Research Article

Effect of Flow Attack Angle of V-Ribs Vortex Generators in a Square Duct on Flow Structure, Heat Transfer, and Performance Improvement

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Received 27 July 2013; Accepted 2 December 2013; Published 2 February 2014

Academic Editor: Joseph Virgone

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A numerical investigation has been carried out to examine the periodic laminar flow and heat transfer characteristics in a three-dimensional isothermal wall square duct with 20° inline V-ribs. The computations are based on the finite volume method, and the SIMPLE algorithm has been implemented. The fluid flow and heat transfer characteristics are presented for Reynolds numbers based on the hydraulic diameter of the square duct ranging from 100 to 2000. To generate main streamwise vortex flows through the tested section, V-ribs with an attack angle of 20° are mounted in tandem with inline arrangement, pointing downstream (V-Downstream) and pointing upstream (V-Upstream) placed on both the upper and lower walls. Effects of different blockage ratio (b/H , BR) with a single pitch ratio (P/H , PR) of 1 on heat transfer, pressure loss, and performance in the ribbed tube are studied. Apparently in each of the main vortex flows, streamwise twisted vortex flows can induce impinging flows on the walls of the interbaffle cavity leading to drastic increase in heat transfer rate over the square duct. In addition, the rise in the V-baffle height results in the increase in the Nusselt number and friction factor values. The computational results show that the optimum thermal enhancement factor is about 4.2 at BR = 0.20 and 0.15 for the V-Downstream and V-Upstream, respectively.

1. Introduction

The increasing necessity for saving energy and material imposed by the diminishing world resources and environmental concerns have prompted the development of more effective heat transfer equipment with improved heat transfer rates. In many industrial systems, heat must be transferred either to input energy into the system or to remove the energy produced in the system. Considering the rapid increase in energy demand world-wide, both reducing energy lost due to ineffective use and enhancement of the energy transfer in the form of heat has become an increasingly important task for the design and operation engineers for such systems.

From the reasons above, the performance improvement investigation into the heat exchanger system has been interested. Laminar flow behaviors in a channel fitted with 90° transverse baffles mounted on two opposite walls with a staggered array were studied by Berner et al. [1] who found that the flow is free of vortex shedding at a Reynolds number

below 600. Webb and Ramadhyani [2] numerically investigated the fluid flow and heat transfer characteristics in a smooth channel attached with staggered baffles. Kelkar and Patankar [3] reported that the heat transfer in a channel with staggered baffle increases with the rise in baffle height and with the decrease in baffle spacing. Lopez et al. [4] carried out a numerical investigation on laminar forced convection in a three-dimensional channel with baffles for periodically fully developed flow and with a uniform heat flux at the top and bottom walls. The effect of a single baffle in the entrance region on thermal behaviors in a channel was studied by Guo and Anand [5]. Ko and Anand [6] experimentally studied on turbulent channel flow with porous baffles. They found that the porous baffles give a higher level of turbulent flow than solid baffles. Mousavi and Hooman [7] numerically examined the heat transfer behavior in the entrance region of a laminar channel flow over staggered baffles and reported that the Prandtl number affects the precise location of the periodically fully developed region.

Promvonge et al. [8] presented a numerical investigation on laminar flow and heat transfer characteristics in a three-dimensional isothermal wall square-channel fitted with inline 45° V-shaped baffles on two opposite walls. Apparently the longitudinal counter-rotating vortex flows created by the V-baffle can induce impingement/attachment flows over the walls resulting in a greater increase in heat transfer over the test channel. They found that the V-baffle with blockage ratio $BR = 0.2$ and pitch ratio $PR = 1.5$ yields the maximum thermal enhancement factor (TEF) about 3.8, whereas the Nu/Nu_0 is around 14 times above the smooth channel at higher Re .

Promvonge and Kwankaomeng [9] also numerically studied periodic laminar flow and heat transfer characteristics in a three-dimensional isothermal wall channel of aspect ratio, $AR = 2$ with 45° staggered V-baffle. They reported that the optimum thermal enhancement factor is around 2.6 at the baffle height of 0.15 times of the channel height for the V-baffle pointing upstream while it is about 2.75 at the baffle height of 0.2 times for the V-baffle pointing downstream.

In [8, 9], the heat transfer augmentation leads to the increase in enlarging pressure. Therefore, the improvement of vortex generators is to reduce the pressure loss that is done by study parameters of the vortex generators, especially, flow attack angle of the vortex generators.

The main aim of the present research is to study numerically the 20° V-ribs configuration effect on heat transfer, pressure loss, and thermal performance in the square duct is presented. This was decided after the literature search that has revealed that no work has been reported on the numerical computation of the flow in 20° V-ribbed square duct using the full form of the Navier-Stokes equations. Regarding the above literature reviews, the flow attack angle of 20° V-ribs leads to significant influence on heat transfer characteristic and helps to reduce the pressure loss. The main objective of the present investigation is to study the influence of the 20° V-rib with rib height ratio $BR = b/H = 0.1-0.3$ at single pitch spacing ratio $PR = 1$ for pointing downstream (V-Downstream) and pointing upstream (V-Upstream) on the flow field, temperature field, heat transfer rate, friction characteristics, and thermal performance.

2. Flow Description

2.1. Rib Geometry and Arrangement. The system of interest is a square duct with a 20° V-rib pair placed on both the upper and lower walls in tandem for inline arrangement and pointing on two different positions, V pointing downstream (V-Downstream) and V pointing upstream (V-Upstream) as shown in Figure 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 [10]. The air enters the square duct at an inlet temperature, T_{in} , and flows over a 20° inline V-rib pair, where b is the rib height, H set to 0.05 m is the square duct hydraulic diameter, and b/H is known as the blockage ratio, $BR = 0.1-0.3$. The axial pitch, L , or distance between the rib cell

is set to $L = H$ in which L/H is defined as the pitch spacing ratio, $PR = 1$.

2.2. Boundary Conditions. Periodic boundaries are used for the inlet and outlet of the flow domain. The constant mass flow rate of air with 300 K ($Pr = 0.7$) is assumed in the inlet 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 rib. The constant temperature of all the duct walls is maintained at 310 K while the rib plate is assumed at adiabatic wall conditions.

2.3. Grid Independent. A grid independence procedure was implemented using the Richardson extrapolation technique over grids with different number of cells. The characteristics of three grids, 87,320, 126,000, and 186,000 cells, are used in the simulation. The variation in Nu and f values for 20° V-ribs at $PR = 1$, $Re = 800$, and $BR = 0.20$ is less than 0.25% when increasing the number of cells from 126,000 to 186,000; hence there is no advantage in increasing the number of cells beyond this value. Considering both convergent time and solution precision, the grid system of 126,000 cells was adopted for the current computational model.

3. Mathematical Foundation

The numerical model for fluid flow and heat transfer in a square duct was developed under the following assumptions.

- (i) Steady three-dimensional fluid flow and heat transfer.
- (ii) The flow is laminar and incompressible.
- (iii) Constant fluid properties.
- (iv) Body forces and viscous dissipation are ignored.
- (v) Negligible radiation heat transfer.

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:

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

Momentum equation:

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

Energy equation:

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

where Γ is the thermal diffusivity and is given by

$$\Gamma = \frac{\mu}{Pr}. \quad (4)$$

FIGURE 1: Square duct geometry and computational domain of periodic flow for 20° V-rib.

Apart from the energy equation discretized by the QUICK scheme, the governing equations were discretized by the second order upwind 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

$$Re = \frac{\rho \bar{u} D}{\mu}. \quad (5)$$

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

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

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

$$Nu_x = \frac{h_x D}{k}. \quad (7)$$

The average Nusselt number can be obtained by

$$Nu = \frac{1}{A} \int Nu_x \partial A. \quad (8)$$

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 is given by [12]

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

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

4. Numerical Results

4.1. Validations Test. In this chapter, the verification is performed to ensure that the computation is reliable. Verification of the heat transfer and friction factor of the square duct without V-rib is performed by comparison with the previous values under similar operating condition as shown in

FIGURE 2: Verification of (a) Nusselt number and (b) friction factor for smooth square duct.

Figures 2(a) and 2(b) for Nu and f , respectively. The present numerical result is found to be in excellent agreement with exact solution values obtained from the open literature [13] for both Nu and f , less than $\pm 0.30\%$ deviation. This provides a strong confidence in further investigation of the duct flow over the V-ribs. The exact solutions of the Nusselt number and the friction factor for laminar flows over smooth square duct with constant wall temperature are as follows [13]:

$$\begin{aligned} Nu_0 &= 2.98, \\ f_0 &= \frac{57}{Re}. \end{aligned} \quad (10)$$

4.2. Flow Topology. The flow configurations in a square duct with V-ribs placed on both the upper and lower walls of square duct can be displayed by considering the streamline plots as depicted in Figures 3 to 5.

In Figures 3(a) and 3(b), the streamline in the transverse plane for 20° V-rib was plotted at $Re = 800$, $BR = 0.20$ and $PR = 1$ for V-Downstream and V-Upstream, respectively. It is found that the 20° V-rib case shows similar flow structure as 45° V-ribs [8] in Figure 4. Promvong et al. [8] also presented the flow structures of 45° V-rib; two counter-rotating vortex pairs caused by the V-rib appear on the lower and upper parts of the module channel in the first plane to last plane. This vortex flow pattern is similar to the V-Upstream case in Figure 3(b) but different in rotating direction. They concluded that the appearance of the vortex flows can help to increase higher heat transfer in the square duct because of highly transporting the fluid from the central core to the near wall regimes.

The plots of streamlines impingement flows on the lower for the V-Downstream and V-Upstream cases are shown in Figures 5(a) and 5(b), respectively. In the figures, it can be noted that impinging jets occur periodically in a region on the lower wall in the rib cavity (as well on the upper wall due to

symmetry). A close look shows that the impinging jet on the wall comes from the helical flows rolling up at the side. The helical vortex flow moves along the rib cavity to the RTE (rib trailing edge) side and rolls up to impinge on the wall. After impingement, the jet splits over the wall and recombines into two helical streams at the nearby rib end to create vortex flows again. The helical pitch length of the main vortex flow is about $5H$ before impingement and becomes shorter (about $3H$) after impingement. This means that the helical vortex flow passes five rib modules from a RTE side to the other RLE (rib leading edge) side before impingement. This behavior is identical on both the upper and lower parts so two streamwise vortices with nonuniform helical pitch are formed throughout the square duct. It can be concluded that vortex flows with nonuniform helical pitches can induce two impingement flows leads to the heat transfer augmentation.

4.3. Heat Transfer Characteristics. Figures 6(a) and 6(b) display the contour plots of temperature field in transverse planes for the V-Downstream and V-Upstream ribs at $Re = 800$ and $BR = 0.2$, respectively. The figures show that there is a major change in the temperature field over the duct for both baffle cases. 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. The higher temperature gradient can be observed where the flow impinges the walls, while the lower one is found at the RTE sidewall area for V-Downstream cases and on the upper and lower walls for V-Upstream case where the temperature in this region is somewhat high. The results show similar trends with 45° V-ribs [8] cases as presented in Figure 7.

Local Nu_x contours of the square duct walls with the V-rib at $Re = 800$, $PR = 1$, and $BR = 0.20$ are presented in Figures 8(a) and 8(b) for V-Downstream and V-Upstream,

FIGURE 3: Streamlines in transverse planes for (a) V-Downstream and (b) V-Upstream of 20° V-rib at $Re = 800$, $BR = 0.20$, and $PR = 1$.

respectively. In these figures, it appears that the V-Downstream rib shows higher Nu_x values over the square duct walls especially on two sidewalls, while the V-Upstream baffle presents higher Nu_x values on both the upper and lower walls of square duct. This indicates a merit of employing the V-discrete baffle over the smooth tube for enhancing heat transfer. For 45° V-Downstream rib, as shown in Figure 9, the higher heat transfer area is found to be at the RTE sidewall similar to 20° V-rib case but of different values of heat transfer rate.

The variation of the average Nu/Nu_0 ratio with BRs at different Reynolds number values is depicted in Figures 10(a) and 10(b) for V-Downstream and V-Upstream, respectively. It is worth noting that the Nu/Nu_0 value tends to increase with the rise of Reynolds number for all cases. The higher BR value results in the increase in the Nu/Nu_0 value. The maximum

heat transfer rate is found to be about 13 and 12 times higher than smooth square duct with no ribs for V-Downstream and V-Upstream cases, respectively. Thus, the generation of vortex flows from using the V-rib as well as the role of better fluid mixing and the impingement is the main reason for the augmentation in heat transfer of the square duct. The use of the V-rib with range studied yields heat transfer rate of about 1–13 times higher than the smooth square duct with no rib, while the 45° V-rib gives the highest heat transfer rate around 1–20 times, reported by Promvonge et al. [8].

4.4. Pressure Loss. Figures 11(a) and 11(b) present the variation of the normalized friction factor, f/f_0 with BR values for various Reynolds number values of 20° V-rib for V-Downstream and V-Upstream, respectively. In the figure, it is found that f/f_0 tends to increase with the rise of Re and BR

FIGURE 4: Streamlines in transverse planes for $\alpha = 45^\circ$ of V-Downstream rib at $Re = 800$, $BR = 0.20$, and $PR = 1$.

FIGURE 5: Streamlines impinging jet on the lower wall for (a) V-Downstream and (b) V-Upstream of 20° V-rib at $Re = 800$, $BR = 0.20$, and $PR = 1$.

values. The f/f_0 value for both V-ribs is found to be about 1–52 times over the smooth square duct while the 45° V-rib gives very enlarged pressure loss of about 1.1–225 times higher than smooth duct [8]. This means that the use of lower flow attack angle helps to reduce the pressure of the system.

4.5. Thermal Enhancement Factor. Figure 12 exhibits the variation of thermal enhancement factor (TEF) for air flowing in the ribbed square duct. In the figure, the enhancement factor of both V-discrete baffles tends to increase with the rise of Re values. The maximum TEF of 20° V-rib is about 4.2 at $BR = 0.20$ and 0.15 for the V-Downstream and V-Upstream, respectively. In comparison with 45° V-rib, on both 20° and 45° V-rib

gives nearly value of the maximum TEF although the 20° V-rib provides lower heat transfer rate.

5. Conclusions

Laminar periodic flow configurations and heat transfer characteristics in a square duct fitted with 20° V-Downstream and V-Upstream rib elements in tandem, inline arrangements placed on both the upper and lower walls of the tested duct have been investigated numerically.

- (i) The vortex flows created by using the 20° V-rib baffles help to induce impingement flows on the walls in the interbaffle cavity leading to a drastic increase in heat

(a)

(b)

FIGURE 6: Contour plot of temperature in transverse plane for (a) V-Downstream and (b) V-Upstream of 20° V-rib at $Re = 800$, $BR = 0.20$, and $PR = 1$.

FIGURE 7: Contour plot of temperature in transverse plane for $\alpha = 45^\circ$ of V-Downstream rib at $Re = 800$, $BR = 0.20$, and $PR = 1$.

FIGURE 8: Contour plot Nu_x for (a) V-Downstream and (b) V-Upstream of 20° V-rib at $Re = 800$, $BR = 0.20$, and $PR = 1$.

transfer in the ribbed duct on both V-Downstream and V-Upstream cases.

- (ii) The order of heat transfer enhancement is 1–13 time higher than smooth duct for using the V-ribs at $BR = 0.10$ – 0.20 with single pitch ratio, $PR = 1$.
- (iii) The pressure loss in the range studied is ranging from 1 to 52 times above the smooth plain duct that lower than 45° flow attack angle of the V-rib.
- (iv) Thermal enhancement factors for both V-ribs are found to be in a range of 1.00–4.20 and the maximum TEF found at $BR = 0.20$ and 0.15 for the V-Downstream and V-Upstream, respectively, at the highest Reynolds number.
- (v) It is noted that the use of 20° V-rib gives lower heat transfer rate than 45° V-rib but can help to reduce the pressure loss of the system.

Nomenclature

A :	Heat transfer area, m^2
BR :	Blockage ratio, (b/H)
b :	Rib height, m
H :	Hydraulic diameter of square duct
f :	Friction factor
GCI:	Grid convergence index
h :	Convective heat transfer coefficient, $W m^{-2} K^{-1}$
k :	Thermal conductivity, $W m^{-1} K^{-1}$
L :	Cyclic length of one cell (or axial pitch length, H), m
Nu :	Nusselt number
p :	Static pressure, Pa
Pr :	Prandtl number
PR :	Pitch ratio, L/H
Re :	Reynolds number, $(\rho \bar{u} D / \mu)$

FIGURE 9: Contour plot of temperature in transverse plane for $\alpha = 45^\circ$ of V-Downstream rib at $Re = 800$, $BR = 0.20$, and $PR = 1$.

FIGURE 10: Nu/Nu_0 versus BRs at various Reynolds number values for (a) V-Downstream and (b) V-Upstream.

RLE: Rib leading edge

RTE: Rib trailing edge

T : Temperature, K

u_i : Velocity in x_i -direction, $m s^{-1}$

$\bar{u}$: Mean velocity in channel, $m s^{-1}$

Greek Letter

μ : Dynamic viscosity, $kg s^{-1} m^{-1}$

Γ : Thermal diffusivity

α : Rib inclination angle or angle of attack, degree

FIGURE 11: f/f_0 versus BRs at various Reynolds number values for (a) V-Downstream and (b) V-Upstream.

FIGURE 12: TEF versus BRs at various Reynolds number values for (a) V-Downstream and (b) V-Upstream.

TEF: Thermal enhancement factor, $(= (\text{Nu}/\text{Nu}_0)/(f/f_0)^{1/3})$
 ρ : Density, kg m^{-3}

Subscript

in: Inlet
 0: Smooth duct
 w : Wall
 pp: Pumping power.

Conflict of Interests

The author declares that there is no conflict of interests regarding the publication of this paper.

Acknowledgments

This research was funded by the College of Industrial Technology (CIT), King Mongkut's University of Technology North Bangkok which is gratefully acknowledged. The author would like to thank Associate Professor Dr. Pongjet Promvong and Dr. Withada Jedsadaratanachai, KMITL for kind suggestions.

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