found that the 30° angled baffles provide the highest thermal performance. In addition, Promvonge et al. [8] found that in a 3D square-duct with inline 60° V-shaped discrete thin ribs placed on two opposite walls, the ribbed duct flow showed fully developed periodic flow and heat transfer profiles at about $x/D = 7-10$ downstream of the inlet.

Most of the numerical investigations, cited above, have considered the heat transfer characteristics for circular ducts fitted with helical screw-tape inserts and angled-baffles attached to the duct wall only. Therefore, the use of inclined flower baffles mounted repeatedly in the round tube has rarely been reported. In the present work, numerical computations for three dimensional turbulent flows through the four leaves of flower baffle mounted periodically in a round tube are conducted with the main aim being to examine the changes in the flow structure and heat transfer behaviors for turbulent region.

2. MODEL DESCRIPTION

2.1 Flower baffle geometry and arrangement

The flower baffles having four leaves each are inserted periodically in a round tube as shown in Fig. 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 as in [1]. The air enters the tube at an inlet temperature, T_{in} , and flows through flower baffles where θ is a closure angle of flower leaf, D is the tube diameter set to 0.05 m. The gap between the leaf and wall is set to 1 mm. The inclination angle of flower baffles are set to $\alpha = 30^\circ$. The axial pitch, p is a distance between the flower baffle cell set to $p = 0.75D$ to $2D$ in which p/D is defined as the baffle pitch spacing ratio, PR. To investigate an effect of the interaction between flower baffles, the pitch spacing ratio, PR is varied to PR = 0.75, 1.0, 1.5 and 2.0. In the present investigation, the closure angles are set to two angles, $\theta = 25^\circ$ and 35° , both at $\alpha = 30^\circ$.

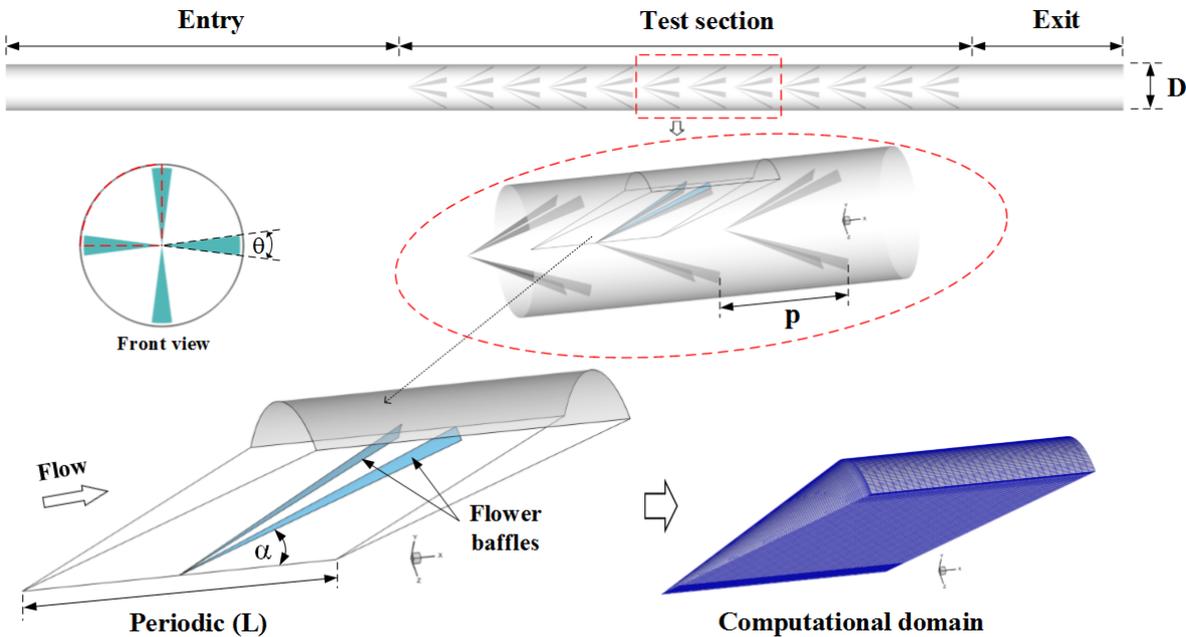


Fig. 1. Tube geometry and computational domain of periodic flow.

2.2 Boundary conditions

Periodic boundaries are used for the inlet and outlet of the flow domain. Constant mass flow rate of air at 300 K ($Pr = 0.707$) is assumed due to periodic flow conditions. The physical properties of the air are assumed to remain constant at mean bulk temperature (300 K). Impermeable boundary and no-slip wall conditions have been implemented over the tube wall as well as the baffle surfaces. The constant heat-flux on the tube wall is maintained at 600 W/m^2 while the baffles are assumed at adiabatic wall condition.

2.3 Mathematical foundation

The numerical model for fluid flow and heat transfer in a round tube was developed under the following assumptions: steady three-dimensional, turbulent and incompressible fluid flow by ignoring body forces, viscous dissipation and radiation heat transfer. Based on the assumptions above, the tube 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:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = 0 \quad (1)$$

Momentum equation:

$$\frac{\partial (\rho \vec{v})}{\partial t} + \nabla \cdot (\rho \vec{v} \vec{v}) = \rho \vec{g} - \nabla p - \nabla \cdot (\bar{\vec{\tau}}) \quad (2)$$

Energy equation:

$$\frac{\partial (\rho E)}{\partial t} + \nabla \cdot (\vec{v} (\rho E + p)) = \nabla \cdot (k_{eff} \nabla T + (\bar{\vec{\tau}}_{eff} \cdot \vec{v})) \quad (3)$$

When

$$\bar{\vec{\tau}} = \mu \left((\nabla \vec{v} + \nabla \vec{v}^T) - \frac{2}{3} \nabla \cdot \vec{v} I \right) \text{ and } E = h - \frac{p}{\rho} + \frac{v^2}{2}$$

Realizable $k - \varepsilon$ equations:

$$\frac{\partial (\rho k)}{\partial t} + \frac{\partial}{\partial x_i} (\rho k v_i) = \frac{\partial}{\partial x_j} \left(\alpha_k \mu_{eff} \frac{\partial k}{\partial x_j} \right) + G_k + G_b - \rho \varepsilon - Y_M \quad (4)$$

$$\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial}{\partial x_i} (\rho \varepsilon v_i) = \frac{\partial}{\partial x_j} \left(\alpha_\varepsilon \mu_{eff} \frac{\partial \varepsilon}{\partial x_j} \right) + C_{1\varepsilon} \frac{\varepsilon}{k} (G_k + C_{3\varepsilon} G_b) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} - R_\varepsilon + S_\varepsilon \quad (5)$$

All the governing equations were discretized by the QUICK differencing scheme while the velocity-pressure coupling in the equations is handled by the SIMPLE algorithm and solved using a finite volume approach [9]. The solutions were considered to be converged when the normalized residual values were less than 10^{-5} for all variables but less than 10^{-9} only for the energy equation.

Four parameters of interest in the present work are the Reynolds number (Re), friction factor (f), Nusselt number (Nu) and thermal enhancement factor (TEF). The Reynolds number is defined as

$$Re = \rho \bar{u} D / \mu \quad (6)$$

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

$$f = \frac{(\Delta p / L) D}{\frac{1}{2} \rho \bar{u}^2} \quad (7)$$

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

$$Nu_x = h_x D / k \quad (8)$$

The area-averaged Nusselt number can be obtained by

$$Nu = \frac{1}{A} \int Nu_x \partial A \quad (9)$$

The thermal enhancement factor (TEF) 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

$$TEF = \frac{h}{h_0} \bigg|_{pp} = \frac{Nu}{Nu_0} \bigg|_{pp} = (Nu/Nu_0) / (f/f_0)^{1/3} \quad (10)$$

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

The computational domain is resolved by regular Cartesian elements or hexagonal elements. A grid independence solution was obtained by comparing the solution for different grid levels. It was found that the difference in Nu between the results of grid system of about 168,000 and 320,000 is less than 0.5%. Considering both convergence time and solution precision, the grid system of 168,000 with finer resolution near the wall was adopted for the current computation as shown in Fig. 1.

3. RESULT AND DISCUSSION

3.1 Verification of smooth tube

Verification of the heat transfer and friction factor of the smooth tube with no baffle is performed by comparing with the previous values as shown in Fig. 2. The current numerical smooth tube result is found to be in excellent agreement with exact solutions obtained from the open literature [10] for both the Nu and f , less than 6.1 and 4.8 % deviation, respectively.

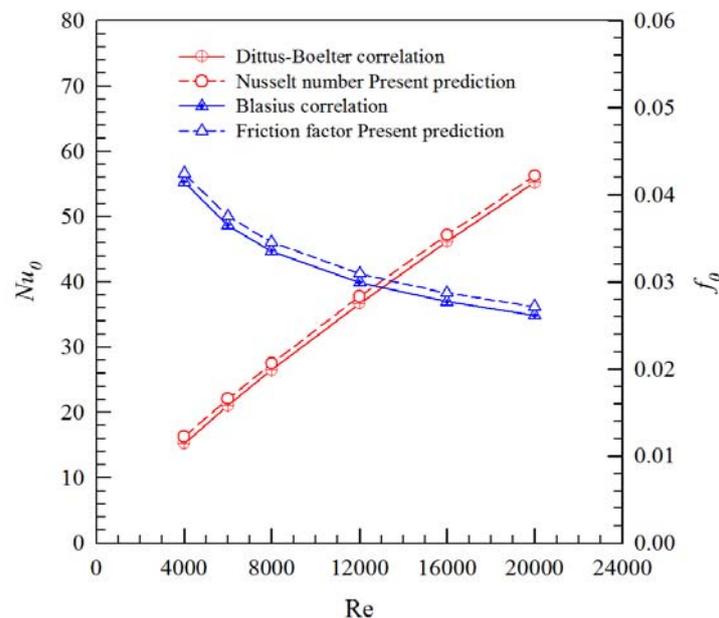


Fig. 2. Verification of Nusselt number and friction factor for smooth tube.

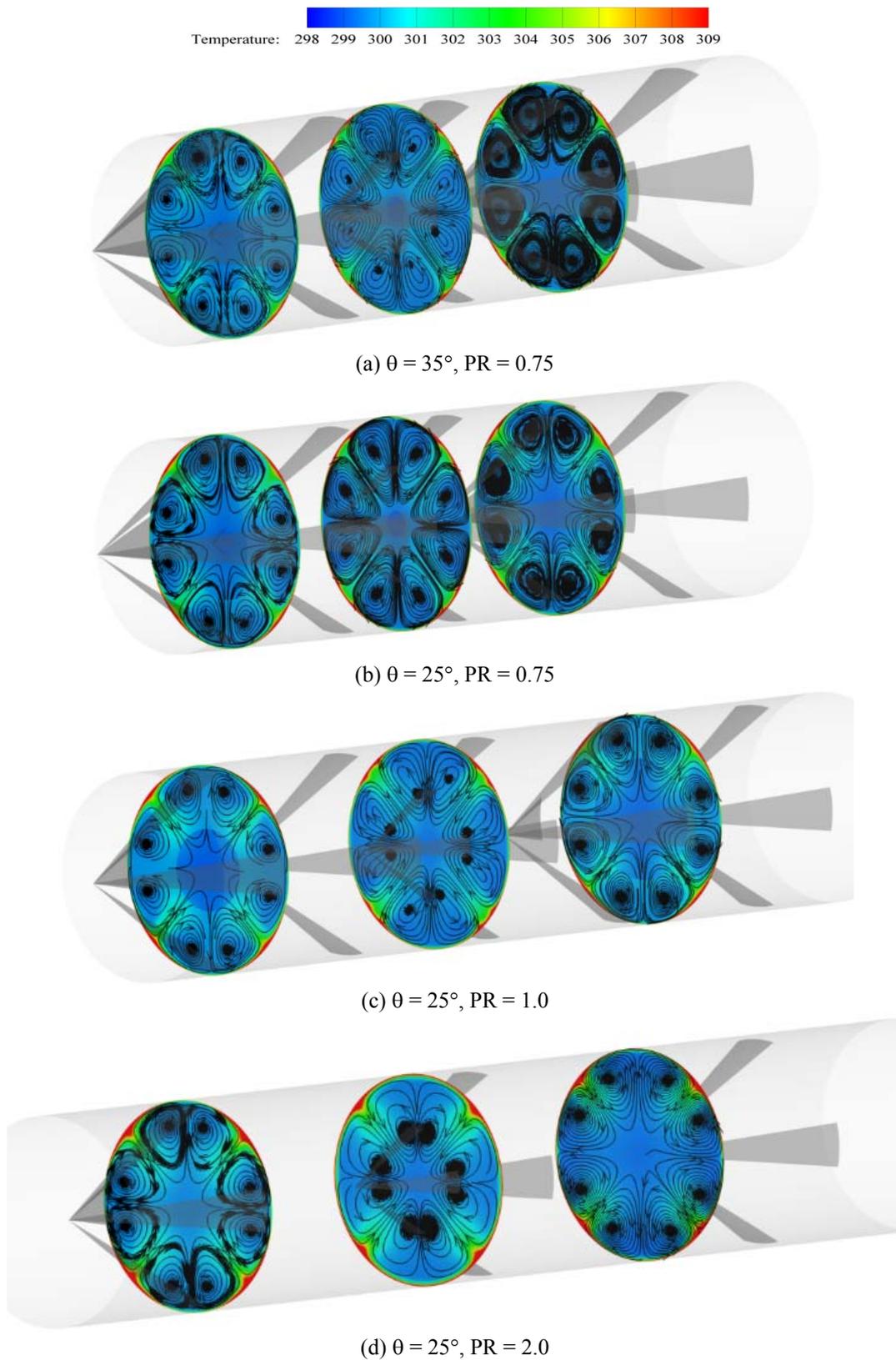


Fig. 3. Temperature contours with streamlines in transverse planes for flower baffles at $Re = 12,000$.

3.2 Flow structure

The flow structure and vortex coherent in a tube with four leaves of flower baffle for $\theta = 35^\circ$, PR = 0.75 and $\theta = 25^\circ$, PR = 0.75-2.0 at $\alpha = 30^\circ$, Re = 12,000 can be displayed by considering the temperature contours with streamline plots in transvers plane as depicted in Fig. 3. The transverse planes are extracted at the tailing, middle and leading areas of the flower baffle. The temperature contours with streamline plots of the four baffle leaves can create four main counter-rotating vortices over the tube that help induce impingement flows over certain regions of the wall leading to drastic increase in heat transfer. In a flow module, two centers of the main counter-rotating vortices (eyes of the vortex cores) starting at the baffle leading end location plane moves up and becomes close each other at the middle plane. When the flow reaches the next cell/module and then they repeats themselves again as can be observed in Fig.

3.3 Heat transfer, pressure loss and performance evaluation

The local Nusselt number (Nu_x) contour plots of the round tube with four leaves of flower baffles at $\alpha = 30^\circ$, Re = 12,000 for $\theta=25^\circ$, PR=0.75-2.0 and for $\theta = 35^\circ$, PR = 0.75 are presented in Fig. 4. In the figure, it appears that the high Nu_x values for the tube with flower baffles are seen in large areas over the tube wall. The peaks can be observed at the impingement areas on the tube wall where the baffle trailing ends attached. This means that the vortex flows provide a significant influence on the temperature field, because it can induce better fluid mixing between the wall and the core flow regions, leading to a high temperature gradient over the heating wall. In the case studied, it appears that the flower baffle with $\theta = 35^\circ$, PR = 0.75 gives the highest Nu as can be seen in Fig. 4a.

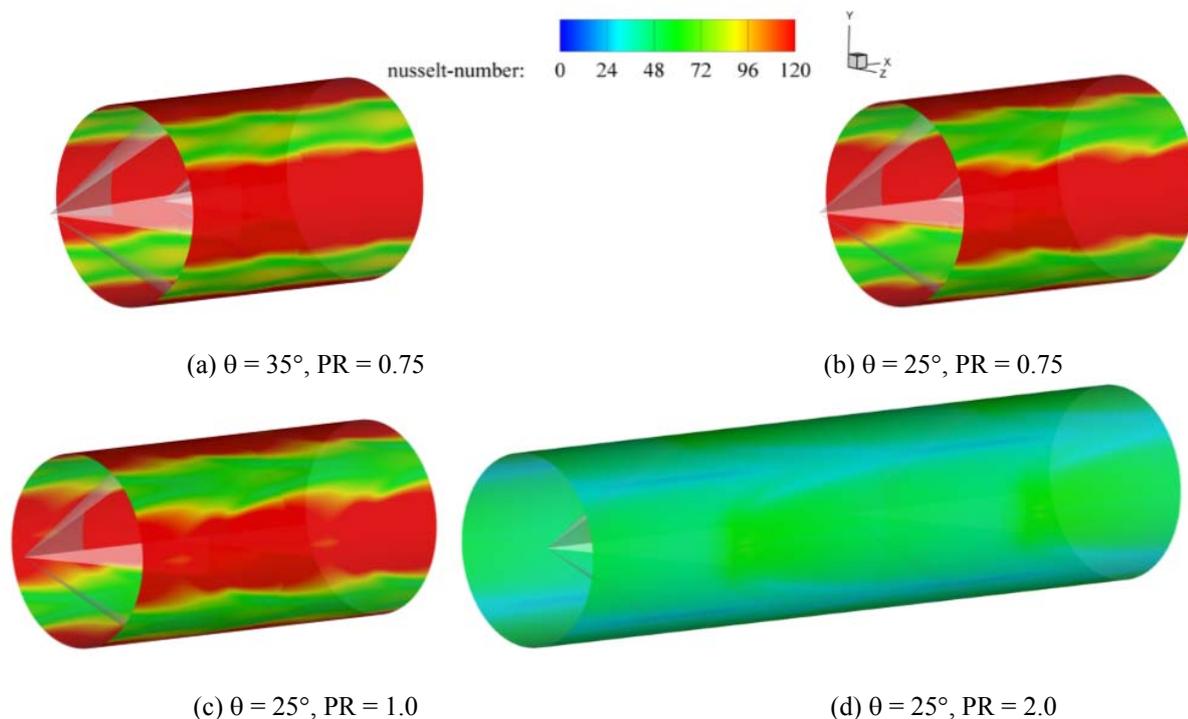
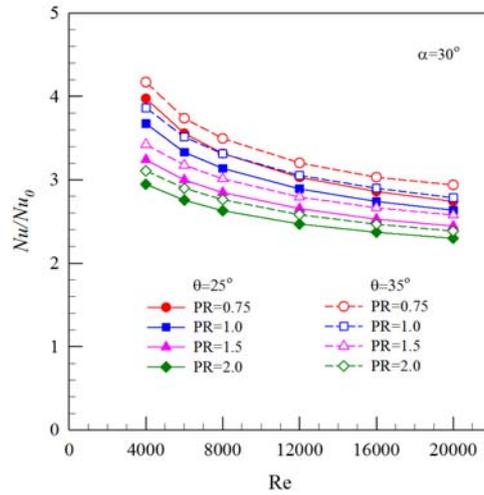
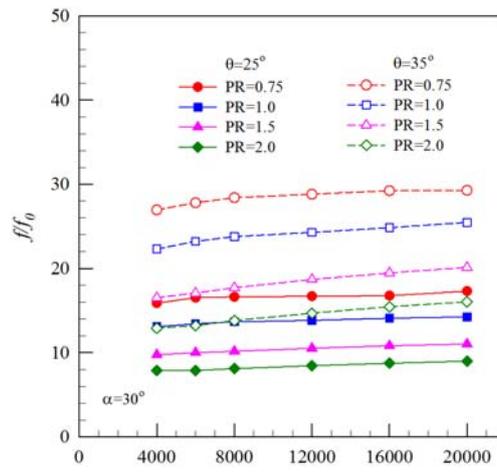


Fig. 4. Nu_x contours for flower baffle at Re = 12,000.

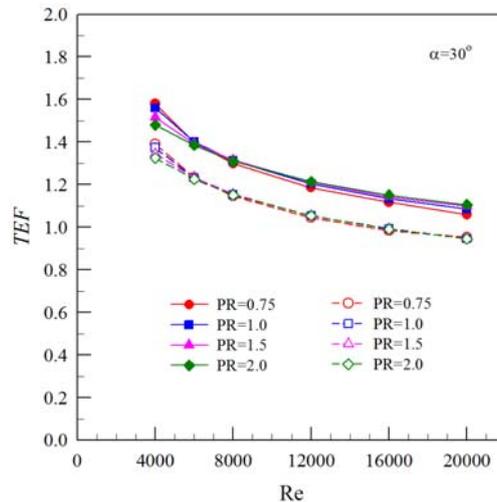
The variation of the average Nu/Nu_0 ratio with Re at different θ and PR values are depicted in Fig. 5a. It is worth noting that the Nu/Nu_0 tends to decrease with the rise of Re and PR for all θ values. The higher PR value yields the decrease of Nu/Nu_0 . The highest Nu/Nu_0 around 4.2 times is found at $\theta=35^\circ$, PR=0.75. The baffle provides the increase in heat transfer rate of about 2.4-4.2 times the smooth tube. Figure 5b presents the variation of the friction factor ratio, f/f_0 with Re for various θ and PR values. In the figure, it is noted that the f/f_0 tends to slightly increase with the rise in Re. The use of flower baffles leads to much higher friction factor than the smooth tube. The increase in PR gives rise to the reduction of friction factor. The f/f_0 for using the flower baffles is about 8-30 times depending on the θ , PR and Re values.



(a) Nu/Nu_0



(b) f/f_0



(c) TEF

Fig. 5. Variation of Nu/Nu_0 , f/f_0 and TEF with Re at various θ and PR values.

Figure 5c displays the variation of TEF with Re at different θ and PR values. It is visible that the TEF tends to decrease with the rise of Re and θ values. It is interesting to note that the TEF is nearly free from PR but is strongly dependent on θ as seen in Fig. 5c. The TEF for the flower baffles is found to be about 1.0-1.6, depending on the PR, θ and Re values. This indicates the merit of using the flower baffle for thermal performance improvement in a heat exchanger tube, especially for $\theta = 25^\circ$, PR = 0.75 to obtain higher TEF.

4. CONCLUSION

A numerical computation of turbulent periodic flow and heat transfer characteristics in a round tube fitted with flower baffles has been carried out. The use of four baffle leaves can create four main counter-rotating vortex flows over the tube that can help induce impingement/reattachment flows over the tube wall leading to greater increase in heat transfer. The order of heat transfer enhancement is about 2.4-4.2 times for $\theta = 25^\circ$ - 35° at PR = 0.75-2.0 while the pressure loss is enlarged in a range of 8-30 times above the smooth tube. The maximum thermal enhancement factor (TEF) is about 1.6 for $\theta = 25^\circ$, PR = 0.75 at Re = 4000, indicating much higher thermal performance over the smooth tube.

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