Thermal Performance Analysis for Laminar Forced Convection in a Square Channel with PD-RWVG

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ABSTRACT

The numerical investigation on heat transfer characteristics, flow configurations and thermal evaluations in a square channel with 30^o pointing–downstream rectangular winglet vortex generators (PD–RWVG) is presented. The RWVGs are placed on both the upper and lower walls of the square channel with in–line arrangement. The different RWVG lengths, t/W=0.1 – 0.9, with single blockage ratio, b/H = BR = 0.15, and pitch ratio, L/H = PR = 1, are investigated in three dimensional for Reynolds number based on the hydraulic diameter of the square channel, D_h, Re = 100 – 1000. The finite volume method and periodic boundary apply for the current computational domain. The numerical results are reported in three parts; flow configurations, heat transfer characteristics and performance evaluations. In the part of performance evaluations, the heat transfer, pressure loss and thermal performance are presented in forms of Nusselt number ratio (Nu/Nu_0), friction factor ratio (f/f_0) and the thermal enhancement factor (TEF), respectively. At similar conditions, the use of the channel equipped with PD–RWVG consistently results in higher heat transfer rate and friction factor than those of the smooth channel. The increases of t/W value and Reynolds number result in the rising heat transfer rate and friction factor. The highest Nu/Nu_0 of 6.23, f/f_0 of 9.3 and TEF of 2.96 are obtained at t/W = 0.90, Re = 1000.

Keywords:

1. INTRODUCTION

The subject of heat transfer augmentation has significant interest to develop the compact heat exchangers in order to obtain a high performance, low cost, light weight, and size as small as possible. Hence, energy cost and environmental considerations are going on to encourage attempts to invent better performance over the existing designs. The vortex generators; rib, baffle, winglet, groove, etc., are widely use for enhancing thermal performance in the heat exchanger channel by changing the flow configuration and creating secondary flow, vortex flow, swirling flow, impinging jet result in the increase in heat transfer rate over the plain channel. Both numerical and experimental investigations had been extensively studied on the effects of vortex generators. The numerical method offers the flow configuration describing the mechanisms behind the thermal performance improvement by vortex generators. The numerical results of the heat transfer enhancement by vortex generators from the published works are shown in the table. 1.

Table 1

Authors	**Studied cases**	***Nu/Nu*₀**	***f/f*₀**	***TEF***
Jedsadaratanachai . [1]	30° inclined baffle Inline, two opposite walls, square channel = 0.2 = 0.5 – 2.5 = 100 – 2000	1 – 9.2	1 – 21.5	3.78
Kwankaomeng and Promvonge [2]	30° inclined baffle One side, square channel = 0.1 – 0.5 = 1.0 – 2.0 = 100 – 1000	1 – 9.23	1.09 – 45.31	3.1
Promvonge . [3]	30° inclined baffle Inline, two opposite walls, square channel = 0.1 – 0.3 = 1.0 – 2.0 = 100 – 2000	1.2 – 11.0	2 – 54	4
Promvonge and Kwankaomeng [4]	45° V–baffle Staggered, two opposite walls, = 2 channel = 0.05 – 0.3 = 1.0 = 100 – 1200	1 – 11	2 – 90	2.75
Promvonge . [5]	45° inclined baffle Inline – staggered, two opposite walls, square channel = 0.05 – 0.3 = 1.0 = 100 – 1000	1.5 – 8.5	2 – 70	2.6
Promvonge . [6]	45° V–baffle Inline Downstream, two opposite walls, square channel = 0.1 – 0.3 = 1.0 – 2.0 = 100 – 2000	1 – 21	1.1 – 225	3.8
Boonloi [7]	20° V–baffle Inline Downstream-Upstream, two opposite walls, square channel = 0.1 – 0.3 = 1.0 = 100 – 2000	1 – 13	1 – 52	4.2
Boonloi and Jedsadaratanachai [8]	30° V–baffle Downstream, One side, square channel = 0.1 – 0.5 = 1.0 – 2.0 = 100 – 1200	1 – 14.49	2.18 – 313.24	2.44

According to the above results, the use of V baffles resulted in heat transfer enhancement with significant pressure loss penalty. Therefore, this work focuses on the modification of V shaped vortex generators, in order to reduce pressure loss by trimming the V tip in form of rectangular winglet. The modified vortex generators, namely "rectangular winglet vortex generators, " with 30^o of attack angle and pointing downstream ( ) are placed on both the upper and lower walls of the square channel with inline arrangement. The use of is expected to generate the vortex flow and the impinging jet flow over the channel and help to increase in heat transfer rate and the thermal performance in the heat transfer system. The influences of width ratio, / , are investigated in three dimensional simulation. The results of the channel with are also compared with the smooth square channel.

2. FLOW DESCRIPTIONS

2.1 Channel Geometry and Case Studies

The configuration of 30^o s in a square channel are developed from s. [1] [8]. The s are placed on the channel walls with inline arrangement and pointing downstream as Fig. 1. The square channel height, is 0.05 m which equal to distance between , = . The current study, the blockage ratio (/ ,), the pitch ratio (/ ,) and the gap at V tip ( ) are fixed at 0.15, 1 and 0.1 , respectively. The length ratios, / , vary from 0.1 – 0.9 as displayed in Fig. 2.

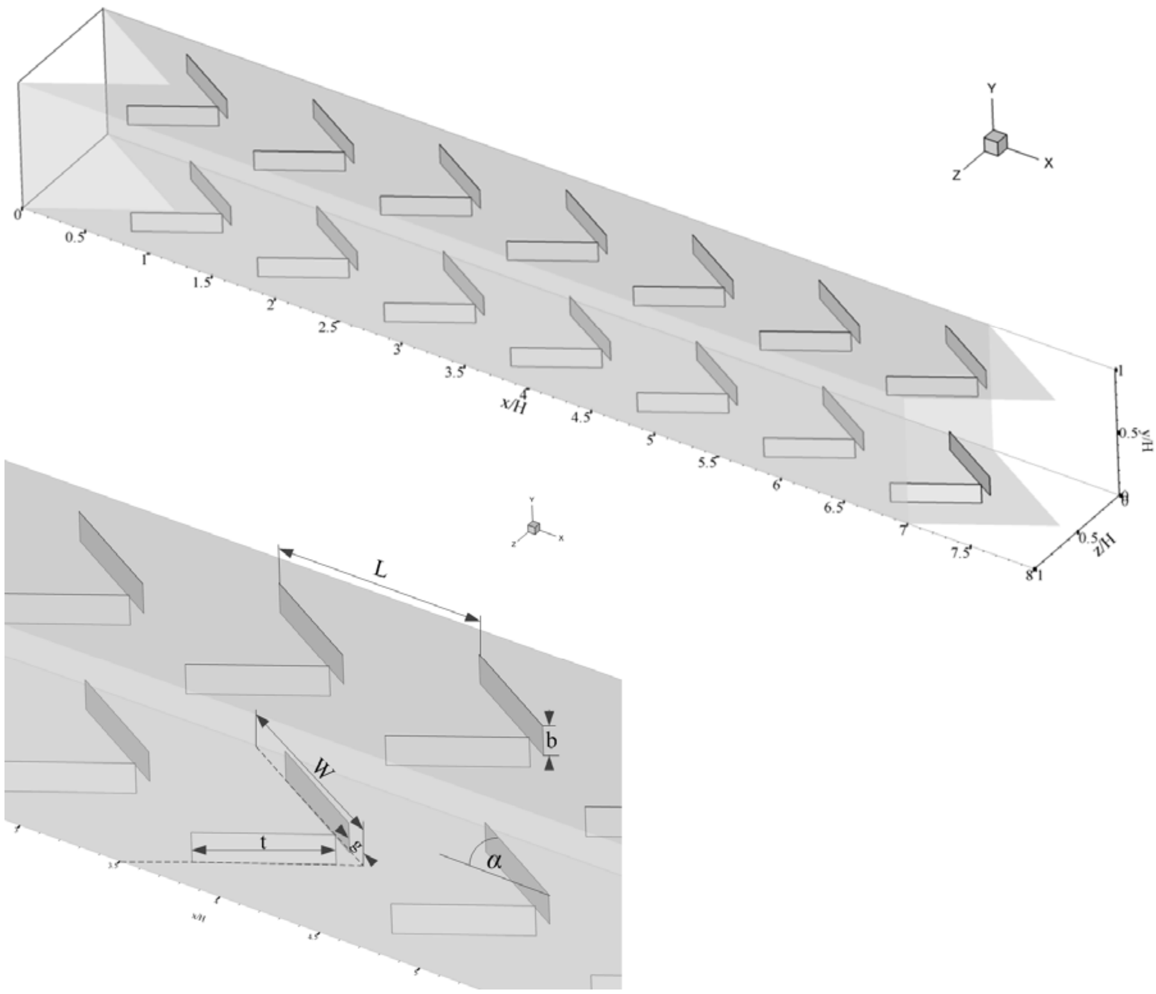


Fig. 1 .

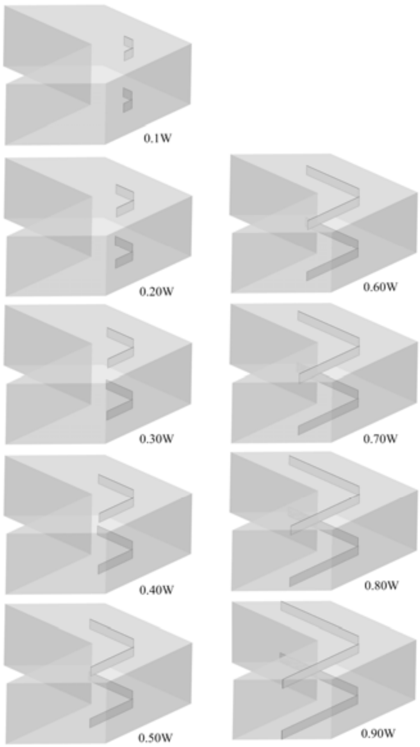


Fig. 2

2.2Boundary Conditions

The boundary conditions of the present work are cited form [1] – [8]. The boundary conditions are summarized in table 2.

Table 2.

Zone	Boundary conditions
Inlet	Periodic
Outlet	Periodic
Square channel walls	- Constant temperature 310K - No–slip wall
RWVG	Adiabatic wall
Tested fluid	- Constant mass flow rate of air with 300 K (= 0.7) - Constant physical properties

3. MATHEMATICAL FOUNDATION

From . [1] – [8], 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 is laminar and incompressible.
- Constant fluid properties.
- Body forces and viscous dissipation are ignored.
- Negligible radiation heat transfer.

Based on the assumptions, the channel flow is governed by the continuity, the Navier–Stokes equations and the energy equation. In the Cartesian tensor form these equations can be written as follows:
Continuity equation:

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

Momentum equation:

$$\frac{\partial(\rho \quad)}{\partial} = -\frac{\partial}{\partial} + \frac{\partial}{\partial}\left[\mu\left(\frac{\partial}{\partial} + \frac{\partial}{\partial}\right)\right] \tag{2}$$

Energy equation:

$$\frac{\partial}{\partial}(\rho \quad) = \frac{\partial}{\partial}\left(\Gamma\frac{\partial}{\partial}\right) \tag{3}$$

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 [9]. 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:

$$\mathrm{Re} = \rho^{-} \quad / \mu \tag{5}$$

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

$$= \frac{(\Delta \ / \ )}{\frac{1}{2}\rho^{-2}} \tag{6}$$

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

$$= \frac{}{} \quad (7)$$

The average Nusselt number can be obtained by

$$= \frac{1}{}\int \partial \quad (8)$$

The thermal enhancement factor ( ) is defined as the ratio of the heat transfer coefficient of an augmented surface, to that of a smooth surface, $_0$, at an equal pumping power and given by

$$= \frac{}{}\Big|_0 = \frac{}{}\Big|_0 = (/ _0)/(/ _0)^{1/3} \quad (9)$$

Where, and stand for Nusselt number and friction factor for the smooth square channel, respectively.

The computational domain is resolved by regular Cartesian elements. For this channel flow, however, regular grid was applied throughout the domain. A grid independence procedure was implemented by using Richardson extrapolation technique over grids with different numbers of cells. The characteristics of four grids; 54,000, 82,000, 122,400 and 250,000 cells, are used in the simulations for using the grid convergence index (GCI) [10]. The variation in and values for the 30° at = 0.6 is less than 0.15% when increasing the number of cells from 122,400 to 250,000, thus there is no such advantage in increasing the number of cells beyond this value. Considering both convergent time and solution precision, the grid system of 122,400 cells was adopted for the current computational model.

4. RESULT AND DISCUSSION

4.1Verification of Smooth Square Channel

The validations of the heat transfer and friction factor of the smooth square channel without by comparison with the previous values under a similar operating condition. The results are found to be in excellent agreement with exact solution values obtained from the open literature [11] for both the Nusselt number and the friction factor, less than ±0.25 % deviation. The exact solutions of the Nusselt number and the friction factor for laminar flows in a smooth square channel with constant wall temperature are shown in equations 10 and 11, respectively.

$$_0 = 2.98 \quad (10)$$

$$_0 = 57/\mathrm{Re} \quad (11)$$

4.2Flow Configuration

The flow configurations of the – in square channel are presented in forms of streamlines in transverse planes and streamlines impinging jet on the square channel walls as presented in Fig. 3 and 4, respectively. Figs. 3 , , and show the streamlines in transverse planes at = 1000 for / = 0.2, 0.4, 0.6 and 0.8, respectively. The results reveal that the flow fields in the channel with are considerably different from that in the smooth square channel. In general, each induces four main vortex flows in form of counter–rotating with common–flow–up. The apparent of the vortex flows is a key for enhancing heat transfer rate and thermal performance.

Figs. 4 , , and display the streamlines impinging jet on the lower wall with $_x$ contours of the square channel at the highest Reynolds number, = 1000 for / = 0.2, 0.4, 0.6 and 0.8, respectively. Evidently, induces impinging jet flow over the square channel wall that lead to a higher heat transfer rate due to the decreasing boundary layer thickness. The rising / results in the increasing vortex intensity and heat transfer rate. Among the examined , the one with / = 0.8 provides the highest on vortex strength and heat transfer rate while the one with / = 0.2 gives the opposite result.

4.3Heat Transfer Behavior

The heat transfer behaviors in forms of temperature contours in transverse planes and Nu_x contours on the channel walls as presented Figs. 5 and 6, respectively. Figs. 5 , , and show the temperature contours in transverse planes at = 1000 for / = 0.2, 0.4, 0.6 and 0.8, respectively. As seen, the gives better mixing of the fluid flow when comparing with the smooth square channel, especially, the / = 0.8 in which the thermal boundary layer thickness is considerably smaller than those in other cases.

The $_x$ contours on the square channel walls at = 1000 for / = 0.2, 0.4, 0.6 and 0.8 are given in Figs. 6 , , and , respectively. It is found that the increase of / leads to higher heat transfer rate indicated by the large red area in the contour.

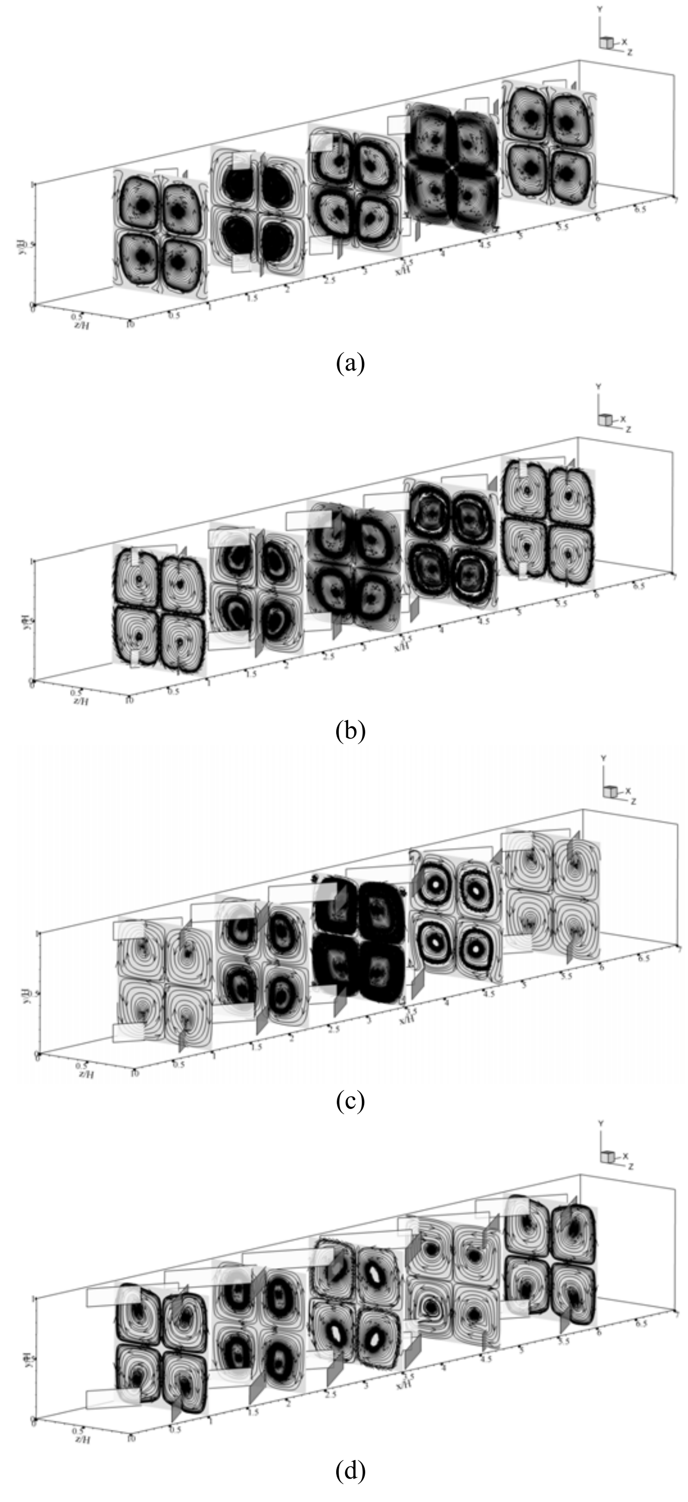
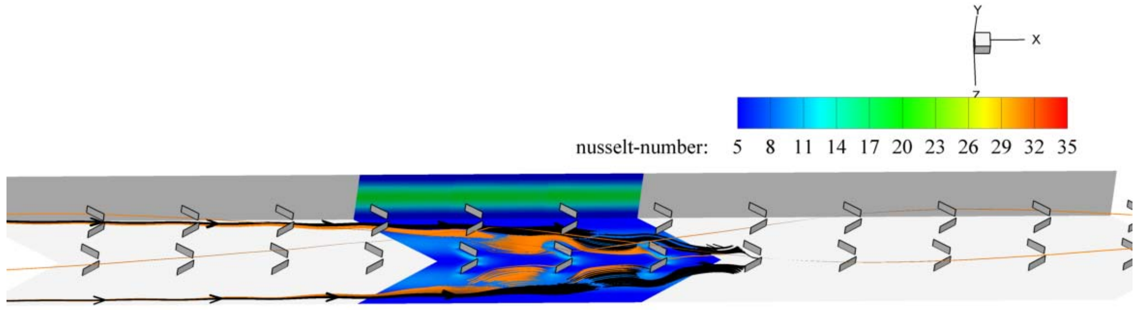
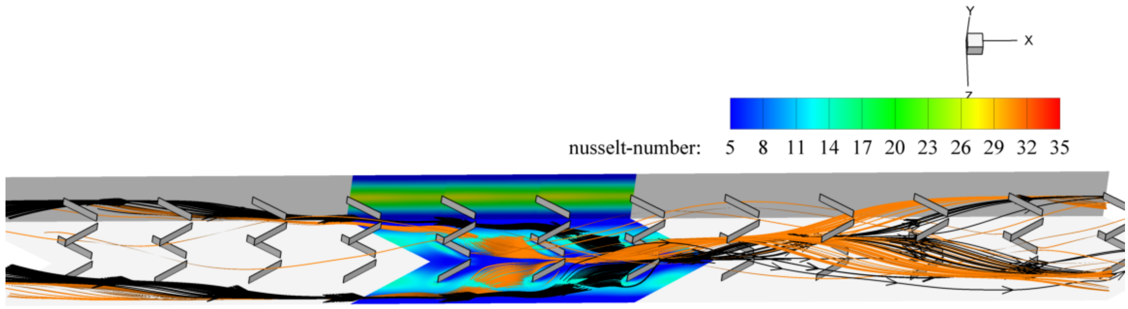


Fig. 3


Y
X
Z
nusselt-number: 5 8 11 14 17 20 23 26 29 32 35

(a)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X
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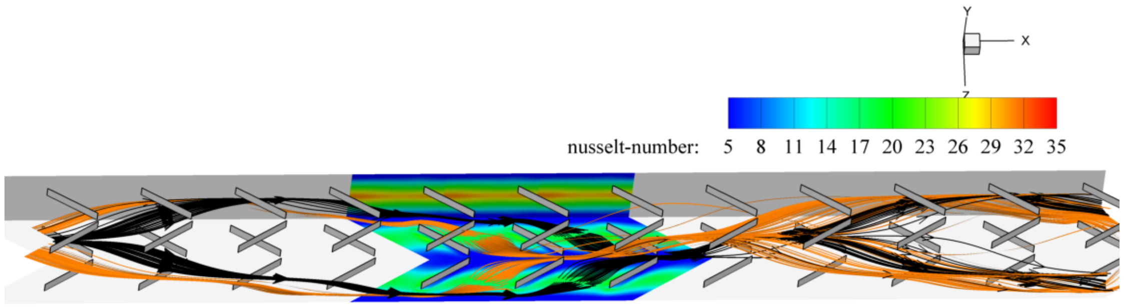
nusselt-number: 5 8 11 14 17 20 23 26 29 32 35

(b)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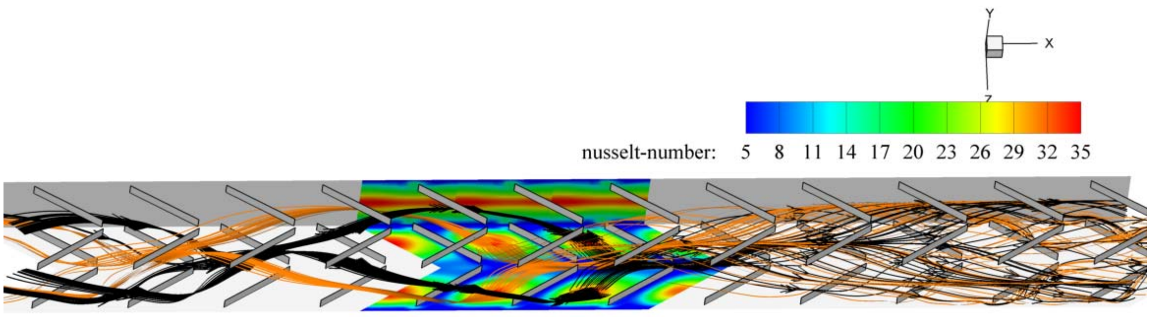
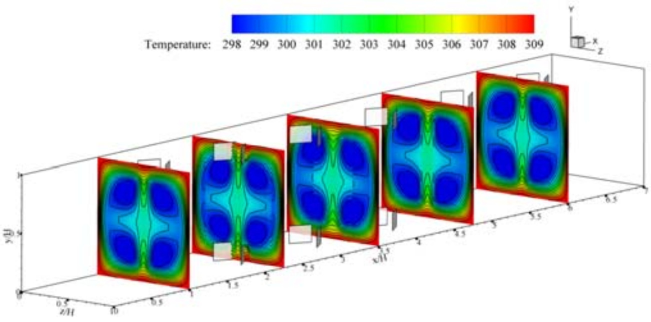
nusselt-number: 5 8 11 14 17 20 23 26 29 32 35

(c)

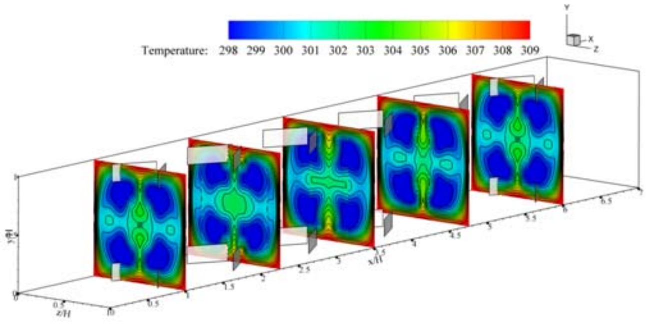

Y
X
Z
nusselt-number: 5 8 11 14 17 20 23 26 29 32 35

(d)

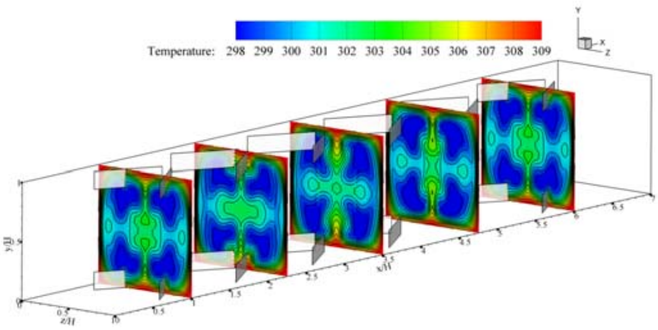
Fig. 4


Temperature: 298 299 300 301 302 303 304 305 306 307 308 309

(a)


Temperature: 298 299 300 301 302 303 304 305 306 307 308 309

(b)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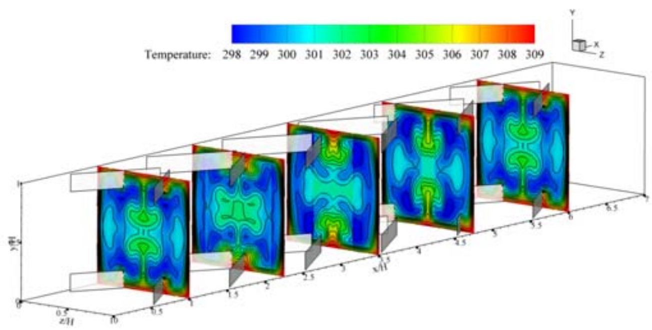
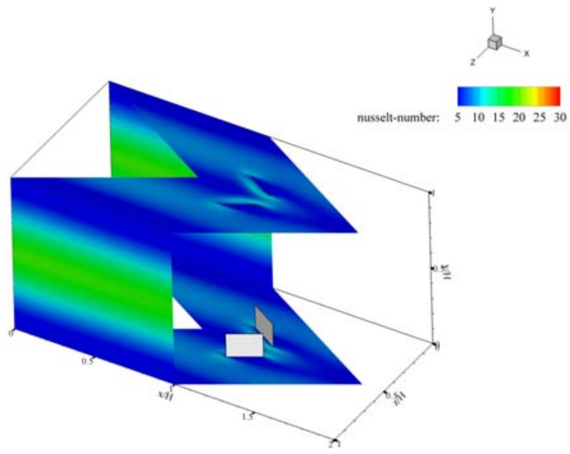

Temperature: 298 299 300 301 302 303 304 305 306 307 308 309

(d)

Fig. 5



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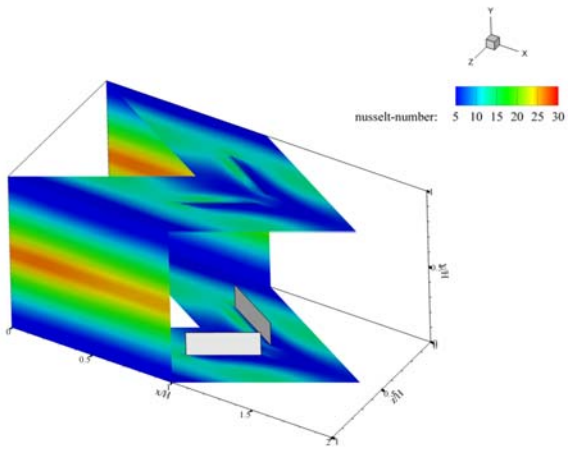

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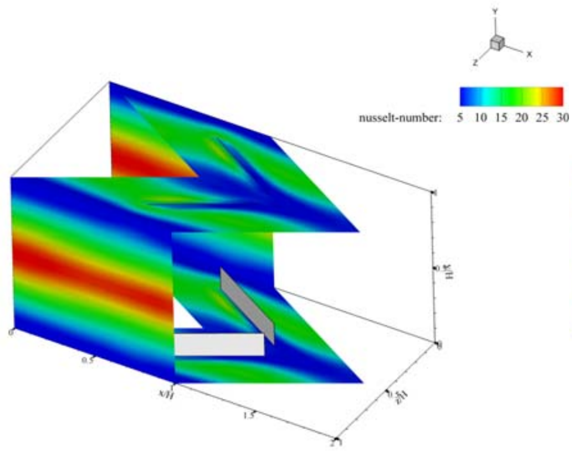

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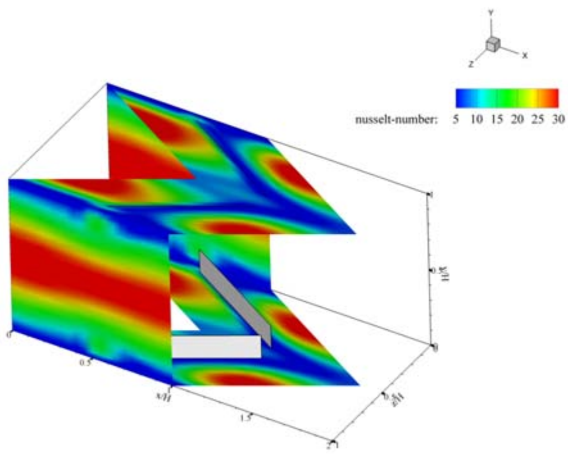


(b)

Fig. 6

4.3Performance Evaluation

The parameter relevant to the thermal performance evaluation including heat transfer, pressure loss and thermal performance in forms of / $_0$, / $_0$ and thermal enhancement factor, , are shown Figs 7, 8 and 9, respectively.

Figs. 7 and present the variations of the / $_0$ with Reynolds number and / value, respectively. In general, the increase of Reynolds number and / leads to the increase of / $_0$. The highest value of / $_0$ of 6.23 is achieved at / = 0.9 at the highest Reynolds number, = 1000, while the lowest / $_0$ of 1.9 is obtained at / = 0.1. In range studied, the / $_0$ is varied from 1 to 6.23 depending on Reynolds number and / value.

The variations of the / $_0$ with Reynolds number and / are shown in Figs. 8 and , respectively. The / $_0$ value tends to increase with the rise of Reynolds number and / value for all cases. The maximum of / $_0$ of 9.3 is found at / = 0.9 and = 1000. s with / from 0.1 to 0.9 give / $_0$ from 1 to 9.3 when is varied from 100 to1000.

Figs. 9 and display the variations of with Reynolds number and / , respectively. As shown, as Reynolds number and / increase, increases. At = 100, s with all / give of 1. The in range studied is varied from 1 – 2.96 depending on Reynolds number and / value. The maximum of 2.96 is achieved at / = 0.90 and e = 1000.

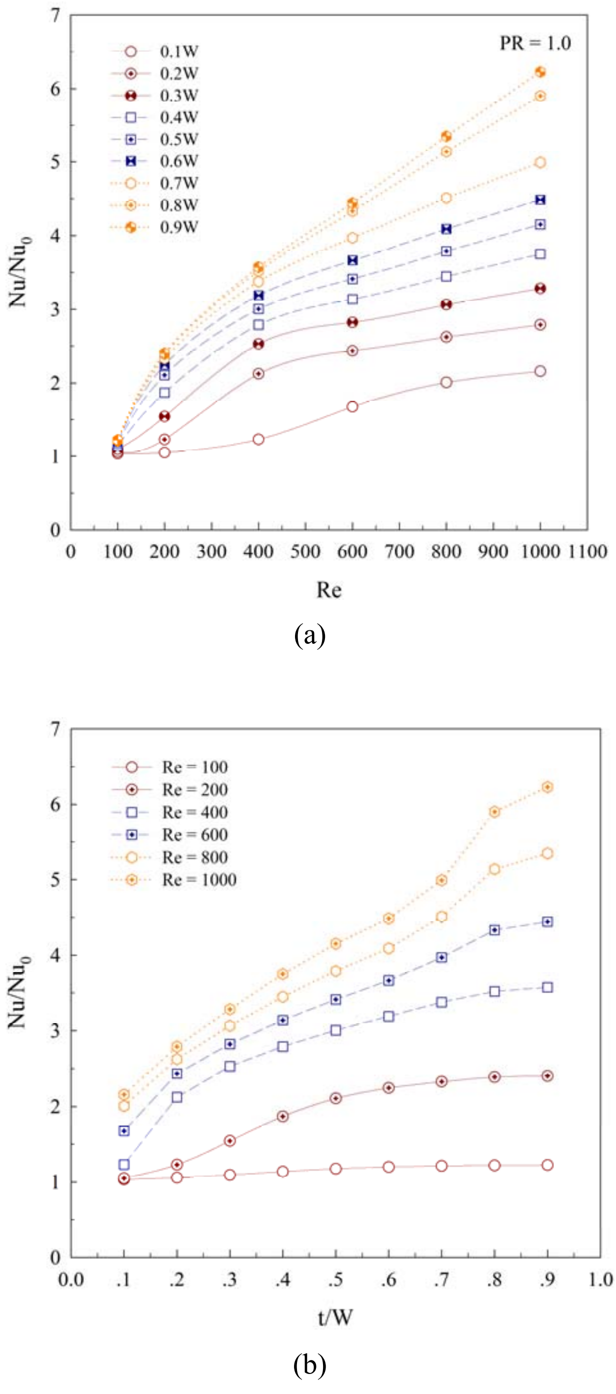

PR = 1.0
0.1W
0.2W
0.3W
0.4W
0.5W
0.6W
0.7W
0.8W
0.9W
Nu/Nu0
Re
(a)
Re = 100
Re = 200
Re = 400
Re = 600
Re = 800
Re = 1000
Nu/Nu0
t/W
(b)

Fig. 7

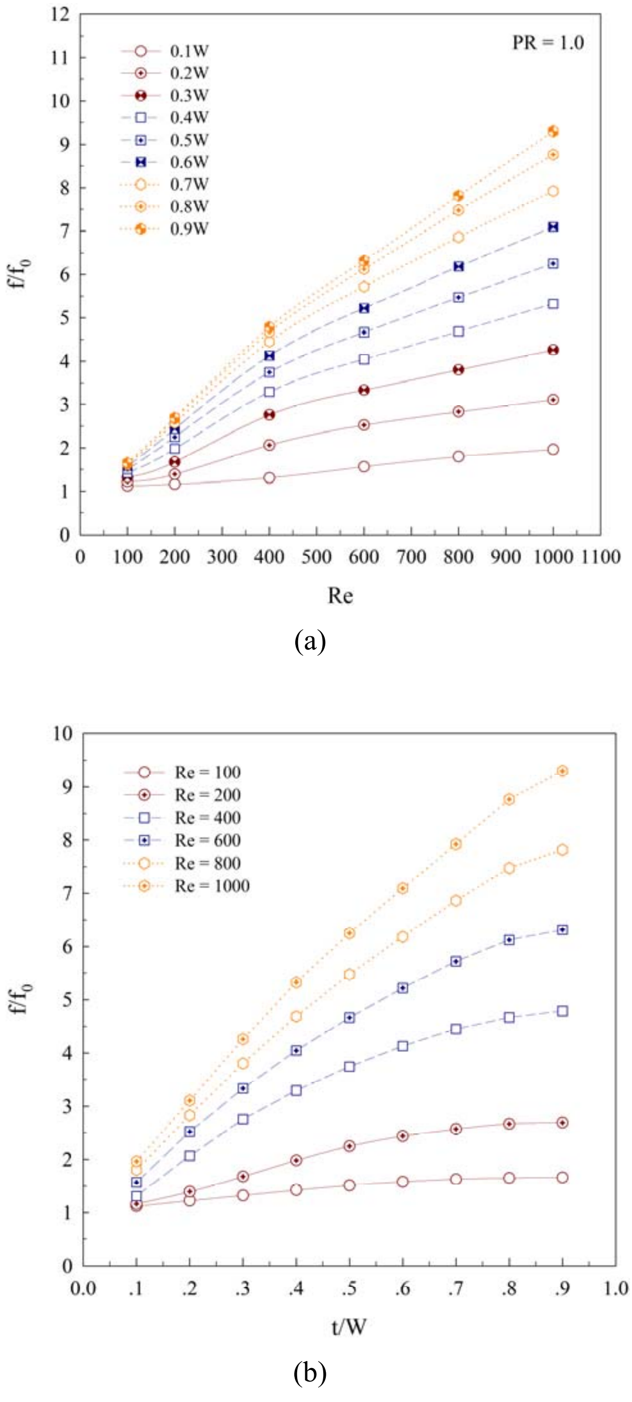
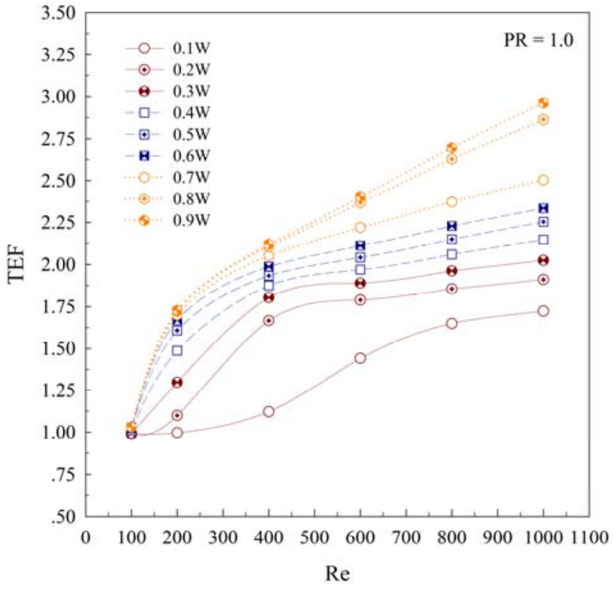
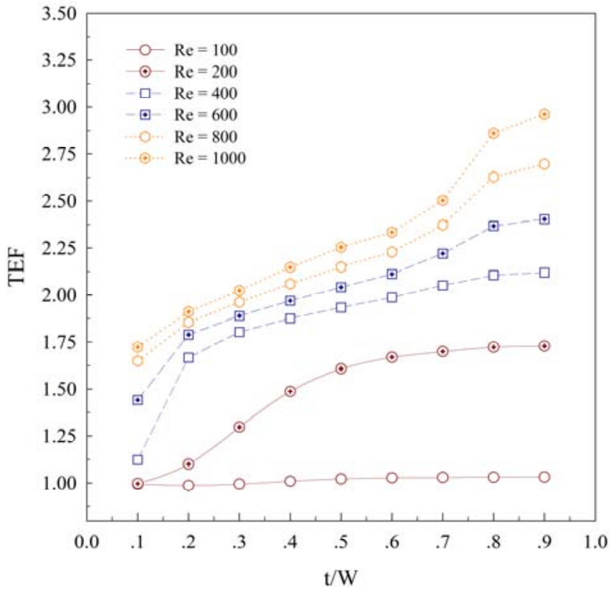

PR = 1.0
0.1W
0.2W
0.3W
0.4W
0.5W
0.6W
0.7W
0.8W
0.9W
f/f0
Re
(a)
Re = 100
Re = 200
Re = 400
Re = 600
Re = 800
Re = 1000
f/f0
t/W
(b)

Fig. 8



(a)



(b)

Fig. 9

5. CONCLUSION

The heat transfer characteristics, flow configurations and performance evaluations for 30° – in the square channel with the inline arrangement for Reynolds number ranging 100 – 1000 are investigated numerically. The main findings are as follows:

The use of – results in better mixing of the fluid flow compared to that in the smooth square channel with no , leading to the increasing heat transfer rate and thermal performance.

NOMENCLATURE

	flow blockage ratio, (/)
	baffle height, m
h	hydraulic diameter of the square channel
	friction factor
GCI	grid convergence index
	gap at V–tip
	convective heat transfer coefficient, W m^{-2} K^{-1}
	thermal conductivity, W m^{-1} K^{-1}
	cyclic length of one cell (or axial pitch length,), m
	Nusselt number
	static pressure, Pa
	pointing downstream
	Prandtl number
	pitch or spacing ratio, /
	Reynolds number, $(\rho^{-}\ /\mu)$
	rectangular winglet vortex generators
	length
	temperature, K
	velocity in -direction, m s^{-1}
–	mean velocity in channel, m s^{-1}
	fully length of
	dynamic viscosity, kg $s^{-1}m^{-1}$
Γ	thermal diffusivity
α	baffle inclination angle or angle of attack, degree
	thermal enhancement factor, (=)
ρ	density, kg m^{-3}
in	inlet
0	smooth tube
w	wall
pp	pumping power

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