The Second TSME International Conference on Mechanical Engineering
19-21 October, 2011, Krabi

Numerical Heat Transfer Study in a Square Channel with Zigzag-Angled Baffles

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Abstract

The article presents a numerical investigation on laminar periodic flow and heat transfer behaviors in a three-dimensional isothermal-wall square-channel fitted with zigzag-inline angled baffles having a constant length equal to channel height, (H) for six flow blockage ratios, (BR=b/H), and a pitch spacing ratio, *s*=L/H=2 and an attack angle, α=45°. The inline angled baffles are mounted repeatedly on the upper and lower channel walls to create longitudinal vortex flows throughout the test channel. The computations based on the finite volume method with the SIMPLE algorithm have been conducted for the airflow in the form of Reynolds numbers ranging from 100 to 1000. Effects of six different baffle heights on heat transfer and flow behaviors in the channel are examined. The computational results reveal that the maximum heat transfer is at *BR* = 0.30. For the given conditions, the maximum thermal performance enhancement factor (*TEF*) of the zigzag-angled baffle is found to be about 2.7 at *BR*=0.15 and Re=1000.

1. Introduction

The need of high performance thermal systems leads to interest in developing techniques for heat transfer enhancement resulting in the reduction of overall heat exchanger dimensions and increasing efficiency. For decades, baffles or ribs have been used in many thermal systems 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. The use of baffles completely makes the change of the flow field and thus the distribution of the local heat transfer coefficient. Apart from inducing the mainstream separation, a single transverse baffle can create a recirculation zone ahead of it, a second recirculation zone behind the baffle and a reattachment at the channel wall. In addition, if the baffle is placed at an inclination angle with respect to the axial direction, secondary flows are induced over the channel, resulting in the rise in the heat transfer rate towards the upstream baffle region with respect to the downstream one. Although the heat transfer is increased through the baffle arrangement, the pressure drop across the channel is also increased due to the decreased flow area effects. Therefore, the baffle spacing, angle of attack and height are among the most important parameters used in the design of channel heat exchangers. It is, thus, difficult to realize the advantage of baffle arrangements or geometry. The use of staggered transverse baffles with height and pitch spacing of 0.1-0.5 and 1 times of the channel height, respectively, is often suggested in most previous works.

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 transverse baffles mounted on two opposite walls is found at a Reynolds number below 600 and for such conditions the flow is free of vortex shedding. Webb and Ramadhyani [3] numerically investigated the fluid flow and heat transfer characteristics in a channel with staggered baffles, based on the periodically fully developed flow conditions of Patankar et al. [1]. Kellar and Patankar [4] studied the heat transfer behaviors in a channel with staggered baffles and reported that the heat transfer increases with the rise in baffle height and with the decrease in baffle spacing. Their results showed similar behaviors as results of Webb and Ramadhyani [3]. 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. Mousavi and Hooman [6] numerically studied the heat transfer behavior in the entrance region of a channel with staggered baffles for Reynolds numbers ranging from 50 to 500 and baffle heights between 0 and 0.75. A numerical study of laminar periodic flow and thermal behaviors in a two-dimensional channel fitted with staggered diamond-shaped baffles was performed by Sripattanapipat and Promvonge [7]. They reported that the diamond baffle with half apex angle of 5°-10° provided slightly better thermal performance than the flat baffle. Promvonge et al. [8] also investigated numerically the laminar heat transfer enhancement in a square channel with 45° inclined baffle mounted on one wall. They found that a single main stream wise vortex flow occurs for using the periodic baffles and P-vortex exists and helps to induce impingement jets on the wall of the baffle cavity and the BTE sidewall. The appearance of the vortex-induced impingement flows led to the thermal enhancement factor of about 2.2 at BR=0.4 and Re=1200.

Most of the investigations on laminar flow as mentioned earlier have considered the heat transfer characteristics for blockage ratio and spacing ratio values for porous or solid transverse baffles only. Although inclined, V-shaped, continuous or broken ribs have been studied extensively; most of the ribs investigated were based on square cross section or thick ribs only. Therefore, the study on inclined short-length baffles (or very thin ribs) in square channels has rarely been reported. In the present work, the numerical computations for three dimensional laminar periodic channel flows over a 45° zigzag-angled baffle pair mounted on two opposite walls are conducted with the main aim being to examine the flow structure and heat transfer performance. The use of the inline 45° baffles placed periodically on the opposite walls of the tested channel is expected to generate a pair of longitudinal vortex flows through the channel to better mixing of fluid flows between the core and the wall.

2. Flow configuration and mathematical foundation

2.1 Baffle geometry and arrangement

The system of interest is a horizontal square channel with two 45°-baffle pairs placed on the upper and lower channel walls in tandem for inline arrangements as shown in Fig. 1. The left baffle end of the first baffle pair are attached on the left sidewall while the right baffle end of the second baffle pair are attached on the right sidewall. The computational domain and the grid size for all baffles high are similar. 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 air enters the channel at an inlet temperature, T_{in}, and flows over 45°-angled baffles where b is the baffle height, H set to 0.05 m, is the channel height and b/H is known as the blockage ratio, BR. The baffle having a constant length, (H) and the axial pitch, L or distance between the baffle cell is set to L=2H in which L/H is defined as the pitch ratio, s = 2. To investigate an arrangement effect of the interaction between baffles, the baffle blockage ratio, BR is varied in a range of BR = 0.05 - 0.30 for α = 45° in the present investigation.

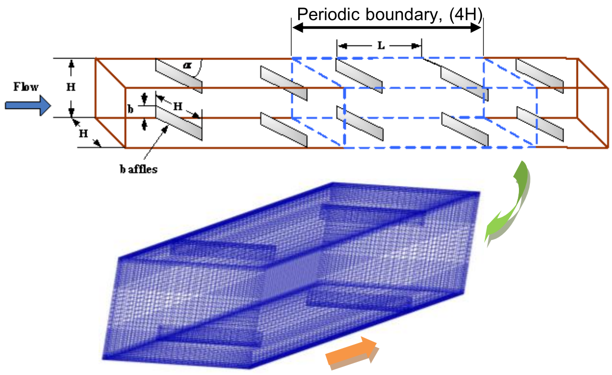


Fig. 1. Channel geometry and computational domain of periodic flow

2.2. Mathematical foundation

The numerical model for fluid flow and heat transfer in the square channel was developed under the following assumptions:

- Steady three-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 Naviere 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

$$\Gamma = \frac{\mu}{\Pr} \tag{4}$$

Apart from the energy equation discretized by the QUICK scheme, the governing equations were discretized by the second order upwind differencing scheme, decoupling with 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^{-8} 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

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

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

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

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

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

The average Nusselt number can be obtained by

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

The thermal performance 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

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

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

The computational domain is resolved by regular Cartesian elements. For this channel flow, however, regular grid was applied throughout the domain. A grid independence solution was obtained by comparing the solution for different grid levels. It was found that the difference in heat transfer coefficient between the results of grid system of about 156,800 and 258,000 is less than 1.2%. Considering both convergent time and solution precision, the grid system of 156,800 with finer resolution near the walls was adopted for the computational model.

2.3. Boundary conditions

Periodic boundaries were used for the inlet and outlet of the flow domain. Constant mass flow rate of air with 300 K (Pr = 0.7) was 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 were assumed to remain constant at average bulk temperature. Impermeable boundary and no-slip wall conditions have been implemented over the channel walls as well as the baffle. The constant temperature of all the channel walls was maintained at 310 K while the baffle plate was assumed at adiabatic wall (high thermal resistance) conditions.

3. Results and discussion

3.1. Verification of smooth square-channel

Verification of the heat transfer and friction factor of the smooth square-channel without baffle is performed by comparing with the previous values under a similar operating condition as shown in Figure 2a and b, respectively. The present smooth square-channel result is found to be in excellent agreement with exact solution values obtained from the open literature [10] for both the Nusselt number and friction factor, less than ±0.5% deviations. The exact solutions of the Nusselt number and friction factor for laminar flow over square channels with constant wall temperature are as follows [10]:

$$Nu_0 = 2.98 \quad (10)$$

$$f_0 = 57/Re \quad (11)$$

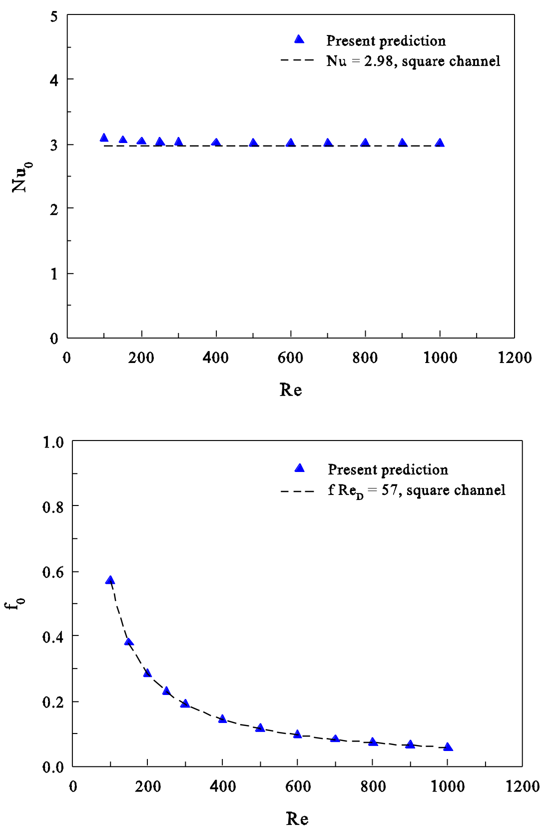


Fig. 2. Verification of (a) Nusselt number and (b) friction factor of smooth square channel.

3.2. Flow structure

It is necessary to understand the flow structure and behavior in the baffled channel before discussing the results. The flow structure in the channel mounted periodically with various baffles on the lower and upper walls can be displayed by considering the streamline plots as depicted in Fig. 3 show the streamlines in each transverse planes for 45° zigzag-angled baffles at Re = 1000, BR = 0.15 and s = 2 respectively.

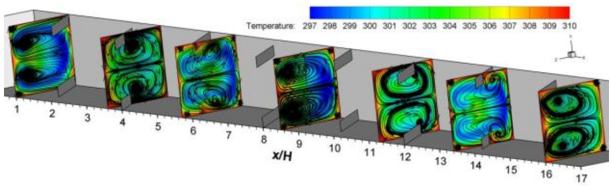
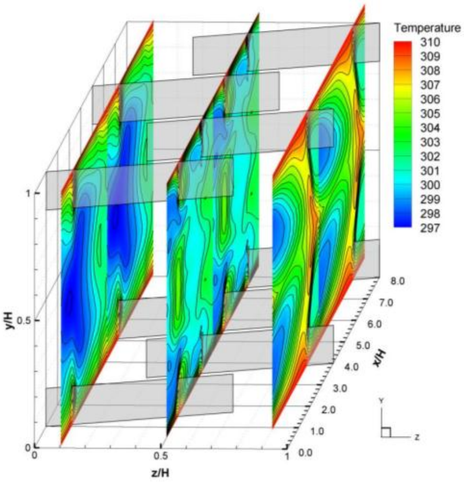


Fig. 3. Streamlines in transverse planes for 45° zigzag-angled baffles at Re=1000 and BR=0.15.

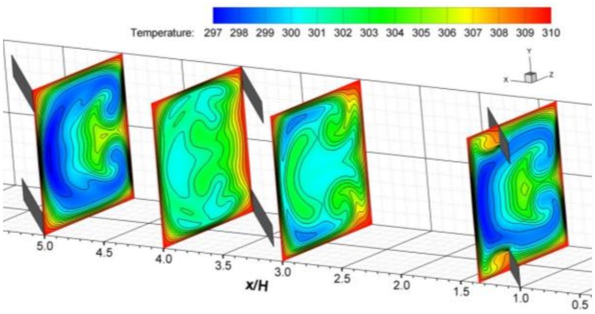
3.3. Heat transfer

Fig. 4a and b displays the contour plots of temperature field for the 45° zigzag-angled baffle at Re=1000 and BR=0.15 in (a) axial and (b) transverse planes, respectively. The figure shows that there is a major change in the temperature field over the channel. This means that the P-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 channel wall. The higher temperature gradient can be observed where the flow impinges the channel walls while the lower one is found at the right sidewall area.

Local Nu_x contours for the channel walls with the 45° zigzag-angled baffle at Re=1000 and BR=0.15 is presented in Fig. 5. In the figure, it is apparent that the higher Nu_x distributions over the walls with the 45° zigzag-angled baffle are seen to be in a larger area, except for the corner region and the baffle trailing end (right sidewall) area. The peaks are observed at the impingement areas on the side wall attached to the baffle leading end (left sidewall). This indicates a merit of employing the 45°-baffle over the smooth channel with no baffle for enhancing heat transfer.



(a) Axial plane



(b) Transverse plane

Fig.4 Temperature contours for 45° zigzag-angled baffle, BR = 0.15 at Re = 1000.

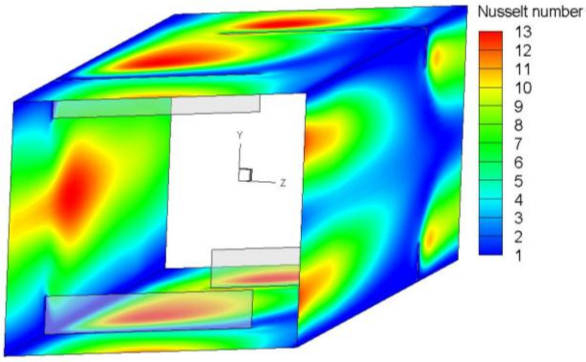


Fig. 5 Local Nusselt number contours for zigzag-angled baffle, BR=0.15 at Re=1000.

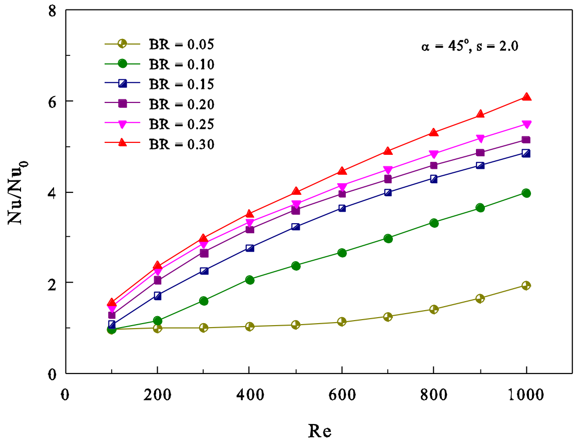


Fig. 6 Variation of Nu/Nu_0 with Reynolds number for 45° zigzag-angled baffles at various BRs.

The variation of the average Nu/Nu_0 ratio with Reynolds number for various baffle BRs is depicted in Fig. 6. It is worth noting that the Nu/Nu_0 value tends to increase with the rise of Reynolds number for all BR values. The higher BR value results in the increase in the Nu/Nu_0 value. The scrutiny of Fig. 6 reveals that the use of the 45° zigzag- angled baffles with the BR range studied yields heat transfer rate of about 1-6 times higher than the smooth channel with no baffle, depending on the BR values.

3.4. Pressure loss

In general, the heat transfer augmentation is concerned with penalty in terms of increased friction coefficient resulting in higher pressure drop. The wall static pressure contour over the channel wall for the 45° zigzag-angled baffles at Re=1000, BR=0.15 is portrayed in Fig. 7. In the figure, it is visible that the high wall pressure regions are found at the upstream corner regimes where the baffle end is attached and at the upstream area of the baffle base. The smallest wall pressure regimes can be seen at the corner area downstream of the baffle-attached walls and at the area behind the baffle.

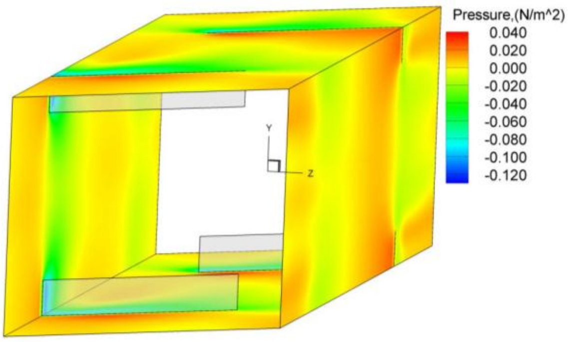


Fig. 7 Static pressure contour for 45° zigzag angled baffle at Re=1000 and BR=0.15.

Fig. 8 presents the variation of the friction factor ratio, f/f_0 with Reynolds number for various BRs. In the figure, it is noted that the f/f_0 tends to increase with the rise of Reynolds number and BR values. The use of the zigzag-angled baffle leads to an extreme increase in friction factor in comparison with the plain channel with no baffle. The decrease in the BR value gives rise to the reduction in friction factor. The friction factor ratio for the 45° zigzag-angled baffle is found to be about 1-18 times over the smooth channel depending on the BR and Reynolds number values. Thus the flow blockage due to the existence of the baffle is a vital factor to cause an extreme pressure drop.

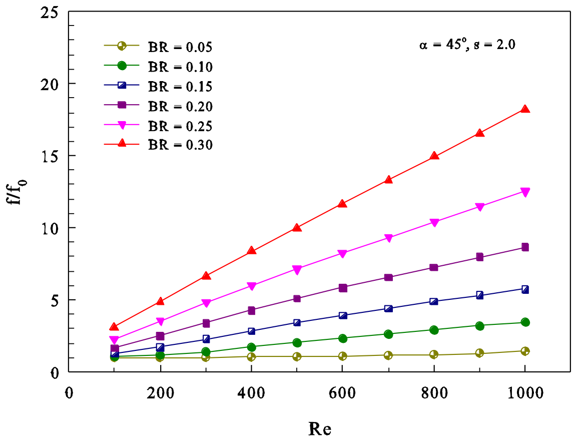


Fig. 8 Variation of f/f_0 with Reynolds number for 45° zigzag-angled baffles at various BRs.

3.5. Performance evaluation

Fig. 9 shows the variation of thermal performance enhancement factor (*TEF*) for air flowing through the square-channel with 45° zigzag-angled baffle. In the figure, the *TEF* tends to increase with the rise in Reynolds number. The BR=0.15 baffle at Re>450 provides the highest *TEF* because of lower friction factor. At Re<450, the *TEF* of the BR=0.15 is less than that of the BR=0.20 because the Nu/Nu_0 of the BR=0.20 is higher than that of the BR=0.15 while the f/f_0 is slightly larger. The *TEF*s of the zigzag-angled baffle are seen to be above unity for BR=0.1-0.3, Re>200 and vary between 1.1 and 2.7, depending on the BR and Re values. For the results investigated, the zigzag-angled baffle with BR=0.15, Re>450 gives the best overall thermal performance and one with BR=0.20 yields slightly lower. This suggests that the 45° zigzag-angled baffle with BR=0.10-0.15 should be used to obtain higher thermal performance.

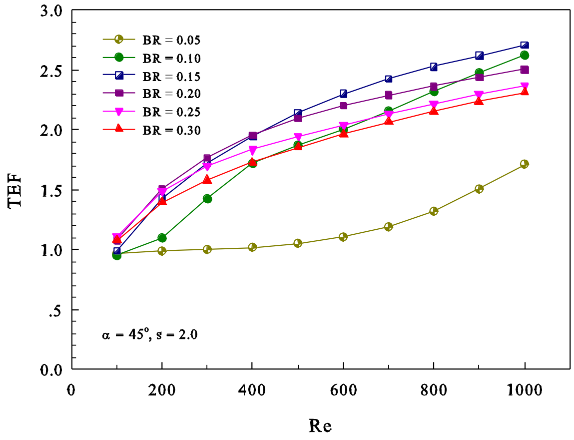


Fig. 9 Thermal enhancement factor for 45° zigzag-angled baffles at various BRs.

4. Conclusions

Laminar periodic flow and heat transfer characteristics in a square channel fitted with 45° angled baffle elements in tandem, zigzag arrangements on two opposite walls have been investigated numerically. The P-vortex flows created by using the zigzag-angled baffles help to induce impingement flows on the sidewall and wall in the baffle cavity leading to drastic increase in heat transfer. The order of enhancement is about 100-600% for using the 45° zigzag-angled baffle with BR=0.05-0.30. However, as expected, the heat transfer augmentation is associated with enlarged pressure loss ranging from 1 to 18 times above the smooth channel depending on the BR and Re values. The heat transfer enhancement for the 45° zigzag-angled baffle is around 100-270% higher than smooth channel and the *TEF* is found to be much higher than unity and its maximum value is about 2.7 indicating much higher performance over the smooth channel.

5. Acknowledgement

The author would like to acknowledge with appreciation, the Thai-Nichi Institute of Technology (TNI) and the Thailand Research Fund (TRF) for the financial support of this work.

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