Numerical Analysis of Laminar Heat Transfer Augmentation in a Square Channel fitted with V-Baffles

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Abstract

This article presents a numerical analysis of laminar periodic flow and heat transfer in a constant temperature-surfaced square duct fitted with V-baffle (or winglets) vortex generators. The computations are based on a finite volume method, and the SIMPLE algorithm has been implemented. The laminar fluid flow and heat transfer characteristics are presented for Reynolds numbers based on the hydraulic diameter of the channel ranging from 100 to 2000. To generate a pair of streamwise counter-rotating vortex (P-vortex) flows through the tested channel, the V-baffle with the attack angle of 30° are mounted in tandem and inline arrangement on both upper and lower walls of the tested channel with the V-baffle tip pointing downstream (V-Downstream). Effects of different baffle heights, BR in range from 0.1-0.4 at a single pitch ratio of 1.5 on heat transfer and pressure loss in the square channel are studied. It is apparent that the P-vortex flows exist and help to induce impinging flows on a side wall and the upper and lower wall leading to drastic increase in heat transfer rate over the test channel. In addition, the increase in the baffle height results in the rise of Nusselt number and friction factor values. The computational results reveal that the optimum thermal enhancement factor of the V-baffle is about 4.25 at BR=0.2 and Re=2000.

***Keywords**:* Periodic flow, Square channel, Laminar flow, Heat transfer, Winglet.

1. Introduction

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. 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. [1] 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 spacing [2,3].

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. [4]. Sripattanapipat and Promvonge [5] 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. [6] also examined numerically the laminar heat transfer in a square channel with 45 deg angled baffle placed on one wall and reported that a single streamwise vortex flow occurs and induces impingement jets on the wall of the interbaffle cavity and the BTE sidewall. Again, Promvonge et al. [7] and [8] also investigated numerically the laminar flow structure and thermal behaviors in a square channel with 30° or 45° inline baffles on two opposite walls. Two streamwise counter-rotating vortex flows were created along the channel and vortex-induced-impingment 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; and of around 4.0 at BR = 0.15, PR = 2 and Re = 2000 for using the 45° and 30° baffles were reported, respectively.

Most of previous investigations on laminar flow have considered the heat transfer characteristics for transverse or inclined baffles only. Therefore, the study on V–baffles channels has rarely been reported. In the present work, the numerical computations for three dimensional laminar periodic channel flows over a 30° inline V–baffle pair mounted on two opposite channel walls are conducted to examine the changes in the flow structure and its thermal performance.

2. Flow description

2.1 Baffle geometry and arrangement

The system of interest is a horizontal rectangular channel with a 30° V–Upstream baffle pair placed on the upper and lower channel walls in tandem and inline arrangement 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 in Ref. [1]. The air enters the channel at an inlet temperature, T_{in}, and flows over a 30° inline V–baffle pair 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 axial pitch, L or distance between the baffle cell is set to L = 1.5H in which L/H is defined as the pitch spacing ratio, PR = 1.5 and the width of the channel, W is equal to H for the channel aspect ratio, AR of 1. To investigate an effect of the interaction between baffles, the flow blockage ratio, BR is varied in a range of BR = 0.1–0.4 for α = 30° in the present investigation.

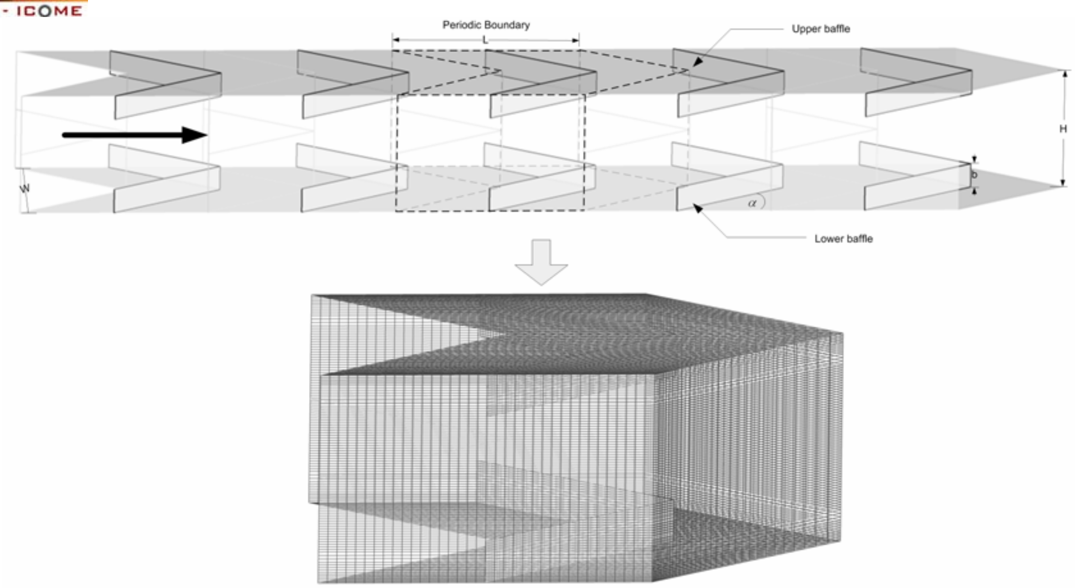


Fig. 1 Channel geometry and computational domain of periodic flow.

2.2 Boundary conditions

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

3. Mathematical foundation

The numerical model for fluid flow and heat transfer in a 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 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

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

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 [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, 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 channel, 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

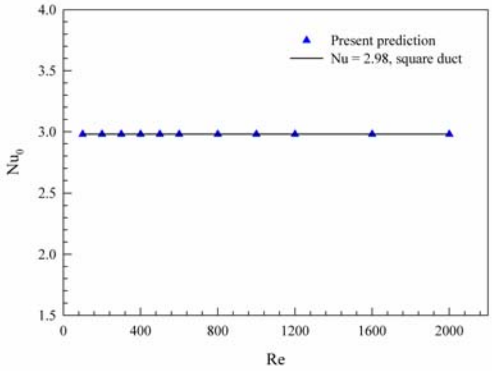
$$Nu = \frac{1}{A} \int Nu_x \partial A \tag{8}$$

The thermal enhancement factor (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

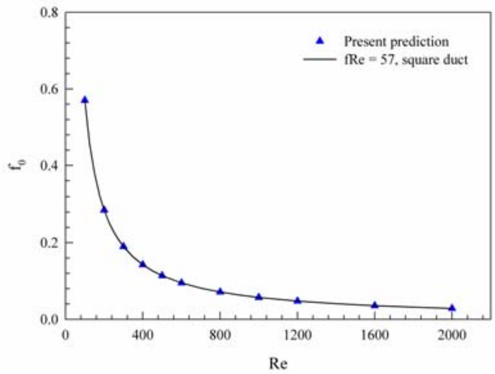
$$\eta = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{Nu}{Nu_0}\right|_{pp} = \left(Nu/Nu_0\right)/\left(f/f_0\right)^{1/3} \tag{9}$$

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

The variation in Nu and f values for the 30° inline V–baffles at BR = 0.1 and Re=500 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. Considering both convergent time and solution precision, the grid system of 120,000 cells was adopted for the current computational model.



(a)



(b)

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

4. Results and discussion

4.1 Verification of a smooth channel

Verification of the heat transfer and friction factor of the smooth channel without baffle 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 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 the friction factor, less than ±0.25% deviation. The exact solutions of the Nusselt number and the friction factor for laminar flows over smooth channels with constant wall temperature are as follows [10]:

$$Nu_0 = 2.98 \tag{10}$$

$$f_0 = 57/\mathrm{Re} \tag{11}$$

4.2 Flow structure

The flow and vortex coherent structure in the channel with V–baffles on the lower and upper walls can be displayed by considering the streamline plots as depicted in Fig. 3. Here the velocity vectors, contours and streamlines of the V–baffle modules are presented at Re = 1000, BR = 0.2 and PR = 1.5. It is visible in Fig. 3 that there are four main vortex flows in the channel, four small vortices at the corners of the channel, and four ones at the middle of the upper and lower walls. Considering two main vortex flows at the lower part of a module with a single pitch of 1.5H, two centers of the main counter-vortex flows (common-vortex flow-up) at the BLE plane, plane A1 in Fig. 3, are at about the middle region above the baffle tip of the lower part while two small vortices appear on the center part of the lower wall. When moving to the quarter module pitch location, plane A2, two vortex core centers appear to spirally move apart to the lower corners. The upstream vortex core centers and the two small centered-wall vortices are gradually vanishing while the downstream ones including the two small corner vortices are appearing at the half module pitch location as seen in plane A3. Then, both downstream ones make a helical move close together until passing the BTE to the three-quarter pitch location, plane A4 and the vortex core centers repeat itself when getting to the BLE of the next module, plane A5.

4.3 Heat transfer

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 channel for using the V-baffle. 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.

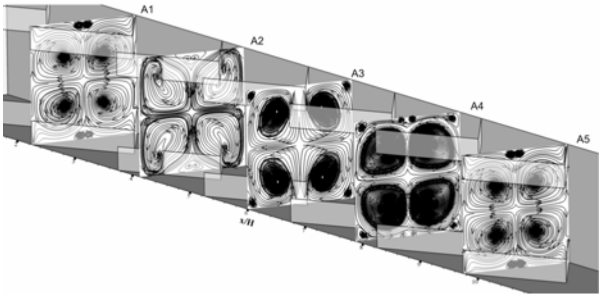


Fig. 3. Streamlines in transverse planes for V–baffle at Re = 1000 and BR = 0.2.

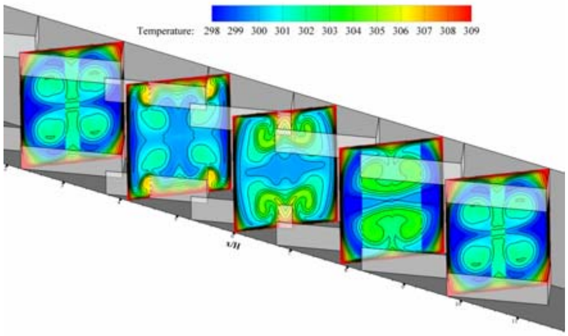


Fig. 4 Temperature contour in transverse planes, at Re=1000 and BR=0.2.

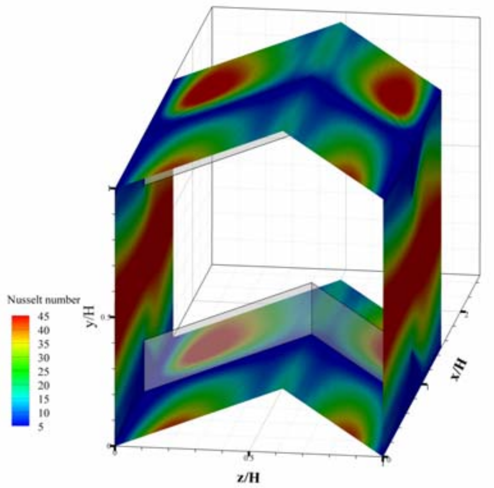


Fig. 5 Nu_x Contours for BR=0.2 and Re=1000.

Local Nu_x contours for the channel walls with the V–baffle 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 in the corner and the baffle base. The peaks are observed at the impingement areas on the sidewall and walls of the interbaffle cavities.

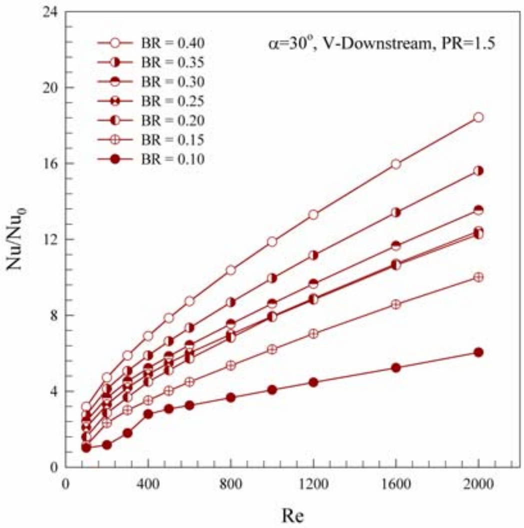


Fig. 6 Variation of Nu/Nu_0 with Reynolds number for various baffle BRs.

The variation of the average Nu/Nu_0 ratio with Reynolds number at different BR values is depicted in Fig. 6. 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 V-baffle with BR = 0.4 is found to be about 18 times over the smooth channel. Thus, the generation of vortex flows from using the V–baffles as well as the role of better fluid mixing and the impingement is the main reason for the augmentation in heat transfer of the channel. The use of the V–baffle with the BR range studied yields heat transfer rate of about 1–18 times higher than the smooth channel with no baffle.

4.4 Pressure loss

Fig. 7 presents the variation of the friction factor ratio, f/f_0 with Reynolds number for various baffle 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 V–baffle leads to considerable increase in friction factor in comparison with the smooth channel with no baffle. The decrease in the BR value gives rise to the reduction of friction factor. The BR=0.4 provides the highest value of friction factor around 300 times above the smooth channel. The f/f_0 value for using the V–baffle is found to be about 1–300 times over the smooth channel depending on the BR and Reynolds number values.

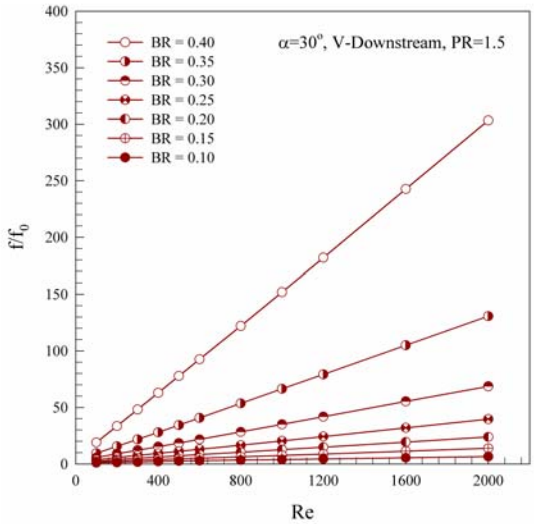


Fig. 7 Variation of f/f_0 with Reynolds number for various baffle BRs.

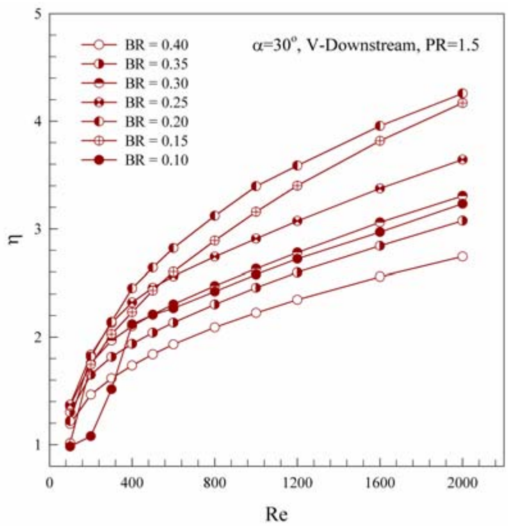


Fig. 8 Comparison of thermal enhancement factor at various BRs.

4.5 Performance evaluation

Fig. 8 exhibits the variation of thermal enhancement factor (η) for air flowing in the baffled channel. In the figure, the enhancement factor for all BR values tends to increase with the rise of Re. The BR=0.2 V–baffle gives the highest enhancement factor of about 4.25 at the highest Re. The enhancement factor of the V–baffle is seen to be above unity for all BRs and varied between 1.0 and 4.25, depending on the BR and Re values.

5. Conclusions

Laminar periodic flow and heat transfer characteristics in a horizontal channel fitted with 30° inline V–downstream baffles mounted periodically on two opposite walls have been investigated numerically. The counter-vortex flows created by using the 30° V–baffles help to induce impingement flows on the sidewall and the wall in the interbaffle cavity leading to drastic increase in heat transfer rate in the channel. The order of enhancement is about 1–18 times the smooth channel for using the V–baffles with BR=0.1–0.4. However, the heat transfer augmentation is associated with enlarged pressure loss ranging from 1 to 300 times above the smooth channel. The highest thermal enhancement factor for the V–baffle with BR=0.2 is found to be about 4.25.

Acknowledgement

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Nomenclature

A	channel wall area, m^2
BR	flow blockage ratio, (b/H)
BLE	baffle leading end
BTE	baffle trailing end
b	baffle height, m
D	hydraulic diameter, (=$2HW/(H+W)$)
f	friction factor
H	channel height, m
h	heat transfer coefficient, W m^{-2} K^{-1}
k	thermal conductivity, W m^{-1} K^{-1}
L	cyclic length of one module, m
Nu	Nusselt number
Pr	Prandtl number
PR	pitch or spacing ratio, L/H
Re	Reynolds number, $(\rho\bar{u}D/\mu)$
T	temperature, K
u_i	velocity in x_i-direction, m s^{-1}
$\bar{u}$	mean velocity in channel, m s^{-1}
W	channel width (=$2H$), m

Greek letter

μ	dynamic viscosity, kg s^{-1} m^{-1}
Γ	thermal diffusivity
α	baffle angle of attack, degree
η	thermal enhancement factor
ρ	density, kg m^{-3}

Subscript

in	inlet
0	smooth channel
w	wall
pp	pumping power