

Effect of Flow Attack Angle on Heat Transfer, Friction Loss and Thermal Performance in Square Channel with Inclined Wavy Surface

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ABSTRACT— *Effects of flow attack angles for inclined wavy surface on heat transfer rate, friction loss and thermal performance in a square channel are investigated numerically. The inclined wavy surface with square profile is inserted in the middle of the square channel to increase heat transfer rate and performance. The current study is solved by finite volume method with SIMPLE algorithm. The convective heat transfer in turbulent regime, $Re = 3000 - 10,000$, is considered. The flow structure and heat transfer behavior in the test section are presented in the numerical results. The thermal performance is also analyzed in terms of Nusselt number ratio (Nu/Nu_0), friction factor ratio (f/f_0) and thermal enhancement factor (TEF). The maximum heat transfer rate is detected at the flow attack angle of 45° , while the extreme pressure loss is found at the flow attack angle of 60° . The optimum TEF are found at the flow attack angle of 30° and 45° around 1.9.*

Keywords— wavy surface, heat transfer, thermal performance, Nusselt number, square channel

1. INTRODUCTION

The addition of the turbulators in heat exchanger to improve heat transfer rate and thermal performance is always found in various industries and engineering applications such as solar air heater. The turbulators such as rib, baffle, roughness surface, fin, etc. can develop the thermal performance due to the turbulators can disturb the thermal boundary layer of the heat transfer surface. The selection of the turbulators depends on the application of the heat exchanger. Moreover, the manufacture and maintenance of the turbulators are important points for the selection of the turbulators.

The baffle/rib [1-2] is usually selected to enhance heat transfer rate and performance in the heating/cooling section. Many works presented the heat transfer, pressure loss and thermal performance in heat exchangers with inclined-shape baffle. Jedsadaratanachai *et al.* [3] numerically investigated on heat transfer, pressure loss and thermal performance in a square channel inserted with 30° inclined baffle. The baffles were placed on both the upper and lower walls of the channel with inline arrangement. The pitch ratios were varied with single blockage ratio of 0.2. They concluded that the augmentation on heat transfer is around 1 – 9.2 times above the smooth channel. Kwankaomeng and Promvong [4] numerically studied the thermal performance improvement in a square channel with 30° inclined baffle. The baffles were placed on one side of the channel wall. The influences of the blockage ratios and pitch ratios for the inclined baffle were presented for laminar regime. Promvong *et al.* [5] studied the effects of baffle height and pitch spacing for 30° inclined baffle, which placed on the opposite walls of the square channel. They summarized that the augmentation on heat transfer rate is around 1.2 – 11 times over the plain channel. They also pointed out the thermal performance is around 4 when considering at similar pumping power. Promvong *et al.* [6] studied the effects of baffle height on heat transfer and pressure loss in a square channel heat exchanger with numerical method. The 45° inclined baffles were placed on the upper and lower surfaces of the channel with inline and staggered arrangements. The optimum thermal enhancement factor is around 3.8.

The wavy surface or roughness surface is type of the turbulator, which always use in the fin-and-tube heat exchanger to augment heat transfer rate and thermal performance. For examples, the investigations on flow configuration and heat transfer behavior in a smooth wavy fin-and-elliptical tube heat exchanger with various type vortex generators were reported by Lotfi *et al.* [7]. Dong *et al.* [8] experimentally studied a wavy-fin-flat-tube heat exchanger on heat transfer and thermal performance. They concluded that the amplitude and length of the wavy fin are most important factor for thermal performance improvement. Dong *et al.* [9] also examined the thermal hydraulic performance of a wavy fin-and-flat tube heat exchanger by both numerical and experimental methods. They concluded that the waviness amplitude is a key for augmenting the heat transfer rate and pressure loss, while the wavy fin profiles; triangular, sinusoidal and triangular round corner, have slightly effect for thermal performance. The numerical investigations on the heat transfer behaviors in a wavy fin-and-tube heat exchanger with combined rectangular winglet pairs were presented by Gong *et al.*

[10]. They found that combined vortex generators can generate larger and stronger vortex flow than the single rectangular winglet pairs. Du *et al.* [11] studied the heat transfer increment in a wavy fin-and-flat-tube heat exchanger with punched longitudinal vortex generators. They explained that the best thermal performance of the heating system is around 1.23. Du *et al.* [12] experimentally examined the longitudinal vortex flow in a wavy fin-and-flat-tube heat exchanger on thermal performance improvement at $Re = 1500 - 4500$. They stated that the augmentations on the Nusselt number and friction factor are around 21 – 60% and 13 – 83%, respectively, while the thermal performance is around 1.31.

In the present research, the inclined wavy surface (combination of the inclined baffle with staggered arrangement and wavy surface) with square profile ($0.2H \times 0.2H$) is selected to develop heat transfer rate and thermal performance in a square channel. The square profile is created due to the Ref. [3] reported that the optimum height of the baffle is around 0.15 – 0.2. The wavy surface is inserted in the middle of the square channel heat exchanger. The manufacture, maintenance and installation of the wavy surface are more convenient than the baffle or rib, which places on the channel surface. The influences of the flow attack angles ($15^\circ - 60^\circ$) for the inclined wavy surface are investigated for $Re = 3000 - 10,000$.

2. PHYSICAL MODEL

The inclined wavy surface with square profile is inserted in the middle of the square channel heat exchanger as depicted in Fig. 1. The square profile is set around $0.2H \times 0.2H$ in all cases. The hydraulic diameter of the square channel is equal to the channel height (H). The flow attack angles are varied as $15^\circ, 20^\circ, 25^\circ, 30^\circ, 35^\circ, 40^\circ, 45^\circ, 50^\circ, 55^\circ$ and 60° . The present study is imposed for turbulent regime, $Re = 3000 - 10,000$. The computational domain for the test section is also illustrated in the Fig. 1.

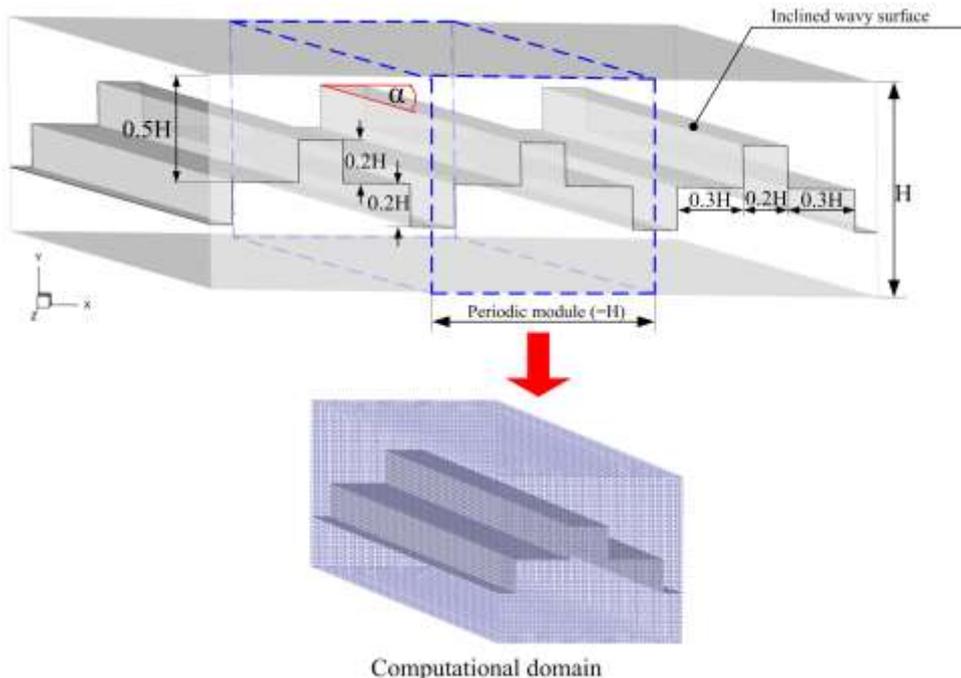


Figure 1: Square channel inserted with inclined wavy surface and computational domain.

3. ASSUMPTION AND BOUNDARY CONDITION

The tested fluid is air ($300K, Pr = 0.707$, incompressible) in turbulent regime. The flow and heat transfer in the heating tube are steady in three dimensions. The body force, viscous dissipation, natural convection and radiation heat transfer are ignored. The boundary conditions of the computational domain are performed as Table. 1

Table 1: Boundary condition of the computational domain.

Zone	Boundary condition
Inlet, outlet	Periodic condition
Channel walls	No slip wall Uniform heat flux around 600 W/m^2
Wavy surface	No slip wall Insulators

4. MATHEMATICAL FOUNDATION

The turbulent model is realizable k - ε with enhanced wall treatment as following equations;

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho \varepsilon - Y_M + S_k \quad (1)$$

and

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_j}(\rho \varepsilon u_j) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + \rho C_1 S \varepsilon - \rho C_2 \frac{\varepsilon^2}{k + \sqrt{\nu \varepsilon}} + C_{1\varepsilon} \frac{\varepsilon}{k} C_{3\varepsilon} G_b + S_\varepsilon \quad (2)$$

where

$$C_1 = \max \left[0.43, \frac{\eta}{\eta + 5} \right], \eta = S \frac{k}{\varepsilon}, S = \sqrt{2 S_{ij} S_{ij}} \quad (3)$$

the constants in the model are given as follows:

$$C_{1\varepsilon} = 1.44, C_2 = 1.9, \sigma_k = 1.0, \sigma_\varepsilon = 1.2 \quad (4)$$

The second order upwind (SOU) numerical scheme is selected for all governing equations, decoupling with the SIMPLE algorithm using a finite volume method. The solutions are measured to be converged when the normalized residual are less than 10^{-9} and 10^{-5} for the energy equation and the other variables, respectively.

The important parameters are Reynolds number (Re), friction factor (f), local Nusselt number (Nu_x), Nusselt number (Nu) and thermal enhancement factor (TEF).

The Reynolds number is computed from Eq. 5.

$$Re = \frac{\rho u_0 H}{\mu} \quad (5)$$

ρ , μ and u_0 are density, viscosity and inlet velocity of the fluid, respectively, while H is the hydraulic diameter/height of the square channel.

The pressure loss in the heating channel is presented as friction factor. The friction factor can calculate as Eq. 6.

$$f = \frac{(\Delta P / L) D_h}{2 \rho \bar{u}^2} \quad (6)$$

ΔP is the pressure drop across the periodic module, L , and u is mean flow velocity.

The heat transfer rate in the tested section is shown in terms of Nusselt number.

The local Nusselt number is computed by

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

h_x is local heat transfer coefficient based on bulk temperature and k is thermal conductivity of the air.

The average Nusselt number can be printed by

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

where, A is heat transfer area of the square channel heat exchanger.

Thermal performance is presented in form of thermal enhancement factor (TEF).

The TEF is defined as the ratio of the heat transfer coefficient of an augmented surface, h to that of a smooth surface, h_0 , under the constant pumping power condition. The TEF can be stated as follow;

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

5. NUMERICAL VALIDATION

The verifications of the computational domain for the square channel heat exchanger inserted with the wavy surface are divided into two parts;

- Validation with the smooth channel on flow and heat transfer
- Grid independence

The verifications on heat transfer and pressure loss are done by compared the present results with the values from the

correlations [13]. The deviations are around 0.3% and 0.7% for heat transfer and friction loss, respectively.

The numerical models with different grid numbers (80000, 120000, 180000, 240000 and 310000) for the square channel with the wavy surface at $Re = 3000$, $\alpha = 45^\circ$ are compared. The result reveals that the augmentation of grid from 120000 to 180000 gives nearly results on both heat transfer and pressure loss. Therefore, the grid around 120000 is applied for all domain of the present investigation.

6. NUMERICAL RESULT

Numerical results of the heating channel inserted with various flow attack angle of the wavy surface are separated into two sections; mechanisms in the tested channel and performance analysis. The mechanisms in the tested section are shown with the streamline in transverse planes, impinging jet on the channel walls, temperature contour, local Nusselt number contour and turbulent kinetic energy contour (*TKE*). The understandings on flow and heat transfer behaviors are key factor to help to design the compact heat exchanger. The performance analysis in the tested section are presented with the Nusselt number ratio (Nu/Nu_0), friction factor ratio (f/f_0) and thermal enhancement factor (*TEF*).

6.1 Flow and heat transfer mechanisms

Streamline in cross sectional planes in the square channel heat exchanger inserted with wavy surface at $Re = 4000$ and $\alpha = 45^\circ$ is depicted as Fig. 2. The vortex flow or swirling flow is found in all segments due to the insertion of the wavy surface in the tested section. The vortex flow can be separated into two zones; upper and lower flows. The core of the vortex flow depends on the position in the heating section. The vortex flow disrupts the thermal boundary layer on the heat transfer surface of the square channel. The disturbance of the thermal boundary layer is reason for heat transfer augmentation. The similar flow pattern is detected in all flow attack angles of the wavy surface, but the strength of the vortex flow is not equally.

Fig. 3 shows the streamline impinging on the sidewalls with the contour plot of local Nusselt number for the square channel at $Re = 4000$ and $\alpha = 45^\circ$. The high heat transfer area is plotted with red contour. The impingement of the flow on the channel walls extremely effects for heat transfer enhancement. The square channel inserted with various flow attack angle of the wavy surface provides nearly pattern of the heat transfer and flow mechanisms, but the strength of the impingement is not identically. The impingement of the flow is found in all sides of the channel as presented in Fig. 4 (impingement of the flow on the upper wall).

The thermal boundary layer disturbance is presented with the distribution of the temperature in transverse planes as Fig. 5 for the square channel inserted with wavy surface at $Re = 4000$ and $\alpha = 45^\circ$. For smooth channel, the red layer is found near the channel wall, while the blue layer is detected at the middle of the channel. The insertion of the wavy surface in the heating channel changes the heat transfer behavior. The red layer of the temperature performs thinner, while the blue layer distributes from the core to the channel wall. This means that the wavy surface makes better fluid mixing and creates the vortex flow, which disturbs the thermal boundary layer on the channel wall.

Fig. 6 shows turbulent kinetic energy (*TKE*) distribution in transverse plane of the square channel inserted with 45° of the wavy surface at $Re = 4000$. The *TKE* indicates the strength of the flow in the heating channel. As the figure, the high *TKE* is detected near upper-lower walls and right sidewall of the channel. Therefore, the peak of heat transfer areas is found at the upper-lower walls and right sidewall of the channel as seen in Fig. 7 which, presents the local Nusselt number distribution on the channel walls.

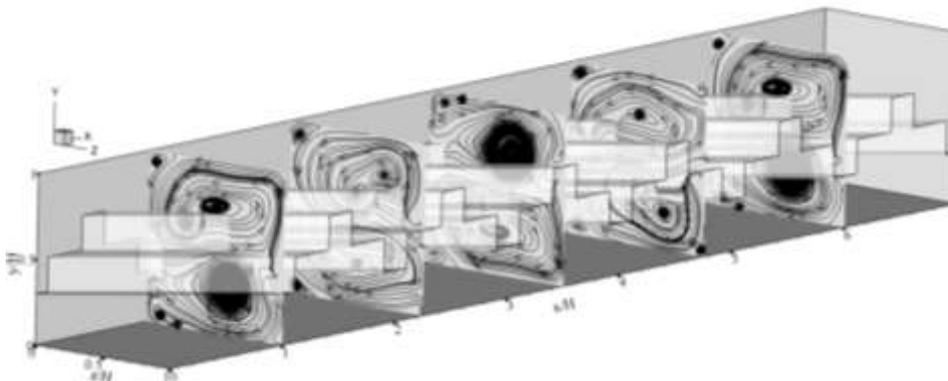


Figure 2: Streamline in cross sectional planes at various positions (x/H) for square channel inserted with inclined wavy surface at $Re = 4000$ and $\alpha = 45^\circ$.

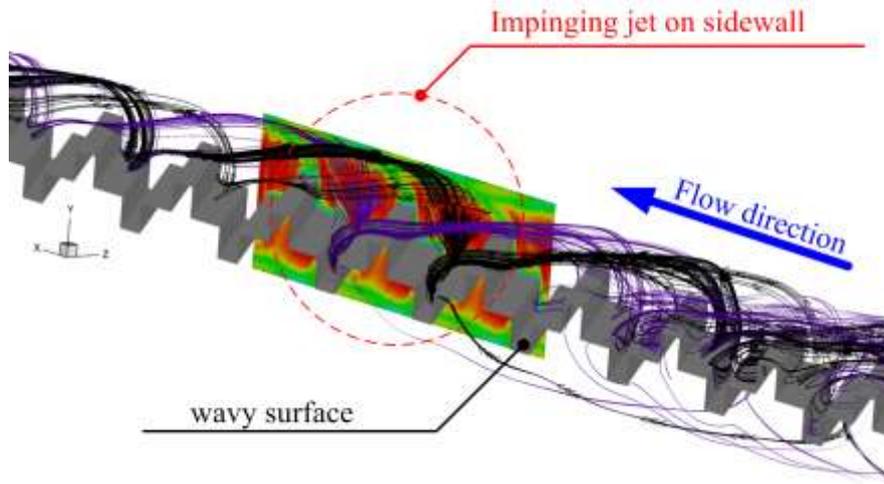


Figure 3: Impinging jet on the sidewall for square channel inserted with inclined wavy surface at $Re = 4000$ and $\alpha = 45^\circ$.

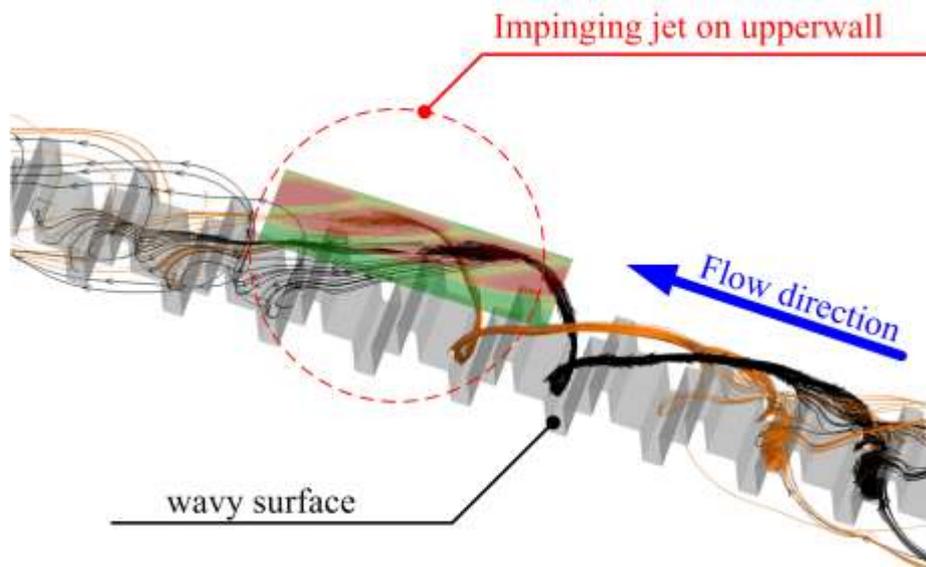


Figure 4: Impinging jet on the upper wall for square channel inserted with inclined wavy surface at $Re = 4000$ and $\alpha = 45^\circ$.

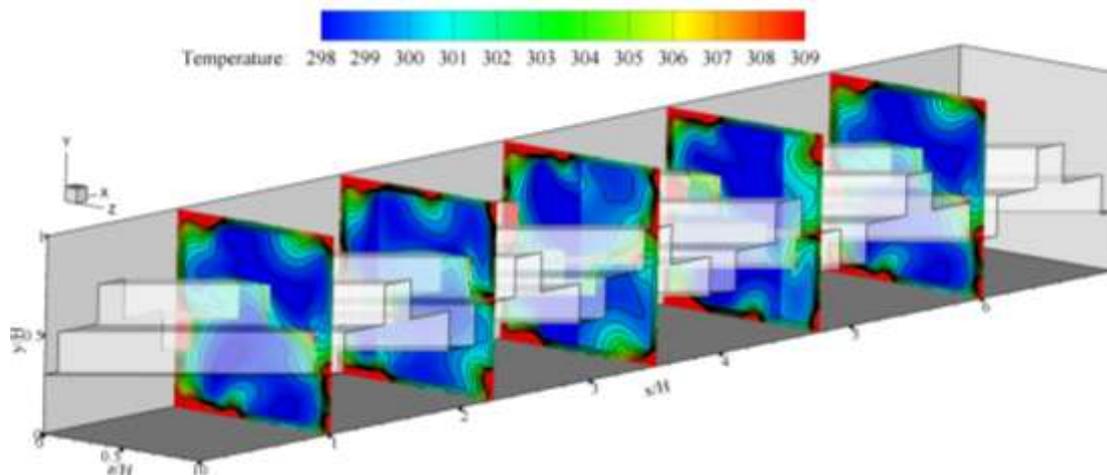


Figure 5: Temperature distributions in cross sectional planes at various positions (x/H) for square channel inserted with inclined wavy surface at $Re = 4000$ and $\alpha = 45^\circ$.

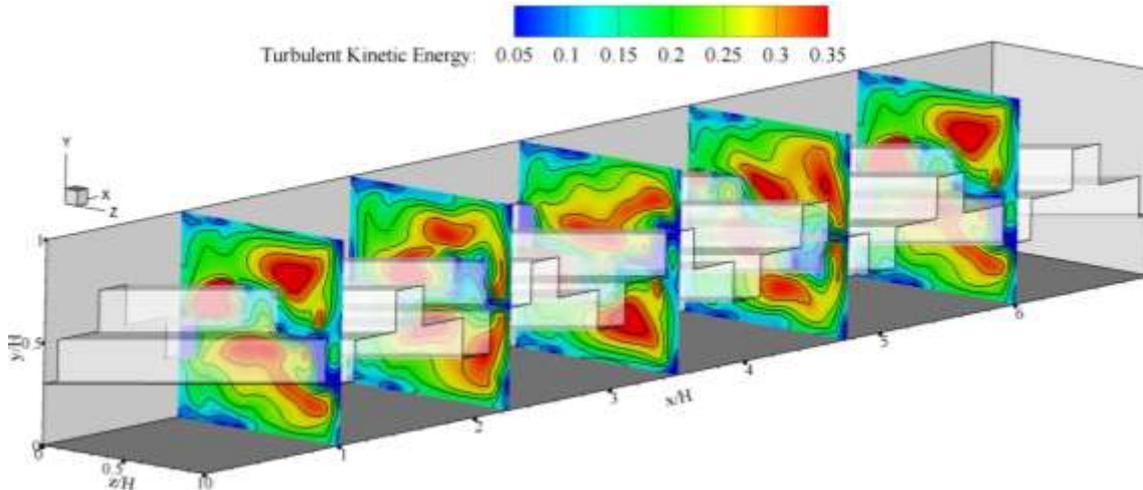


Figure 6: Turbulent kinetic energy (*TKE*) distributions in cross sectional planes at various positions (x/H) for square channel inserted with inclined wavy surface at $Re = 4000$ and $\alpha = 45^\circ$.

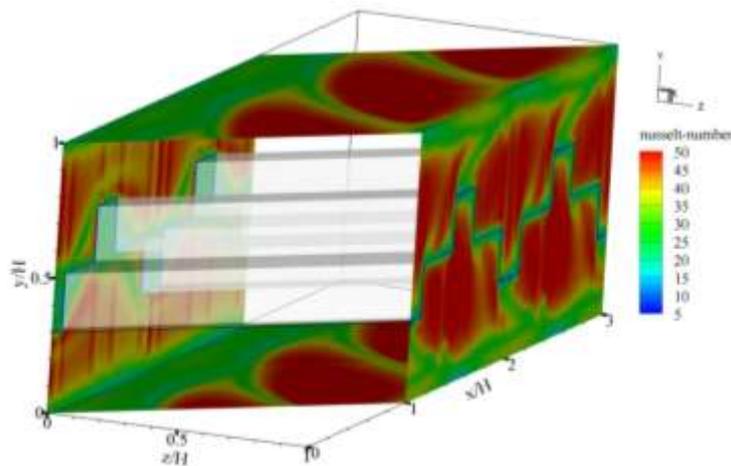


Figure 7: Local Nusselt number distributions on the channel walls for square channel inserted with inclined wavy surface at $Re = 4000$ and $\alpha = 45^\circ$.

6.2 Performance analysis

The performance analysis in the heating section inserted with various flow attack angles of the wavy surface is separated into three parts; Nusselt number ratio (Nu/Nu_0), friction factor ratio (f/f_0) and thermal enhancement factor (*TEF*).

Fig. 7 presented the relation of the Nu/Nu_0 with the flow attack angle. The addition of the wavy surface in the test section improves heat transfer rate higher than the smooth channel in all cases ($Nu/Nu_0 > 1$). In range $15^\circ \leq \alpha \leq 45^\circ$, the Nu/Nu_0 increases when growing the flow attack angle and Re . The heat transfer rate decreases when $\alpha > 45^\circ$. The $\alpha = 45^\circ$ provides the highest heat transfer rate, while the $\alpha = 15^\circ$ gives the reverse result. The reason is the $\alpha = 45^\circ$ can create the highest strength of the vortex flow in the tested channel. In range investigates the maximum and minimum heat transfer rates are around 6 and 2.2 times higher than the smooth channel, respectively.

Fig. 8 shows the variation of the f/f_0 with α at various Re for the square channel heat exchanger inserted with wavy surface. The insertion of the wavy surface in the test section gives higher friction loss than the smooth channel in all cases ($f/f_0 > 1$). The f/f_0 tends to increase when increasing α and Re . The maximum and minimum friction losses are detected at the flow attack angle of 60° and 15° , respectively. In range study, the f/f_0 is around 9 – 48 times above the smooth square channel.

Fig. 9 depicts the relation of the *TEF* with the flow attack angle of the wavy surface in the square channel heat exchanger. The *TEF* is calculated from the augmentations on both heat transfer rate and friction loss when inserting the wavy surface in the test section at similar pumping power. The *TEF* reduces when increasing the Reynolds number in all flow attack angles. The addition of the wavy surface in the square channel improves thermal performance higher than the smooth channel in all flow attack angles ($TEF > 1$). In range investigates the optimum *TEF* is found at the flow attack

angle of 30° and 45° around 1.9.

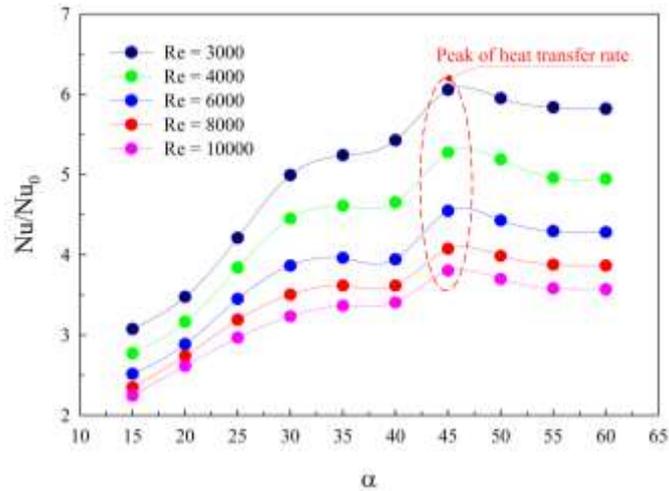


Figure 8: Variations of the Nu/Nu_0 with α .

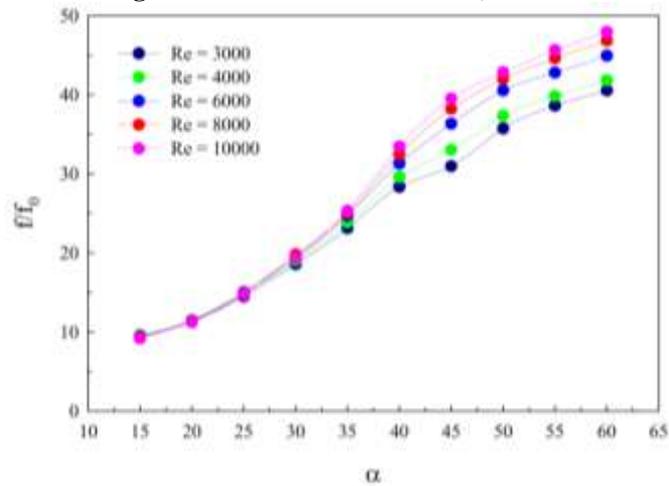


Figure 9: Variations of the ff_0 with α .

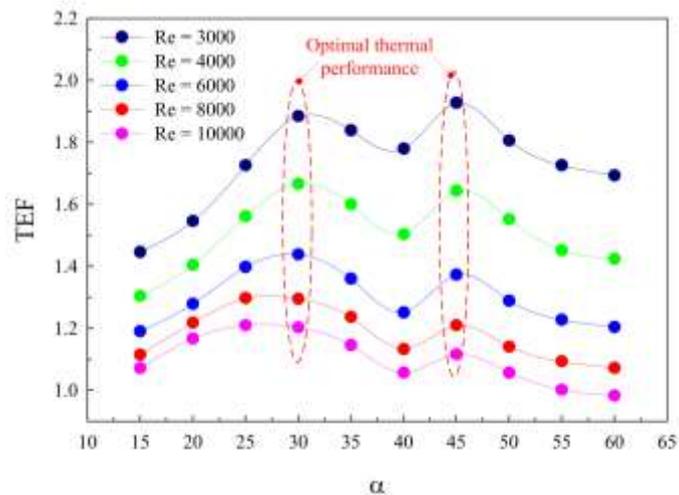


Figure 10: Variations of the TEF with α .

7. CONCLUSION

Numerical investigations on flow and heat transfer mechanisms in the square channel inserted with various flow attack angle of the wavy surface are presented. The turbulent flow with the Reynolds number in range 3000 – 10,000 is considered for the present study. The flow attack angles of the wavy surface are varied; 15°, 20°, 25°, 30°, 35°, 40°, 45°,

50°, 55° and 60°. The major findings are as follows;

The flow and heat transfer behaviors in the test section added with the wavy surface are similarly found, but the strengths of the vortex flow is not equal. The flow attack angle of 45° performs the highest vortex strength, while the flow attack angle around 15° provides the opposite result.

The addition of the wavy surface increases heat transfer rate, friction loss and thermal performance higher than the smooth square channel. The flow attack angle of 45° gives the highest Nusselt number, while the flow attack angle of 15° performs the reversed trend. The maximum and minimum friction losses are detected at the flow attack angle of 60° and 15°, respectively. The optimum *TEF* is found at the flow attack angle of 30° and 45°.

In range study, the heat transfer rate and friction loss are around 2.2 – 6 times and 9 – 48 times over the plain channel, respectively. The maximum *TEF* is around 1.9 at $Re = 3000$ for the flow attack angle of 30° and 45°.

NOMENCLATURE

H	channel height/hydraulic diameter of the channel
f	friction factor
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
Re	Reynolds number, $(\rho \bar{u} H / \mu)$
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	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 duct
w	wall
pp	pumping power

8. ACKNOWLEDGMENT

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