3D ANALYSIS OF LAMINAR PERIODIC FLOW ON HEAT TRANSFER IN A CIRCULAR TUBE WITH ANGLED ORIFICE INSERTED

Withada Jedsadaratanachai[1], Ratsak Poomsalood[1], Amnart Boonloi[2], Pongjet Promvonge[1]*

[1]Department of Mechanical Engineering, Faculty of Engineering, King Mongkut's Institute of Technology Ladkrabang, Bangkok 10520, Thailand
[2] Department of Mechanical Engineering Technology, College of Industrial Technology King Mongkut's University of Technology North Bangkok, Bangkok 10800, Thailand.
*e-mail: kppongje@kmitl.ac.th

Abstract
The article presents a numerical analysis of laminar periodic flow and heat transfer characteristics in a constant temperature-surfaced circular tube mounted with 30° angled orifices of various blockage ratio. The computations were based on a finite volume method and the SIMPLE algorithm implemented. The fluid flow and heat transfer characteristics are presented for Reynolds number based on the circular tube diameter ranging from 100 to 1200. The angled orifices were placed repeatedly on the inner tube walls to generate a pair of streamwise counter–rotating vortex flows. Effects of different blockage ratios (BR=b/D) with a single pitch ratio of 1.5 on heat transfer and pressure loss in the tube were examined. The computation revealed that the pair of vortex flows created by the orifices exists and induces impingement and attachment flows on a side and lower walls leading to substantial increase in heat transfer rate. The increase in the BR leads to the rise of heat transfer rate and friction factor. The result showed that the optimum thermal performance enhancement factor of about 2.63 is found at Re 1200.

Keywords: periodic flow, circular tube, laminar flow, heat transfer, angled orifice

Introduction

Vortex/swirl flow has been of considerable interest over the past decades because of their occurrence in industrial applications, such as gas-turbines, swirl burner, cyclone combustor, and heat exchangers. In tube heat exchangers, the vortex flow created by wire-coil/twisted tape inserts [1,2] is one of the commonly used passive heat transfer enhancement technique while ribs/baffles are widely employed in channel heat exchangers [3]. Ribs/baffles used in many engineering applications of heat exchanger systems are due to their high thermal loads and decreased dimensions. The cooling or heating fluid is supplied into the channels mostly mounted with several 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, angle of attack and height are among the most important parameters in the design of channel heat exchangers. The concept of periodically fully developed flow was first introduced by Patankar et al. [4] to numerically investigate the heat transfer and flow characteristics in a duct. Since then, the periodically fully developed flow condition has been widely used to study thermal characteristics in staggered transverse-baffled channels with different baffle heights and pitch spacing lengths [5]. 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. [6].

Sripattanapipat and Promvonge [7] numerically studied the laminar periodic flow and thermal behaviors in a two dimensional channel fitted with staggered diamond–shaped baffles and found that the diamond baffle with half apex angle of 5 – 10° performs slightly better than the flat baffle. Promvonge et al. [8] also examined numerically the laminar heat transfer in a square channel with 30° angled baffle placed on two opposite walls and reported that a pair of streamwise vortex flows occurs and induces impingement jets on the walls of the inter-baffle cavity and the BTE sidewall. The maximum thermal enhancement factor was around 4.0 at BR = 0.15, PR = 2 and Re = 2000. Again, Promvonge et al. [9, 10] investigated numerically the laminar flow structure and thermal behaviors in a square channel with 45° angled baffles on one/two walls. Longitudinal counter-rotating vortex flows were created along the channel and vortex-induced-impingement jets appeared on the upper, lower and BLE side walls while the maximum thermal enhancement factors of about 2.6 at BR = 0.2, PR = 1 and Re = 1000 for using the 45° inline baffles on two walls were reported. From Ref. [8], the studies about the effect of BR and PR in range from 1 to 2 were report. Therefore, the study on effect of BRs in range from 0.10 – 0.25 for angled orifice has rarely been reported. In the present work, the numerical computations for three dimensional laminar periodic tube flows over a 30° angled orifice with different BRs for single PR = 1.5 placed on the tube walls were conducted to examine the changes in the flow structure and its thermal performance.

Flow description

Angled orifice geometry and arrangement

The flow system of interest was a circular tube with angled orifices repeatedly placed on the inner walls as depicted 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 air enters the tube at an inlet temperature, T_{in}, and flows over a 30° angled orifice where b is the orifice height, D set to 0.05 m, is the tube diameter and b/D is known as the blockage ratio, BR. The axial pitch, L or distance between the 30° angled orifice cell is set to L = 1.5D in which L/D is defined as the pitch spacing ratio, PR = 1.5. To investigate an effect of blockage ratio, BR is varied in a range of BR = 0.10 – 0.25 for α = 30° in the present investigation.

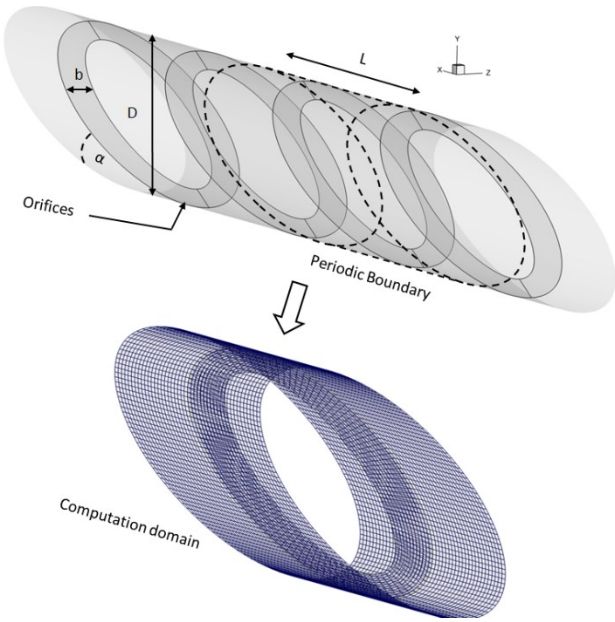


Figure 1 Circular tube geometry and computational domain of periodic flow.

Boundary conditions

Periodic boundaries are used for the inlet and outlet of the flow domain. Constant mass flow rate of air with 300K (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 tube walls as well as the orifice. The constant temperature of all the tube walls is maintained at 310K while the orifice plate is assumed at adiabatic wall conditions.

Mathematical foundation

The numerical model for fluid flow and heat transfer in a circular tube was developed under the following assumptions:

- Steady, three-dimensional, laminar and incompressible fluid flow.
- Constant fluid properties.
- Body forces, radiation heat transfer and viscous dissipation are ignored.

Based on the above assumptions, 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:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

Momentum equation:

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] \tag{2}$$

Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_j}\left(\Gamma \frac{\partial T}{\partial x_j}\right) \tag{3}$$

where Γ is the thermal diffusivity and is given by $\Gamma = \mu / \mathrm{Pr}$.

Apart from the energy equation discretized by the QUICK scheme, the governing equations were discretized by the power law scheme, decoupling with the SIMPLE algorithm and solved using a finite volume approach [11]. 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, friction factor, Nusselt number and thermal enhancement factor. The Reynolds number is defined as

$$\mathrm{Re} = \rho \bar{u} D / \mu \tag{5}$$

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

$$f = \frac{(\Delta p / L) D}{\frac{1}{2}\rho \bar{u}^2} \tag{6}$$

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

$$Nu_x = \frac{h_x D}{k} \tag{7}$$

The area-averaged Nusselt number can be obtained by

$$Nu = \frac{1}{A}\int Nu_x \partial A \tag{8}$$

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

$$TEF = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{Nu}{Nu_0}\right|_{pp} = \left(Nu/Nu_0\right)/\left(f/f_0\right)^{1/3} \quad (9)$$

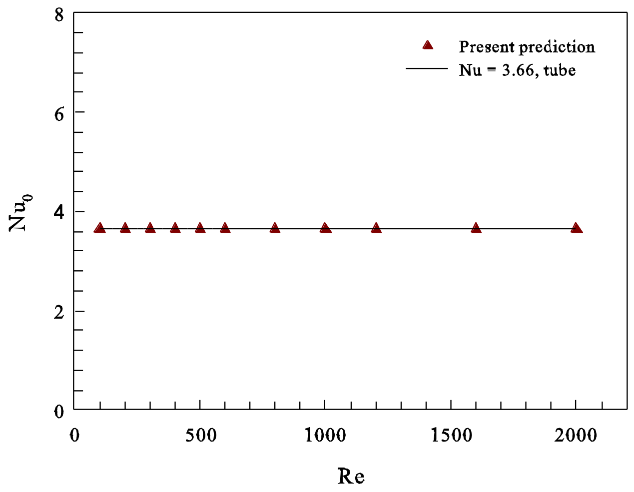
where Nu_0 and f_0 stand for Nusselt number and friction factor for the smooth tube, respectively.

The variation in Nu and f values for the 30° angled orifice at BR = 0.2 and Re = 1000 is less than 0.2% when increasing the number of cells from 120,000 to 240,000, hence there is no such advantage in increasing the number of cells beyond this value and thus, the grid system of 120,000 cells was adopted for the current computation.

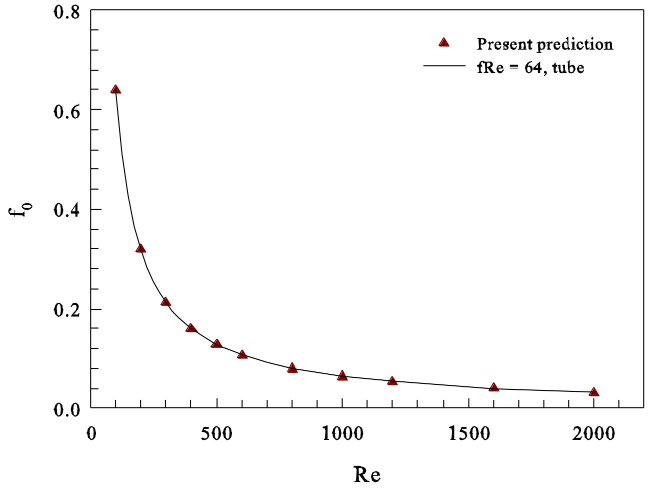
Results and discussion

Verification of a smooth tube

Verification of the heat transfer and friction factor of the smooth circular tube without orifice is performed by comparing with the previous values under a similar operating condition as shown in Fig. 2a and b, respectively. The current numerical smooth tube result is found to be in excellent agreement with exact solution values obtained from the open literature [12] for both the Nusselt number and the friction factor, less than ±0.25% deviation. The exact solutions of the Nu and f for laminar flows over the smooth circular tube with constant wall temperature are, respectively, $Nu_0 = 3.66$ and $f_0 = 64/\mathrm{Re}$, [12].



(a)



(b)

Figure 2 Verification of (a) Nusselt number and (b) friction factor for smooth tube

Flow structure

The flow and vortex coherent structure in the circular tube with angled orifice on the walls can be displayed by considering the streamline plots as depicted in Fig. 3. Here the streamlines plot of the angled orifice modules are presented at Re = 1000, BR = 0.2 and PR = 1.5. It is visible in Fig. 3 that there are two centers main vortex flows in the tube (or an "eye" of vortex core) at the orifice leading end (OLE) plane, plane 1 as you seen in Fig. 3. When moving to the plane 2, two vortex cores appear to spirally move apart at the lower walls, while the two centers main vortex flows are gradually vanishing. Then at the plane 3, the two centers main vortex flows become the weak strength of secondary flow, while two small vortex cores are generated at the upper walls and get stronger as you seen in plane 4. Then, the both appear and vortex flow repeats itself when getting to the OLE of the next module, plane 5. This vortex flow is similar to the all cases.

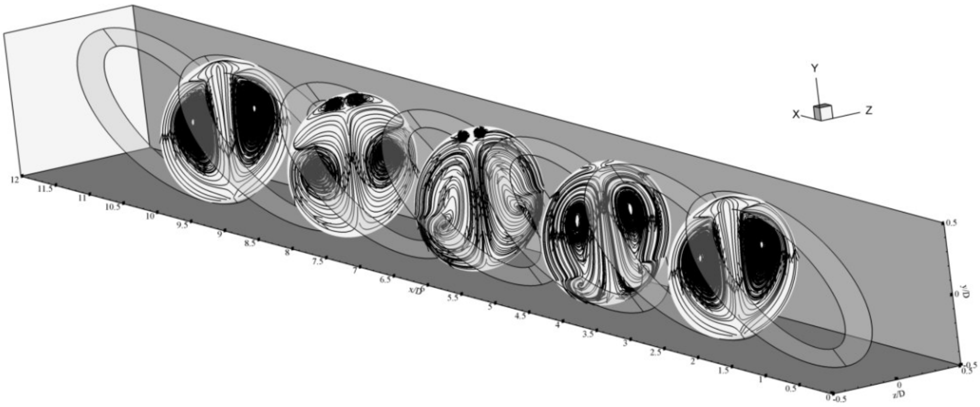


Figure 3 Streamlines in transverse planes for angled orifice at Re = 1000 and BR = 0.2

Heat transfer, pressure loss and performance evaluation

Fig. 4. displays the contour plots of temperature field in transverse planes for Re = 1000 and BR = 0.2. The figure shows that there is a major change in the temperature field over the tube for using the angled orifice. 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.

Local Nu_x contours for the tube walls with the angled orifice for BR = 0.2 and Re = 1000 are presented in Fig. 5. In the figure, it appears that the higher Nu_x values over the walls are seen in a larger area, except for small regions at the upper walls and the orifice base. The peaks are observed at the impingement areas on the sidewall and the lower wall by show the red areas.

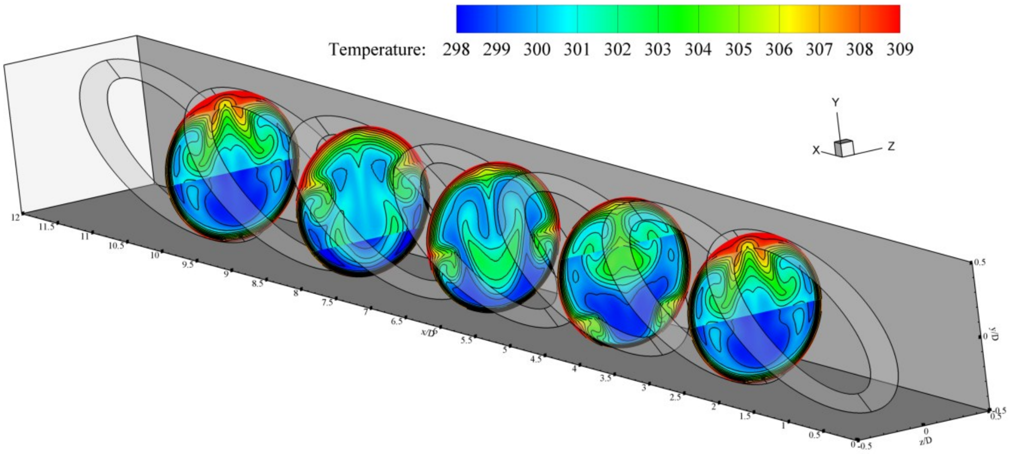


Figure 4 Temperature contours in transverse planes at Re = 1000 and BR = 0.2

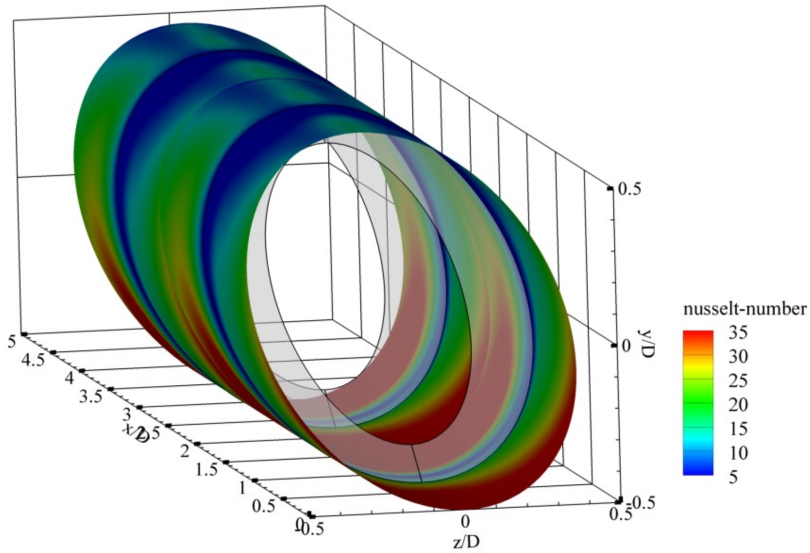


Figure 5 Nu_x Contours for BR = 0.2 and Re = 1000

Fig. 6. exhibits the boundary heat flux, the stream line impinging on the tube wall. The red area show impinging of the vortex flows and provide higher heat transfer rate than other areas.

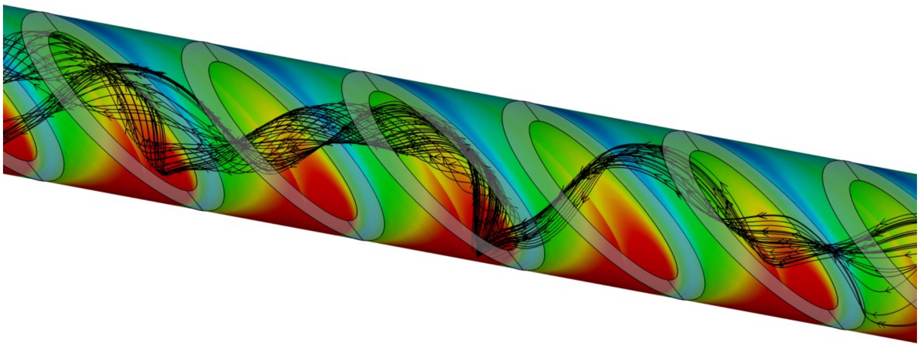


Figure 6 Streamlines impinging on the tube walls for BR=0.2, Re=1000

The variation of the average Nu/Nu_0 ratio with Reynolds number at different BR values is depicted in Fig. 7. It is worth noting that the Nu/Nu_0 value tends to increase with the rise in Reynolds number for all BR values. The higher BR results in the increase in the Nu/Nu_0 value. The Nu value for the angled orifice with BR = 0.25 is found to be about 6.6 times over the smooth tube. Thus, the generation of vortex flows from using the angled orifice as well as the role of better fluid mixing and the impingement is the main reason for the augmentation in heat transfer of the tube. The use of the angled orifice with the BR range studied yields heat transfer rate of about 1–6.6 times higher than the smooth tube with no orifice.

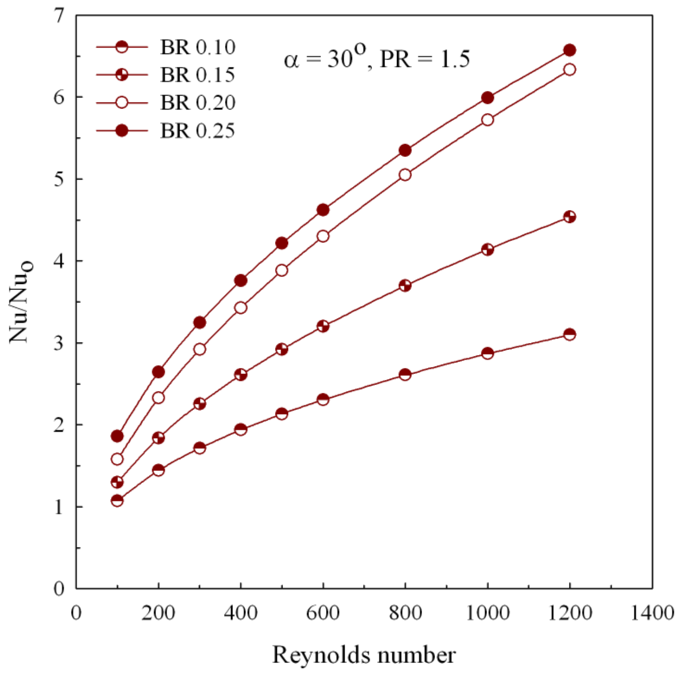


Figure 7 Variation of Nu/Nu_0 with Reynolds number at various orifice BRs

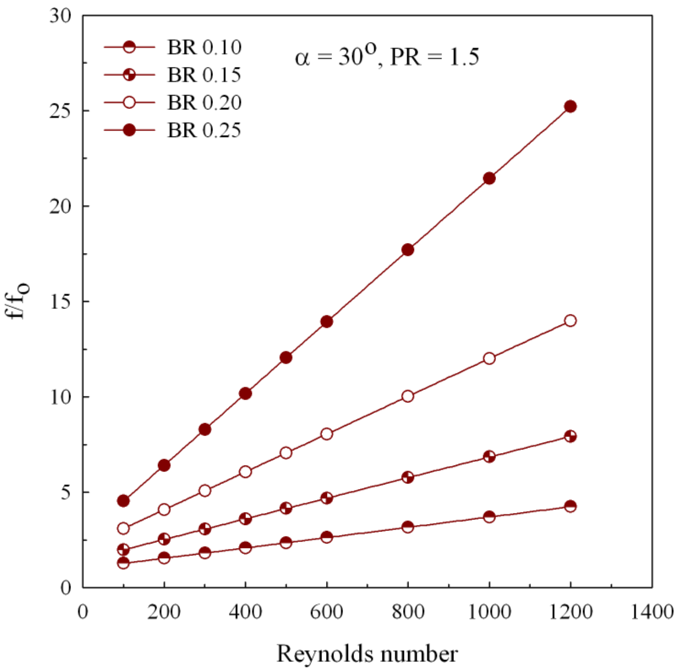


Figure 8 Variation of f/f_0 with Reynolds number at various orifice BRs

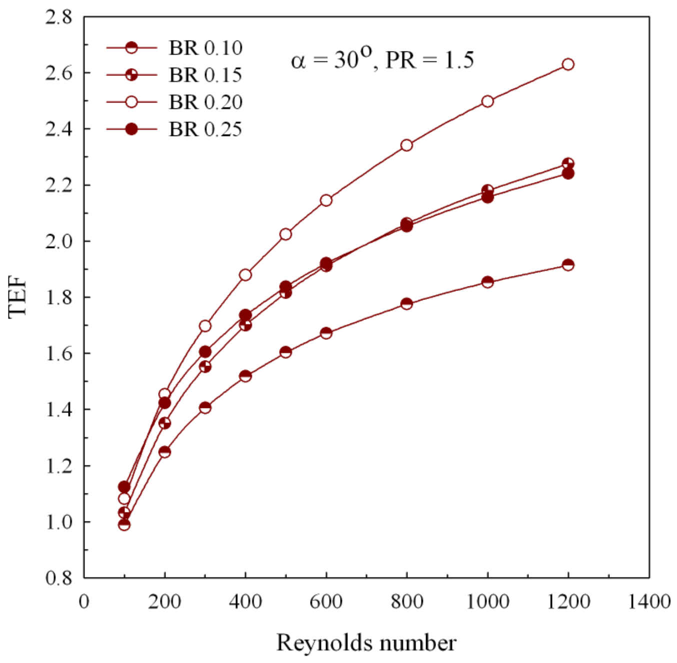


Figure 9 Variation of TEF with Reynolds number at various orifice BRs

Fig. 8. presents the variation of the friction factor ratio, f/f_0 with Reynolds number for various angled orifice BRs. In the figure, it is noted that the f/f_0 tends to increase with the rise of Reynolds number and with increase of BR values. The use of the angled orifice leads to considerable increase in friction factor in comparison with the smooth tube with no angled orifice. The increase in the BR value gives rise to the increase of friction factor. The BR = 0.25 provides the highest value of friction factor around 25 times above the smooth tube. The f/f_0 value for using the angled orifice is found to be about 1–25 times over the smooth tube depending on the BR and Reynolds number values.

Fig. 9. exhibits the variation of thermal enhancement factor (TEF) for air flowing in the angled orifice tube. In the figure, the thermal enhancement factor for all BR values tends to increase with the rise of Re. The BR = 0.2 angled orifice gives the highest enhancement factor of about 2.63 at the highest Re. The thermal enhancement factor of the angled orifice is seen to be above unity for all BRs and varied between 1.0 to 2.63, depending on the BR and Re values.

Conclusions

Numerical computations of laminar periodic flow and heat transfer characteristics in a round tube inserted with 30° angled orifice on the tube wall are performed. The main vortex flow created by the angled orifice exists and helps to induce impingement flows over the tube wall leading to drastic increase in heat transfer in the round tube. The order of heat transfer enhancement is about 1 – 6.6 times for using the angle orifice with BR = 0.10 – 0.25 at PR = 1.5. The enlarged pressure loss is in a range of 1 to 25 times above the smooth tube. Thermal enhancement factor for the angled orifice is higher than unity and its maximum value is about 2.63 at BR = 0.20, Re = 1200 and PR = 1.5, indicating higher performance over the smooth tube.

References

1. Promvonge P. Thermal performance in circular tube fitted with coiled square wires. *Energ Convers Manage* 2008; **49**: 980–987.
2. Eiamsa-ard S, Thianpong C, Eiamsa-ard P and Promvonge P. Convective heat transfer in a circular tube with short-length twisted tape insert. *Int Commun Heat Mass Transf* 2009; **36**: 365 – 371.
3. Promvonge P and Thianpong C. Thermal performance assessment of turbulent channel flows over different shaped ribs. *Int Commun Heat Mass Transf* 2008; **35**: 1327 – 1334.
4. Patankar SV, Liu CH and Sparrow EM. Fully developed flow and heat transfer in ducts having streamwise-periodic variations of cross sectional area. *ASME J Heat Transf* 1977; **99**: 180 – 186.
5. Webb BW and Ramadhyani S. Conjugate heat transfer in a channel with staggered ribs. *Int J Heat Mass Trans* 1985; **28**: 1679 – 1687.
6. Lopez JR, Anand NK and Fletcher LS. Heat transfer in a three-dimensional channel with baffles. *Numerl Heat Transf A: Appl* 1996; **30**: 189 – 205.
7. Sripattanapipat S and Promvonge P. Numerical analysis of laminar heat transfer in a channel with diamond-shaped baffles. *Int Commun Heat Mass Transf* 2009; **36**: 32 – 38.
8. Promvonge P, Jedsadaratanachai W and Kwankaomeng S. Numerical study of laminar flow and heat transfer in square channel with 30° inline angled baffle turbulators. *Appl Therm Eng* 2010; **30**: 1292 – 1303.
9. Promvonge P, Sripattanapipat S, Tamna S, Kwankaomeng S and Thianpong C. Numerical investigation of laminar heat transfer in a square channel with 45 deg inclined baffles. *Int Commun Heat Mass Transf* 2010; **37(2)**: 170 – 177.
10. Promvonge P, Sripattanapipat S and Kwankaomeng S. Laminar periodic flow and heat transfer in square channel with 45° inline baffles on two opposite. *Int J Therm Sci* 2010; **49**: 963 – 975.
11. Patankar SV. Numerical heat transfer and fluid flow (McGraw–Hill). 1980.
12. Incropera F and Dewitt PD. Introduction to heat transfer 5rd edition (John Wiley & Sons Inc). 2006.