# Numerical Study of Periodic Fluid Flow, Heat Transfer, Friction Loss and Thermal Performance Characteristics in Square Channels Equipped with V-Ribs with Tail-End Cut

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**ABSTRACT**

***Periodic fluid flow, heat transfer, friction loss and thermal performance characteristics in square channels with V-ribs with tail-end cut were numerically investigated. The V-ribs were installed on a plate which was diagonally placed in a channel. The objective of the present work is to study the effects of the angle of tail-end cut of ribs (β = 0°, 20°, 30°, 45° and 60°) on heat transfer enhancement. The study was carried out under constant heat flux condition for Reynolds numbers (Re) from 3000 to 20,000. The pitch ratio (PR= p/H) and blockage ratio (BR=b/H) of ribs were fixed at 1.0 and 0.15, respectively. As compared to the channel without rib, the one with ribs possessed considerably higher heat transfer and friction loss. At similar conditions, the ribs with larger angle of cut (β) gave lower heat transfer (Nusselt number, Nu) and friction loss (friction factor, f) due to the weaker turbulence and lower resistance to the flow. Among the ribs considered the ones with β of 0° (the ribs without cut) offered the maximum thermal enhancement factor (overall energy performance) of 1.65 at Reynolds number of 3000.***

***Keywords:*** *Periodic flow, Heat transfer, Friction loss, V-rib*

## 1. INTRODUCTION

Several techniques have been employed to improve heat transfer in heat transfer devices for more compact heat transfer system and cost/energy saving. Among the heat transfer enhancement techniques, the use of rib/baffle roughened surfaces is one of the most efficient techniques. In general, the modified surfaces induce secondary flows which effectively enhance heat transfer with an increase of friction loss penalty. Rib/baffle roughened surfaces are widely used in engineering applications especially, for cooling gas turbine blades.

Most of researches relevant to heat transfer enhancement by ribbed/baffled surfaces have focused on optimizing geometric parameter and arrangement of ribs/baffles to yield great heat transfer enhancement with reasonable friction loss. Lau *et al.* [1] and Sara *et al.* [2] reported that the use of solid ribs with and angle of attack of 90° to the axial flow resulted in hot zones in the wake of the ribs due to the recirculation flow with low speed. They showed that as the angle of attack decreased, the hot zones became smaller. The idea was adopted to design V-shaped ribs in several research works. Han *et al.* [3] studied the effects of rib shape, rib angle on heat transfer enhancement in a square channel. It was found that for the same friction loss the 45° V-shaped ribs gave heat transfer than the 90° transverse ribs. Han and Zhang [4] investigated the heat transfer enhancement in a square channel with 60° V-broken ribs in comparison with that of continuous ribs. Their results revealed that the 60° V-broken ribs with pitch to height ratio of 10 and rib height to hydraulic diameter ratio of 0.0625 yielded greater heat transfer enhancement than the continuous ribs. Similarly, SriHarsha *et al.* [5], Tanda [6] and Gupta *et al.* [7] found that V-broken ribs gave significantly higher thermal performance than the continuous ribs. Lau *et al.* [8] reported that the 60° V-shaped ribs with pitch to height ratio of 10 offered a promising thermal performance. Details of effects of 60° V-shaped baffles on thermal and friction characteristics in a channel were reported by Promvonge [9].

As mentioned above, the heat transfer enhancement is accompanied with an increased friction loss. Energy lost caused by friction loss may exceed that gained from increased heat transfer. Therefore, for practical applications, it is needed to determine overall thermal performance as a net benefit by comparison of the heat transfer at constant pumping power. Another approach to improve thermal performance is reducing friction loss due to the flow resistance. Sara *et al.* [10] investigated the heat-transfer enhancement and the corresponding pressure drop over a flat surface in a channel flow due to perforated blocks with different inclination angles. As compared to solid blocks, the inclined perforated blocks possessed considerably better overall thermal performance because of the higher heat transfer enhancement and also the lower pressure drop. The higher heat transfer enhancement was gain because the inclined perforated blocks induced the jet-like flow toward the heated surface in the wake of the blocks while the lower pressure drop was caused by the lower resistance to the flow as the existing holes allowed the flow to pass through the blocks. Yang and Hwang [11] examined numerically heat transfer characteristics in a rectangular duct with slit and solid ribs on one wall and reported that the slit rib performed better than the solid one. Liou *et al.* [12] reported heat transfer enhancement by vortex generators and ribs in first passage of a two pass square channel. The effects of the vortex generators and rib configurations (60°oblique rib, ‘V’ shaped ribs of 60°and 45° and delta wing) were examined. Among the rib configurations considered, it was that the 45°V-shaped rib gave the highest heat transfer enhancement while the delta wing configuration gave the lowest heat transfer augmentation. However, the delta wing configurations offered superior of overall thermal performance because of the lower pressure drops.

The present work aimed to improve the flow conditions and thermal performance by allowing some portions of flow through the spaces on V-shaped ribs in order to reduce the flow resistance. The spaces on the V-shaped ribs were created by partial cutting off tail-end of V-ribs. The investigation was carried on via numerical simulation to examine periodic fluid flow, heat transfer, friction loss and thermal performance characteristics in square channels equipped with tail-end cut V-ribs with different angles of cut (β = 0°, 20°, 30°, 45° and 60°).

## 2. FLOW DESCRIPTION

### 2.1 Geometries and Configurations of V-Ribs and Channel

The system of interest is a square channel with tail-end cut V-ribs. The V-ribs were installed on a plate which was diagonally placed in a channel as shown in Fig. 1. The height of the V-rib (b), the height of the channel (H) and the rib pitch (p) were kept at 7.5, 50 and 50 mm, respectively. These resulted in constant pitch ratio (PR= p/H) and blockage ratio (BR=b/H) of 1.0 and 0.15, respectively. The angle of attack (α) was fixed at 45° while the angle of tail-end cut (β), was varied from 0° to 60° (0°, 20°, 30°, 45° and 60°). The flow was assumed to be under a periodic flow condition in which the velocity field repeats itself from one cell to another. The simulation was performed using the concept of periodically fully developed flow which was introduced by Patankar *et al.* [13], Webb and Ramadhyani [14] and Kelkar and Patankar [15].

### 2.2 Boundary Condition

Periodic boundary condition was applied throughout the flow domain. Constant mass flow rate of air with 300K (Pr=0.7) was assumed in the flow direction rather than constant pressure drop due to periodic flow condition. The inlet and outlet profiles for the velocities were identical. The physical properties of the air were assumed to remain constant at average bulk temperature. Impermeable boundary and no-slip wall conditions were implemented over the square channel walls as well as the ribs. All the square channel walls were subjected to a constant heat flux of 600 W/m$^2$ while the rib surfaces were subjected to an adiabatic wall condition.

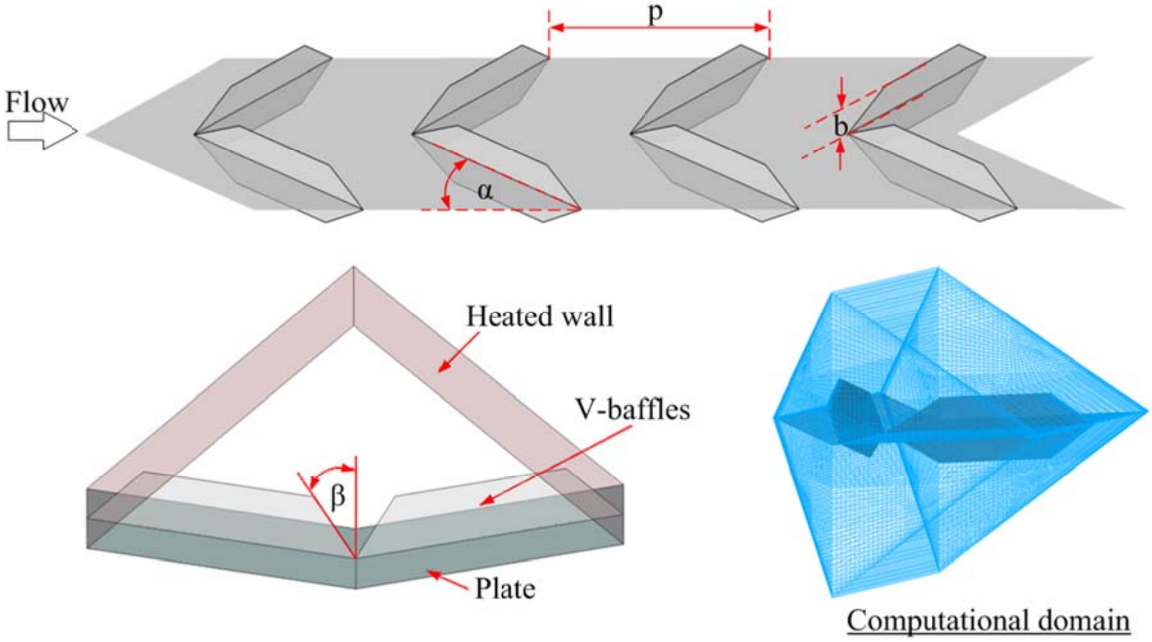


***Fig. 1*** *V-rib geometry and computational domain of the periodic flow.*

## 3. MATHEMATICAL FOUNDATION

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

- Steady three-dimensional fluid flow and heat transfer.
- The flow was turbulent and incompressible.
- Constant fluid properties.
- Body forces and viscous dissipation were ignored.
- Negligible radiation heat transfer.

Based on the above assumptions, the square 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:

*Continuity equation:*

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

*Momentum equation:*

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

Where $\rho$ is the density of fluid, and $u_i$ is a mean component of velocity in the direction $x_i$, p is the pressure, $\mu$ is the dynamic viscosity, and u' is a fluctuating component of velocity.

*Energy equation:*

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

Where $\Gamma$is molecular thermal diffusivity and $\Gamma_t$ is turbulent thermal diffusivity. Their definitions are

$$\Gamma = \mu/\mathrm{Pr} \text{and} \Gamma_t = \mu_t/\mathrm{Pr}_t \tag{4}$$

For the Reynolds-averaged approach to turbulent modeling, the Reynolds stresses. $-\rho\overline{u_i' u_j'}$ in Eq. (2) need to be modeled. The Boussinesq hypothesis relates the Reynolds stresses to the mean velocity gradients can be written as

$$-\rho\overline{u_i' u_j'} = \mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_i}{\partial x_i}\right) - \frac{2}{3}\left(\rho k + \mu_t\frac{\partial u_i}{\partial x_i}\right)\delta_{ij} \tag{5}$$

Where k is the turbulent kinetic energy, defined as $k = \frac{1}{2}\overline{u_i' u_j'}$ and $\delta_{ij}$is a Kronecker delta.

The Boussinesq approach with the computation of the relatively low computational cost is associated with the computation of the turbulent viscosity ($\mu_t$), defined as $\mu_t = \rho c_\mu k^2/\varepsilon$. The RNG $k-\varepsilon$ model under the Boussinesq hypothesis is derived from the instantaneous Navier-Stokes equation using the "renormalization group" (RNG) method. The steady state transport equations are expressed as:

$$\frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left(\alpha_k \mu_{eff}\frac{\partial}{\partial x_j}\right) + G_k - \rho\varepsilon \tag{6}$$

$$\frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_j}\left(\alpha_\varepsilon \mu_{eff}\frac{\partial}{\partial x_j}\right) + G_{1\varepsilon}\frac{\varepsilon}{k}G_k - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} - R_e \tag{7}$$

In the above equations, $\alpha_k$ and $\alpha_\varepsilon$ are the inverse effective Prandtl numbers for k and ε, respectively. $C_{1\varepsilon}$and$C_{2\varepsilon}$ are constants. The effective viscosity $\mu_{eff}$ can beexpressed as

$$\mu_{eff} = \mu + \mu_f = \mu + \rho C_\mu \frac{k^2}{\varepsilon} \tag{8}$$

where$c_\mu$ is a constants and set to 0.0845, derived using the RNG theory.

All the governing equations were discretized by the QUICK numerical scheme, decoupling with the SIMPLE algorithm and solved using a finite volume approach [16]. The solutions were converged when the normalized residual values of for all variables were less than $10^{-5}$.

Four parameters of interest in the present work included Reynolds number, friction factor, Nusselt number and thermal performance enhancement factor. The Reynolds number is defined as

$$Re = \rho u_0 D/\mu \tag{9}$$

The friction factor, f was computed using pressure drop, ΔP across the length of the periodic square duct, L as

$$f = \frac{\left(\frac{\Delta P}{L}\right)D}{2\rho u_0} \tag{10}$$

The local heat transfer was evaluated from the local Nusselt number which can be written as

$$Nu_x = \frac{h_x D}{k} \tag{11}$$

The area-average Nusselt number can be obtained by

$$Nu = \frac{1}{A}\int Nu_x dA \tag{12}$$

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 the same pumping power. TEF can be expressed as

$$TEF = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{Nu}{Nu_0}\right|_{pp} = \left(\frac{Nu}{Nu_0}\right)/\left(\frac{f}{f_0}\right)^{1/3} \tag{13}$$

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

For the present work the use of grid system of 320,000 cells resulted in variation less than for 0.2% Nu and f values. The comparable results were obtained when cell numbers were increased from 320,000 to 800,000. Considering both convergent time and solution precision, the grid system of 320,000 cells was adopted for the current computational model.

## 4. RESULT AND DISCUSSION

### 4.1 Validation

To prove the reliability of the present simulation, the data validation was performed using smooth square channel (the channel without rib). The predicted Nusselt number and friction factor were compared with those obtained from the standard correlations given in open literature [17] as shown below.

*Correlation of Dittus-Boelter,*

$$Nu_0 = 0.023 Re^{0.8} Pr^{0.4} for Re \geq 10{,}000 \tag{14}$$

*Correlation of Blasius,*

$$f_0 = 0.079 Re^{-0.25} for Re \leq 20{,}000 \tag{15}$$

$$f_0 = 0.046 Re^{-0.2} for Re\ b \leq N20{,}000 \tag{16}$$

The comparisons shown in Fig. 2, indicated that the predicted Nusselt number and friction factor were in good agreements with the standard correlations within ±5.3 and ±4.8% maximum deviations, respectively.

### 4.2 Flow Structure and Temperature Contours

Fig. 3 displays the flow structure and contour plots of temperature field in longitudinal planes for (a) β = 0°, (b) β = 20°,(c) β = 30°, (d) β = 45° and (e) β = 60°at α= 45°, BR = 0.15,PR = 1.0 and Re = 3000. The results revealed that as the angle of tail-cut (β) increased, the intensities of vortex and turbulent flows decreased. In addition, the centers of vortices shifted away from channel walls to tail-cut regions. These resulted in poorer fluid mixing near channel walls, which can be observed from the longer bands of thermal boundary layers along the channel walls.

### 4.3 Heat Transfer

Local $Nu_x$ contours for the square channel walls with (a) β = 0, (b) β = 20°,(c) β = 30°, (d) β = 45° and (e) β = 60° at BR=0.15, PR = 1.0 and Re=3000 are presented in Fig. 4. For all cases, heat transfer was intense around the wall middles with the help of ribs by inducing impingement flows toward to the areas. However, heat transfer became poorer around the corners as the hot zones. As the angle of tail-cut (β) increased, the high heat transfer area became smaller. This can be explained that the tail-cuts with larger β possess smaller flow blockage and disturbing effects, thus they allow the fluid flows through the cuts more easily, resulting in weaker turbulence and fluid mixing and consequently less efficient heat transfer. The variation of $Nu/Nu_0$ ratio with β at various Re values is depicted in Fig. 5. At the given Re, $Nu/Nu_0$ ratio decreased with the increase of β. For the range considered, the use of V-ribs resulted in heat enhancement from 3.46 to 5.39 times of those in the smooth square channel. The maximum $Nu/Nu_0$ ratio (5.39) was achieved by using the ribs with β of 0 (the ribs without cut) at Re = 3000.

### 4.4 Pressure loss

The variation of the average $f/f_0$ ratio with β at various Re values is depicted in Fig. 6. At the same Re, $f/f_0$ ratio decreased with the increase of β, as the cut with larger β possessed lower resistance to the flow. For the range considered, the use of V-ribs resulted in increased friction factor from 25.1-37.9 times of those in the smooth square channel. The maximum $f/f_0$ ratio (37.9) was caused by using the ribs with β of 0 (the ribs without cut) at Re = 3000.

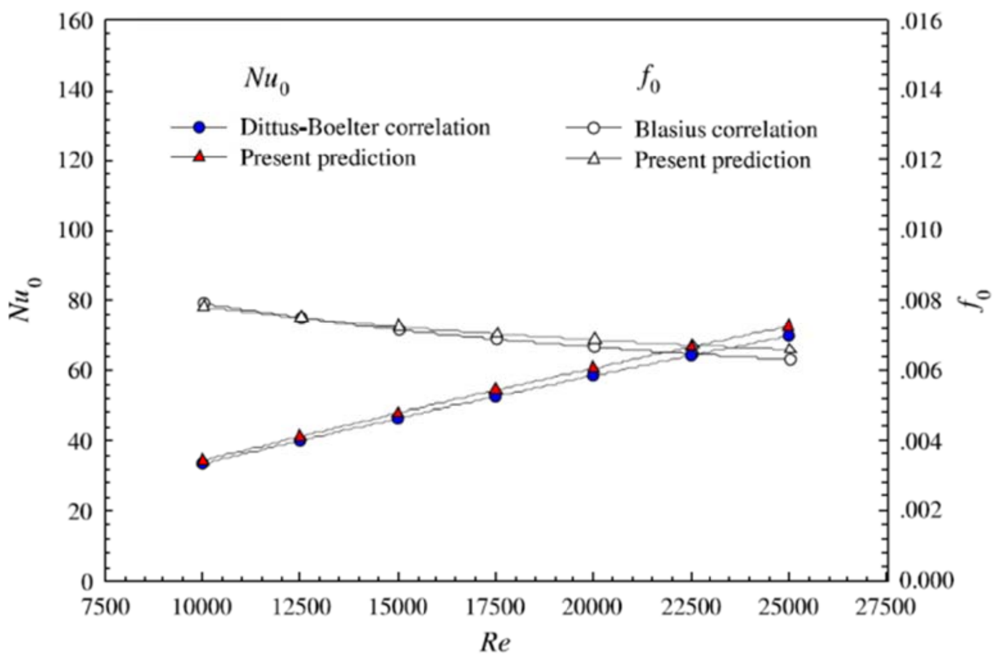


***Fig. 2*** *Verification of (a) Nusselt number and (b) friction factor for smooth square channel.*

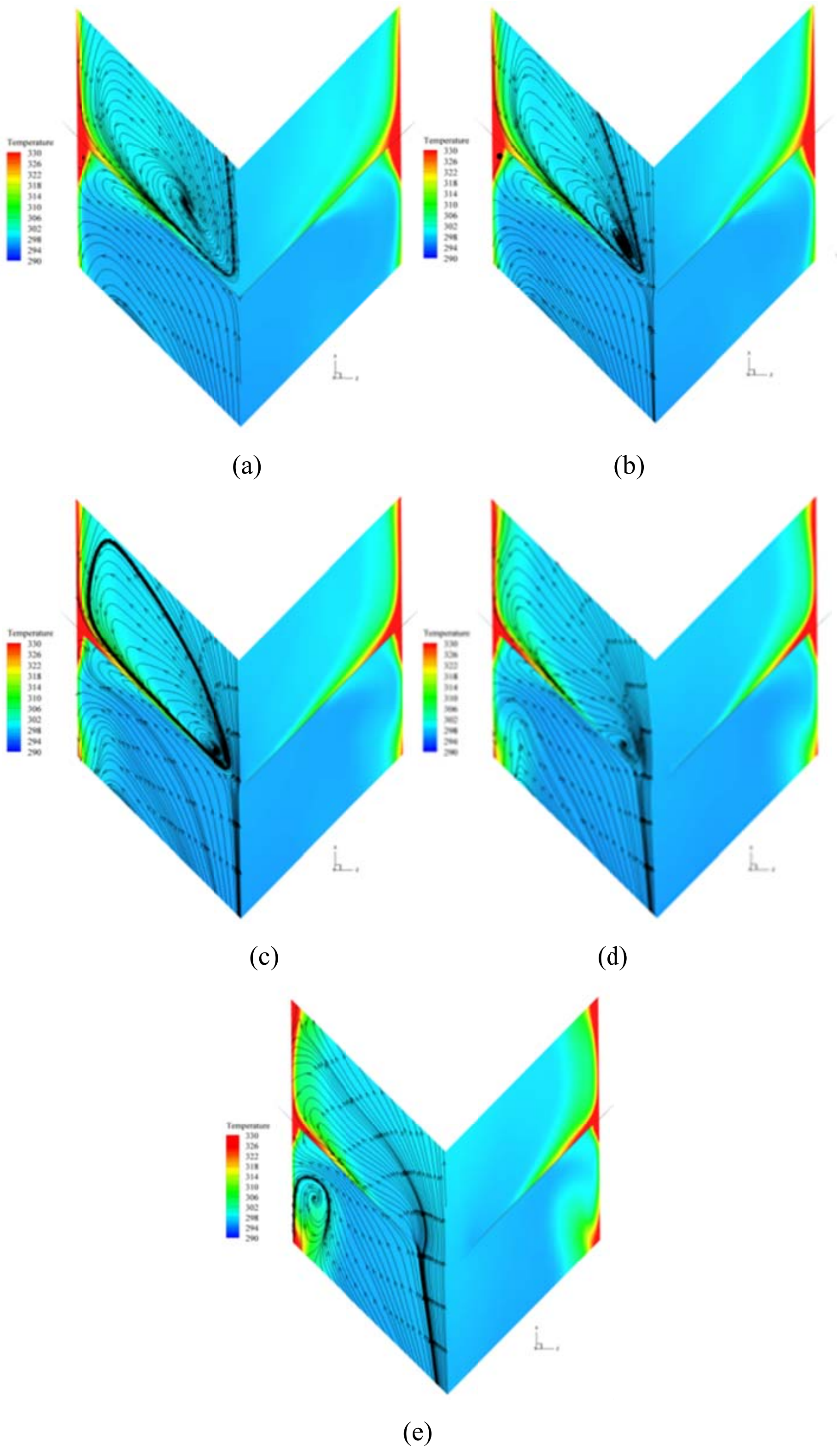


***Fig. 3*** *Flow structure and temperature contours for (a) $\beta = 0^o$, (b) $\beta = 20^o$,(c) $\beta = 30^o$, (d) $\beta = 45^o$ and (e) $\beta = 60^o$ at $\alpha = 45^o$, BR = 0.15,PR = 1.0 and Re = 3000*

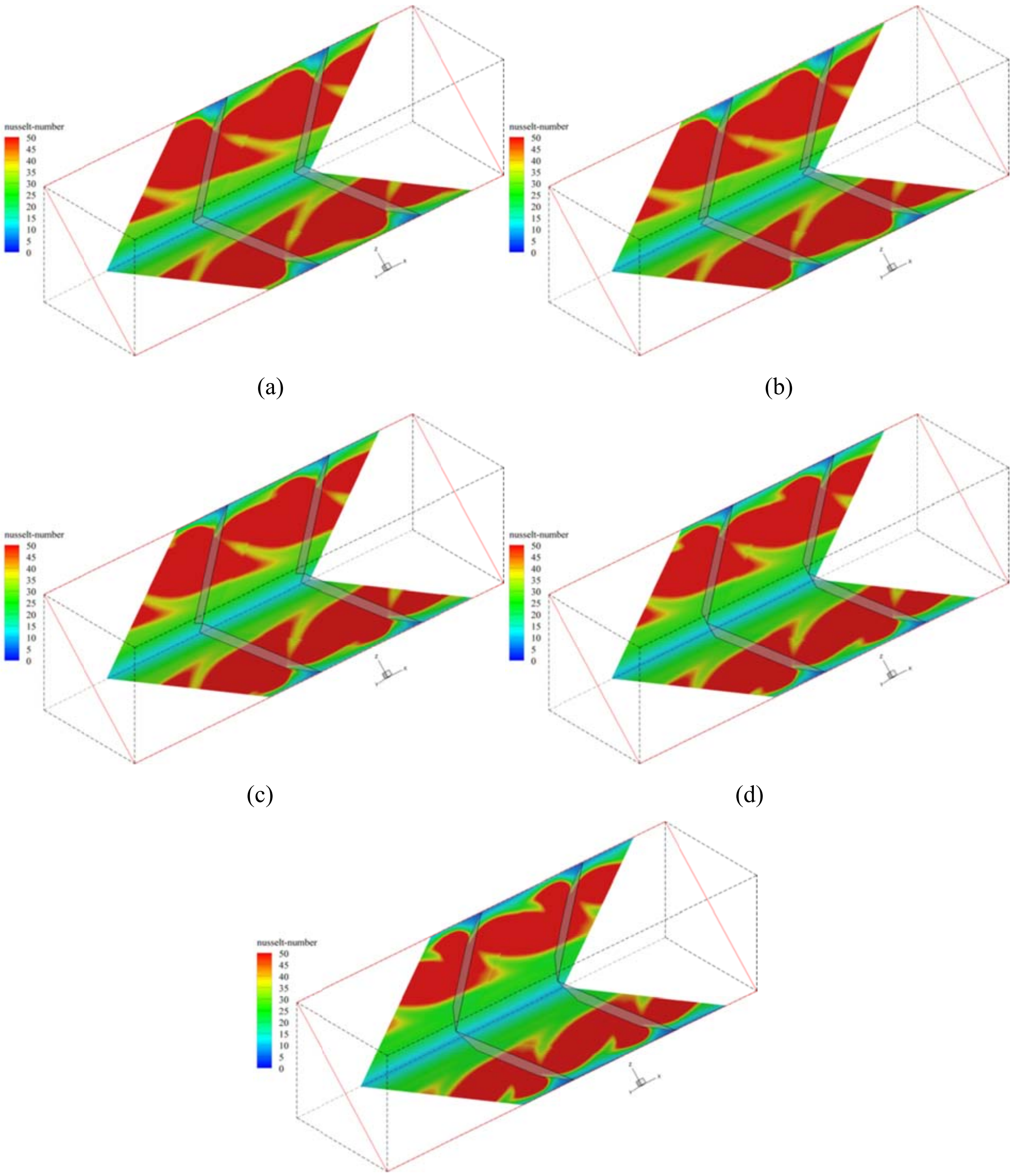


***Fig. 4*** *$Nu_x$ contours for (a) $\beta = 0^o$, (b) $\beta = 20^o$, (c) $\beta = 30^o$, (d) $\beta = 45^o$ and (e) $\beta = 60^o$ at $\alpha = 45^o$, $BR = 0.15$, $PR = 1.0$ and $Re = 3000$*

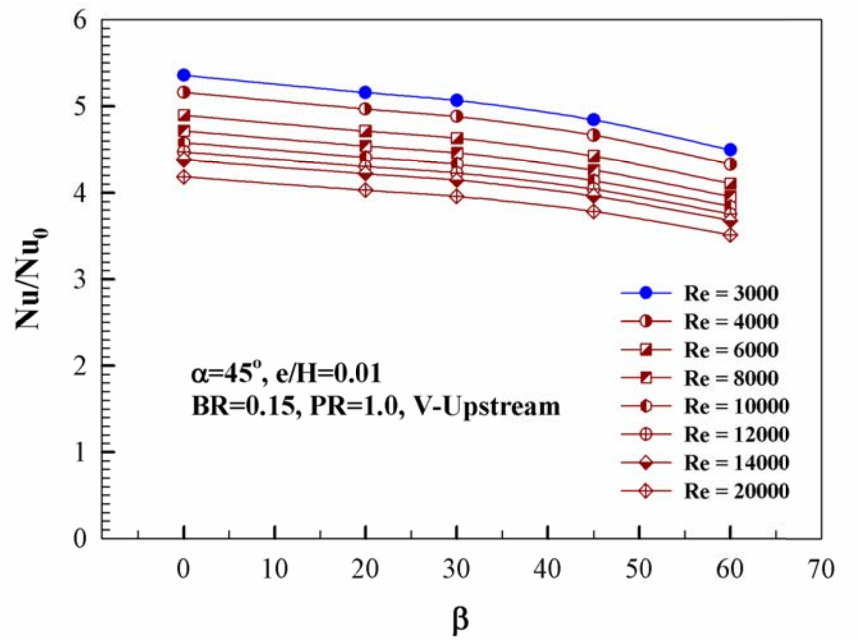


***Fig. 5*** *Variation of $Nu/Nu_0$ with $\beta$ at various Re.*

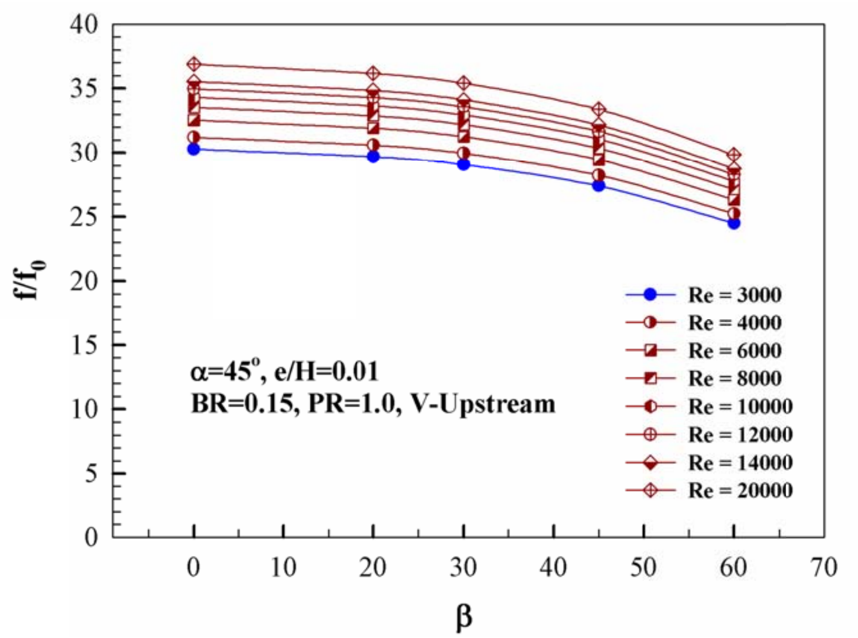


***Fig. 6*** *Variation of $f/f_0$ with $\beta$ at various Re.*

### 4.5 Thermal Performance Evalution

Fig. 7 exhibits the variation of thermal enhancement factor (TEF) at the same pumping power associated with the use of the V-ribs. For all cases, TEFs were higher than unity (varied between 1.10 and 1.65), this indicated the net benefit in the view point of energy saving by using the channel with ribs instead of the smooth one (the one without ribs). Evidently, the TEF as an overall thermal performance decreased with increasing β. In fact, the V-ribs with larger βcaused lower friction loss and gave poorer heat transfer enhancement. The effect of β on TEF signified that the influence of decreasing heat transfer enhancement was more significant than that of decreasing friction loss, when β increased. For range determined, the maximum TEF of 1.65 was achieved by using the ribs with β of $0^{o}$ (the ribs without cut) at Re = 3000.

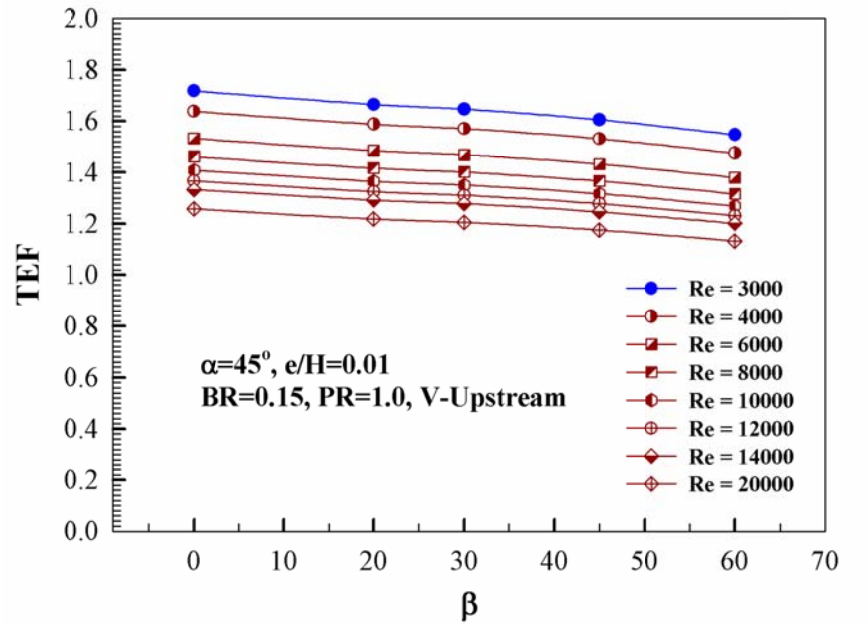


***Fig.** 7 Variation of thermal enhancement factor with β at various Re.*

## 5. CONCLUSIONS

A numerical investigation was conducted to examine periodic fluid flow, heat transfer, friction loss and thermal performance characteristics associated with V-ribs with tail-end cut in square channels. The numerical results revealed that the ribs induced impingement flows toward to the channel middles, resulting in considerable heat transfer enhancement which accompanied with friction loss penalty. However, the increase of angle of tail-end cut (β) resulted in the decrease of heat transfer enhancement and also friction loss. For the range determined, Nusselt number and friction factor associated with the use of the ribs respectively varied from 3.46-5.39 times and 25.1-37.9 times of those in the smooth channel. All ribs gave TEF above unity, indicating the net benefit in the view point of energy saving of the enhancement approach. Among the ribs considered the ones with β of 0° (the ribs without cut) offered the maximum thermal enhancement factor (overall energy performance) of 1.65 at Reynolds number of 3000.


## 6. ACKNOWLEDGEMENTS

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