Thermal Performance Improvement in Square Duct with Inclined Ribs Inserted Diagonally

Amnart Boonloi

Department of Mechanical Engineering Technology, College of Industrial Technology,
King Mongkut's University of Technology North Bangkok, Bangkok 10800, Thailand.
Email: amnartb [AT] kmutnb.ac.th

ABSTRACT—*In the current work, a numerical simulation is performed to investigate periodic laminar flow and heat transfer characteristics in a square channel with diagonally 45° full−ribbed and discrete−ribbed tape inserts. The rib height (b/H, BR) is in a range of 0.1 to 0.3 with a single pitch ratio of 1.0. The uniform heat-flux condition is applied to the duct walls. The computations based on the finite volume method and the SIMPLE algorithm has been implemented. Air is used as the test fluid with the airflow rate in terms of Reynolds number ranging from 4000 to 20,000. The numerical results are obtained from using the RNG k−ε turbulence model. Effects of ribbed−tape inserts on heat transfer and pressure loss in the tube are examined and their results are also compared with the smooth tube. The numerical results reveal that the tube with inclined−ribbed tape inserts gives the heat transfer rate and pressure loss higher than the smooth tube. The full−ribbed tape performs slightly better than the discrete one. In addition, the TEF is found to be about 1 – 1.25 and 1 –1.22 for the full−ribbed and the discrete−ribbed tapes, respectively.*

Keywords— heat transfer, square−duct, full rib, discrete rib

1. INTRODUCTION

Rib is a type of turbulator which always use to improve heat transfer rate and thermal performance in a compact heat exchanger. The rib turbulator can change the flow structure and create the longitudinal vortex flow though the heat exchanger. The vortex flow, which generated by the rib turbulator help to a better mixing of the fluid flow between the core of the vortex flow and near the wall regime, results in the increase in heat transfer rate in the system.

Many researches have been reported the use of rib turbulator for improving the heating system. For examples, Wang et al. [1] presented the thermal performance in a mini−channel with discrete double−inclined ribs by using numerical method. They reported that the discrete double−inclined ribs can produce the longitudinal vortex flow though the test section that help to augmenting heat transfer rate. Yongsiri et al. [2] numerical studied the effect of the flow attack angle for inclined detached−ribs on heat transfer, pressure loss and thermal performance. They concluded that the ribs with the flow attach angle of 60^o and 120^o give the highest on both heat transfer rate and thermal performance at high Reynolds number. Aharwal et al. [3] investigated a solar air heater duct with inclined discrete ribs on heat transfer and friction loss. They presented that the heat transfer and friction loss are found to be around 2.83 and 3.6 times higher than the smooth duct, respectively. Tang and Zhu [4] reported the influences of discrete rib on flow structure and heat transfer in a rectangular channel. They summarized that the heat transfer rate of the inclined broken rib is about 160−230% in comparison with the other types. Satta et al. [5] experimental studied the flow configurations and heat transfer characteristics in a rectangular channel with 45^o angled ribs on the channel walls. They found that the ribs whicg placed on two walls provide higher heat transfer rate than only one wall. Tanda[6] presented the effects of pitch−to−height ratios of 45^o angled rib turbulators. Gao and Sunden [7] studied the thermal performance in a rectangular duct with rib turbulators. They pointed out that the secondary flow which created by rib turbulators results in the increase in heat transfer rate. Dutta and Hossain[8] experimental study the heat transfer augmentation in a rectangular channel with two inclined baffles. They found that the two inclined baffles give higher heat transfer rate than the single rib turbulators. Yadav and Bhagira[9] presented that the transverse rib which placed in a solar air heater channel help to improve the thermal performance, especially, at case of P/e = 10.71 and e/D = 0.042. Lu and Jiang [10] investigated on both experimental and numerical methods for heat transfer in a rectangular channel with angled ribs. They conclude that the maximum thermal performance is found in the case of 20^o ribs with 1 – 2 mm spacing ribs. Wong et al. [11] claimed that the secondary flow and vertical motion are the causes of heat transfer enhancement in a channel with rib turbulators.

As the previous works, almost of the investigations reported the use of rib and baffle turbulators placed on the channel walls. The effect of the inclined rib turbulators inserted diagonally in the square channel has not been reported. In the current work, the numerical investigations on turbulent forced convection, heat transfer and thermal performance in the square channel with rib turbulators are presented. The influences of rib configuration and blockage ratio, b/H, BR = 0.10, 0.15, 0.20, 0.25 and 0.30 are investigated for Reynolds number, Re = 4000 – 20,000.

2. COMPUTATIONAL MODEL AND NUMERICAL METHOD

2.1 Computational domain

Two types of 45° inclined ribs insert diagonal in a square channel heat exchanger are depicted in Figures 1a and b for full type and discrete type, respectively. The periodic concept which reported by Patankar [12], is applied to the computational domain. The air enters the channel at an inlet temperature, T_{in}, and flows over the 45° inclined ribs where D is the hydraulic diameter of the channel, $D = H$ set to 0.05 m. The axial pitch, P or the distance between the inclined rib is set to $L = H$ in which L/H is defined as the pitch spacing ratio, PR = 1 and the width of the duct, $W = H$, The blockage ratio, b/H, BR is varied in a range of BR = 0.1 – 0.3 for α = 45° in the present investigation.

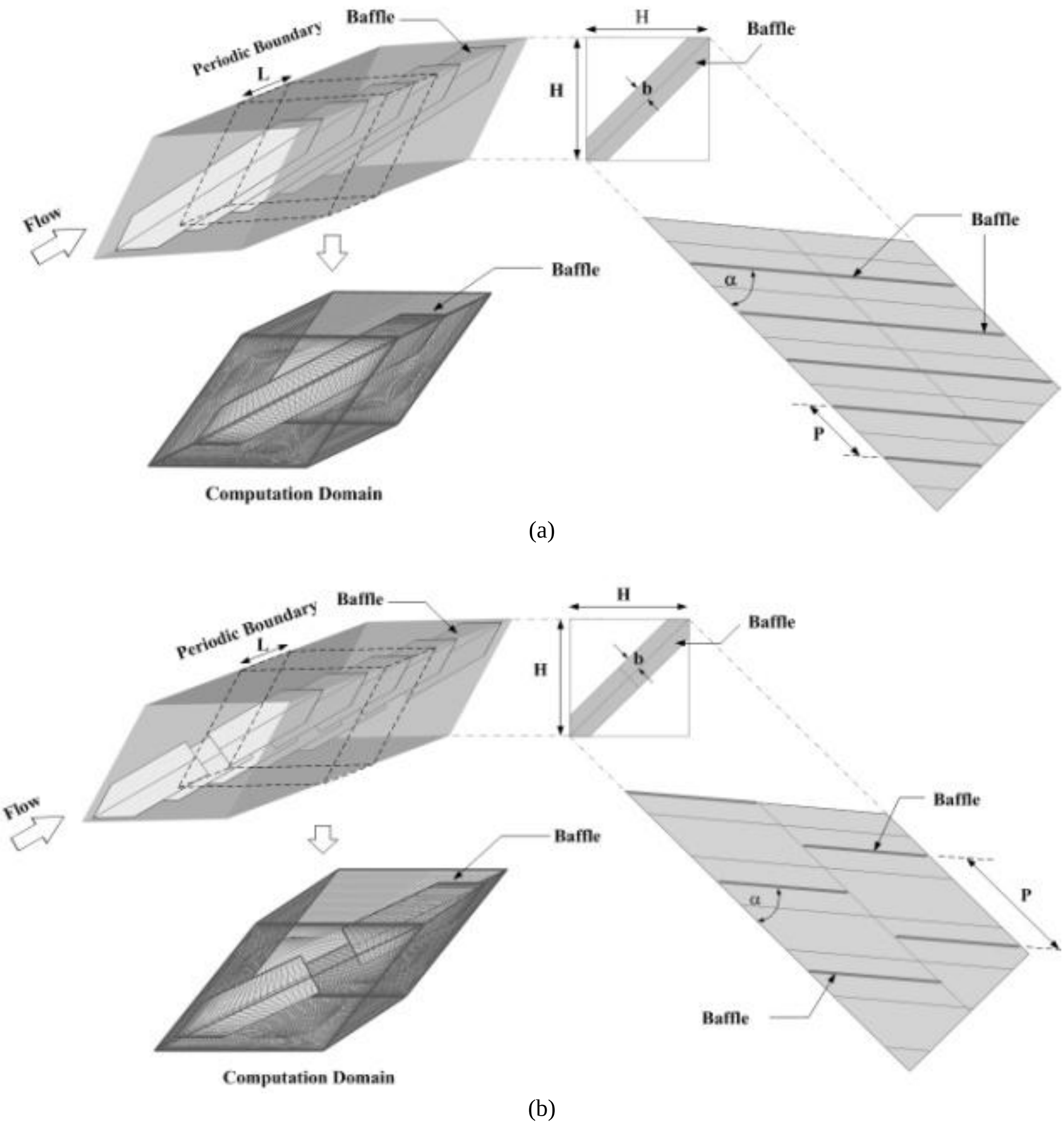


Figure 1: Physical structure and computational domain of 45° inclined rib insert diagonally for (a) full type and (b) discrete type.

2.2 Boundary conditions

The boundary conditions of the current computational domain are as follows:

- The inlet and outlet zones of the computational domain are set as periodic boundary.
- 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 physical properties of the air are assumed to remain constant at an average bulk temperature.
- Impermeable boundary and no–slip wall conditions are implemented over the duct walls and ribs.
- The constant heat flux of all the channel walls is maintained at 600 W/m^{2}while the rib plate is assumed at adiabatic wall conditions

2.3Grid system

Four sets of grid cells; 120,000, 240,000, 360,000 and 480,000 are used to test the grid system. The case of BR = 0.2, PR = 1 and Re = 8000 is selected to this check. As the results, the variations on both the Nusselt number and friction factor are found to be around 0.74% and 0.54%, respectively, when increasing the cell from 360,000 to 480,000. The number of grid cells about 360,000 is used for the current computational domain.

2.4Verification of the smooth channel

Verifications of the heat transfer and friction factor of the smooth square channel with no rib are done by comparing with the values from correlations. The numerical results on both heat transfer rate and friction loss are found to be in excellent agreement with the correlation solutions for the Nusselt number and friction factor within ±3.3 and ±4.8%, respectively.

3. MATHEMATICAL FOUNDATIONS

The channel flow is governed by the Navier–Stokes equations and energy equation. In the Cartesian tensor system these equations can be written as follows:

Continuity equation:

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

Navier–Stokes equations:

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

Energy equation:

$$\frac{\partial}{\partial x_1}(\rho u_i T)=\frac{\partial}{\partial x_j}\left((\Gamma+\Gamma_t)\frac{\partial T}{\partial x_j}\right) \tag{3}$$

where Γ is the thermal diffusivity, and Γ_t is turbulent thermal diffusivity as given by

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

Apart from the energy equation discretized by the QUICK scheme, the governing equations are discretized by the power law scheme, coupling pressure–velocity with the SIMPLE algorithm and solved using the finite volume approach. The solutions are considered to be converged when the normalized residual values are less than 10^{-5} for all variables, but less than 10^{-9} only for the energy equation. In Equation. 2, the Reynolds stress terms, $-\rho\overline{u_i^{'}u_j^{'}}$, is defined by using the Boussinesq hypothesis that relates the Reynolds stresses to the mean velocity gradients as seen in the equation below:

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

where$k$ is the turbulent kinetic energy, as defined by $k=\frac{1}{2}\overline{u_i^{'}u_j^{'}}$ and δ_{ij} is the Kronecker delta, μ_t is the turbulent viscosity as $\mu_t=\rho c_\mu k^2/\varepsilon$. The RNG k–ε turbulence model is derived from the instantaneous Navier–Stokes equations using the "renormalization group" (RNG) method. The steady state transport equations for closure of the problem are expressed as

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

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

In the above equations, α_k and α_ε are the inverse effective Prandtl numbers *k* and *ε*, respectively. $C_{1\varepsilon}$ and $C_{2\varepsilon}$ are constants. The effective viscosity μ_{eff} is written by

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

Where C_μ is a constant and set to 0.0845, derived from the RNG theory.

There are four parameters of interest in the present work; Reynolds number, friction factor, Nusselt number and thermal performance enhancement factor. The Reynolds number is defined as

$$Re = \rho \bar{u} D / \mu \quad (9)$$

The friction factor, *f*, is computed by pressure drop, Δ*p*, across the length of the periodic channel, *L*, as

$$f = \frac{(\Delta p / L) D}{\frac{1}{2}\rho \bar{u}^2} \quad (10)$$

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

$$Nu_x = \frac{h_x D}{k_a} \quad (11)$$

The area–average Nusselt number can be obtained by

$$Nu = \frac{1}{A}\int Nu_x dA \quad (12)$$

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

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

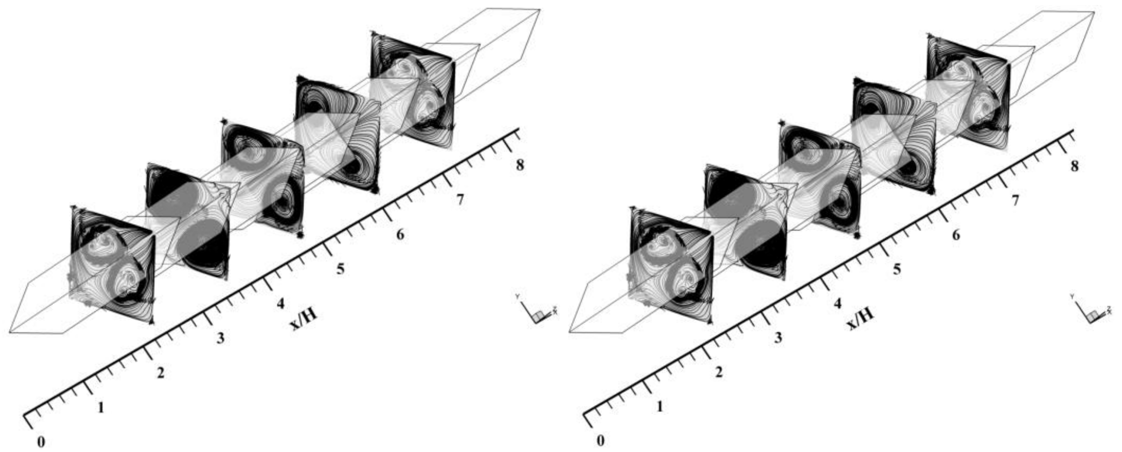
whereNu_0 and f_0 stand for Nusselt number and friction factor for the smooth square duct, respectively.

4. RESULTS AND DISCUSSION

4.1 Flow configuration and heat transfer characteristic

The flow configuration in the square channel with the 45^o inclined rib is presented in term of streamlines in transverse planes, while the heat transfer characteristics are displayed in forms of temperature contours and Nusselt number contours over the channel walls.

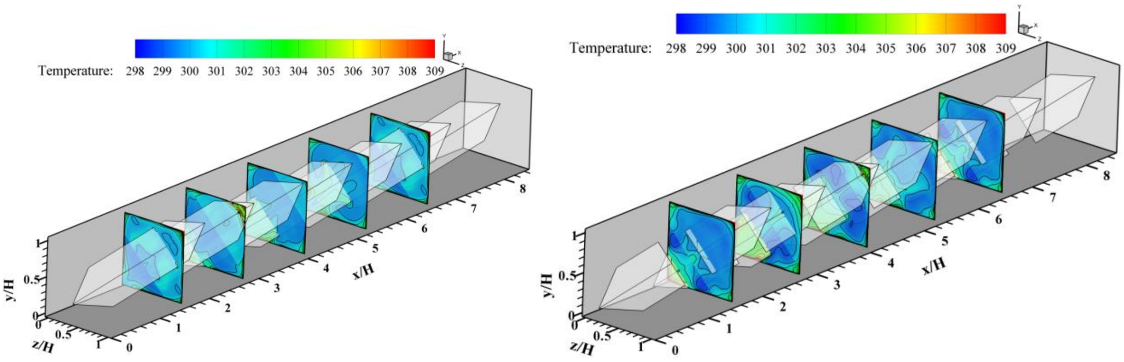
The streamlines in transverse planes for the 45^o inclined rib with full type and discrete type are displayed in the Figure 2*a* and *b*, respectively, at *BR* = 0.2, *PR* = 1 and *Re* = 8000. As the Figures, on both types of inclined ribs can generate the vortex flow though the test section. The two main vortex flows and small vortices at the corner of the square channel are found for all cases.



(a) (b)

Figure 2: Streamlines in transverse planes for 45^o inclined rib at *Re* = 8000, *PR* = 1 and *BR* = 0.2 for (a) full type and (b) discrete type.

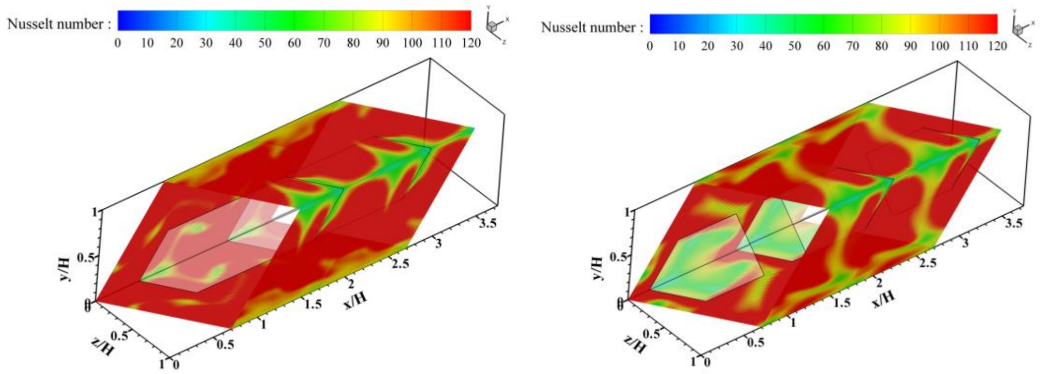
Figures 3*a* and *b* display temperature contours in transverse planes for inclined ribs at *Re* = 8000, *BR* = 0.2 and *PR* = 1 of the full type and discrete type, respectively. The temperature distributions in all cases are seem to be similar. The use of 45^{o} inclined rib provides better mixing of the fluid flow between the core of the vortex and near the wall regime when comparing with the smooth channel.



(a) (b)

Figure 3: Temperature contours in transverse planes for 45^{o} inclined rib at *Re* = 8000, *PR* = 1 and *BR* = 0.2 for (a) full type and (b) discrete type.

Figures 4*a* and *b* report the Nusselt number contours over the channel walls of the 45^{o} inclined ribs for full and discrete types, respectively, at similar case; *Re* = 8000, *BR* = 0.2 and *PR* = 1. For both types, the 45^{o} inclined ribs show higher heat transfer rate than the smooth channel with no rib turbulators. In addition, the full type performs a slightly higher Nusselt number than the discrete type.



(a) (b)

Figure 4: Nu_x contours 45^{o} inclined rib at *Re* = 8000, *PR* = 1 and *BR* = 0.2 for (a) full type and (b) discrete type.

4.2 Performance evaluation

The performance evaluation for the 45^{o} inclined ribs is divided into three parts; heat transfer, pressure loss and performance, which presented in forms of Nusselt number ratio (Nu/Nu_0), friction factor ratio (f/f_0) and thermal enhancement factor (*TEF*), respectively.

The variations of the Nu/Nu_0, f/f_0 and *TEF* with the Reynolds number are presented as Figures 5, 6 and 7, respectively. In general, the Nu/Nu_0 tends to decrease with the rise of Reynolds number for all cases. The rise of *BR* leads to the increase in Nusselt number at all Reynolds numbers for two types of inclined ribs. The *BR* = 0.30 shows the highest heat transfer rate while the *BR* = 0.10 performs the lowest value. Considering at similar *BR* value, the full type performs higher heat transfer rate than the discrete type. In addition, the use of 45^{o} inclined rib with full and discrete types provides the augmentation on heat transfer around 2.4 – 6.5 times higher than the smooth channel depended on *BR*, *Re* and configuration of rib.

The f/f_0 increase when increasing Reynolds number and *BR* values. The *BR* = 0.25 and 0.30 give very enlarge pressure loss, especially, at the highest Reynolds number, *Re* = 20000. The full type provides higher friction factor than the discrete type for all *Re* and *BR* values. Additionally, the use of the two types for the 45^{o} inclined ribs gives the enhancing friction loss around 12 – 250 times higher than the smooth channel. The *TEF* tends to decrease when rising Reynolds number. The optimum *TEF* is found at case *BR* = 0.15 for discrete type around 1.43 at the lowest Reynolds number.

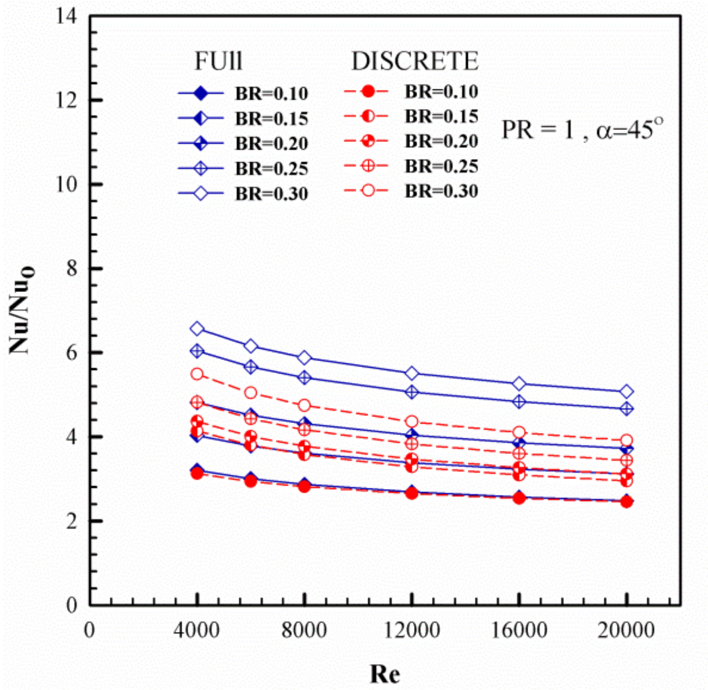


Figure 5: Variation of Nu/Nu_0 with Reynolds number.

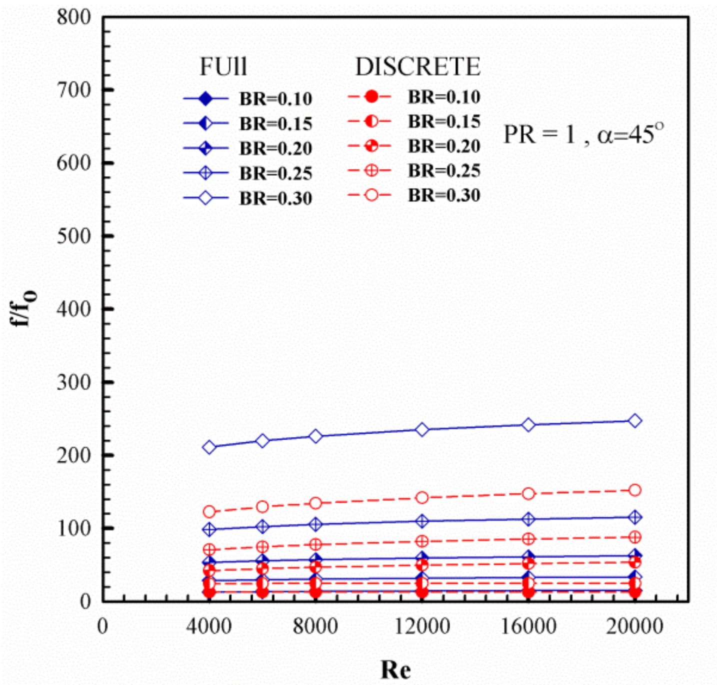


Figure 6: Variation of f/f_0 with Reynolds number.

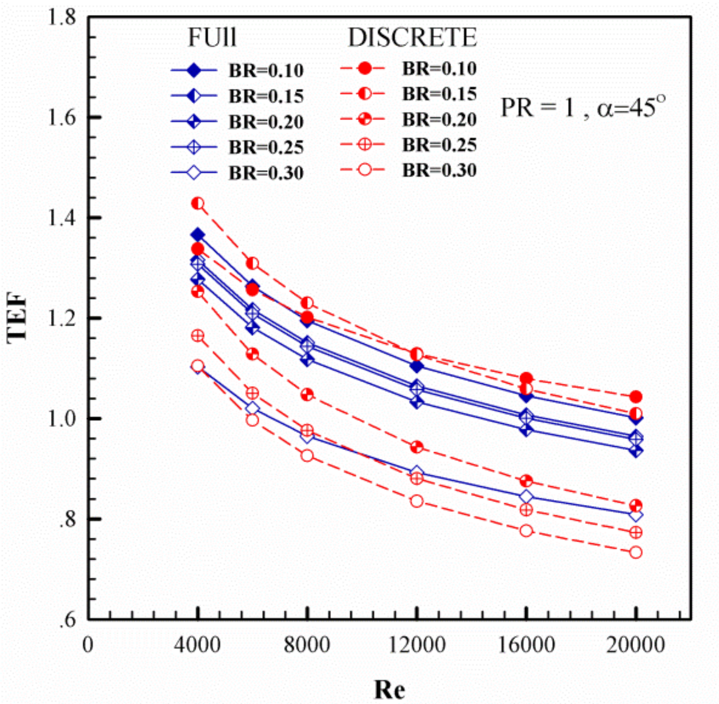


Figure7: Variation of *TEF* with Reynolds number.

5. DISCUSSION AND CONCLUSION

As the numerical results, the use of the 45° inclined ribs with full and discrete types can generate the two main vortex flow and small vortices at the corner of the square channel though the test square channel that help to increase in heat transfer rate. The 45° inclined ribs not only increase in heat transfer rate, but also increase in pressure loss, especially, at high Reynolds number and *BR* values. The full type create a higher level of the vortex strength than the discrete type, that results in a higher heat transfer rate over the channel walls as seen in the Figure 4. The optimization between the rising heat transfer rate and friction loss can be considered by the thermal enhancement factor. The *BR* = 0.15 for discrete type gives the optimum *TEF* at *Re* = 4000.

The Nusselt number ratio is around 2.7 – 6.8 and 2.7 – 5.7 for full type and discrete type, respectively. The present of inclined rib turbulators leads to very enlarge pressure loss ranging from 12.93 – 247 and 12.74 – 152 times in comparison with the smooth channel for full and discrete types, respectively. The *TEF* is found to be about 1 – 1.36 and 1 – 1.43 for full and discrete types, respectively, with *BR* = 0.1 – 0.3, *Re* = 4000 – 20,000 and *PR* = 1.

NOMENCLATURE

A	convection heat transfer area, m^2
BR	blockage ratio,(b/H)
B	rib height, m
D	hydraulic diameter of square duct, (= H)
f	friction factor
H	channel height, m
h	convective heat transfer coefficient, W m^{-2} K^{-1}
k	turbulent kinetic energy
k_a	thermal conductivity of air, W m^{-1} K^{-1}
L	cyclic length of one cell (or axial pitch length), m
Nu	Nusselt number (=hD/k)
p	static pressure, Pa
Pr	Prandtl number(Pr = 0.707)
PR	spacing pitch ratio, L/H
Re	Reynolds number(=$\rho\bar{u}D/\mu$)
T	temperature, K
TEF	thermal enhancement factor, (=$(Nu/Nu_0) / (f/f_0)^{1/3}$)
u_i	velocity component in xi-direction, m s^{-1}
$\bar{u}$	mean velocity in duct, m s^{-1}

W	width of the channel, m
Greek letter	
μ	dynamic viscosity, kg $s^{-1}m^{-1}$
Γ	thermal diffusivity
α	fin inclination angle or angle of attack, degree
ρ	density, kg m^{-3}
Subscript	
0	smooth duct
in	inlet
pp	pumping power
w	wall
t	turbulence

6. ACKNOWLEDGEMENT

This researcher was funded by College of Industrial Technology, King Mongkut's University of Technology North Bangkok, Thailand. The researchers would like to thank Dr. WithadaJedsadaratanachai and Assoc. Prof. Dr. PongjetPromvonge for suggestions.

7. REFERENCES

[1] Y.Wang, B.Zhou, Z.Liu, Z.Tu, and W.Liu, "Numerical study and performance analyses of the mini-channel with discrete double-inclined ribs", International Journal of Heat and Mass Transfer, Elsevier,vol. 78, pp. 498–505, 2014.

[2] K.Yongsiri, P.Eiamsa–ard, K.Wongcharee, and S. Eiamsa–ard, "Augmented heat transfer in a turbulent channel flow with inclined detached–ribs", Case Studies in Thermal Engineering, Elsevier,vol. 3, pp. 1–10, 2014.

[3] K.R.Aharwal, B.K. Gandhi, and J.S. Saini, "Heat transfer and friction characteristics of solar air heater ducts having integral inclineddiscrete ribs on absorber plate", International Journal of Heat and Mass Transfer, Elsevier,vol. 52, Issue 25–26, pp. 5970–5977,2009.

[4] X.Y.Tang, and D.S. Zhu, "Flow structure and heat transfer in a narrow rectangular channel with different discrete rib arrays", Chemical Engineering and Processing: Process Intensification, vol. 69, pp. 1–14, 2013.

[5] F.Satta, D. Simoni, and G. Tanda, "Experimental investigation of flow and heat transfer in a rectangular channel with 45^o angled ribs on one/two walls", Experimental Thermal and Fluid Science, vol. 37, pp.46–56, 2012.

[6] G.Tanda, "Effect of rib spacing on heat transfer and friction in a rectangular channel with 45^o angled rib turbulators on one/two walls", International Journal of Heat and Mass Transfer, Elsevier, vol. 54, Issue5–6, pp. 1081–1090, 2011.

[7] X. Gao, and B. Sunden, "Heat transfer and pressure drop measurements in rib-roughened rectangular ducts", Experimental Thermal and Fluid Science, vol. 24, Issue 1–2, pp. 25–34, 2001.

[8] P. Dutta, and A.Hossain, "Internal cooling augmentation in rectangular channel using two inclined baffles", International Journal of Heat and Fluid Flow, vol. 26, Issue 2, pp. 223–232, 2005.

[9] A.S. Yadav, and J.L. Bhagoria, "A numerical investigation of square sectioned transverse rib roughened solar air heater", International Journal of Thermal Sciences, vol. 79, pp. 111–131, 2014.

[10] B. Lu, and P.X. Jiang, "Experimental and numerical investigation of convection heat transfer in a rectangular channel with angled ribs", Experimental Thermal and Fluid Science, vol. 30, Issue 6, pp. 513–521, 2006.

[11] S.Y. Won, G.I.Mahmood, and P.M. Ligrani, "Flow structure and local Nusselt number variations in a channel with angled crossed–rib turbulators", International Journal of Heat and Mass Transfer, vol. 46, Issue 17, pp. 3153–3166, 2003.

[12] S.V.Patankar, C.H. Liu, and E.M. Sparrow, "Fully developed flow and heat transfer in ducts having streamwise–periodic variations of cross–sectional area", ASME Journal of Heat Transfer, vol. 99, pp.180–186, 1977.