

Laminar Flow and Heat Transfer in a Square Channel Installed with Inclined Wavy Surface

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Abstract: Laminar flow and heat transfer characteristic in a square channel heat exchanger installed with inclined wavy surface are investigated numerically. The inclined wavy surface is designed with the main the aim to help the installation, production and maintenance when using the inclined wavy surface in the heat exchanger. The finite volume method with SIMPLE algorithm is selected to solve the present problem. The effects of flow attack angle ($\alpha = 15^\circ$ - 60°) and Reynolds number ($Re = 100$ - 1200) on heat transfer, friction loss and thermal performance are considered. The numerical model is validated with the smooth channel. The grid independence of the computational domain is checked. The numerical results show that the heat transfer enhancement is around 1.9-7.5 times above the smooth channel. The optimum thermal performance is found at the flow attack angle of 45° around 2.35 at $Re = 1200$.

Keywords: Inclined Wavy Surface, Heat Exchanger, Thermal Performance, Laminar Flow, Finite Volume Method

Introduction

The vortex flow, swirling flow and thermal boundary layer disturbance are the mechanisms when installed the vortex generators in the heat exchanger. These behaviors can help to improve the heat transfer rate and thermal performance in the heat exchanger. The enhancements of the heat transfer rate and thermal performance depend on the parameter, type, shape, etc. of the vortex generators.

Many researchers reported the investigations on flow and heat transfer in the heat exchanger inserted with the vortex generators, especially, inclined rib/baffle. Zheng *et al.* (2015) numerically investigated on flow and heat transfer mechanisms in a tube heat exchanger inserted with discrete double inclined ribs for $Re = 3390$ - 20340 . They presented that the augmentations on heat transfer rate and friction loss in the tube installed with the discrete double inclined ribs are around 1.8-3.6 and 2.1-5.6 times above the smooth tube, respectively, with the performance evaluation criterion around 1.3-2.3. They also reported that the optimum parameters of the discrete double inclined ribs are as follows; rib length ratio of 4, rib pitch ratio of 5 and rib inclination angle of

37.5° . Ary *et al.* (2012) reported the effects of inclined perforated baffle on heat transfer and flow pattern in a rectangular channel. They summarized that the two baffles perform higher heat transfer rate than the single baffle. Promvong *et al.* (2010) selected the 45° inclined baffles to enhance heat transfer rate and thermal performance in a square channel heat exchanger. They reported that the 45° inclined baffles can generate the vortex flow through the test section that the reason for heat transfer augmentation. They also pointed out that the optimum thermal enhancement factor is around 2.2 at the baffle to channel height ratio of 0.4 and $Re = 1200$. Yongsiri *et al.* (2014) numerically studied the influences of inclined detached ribs on heat transfer rate, pressure loss and thermal performance in a turbulent channel flow. They found that the flow attack angle of the rib is insignificant at low Reynolds number. Jedsadaratanachai *et al.* (2011) numerically investigated the heat transfer, friction factor and thermal enhancement factor in a square channel heat exchanger inserted with 30° inclined baffle on two opposite walls with inline arrangement. The effects of pitch to channel height ratio (0.5-2.5) were considered for $Re = 100$ -

2000. They detected that the heat transfer augmentation is around 1-9.2 times higher the smooth channel. Kwankaomeng and Promvonge (2010) studied the heat transfer and performance improvement in a square channel inserted with 30° inclined baffle on one wall. The effects of rib to channel height ratio, pitch to channel height ratio were considered for laminar region, $Re = 100-1000$. They found that the highest heat transfer rate is around 9.23 times above the smooth channel, while the optimum thermal performance is around 3.1 when considered at similar pumping power.

The wavy surface is another type of the turbulator, which always use in the fin-and-tube heat exchanger. For example, Lotfi *et al.* (2014) studied the mechanisms in the wavy fin-and-elliptical tube heat exchanger with various type vortex generators. Dong *et al.* (2013) experimentally examined a wavy-fin-flat-tube heat exchanger on thermo-hydraulic performance. They summarized that the amplitude and length of the wavy fin are most important factors for heat transfer rate improvement. Dong *et al.* (2010) also claimed that the waviness amplitude is a key for heat transfer rate augmentation, while the wavy fin profiles; triangular, sinusoidal and triangular round corner, have slightly effect for thermal performance. Gong *et al.* (2013) reported the numerical investigation on the heat transfer characteristic in a wavy fin-and-tube heat exchanger with combined rectangular winglet pairs. They detected that the combined vortex generators perform larger and stronger vortex flow than the single rectangular winglet pairs. Du *et al.* (2014) studied the heat transfer enhancement in a wavy fin-and-flat-tube heat exchanger with punched longitudinal vortex generators. They found that the best thermal performance of the system is around 1.23. Du *et al.* (2013) experimentally studied the flow pattern in a wavy fin-and-flat-tube heat exchanger at $Re = 1500-4500$. They stated that the Nusselt number and friction factor are around 21-60 and 13-83%, respectively, while the thermal performance is around 1.31.

As the literature reviews above, it is found that the inclined baffle in the heat exchanger has high effective to enhance heat transfer rate and thermal performance. However, the difficulty of the production, installation and maintenance for the baffle in the heat exchanger is found in real system. In the present research, the design of the vortex generators is improved. The combination concepts between inclined rib and wavy surface (called "inclined wavy surface") are presented. The inclined wavy surface is inserted in the middle of the square channel heat exchanger to enhance heat transfer rate and thermal performance. The design of inclined wavy surface focuses on the generator production, maintenance, installation and remains the thermal performance as inclined rib. The influences of the flow attack angles for

the inclined wavy surface are considered at laminar regime, $Re = 100-1200$. The numerical method is selected to study the present problem and to describe the mechanism in the heating section.

The contents of the present work are as follows:

- Introduction
- Physical model
- Assumption and boundary condition
- Mathematical foundation
- Model validation
- Numerical result
 - Flow and heat transfer mechanism
 - Performance assessment
- Conclusion

Physical Model

Physical model of the square channel heat exchanger installed with inclined wavy surface is presented in Fig. 1. The height of the square channel, H , is around 0.05 m. The hydraulic diameter of the square channel and periodic module (L) are equal to H . The flow attack angles of the inclined wavy surface are varied; 15°, 20°, 25°, 30°, 35°, 40°, 45°, 50°, 55° and 60°. The laminar flow with the Reynolds number around 100-1200 is considered. The computational model of the square channel heat exchanger inserted with inclined wavy surface is also depicted in the Fig. 1.

Assumption and Boundary Condition

The assumptions for the present study are as follows:

- Flow and heat transfer are steady in three dimensions
- Flow is laminar and incompressible
- Body force, viscous dissipation, radiation heat transfer and natural convection are disregarded
- The test fluid is air with 300 K ($Pr = 0.707$)
- The thermal properties of the fluid remain constant at average bulk mean temperature

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

- Inlet and outlet of the domain are set with periodic boundary
- Constant temperature around 310 K is applied for all sides of the channel walls
- No slip wall condition is used for all surfaces of the computational domain
- The inclined wavy surface is assumed as adiabatic wall (insulator)

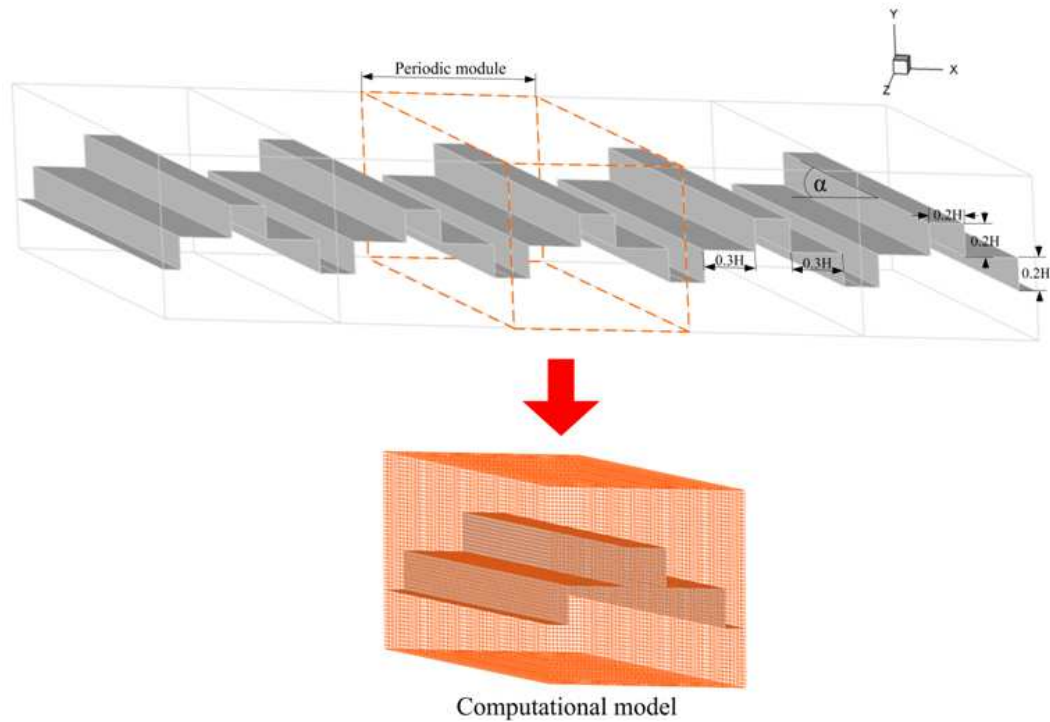


Fig. 1. Physical domain of the square channel heat exchanger installed with inclined wavy surface and computational domain

Mathematical Foundation

The channel flow is governed by the continuity, the Navier-Stokes equations and the energy equation as Equation 1-3, respectively.

Continuity equation:

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

Momentum equation:

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] \quad (2)$$

Energy equation:

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

Γ is the thermal diffusivity, which equal to:

$$\Gamma = \frac{\mu}{Pr} \quad (4)$$

The energy equation is discretized by the QUICK scheme, while the governing equations are discretized

by power law scheme. The present problem is solved by finite volume method with SIMPLE algorithm. 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. The important parameters are Reynolds number, friction factor, local Nusselt number, average Nusselt number and thermal enhancement factor.

The Reynolds number is calculated by:

$$Re = \rho \bar{u} H / \mu \quad (5)$$

The friction factor, f , is measured by pressure drop, Δp , across the periodic module, L :

$$f = \frac{(\Delta p / L) D}{\frac{1}{2} \rho \bar{u}^2} \quad (6)$$

The local heat transfer is written as:

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

The average Nusselt number can be obtained by:

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

The Thermal Enhancement Factor (*TEF*) is calculated by the augmentations on both heat transfer and friction factor at similar pumping power:

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

The Nu_0 and f_0 is the Nusselt number and friction factor for the smooth square channel, respectively.

Model Validation

The computational domain (with mesh around 240000) of the smooth square channel is validated on both Nusselt number and friction factor as Fig. 2. The

deviations of the Nusselt number and friction factor are around ± 0.02 and $\pm 0.035\%$, respectively.

The grid independences of the square channel heat exchanger installed with 45° inclined wavy surface for heat transfer and friction factor are presented in Fig. 3a and 3b, respectively. The difference numbers of grid cells; 80000, 120000, 240000, 360000 and 480000, are compared. The numerical results reveal that the increasing cells from 240000 to 360000 has a few effects on both flow and heat transfer. Therefore, the grids around 240000 cells are applied for all cases of the computational domain of the square channel heat exchanger installed with inclined wavy surface. The optimum grid of the computational domain can help to save time for investigation and computational resource.

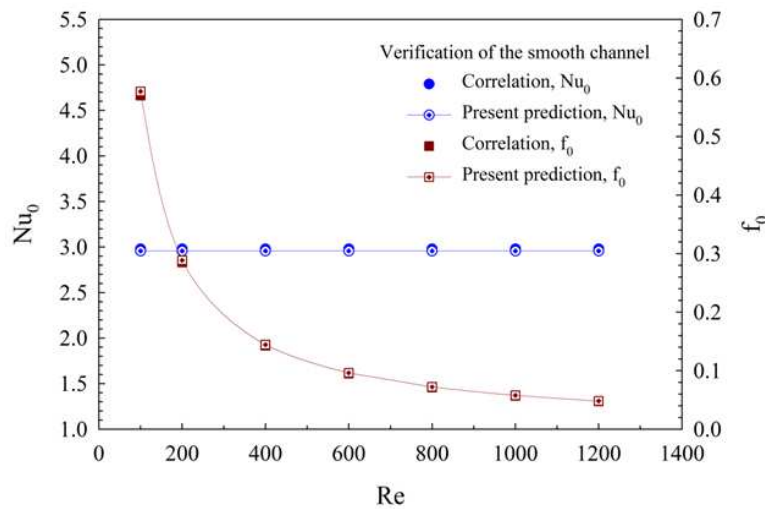


Fig. 2. Validations of the smooth square channel on heat transfer and friction factor

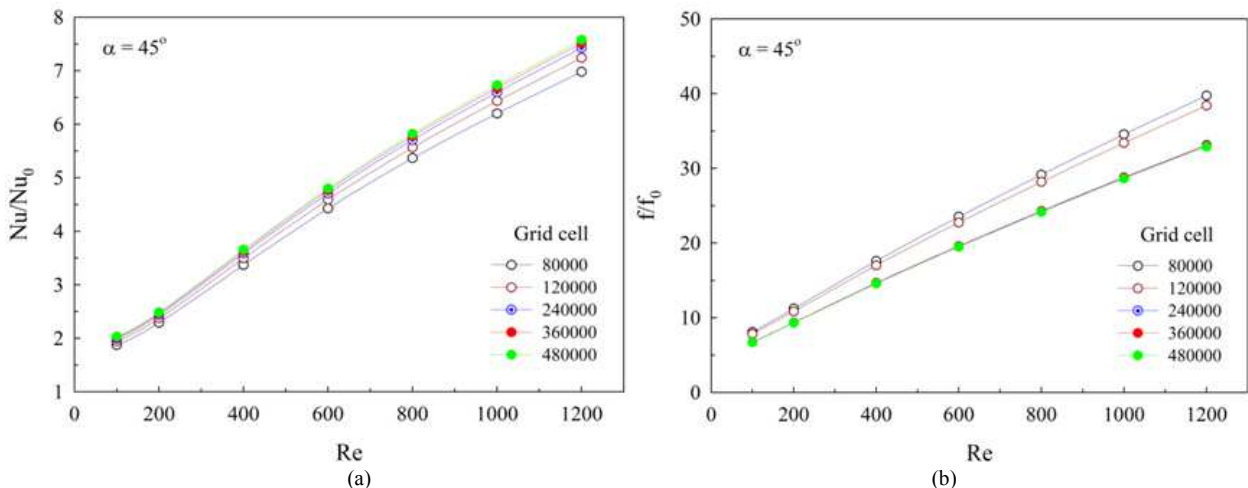


Fig. 3. Grid independences of the square channel heat exchanger installed with 45° inclined wavy surface for (a) Nusselt number ratio and (b) friction factor ratio

Numerical Result

Flow and Heat Transfer Mechanism

The flow topology in the square channel heat exchanger installed with inclined wavy surface is presented with tangential velocity vector in transverse planes and longitudinal vortex flow, while the heat transfer mechanism is plotted with temperature distributions in transverse planes and local Nusselt number distributions on the channel walls. The mechanisms in the test section are important data to develop the compact heat exchangers.

Figure 4 reports the tangential velocity vector in y - z planes of the square channel heat exchanger inserted with 45° inclined wavy surface at $Re = 800$. As the figure, the inclined wavy surface generates the vortex

flow through the test section. The core of the vortex depends on the x -position in the square channel. The generation of the vortex flow in the heating section is the thermal boundary layer disturbance that results in the increment of the heat transfer rate and thermal performance. The strength of the vortex flow depends on the flow attack angle of the inclined wavy surface and the Reynolds number. Figure 5 shows the longitudinal vortex flow in the square channel heat exchanger inserted with 45° inclined wavy surface at $Re = 800$. It clearly seen in the figure that the vortex flow disturbs the thermal boundary layer on the channel wall that the cause for heat transfer augmentation. In conclusion, the similar flow topology is found in all cases, but the vortex strength is not equal.

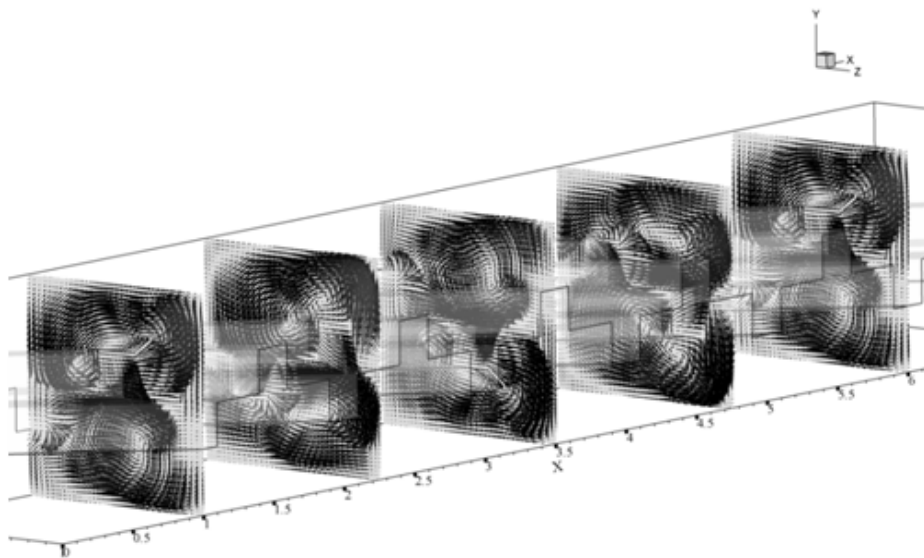


Fig. 4. Tangential velocity vector in y - z plane of the square channel heat exchanger installed with 45° inclined wavy surface at $Re = 800$

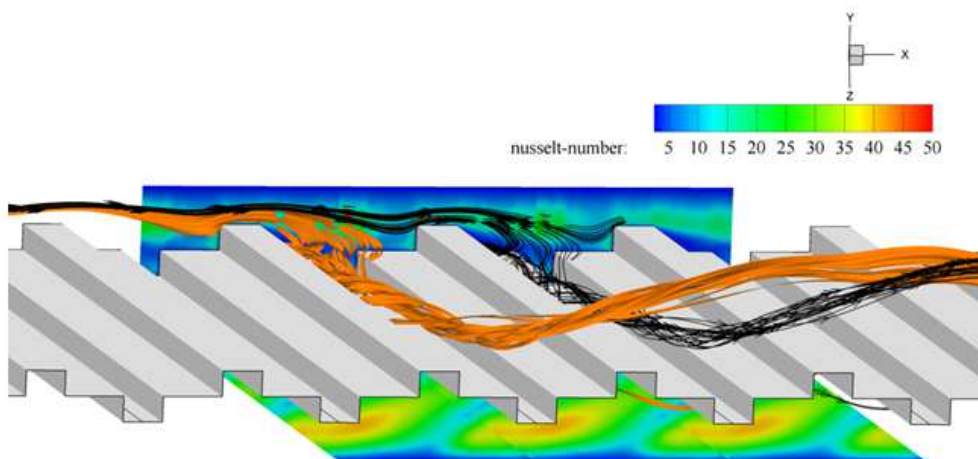


Fig. 5. Longitudinal vortex flow of the square channel heat exchanger installed with 45° inclined wavy surface at $Re = 800$

The temperature distributions in y - z planes in the square channel heat exchanger installed with 45° wavy surface at $Re = 800$ is illustrated in Fig. 6. From the figure, the blue layer (low temperature) distributes from the center of the square channel, while the red layer (high temperature) near the channel wall performs thinner. This means that the inclined wavy surface provides better fluid mixing between the core of the channel and near the heat transfer surface. The reduction of high temperature layer near the channel wall also

indicates the disturbance of the thermal boundary layer. Figure 7a-7j present the local Nusselt number on the channel wall that inserted with inclined wavy surface for $\alpha = 15^\circ, 20^\circ, 25^\circ, 30^\circ, 35^\circ, 40^\circ, 45^\circ, 50^\circ, 55^\circ$ and 60° , respectively, at $Re = 800$. The peak of heat transfer rate is found similarly at the right sidewall of the square channel heat exchanger for all flow attack angles of the inclined wavy surface. This means that the right sidewall of the square channel inserted with the wavy surface is extremely disturbed by the vortex flow.

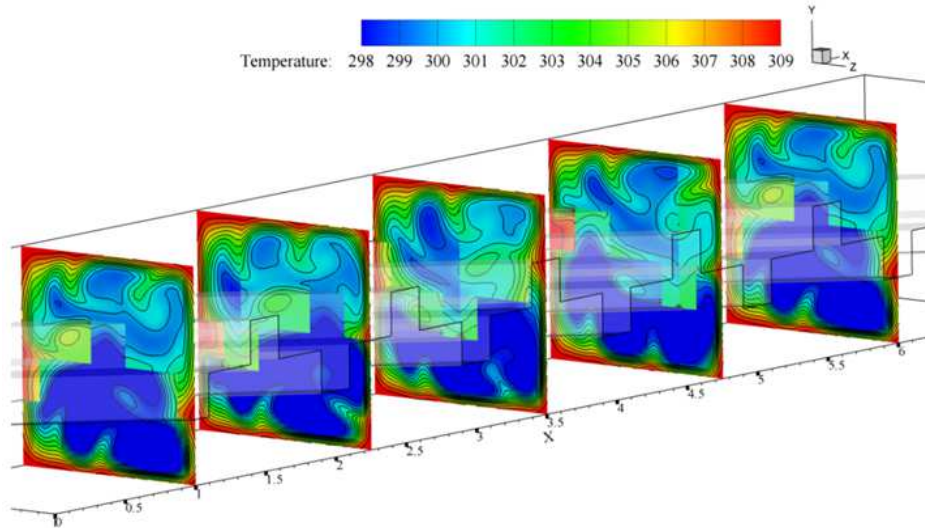
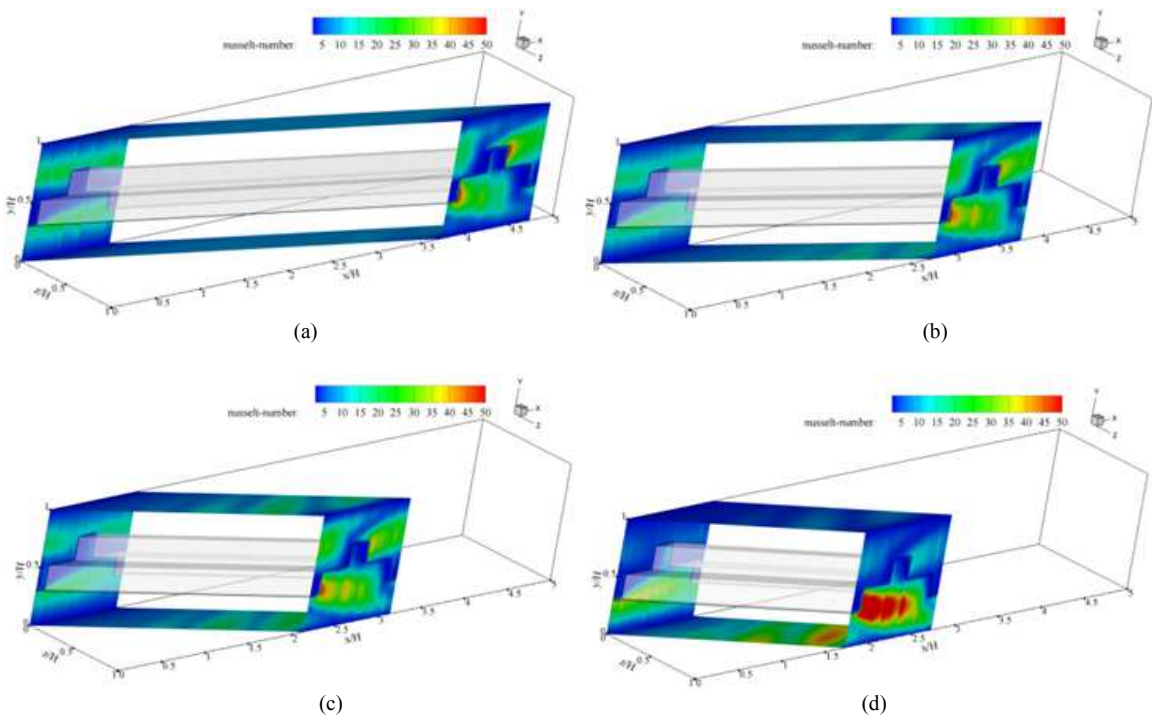


Fig. 6. Temperature distributions in y - z plane of the square channel heat exchanger installed with 45° inclined wavy surface at $Re = 800$



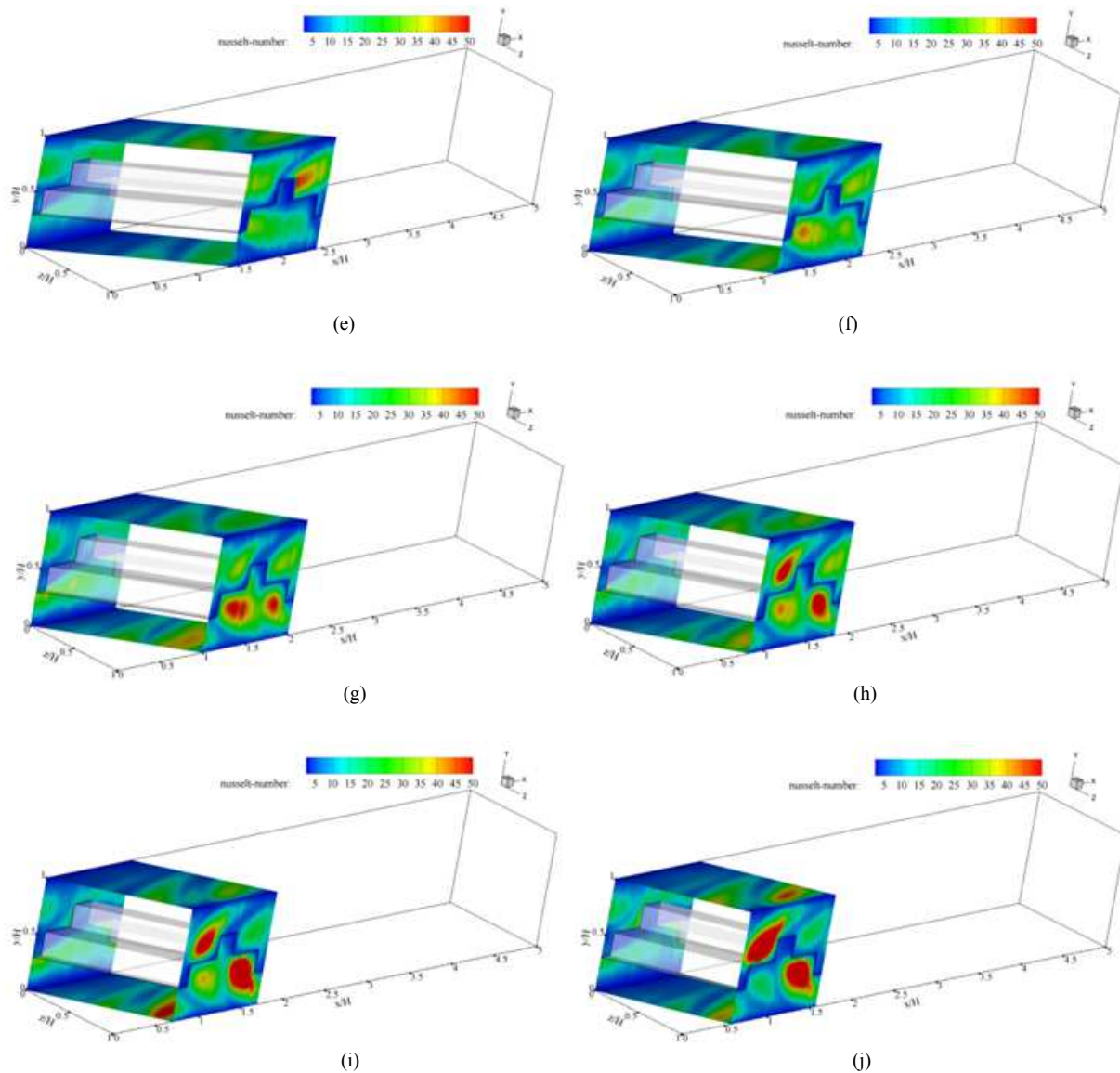


Fig. 7. Local Nusselt number distributions on the channel walls of the square channel heat exchanger inserted with inclined wavy surface at $Re = 800$ for (a) $\alpha = 15^\circ$, (b) $\alpha = 20^\circ$, (c) $\alpha = 25^\circ$, (d) $\alpha = 30^\circ$, (e) $\alpha = 35^\circ$, (f) $\alpha = 40^\circ$, (g) $\alpha = 45^\circ$, (h) $\alpha = 50^\circ$

Performance Assessment

Performance assessments in the square channel heat exchanger inserted with various flow attack angle of the inclined wavy surface are reported in terms of Nusselt number ratio (Nu/Nu_0), friction factor ratio (f/f_0) and Thermal Enhancement Factor (TEF).

Figure 8a-8c report the relations of the Nu/Nu_0 , f/f_0 and TEF , respectively, with the Reynolds numbers at various flow attack angles of the wavy surface. In general, the addition of the inclined wavy surface provides higher heat transfer rate and friction loss higher than the plain channel. The heat transfer rate, friction loss and thermal performance tend to increase

when increasing the Reynolds number for all cases. The maximum values of the Nusselt number, friction factor and thermal enhancement factor are found at $Re = 1200$, while the minimum values are detected at $Re = 100$. The strength of the vortex flow increases when augmenting the Reynolds number.

Figure 9a-9c plot the variations of the Nu/Nu_0 , f/f_0 and TEF with the flow attack angle of the inclined wavy surface, respectively, at various Reynolds number. At $Re = 100-200$, the Nusselt number values are found closely for all flow attack angles. When $Re > 200$, the flow attack angle of 45° performs the highest heat transfer rate, while the flow attack angle

of 15° provides opposite result. This means that the 45° inclined wavy surface can generate toughest strength of the vortex flow. In addition, the insertion of the inclined wavy surface in the square channel heat exchanger can enhance heat transfer rate around 1.9-7.5 times above the smooth channel with no wavy surface for $Re = 100-1200$ and $\alpha = 15^\circ-60^\circ$. The friction factor increases when enhancing the flow

attack angle. The 60° inclined wavy surface shows the highest friction loss, while the 15° inclined wavy surface performs the reverse result. In the range investigates, the friction factor is around 5-40 times over the smooth channel depended on Re and α . At similar pumping power, the maximum TEF is detected at 45° inclined wavy surface around 2.35 when considered at $Re = 1200$.

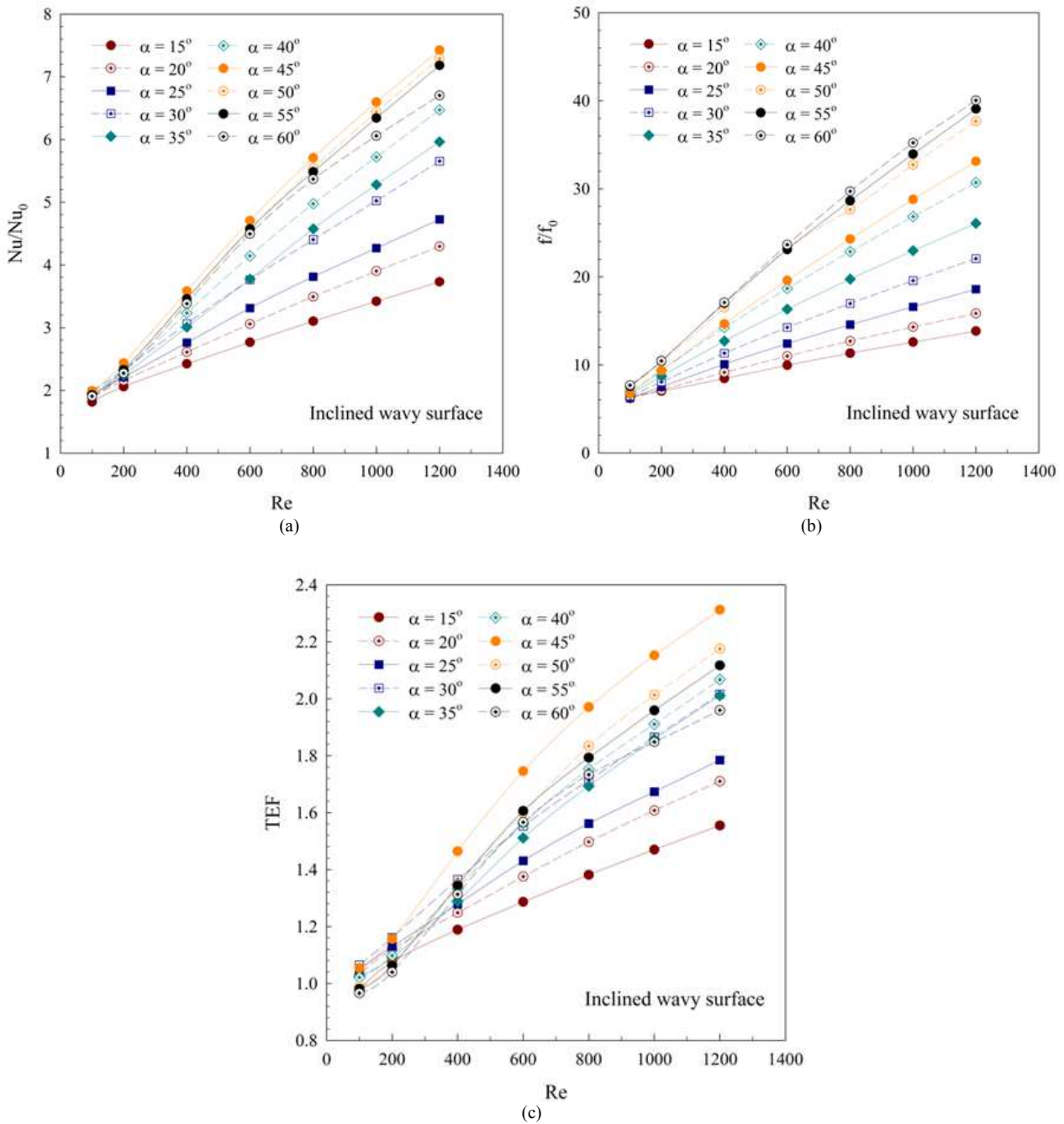


Fig. 8. (a) Nu/Nu_0 Vs Re , (b) f/f_0 Vs Re and (c) TEF Vs Re

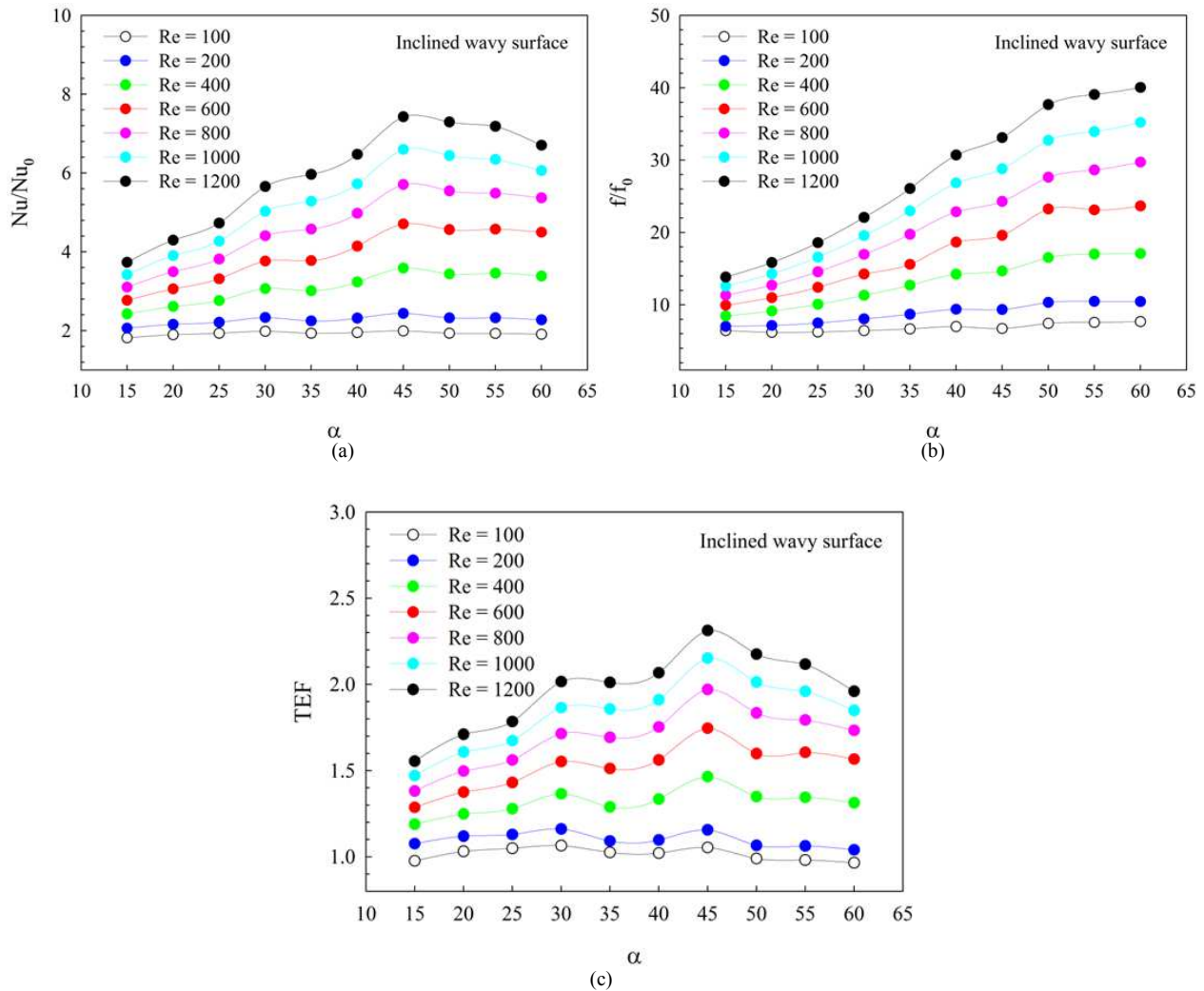


Fig. 9. (a) Nu/Nu_0 Vs α , (b) f/f_0 Vs α and (c) TEF Vs α

Conclusion

Numerical predictions on laminar flow, heat transfer and thermal performance in the square channel heat exchanger installed with various flow attack angles of the inclined wavy surface are presented. The effects of the flow attack angles ($\alpha = 15^\circ$ - 60°) and Reynolds numbers (laminar, $Re = 100$ - 1200) are considered. The major findings are concluded as follows;

The insertion of the inclined wavy surface in the square channel heat exchanger can improve the heat transfer rate and thermal performance higher than the smooth channel due to the inclined wavy surface can create the vortex flow that disturbs the thermal boundary layer on the heat transfer surface.

The augmentation of the heat transfer rate is around 1.9-7.5 times above the plain channel depended on the Reynolds numbers and flow attack angles. At similar

pumping power, the optimum thermal performance is detected at 45° inclined wavy surface around 2.35 when consider at the highest Reynolds number, $Re = 1200$.

The design of the inclined wavy surface can help to manufacture and install in real system in comparison with other turbulators.

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Ethics

This article is original and contains unpublished material. The corresponding author confirms that all of the other authors have read and approved the manuscript and no ethical issues involved.

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Nomenclature

H	duct height/hydraulic diameter of the square duct, m
f	friction factor
h	convective heat transfer coefficient, $W m^{-2} K^{-1}$
L	periodic length, m
Nu	Nusselt number
P	static pressure, Pa
Pr	Prandtl number
Re	Reynolds number, $(\rho u_0 H / \mu)$
T	temperature, K
TEF	thermal performance enhancement factor, $(Nu/Nu_0) / (f/f_0)^{1/3}$

Greek Letter

μ	dynamic viscosity, $kg s^{-1} m^{-1}$
ρ	density, $kg m^{-3}$
α	flow attack angle

Subscript

0	smooth duct
pp	pumping power