Mathematical Investigation on Laminar Forced Convection in Square Channel with U-Shaped Rib Turbulator

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ABSTRACT

Periodic laminar flow and heat transfer characteristics in a three–dimensional (3D) isothermal square channel walls with 45° U-shaped rib turbulators, URT, were investigated numerically. The computations are based on the finite volume method (FVM), and the SIMPLE algorithm has been implemented. The fluid flow and heat transfer characteristics are shown for Reynolds numbers based on the hydraulic diameter of the square channel, Re = 100 to 1000. To generate main longitudinal vortices flows through the tested section, URT with an attack angle of 45° are inserted in the tested square channel. Effects of different blockage ratio (b/H, BR) with a single pitch ratio (P/H, PR) of 1 on heat transfer and pressure loss in the channel are studied. It is apparent that in each of the main vortex flows, longitudinal twisted vortex flows can induce impinging flows on a wall of the interbaffle cavity leading to a drastic increase in heat transfer rate over the channel. In addition, the rise in the URT height results in the increase in the Nusselt number and friction factor values. The flow structures with common–flow–down are appeared by using URT. The computational results show that the optimum thermal enhancement factor is around 2.5 at BR = 0.25 and, Re = 1000.

Keywords: *Forced convection; Heat transfer; Laminar flow; Periodic flow; Turbulators;*

1. INTRODUCTION

The required for more efficient and performance improvement of heat exchanger had been done by increase heat transfer rate with turbulators or vortex generators. Many types of turbulators such as rib, fin, groove, baffle, winglet, etc. were installed into tube or channel of heat exchanger to increase higher level both of turbulence vortex flow and degree of impinging jet flows over the walls of the channel. The impinging jet of flows in heat exchanger system can be augmented not only heat transfer rate but also increase in pressure loss. The designs of turbulators and their parameters; shape, attach angle, height and space have been interesting by researchers. Array of V–shaped baffle were used to insert into the channel or tube in heat exchanger to create impinging jet on the wall leads to the increase in heat transfer rate and efficiencies. The higher of blockage ratio (*b*/*H*, *BR*) of V–baffle give the highest heat transfer rate but also provide very enlarge pressure loss. To reduce the pressure loss of the system, the investigation of pitch spacing or pitch ratio (*P*/*H*, *PR*) was reported by Promvonge *et al*. [1], [2]. The results showed that the rise of *PR* leads to the decrease on both pressure loss and heat transfer. Han and Zhang [3] experimentally investigated the effect of 45^o and 60^o parallel ribs and V–shaped ribs in the square channel. They reported that all of the turbulators increase in heat transfer rate with not as much of the rise up flow resistance. Taslim *et al*. [4] experimentally investigated oblique ribs and V–shaped ribs. They showed that the high heat transfer rate cause of the secondary flows that generated by oblique ribs and V–shaped ribs. The experimental examination of the heat transfer coefficient in rib–roughened channels with continuous ribs, interrupted ribs and V–shaped interrupted ribs were reported by Giovanni [5]. The experimental results indicated that the shape of turbulators have effect for flow structure and heat transfer behavior and also referred that the transverse interrupted ribs give the highest heat transfer. The experimental investigation of V–shaped and angled ribs in rotating rectangular channels by Lee *et al*. [6] also confirmed that the V–shaped rib gave better heat transfer enhancement than the angled rib configurations. Fu *et al*. [7] showed the related results. They reported that the V–shaped ribs and the discrete V–shaped ribs provide better heat transfer enhancement. Peng *et al*. [8] studied on both experimentally and numerically for convection heat transfer in channels with different types of ribs; 90^o continuous, 90^o interrupted, 45^o V–shaped continuous, 45^o V–shaped interrupted, 60^o V–shaped continuous and 60^o V–shaped interrupted ribs. The results indicated that the 45^o V–shaped continuous ribs have the highest thermal performance. They also reported the comparison of continuous and interrupted ribs. The V–shaped interrupted ribs give lower heat transfer rate than that from the V-shaped continuous ribs, while the 90^o ribs had the opposite result. Effects of staggered discrete V–apex up (V–Upstream) and down (V–Downstream) rib and transverse staggered discrete ribs on heat transfer were reported by Muluwork [9]. The results showed that the V–down discrete ribs performed better in comparison with V–up and transverse discrete ribs. Momin *et al*. [10] displayed that the 60^o V–ribs give higher heat transfer rate and pressure loss than inclined ribs. The heat transfer of rotating rectangular duct with compound scaled roughness and V–ribs at high rotation numbers was reported by Chang *et al*. [11]. Chang *et al.* [12] also reported the heat transfer and pressure drop in a rectangular channel with the compound roughness of V–shaped ribs and deepened scales. Karwa and Chitoshiya [13] experimentally studied on thermo–hydraulic performance of a solar air heater with 60^o V–down discrete rib roughness on the air flow side of the absorber plate. The enhancement in the thermal efficiency due to the roughness on the absorber plate is found to be 12.5–20%, depending on the airflow rate. Higher enhancement is at the lower flow rate were also presented by Karwa and Chitoshiya [13]. Singh *et al.* [14] studied on thermo-hydraulic performance caused by flow–attack–angle in V–down rib with a gap in a rectangular duct of solar air heater. They concluded that the thermal enhancement factor was found to be highest at the attach angle of 60^o.

From *Ref.* [2], the use of the inclined baffle placed on the channel walls gave the higher *TEF* in comparison with the smooth square channel, but the installation may be difficult similarly as V–baffle. Therefore, the improvements of the baffle turbulators are focused on more comfort and also suitable for applying with equipments and industrials.

The objectives of the present work are as follows:

- Numerical study the effect of 45^o U–shaped rib turbulators, *URT*, on heat transfer, pressure loss, thermal enhancement factor and flow structure in isothermal wall square channel.
- The effects of *URT* height to diameter ratio (blockage ratio, b/H, BR = 0.05 – 0.30) with single pitch to diameter ratio (P/H, PR) of 1 are studied for Re = 100 – 1000.

2. FLOW DESCRIPTION

2.1 URT Geometry and Arrangement

The square channel geometry with *URT* refers from Promvonge *et al*. [2]. Fig. 1 shows the square channel with the *URT* inserted which the configuration of the *URT* similar to inclined baffle with clamping plate. The flow under consideration is expected to make a periodic flow condition in which the velocity field and thermal profile repeat itself from one cell to another. The periodically fully developed flow concepts and its solution procedure have been described in *Ref.* [15]. The air enters the channel at an inlet temperature, T_{in}, and flows over 45° *URT* where b is the *URT* height, H which is the channel height and is set to 0.05. The 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 spacing ratio, PR = 1. To investigate an effect of the flow blockage ratio, BR is varied in a range of BR = 0.05 – 0.30 with single pitch ratio for α = 45° in the present investigation.

2.2 Boundary Conditions

From *Ref.* [2], inlet and outlet of the computational domain are periodic boundaries. The 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 channel walls as well as the *URT*. The constant temperature of all the square channel walls is maintained at 310 K while the *URT* are assumed at adiabatic wall conditions.

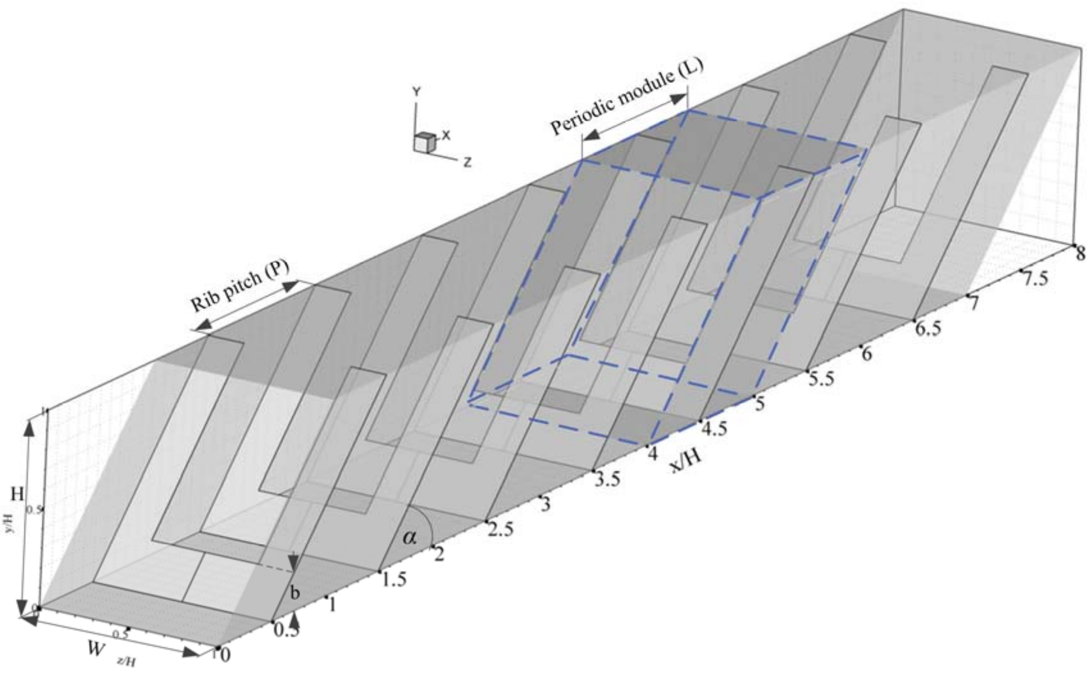


***Fig. 1** Square channel geometry with 45^o of URT inserted.*

3. MATHEMATICAL FOUNDATIONS

From *Refs*. [2], the numerical model for fluid flow and heat transfer in 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 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

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

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 [16]. 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 the 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

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

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 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.

Considering both convergent time and solution precision, the grid system of 80,400 cells was adopted for the current computational model.

4. RESULTS AND DISCUSSIONS

4.1 Verification of Smooth Channel

Validations of the heat transfer and friction factor of the smooth channel without *URT* are studied by comparison with the previous values under a similar operating condition. The current numerical smooth channel results are found to be in excellent agreement of which less than ±0.25 % deviation with exact solution values obtained from the open literature [18] for both the Nusselt number and the friction factor.

4.2 Flow Configuration

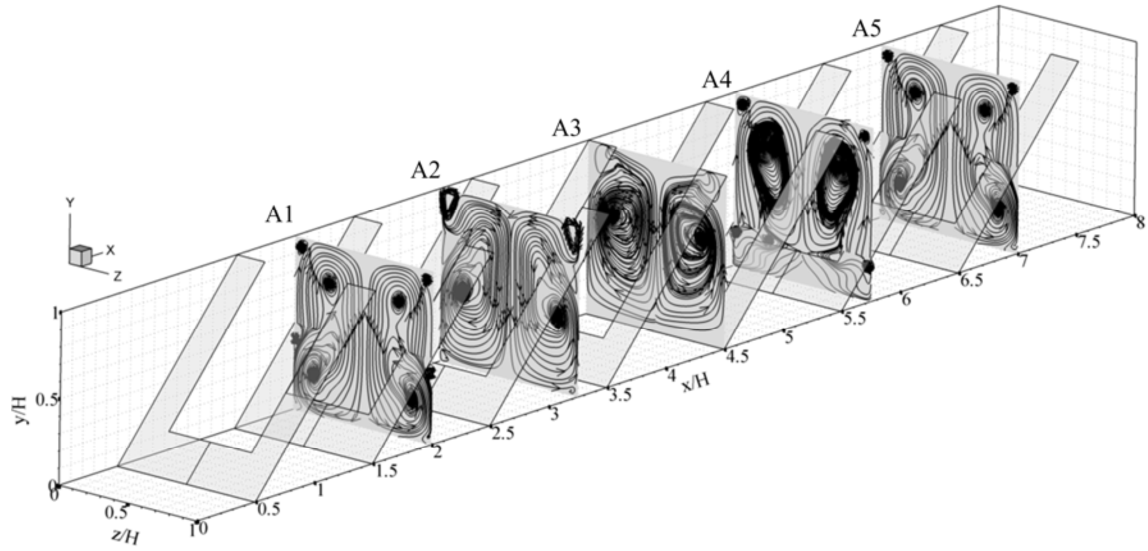
The flow configurations in square channel with *URT* inserted in testing channel can be displayed by considering the streamline plots as depicted in Fig. 2 and 3 for streamlines in transverse planes and streamlines impinging jets on the channel walls, respectively.

Fig. 2*a* and *b* present the streamlines in transverse planes and the details for each plane, respectively, for *BR* = 0.20 and *Re* = 800. It is visible that there are two longitudinal main vortex flows in the tested channel. Similar two counter–rotating vortices visible on left and right parts by reason of *URT* symmetry as depicted in planes A1 to A5. It can be concluded that the *URT* case produces two counter–vortex flows having a rotating direction down to the channel wall, called “common–flow–down”. The appearance of the vortex flows can help to increase higher heat transfer in the tested channel because of highly transporting the fluid from the vortex core to the wall regions.

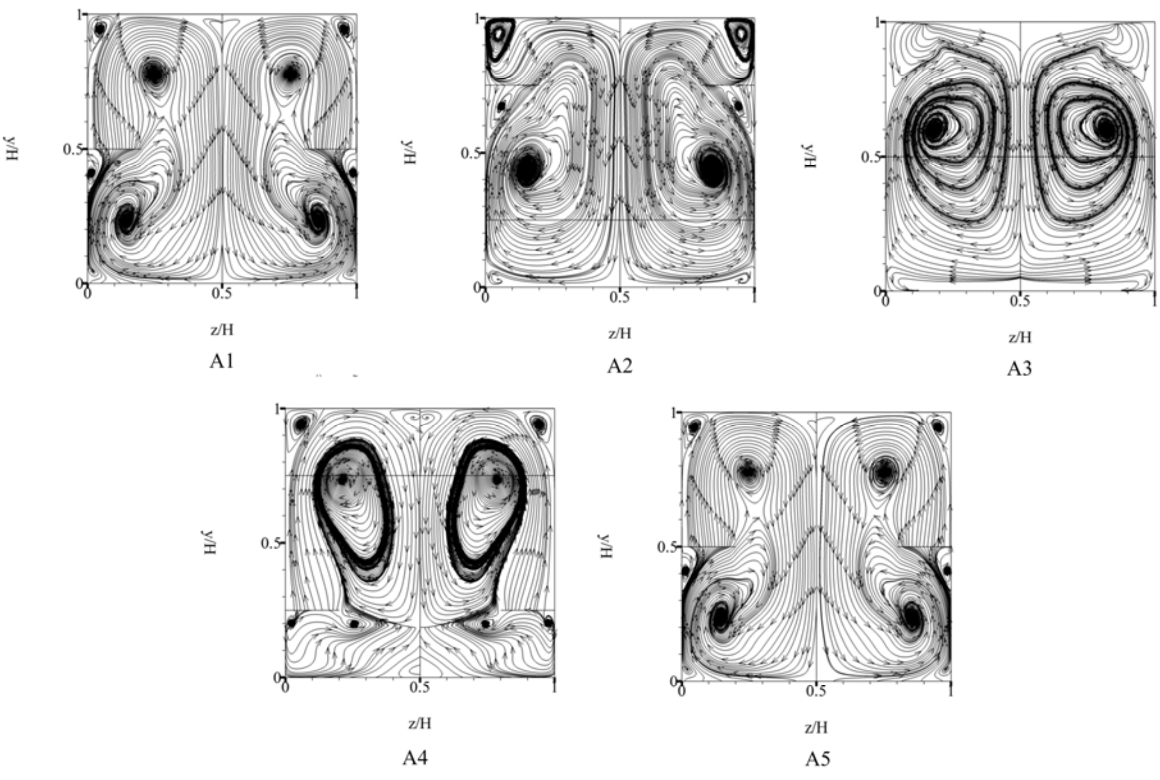
The schemes of streamlines impingement flows on the channel walls are shown in Fig. 3*a*, and *b*, respectively, for sidewall and lower wall. It is visible that impinging jets happen periodically in the *URT* spacing. The helical pitch length of the main vortex flow is about 2*H* before impingement and becomes shorter (about 1*H*) after impingement. This behavior is identical on both the left and right parts.

4.3 Heat Transfer Behavior

The contours temperature in transverse planes for *URT* at *BR* = 0.20 and *Re* = 800 is presented in Fig. 4. It is found that the *URT* lead to good mixing between the temperature in the core flow and near the wall regimes as the distributions of the contours temperature in comparison with the smooth square channel. Local Nu_x contours of the channel walls with *URT* at *Re* = 800 and *BR*=0.20 are presented in Fig. 5. The maximum heat transfer areas are found on the sidewalls and the lower wall of the channel due to the impinging jets of the fluid flow. These trends are found similar for all *BR*s and Reynolds number. The variations of the average Nu/Nu_0 ratio with Reynolds number at different *BR*s is depicted in Fig. 6 and 7, for average values and on each side of the channel walls, respectively. It appears that the Nu/Nu_0 value tends to increase with the rise of Reynolds number for all cases. The increasing *BR* value results in the augmenting Nu/Nu_0 value. Thus, the generation of longitudinal vortex flows from using the *URT* as well as the role of better fluid mixing and the impingement is the main cause for the augmentation in heat transfer of the tested channel. The use of the *URT* with range studied gives heat transfer rate of about 1 – 14 times higher than the plain channel with no *URT*. In addition, the sidewalls provide a higher heat transfer rate than the upper and lower walls of the channel for *BR* > 0.2 while $0.05 \leq$ BR ≤ 0.2, the heat transfer rate for all sides perform nearly values.

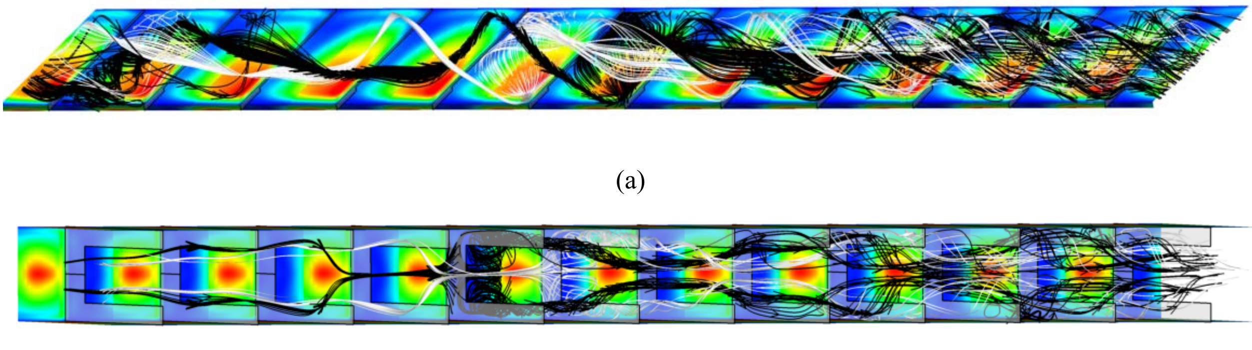


(a)



(b)

Fig. 2 *(a) Streamlines in transverse planes and (b) detail of each plane for URT at Re=800 and BR=0.20.*



(a)

(b)

Fig. 3 *Streamlines of impinging jet on the channel walls (a) sidewall and (b) lower wall at BR=0.20 and Re = 800.*

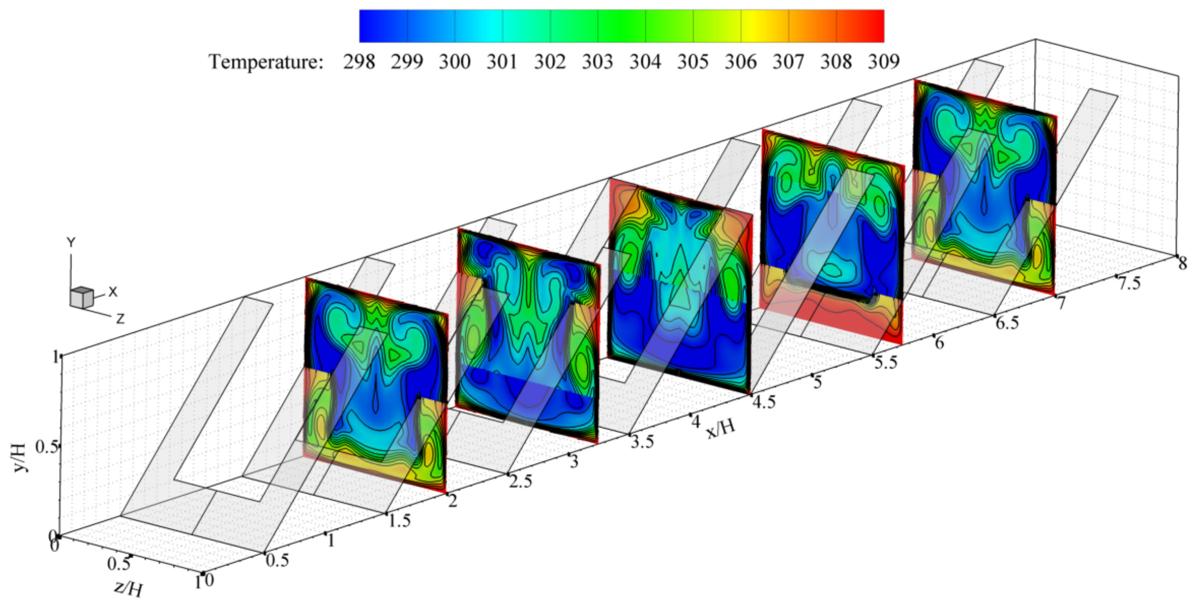


Fig. 4 *Contours temperature in transverse planes for URT at BR=0.20 and Re = 800.*

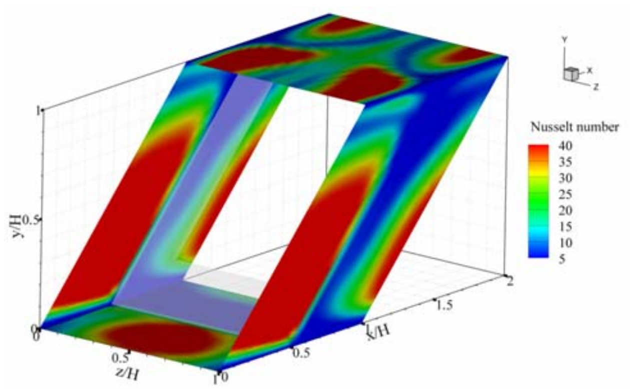


Fig. 5 *Nu_x Contours at the channel walls for URT at BR=0.20 and Re = 800.*

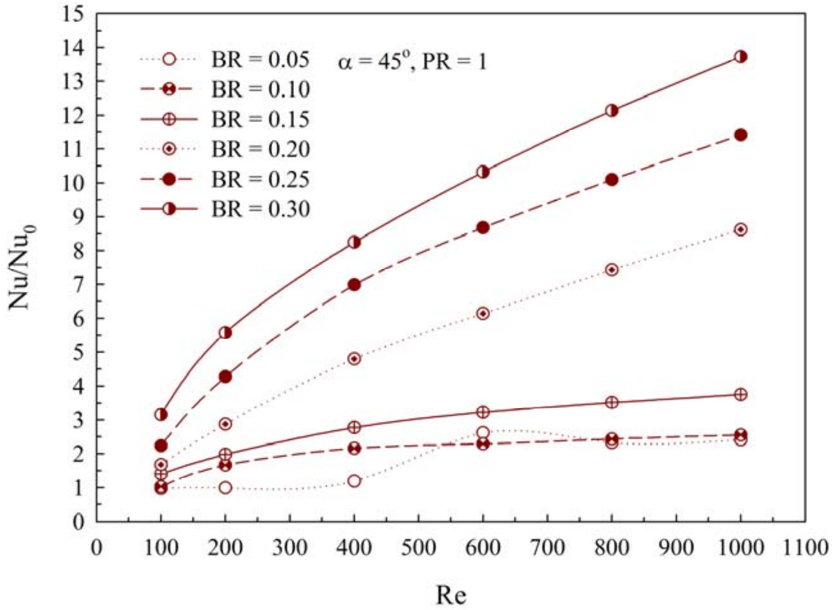


Fig. 6 *Variations of Nu/Nu_0 with Reynolds number for URT at various BRs.*

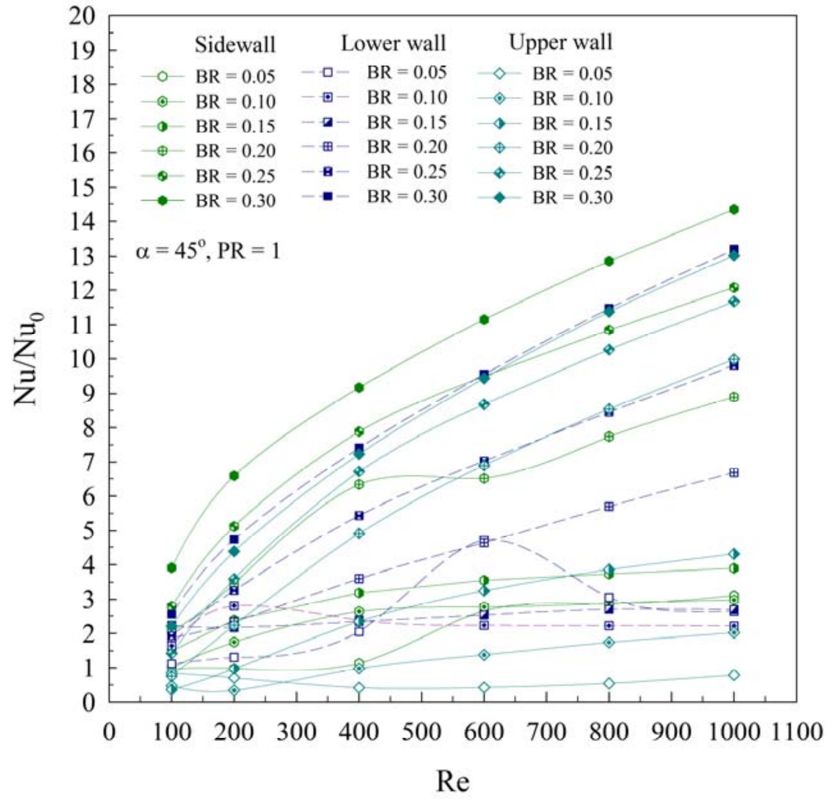


Fig. 7 *Variations of Nu/Nu_0 with Reynolds number for various BRs on each side of the channel walls.*

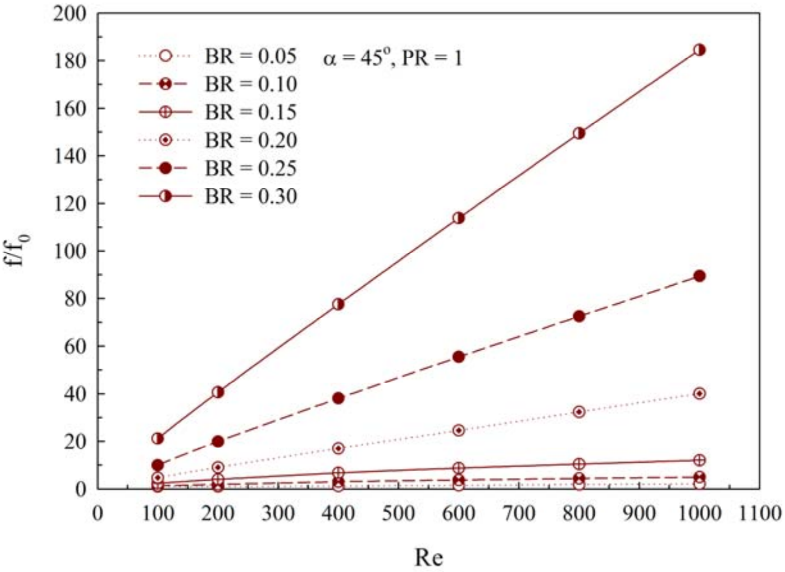


Fig. 8 *Variations of f/f_0 with Reynolds number for URT at various BRs.*

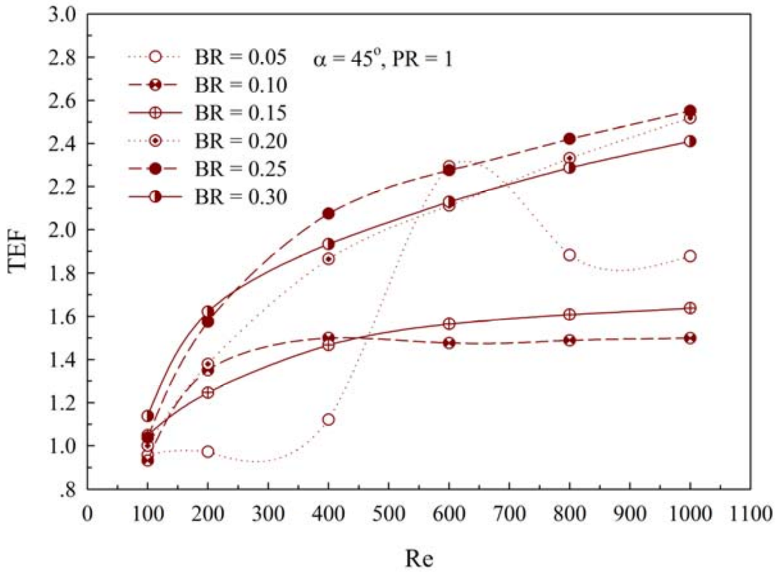


Fig. 9 *Comparison of thermal enhancement factor for URT at various BRs.*

4.4 Pressure Loss

Fig. 8 displays the variations of the friction factor, f/f_0 with Reynolds number values for various *BR*s of *URT*. In the figure, it is visible that the f/f_0 tends to increase with the rise of *Re* and *BR* values. The use of the *URT* leads to a considerable increase in friction factor in comparison to the plain channel with no *URT*. The f/f_0 values for all cases are found enlarge around 1–185 times over the smooth channel depending on the *BR* and Reynolds number values. This means that the height of *URT* is the key for the augmentation for the pressure loss of the system.

4.5 Thermal Enhancement Factor

Fig. 9 exhibits the variations of thermal enhancement factor (*TEF*) for air flowing in the *URT* channel. The enhancement factor of both *URT* tends to increase with the rise of *Re* and *BR* values except for *BR* = 0.05. All of the *UTR* cases provide the highest enhancement factor at the highest *Re*, *Re* = 1000. The enhancement factors are seen to be above unity for all cases and vary between 1.0 and 2.5, depending on the *BR* and *Re* values. The maximum *TEF* is found in the case of *BR* = 0.25.

5. CONCLUSIONS

Fully developed periodic laminar flow configurations and heat transfer characteristics in square channel fitted with 45° *URT* inserted in the tested channel have been investigated numerically.

- The longitudinal vortex flows created by using the 45° *URT* cases help to induce impingement flows on the walls leading to a drastic increase in heat transfer in the square channel.
- The augmentation on heat transfer is about 1 – 14 times higher than a smooth square channel for using *URT* in the range studied, *BR* = 0.05 – 0.30 with single pitch ratio, *PR* = 1 and *Re* = 100 – 1000.
- The pressure loss in the range studied varied from 1 to 185 times above the smooth plain channel.
- Thermal enhancement factors for both the *URT* cases are found to be in a range of 1.0–2.5 and the maximum *TEF* found at *BR* = 0.25 at the highest *Re*.
- The installation for *URT* in the square channel is more convenient than the baffle turbulators placing on the upper and lower walls of the square channel and also suitable for applying with industries.

NOMENCLATURE

BR	flow blockage ratio, (*b*/*H*)
b	*URT* height, m
D_h	hydraulic diameter of square channel
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, *D*), m
Nu	Nusselt number
p	static pressure, Pa
Pr	Prandtl number
PR	pitch or spacing ratio, *L*/*D*
Re	Reynolds number, $(\rho \bar{u} D/\mu)$
T	temperature, K
URT	U–shaped rib turbulators
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	*URT* inclination angle or angle of attack, degree
TEF	thermal enhancement factor, $(=(Nu/Nu_0)/(f/f_0)^{1/3})$
ρ	density, kg m^{-3}
Subscript	
in	inlet
0	smooth channel
w	wall
pp	pumping power

6. ACKNOWLEDGEMENTS

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REFERENCES

[1] P. Promvonge, W. Jedsadaratanachai, S. Kwankaomeng and C. Thianpong, 3D simulation of laminar flow and heat transfer in V–baffled square channel, Int. Commun. Heat Mass Transfer 39 (2012) 85–93.
[2] P. Promvonge, W. Jedsadaratanachai and S. Kwankaomeng, Numerical study of laminar flow and heat transfer in square channel with 30° inline angled baffle turbulators, Applied Thermal Engineering 30 (2010) 1292–1303.
[3] J.C. Han and Y.M. Zhang, High performance heat transfer ducts with parallel and V-shaped interrupted ribs, Int. J. Heat Mass Transfer 35 (2) (1992) 513-523.
[4] M.E. Taslim, T. Li and D.M. Kercher, Experimental heat transfer and friction in channels roughened with angled, V-shaped, and discrete ribs on two opposite walls, Journal of Turbomachinery 8 (1) (1996) 20-28.

[5] T. Giovanni, Heat transfer in rectangular channels with transverse and V-shaped interrupted ribs, Int. J. Heat Mass Transfer 47 (2004) 229-243.
[6] E. Lee, L.M. Wright and J.C. Han, Heat transfer in rotating rectangular channels with V-shaped and angled ribs, Journal of Thermophysics and Heat Transfer 19 (1) (2005) 48-56.
[7] W.L. Fu, L.M. Wright and J.C. Han, Heat transfer in two-pass rotating rectangular channels (AR = 2:1) with discrete ribs, Journal of Thermophysics and Heat Transfer 20 (3) (2006) 569-582.
[8] W. Peng, P.X. Jiang , Y. P. Wang and B. Y. Wei, Experimental and numerical investigation of convection heat transfer in channels with different types of ribs, Applied Thermal Engineering, 31 (2011) 2702-2708.
[9] K.B. Muluwork, Investigations on fluid flow and heat transfer in roughened absorber solar heaters, Ph.D. Dissertation; IIT, Roorkee (2000).
[10] A.M.E. Momin, J.S. Saini and S.C. Solanki, Heat transfer and friction in solar air heater duct with V-shaped rib roughness on absorber plate, Int. J. Heat Mass Transfer 45(2002) 3383–3396.
[11] S. W. Chang, T. L. Yang, T. M. Liou and G. F. Hong, Heat transfer of rotating rectangular duct with compound scaled roughness and V-ribs at high rotation numbers, International Journal of Thermal Sciences 48 (2009) 174–187.
[12] S.W. Chang, T.M. Liou, K.F. Chiang, G.F. Hong, Heat transfer and pressure drop in rectangular channel with compound roughness of V-shaped ribs and deepened scales, International Journal of Heat and Mass Transfer, 51 (2008) 457–468.
[13] R. Karwa and G. Chitoshiya, Performance study of solar air heater having v-down discrete ribs on absorber plate, Energy, 55 (2013) 939-955.
[14] S. Singh, S. Chander and J.S. Saini, Investigations on thermo-hydraulic performance due to flow-attack-angle in V-down rib with gap in a rectangular duct of solar air heater, Applied Energy, 97 (2012) 907–912.
[15] S.V. Patankar, C.H. Liu, E.M. Sparrow, Fully developed flow and heat transfer in ducts having streamwise-periodic variations of cross–sectional area, ASME J. Heat Transfer, 99 (1977) 180-186.
[16] S.V. Patankar, Numerical heat transfer and fluid flow, McGraw-Hill, New York, 1980.
[17] P.J. Roache, Verification and Validation in Computational Science and Engineering, Hermosa Publishers, Albuquerque, NM, 1998, ISBN 0913478083.
[18] F. Incropera, P.D. Dewitt, Introduction to heat transfer, 5rd edition John Wiley & Sons Inc, 2006.

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