



NUMERICAL PREDICTIONS ON FLOW AND HEAT TRANSFER IN HEAT EXCHANGER TUBE EQUIPPED WITH VARIOUS FLOW ATTACK ANGLES OF INCLINED-WAVY SURFACE

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ABSTRACT

Numerical analysis on flow configuration, heat transfer behavior and thermal performance in the heat exchanger tube equipped with various flow attack angles of the inclined wavy surface are presented. The laminar flow ($Re = 100 - 1200$) and turbulent flow ($Re = 3000 - 10000$) are considered for the present investigation. The flow attack angles of the inclined wavy surface are varied as $15^\circ - 60^\circ$. The finite volume method with SIMPLE algorithm is selected to evaluate the current problem. The numerical results are reported in terms of flow and heat transfer mechanisms. The performance evaluations in forms of the Nusselt number ratio (Nu/Nu_0), friction factor ratio (f/f_0) and thermal enhancement factor (TEF) are also concluded. As the results, the vortex flow, impinging flow and thermal boundary layer disturbance are detected when inserted the inclined wavy surface in the heat exchanger tube. These behaviors effect for the augmentation of the heat transfer rate, pressure loss and thermal performance. The optimum flow attack angle of the inclined wavy surface for the laminar and turbulent flows are also concluded.

Keywords: flow attack angle, heat transfer, wavy surface, thermal performance, heat exchanger.

1. INTRODUCTION

The development of the heat exchanger in various industries is very important for the energy saving process. The thermal performance improvement in the heat exchanger can do by various methods. The insertion of the vortex generator or turbulator in the heat exchanger or tube/channel heat exchanger is one of the methods to enhance heat transfer rate and thermal performance. The generator can produce the vortex flow and impinging flow, which disturbs the thermal boundary layer on the heat transfer surface. The type, shape, dimension, arrangement, etc., of the vortex generators effect for the flow structure and heat transfer behavior. The examples for the investigations on heat transfer enhancement in the heat exchangers with various types of the vortex generators are performed as follows.

Menasria *et al.* (2017) investigated the turbulent flow and heat transfer in a solar heater channel with rectangular baffle. The Reynolds number, blockage ratio and pitch ratio were varied. They reported that the blockage ratio of 0.7 with the pitch spacing ratio of 2 gives the highest thermo-hydraulic performance factor around 0.875. The augmentations on heat transfer and friction factor are around 2.16 and 15.95 times above the smooth channel, respectively. Alam and Kim (2016) numerically studied thermal performance in a solar air heater duct with semi ellipse shaped obstacles. The influences of the flow attack angle ($30^\circ - 90^\circ$), arrangement (inline and staggered) and Reynolds number (6000 - 18,000) on heat transfer and friction loss were presented. They concluded that the flow attack angle of 75° performs the greatest heat transfer rate. They also stated that the maximum heat transfer rate and friction factor are around 2.05 and

6.93, respectively for staggered arrangement, while around 1.73 and 6.12, respectively, for inline arrangement. Chai *et al.* (2016a) presented laminar flow and thermo-hydraulic performance in a microchannel heat sink with fan shaped ribs. They reported that the average Nusselt number increases around 6 - 101% and the total thermal resistance decreases around 3 - 40% for the aligned fan-shaped ribs, while the average Nusselt number enhances around 4 - 103% and total thermal resistance reduces around 2 - 42% for the offset fan-shaped ribs when compared with the smooth channel. Rajaseenivasan *et al.* (2015) compared between circular and V-shaped tabulators on heat transfer and pressure loss in a solar air heater channel. Chai *et al.* (2016b) numerically investigated on convective heat transfer in a microchannel heat sink with offset ribs for laminar flow regime ($190 \leq Re \leq 838$). They summarized that the Nusselt number and friction factor are around 1.42 - 1.95 and 1.93 - 4.57 times above the smooth channel, respectively, while the performance evaluation criteria is around 1.02 - 1.48. Zhan and Park (2016) presented the effects of plate angle on flow bifurcations and heat transfer characteristics in a channel with inclined plates. The flow configuration and heat transfer characteristic with different angle of the inclined plate were concluded. Sabzpooshani *et al.* (2014) reported the exergetic performance evaluation of a single pass baffled solar air heater. They found that the addition of the fin and baffle can lead to noticeable enhancement of the exergy efficiency. Hong *et al.* (2017) studied the heat transfer and flow structure in a wavy corrugated tube for $Re = 7500 - 20,000$. They showed that the wavy configuration can help to improve heat transfer rate and thermal performance when compared with the smooth tube. They also reported that the greatest performance evaluation criterion for the wavy corrugated tube is around 1.56.

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Sawhney *et al.* (2017) experimentally investigated the pressure loss, heat transfer configuration and thermal performance in a solar air heater with wavy delta winglet. The number of wave (3, 5 and 7), Reynolds number (4000 – 17,300), arrangement (inline and staggered) and relative longitudinal pitch (3, 4, 5 and 6) were varied. They reported that the maximum Nusselt number is around 223% over the flat plate with the friction loss around 10.3 times above the base case. They also stated that the optimum thermo-hydraulic performance is found to be around 2.09. Li *et al.* (2016) numerically reported the laminar flow and heat transfer in a microchannel heat sink with triangular cavities and rectangular ribs for $Re = 173 - 635$. The influences of relative rib width and relative cavity width on the flow structure and performance were investigated. Shirvan *et al.* (2017) summarized the effects of wavy surface on natural convection heat transfer in a cosine corrugated square cavity filled with nanofluid. Xu *et al.* (2015) experimentally investigated on thermal performance enhancement in a wavy finned flat tube heat exchanger for the Reynolds number in the range 1340 – 13,476. They stated that the Nusselt numbers of discontinuous type, staggered type and vortex-generator type are increased by 11.29%, 28.61% and 56.46% on average, and the friction factors are increased by 3.23%, 66.26% and 48.58% on average, respectively. Xiao *et al.* (2017) studied the augmentations on heat transfer and performance in a wavy finned flat tube by water spray cooling for $Re = 210 - 680$. They claimed that the Nusselt numbers of three water flow rates are increased by 48 - 68% on average when compared to the case without spray. Du *et al.* (2014) performed the investigations on heat transfer augmentation in a wavy finned flat tube with punched longitudinal vortex generators. The flow attack angle of the delta winglet vortex generators was considered. They claimed that the optimum performance evaluation

criterion is detected at the flow attack angle of 25° around 1.23. Khan *et al.* (2015) studied thermal performance of 3D wavy channel based printed circuit heat exchanger. They stated that the wavy surface of the channel can improve the heat transfer rate and thermal performance higher than the plain channel.

In the present work, the combinations of the inclined baffle and wavy surface called “inclined wavy surface” are presented. The inclined wavy surface is equipped in the heat exchanger tube to change the flow structure. The presumption for the present research, the inclined wavy surface can create the flow, which disturbs the thermal boundary layer on the heat transfer surface. The disturbance of the thermal boundary layer is the reason for heat transfer and thermal performance augmentations. The influences of the flow attack angle for the inclined wavy surface on heat transfer, pressure loss and thermal performance are considered. The investigations are done on both laminar and turbulent flows with the Reynolds number in range 100 – 1200 and 3000 – 10,000, respectively.

2. PHYSICAL MODEL

The inclined wavy surface with various flow attack angles ($\alpha = 15^\circ, 20^\circ, 25^\circ, 30^\circ, 35^\circ, 40^\circ, 45^\circ, 50^\circ, 55^\circ$ and 60°) is inserted in the heat exchanger tube as Fig. 1a, while the computational domain for the present investigation is depicted as Fig. 1b. The profile of the inclined wavy surface is square with $0.2D \times 0.2D$. The laminar flow with the Reynolds number around 100 – 1200 and the turbulent flow with the Reynolds number around 3000 – 10,000 are considered.

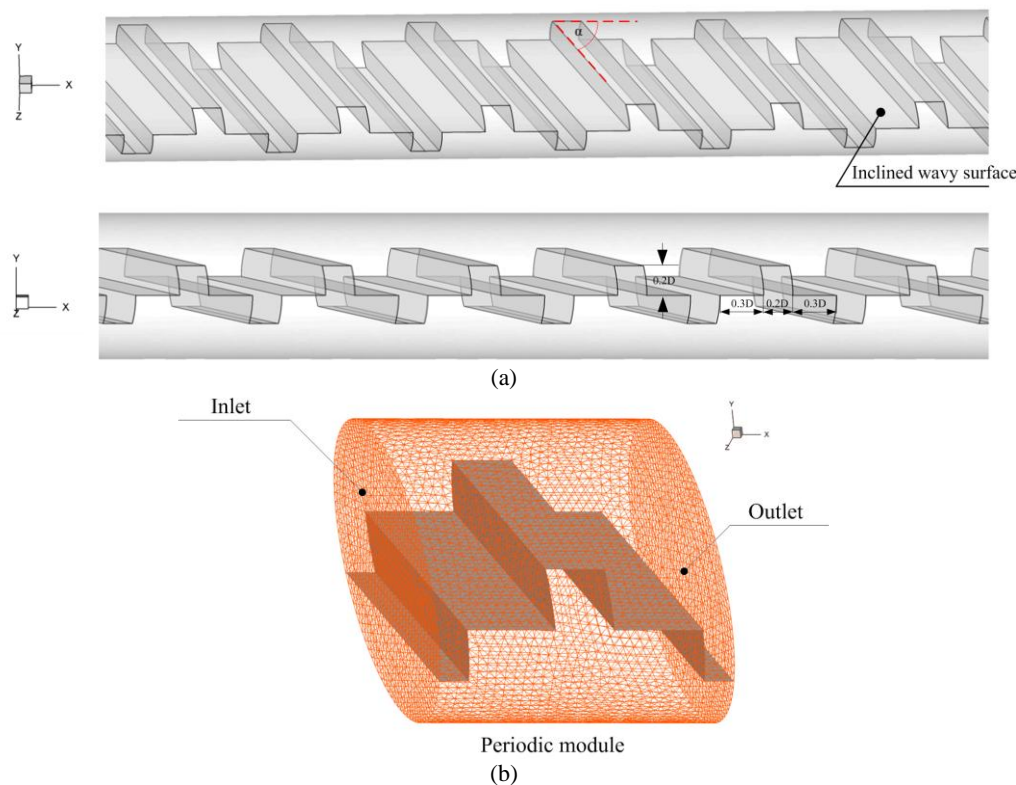


Fig. 1 (a) heat exchanger tube equipped with the inclined wavy surface and (b) computational domain.

3. BOUNDARY CONDITION AND ASSUMPTION

The assumptions for the present investigation are as follows;

1. The flow and heat transfer are steady in three dimensions.
2. The test fluid is air at 300K with the Prandtl number around 0.707.

3. The air is set as incompressible fluid on both laminar and turbulent flows.
4. The thermal properties of the air assume to be constant at the average bulk mean temperature.
5. The forced convective heat transfer is considered, while the natural convection and radiation are ignored.
6. The body force and viscous dissipation are uncouncted.

The boundary conditions for the computational domain on both laminar and turbulent flow are concluded as Table. 1

Table 1 Boundary condition for the computational domain of the heat exchanger tube equipped with the inclined wavy surface

Zone	Boundary condition
Inlet	Periodic
Outlet	Periodic
Tube wall	Constant temperature at 310K, no slip wall condition
Inclined wavy surface	Insulator, no slip wall condition

4. MATHEMATICAL FOUNDATION

The heat exchanger tube equipped with the inclined wavy surface is governed by the continuity, the Navier–Stokes equations and the energy equation.

For laminar flow, the energy equation is discretized by the SOU scheme, while the governing equations are discretized by power law scheme. All governing equations are discretized with the SOU numerical scheme for the turbulent flow. The current investigation is resolved by finite volume method with SIMPLE algorithm. The solutions are measured 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 realizable k-ε turbulent model is selected for the turbulent part:

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

and:

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

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 constant values are as follows:

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

The main parameters are Reynolds number, friction factor, local Nusselt number, average Nusselt number and thermal enhancement factor.

The Reynolds number is calculated as:

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

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 are the Nusselt number and friction factor for the smooth circular tube (Cengel and Ghajar (2015)), respectively.

$$Nu_0 = 3.66, \text{ laminar flow} \quad (10)$$

$$f_0 = 64 / Re, \text{ laminar flow} \quad (11)$$

$$Nu_0 = 0.023 Re^{0.8} Pr^{1/3}, \text{ turbulent flow} \quad (12)$$

$$f_0 = (0.79 \ln Re - 1.64)^{-2}, \text{ turbulent flow} \quad (13)$$

5. NUMERICAL VALIDATION

The validations of the numerical model on flow and heat transfer are an important part for the numerical investigation. The numerical validations include grid independence and verification of the smooth tube (on both flow and heat transfer). The five sets of the numerical model ($\alpha = 30^\circ$ with $Re = 600$ and 6000 for laminar and turbulent flows, respectively) with different cells, 80000, 120000, 240000, 320000 and 480000 are compared. The numerical results reveal that the Nusselt number and friction factor values for the numerical models with grid around 120000 – 480000 are not different on both laminar and turbulent regions. The grid around 120000 is selected for all cases of the present investigation.

Figs. 2 and 3 report the verifications of the smooth tube for laminar and turbulent flows, respectively. The differences of the Nusselt number and friction factor between the correlation values and present values are around $\pm 0.03\%$ and $\pm 0.05\%$, respectively, for laminar flow and around $\pm 5\%$ and $\pm 11\%$, respectively, for turbulent flow. As the results above, it can be concluded that the numerical model of the heat exchanger tube with wavy surface has reliability to predict flow and heat transfer mechanisms.

6. NUMERICAL RESULT

The numerical results are separated into two parts; laminar and turbulent flows. The flow configuration, heat transfer behavior and thermal performance evaluation are stated in each part. The content for the numerical result is illustrated as follows;

- Laminar flow
 - Flow and heat transfer behavior
 - Performance analysis
- Turbulent flow
 - Flow and heat transfer behavior
 - Performance analysis

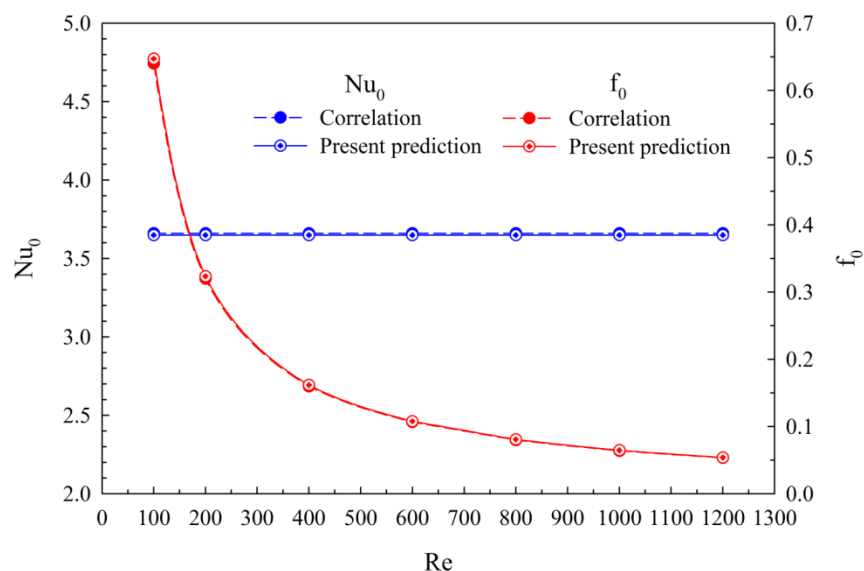


Fig. 2 Verifications on heat transfer and friction loss of the smooth circular tube for laminar regime.

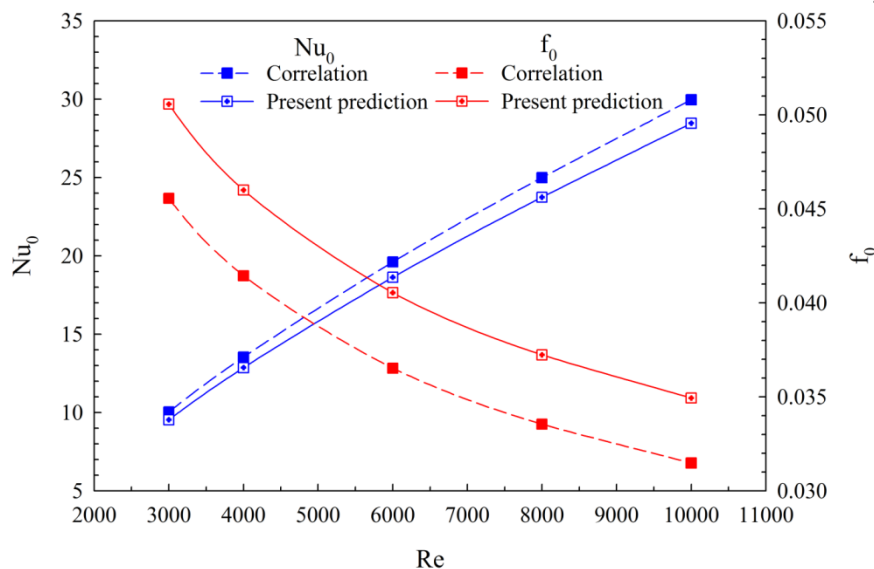


Fig. 3 Verifications on heat transfer and friction loss of the smooth circular tube for turbulent regime.

6.1 Laminar flow

6.1.1 Flow and heat transfer behavior

The flow configuration in the heat exchanger tube equipped with inclined wavy surface is presented in terms of iso-surface, tangential velocity vector in transverse plane and longitudinal vortex flow. Figs. 4a, b, c and d plot the λ_2 iso-surface for the heat exchanger tube equipped with inclined wavy surface of the flow attack angle of 15°, 30°, 45° and 60°, respectively, at Re = 600. The λ_2 iso-surface is an indicator to consider the core of the vortex flow. As seen in the figures, the inclined wavy surface changes the flow structure in the heat exchanger tube. The vortex flow is clearly detected for the inclined wavy surface with the flow attack angles of 30°, 45° and 60°, while the flow attack angle of 15° performs the reverse result.

Figs. 5a, b, c and d present the tangential velocity vector in transverse plane for the heat exchanger tube equipped with the inclined wavy surface with the flow attack angle of 15°, 30°, 45° and 60°, respectively, at Re = 600. The two vortex flows are detected at the upper-lower parts of the plane in all cases. The flow configuration for all cases is found similarly. The strength of the vortex flow increases when increasing the flow attack angle. The flow attack angle of 60°

provides the highest vortex strength, while the flow attack angle of 15° gives the opposite result.

The longitudinal vortex flow in the heat exchanger tube equipped with inclined wavy surface for the flow attack angles of 15°, 30°, 45° and 60° is depicted in the Figs. 6a, b, c and d respectively, at Re = 600. The impinging of the flow on the tube wall and wavy surface is detected in all cases. The impingement of the flow on the heat transfer surface and the vortex strength are important factors for heat transfer enhancement in the heat exchanger.

The heat transfer behaviors in the heat exchanger tube equipped with the inclined wavy surface are presented in terms of temperature distributions in transverse planes and local Nusselt number distributions on the tube wall. The temperature distributions in cross sectional planes for the heat exchanger tube equipped with the inclined wavy surface are depicted as Figs. 7a, b, c and d for the flow attack angle of 15°, 30°, 45° and 60°, respectively. In general, the inclined wavy surface changes the heat transfer behavior in the tube heat exchanger when compared with the smooth circular tube with no generators. For all cases, the blue layer (cold fluid) distributes from the center of the test section, while the red layer (hot fluid) near the tube wall performs thinner. The mixing of the fluid in the test tube is found to be better. This leads to the enhancements of the heat transfer rate and thermal performance. The

worst mixing of the air in the test section is detected in the case of the flow attack angle of 15° .

The local Nusselt number distributions on the heat transfer surface for the heat exchanger tube equipped with the inclined wavy surface of the flow attack angle of 15° , 30° , 45° and 60° are reported as Figs. 8a, b, c and d, respectively, for the Reynolds number of 600. The level of the local Nusselt number ($Nu = 5 - 45$) for all flow attack angles is similarly shown. The insertion of the inclined wavy surface in the heat exchanger tube can increase the heat transfer rate in all flow attack

angles. The lowest values of the Nusselt number are detected at the inclined wavy surface with the flow attack angle of 15° . The inclined wavy surfaces with the flow attack angles of 45° and 60° give the nearly values of the Nusselt number. The peak of the high Nusselt number value on the tube wall (red contour) is due to the impingement of the vortex flow which created by the inclined wavy surface.

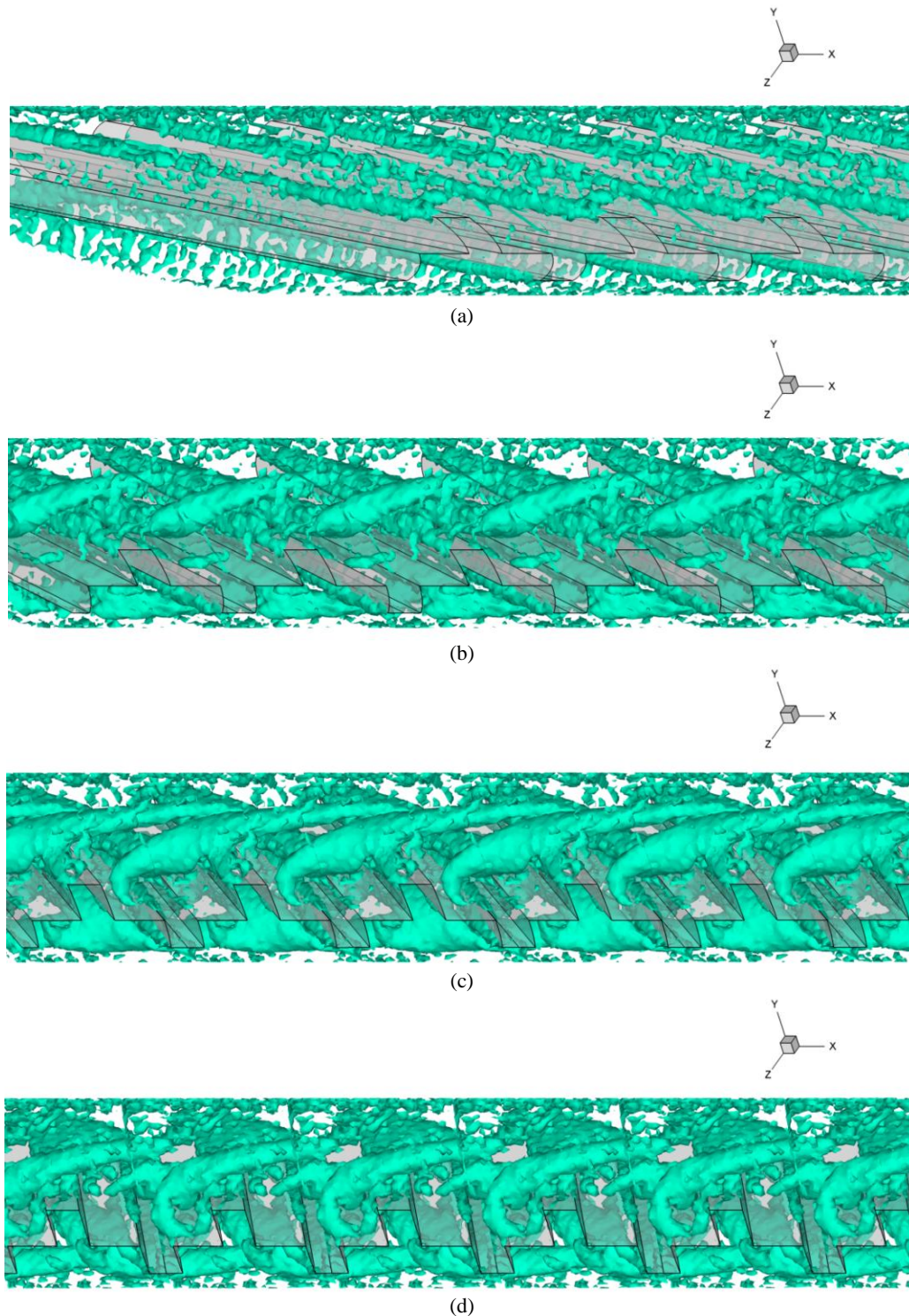


Fig. 4 λ_2 iso-surface of the heat exchanger tube equipped with the inclined wavy surface for (a) $\alpha = 15^\circ$, (b) $\alpha = 30^\circ$, (c) $\alpha = 45^\circ$ and (d) $\alpha = 60^\circ$ at $Re = 600$.

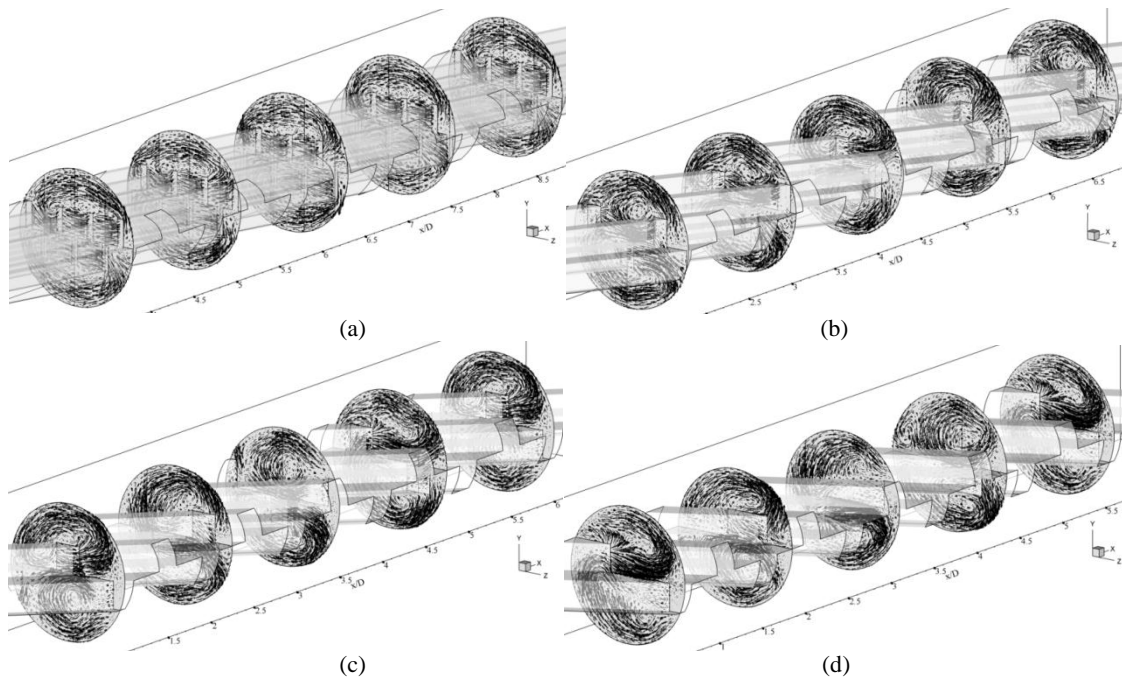


Fig. 5 Tangential velocity vector in transverse planes of the heat exchanger tube equipped with the inclined wavy surface for (a) $\alpha = 15^\circ$, (b) $\alpha = 30^\circ$, (c) $\alpha = 45^\circ$ and (d) $\alpha = 60^\circ$ at $Re = 600$.

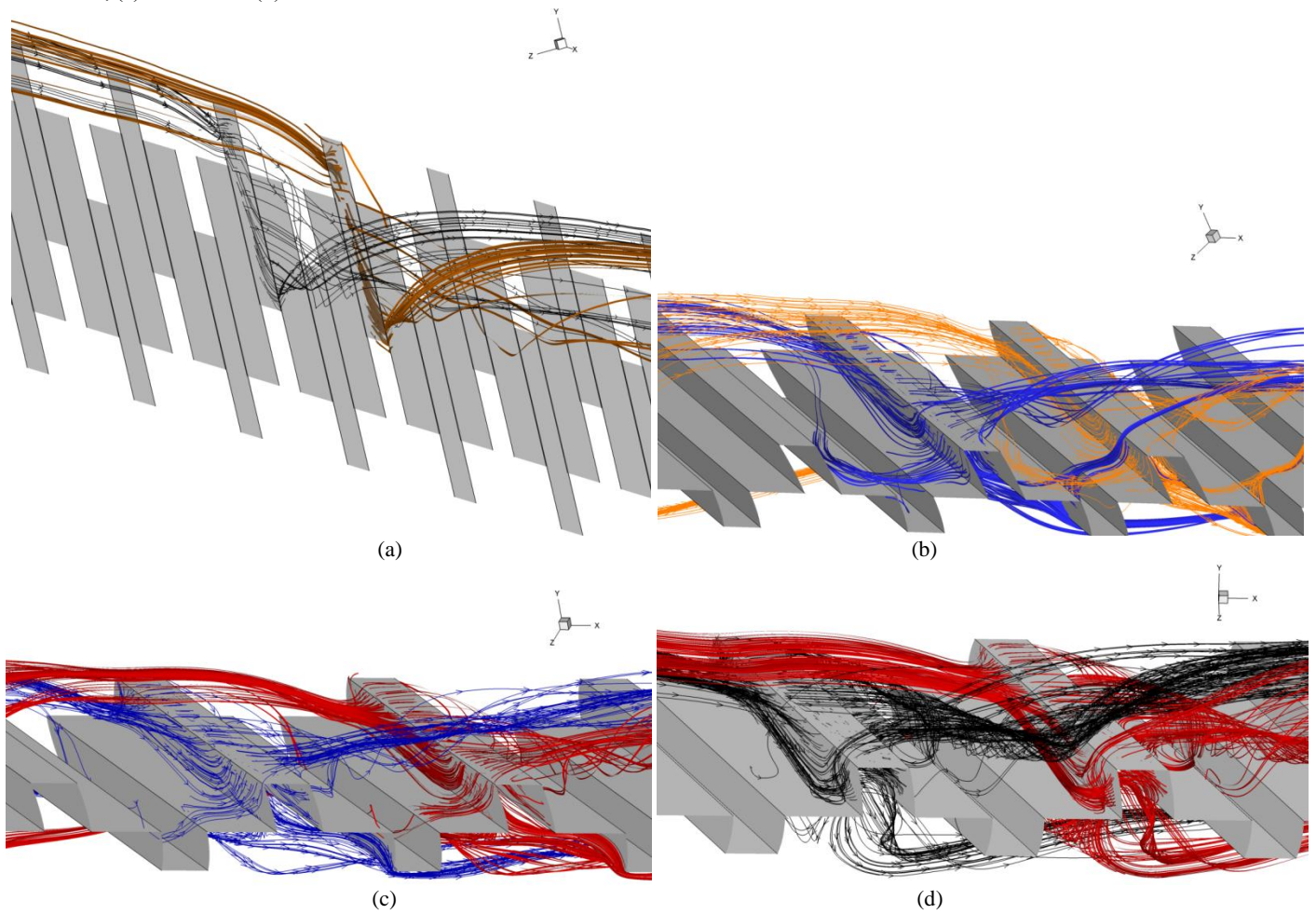


Fig. 6 Longitudinal vortex flow of the heat exchanger tube equipped with the inclined wavy surface for (a) $\alpha = 15^\circ$, (b) $\alpha = 30^\circ$, (c) $\alpha = 45^\circ$ and (d) $\alpha = 60^\circ$ at $Re = 600$.

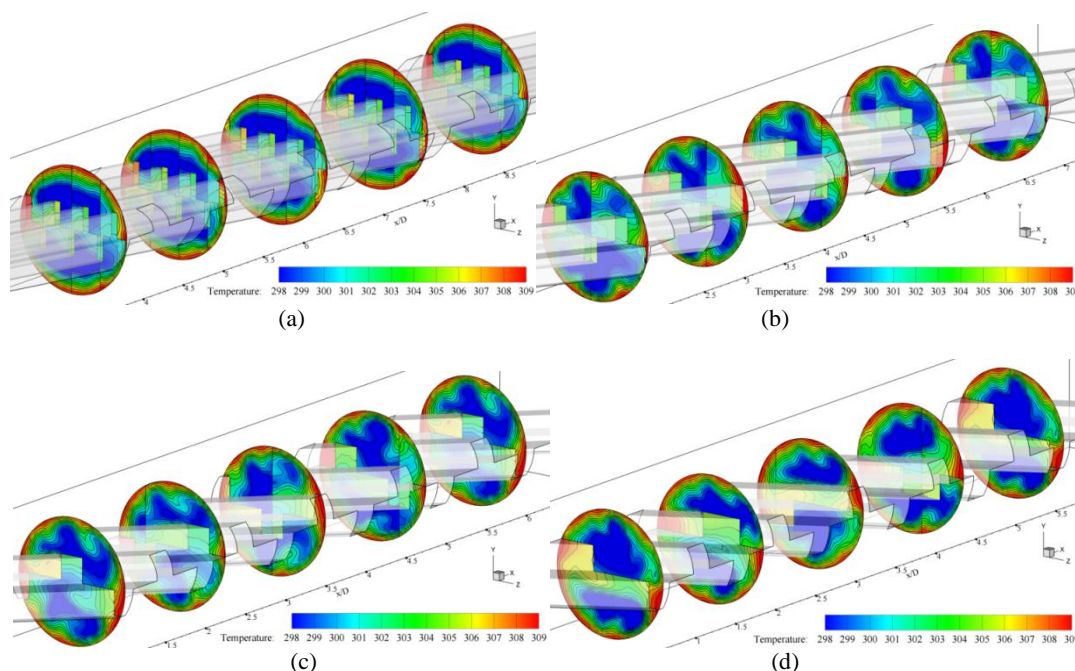


Fig. 7 Temperature distributions in transverse planes of the heat exchanger tube equipped with the inclined wavy surface for (a) $\alpha = 15^\circ$, (b) $\alpha = 30^\circ$, (c) $\alpha = 45^\circ$ and (d) $\alpha = 60^\circ$ at $Re = 600$.

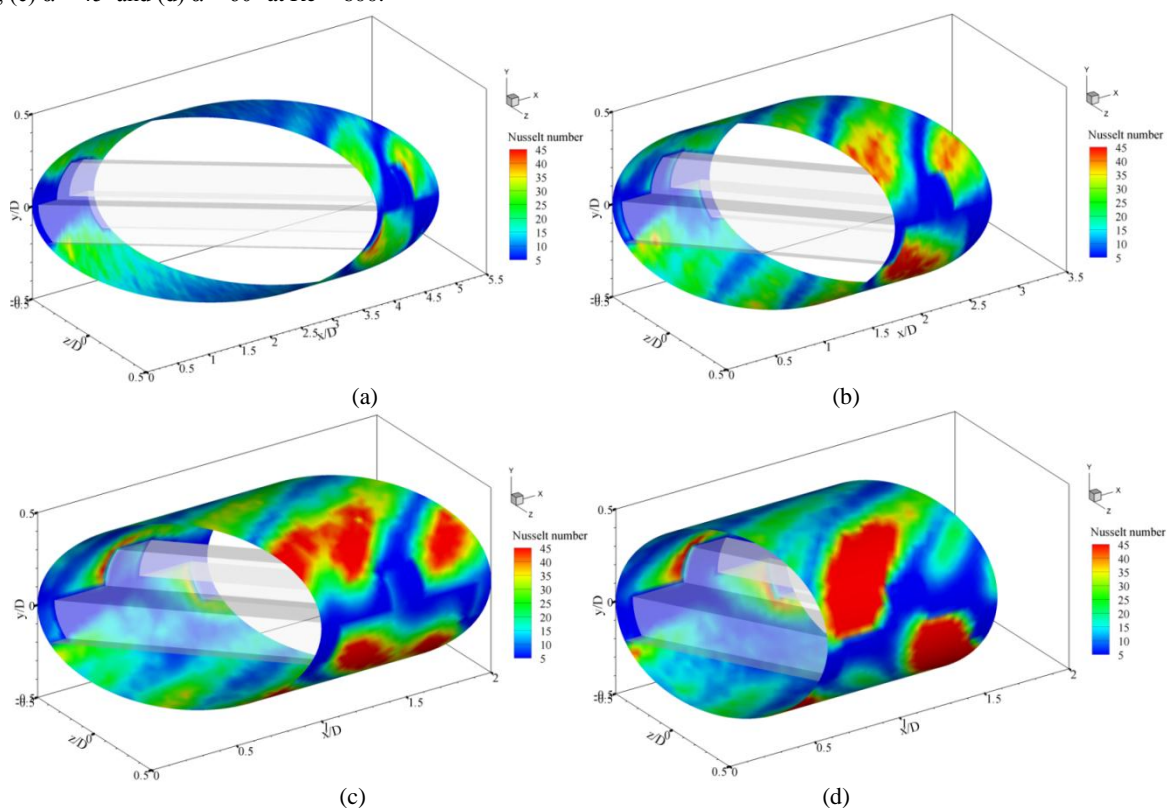


Fig. 8 Local Nusselt number distributions on the tube wall of the heat exchanger tube equipped with the inclined wavy surface for (a) $\alpha = 15^\circ$, (b) $\alpha = 30^\circ$, (c) $\alpha = 45^\circ$ and (d) $\alpha = 60^\circ$ at $Re = 600$.

6.1.2 Performance analysis

The performance evaluations in the heat exchanger tube equipped with the inclined wavy surface at various flow attack angles are reported in terms of Nusselt number ratio (Nu/Nu_0), friction factor ratio (f/f_0) and thermal enhancement factor (TEF). The relations of the Nu/Nu_0 , f/f_0 and TEF with the Reynolds number for the heat exchanger tube inserted with the inclined wavy surface are presented as Figs. 9a, b and c, respectively. Generally, the Nusselt number ratio increases when

enhancing the Reynolds numbers for all flow attack angles. In comparison with the smooth tube, the insertion of the inclined wavy surface in the test section performs greater heat transfer rate in all cases ($Nu/Nu_0 > 1$). The maximum Nusselt number ratio is found at the flow attack angle of 40° , while the lowest value is found at the flow attack angle of 15° . The reason is the flow attack angle of 40° can produce the strongest vortex flow. Considering at $Re = 1200$, the maximum and minimum values of the Nusselt number ratio are around 9.0 and 4.7, respectively. In the range studies, the insertion of the inclined wavy

surface in the heat exchanger tube gives the Nusselt number around 2 – 9 times above the smooth tube with no generators.

The addition of the inclined wavy surface in the heat exchanger tube not only increases in heat transfer rate, but also augments pressure loss. The present of the inclined wavy surface in the heat exchanger tube provides higher friction loss than the smooth circular tube in all cases ($f/f_0 > 1$). As the figure, the pressure loss increases when enhancing the Reynolds number. The maximum and minimum values of the f/f_0 are found at the flow attack angles of 60° and 15° , respectively. In the range studies, the present of the inclined wavy surface in the test section is around 7 – 65 times higher than the smooth tube with no generator. In addition, the low flow attack angle can help to reduce the friction loss in the heat exchanger.

Almost cases, the installation of the inclined wavy surface in the heat exchanger tube enhances the thermal performance higher than the smooth tube ($TEF > 1$). Considering at $Re = 1200$, the lowest TEF is detected at the inclined wavy surface with the flow attack angle of 15° , while the highest TEF is found at the flow attack angles of 30° , 35° and 40° . This means that the flow attack angles of 30° , 35° and 40° give the optimum values of the heat transfer rate and friction loss in the test tube. The insertion of the inclined wavy surface with the flow attack angles of 30° , 35° and 40° performs the optimum TEF is around 2.50. Additionally, the TEF is around 0.75 – 2.50 depended on the Reynolds number and flow attack angle.

The relations of the Nu/Nu_0 , f/f_0 and TEF with the flow attack angle for the heat exchanger tube inserted with the inclined wavy surface are depicted as Figs. 10a, b and c, respectively. As seen in the figures, it can be concluded that the suggested flow attack angle for the inclined wavy surface of the circular tube heat exchanger is around $30^\circ - 40^\circ$ due to these flow attack angles give high heat transfer rate and thermal performance.

6.2 Turbulent flow

6.2.1 Flow and heat transfer behavior

The λ_2 iso-surface, tangential velocity vector and longitudinal vortex flow are plotted for the heat exchanger tube inserted with the inclined wavy surface at turbulent regime to describe the flow mechanism in the test tube. Figs. 11a, b, c and d present the λ_2 iso-surface for the heat exchanger tube equipped with the inclined wavy surface for the flow attack angles of 15° , 30° , 45° and 60° , respectively, at $Re = 6000$. The core of the vortex flow is clearly detected in cases of the flow attack angles of 30° , 45° and 60° , while the flow attack angle of 15° performs the reverse result.

The tangential velocity in transverse planes for the heat exchanger tube equipped with the 15° , 30° , 45° and 60° inclined wavy surfaces are presented as Figs. 12a, b, c and d, respectively, at $Re = 6000$. The inclined wavy surface generates the similarity vortex flow at the upper-lower parts of the planes for all cases. The vortex flow, which created by the inclined wavy surface, can help to improve the quantity of the fluid mixing.

Figs. 13a, b, c and d show the turbulent kinetic energy (TKE) in transverse planes for the heat exchanger tube inserted with the inclined wavy surface for the flow attack angle of 15° , 30° , 45° and 60° , respectively, at $Re = 6000$. The high TKE is detected at the core flow for all angles. The peak of the TKE is found in case of the 60° inclined wavy surface, while the 15° inclined wavy surface performs the reverse trend. This means that the flow attack angle of 60° for the inclined wavy surface can create the greatest strength of the vortex flow.

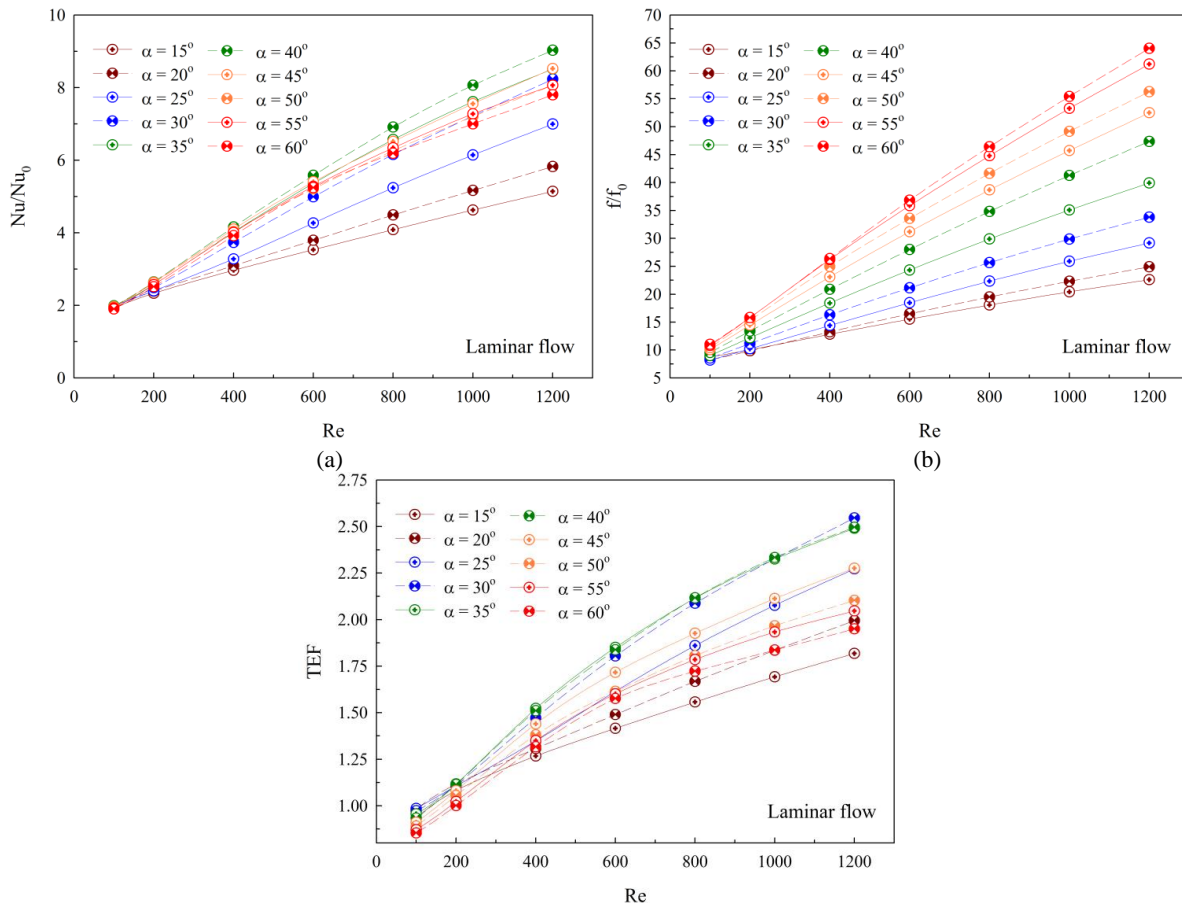


Fig. 9 (a) Nu/Nu_0 vs Re , (b) f/f_0 vs Re and (c) TEF vs Re .

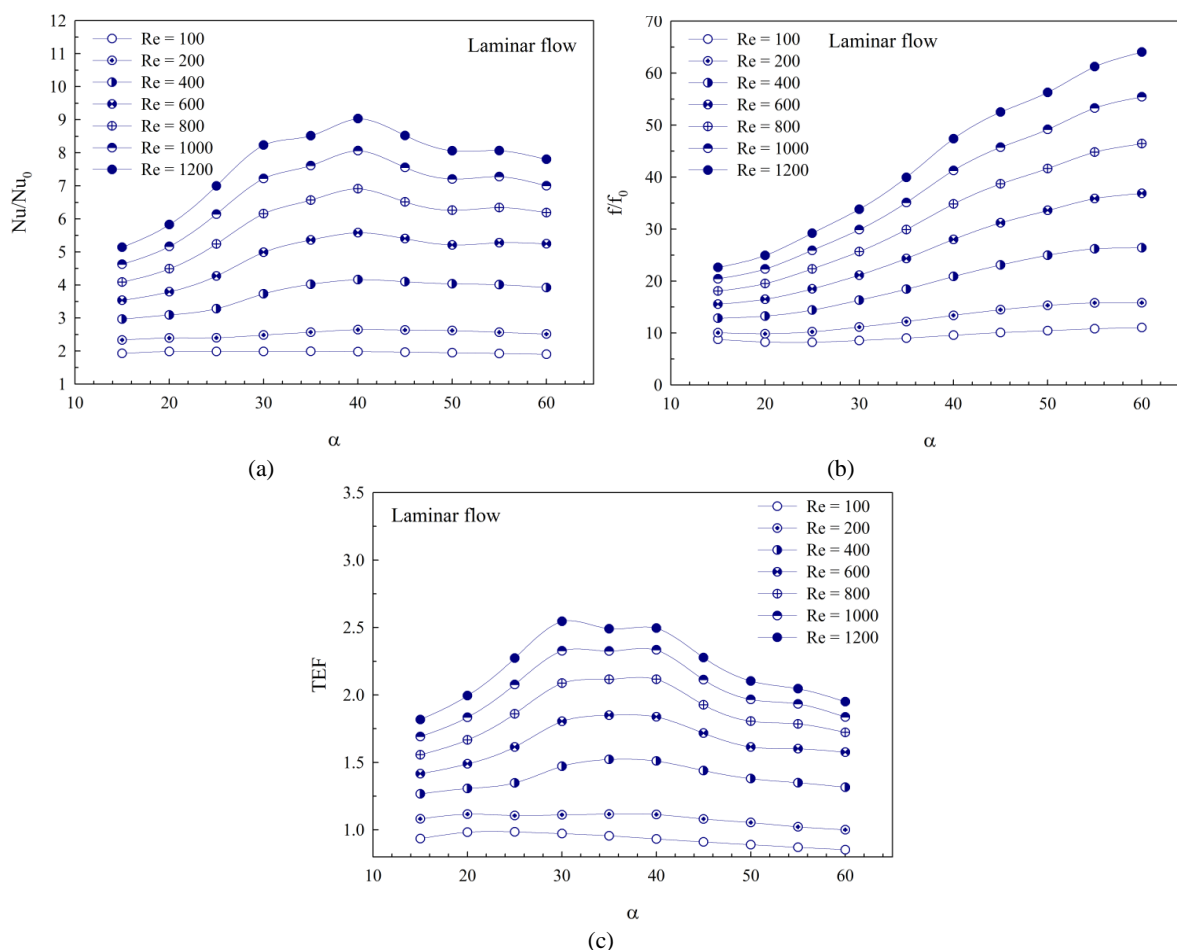


Fig. 10 (a) Nu/Nu_0 vs α , (b) f/f_0 vs α and (c) TEF vs α .

Fig. 14 reports the longitudinal vortex flow in the test section inserted with the 60° inclined wavy surface at $Re = 6000$. The impingement of the flow on the heat transfer surface and on the wavy surface is detected. This phenomenon is the main factor for heat transfer and thermal performance augmentation. The flow pattern for all attack angles is found in similar, but the strength of the flow is not equal.

The addition of the inclined wavy surface in the heat exchanger tube provides a better fluid mixing similarly as the laminar regime for all flow attack angles (see Fig. 15). The worse fluid mixing is detected at the 15° inclined wavy surface. The temperature distribution in transverse planes for the 30°, 45° and 60° inclined wavy surfaces performs nearly pattern.

Figs. 16a, b, c and d report the local Nusselt number distributions on the heat transfer surface of the heat exchanger tube equipped with the inclined wavy surface for the flow attack angles of 15°, 30°, 45° and 60°, respectively, at $Re = 6000$. The 45° and 60° inclined wavy surfaces give the highest heat transfer rate, while the 15° inclined wavy surface provides the opposite result. This is because the 45° and 60° inclined wavy surfaces can produce the strongest both vortex flow and impinging flow.

6.2.2 Performance analysis

The performance evaluations in the heat exchanger tube equipped with the inclined wavy surface at various flow attack angles are reported in terms of the Nusselt number ratio (Nu/Nu_0), friction factor ratio (f/f_0) and thermal enhancement factor (TEF) similarly as the laminar regime.

The relations of the Nu/Nu_0 , f/f_0 and TEF with the Reynolds number at various flow attack angles of the inclined wavy surface are depicted as Figs. 17a, b and c, respectively. In general, the Nu/Nu_0

decreases when increasing the Reynolds number for all flow attack angles. The insertion of the inclined wavy surface in the heat exchanger tube performs higher heat transfer rate than the smooth circular tube for all cases ($Nu/Nu_0 > 1$). The peak of heat transfer rate is detected for the flow attack angles in the range 45° – 60°, while the reverse trend is found at the inclined wavy surface with the flow attack angle of 15°. The augmentation on heat transfer rate depends on the strength of the vortex flow and impinging flow. The high flow attack angle can perform the strong vortex flow and impinging flow in the test section. In the range studies, the Nusselt number is around 3.0 – 6.7 times above the smooth tube when inserts the inclined wavy surface in the heat exchanger tube.

The insertion of the inclined wavy surface in the tube leads to augment pressure loss across the test section. The present of the inclined wavy surface in the heat exchanger tube gives higher friction loss than the smooth tube in all cases ($f/f_0 > 1$). As seen, the f/f_0 slightly increases when enhancing the Reynolds number for all flow attack angles. The maximum and minimum friction factor values are found at the flow attack angle of 60° and 15°, respectively. For the present-study condition, the friction factor is around 15 – 80 times over the smooth tube. In addition, the low flow attack angle of the inclined wavy surface can help to reduce the pressure loss in the test tube.

Due to the insertion of the inclined wavy surface in the heat exchanger tube leads to the augmentations on both heat transfer rate and friction loss, therefore the thermal enhancement factor is selected to decide the performance in the test section. Generally, the TEF decrease when augmenting the Reynolds number for all flow attack angles. The optimum TEF is detected for the flow attack angle in the range 20° – 30°. The TEF is found to be maximum at the flow attack angle of 30°

and $Re = 3000$ around 1.9. In the range studies, the TEF is around 0.9 – 1.9 depends on the flow attack angle and Reynolds number. The relations of the Nu/Nu_0 , f/f_0 and TEF with the flow attack angle for the heat exchanger tube inserted with the inclined wavy surface at various Reynolds numbers are plotted as Figs. 18a, b and c,

respectively. It can conclude that the high heat transfer rate and friction loss are detected at the flow attack angles around $45^\circ - 60^\circ$, while the optimum TEF is found at the flow attack angles around $20^\circ - 30^\circ$.

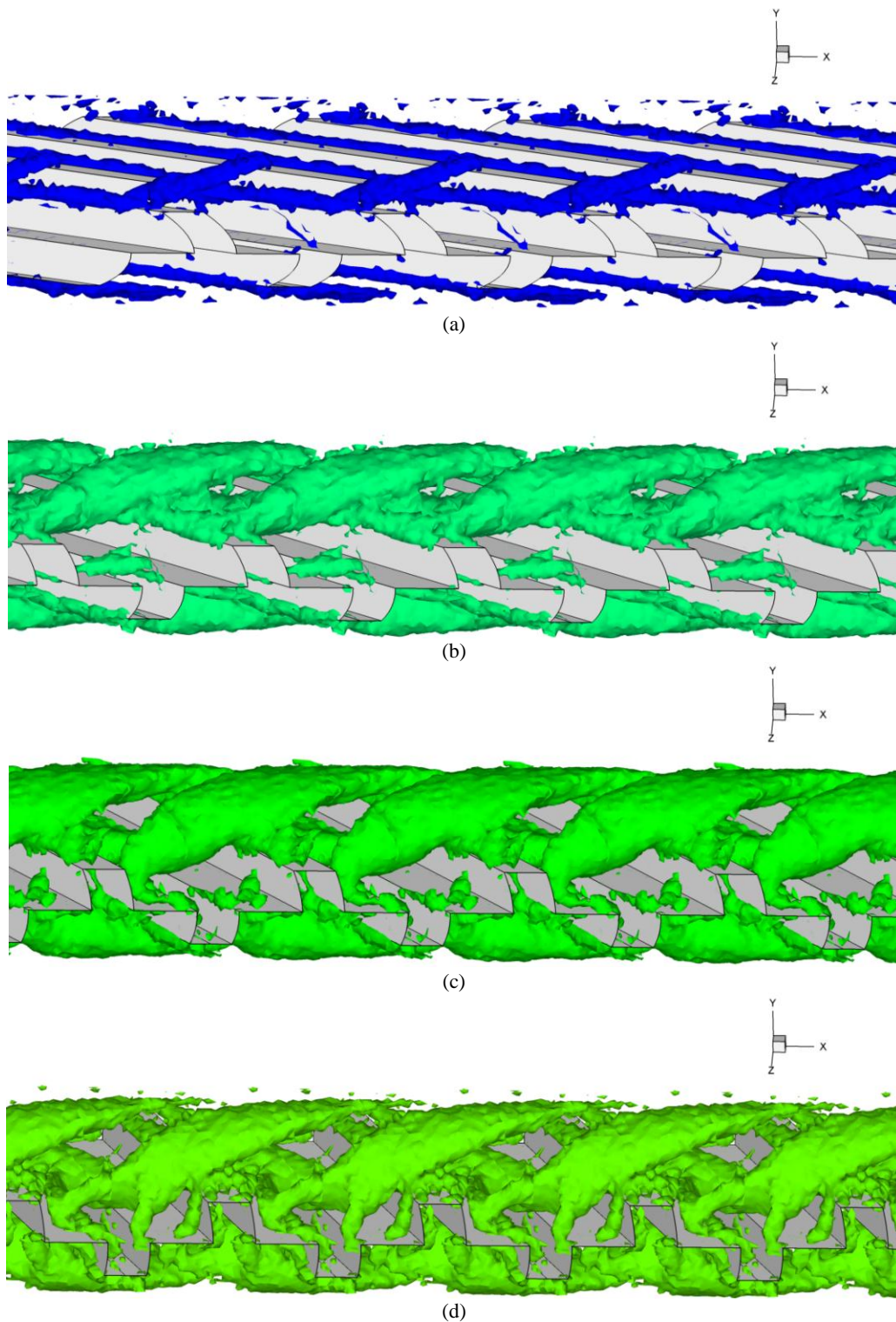


Fig. 11 λ_2 iso-surface of the heat exchanger tube equipped with the inclined wavy surface for (a) $\alpha = 15^\circ$, (b) $\alpha = 30^\circ$, (c) $\alpha = 45^\circ$ and (d) $\alpha = 60^\circ$ at $Re = 6000$.

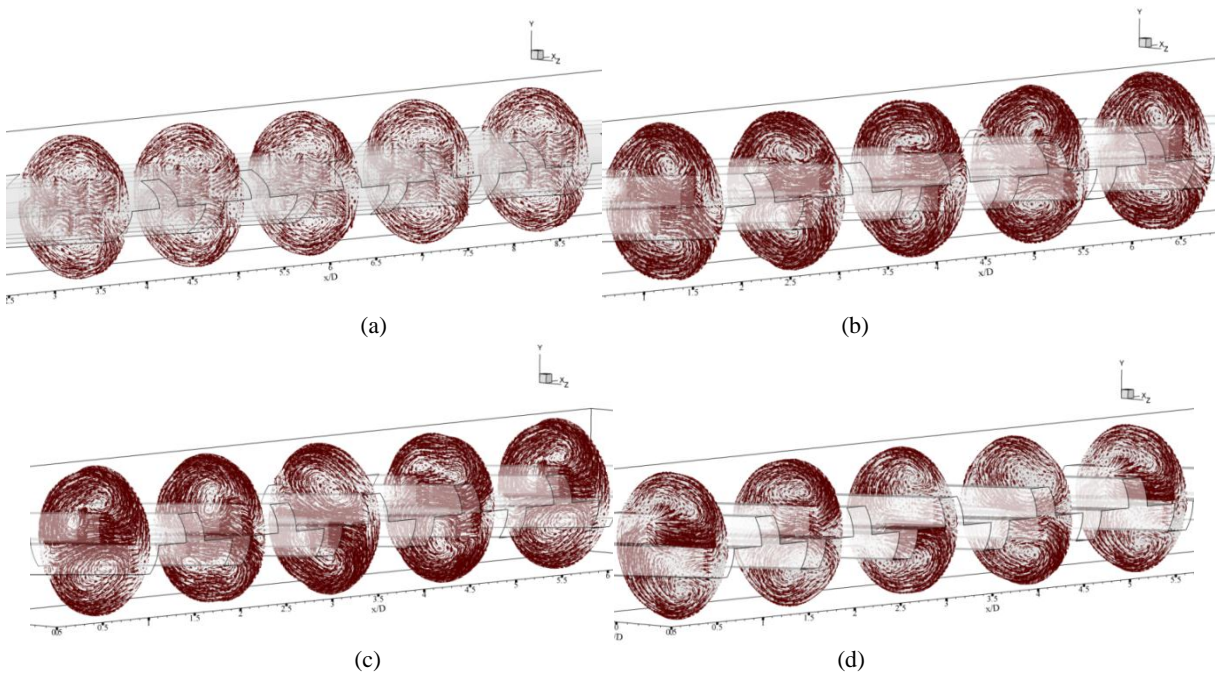


Fig. 12 Tangential velocity vector in transverse planes of the heat exchanger tube equipped with the inclined wavy surface for (a) $\alpha = 15^\circ$, (b) $\alpha = 30^\circ$, (c) $\alpha = 45^\circ$ and (d) $\alpha = 60^\circ$ at $Re = 6000$.

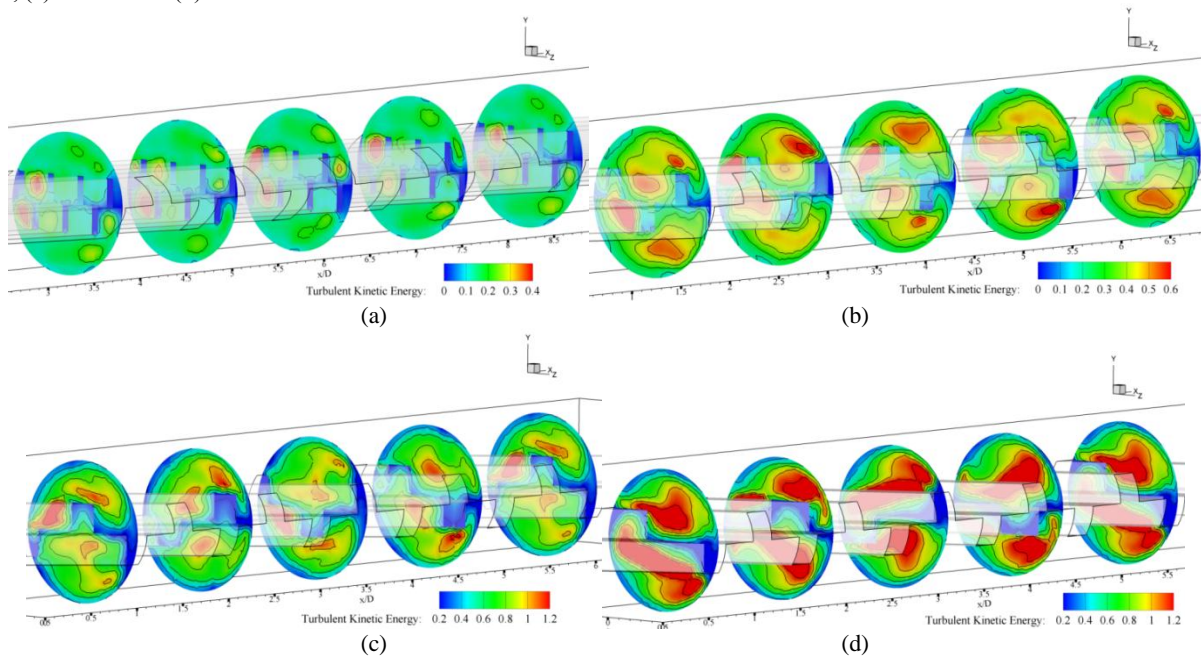


Fig. 13 Turbulent kinetic energy in transverse planes of the heat exchanger tube equipped with the inclined wavy surface for (a) $\alpha = 15^\circ$, (b) $\alpha = 30^\circ$, (c) $\alpha = 45^\circ$ and (d) $\alpha = 60^\circ$ at $Re = 6000$.



Fig. 14 Longitudinal vortex flow of the heat exchanger tube equipped with the inclined wavy surface for $\alpha = 60^\circ$ at $Re = 6000$.

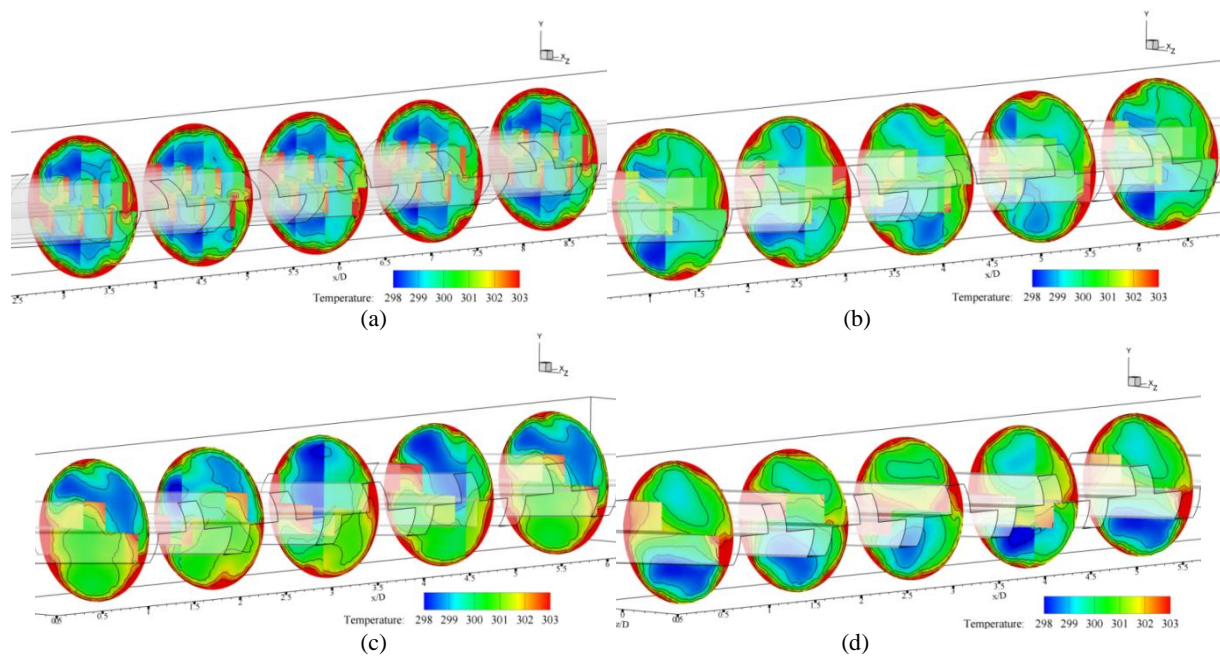


Fig. 15 Temperature distributions in transverse planes of the heat exchanger tube equipped with the inclined wavy surface for (a) $\alpha = 15^\circ$, (b) $\alpha = 30^\circ$, (c) $\alpha = 45^\circ$ and (d) $\alpha = 60^\circ$ at $Re = 6000$.

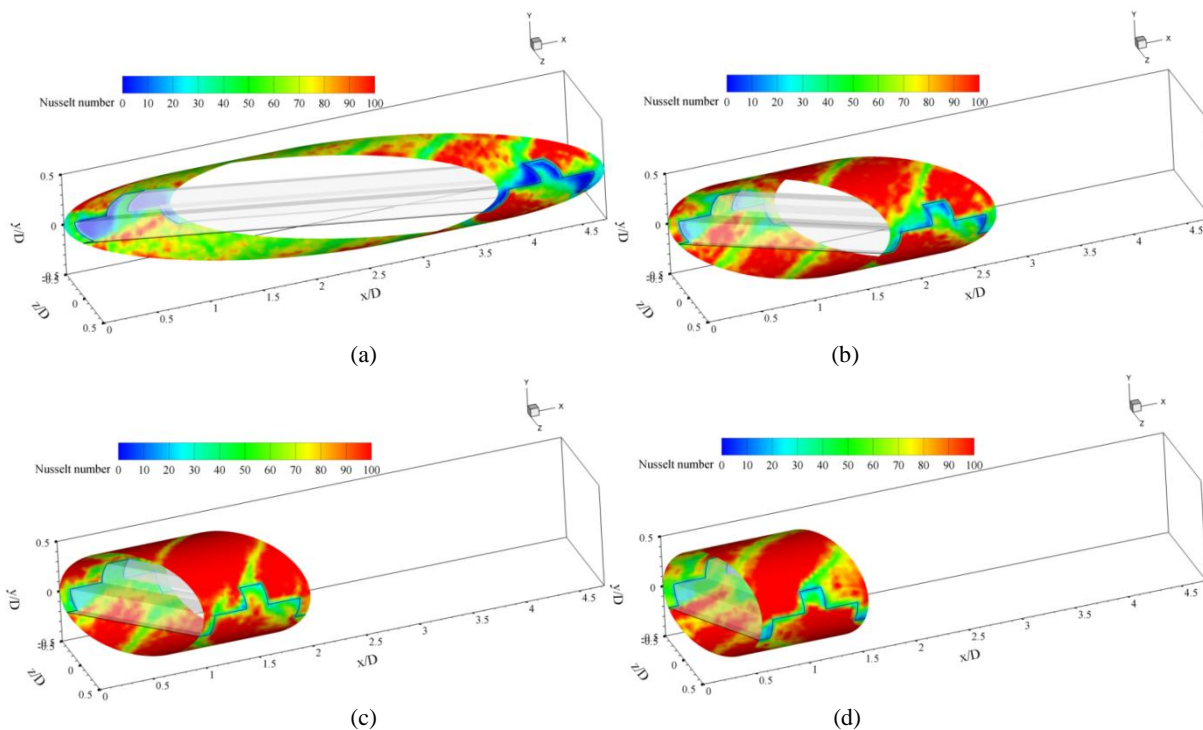


Fig. 16 Local Nusselt number distributions on the tube wall of the heat exchanger tube equipped with the inclined wavy surface for (a) $\alpha = 15^\circ$, (b) $\alpha = 30^\circ$, (c) $\alpha = 45^\circ$ and (d) $\alpha = 60^\circ$ at $Re = 6000$.

7. CONCLUSION

The numerical investigations on heat transfer, flow structure and performance assessment in the heat exchanger tube inserted with various flow attack angles ($\alpha = 15^\circ, 20^\circ, 25^\circ, 30^\circ, 35^\circ, 40^\circ, 45^\circ, 50^\circ, 55^\circ$ and 60°) of the inclined wavy surface are presented. The laminar flow ($Re = 100 - 1200$) and turbulent flow ($Re = 3000 - 10000$) are considered for the present study. The major findings are concluded as follows:

The insertion of the inclined wavy surface in the heat exchanger tube changes the flow structure and heat transfer behavior for both laminar and turbulent flows. The vortex flow, impinging flow and thermal boundary layer disturbance are detected when inserting the inclined wavy surface in the heat exchanger tube. These behaviors lead to higher heat transfer rate and thermal performance.

The flow and heat transfer patterns for the laminar and turbulent flows are found closely, but the strength of the flow is not equal.

The suggested attack angle for the laminar region is around $30^\circ - 40^\circ$, while around $20^\circ - 30^\circ$ for the turbulent flow when considered at the thermal enhancement factor.

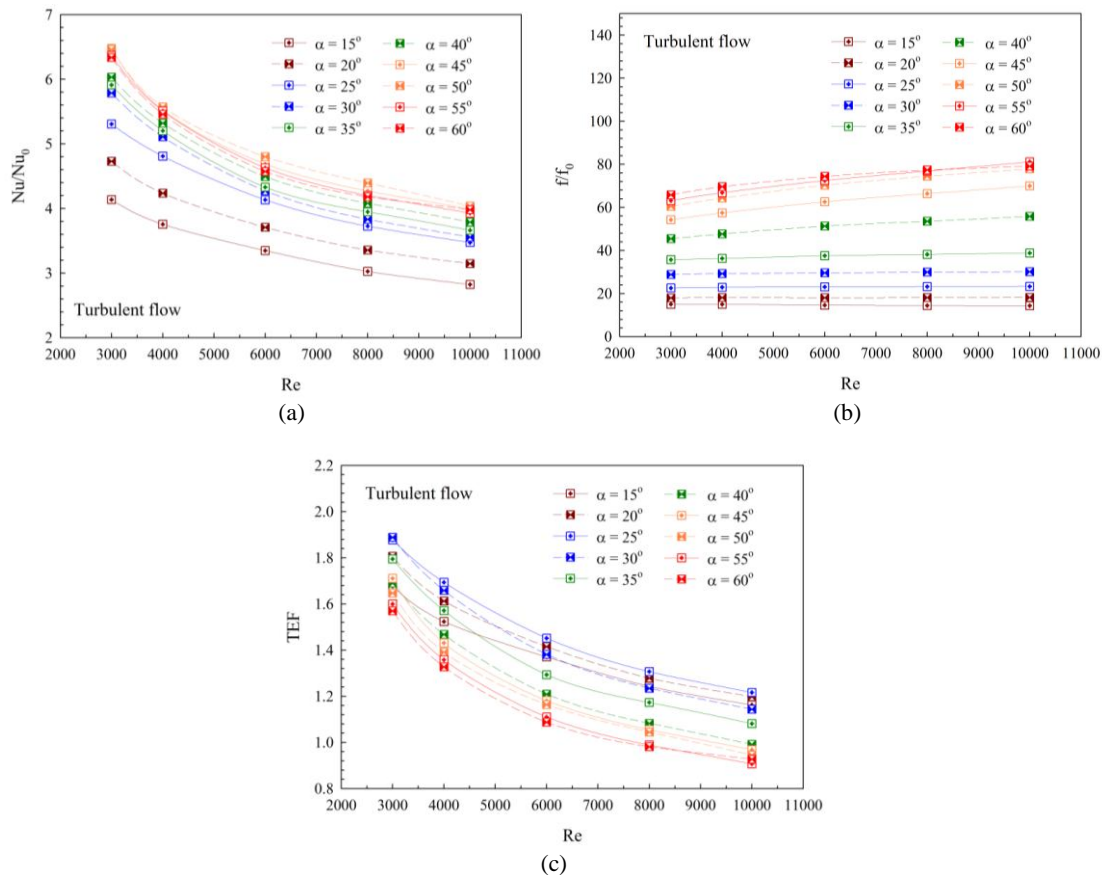


Fig. 17 (a) Nu/Nu_0 vs Re, (b) f/f_0 vs Re and (c) TEF vs Re.

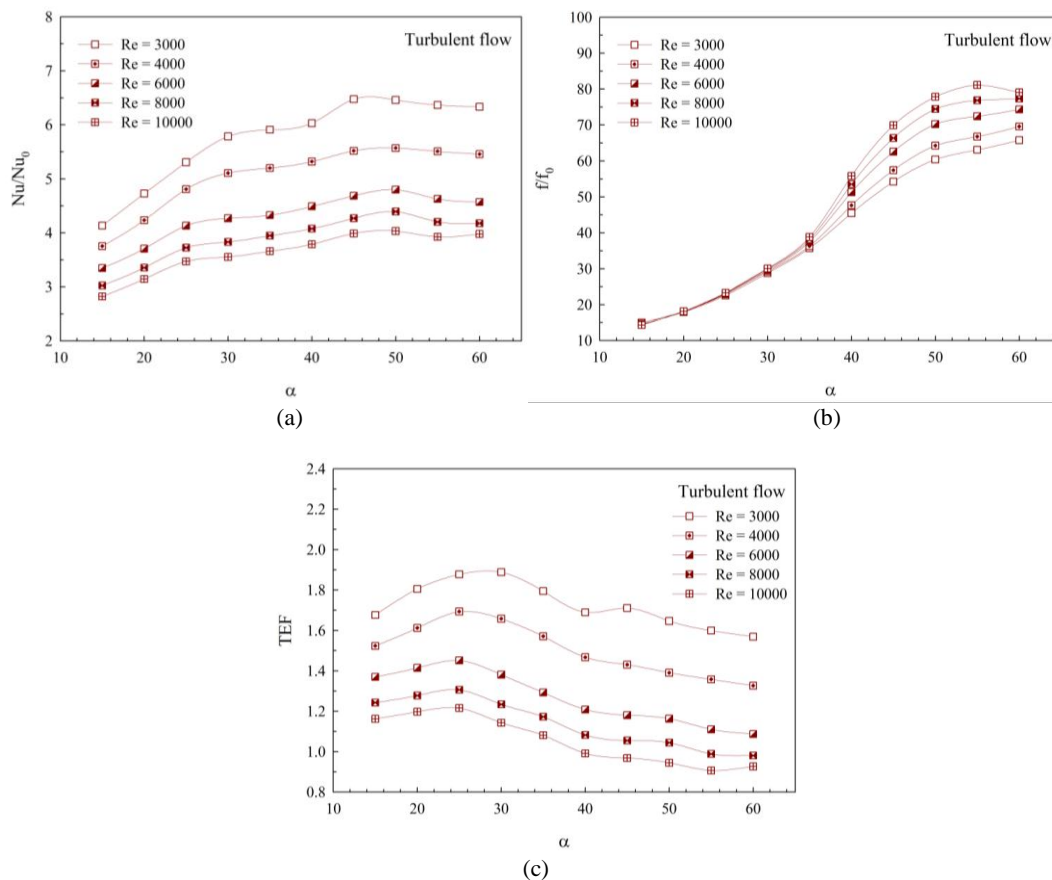


Fig. 18 (a) Nu/Nu_0 vs α , (b) f/f_0 vs α and (c) TEF vs α .

ACKNOWLEDGEMENTS

The authors would like to acknowledge Assoc. Prof. Dr. Pongjet Promvong for suggestions. The funding of this work is supported by King Mongkut's Institute of Technology Ladkrabang research funds (contract no. KREF046006).

NOMENCLATURE

D	diameter of tube
f	friction factor
h	convective heat transfer coefficient, $W\ m^{-2}\ K^{-1}$
k	thermal conductivity, $W\ m^{-1}\ K^{-1}$
Nu	Nusselt number ($=hD/k$)
p	static pressure, Pa
Pr	Prandtl number ($Pr = 0.707$)
Re	Reynolds number
T	temperature, K
u_i	velocity in x_i -direction, $m\ s^{-1}$
\bar{u}	mean velocity in channel, $m\ s^{-1}$
Greek letter	
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 tube
pp	pumping power

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