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A numerical study on heat transfer and friction factor characteristics of laminar flow in a circular tube fitted with center-cleared twisted tape

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ABSTRACT

Twisted tape is a widely used technique for heat transfer enhancement. In the present paper, we proposed a center-cleared twisted tape aiming at acxhieving good thermohydraulic performance. A comparative study between this type and the short-width twisted tape was performed numerically in laminar tubular flows. The computation results demonstrated that the flow resistance can be reduced by both methods; however, the thermal behaviors are very different from each other. For tubes with short-width twisted tapes, the heat transfer and thermohydraulic performance are weakened by cutting off the tape edge. Contrarily, for tubes with center-cleared twisted tapes, the heat transfer can be even enhanced in the cases with a suitable central clearance ratio. The thermal performance factor of the tube with center-cleared twisted tape can be enhanced by 7–20% as compared with the tube with conventional twisted tape. All these demonstrated that the center-cleared twisted tape is a promising technique for laminar convective heat transfer enhancement.

1. Introduction

Twisted tapes, as one of the passive heat transfer enhancement technology, have been extensively studied due to their advantages of steady performance, simple configuration and ease of installation. By generating swirls which enhance the fluid mixing of the near-wall and central regions, the heat transfer in tubes with twisted tapes could be enhanced [1–3]. Moreover, the twisted tape can partition and block the flow, reduce the hydraulic diameter, elongate the twisted flow path and generate a fin effect [1,2]. All these lead to additional heat transfer improvements. However, the thermal improvements are accompanied by increased pressure drop.

How to optimize the thermohydraulic performance of tubes fitted with twisted tapes has gained increasing attention [4–21]. For instance, Saha et al. [4] experimentally studied the heat transfer and pressure drop characteristics of laminar flow in a circular tube fitted with regularly spaced twisted tape elements connected with rod. The results showed that the pressure drop of the tube fitted with the segmented twisted tape elements is 40% smaller than that of the tube fitted with a continuous twisted tape, and the former has a better thermohydraulic performance. Eiamsa-ard et al. [5] experimentally investigated the convective heat transfer behaviors in a circular tube fitted with regularly spaced twisted tape elements in laminar and turbulent flows, and they found that the heat transfer coefficient and friction factor were both significantly reduced as compared with those of the tube fitted with a continuous twisted tape. Later, Saha et al. [6] further investigated the effects of the width of tape elements and the diameter of connecting rod on heat transfer and pressure drop characteristics. This work indicated that a narrower width of tape elements led to a worse thermohydraulic performance, while a thinner connecting rod resulted in a better one. Therefore, he proposed to abolish the connecting rod and 'pinch' the tube to fix the segmented twisted tape elements. Jaisankar et al. [7] studied the heat transfer and friction factor characteristics of thermosyphon solar water heater system using full-length twisted tape, and short-length twisted tapes with and without connecting rod. In their experiment the segmented twisted tape elements with several twist cycles were used, while in Saha and Eiamsa-ard's investigations the segmented twisted tape elements only have half twist cycle. Jaisankar et al. found that whether it is fitted with rod or not, the heat transfer coefficient and friction factor for the segmented twisted tape elements were both much smaller than those for the full-length twisted tape. Moreover, the overall performance for the segmented twisted tape elements connected with rod was better than that for the elements without rod.

Other researchers investigated the convective heat transfer behaviors in tubes fitted with short-width twisted tapes. In these cases, there is a gap between the twisted tape edge and tube wall,

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which might lead to a reduction in the flow resistance. For example, Ayub and Al-Fahed [8] experimentally investigated the effect of gap width on the pressure drop in turbulent flow. The results showed that increasing the gap width could effectively reduce the friction factor. Eiamsa-ard et al. [9] numerically studied the swirling flow and convective heat transfer in a circular tube fitted with loose-fit twisted tapes. They found that it was beneficial to reduce the pressure drop by reducing the width of twisted tape. However, the heat transfer coefficient and overall performance were also weakened. Later, Liu et al. [10] conducted a numerical investigation on heat transfer behaviors of laminar and turbulent flows in a circular tube fitted with short-width twisted tapes. Their results of laminar flow also showed a worse overall performance of this method.

Recently, some new types of twisted tapes were developed by some investigators [11–16]. Chang et al. [11] designed a serrated twisted tape with square-sectioned serrations on each side. Their investigation showed that the heat transfer enhancement attributed to the serrated twisted tape is about 1.25–1.67 times that of the tube fitted with smooth twisted tape in a *Re* range of 5000–25,000. Different from Chang et al.'s work [11], Eiamsa-ard and Promvonge [12] developed a twisted tape with serrated edges. Their experiment demonstrated that the heat transfer rate and thermal performance factor in the tube with this type of twisted tape insert were about 1.04–1.27 and 1.02–1.12 times those in the tube with smooth twisted tape insert, respectively. Later, Rahimia et al. [13] studied the heat transfer and friction factor characteristics of the tube fitted with perforated, notched and jagged twisted tapes. The results revealed that only the jagged insert was better than the conventional twisted tape in the heat transfer coefficient and thermal performance factor. More recently, Chang et al. [14] invented a broken twisted tape insert which can induce a better fluid mixing. They reported that, as compared with those of the tube fitted with smooth twisted tape, the heat transfer coefficient, mean Fanning friction factor and thermal performance factor of the tube fitted with broken twisted tape were augmented up to 1.28–2.4, 2.0–4.7 and 0.99–1.8 times, respectively, in a *Re* range of 1000–40000. Sivashanmugam and Suresh [15,16] proposed a helical screw tape which made the swirl flow rotate in only a single way smooth direction. Their experimental results indicated that the helical screw tape had better thermohydraulic performances as compared with the conventional twisted tape for both laminar and turbulent flows.

The compound heat transfer enhancement technologies have broadened the ability of twisted tapes [17–22]. Ray and Date [17,18] 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 [19,20] 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 [21] 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. [22] 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 laminar tubeside flow of highly viscous fluid.

Although tremendous efforts have been made to enhance the convective heat transfer in tubes and ducts with the twisted tape inserts, to achieve a good heat transfer performance as well as a low friction factor is still admirable. For these objects, we proposed a center-cleared twisted tape in the present work. In this method, the upwind area blocking the flow in the tube is reduced. At the meantime, the disturbance of boundary layers is not weakened that much. Therefore, it is expected that this kind of twisted tape might be capable of reducing the flow resistance and maintaining a high heat transfer rate as well. The heat transfer and friction factor characteristics of laminar flow in a circular tube fitted with the center-cleared twisted tape will be investigated through numerical simulation. Comparisons will also be made with the short-width twisted tape which has been demonstrated to be able to reduce the flow resistance by partly cutting off the tape edge [8,9].

2. Physical model

The geometries of the conventional, short-width and center-cleared twisted tapes are depicted in Fig. 1. Twisted tapes with thickness (δ) of 0.001 m are fitted in the full length of all tubes. The diameter (D) and length (L) of the tube are 0.02 m and 0.5 m, respectively. The 180 deg twist pitch (H) is 0.05 m and thus the relative twisted ratio (H/D) is 2.5. The effects of the tape width ratio ($w = W/D$) and central clearance ratio ($c = C/D$) on the heat transfer and friction factor characteristics will be investigated.

The Reynolds number (Re), Nusselt number (Nu), friction factor (f) and thermal performance factor (η) proposed by Webb [23] are defined as follows:

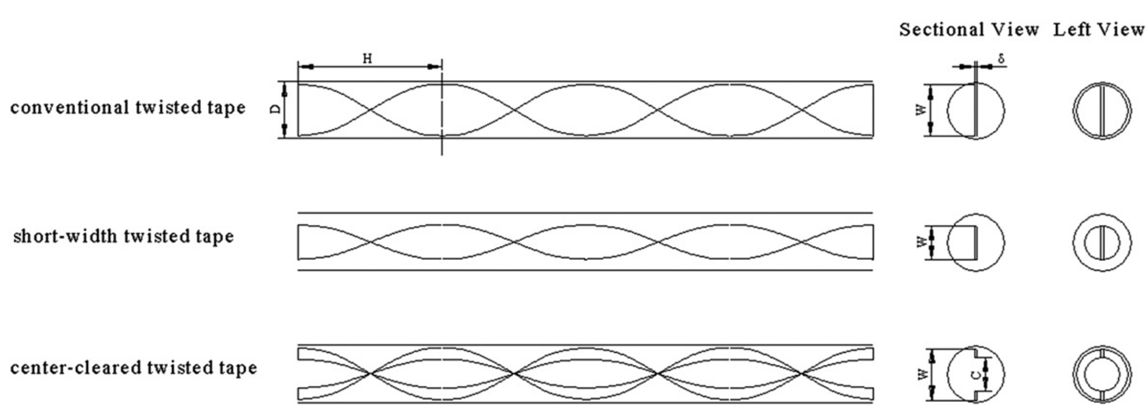

Fig. 1. The enhanced tube fitted with conventional, short-width and center-cleared twisted tapes.

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

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

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

$$\eta = \frac{Nu/Nu_0}{(f/f_0)^{(1/3)}} \tag{4}$$

where, Nu_0 and f_0 are the Nusselt number and the friction factor of the plain tube, respectively. Water is selected as the working fluid which is assumed to be incompressible. The natural convection has been neglected and the thermophysical properties of fluid are assumed to be temperature independent. The dynamic viscosity (μ), thermal conductivity (k), density (ρ) and specific heat at constant pressure (C_p) of water are given as $\mu = 1.003 \times 10^{-3}$ kg m^{-1} s^{-1}, $k = 0.6$ W m^{-1} K^{-1}, $\rho = 998.2$ kg m^{-3} and $C_p = 4182$ J kg^{-1} K^{-1}, respectively. The Reynolds numbers referred to the inlet values are set at 500, 750, 1000, 1250, 1500 and 1750 in the computations.

3. Mathematical model and numerical method

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:

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0 \tag{5}$$

Momentum equation:

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] \tag{6}$$

Energy equation:

$$\frac{\partial}{\partial x_j}\left(\rho u_j C_p T - k\frac{\partial T}{\partial x_j}\right) = 0 \tag{7}$$

A constant heat flux is imposed on the tube wall. At the inlet, the fully developed profiles of velocity and temperature are specified [24], as shown in Equations (8) and (9), respectively. At the outlet, a pressure-outlet condition is used. On the surfaces of the tube wall and twisted tape, no slip conditions are applied.

$$u = u_c\left(1 - \frac{r^2}{R^2}\right) \tag{8}$$

$$T = T_c + \frac{qR}{k}\left[\left(\frac{r}{R}\right)^2 - \frac{1}{4}\left(\frac{r}{R}\right)^4\right] \tag{9}$$

where, u_c and T_c are the velocity and temperature at the centerline of the tube respectively. q is the heat flux density on the tube wall. R is the inner radius of the tube, and r is the radial distance.

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,

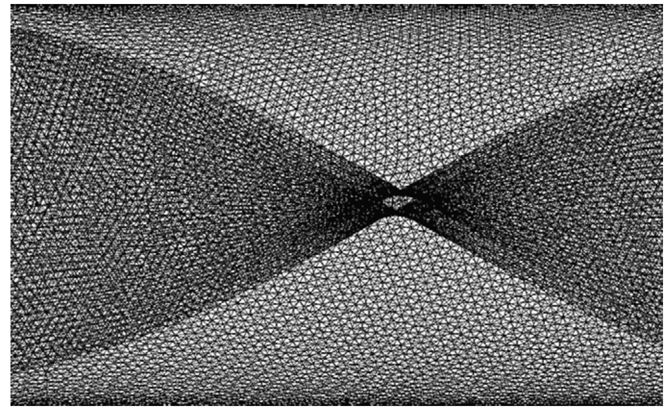

Fig. 2. Part of the tube with twisted tape and tetrahedral grid generation.

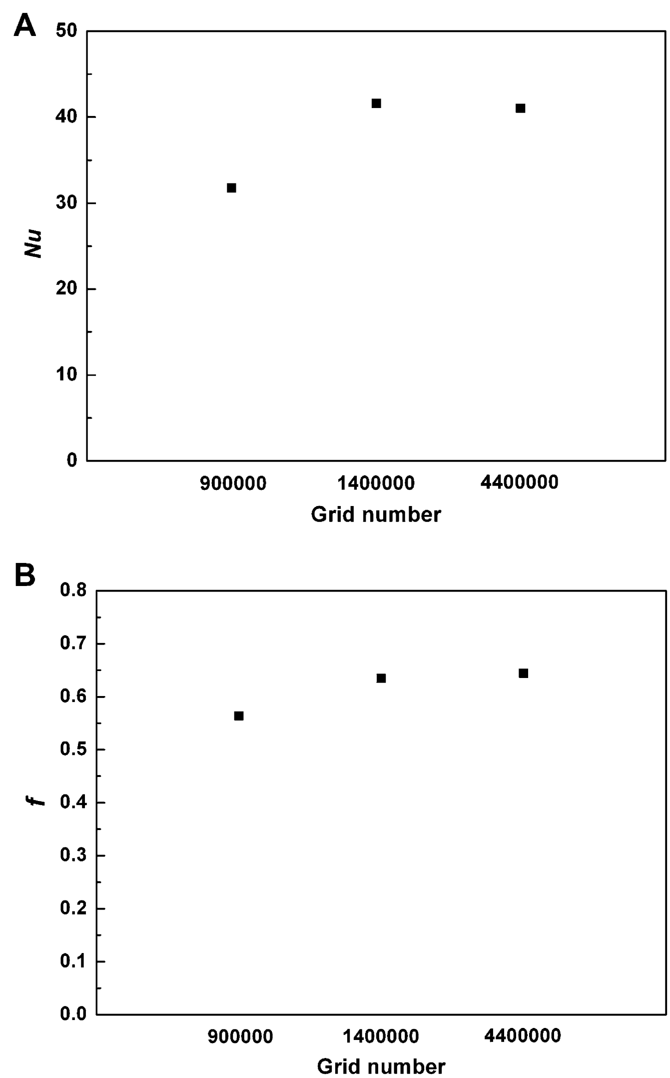

Fig. 3. The Nusselt number (Nu) and the friction factor (f) calculated by different grid systems: (a) Nu, and (b) f.

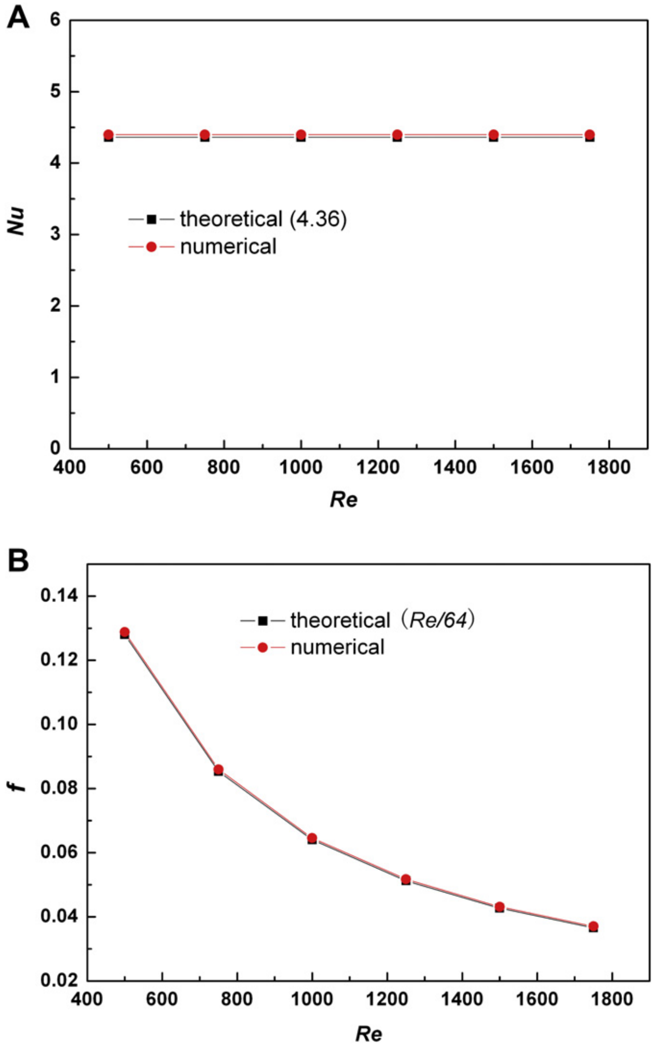

Fig. 4. Comparison of the numerical results and the theoretical data of the Nusselt number (Nu) and the friction factor (f) of the plain tube: (a) Nu, (b) f.

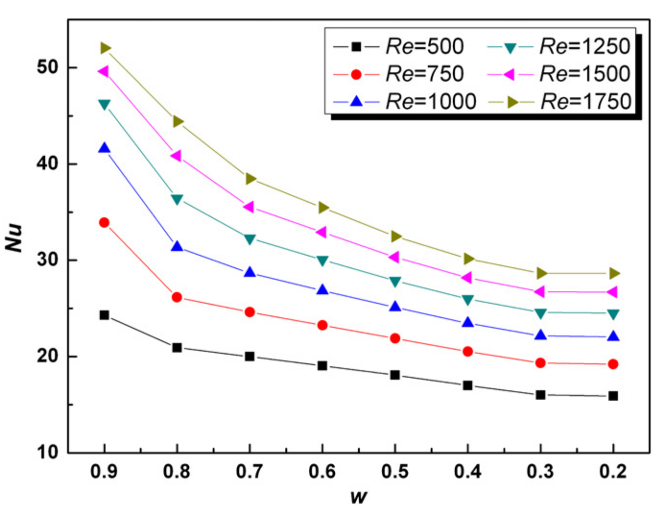

Fig. 5. Variation of the Nusselt number (Nu) versus the tape width ratio (w) for tube fitted with short-width twisted tape.

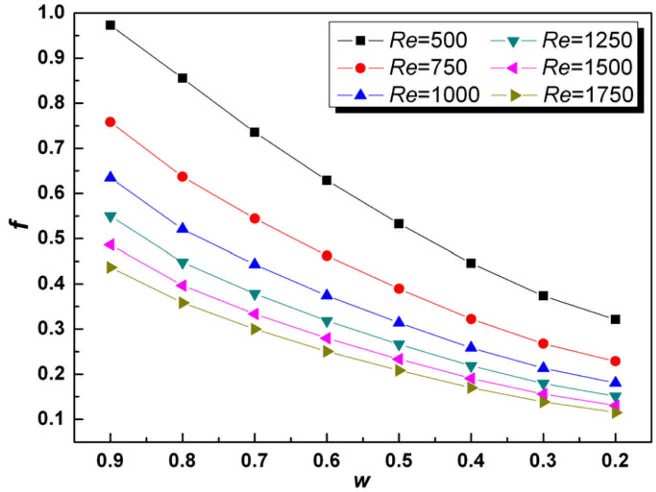

Fig. 6. Variation of the friction factor (f) versus the tape width ratio (w) for tube fitted with short-width twisted tape.

convergence criteria of 10^{-5} for continuity and velocity components and 10^{-8} for energy are used, respectively.

4. Results and discussions

Grid independent test has been performed for the physical model. The tetrahedral grid is used for meshing, as depicted in Fig. 2. The grid is highly concentrated near the wall and in the vicinity of the twisted tape. Three grid systems with about 900,000, 1,400,000, 4,400,000 cells are adopted to calculate a baseline case in which the Reynolds number is 1000 and the tape width ratio is 0.9. The Nusselt numbers (Nu) and the friction factors (f) of the baseline case are shown in Fig. 3, which demonstrates that the difference between the calculated results of 1,400,000 and 4,400,000 cells is very small. Therefore, the gird system with 1,400,000 cells is adopted for 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 condition. From Fig. 4 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.

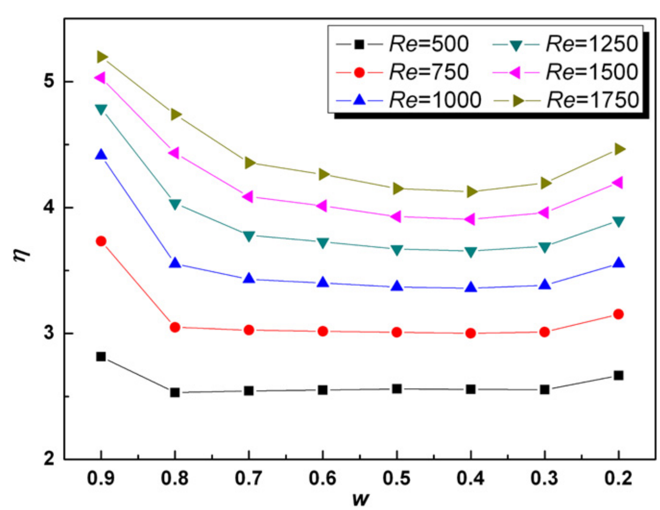

Fig. 7. Variation of the thermal performance factor (η) versus the tape width ratio (w) for tube fitted with short-width twisted tape.

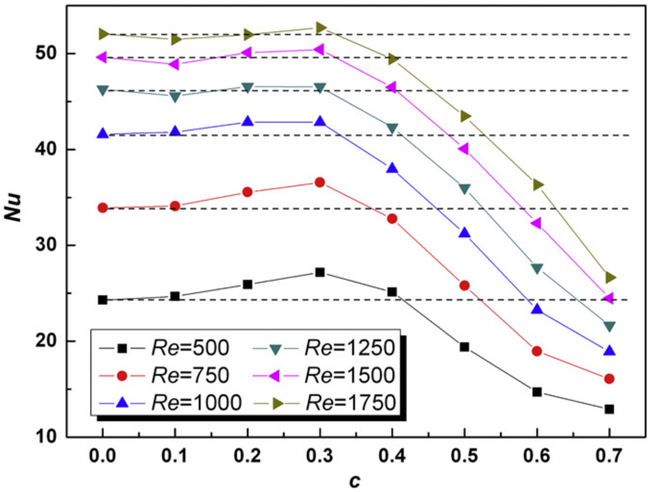

Fig. 8. Variation of the Nusselt number (Nu) versus the central clearance ratio (c) for tube fitted with center-cleared twisted tape.

For the convenience of comparison, we first present the results of the short-width twisted tape in subsection 4.1. Then, the results of the center-cleared twisted tape will be reported in subsection 4.2. Analyses of the flow and temperature fields will be discussed in the last subsection.

4.1. Effects of the tape width ratio

Figs. 5–7 show the variations of the Nusselt number (Nu), the friction factor (f) and the thermal performance factor (η) with the tape width ratio (w) at different Reynolds numbers (Re) of laminar flow. From these figures it is seen that the Nusselt number and the thermal performance factor increase as the Reynolds number increases, while the friction factor decreases with the increase of the Reynolds number.

From Fig. 5 it is seen that, for each Re, the Nusselt number decreases as the width ratio decreases, and the larger the width ratio, the better the heat transfer enhancement is. Moreover, it is also noted that Nu decreases more rapidly when w is relatively large and then the decrease of Nu slows down with further reduction of w. For example, when w is reduced from 0.9 to 0.8, Nu decreases by 14–25%. However, when w is reduced to 0.2, Nu decreases by 35–47% as compared with the value when $w = 0.9$. This demonstrates that the weakening of the heat transfer mainly happens when w is reduced from 0.9 to 0.8, and in some cases ($Re = 500, 750, 1000$) the decrement of Nu in this procedure is even larger than that in the procedure when w is reduced from 0.8 to 0.2. There are two reasons responsible for the weakening in heat transfer by the reduction of w. First, as the width ratio decreases, the disturbance of the boundary layer, as well as the fluid mixing of the boundary layer and core flow is weakened. Second, a decrement of the width ratio leads to an increment of the twist ratio (H/W), which results in weaker swirls generated by the twisted tape.

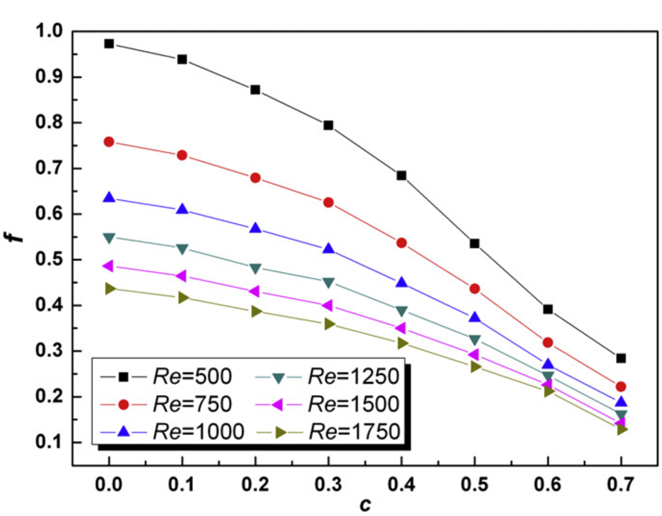

Fig. 9. Variation of the friction factor (f) versus the central clearance ratio (c) for tube fitted with center-cleared twisted tape.

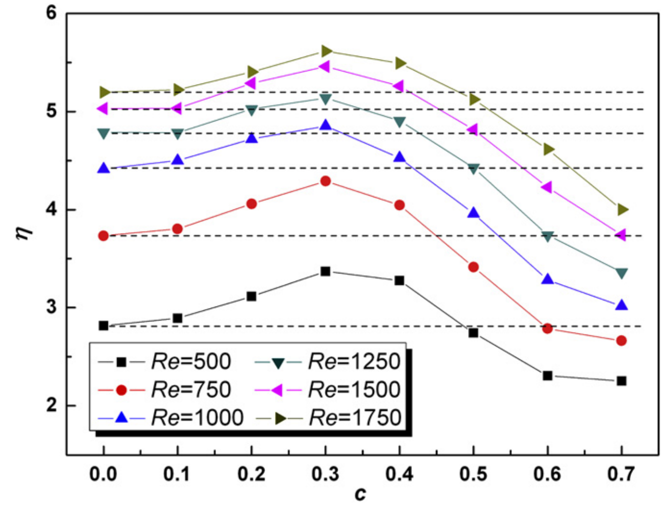

Fig. 10. Variation of the thermal performance factor (η) versus the central clearance ratio (c) for tube fitted with center-cleared twisted tape.

From Fig. 6 it can be seen that the friction factor decreases with the reduction of the width ratio at each Reynolds number. For example, the magnitude of f when $w = 0.2$ decreases by 67–74% as compared with that when $w = 0.9$. It is also noted that the decrement of f is not so rapid as compared with that of Nu when w is relatively large. For instance, when w is reduced from 0.9 to 0.8, the friction factor only decreases by 12–17%, while the decrement of Nu is up to 14–25%. Furthermore, the friction factor drop when w is reduced from 0.9 to 0.8 does not possess that large a proportion as compared with Nu in the total decrement when w is reduced from 0.9 to 0.2.

It is clearly seen from Fig. 7 that variation tendency of the thermal performance factor (η) is quite different from those of Nu and f. In other words, η decreases at first and then increases with the reduction of w. The magnitude of η declines sharply when w is reduced from 0.9 to 0.8, and reaches a minimum value approximately at $w = 0.4$, then it ascends slightly with the further

Table 1
The percentage of thermal performance enhancement for the center-cleared twisted tape compared with the smooth twisted tape.

$\Delta\eta$ (%)	c						
	0.1	0.2	0.3	0.4	0.5	0.6	0.7
Re							
500	2.7	10.5	19.7	16.3	−2.6	−18.1	−20.0
750	1.9	8.7	14.9	8.4	−8.5	−25.4	−28.7
1000	1.9	6.9	9.9	2.5	−10.3	−25.6	−31.7
1250	0.0	5.0	7.3	2.5	−7.5	−21.9	−29.8
1500	0.0	5.1	8.5	4.6	−4.3	−16.0	−25.6
1750	0.5	4.0	8.1	5.7	−1.4	−11.2	−23.0

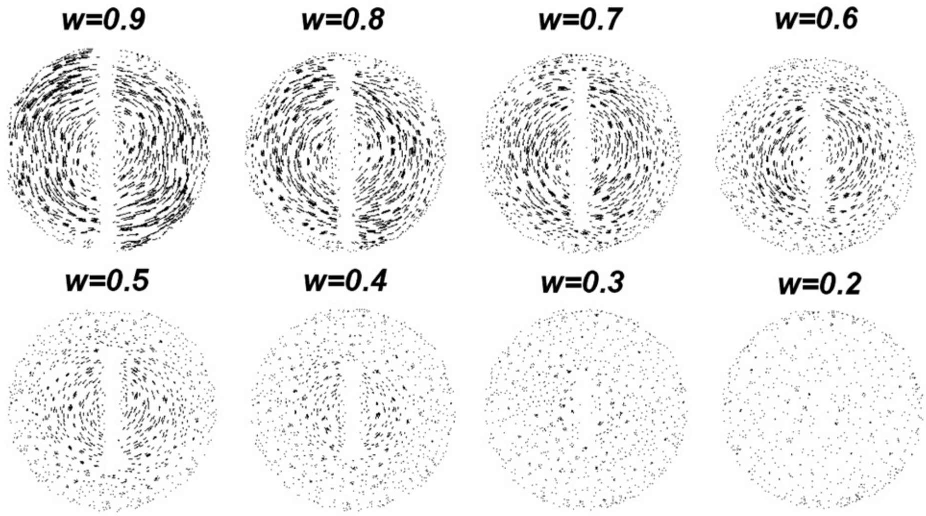

Fig. 11. Plots of the tangential velocity vectors on cross-sectional planes at $L = 0.25$ m in tubes fitted with short-width twisted tapes of different tape width ratios.

reduction of the width ratio. Moreover, it also could be noted that thermal performance factor at $w = 0.9$ is much larger than those when $w < 0.9$. All those show that it is unfavorable to enhance the thermohydraulic performance by reducing the outer edge of the twisted tape.

4.2. Effects of the central clearance ratio

Fig. 8 shows the variation of the Nusselt number (Nu) with the central clearance ratio (c) at different Reynolds numbers (Re) of laminar flow. Comparing the Fig. 5 with Fig. 8, it is obvious that cutting off the edge of the twisted tape leads to a decrement in the heat transfer, but removing away the center of the twisted tape does not mean weakening the heat transfer. In some cases, the heat transfer is even enhanced to some extent.

From Fig. 8 it can be seen that when the clearance ratio is 0.1, the Nusselt numbers are slightly larger than those of the tube with smooth twisted tape ($c = 0$) in Re range of 500–1000. Contrarily, in Re range of 1250–1750, the Nusselt numbers of $c = 0.1$ are slightly smaller as compared with those of $c = 0$. After that, Nu increases gradually for all the Reynolds numbers with the increment of the central clearance ratio until it reaches the peak at $c = 0.3$ and then declines gradually. The value of Nu at $c = 0.3$ is 1–12% larger than that at $c = 0$. This indicates that in terms of heat transfer enhancement, $c = 0.3$ could be the most beneficial. Furthermore, the corresponding value of c (termed c_c), at which the Nusselt number decreases below the horizontal line which is representative for the value of Nu at $c = 0$, is different at different Re. In general, c_c is larger when Re is lower, and it decreases with the increment of Re until $Re = 1250$.

The plausible mechanism responsible for these characteristics is discussed as below. It is well known that the heat transfer enhancement technique with a normal twisted tape is based on the swirls generated by the twisted tape. Once the continuous surface of the normal twisted tape is broken, the swirls in the tube become more complicated, which enhances the mixing of the fluid. In other words, there are two effects of heat transfer enhancement in the case of center-cleared twisted tape. One is the basic effect that generates the swirls, which is slightly weakened because of the tape cutting. And the other is the extra effect that complicates the swirls, which is positive for heat transfer enhancement. This extra function is especially effective for the cases with relatively small Reynolds numbers, under which the original swirling flows are very weak. When the central clearance ratio is too small (i.e. $c = 0.1$), the extra effect has little impact on the heat transfer for the cases with relatively large Reynolds numbers (i.e. $Re = 1250$, 1500, 1750), which leads to a decrease of Nu. However, the impact of the extra effect grows greater with the increment of c; correspondingly, the Nusselt number increases.

Fig. 9 depicts the variation of the friction factor (f) with the central clearance ratio (c) at different Reynolds numbers (Re) of laminar flow. It is seen that the friction factor decreases as the

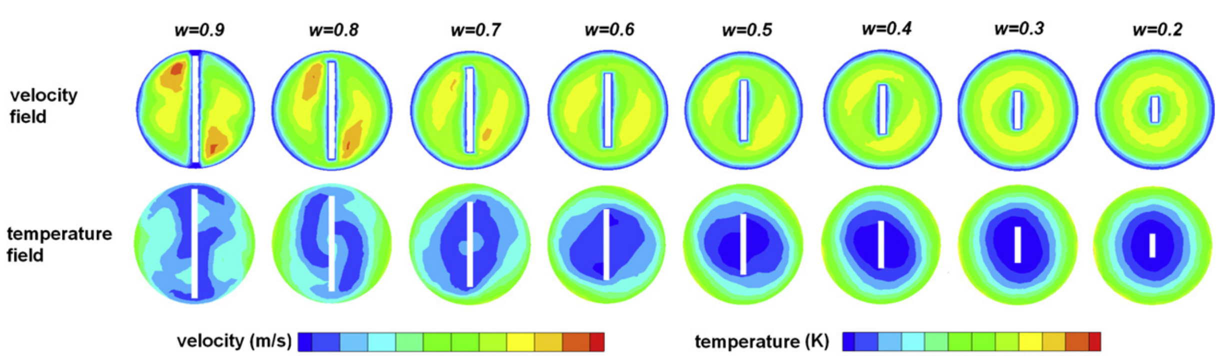

Fig. 12. Plots of the velocity and temperature contours on cross-sectional planes at $L = 0.25$ m in tubes fitted with short-width twisted tapes of different tape width ratios.

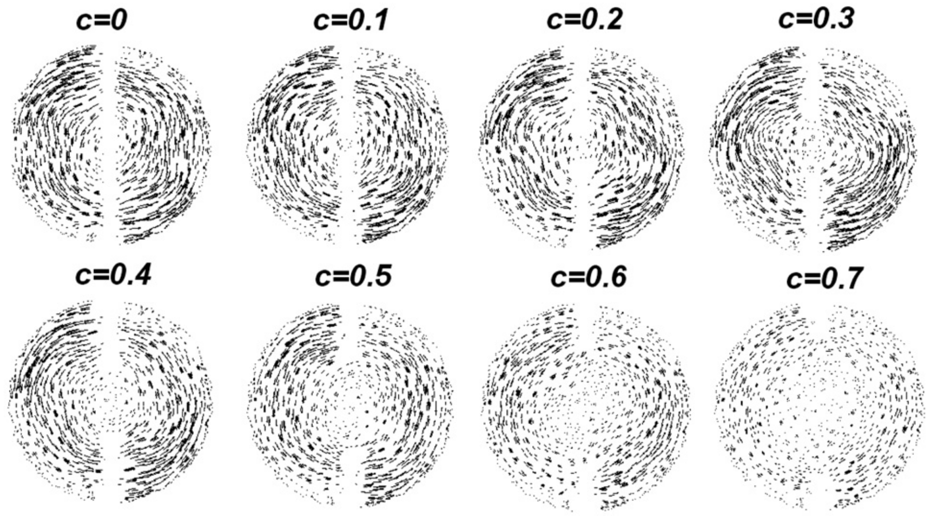

Fig. 13. Plots of the tangential velocity vector on cross-sectional planes at $L = 0.25$ m in tubes fitted with center-cleared twisted tapes of different central clearance ratios.

central clearance ratio increases. This is because more blocking area is reduced at a larger central clearance ratio. Furthermore, the friction reduction effect is more obvious in the cases with smaller Reynolds numbers.

Fig. 10 presents the variation of the thermal performance factor (η) with the central clearance ratio (c) at different Reynolds numbers (Re) of laminar flow. It is shown that as c increases, η increases first until it reaches a peak at $c = 0.3$ and then decreases. The maximum value of η is 1.07–1.20 times of that at $c = 0$ in Re range of present study. More details about the enhancement of thermal performance factor for the center-cleared twisted tape compared with the smooth twisted tape ($c = 0$) are shown in Table 1.

4.3. Analyses on flow and temperature fields

To explain the differences between the results caused by the short-width and center-cleared twisted tapes, we take $Re = 1000$ as an example and present the plots of tangential velocity vectors, velocity contours and temperature contours of the same location at $L = 0.25$ m in Figs. 11–14, respectively. From Fig. 11, it is seen that when $w = 0.8$, the tangential velocity near the wall is much smaller than that when $w = 0.9$, which means a much weaker disturbance and fluid mixing of the boundary layer. A further reduction of the tape width ratio not only makes the near-wall swirls disappeared, but also weakens the swirls in the whole area.

The temperature fields shown in Fig. 12 demonstrate that, as the tape width ratio decreases, the thermal boundary layer grows thicker. In other words, temperature gradient near the wall becomes smaller, especially in the reduction process from $w = 0.9$ to 0.8. The variation tendencies of the flow and temperature fields agree well with the phenomena shown in Fig. 5. This verifies that the disturbance of boundary layer is the key factor for the heat transfer enhancement technique with a twisted tape. Once the outer edge of the twisted tape is far away from the tube wall, the near-wall swirls will be seriously weakened, and thus leading to a great weakening in the heat transfer.

Therefore, instead of cutting off the edge, we clear the central part of the twisted tape. It is expected that this method can maintain the disturbance of the boundary layer and reduce the flow resistance as well. The velocity vector fields of different central clearance ratios are depicted in Fig. 13. From this figure we can see that as the central clearance ratio increases, the swirls near the wall still exist and the intensity of the swirls stays almost unchanged until $c = 0.3$. Looking into the variations of the velocity vector fields (Fig. 13), and the contours of velocity and temperature (Fig. 14), we can divide the whole process into two stages. The first stage is from $c = 0$ to 0.3. In this stage the velocity vector fields, and the contours of velocity and temperature show very small variations, which implies that expansion of the central clearance has slight effect on the heat transfer in this stage. In the second stage ($c = 0.4$ to 0.7), the velocity vector fields, and the contours of velocity and temperature exhibit obvious variations as compared with those of the first stage. It is clearly seen from Fig. 14 that the high-speed flow region completely appears in the central part of the tube and the

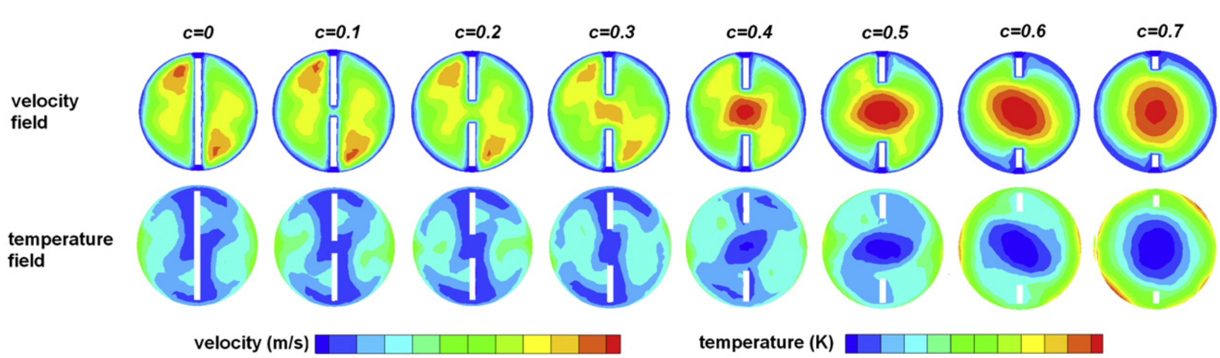

Fig. 14. Plots of the velocity and temperature contours on cross-sectional planes at $L = 0.25$ m in tubes fitted with center-cleared twisted tapes of different central clearance ratios.

near-wall swirls have been greatly weakened. Correspondingly, the thermal boundary layer grows thicker, i.e., the heat transfer at the wall will be seriously deteriorated. The characteristics of these two stages are in accordance with the phenomena shown in Fig. 8.

5. Conclusions

Heat transfer and friction factor characteristics of laminar flow in a circular tube with short-width and center-cleared twisted tapes have been investigated numerically. The computation results show that the flow resistance can be reduced by both methods but the heat transfer features are very different from each other. For tubes with short-width twisted tapes, the heat transfer and the overall performance are weakened by cutting off the tape edge. Contrarily, for the tubes with center-cleared twisted tapes, the heat transfer can be even enhanced in the cases with a suitable central clearance ratio. As compared with the tube with conventional twisted tape, the thermal performance factor of the tube with center-cleared twisted tapes can be enhanced by 7–20%. In summary, one can achieve a satisfying overall performance by using a twisted tape with a suitable central clearance ratio. Therefore, the center-cleared twisted tape is a promising technique for laminar convective heat transfer enhancement.

Acknowledgement

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Nomenclature

W	width of the twisted tape, m
H	180 deg twist pitch of the twisted tape, m
C	width of the central clearance, m
L	length of the tube, m
D	inner diameter of the tube, m
R	inner radius of the tube, m
r	radial distance, m
x_i	space coordinates in Cartesian system, m
w	tape width ratio $= W/D$
c	central clearance ratio $= C/D$
c_c	critical value of the central clearance ratio
k	thermal conductivity of fluid, W m^{-1} K^{-1}
T	temperature, K
T_c	temperature at the centerline of the tube, K
C_p	specific heat at constant pressure, J kg^{-1} K^{-1}
q	heat flux density, W m^{-2}
h	heat transfer coefficient, W m^{-2} K^{-1}
u	flow velocity, ms^{-1}
u_c	flow velocity at the centerline of the tube, m s^{-1}
Δp	pressure drop between tube entry and exit, N m^{-2}
Nu	Nusselt number
Nu_0	Nusselt number of plain tube
f	friction factor
f_0	friction factor of plain tube
Re	Reynolds number
Pr	Prandtl number
δ	thickness of the twisted tape, mm
η	thermal performance factor
ρ	fluid density, kg m^{-3}
μ	fluid dynamic viscosity, kg m^{-1} s^{-1}

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