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Numerical studies on heat transfer and flow characteristics for laminar flow in a tube with multiple regularly spaced twisted tapes

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A B S T R A C T

The principle of heat transfer enhancement in the core flow of tube has been proposed to improve the temperature uniformity and reduce flow resistance, which is different from that of heat transfer enhancement in the boundary flow of tube. This article presents a simulation of multi-longitudinal vortices in a tube induced by triple and quadruple twisted tapes insertion. The simulation is conducted in order to gain an understanding of physical behavior of the thermal and fluid flow in the tube fitted with triple and quadruple twisted tapes for the Reynolds number from 300 to 1800. The obtained results show that, a maximum increase of 171% and 182% are observed in the Nusselt number by using triple and quadruple twisted tapes. And the friction factors of the tube fitted with triple and quadruple twisted tapes are around 4.06–7.02 times as that of the plain tube. The PEC of the tubes varies from 1.64 to 2.46. And the results verify the theory of the core flow heat transfer enhancement. Physical quantity synergy analysis is performed to investigate the mechanism of heat transfer enhancement. The synergy angles β and θ, are calculated, and the numerical results verify the synergy regulation among physical quantities of fluid particle in the flow field of convective heat transfer, which can guide us to get the optimum design.

1. Introduction

Heat exchangers, which are widely used in many fields such as power generation, chemical industry, metallurgy, steel production, refrigeration, air-conditioning etc., are indispensable general devices for heat transfer. The most significant variables in reducing the size and cost of a heat exchanger are heat transfer coefficient and pressure drop or flow resistance. An increase in the heat transfer coefficient often leads to an increase in the flow resistance, thereby reducing energy efficiency. The main challenge for heat exchangers design is to minimize the flow resistance while enhancing the heat transfer coefficients. Therefore, it is essential to develop theory and technique about heat transfer enhancement in the tube flow to raise the performance of a heat exchanger.

Generally speaking, tube flow can be divided into two parts [1]: the boundary flow and the core flow. The boundary flow is a fluid region near the wall in the tube, beyond which in the tube the core flow is defined. Heat transfer enhanced tubes such as [2–5] spiral grooved tube, longitudinal troughed tube, corrugated tube, inner-finned tube, spiral-ribbed tube, micro-ribbed tube and so on, are mainly considered to effectively design and improve heat transfer surface in the boundary flow. Moreover, those improved surfaces dominate convective heat transfer between the fluid and the tube wall. Therefore, this kind of methods can be called surface-based heat transfer enhancement or heat transfer enhancement in the boundary flow. On the contrary, the heat transfer enhancement in the core flow can be called fluid-based heat transfer enhancement. The surface-based heat transfer enhancement is the common method to enhance heat transfer in the tube. While these measures are effective for heat transfer, however, intensifying fluid disturbance in the boundary flow will result in more dissipation of fluid momentum, and enlarging continuously extended surface will cause more frictional resistance and viscosity dissipation. Thus, the flow resistance will be increased by adopting these techniques. If the flow resistance is overlarge, the fluid velocity will become small, which may weaken convective heat transfer between the fluid and the surface. To overcome this inherent weakness of surface-based enhancement, a number of experiments have been conducted on fluid-based enhancement [6–8]. In theory, Liu et. [9,10] have proposed a principle for increasing efficiency for core flow which is mainly expressed as: (1) strengthening temperature uniformity in the core flow; (2) increasing fluid disturbance in the core flow; (3)

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reducing surface area of heat transfer component in the core flow; (4) decreasing fluid disturbance in the boundary flow. Based on the principle, when developing a technique of heat transfer enhancement in the core flow of a tube, we should avoid strongly disturbing the fluid near the wall. As a result, the contact between the heat transfer component and the tube wall should be prevented, and the function of this component without contacting heat source or heat sink is just to disturb the fluid or uniform the fluid temperature. As the component for heat transfer enhancement in the core flow does not conduct heat from the tube wall, no convective heat transfer occurs at any point between the component surface and the fluid. So we can define that (1) surface-based heat transfer enhancement is one type that convective heat transfer occurs between the wall surface and the fluid; (2) fluid-based heat transfer enhancement is another type that there is no convective heat transfer between the component surface and the fluid, which is so-called heat transfer enhancement in the core flow.

In the past work, the twisted tape inserts are extensively used in the heat transfer enhancement of many heat exchangers. Manglik and Bergles [6,7] reported the experimental data for twisted tape and presented predictive correlations for laminar and turbulent flows under uniform wall temperature condition. Saha et al. [11] used the regularly spaced twisted tape elements connected by thin circular rods to investigate heat transfer enhancement in a circular tube. Date and Gaitonde [12] introduced the correlations for predicting characteristics of laminar flow in a tube fitted with regularly spaced twisted tape elements. Chang et al. [13] experimentally studied the axial heat transfer distribution and friction factor for the tubes fitted with broken twisted tapes of different twisted ratios, and found that local Nusselt number and mean friction factor increased with the decrease of twisted ratio. Naphon [14] compared tubes with twisted inserts with those without twisted tape inserts, and proposed non-isothermal correlations for predicting the heat transfer coefficient and friction factor of the horizontal pipe with twisted tape inserts. Eiamsa-ard et al. [15] experimentally studied the heat transfer, flow friction and thermal performance factor characteristics in a tube fitted with delta-winglet twisted tape, using water as working fluid. Eiamsa-ard [16] also studied thermal and fluid flow characteristics in turbulent channel flows with multiple twisted tape vortex generators. Seemawute [17] experimentally investigated the effect of peripherally-cut twisted tape with alternate axis (PT-A) on the fluid flow and heat transfer enhancement characteristic in a uniform heat flux circular tube. Eiamsa-ard et al. [18] presented a comparative investigation of enhanced heat transfer and pressure loss by insertion of single twisted tape, full-length dual and regularly-spaced dual twisted tapes as swirl generators, in a round tube under axially uniform wall heat flux (UHF) conditions. Eiamsa-ard et al. [19] also numerically studied the swirling flow in a tube induced by loose-fit twisted tape insertion. Guo et al. [20] numerically studied the heat transfer and friction factor characteristics of laminar flow in a circular tube fitted with center-cleared twisted tape.

The compound heat transfer enhancement technologies have broadened the ability of twisted tapes [21–26]. Ray and Date [21,22] numerically investigated the convective heat transfer behaviors in square duct with twisted tape insert. In square ducts, the twisted tape contacts with and away from the wall periodically. This structure creates periodical bursting swirls in the gaps which alter the fluid structure near the wall. Therefore, it ensures the effectiveness of the twisted tape under larger *Re* conditions as compared with circular tubes. Zimparov [23,24] experimentally studied the heat transfer and friction factor characteristics of the three-start and single-start spirally corrugated tubes with twisted tape insert. He found that these methods could achieve higher heat transfer coefficient and thermohydraulic performance than the smooth tubes with twisted tape insert. Promvonge and Eiamsa-ard [25] investigated the heat transfer behaviors in a tube with combined conical-ring and twisted tape inserts. Their work verified that this compound technique had better performance than using conical-ring only. Liao et al. [26] studied the heat transfer and friction factor characteristics in tubes with three-dimensional internal extended surfaces and twisted tape inserts. The experimental results showed that this method was of particular advantage to enhance the convective heat transfer for the laminar tubeside flow of highly viscous fluid.

However, as mentioned above, there are extensive literatures investigating the tubes fitted with twisted tape inserts, they mainly focused on the performance of heat transfer and flow resistance, the mechanism of heat transfer enhancement is rarely reported. So it is worthwhile studying the novel mechanism of heat transfer enhancement, which could serve as a guideline to optimize heat exchangers and design new-type heat transfer enhancement apparatus. Guo et al. [27,28] proposed the field synergy principle which indicates that the heat transfer rate depends not only on the velocity and temperature fields but also on their synergy which is related to an integral of the inner product of the temperature gradient and the velocity field. Liu et al. [29–31] developed Guo's field synergy principle and proposed that there exists other synergetic relation among physical quantities besides the velocity and temperature fields. Improving synergetic relation among physical quantities is beneficial to heat transfer enhancement. Guo et al. [32,33] also introduced a new quantity that corresponds to electrical potential energy in a capacitor based on the analogy between electrical and thermal systems. This quantity is called entransy which describes the total 'potential energy" of the thermal energy in an object. Heat transfer is always accompanied by entransy transfer. Thermal energy is conserved, while entransy is not conserved due to dissipation. The entransy dissipation could be used to measure the irreversibility of heat transfer process. The extremum principle of entransy dissipation is put forward by Guo et al. [32]. Based on the extremum principle of entransy dissipation, Meng et al. [34,35] solved the equation of the heat transfer potential capacity numerically for fluid flow in a straight circular tube for fully developed laminar flow and a theoretically optimum flow field was derived where multiple longitudinal vortices appeared in the cross section of the circular tube.

In the present study, we attempt to develop a triple twisted tape and a quadruple twisted tape to produce the multi-longitudinal vortices flow. The synergy regulation among physical quantities of fluid particle in the flow field of convective heat transfer are also applied to guide the optimum design for better heat transfer units and high-efficiency heat exchangers.

2. Physical model

The geometries of the triple twisted tape and quadruple twisted tape are depicted in Fig. 1. The diameter (D) of the tube is 0.02 m. And the length (L) of a period of the tube is 0.025 m. Twisted tapes with thickness (t) of 0.001 m and width (W) of 0.004 m are fitted in the tubes. The 360° twist pitch (s) is 0.02 m and thus the relative twisted ratio ($s/2W$) is 2.5, and the twist direction is right twist. The regularly spaced twisted tape elements have space ($b = L - s$) of 0.005 m with each other and thus the space ratio $b^* = b/L$ is 0.25. From the sectional view of the three and the four twisted tapes, they constitute a regular triangle and a square, respectively, with different clearance ratio $a^* = a/D$.

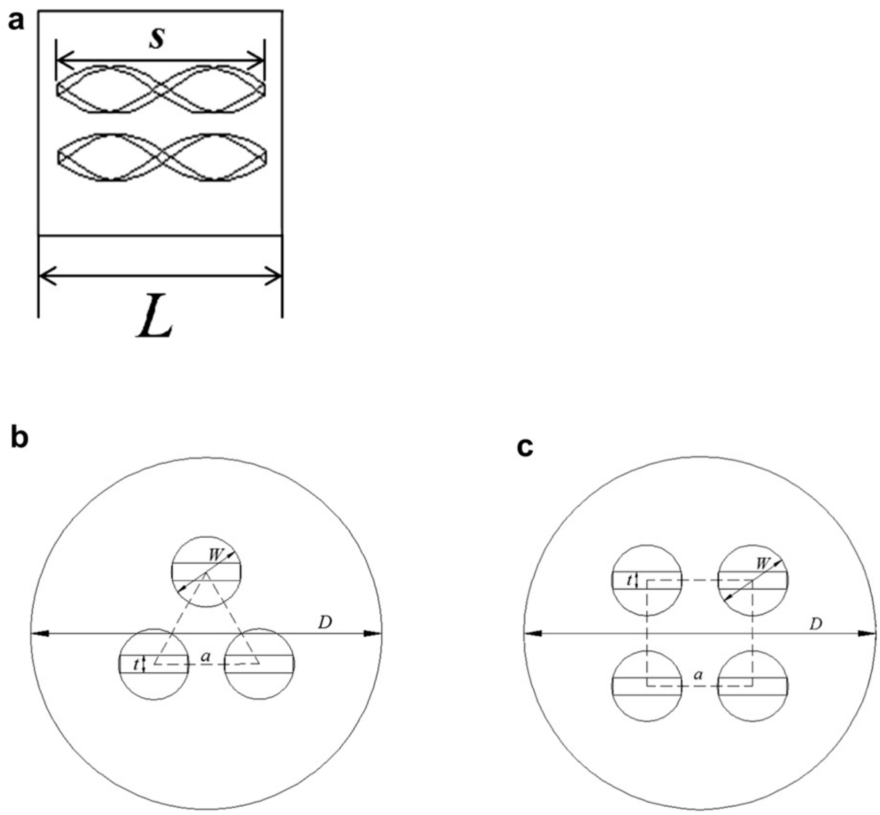

Fig. 1. The model of the tube fitted with triple or quadruple twisted tapes: (a) The side view of the tube fitted with triple or quadruple twisted tapes; (b) The sectional drawing of the tube fitted with triple twisted tapes; (c) The sectional drawing of the tube fitted with quadruple twisted tapes.

The Reynolds number (Re), the Nusselt number (Nu) and the friction factor (f) are defined as follows:

$$Re = \frac{\rho u D}{\mu} \tag{1}$$

$$Nu = \frac{hD}{k} \tag{2}$$

$$f = \frac{\Delta p}{(\rho u^2/2)(L/D)} \tag{3}$$

Air is selected as the working fluid which is assumed to be incompressible. The twisted tapes and the fluid are in local thermodynamic equilibrium. The natural convection has been neglected and the thermophysical properties of fluid are assumed to be temperature independent. The dynamic viscosity (μ), the thermal conductivity (k), the density (ρ) and the specific heat at constant pressure (c_p) of air are given as $\mu = 1.7894 \times 10^{-5}$ kg m^{-1} s^{-1}, $k = 0.0242$ W m^{-1} K^{-1}, $\rho = 1.225$ kg m^{-3} and $c_p = 1006.43$ J kg^{-1} K^{-1}, respectively. The Reynolds numbers referred to the inlet values are set at 300, 600, 900, 1200, 1500 and 1800 in the computations.

3. Numerical simulations

3.1. Governing equations and boundary conditions

The problem under consideration is assumed to be three-dimensional, laminar and steady. Heat conduction in the twisted tape is neglected. Equations of continuity, momentum and energy for the fluid flow are given below in a tensor form,

Continuity equation:

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0 \tag{4}$$

Momentum equations:

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] \tag{5}$$

Energy equation:

$$\frac{\partial}{\partial x_j}\left(\rho u_j c_P T - k\frac{\partial T}{\partial x_j}\right) = 0 \tag{6}$$

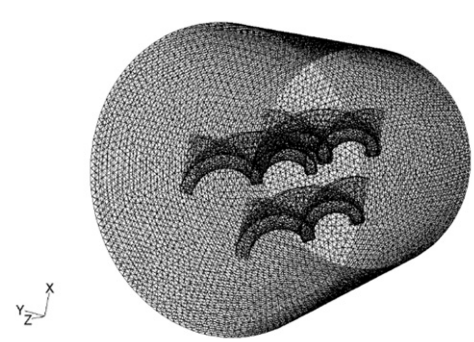

Fig. 2. The mesh of the tube fitted with multiple twisted tapes.

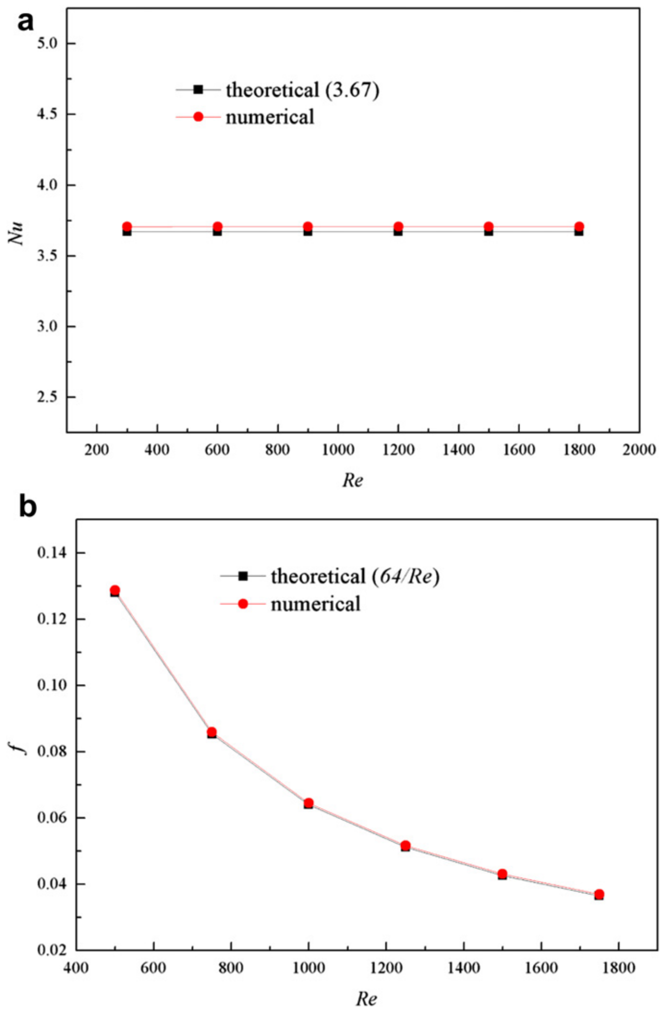

Fig. 3. Comparison of the numerical results and the theoretical data of the Nusselt number (*Nu*) and the friction factor (*f*) of the plain tube: (a) *Nu*; (b) *f*.

At the inlet and outlet, a periodic condition is specified and the upstream temperature is set to 293 K. On the tube walls and the surfaces of the twisted tape, no slip conditions are imposed. And the boundary condition between the fluid and the twisted tape surface is adiabatic condition. The tube wall temperature is constant both peripherally and axially, and it is set to 350 K.

3.2. Numerical method

The commercial software, Fluent 6.3.26, is chosen as the CFD tool for this work. Fluent works on the finite volume method to solve the above-mentioned governing equations accompanied with boundary conditions. The standard pressure and second order upwind discretization schemes for momentum and energy equations are employed in the numerical model. The pressure-velocity coupling is handled by the SIMPLE algorithm. In addition, convergence criteria of 10^{-6} for continuity and velocity components and 10^{-8} for energy are used, respectively. Non-uniform structured mesh was generated using Gambit 2.0, with more grids in the region near the tube wall and twisted tape. Fig. 2 shows the mesh of the tube fitted with multiple twisted tapes.

4. Results and discussions

4.1. Grid independent

For validating the accuracy of numerical solutions, the grid independent test has been performed for the physical model. The grid is highly concentrated near the wall and in the vicinity of the twisted tape. Three grid systems with about 5 993 370, 7 977 180 and 10 356 540 cells are adopted to calculate a baseline case. The type of grid is tetrahedral grid. The test problem is the laminar convection in a tube inserted with triple twisted tape of $a = 5$ mm and the Reynold number is 600. From the calculated values of Nusselt numbers obtained by the three grid systems, the 5 993 370-grid system is found to be dense enough to result in the grid independent solutions. Accordingly, the grid system with 5 993 370 grids is employed to perform the following calculations. To validate the accuracy of the numerical solutions, the Nusselt number (*Nu*) and the friction factor (*f*) of the plain tube are compared with the theoretical data under fully developed periodic condition. From Fig. 3 it is clearly seen that the deviation between the numerical results and the theoretical data is very limited. Therefore, the present numerical predictions have reasonable accuracy.

4.2. Flow structure

4.2.1. Velocity field

Field and vector plots of velocity predicted for triple and quadruple twisted tapes with different clearance ratios are depicted in Figs. 4–7. As seen in the figures, three or four longitudinal vortices are generated around tapes in the core flow area. These longitudinal vortices play a critical role of disturbing the boundary layer and uniforming the temperature in the core flow. And at the same time, it has been found that a new vortex tends to be generated in the center of the tube as the clearance between the twisted tapes increases, the fluid flows through the center of the tube while the fluid velocity have little loss.

Contour plots of streamline through the tube with triple and quadruple twisted tapes are respectively displayed in Fig. 8(a, b). It is clearly seen that the triple twisted tapes induce three swirling flows while the quadruple twisted tapes generate four swirling

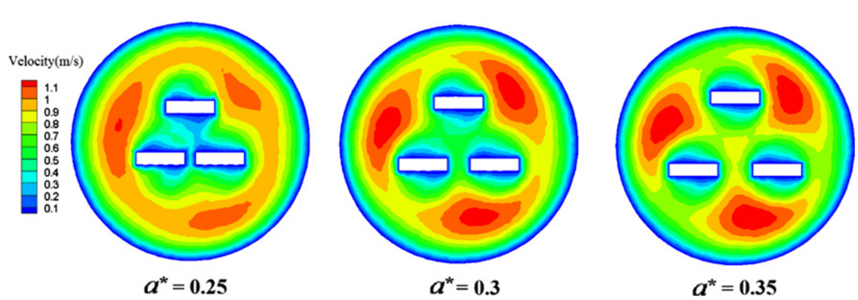

Fig. 4. The velocity field of the tube fitted with triple twisted tapes of various clearance ratios.

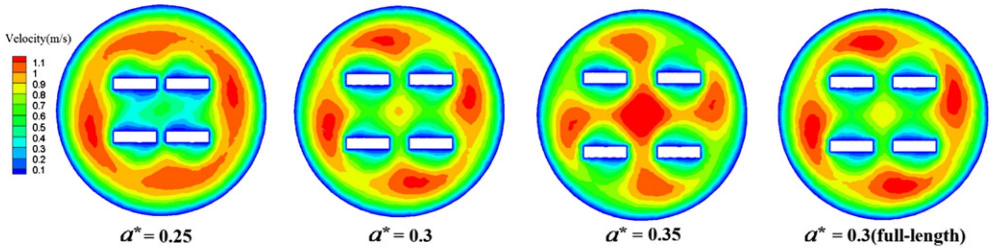

Fig. 5. The velocity field of the tube fitted with quadruple twisted tapes of various clearance ratios.

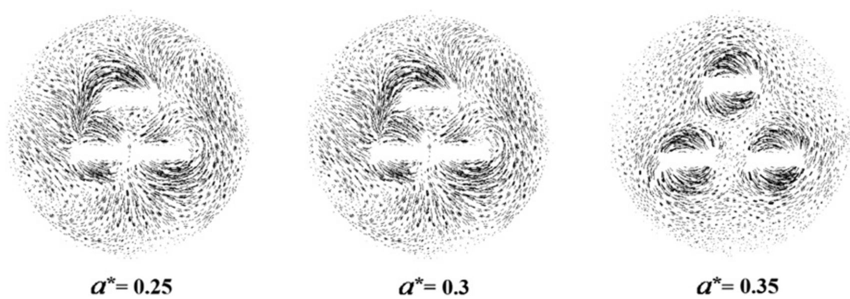

Fig. 6. The velocity vector of the tube fitted with triple twisted tapes of various clearance ratios.

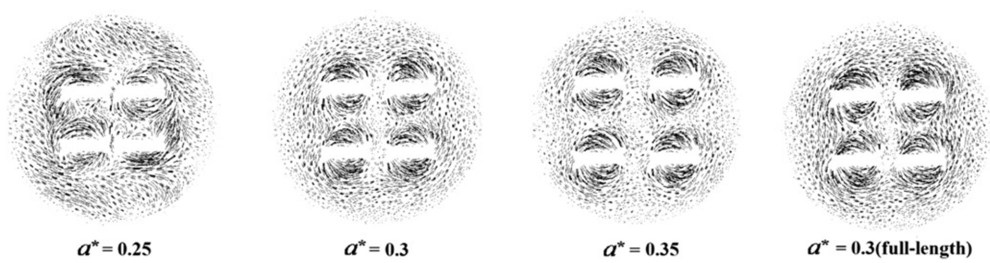

Fig. 7. The velocity vector of the tube fitted with quadruple twisted tapes of various clearance ratios.

flows. And both of the two kinds of twisted tapes generate two types of flows which are (1) a swirling flow and (2) an axial or straight flow near the tube wall.

4.2.2. Temperature field

Fig. 9 and Fig. 10 show the contour plots of temperature fields for triple and quadruple twisted tapes with different clearance ratios. For the two kinds of twisted tapes, the quadruple twisted tapes provide better temperature distribution than the triple twisted tape at the same twisted tapes clearance. And as the clearance between the twisted tapes increases, the temperature distribution is better. But the temperature distributions have little difference between each other among triple twisted tapes or quadruple twisted tapes.

4.2.3. Velocity and temperature profiles analysis

The dimensionless velocity and temperature profiles of laminar flow in a round tube are depicted in Fig. 11 and Fig. 12 under the Reynolds number of 900. Where, the dimensionless temperature and radius are defined as $T_\theta = (T_w - T)/(T_w - T_m)$ and $r^* = 2r/D$ respectively. As shown in the Figs. 11 and 12, the multi-longitudinal vortices generated by the insertion of triple or quadruple twisted tapes lead to the separation of the velocity boundary layer and the temperature boundary layer. Compared with the plain tube, the velocity profile remains unchanged fundamentally while the

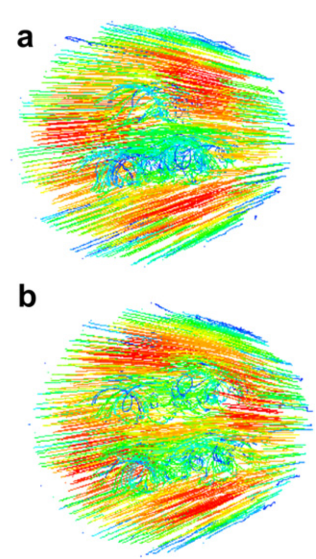

Fig. 8. The streamline of the tube fitted with triple and quadruple twisted tapes. (a) The streamline of the tube fitted with triple twisted tapes. (b) The streamline of the tube fitted with quadruple twisted tapes.

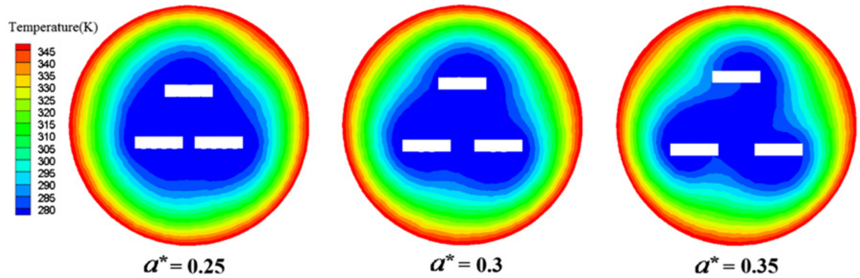

Fig. 9. The temperature field of the tube fitted with triple twisted tapes of various clearance ratios.

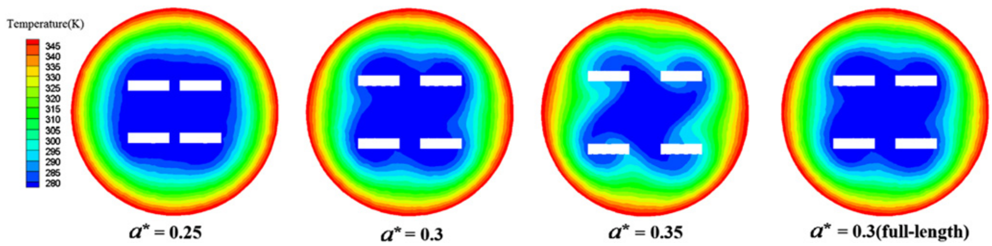

Fig. 10. The temperature field of the tube fitted with quadruple twisted tapes of various clearance ratios.

temperature profile has changed extremely. Thus the thermal diffusivity is enhanced greatly while the momentum diffusivity is almost unchanged. And this is the goal of core-flow heat transfer enhancement. So, the insertion of triple or quadruple twisted tapes enhances the heat transfer greatly, but does not increase the friction resistance very much.

4.3. Heat transfer

Effect of the triple and quadruple twisted tapes on the heat transfer rate is numerically studied and presented in Fig. 13. The heat transfer coefficients from the numerical simulation were defined as:

$$h = \frac{Q}{A_w \Delta T_m} \tag{7}$$

$$Q = \int_A q \mathrm{d}A \tag{8}$$

where $q = k\partial T/\partial r|_w$ is the local heat flux, A_w is the nominal inside tube area, ΔT_m is the logarithmic mean temperature difference and Q is the total heat transfer rate. Then the average Nusselt number was calculated as in Eq. (2).

The results of the tube fitted with all twisted tapes are also compared with those of a plain tube under similar operation conditions. For triple and quadruple twisted tapes, the heat transfer rate in terms of Nusselt numbers of quadruple twisted tapes is better than those for the triple twisted tapes. This is due to the fact that a system with more twisted tapes generates more swirl flows, making the temperature more uniform in the core flow and thus resulting in a thinner thermal or hydrodynamic boundary layer. On the other hand, it can also clearly seen that, for the same number of twisted tape, the Nusselt number decreases as the clearance ratio of the twisted tape decreases, and the larger the clearance, the better the heat transfer enhancement is. There are two reasons responsible for the weakening in heat transfer by the reduction of a^*. First, as the clearance ratio decreases, the disturbance of the boundary layer, as well as the fluid mixing of the boundary layer and core flow

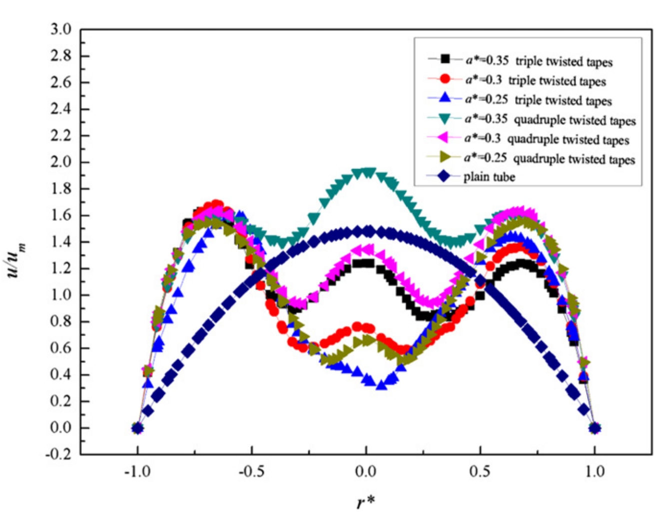

Fig. 11. The velocity profile of the tube fitted with triple and quadruple twisted tapes of various clearance ratios at the outlet.

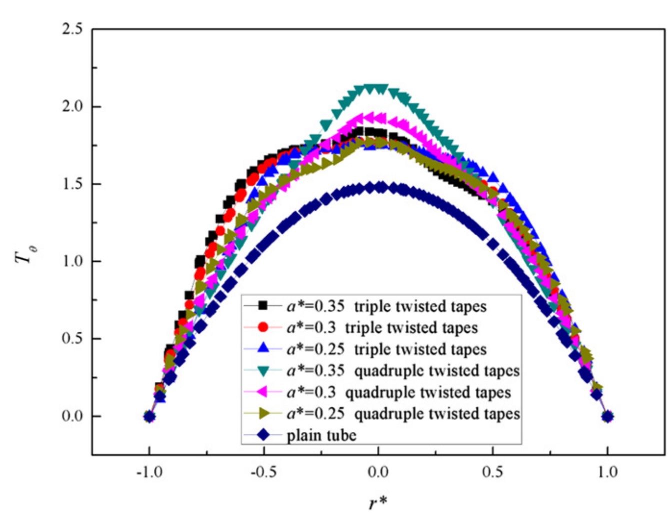

Fig. 12. The temperature profile of the tube fitted with triple and quadruple twisted tapes of various clearance ratios at the outlet.

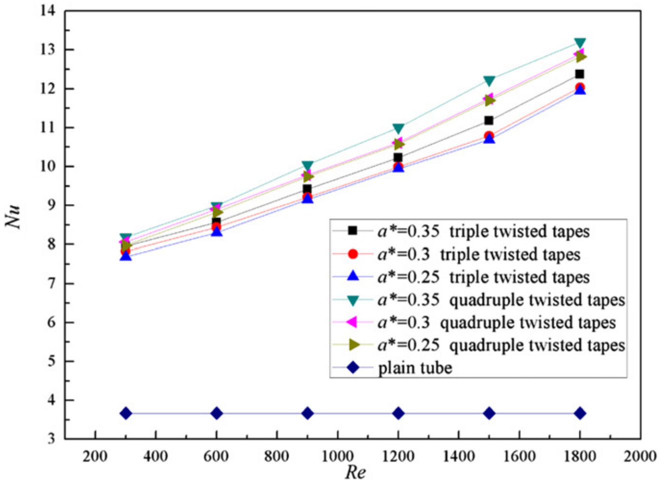

Fig. 13. Variation of Nusselt number with Reynolds number for triple and quadruple twisted tape of different clearance ratios.

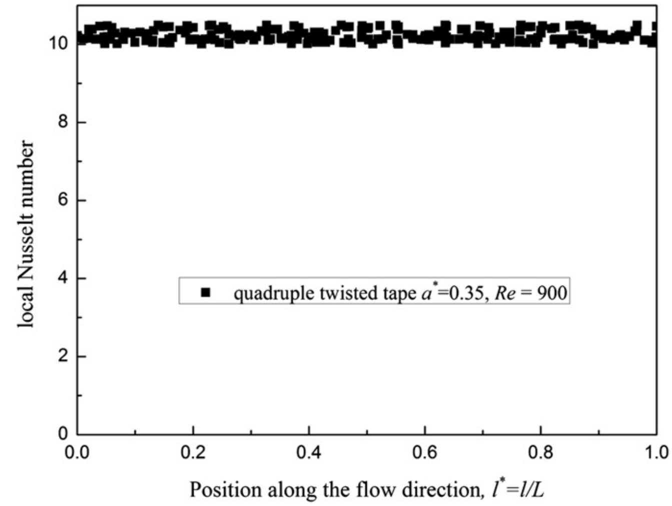

Fig. 15. Local Nusselt numbers along the periodic flow for quadruple twisted tapes, $a^* = 0.35$ at $Re = 900$.

is weakened. Second, a decrement of the clearance ratio leads to an increment of the twist ratio, which results in weaker swirls generated by the twisted tape. However, whether the tube fitted with triple twisted tapes or quadruple twisted tapes, the heat transfer rates of those tubes are much higher than the plain tube. The quantitative results show that, the mean heat transfer coefficients of the tube fitted with triple twisted tapes with $a^* = 0.25$, 0.3 and 0.35 are 162%, 164% and 171%, respectively, higher than that of the plain tube. And the mean heat transfer coefficients of the tube fitted with quadruple twisted tapes with $a^* = 0.25$, 0.3 and 0.35 are 180%, 182% and 189%, respectively, higher than that of the plain tube. It should be noted that the mean heat transfer coefficient of the triple twisted tapes, is slightly lower than that of the quadruple twisted tapes with difference around 10.6%. The correlation of the Nusselt number of the tube fitted with triple or quadruple regularly spaced twisted tapes can be written as follows:

$$\text{triple twisted tapes}: \; Nu = 2.0358Re^{0.2380}(a^*)^{0.0492} \tag{9}$$

$$\text{quadruple twisted tapes}: Nu = 1.5689Re^{0.2629}(a^*)^{-0.0773} \tag{10}$$

Fig. 15 shows that the local Nusselt numbers along the periodic flow for quadruple twisted tapes, $a^* = 0.35$ at $Re = 900$. The local Nusselt numbers are defined as:

$$h_{\text{local}} = \frac{q}{T_w - T_f} \tag{11}$$

$$Nu_x = \frac{h_{\text{local}} D}{k} \tag{12}$$

where $T_f = \rho u A c_p T / \rho u A c_p$. As shown in Fig. 15, the local Nusselt numbers have little difference with each other along the periodic flow. That means the tube with regularly spaced multiple twisted tapes has good heat transfer performance because there is no significant decrease in local Nusselt number. And a 3D plot of the local Nusselt numbers on the surface is displayed in Fig. 16; the area near the twisted tapes has high Nusselt number. This can be due to

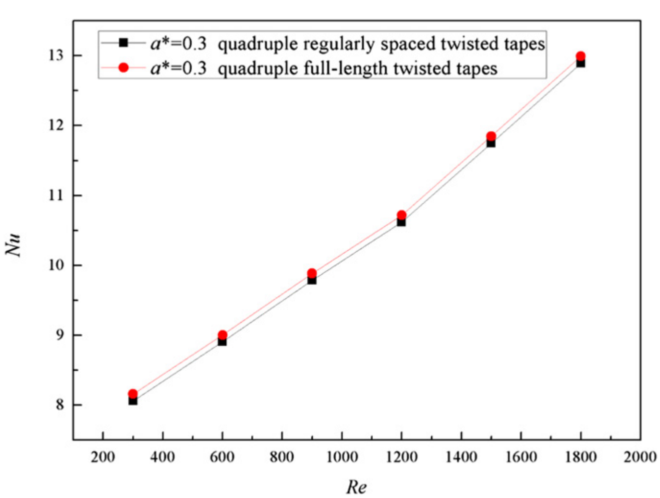

Fig. 14. Variation of Nusselt number with Reynolds number for quadruple regularly spaced and full-length twisted tapes.

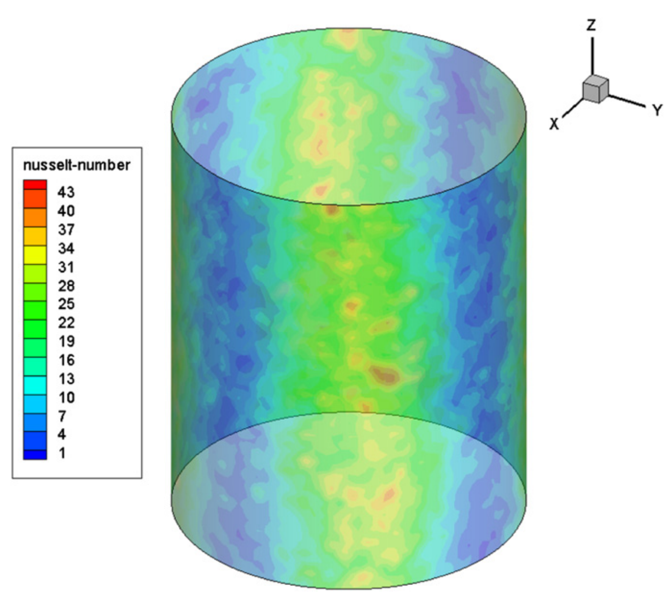

Fig. 16. The 3D-plot of the surface Nusselt numbers for quadruple twisted tapes, $a^* = 0.35$ at $Re = 900$.

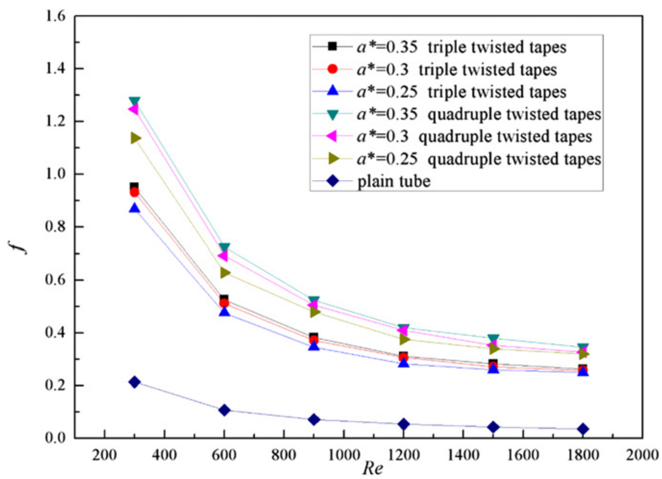

Fig. 17. Variation of friction factor with Reynolds number for triple and quadruple twisted tape of different clearance ratios.

the high temperature difference near the wall surface which is close to the twisted tapes.

4.4. Friction factor

The friction factor characteristic in a round tube fitted with triple twisted tapes or quadruple twisted tapes at various clearances is displayed in Fig. 17. As expected, friction factor decreases with decreasing clearance ratios, for both of triple twisted tapes and quadruple twisted tapes. This is because the larger the clearance ratio, the closer to the boundary the twisted tapes are located, and thus the disturbance of the boundary layer is more severe. The friction factors of the tube fitted with triple twisted tapes with $a^* = 0.25$, 0.3 and 0.35, are, respectively, around 4.06–4.74, 4.36–5.06 and 4.45–5.19 times as that of the plain tube. And the friction factors of the tube fitted with quadruple twisted tapes with $a^* = 0.25$, 0.3 and 0.35, are, respectively, around 5.33–6.27, 5.84–6.76 and 5.99–7.02 times as that of the plain tube. And at the same clearance, the friction factors of the tube fitted with quadruple twisted tapes are about 31.1%–35.2% higher than the tube fitted with triple twisted tapes. This may due to the larger surface area of quadruple twisted tapes compared with the triple twisted tapes. The correlation of the friction factor of the tube fitted

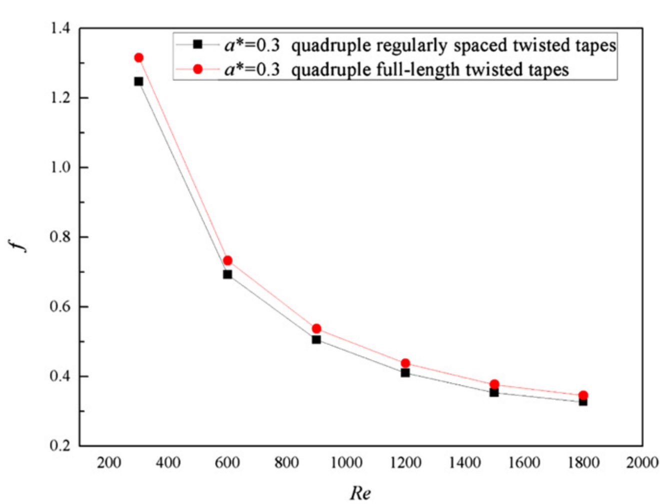

Fig. 18. Variation of friction factor with Reynolds number for quadruple regularly spaced and full-length twisted tapes.

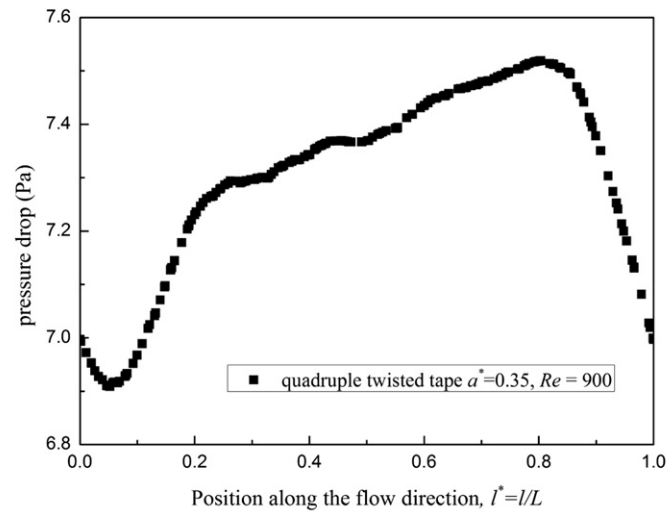

Fig. 19. Pressure drop along the periodic flow for quadruple twisted tapes, $a^* = 0.35$ at $Re = 900$.

with triple or quadruple regularly spaced twisted tapes can be expressed by:

$$\text{triple twisted tapes}: f = 111.2919 Re^{-0.7236}(a^*)^{0.6071} \quad (13)$$

$$\text{quadruple twisted tapes}: f = 181.5216 Re^{-0.7320}(a^*)^{0.7234} \quad (14)$$

Fig. 19 shows that the pressure drop along the periodic flow for quadruple twisted tapes, $a^* = 0.35$ at $Re = 900$. As shown in Fig. 19, there is significant decrease of pressure drop in the space area. That means the tube with regularly spaced multiple twisted tapes has an effect of decreasing flow resistance compared with the tube with the full-length twisted tapes.

4.5. The overall heat transfer performance

The PEC (Performance Evaluation Criterion) [36,37] is defined as follows:

$$\mathrm{PEC} = \frac{Nu/Nu_0}{(f/f_0)^{1/6}} \quad (15)$$

Where Nu_0 and f_0 are the Nusselt number and the friction factor of the plain tube, respectively.

The PEC in a tube fitted with triple or quadruple twisted tapes obtained using numerical simulation are depicted in Fig. 20. It is observed that the PEC value tends to increase with increasing Reynolds number for all twisted tapes. The tubes fitted with twisted tapes of the clearance ratio $a^* = 0.35$ give better overall heat transfer performance than the tubes with twisted tapes of other clearance ratios. This may due to the fact that the flow velocity is very small in laminar flow, so enhancing the disturbance of the boundary layer is dominate in enhancing the overall heat transfer performance while the friction factor increases not so much. And the tubes fitted with quadruple twisted tapes give better overall heat transfer performance than the tubes with triple twisted tapes when Reynolds number is from 900 to 1800. This is because the tubes fitted with more twisted tapes make the temperature in the core flow more uniform, and thus the temperature gradient near the tubeside is larger which means the heat transfer enhancement is greater.

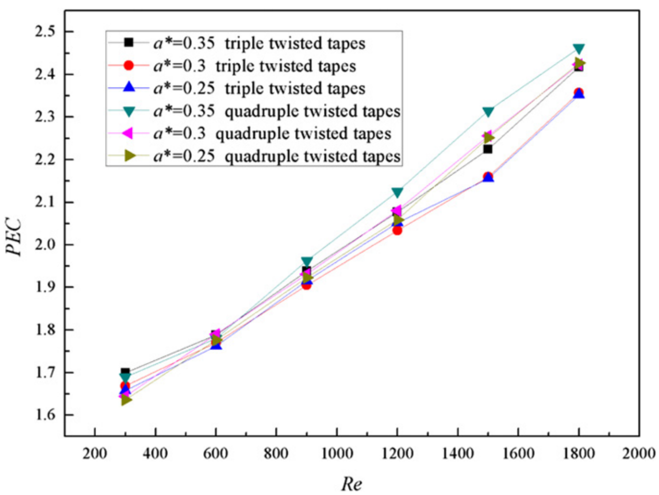

Fig. 20. Variation of PEC value with Reynolds number for triple and quadruple twisted tape of different clearance ratios.

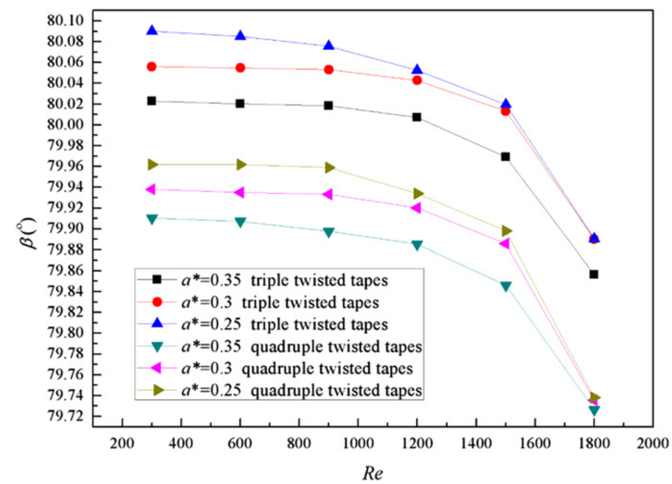

Fig. 22. The variation of average synergy angle β with *Re* for triple and quadruple twisted tape of different clearance ratios.

4.6. The comparison with full-length twisted tapes

The comparison of heat transfer, flow friction and PEC between the quadruple regularly spaced twisted tapes and full-length twisted tapes are shown respectively in, Figs. 14,18 and 21. Compared with full-length twisted tapes, the heat transfer enhancement of the tube fitted with quadruple regularly spaced twisted tapes reduces not obviously while the friction factor is much smaller than the full-length twisted tape. This can also be seen from the velocity and temperature fields (Figs. 5 and 10): the temperature fields of regularly spaced twisted tapes and full-length twisted tapes are almost the same while the velocity field of full-length twisted tapes has higher velocity gradient near the tube wall compared the velocity field of regularly spaced twisted tapes. There are two reasons responsible for this phenomenon. First, for the regularly spaced twisted tapes, as long as the disturbance generated by twisted tapes begins to be weakened in the region of space, the flow goes into the twisted tapes region again and the disturbance could stay still. Second, the area of regularly spaced twisted tapes is smaller than the area of full-length twisted tapes, so the friction factor is smaller too as expected.

5. Physical quantity synergy analysis

On the basis of the principle of field synergy for heat transfer enhancement [27,28], Liu et al. [29–31] set up the concept of physical quantity synergy in the laminar and turbulent flow field according to the physical mechanism of convective heat transfer between fluid and tube wall, which reveals the synergy regulation among physical quantities of fluid particles. The physical nature of enhancing heat transfer and reducing flow resistance, which is directly associated with synergy angles β and θ, is also explained.

The synergy angles among velocity and temperature gradient of a fluid particle M in the flow field can be written as

$$\beta = \arccos\frac{\boldsymbol{U}\cdot\nabla T}{|\boldsymbol{U}||\nabla T|} \tag{16}$$

And the synergy angle between velocity $\boldsymbol{U}$ and pressure gradient ∇p can be expressed as

$$\theta = \arccos\frac{\boldsymbol{U}\cdot\nabla p}{|\boldsymbol{U}||\nabla p|} \tag{17}$$

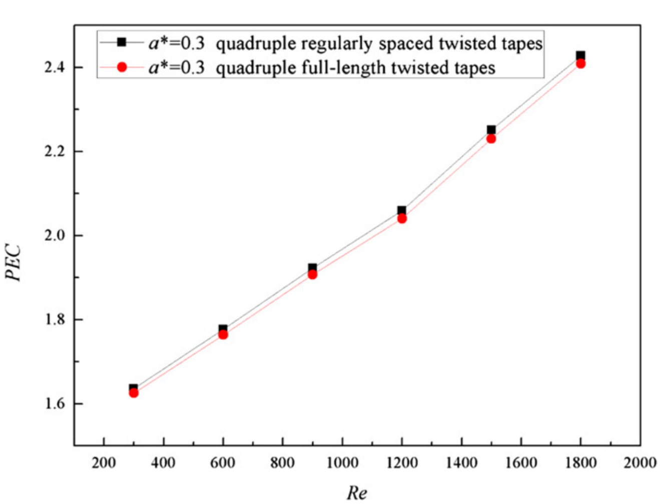

Fig. 21. Variation of PEC value with Reynolds number for quadruple regularly spaced and full-length twisted tapes.

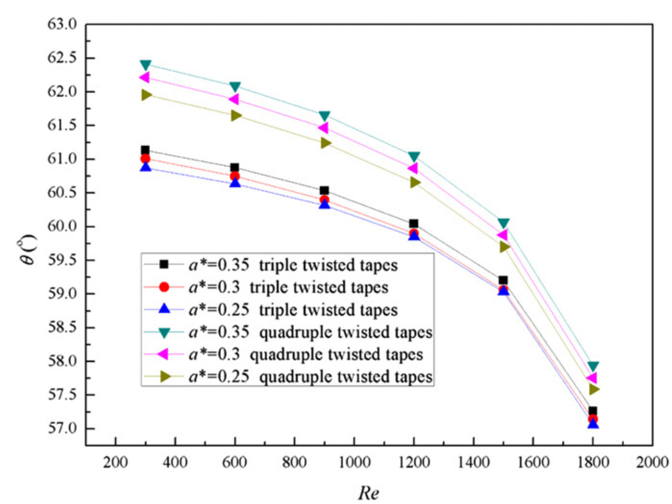

Fig. 23. The variation of average synergy angle θ with *Re* for triple and quadruple twisted tape of different clearance ratios.

Fig. 22 shows the effect of *Re* number on average synergy angle β for tubes with triple or quadruple twisted tapes. As shown in the figure, average synergy angle β between fluid velocity $\boldsymbol{U}$ and temperature gradient ∇T is decreased as the clearance of the twisted tapes increases at the situation of same number of twisted tapes. And the average synergy angle β in the tube of quadruple twisted tapes is smaller than that in the tube of triple twisted tapes. Compared with the Fig. 13, we can conclude that the smaller the average synergy angle β, the higher the heat transfer rate is. This is due to the fact that the synergy of fluid velocity and temperature gradient gets better as the average synergy angle β decreases, and therefore it can be known that heat transfer will be enhanced between the fluid and the tube wall.

Fig. 23 shows the variation of average synergy angle θ with *Re* for heat-transfer enhanced tubes with triple or quadruple twisted tapes. As shown in the figure, average synergy angle θ between fluid velocity $\boldsymbol{U}$ and pressure gradient ∇p is bigger as the clearance of the twisted tapes increases at the situation of same number of twisted tapes, and the average synergy angle θ in the tube of quadruple twisted tapes is bigger than that in the tube of triple twisted tapes. The result shows that the direction of velocity $\boldsymbol{U}$ deviates more greatly from the direction of pressure gradient ∇p, and flow resistance increases more remarkably, which is in good accordance with the friction factor shown in Fig. 17. Therefore, it is necessary to minimize the synergy between vectors $\boldsymbol{U}$ and ∇p for designing lower-resistance heat exchangers.

6. Conclusion

The numerical analysis of heat and fluid-flows through a round tube fitted with triple or quadruple twisted tapes of different clearance is carried out, with the aim to verify the thought of core flow heat transfer enhancement and investigate the effect of multi-longitudinal vortex on the flow, heat transfer and friction loss behaviors. The contour plots of predicted velocity, streamline and temperature are also presented. The major findings are summarized as follows:

1) The numerical results show that the mean heat transfer coefficients of the tube fitted with triple twisted tapes with $a^* = 0.25$, 0.3 and 0.35 are 162%, 164% and 171%, respectively, and the mean heat transfer coefficients of the tube fitted with quadruple twisted tapes with $a^* = 0.25$, 0.3 and 0.35 are 180%, 182% and 189%, respectively, higher than that of the plain tube. And the friction factors of the tube fitted with triple twisted tapes with $a^* = 0.25$, 0.3 and 0.35, are, respectively, around 4.06–4.74, 4.36–5.06 and 4.45–5.19 times as that of the plain tube, while the friction factors of the tube fitted with quadruple twisted tapes with $a^* = 0.25$, 0.3 and 0.35, are, respectively, around 5.33–6.27, 5.84–6.76 and 5.99–7.02 times as that of the plain tube. The PEC of the tubes varies from 1.64 to 2.46.
2) The physical quantity synergy analysis is performed to investigate the mechanism of heat transfer enhancement. The synergy angles β and θ are calculated, and the numerical results verify the synergy regulation among physical quantities of fluid particle in the flow field of convective heat transfer, which can guide us to design the optimum heat transfer units and heat exchangers.
3) The simulation results verify the theory of the core flow heat transfer enhancement which leads to the separation of the velocity boundary layer and the temperature boundary layer, and thus enhances the heat transfer greatly while the flow resistance is not increased very much.

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Nomenclature

a	the clearance of the twisted tapes (m)
a^*	the clearance ratio of the twisted tapes, $a^* = a/D$
A	the section area of the tube (m^2)
A_w	nominal inside tube area (m^2)
b	the broken length of the broken twisted tape (m)
b^*	the broken length ratio of the broken twisted tape, $b^* = b/L$
c_p	the specific heat at constant pressure (kJ/kg K)
D	the tube diameter (m)
f	friction factor
h	the average heat transfer coefficient in the tube (W/m^2 K)
h_{local}	the local heat transfer coefficient in the tube (W/m^2 K)
k	thermal conductivity (W/m K)
l	position along the flow direction (m)
l^*	position along the flow direction, dimensionless, $l^* = l/L$
L	the length of tube (m)
Nu	average Nusselt number
Nu_x	local Nusselt number
p	pressure (N/m^2)
q	local heat flux (W/m^2)
Q	total heat transfer rate (W)
r	radius (m)
r^*	dimensionless radius, $r^* = 2r/D$
Re	Reynolds number
s	pitch of twisted tape (m)
s^*	pitch of twisted tape (m), $s^* = s/2W$
T	temperature (K)
t	the thickness of twisted tape (m)
u	the flow velocity (m/s)
W	the width of twisted tape

Greek symbols

β	synergy angle (°)
θ	synergy angle (°)
ρ	density of water (kg/m^3)

Subscripts

0	plain tube
m	mean
w	wall

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