THREE - DIMENSIONAL NUMERICAL STUDY OF LAMINAR HEAT TRANSFER IN A SQUARE DUCT WITH 30° V-ORIFICE

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Abstract

A three-dimensional numerical investigation has been carried out to examine laminar flow and heat transfer characteristics in an isothermal wall square-duct with 30° V-orifice. The computations based on the finite volume method and the SIMPLE algorithm had been implemented. The fluid flow and heat transfer characteristics are presented for Reynolds numbers based on the hydraulic diameter (H) of the duct ranging from 100 to 2000. The simulations were carried out for a square duct with V-orifice pointing downstream and the attack angle, $\alpha = 30°$, mounted on the lower walls of the test duct, for blockage ratio, BR = b/H = 0.05, 0.10, 0.15 and 0.20 and a single pitch, P = 1H. The significant effects of different orifice heights on heat transfer and pressure loss over the smooth wall duct were studied. It was found that the use of V-orifice led to the considerable increase of heat transfer coefficient and frictional factor in comparison with the smooth duct for all Reynolds numbers. In addition, the rise in the blockage ratio resulted in the increase in the Nusselt number and friction factor values. The computational results indicated that the highest heat transfer augmentation of 30° V-orifice was obtained at BR = 0.20 and the maximum thermal enhancement factor, TEF was found to be about 3.7 at BR = 0.20.

Keywords: Periodic flow, Square duct, Laminar flow, Heat transfer, V-orifice

Introduction

One of the commonly used passive heat transfer enhancement technique in single-phase internal flows is the use of ribs and baffles placing periodically in the cooling/heating ducts or duct heat exchangers. 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 fins have been used in cooling or heating systems due to their high thermal loads and decreased dimensions. The cooling or heating air is supplied into the ducts 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. This is because the use of baffles completely makes the change of the flow field and thus the distribution of the local heat transfer coefficient. Periodically mounted baffles in the duct walls help to interrupt hydrodynamic and thermal boundary layers leading to an increase in heat transfer rate. Downstream of each baffle the flow separates, re-circulates, and impinges on the wall. Flow impingement and baffle effect are the main reasons for improvement of heat transfer enhancement in such ducts. In addition, if the baffles are placed at an inclination angle with respect to the axial direction, secondary flows are induced over the duct, resulting in the rise in the heat transfer rate over the duct. Therefore, baffle height, angle of attack and spacing are among the most important parameters used in the design of duct heat exchanger.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 duct with transverse baffles mounted on two opposite walls was found at a Reynolds number below 600 and for such conditions the flow is free of vortex shedding. Cheng and Huang [3] investigated the case of asymmetrical baffles and indicated that the friction factor showed a great dependence on baffle location, especially for a large height of baffle. A numerical investigation of laminar forced convection in a three-dimensional duct 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]. Tsay et al. [5] investigated numerically by using baffles for enhancement of heat transfer in laminar duct flow over two heated blocks mounted on the lower plate. A numerical study of laminar periodic flow and thermal behaviors in a two dimensional duct fitted with staggered diamond–shaped baffles was performed by Sripattanapipat and Promvonge [6]. They reported that the diamond baffle with half apex angle of 5–10° provided slightly better thermal performance than the flat baffle.

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

Flow configuration and mathematical foundation

Geometry and arrangement

The system of interest is a horizontal square duct with 30° V-orifice placed on the lower duct walls in tandem 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 duct at an inlet temperature, T_{in}, and flows over a 30° angled V-orifice where b is the orifice height, H set to 0.05 m, is the duct 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 = H in which L/H is defined as the pitch ratio, Pr = 1. To investigate an arrangement effect of the interaction between V-orifice, the blockage ratio, BR is varied in a range of BR = 0.05-0.20 for α = 30° in the present investigation.

Mathematical foundation

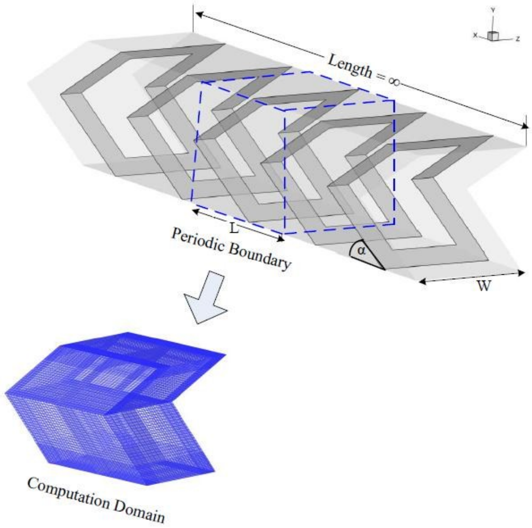

Figure 1 Duct geometry and computational domain of periodic flow.

The numerical model for fluid flow and heat transfer in the square duct 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 duct 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 equation (1) - (3) :

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 upwind differencing scheme, decoupling with the SIMPLE algorithm and solved using a finite volume approach [7]. 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 duct, 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 average Nusselt number can be obtained by

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

The thermal enhancement factor (TEF) is given by

$$TEF = (Nu / Nu_0)/(f / f_0)^{1/3} \tag{9}$$

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

Considering both convergent time and solution precision, the grid system of 120,000 cells was adopted for the current computational model.

Boundary conditions

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

Results and discussion

Verification of smooth square-duct

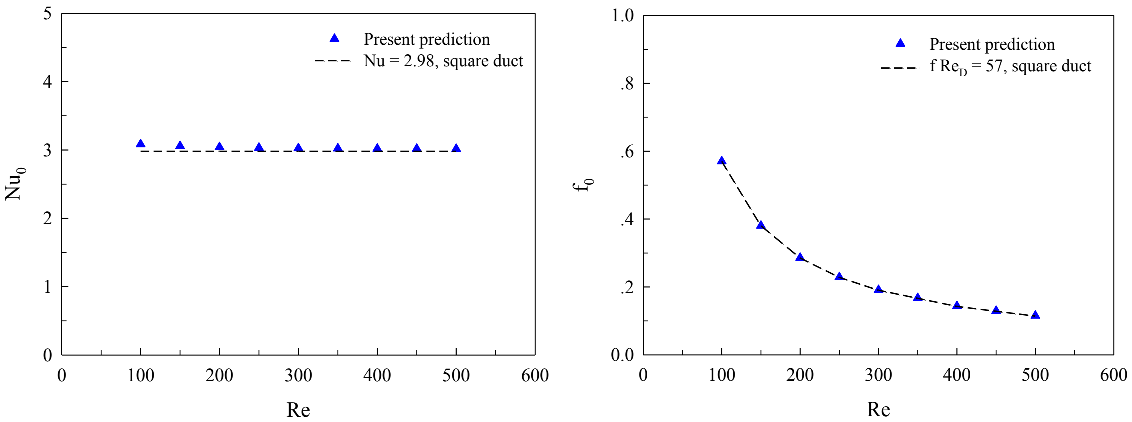

Figure 2 Verification of Nusselt number (left) and friction factor for smooth square duct (right)

Verification of the heat transfer and friction factor of the smooth square duct without baffle is performed by comparing with the previous values under a similar operating condition as shown in Figs. 2(left) and 2(right), respectively. The present numerical smooth square-duct result is found to be in excellent agreement with exact solution values obtained from the open literature [8] for both the Nusselt number and the friction factor, less than ±0.5 % deviation. This provides a strong confidence in further investigation of the duct flow over the baffles. The exact solutions of the Nusselt number and the friction factor for laminar flows over square ducts with constant wall temperature are as follows [8]:

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

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

Flow structure

The flow and vortex coherent structure in the square channel with 30° V-orifice on the walls can be displayed by considering the streamline plots as depicted in Fig. 3. Here the Streamlines in transverse planes for inclined orifice at Re = 2000, BR = 0.20 and PR = 1.

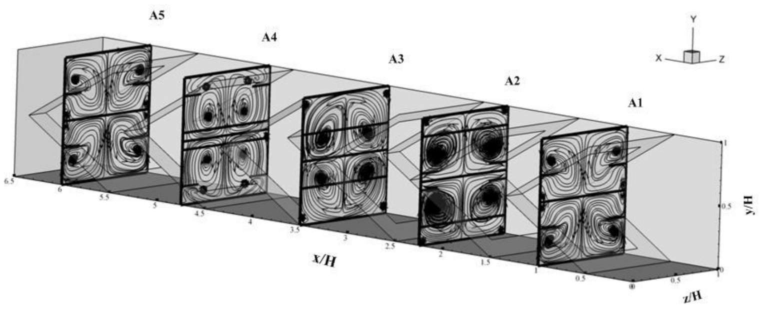

Figure 3 Streamlines in transverse planes for inclined orifice at Re = 2000, BR = 0.20 and PR = 1

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.Considering four main vortex flows of a module with a single pitch, four centers of the main counter-vortex flows (common-vortex flow-up) at the orifice leading end (OLE) plane, plane A1 in Fig. 3, are at about the middle region while four 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 four small corner vortices are appearing in plane A3. Then, both downstream ones make a helical move close together until passing the orifice trailing end (OTE) in plane A4 and the vortex core centers will be drastic counter-rotating flow in plan A5.

Heat transfer

Fig. 4 displays the contour plots of temperature field in transverse planes for Re = 2000 and BR = 0.20. The figure shows that there is a major change in the temperature field over the channel for using the V-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 channel walls with the 30° V-orifice for BR = 0.20 and Re = 2000 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 orifice base. The peaks are observed at the impingement areas on the sidewall and the lower wall.

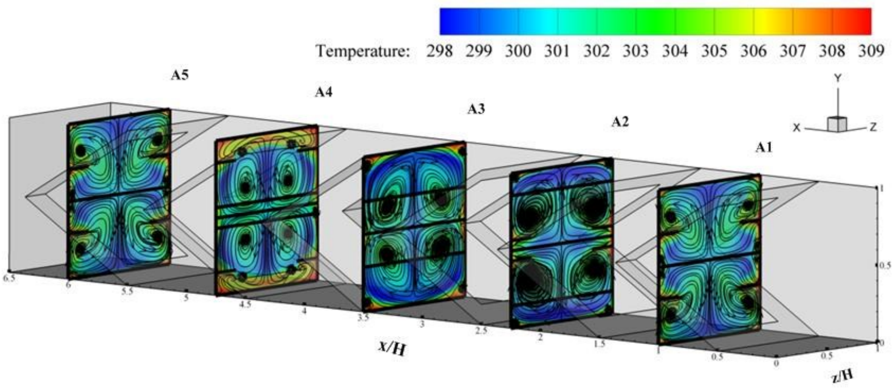

Figure 4 Temperature contours in transverse planes at Re = 2000, BR = 0.20 and PR = 1

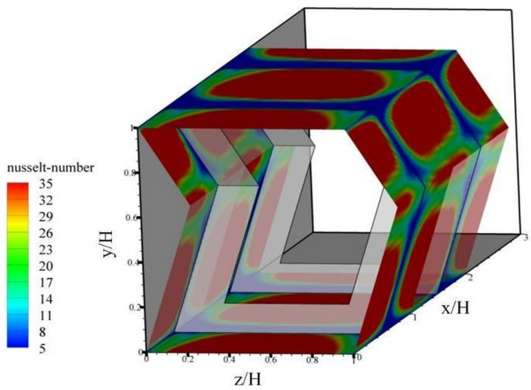

Figure 5 Local Nu_x contours for the channel walls with the V-orifice for BR = 0.20 and Re = 2000

The variation of the average Nu/Nu_0 ratio with Reynolds number for 30° V-orifice at various BRs is depicted in Fig. 6. In the figure, 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 Nu/Nu_0. The scrutiny of Fig. 6 reveals that the use of the 30° V-orifice yields heat transfer rate of about 0.6 - 14 times over that of smooth duct with no orifice, depending on the BR values.

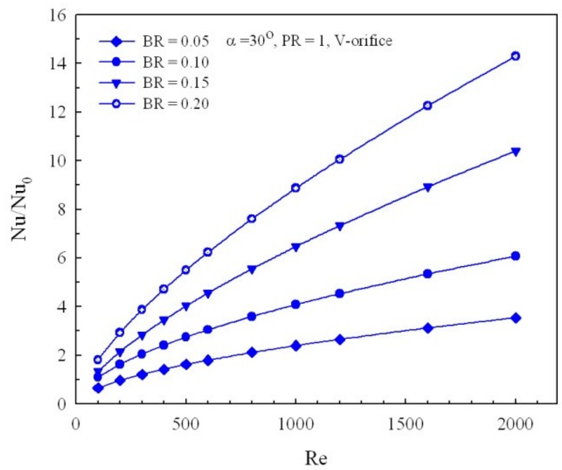

Figure 6 Variation of Nu/Nu_0 with Reynolds number for 30° V-orifice at various BRs.

Pressure loss

Fig. 7 presents the variation of the friction factor ratio, f/f_0 with Reynolds number values for various BRs. In the figure, it is noted that the f/f_0 tends to increase with the rise of Reynolds number and of BR values. The use of the 30° V-orifice leads to an increase in friction factor in comparison with the smooth duct with no orifice. For using the V-orifice, the decrease in the BR value gives rise to the reduction of friction factor. The friction factor ratio value for the 30° V-orifice is found to be higher by about 0.7 - 56 times over the plain duct depending on the BR and Reynolds number values.

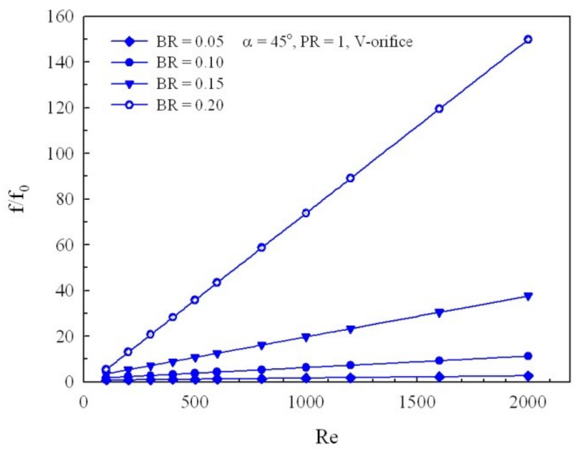

Figure 7 Variation of f/f_0 with Reynolds number for 30° V-orifice at various BRs.

Performance evaluation

Fig. 8 shows the variation of thermal enhancement factor (TEF) for air flowing in the V - orifice square duct. In the figure, the enhancement factor of the 30° V-orifice tends to increase with the rise of Reynolds number. The V-orifice for BR = 0.20 provides the highest value of the enhancement factor. The enhancement factors of the 30° V-orifice are seen to be above unity for BR = 0.05 - 0.20 and vary between 0.7 and 3.7, depending on the BR and Reynolds number values. For the results investigated, the V-orifice with BR = 0.20, Re = 2000 gives the best overall thermal enhancement and one with BR = 0.20 yields slightly lower. This suggests that the 30° V-orifice with BR = 0.05 - 0.20 should be used to obtain higher thermal performance.

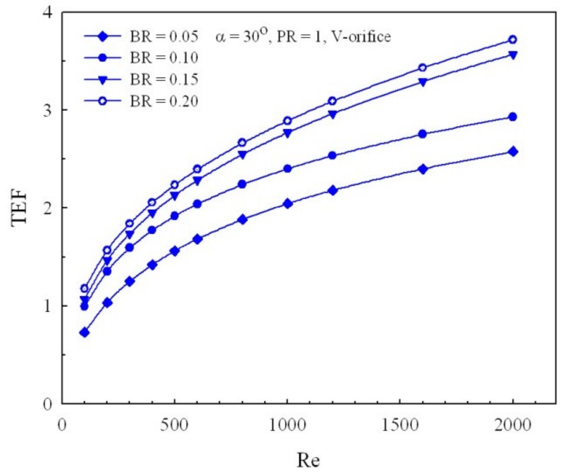

Figure 8 Thermal enhancement factor for 30° V-orifice at various BRs.

Conclusions

Laminar periodic flow and heat transfer characteristics in a square duct fitted with 30° V- orifice elements in tandem on the lower wall is investigated numerically. The order of enhancement is about 0.6 - 14 times the smooth channel for using the 30° V-orifice with BR = 0.05 - 0.20. However, as expected, the heat transfer augmentation is associated with enlarged pressure loss ranging from 0.7 to 56 times above the smooth duct depending on the BR and Reynolds number values. The thermal performance for the 30° V-orifice is around 0.7 to 3.7 times higher than smooth duct, especially for higher Reynolds number. The highest thermal enhancement factor for the 30° V-orifice with BR = 0.20 is found to be about 3.7.

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