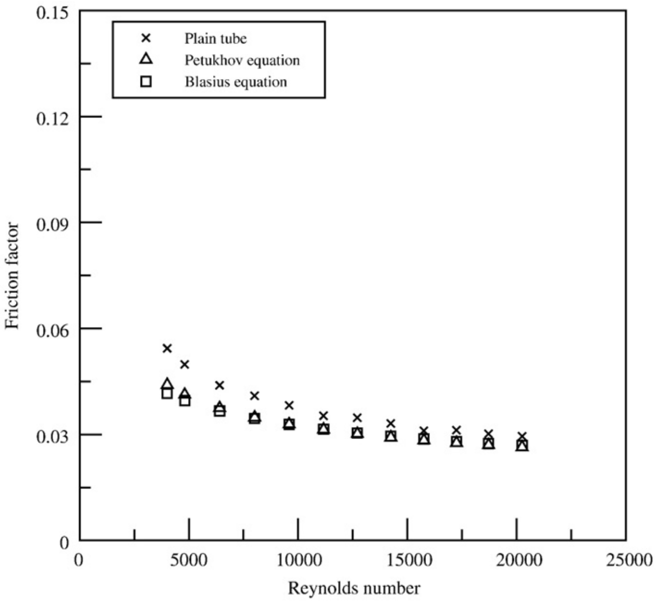

Fig. 4. Confirmation of plain tube for friction factor.

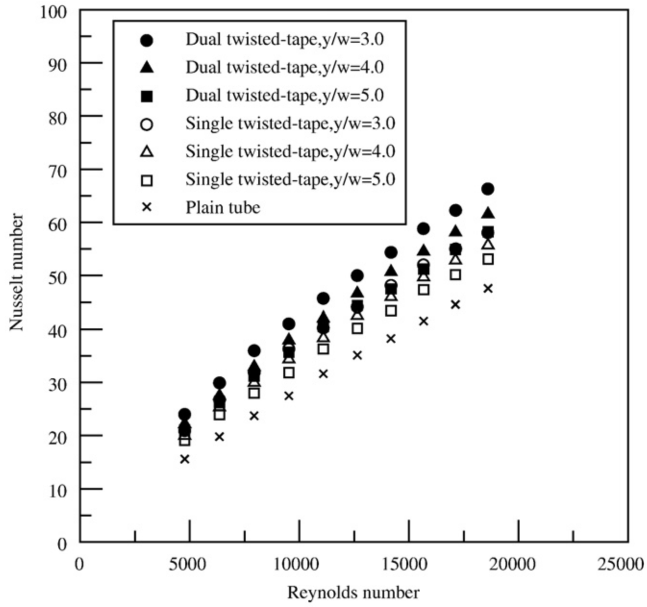

Fig. 5. Effect of single and full-length dual twisted tape inserts on Nusselt number.

in Fig. 5. As seen in this figure, the increase in heat transfer rate with decreasing twist ratio is due to the raise of swirl intensity imparted to the flow at the tube wall. This swirl flow results in the reduction of boundary layer thickness and the increase in flow velocity. Inside the test tube, the boundary layer separation occurs due to the radial velocity components of the flow and the rotation/swirl induced by the twisted tape. In addition, the swirl enhances the flow turbulence intensity, leading to better convection heat transfer than the axial flow of the plain tube. Thus, the higher swirling flow, the greater Nusselt number values become. For all twist ratios, the swirling flow produced by the dual twisted tapes provides higher heat transfer rate than those by the single twisted tape. The dual twisted tapes generates a double swirl flow leading to (1) the increase in residence time of the flow due to its separation into two streams (upper and lower tapes) and (2) increasing turbulent intensity of the flow giving better mixing of fluid between the tube wall and the core. In general, the dual twisted tapes yields higher heat transfer rate than the single one around 12% to 15%. This improvement for $y/w=3.0$ is around 17% to 20% and 24% to 29% higher than those for the $y/w=4.0$, and 5.0, respectively. For the lowest twist ratio ($y/w=3.0$) of the dual twisted tapes, the increase in heat transfer is in a range of 140% to 154% over the plain tube.

The influences of the dual twisted tape inserts on friction factor characteristic are depicted in Fig. 6. The figure shows the relationship between the friction factor and Reynolds number at different twist ratios (y/w) for all tube fitted with/without twisted tapes. It is found that use of the dual twisted tapes gives rise to the friction factor values higher than that of the plain tube and the single one. The higher friction loss mainly comes from the increased surface area and higher swirl intensity. The friction factor is reduced considerably by using lower twist ratio, $y/w=4.0$ and 5.0, instead of $y/w=3.0$. Moreover, the dual one yields higher friction factor than the single one around 1.3, 1.4 and 1.6 times for $y/w=3.0$, 4.0 and 5.0, respectively.

Fig. 7 presents the relationship between the Reynolds number and the heat transfer enhancement efficiency in the tube fitted with dual twisted tapes at several twist ratios. Throughout the results of the experimentation, it appears that use of the smaller twist ratio ($y/w=3.0$) leads to higher enhancement efficiency than that of the larger ones ($y/w=4.0$ and 5.0). Enhancement efficiencies vary from 1.03 to 1.11, 1.01 to 1.08, and 0.99 to 1.06 for $y/w=3.0$, 4.0 and 5.0, respectively. It can be seen that the enhancement efficiency increases with the reduction of twist ratio values. However, use of the dual twisted tapes is not feasible in terms of energy saving at higher Reynolds numbers because it provides higher energy loss in the fluid flow. It is expected that the performance improvement from using the dual twisted tapes in the some intervals of Reynolds number and its sizes is progressively developing. Thus, it is possible to use the device with the various styles of swirl generator inserts to reduce the pressure drop and to increase the heat transfer.

4.3. Effect of space ratio

The effects of the dual twisted tapes in tandem arrangement at the constant twist ratio, $y/w=3.0$ with several space ratios ($s/D=0.0$ (full length), 0.75, 1. 5 and 2.25) on the heat transfer rate in a uniform heat flux tube are shown in Fig. 8. The experimental data on heat transfer in terms of Nu are plotted against the Reynolds number. The influence of the dual twisted tapes on heat transfer is more intense at

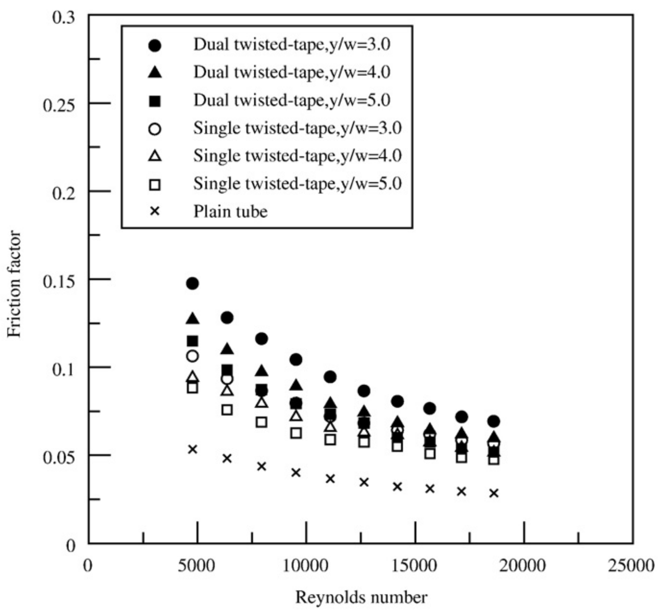

Fig. 6. Effect of single and full-length dual twisted tape inserts on friction factor.

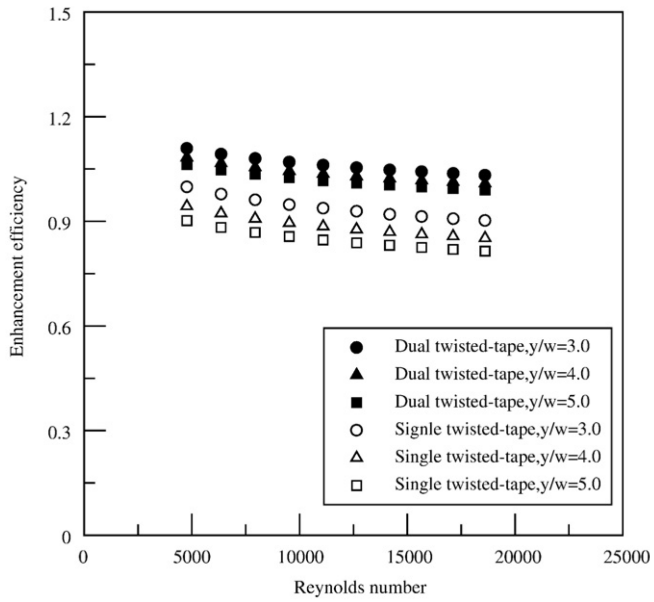

Fig. 7. Effect of single and full-length dual twisted tapes on enhancement efficiency.

higher Reynolds number, related to swirling flow and the breakdown of boundary layers in a shorter time. The tube with dual twisted tape inserts show higher heat transfer than that without twisted tape insert. It can be observed that the mean increase in heat transfer rate is around 146% for the full-length tape ($s/D=0.0$). The increases in heat transfer for using the dual one with space ratios (s/D) of 0.75, 1. 5 and 2.25 are approximately 140%, 137% and 132%, respectively. The Nusselt number values increase with decreasing the space ratio value. The Nusselt number for the dual one with space ratio of 0.75 is about 2.8% to 4.3% lower than that with the full-length tape ($s/D=0.0$).

Fig. 9 shows the relation between the friction factor and the Reynolds number obtained from different space ratios of the tandem tape elements. It is observed that the friction factor tends to reduce rapidly up to the Reynolds number of 14,000. The friction factor for the dual one is higher than that of the plain tube for all space ratios (s/D) used. The maximum value of friction factor is found at the case of

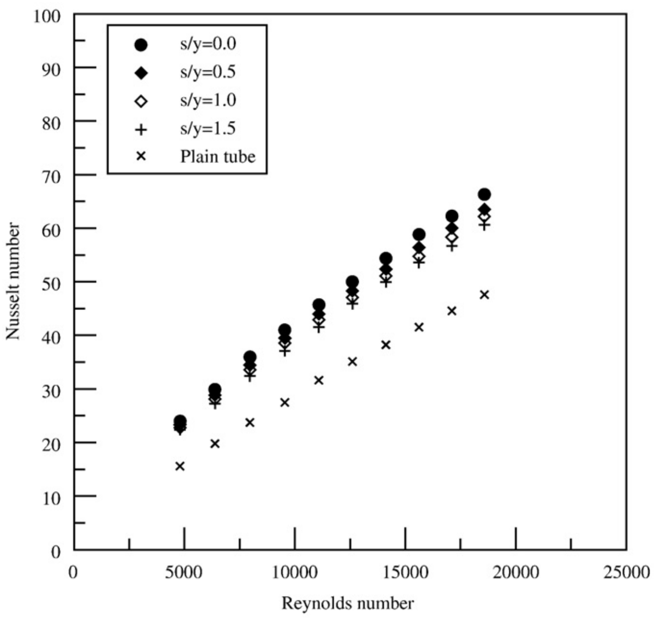

Fig. 8. Effect of dual twisted tape elements in tandem on Nusselt number.

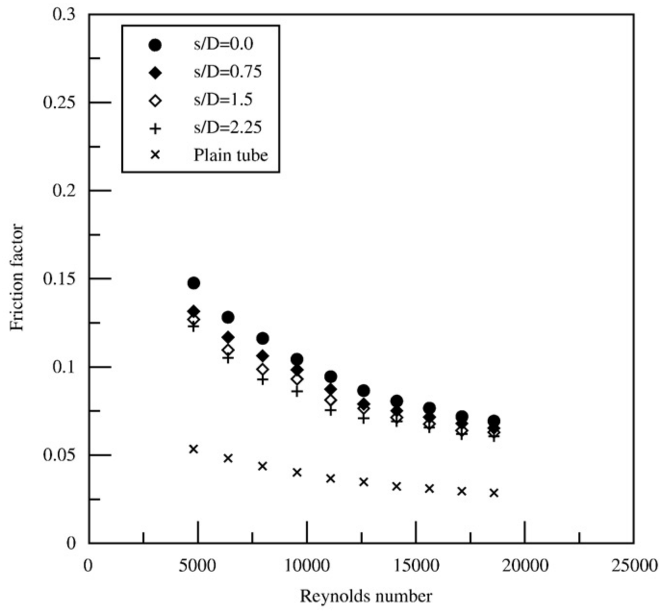

Fig. 9. Effect of dual twisted tape elements in tandem on friction factor.

space ratio, $s/D=0.0$ or without free spacing. The friction factors of all dual ones are similar at high Reynolds number. The mean increases in friction loss of the dual ones are 2.55, 2.36, 2.24 and 2.13 times over the plain tube for $s/D=0.0$, 0.75, 1.5 and 2.25, respectively.

The effects of the dual twisted tape elements in tandem with different space ratios (s/D) on the enhancement efficiency are presented in Fig. 10. The enhancement efficiency is useful to evaluate the quality of the heat transfer enhancement concept. As seen in the figure, the efficiency is higher than unity for lower Reynolds number at all space ratios (s/D) used. The efficiency tends to decrease with the rise in Reynolds number for all space ratios. The efficiency value is found to be 1.03 to 1.11, 1.02 to 1.09, 1.01 to 1.08 and 1.0 to 1.07 for space ratios, $s/D=0.0$ (full-length tape), 0.75, 1.5 and 2.25, respectively. Thus, the dual twisted tape elements in tandem arrangement should be preferably applied at low Reynolds number and space ratio (s/D) to obtain an optimum gain.

The mean Nusselt number and friction factor for the tube with dual twisted tape inserts are correlated as a function of Reynolds number, Prandtl number (Pr), twist ratio (y/w) and space ratios (s/D), and they are formulated as below. The plot of the Nusselt number and friction factor values of the tube with the dual twisted tape inserts, predicted from Eqs. (23) and (24), are showed in Figs. 11 and 12, respectively. In the figures, the majority of the measured data falls within $\pm 10\%$ and $\pm 10\%$, respectively, for the present Nusselt number and friction factor correlations. The correlations of the heat transfer enhancement efficiency of the tube fitted with single twisted tape and the dual twisted tape elements in tandem, are also presented below for several twist ratios (y/w) and space ratios (s/D).

Correlations for Nusselt number, friction and enhancement efficiency of the tube with single twisted tape insert at several twist ratios ($y/w=3.0$, 4.0 and 5.0) are expressed as:

$$Nu = 0.06\,Re^{0.75}Pr^{0.4}(y/w)^{-0.26} \tag{20}$$

$$f = 10.02\,Re^{-0.46}(y/w)^{-0.48} \tag{21}$$

$$\eta = 2.4Re^{-0.08}(y/w)^{-0.2} \tag{22}$$

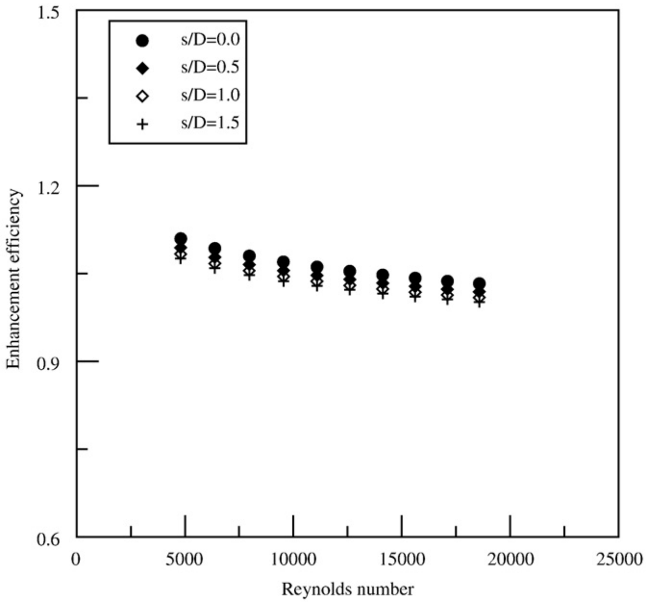

Fig. 10. Effect of dual twisted tape elements in tandem on enhancement efficiency.

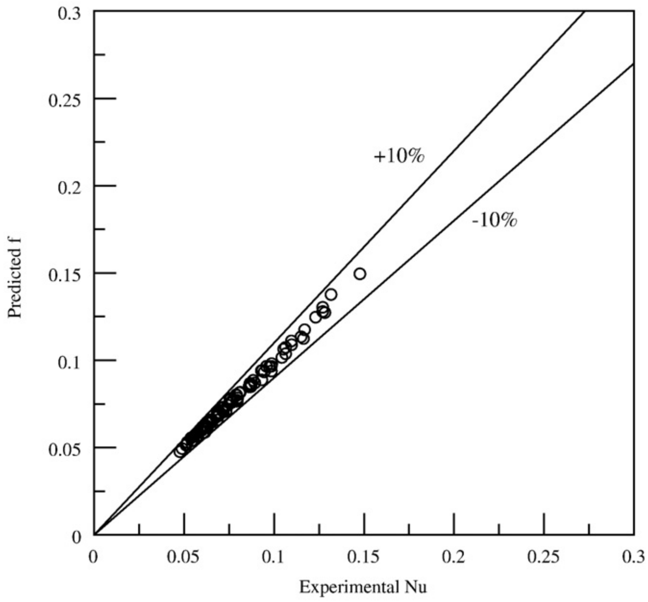

Fig. 12. Prediction of friction factor versus experimental result.

Correlations for Nusselt number, friction and enhancement efficiency of the tube with dual twisted tape elements in tandem at several twist ratios ($y/w=3.0$, 4.0 and 5.0) and space ratios ($s/D=0.0$ (full length), 0.75, 1.5 and 2.25) are written as:

$$Nu = 0.069\,\mathrm{Re}^{0.74}\mathrm{Pr}^{0.4}(y/w)^{-0.26}(1.5(s/D)+1)^{-0.1} \tag{23}$$

$$f = 30.5\,Re^{-0.56}(y/w)^{-0.54}(1.5(s/D)+1)^{-0.2} \tag{24}$$

$$\eta = 1.9\,Re^{-0.05}(y/w)^{-0.08}(1.5(s/D)+1)^{-0.034} \tag{25}$$

5. Conclusions

The heat transfer and the friction factor characteristics in a test tube equipped with dual twisted tape elements in tandem for several twist ratios (y/w) and space ratios (s/D) have been studied. From the experimental results of the present study, the following conclusions can be drawn.

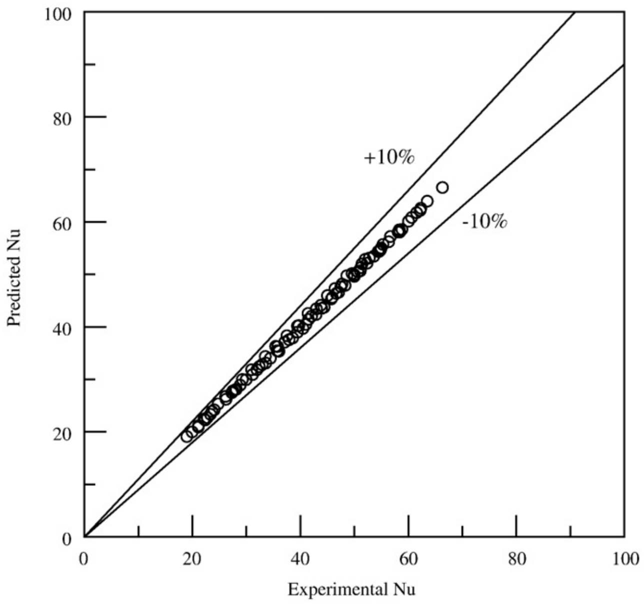

Fig. 11. Prediction of Nusselt number versus experimental result.

- The heat transfer rate for the dual twisted tapes is increased from 12% to 29% in comparison with the single one for $y/w=3.0$ to 5.0 by giving strongly dual swirling flows into the test tube. Depending on the flow conditions and twist ratio (y/w), the increases in heat transfer rate over the plain tube are about 146%, 135% and 128% for $y/w=3.0$, 4.0 and 5.0, respectively.
- The use of the smaller space ratio ($s/D=0.75$) yields the highest heat transfer than the larger space ratio but lower than the full-length tape ($s/D=0.0$). It is also found that at the small space ratio, $s/D=0.75$, there is a slight difference in heat transfer compared with the full-length tape ($s/D=0.0$).
- The Nusselt numbers for the tube with dual twisted tape elements in tandem at $s/D=0.0$, 0.75, 1.5 and 2.25, are about 146%, 140%, 137% and 132% over the plain tube, respectively.
- The friction factor from using the dual twisted tapes is found to increase up to 23% over the single twisted tape. The friction factor tends to decrease with the rise of Reynolds number and twist ratio values.
- The smaller space ratio of the dual twisted tapes in tandem is more attractive in heat transfer application due to higher enhancement efficiency than the single one.

Acknowledgement

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Convective heat transfer in a circular tube with short-length twisted tape insert ☆

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ABSTRACT

This work presents an experimental study on the mean Nusselt number, friction factor and enhancement efficiency characteristics in a round tube with short-length twisted tape insert under uniform wall heat flux boundary conditions. In the experiments, measured data are taken at Reynolds numbers in a turbulent region with air as the test fluid. The full-length twisted tape is inserted into the tested tube at a single twist ratio of $y/w = 4.0$ while the short-length tapes mounted at the entry test section are used at several tape length ratios ($LR = l_s/l_f$) of 0.29, 0.43, 0.57 and 1.0 (full-length tape). The short-length tape is introduced as a swirling flow device for generating a strong swirl flow at the tube entry before decaying along the tube. On the other hand, the full-length tape ($LR = 1.0$) is expected to produce a strongly swirling flow over the whole tube. The variation of heat transfer and pressure loss in the form of Nusselt number (Nu) and friction factor (f) respectively is determined and depicted graphically. The experimental result indicates that the short-length tapes of $LR = 0.29$, 0.43 and 0.57 perform lower heat transfer and friction factor values than the full-length tape around 14%, 9.5% and 6.7%; and 21%, 15.3% and 10.5%, respectively. In addition, it is apparent that the enhancement efficiency of the tube with the short-length tape insert is found to be lower than that with the full-length one. The mean deviation between measured and correlated values of the Nusselt number is in the order of ±7% in the range of Reynolds numbers from 4000 to 20,000.

1. Introduction

Tubes with twisted tape insert have been widely used as the continuous swirl flow devices for augmentation the heat transfer rate in heat exchanger tubes and applied in many engineering applications. Insertion of twisted tape in a tube provides a simple passive technique for enhancing the convective heat transfer by producing swirl into the bulk flow and by disrupting the boundary layer at the tube surface. It has been explained that such tapes induce turbulence and superimposed vortex motion (swirl flow) causing a thinner boundary layer and consequently resulting in higher heat transfer coefficients. However, the increase in friction is seemed to be the penalty of the technique. Thus, tube with twisted tape insert is frequently used in heat exchanger systems because of it low cost, less maintenance and compact.

In the past decades, many researches have investigated the effect of geometry of twisted-tapes on heat transfer and friction in a circular or rectangular tube in both experimental and numerical studies. For the experimental work, Agarwalt and Rao [1] determined the influence of Prandtl number (using Servotherm oil, $Pr = 195$ to 375) on friction factor and heat transfer rate in a tube with twisted tape inserts ($y = 2.41$ to 4.84) under uniform wall temperature (heating and cooling) conditions. Chang et al. [2] investigated the heat transfer and pressure drop characteristics in tube fitted with serrated twisted-tape. In their work, the serrations on two edges of the twisted tape with twist ratios of 1.56, 1.88, 2.81 and ∞ were the square-sectioned ribs with the identical rib pitch and height. Saha et al. [3] conducted the heat transfer and pressure drop characteristics of laminar flow in a tube fitted with regularly spaced twisted-tape elements. Effects of the Reynolds number, Prandtl number, twist ratio, space ratio, and rod-to-tube diameter ratio on heat transfer rate were also reported. Eiamsa-ard et al. [4] studied the heat transfer and friction factor in a tube fitted with regularly spaced twisted tape elements. Effects of the (1) full-length typical twisted tape at different twisted ratios, and (2) twisted tape with various free space ratios ($S = 1.0$, 2.0, and 3.0) were reported. Klaczak [5] investigated the heat transfer and pressure loss characteristics in tubes with a short single-turn turbulator at different length ratios ($S/d = 2.9$ to 4.4) and also compared with those obtained from tube with turbulator mounted all over the tube. Liao and Xin [6] studied the heat transfer and friction characteristics in tubes with three-dimensional internal extended surfaces and twisted-tape inserts with various working fluids such as water, ethylene glycol, and ISO VG46 turbine. Promvonge and Eiamsa-ard [7] investigated the heat

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Nomenclature

A	heat transfer surface area, m^2
C_p	specific heat of fluid, J kg^{-1} K^{-1}
D	inside diameter of test tube, m
f	friction factor $= \Delta P/((L/D)(\rho U^2/2))$
h	heat transfer coefficient, W m^{-2} K^{-1}
I	current, amp
k	thermal conductivity of fluid, W m^{-1} K^{-1}
L	length of test section, m
LR	tape-length ratio $= l_s/l_f$
l_f	full-length twisted tape
l_s	short-length twisted tape
M	mass flow rate, kg s^{-1}
Nu	average Nusselt number $= hD/k$
P	pressure of flow in stationary tube, Pa
ΔP	pressure drop across test tube, Pa
Pr	Prandtl number $= \mu C_p/k$
Q	heat transfer rate, W
Re	Reynolds number $= \rho UD/\mu$
t	thickness of test tube, m
T	temperature, °C
$\tilde{T}$	mean temperature, °C
U	average axial flow velocity, m s^{-1}
V	voltage, V
$\dot{V}$	volume flow rate, m^3 s^{-1}
w	tape width, m
y	pitch length of twisted tape (180° rotation), m
y/w	twist ratio

Greek symbols

ρ	fluid density, kg m^{-3}
δ	tape thickness, m
μ	fluid dynamic viscosity, kg s^{-1} m^{-1}
η	heat transfer enhancement efficiency

Subscripts

a	air
b	bulk
conv	convection
i	inlet
o	outlet
p	plain tube
pp	pumping power
s	swirl generator
VI	heat input
w	wall

transfer, friction factor and enhancement efficiency characteristics in a tube with combined conical-ring turbulator and twisted-tape swirl generator. In their work, the conical-ring was used as turbulator and mounted into the tested tube while the twisted-tape swirl generator placed along the core of the conical-ring. Al-Fahed et al. [8] reported the influence of the microfin tube with twisted-tape inserts on heat transfer and pressure drop behaviors in laminar region under uniform wall temperature conditions with oil as working fluid. Their results were also compared with those obtained from microfin tube alone. Zimparov [9] investigated the heat transfer enhancement of a combination of three-start spirally corrugated tubes with five twisted tape inserts for different relative pitches in the range of Reynolds number 3000 to 60,000. For the numerical work, Date and Gaitonde [10] developed the correlations for predicting the heat transfer coefficient and friction loss characteristics of laminar flow in a tube fitted with regularly spaced twisted-tape elements. Du Plessis and Kroger [11] predicted the friction factor for fully developed laminar twisted-tape flow. Manglik and Bergles [12] conducted on numerical study laminar flow heat transfer and isothermal friction factor in circular-segment ducts with a straight-tape insert at two uniform wall temperature conditions: (1) constant axial and circumferential wall temperature and (2) constant temperature on the curved surface but an adiabatic flat wall.

The present experimental work has been conducted to study the heat transfer rate (Nu) and friction factor (f) characteristics in a round tube fitted with short-length twisted tapes used as a swirling flow generator. Air was employed as the test fluid for the Reynolds number range of 4000 to 20,000. The short-length tape was mounted at the entry test tube with different tape-length ratios ($LR = l_s/l_f = 0.29$, 0.43, 0.57 and 1.0). Moreover, measurements of the tube with full-length twisted tape insert were also conducted for comparison.

2. Experimental strategy

The schematic diagram of the experimental heat exchanger system is depicted in Fig. 1. The experimental system consisted of a 7.5 kW blower, an orifice meter to measure the volume flow rate, inclined or U-tube manometer to measure the pressure drop across the test section, data logger to record the temperature of the inlet and outlet test section and also measured the tube wall temperature of the test tube, and the heat transfer test section. The schematic view of the heat transfer test section and the short-length twisted tape is depicted Fig. 2. The short-length twisted tapes were made of aluminium strips with thickness of 0.8 mm (δ) and width of 46 mm (w). They were fabricated by twisting a straight strip, about its longitudinal axis at constant twist ratio (y/w) of 4.0, while being held under tension. In the test run, the tapes were placed in the tube with different tape arrangements: (1) typical twisted tape or full-length twisted tape ($LR = 1.0$), and (2) short-length twisted tape with several tape-length ratios ($LR = l_s/l_f$) of 0.29, 0.43 and 0.57, where each tape was mounted at the entry test tube. The short-length tape was inserted into the copper tube which has a length of $L = 1250$ mm, with 47.5 mm inner diameter (D), 50.5 mm outer diameter (D_o), and 1.5 mm thickness (t). The tube was heated by continually winding flexible electrical wire to provide a uniform heat flux boundary condition. The outer surface of the test tube was well insulated to minimize convective heat loss to surroundings, and necessary precautions were taken to prevent leakages from the system.

In the apparatus setting above, the inlet bulk air induced by a 7.5 kW blower was directed through the heat transfer test section and passed to an orifice meter. Manometric fluid was used in inclined U-tube manometers with specific gravity (SG) of 0.826 to ensure reasonably accurate measurement of the low pressure drop encountered at low Reynolds numbers. Also, the pressure drop of the test tube was measured with inclined and U-tube manometers. The volumetric air flow rates from the blower were adjusted by varying motor speed through the inverter. During the experiments, the bulk air was heated by an adjustable electrical heater wrapping along the test section. The mean wall temperature was determined by means of calculations based on the reading of Chromel-constantan thermocouples while the bulk air temperatures were measured with the RTD. The calibrated thermocouples were used to measure the temperatures of the fluid at the inlet/outlet of the test section. In the experiments, it was necessary to record the data of temperature, volumetric flow rate and pressure drop of the air at steady state conditions in which the inlet air temperature were maintained at 25 °C. The Reynolds number of the air was varied between 4000 and

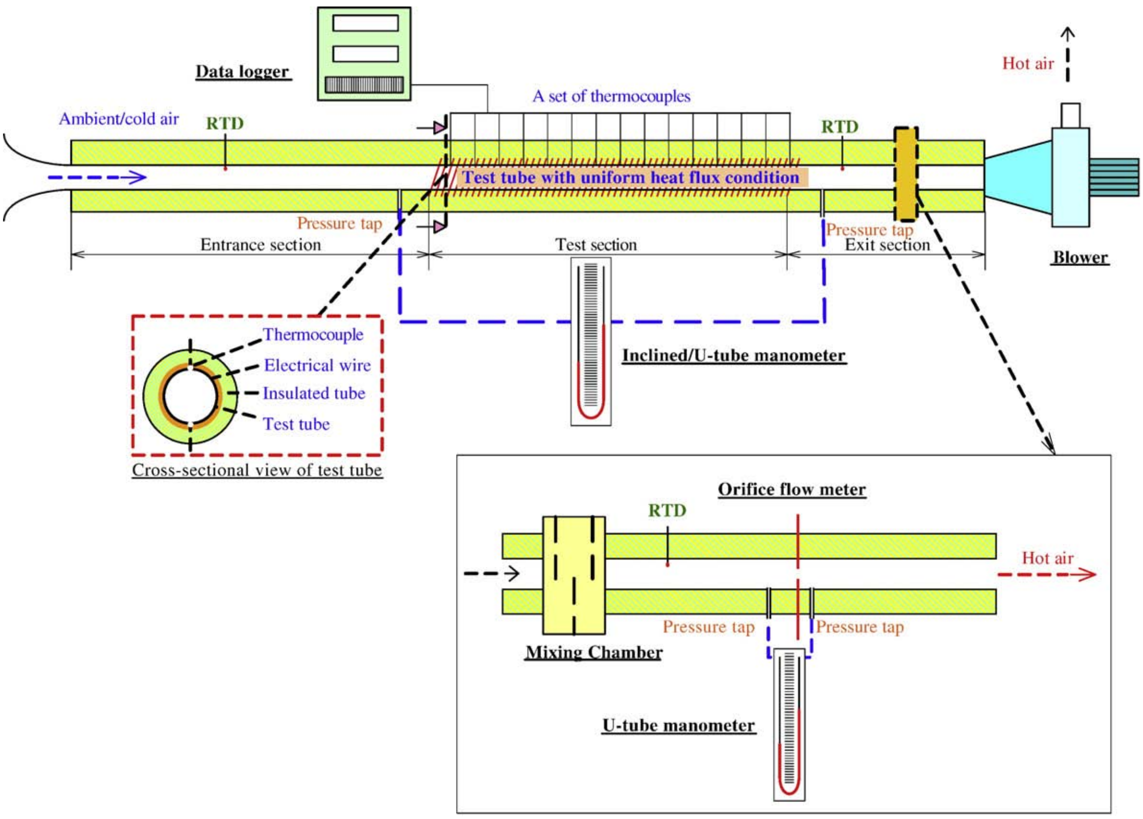

Fig. 1. Experimental set-up.

20,000. The local wall temperature (T_w), inlet and outlet air temperature, the pressure drop across the test section and air flow velocity were measured for heat transfer of the heated tube with twisted tape swirl generators.

3. Data reduction

In order to study the heat transfer rate and friction factor characteristics of a uniform heat flux tube fitted with short-length

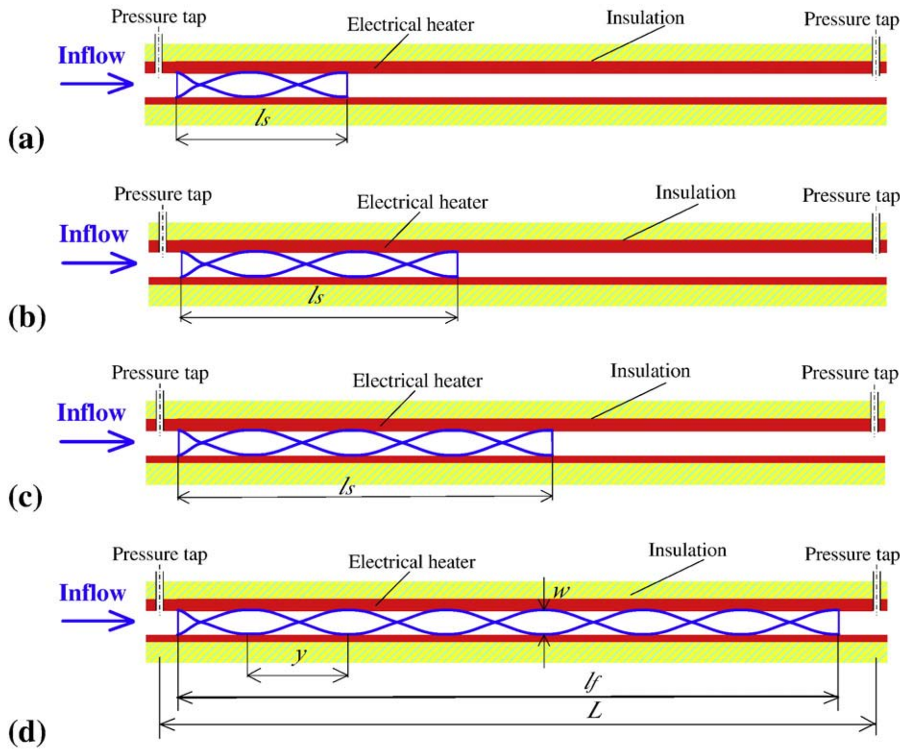

Fig. 2. Test tube with short-length twisted tape inserts at different tape-length ratios: (a) $LR = 0.29$, (b) $LR = 0.43$, (c) $LR = 0.57$ and (d) $LR = 1.0$ or full-length tape.

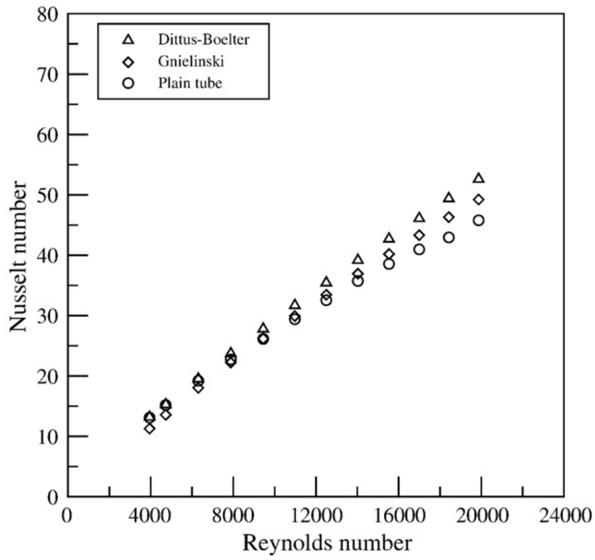

Fig. 3. Verification of Nusselt number of plain tube.

twisted tape, the heat transfer coefficient and pressure drop must be determined. The heat balance between the air (Q_a) and heat input (Q_{VI}) was found to be within 3.0% for all runs. That is

$$\left|\frac{Q_{VI}-Q_a}{Q_{VI}}\right| < 3.0\% \tag{1}$$

where

$$Q_a = MC_{p,a}(T_o - T_i).$$

The steady state of the heat transfer rate is assumed to be equal to the heat loss from the test section which can be expressed as

$$Q_a = Q_{conv}. \tag{2}$$

The convection heat transfer from the test section can be written by

$$Q_{conv} = hA\left(\tilde{T}_w - T_b\right) \tag{3}$$

where,

$$T_b = (T_o + T_i)/2. \tag{4}$$

The inlet temperature (T_i) and outer temperature (T_o) of the bulk air (T_b) were measured at certain points with a data logger in conjunction with the RTDs. The mean wall temperature can be expressed as

$$\tilde{T}_w = \sum T_w / 15 \tag{5}$$

where T_w is the local wall temperature and evaluated at the outer wall surface of the inner tube. Fifteen thermocouples were tapped on the local wall of the tube and the thermocouples were placed round the tube to measure the circumferential temperature variation, which is found to be negligible. The average wall temperatures are calculated from 15 points, lined between the inlet and the exit of the test tube. The average heat transfer coefficient, h and the mean Nusselt number, Nu are estimated as follows:

$$h = Q_a / A\left(\tilde{T}_w - T_b\right) \tag{6}$$

$$Nu = hD/k. \tag{7}$$

The Reynolds number is given by

$$Re = UD/\nu. \tag{8}$$

The pressure drop (ΔP) is determined from the differences in the level of fluid in U-tube manometer while the friction factor (f) is calculated using equation below:

$$f = \frac{\Delta P}{\left(\frac{L}{D}\right)\left(\rho\frac{U^2}{2}\right)} \tag{9}$$

in which U is mean air velocity of the tube. All of thermo-physical properties of the air are determined at the overall bulk air temperature from Eq. (4).

4. Results

In this study, the experimental results of the tube fitted with short-length twisted tapes that effect on the heat transfer rate, friction factor and enhancement efficiency behaviors are investigated experimentally as shown in Figs. 3–9. The tubes fitted with short-length twisted tape of several tape-length ratios ($LR = l_s/l_f = 0.29$, 0.43 and 0.57) are also presented. In the experiment, the short-length twisted tape was mounted at the inlet of the test section to be used as a swirl flow generator to augment heat transfer rate in the tube while the full-length tape ($LR = 1.0$) was also inserted for comparison. The experimental result of the tube without twisted tape insert is first validated with the earlier published correlations for the sake of error assessment.

The results of the heat transfer and friction factor from the plain tube are confirmed with the previous correlations [13], for example, Dittus–Boelter equation or Gnielinski equation for heat transfer and Petukhov equation or Blasius equation for friction factor as depicted in the Figs. 3 and 4, respectively. It is found that the plain tube result of the present work is in good agreement with that of the previous work within $\pm 5.7\%$ deviation for the heat transfer (Nu) and within $\pm 10.7\%$ for the friction factor (f). The present work results of the plain tube are

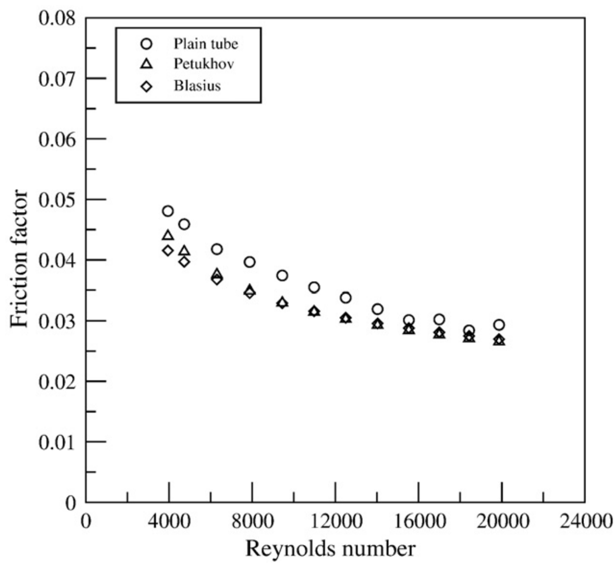

Fig. 4. Verification of friction factor of plain tube.

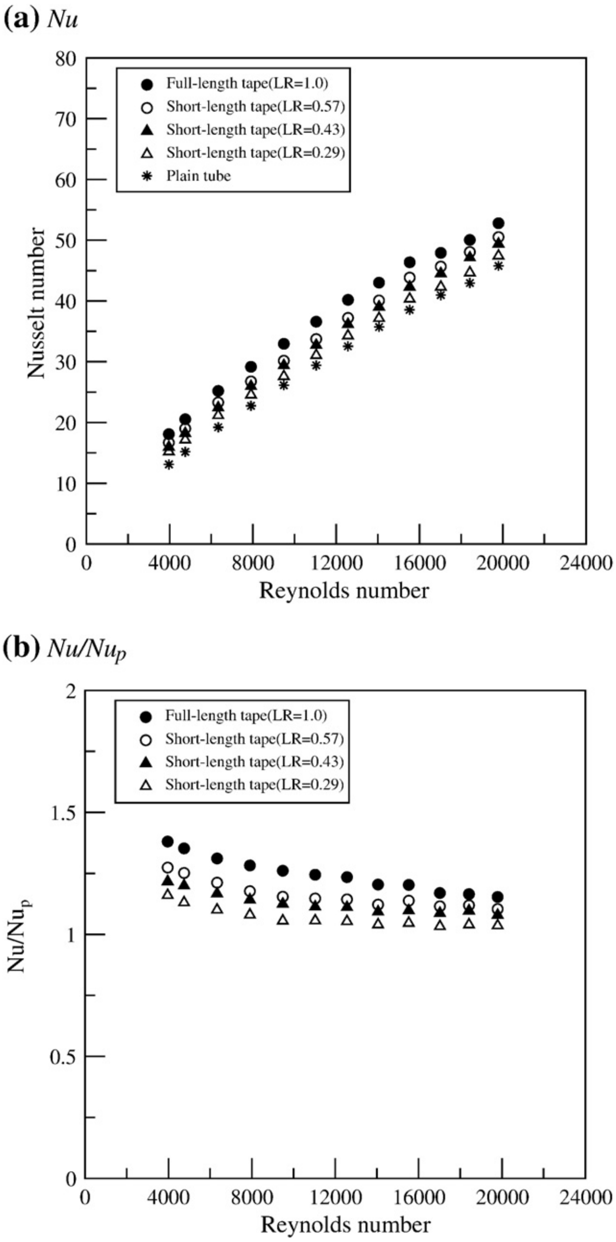

Fig. 5. Nusselt number versus Reynolds number: (a) Nu and (b) Nu/Nu_p.

correlated as expressed in Eqs. (10) and (11) for Nusselt number and friction factor respectively.

$$Nu = 0.025Re^{0.774}Pr^{0.4} \tag{10}$$

$$f = 0.758Re^{-0.331}. \tag{11}$$

The experimental heat transfer of the tube fitted with short-length twisted tape measured under uniform heat flux condition is presented in Fig. 5(a). In the figure, the effects of the short-length twisted tape at different tape-length ratios (LR = 0.29, 0.43 and 0.57) on the mean heat transfer (Nu) are studied. The full-length twisted tape (LR = 1.0) is used as a continuous swirling flow over the test tube length for: (1) increasing the residence time of the flow, (2) reduction or destruction of the thermal boundary layer and (3) increasing the mixing between the core and the tube wall flows. It is found that the Nusselt number increases with the rise of Reynolds number for all twisted tapes. The short twisted tape is expected to generate a strongly swirl flow at the tube entry before it is decaying along the flow leading to the decrease in heat transfer rate at the downstream flow. The short-length twisted tapes with LR = 0.29, 0.43 and 0.57, provide the reduction of heat transfer rate around 14%, 9.5% and 6.7% below the full-length tape, respectively. It is seen that the use of the smallest tape-length ratio yields the lowest heat transfer rate. This can be attributed to the fact that the short tape generates shorter swirl intensity of the downstream flow resulting in the decrease of heat transfer coefficient (h). However, the short-length twisted tape still yields higher heat transfer than the plain tube around 6.9%, 12.7%, and 16.3% for LR = 0.29, 0.43 and 0.57, respectively. Fig. 5(b) shows the variation of the Nusselt number ratio (Nu/Nu_p) with Reynolds number values. The Nusselt number ratio is the ratio of the mean Nusselt number (Nu) obtained from the tube with twisted tape insert to the Nu from plain tube. In the figure, it is apparent that the Nusselt number ratio tends to decrease as Reynolds number increases. This is because the influence of twisted tape insert on the heat transfer enhancement is less significant for increasing Reynolds number.

The friction factors of the plain tube with/without twisted tape insert for different tape-length ratios (LR) are depicted in Fig. 6(a). It is observed that the tube with twisted tape inserts shows a substantial increase in the friction factor (f). The friction factor tends to decrease with increasing

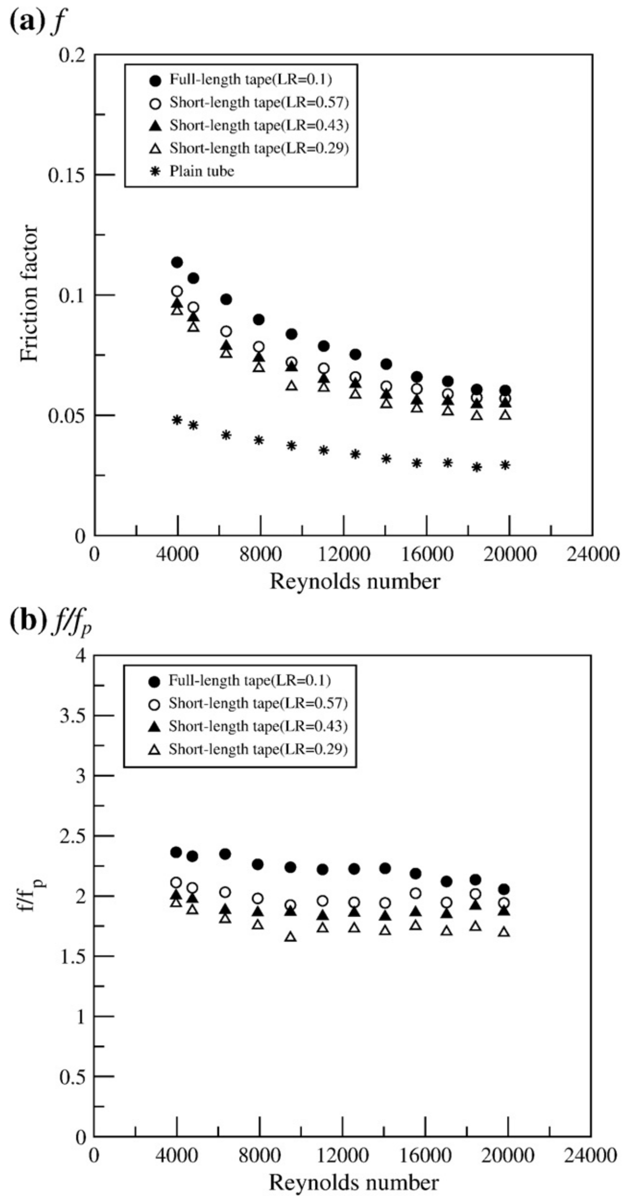

Fig. 6. Friction factor versus Reynolds number: (a) f and (b) f/f_p.

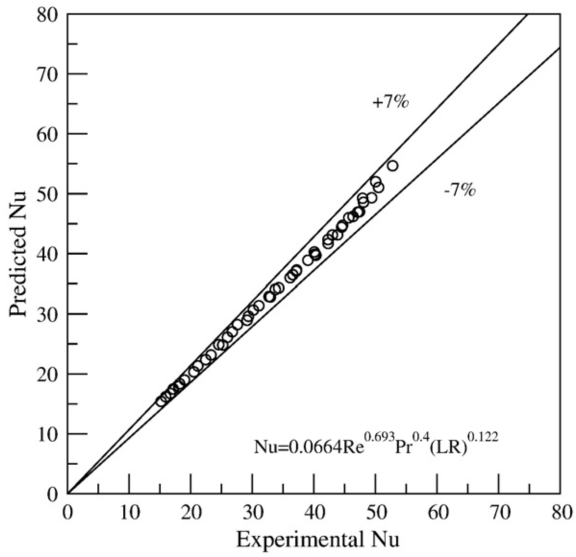

Fig. 7. Prediction of Nusselt number versus experimental result.

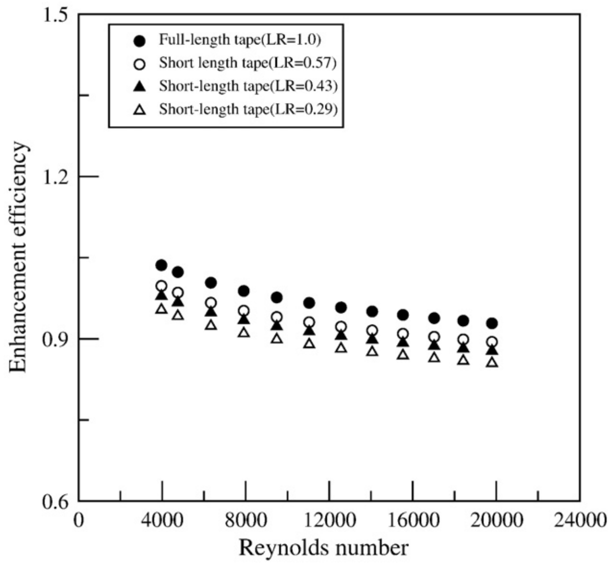

Fig. 9. Heat transfer enhancement efficiency versus Reynolds number.

Reynolds number. The friction factor of the short-length twisted tape with $LR=0.29$, 0.43 and 0.57, respectively, is found to decrease around 21%, 15.3%, and 10.5% in comparison with the full-length tape ($LR=1.0$). This can be explained that the reduction of tape-length ratios (LR) would give rise to lower swirl strength at the downstream flow leading to lower pressure loss. However, the short-length twisted tape insert still provides higher mean friction factor than the plain tube at 1.76, 1.88, and 1.99 times for $LR=0.29$, 0.43 and 0.57, respectively. Fig. 6(b) depicts the variation of the friction factor ratio (f/f_p) with the Reynolds number for different tape inserts. It is seen that the friction factor ratio tends to slightly decrease for the rise of Reynolds number. However, the reduction of friction factor becomes relatively small as heated air mass flow rate increase. Subsequently, the present results of Nusselt number and friction factor values are correlated by Eqs. (12) and (13), respectively.

$$Nu = 0.0664\,Re^{0.693}Pr^{0.4}LR^{0.122} \quad (12)$$

$$f = 2.8\,Re^{-0.386}LR^{0.19}. \quad (13)$$

Figs. 7 and 8 show the relationship between the measured data and the prediction by Eqs. (12) and (13) for Nusselt number and friction

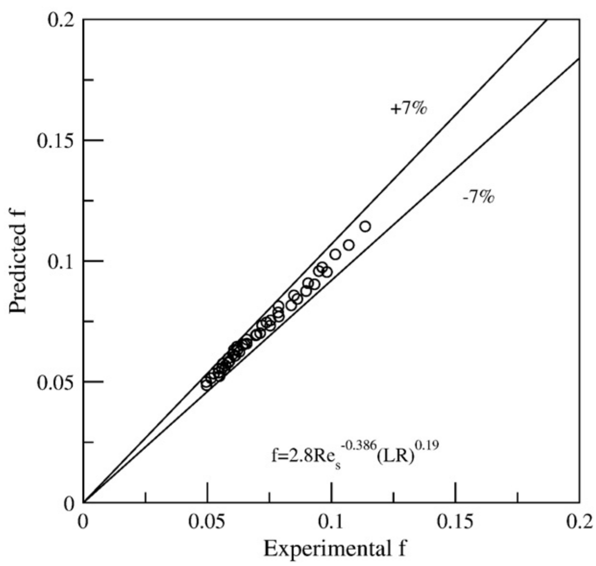

Fig. 8. Prediction of friction factor versus experimental result.

factor values. It is found that the predicted values for both Nusselt number and friction factor are in good agreement with experimental data within $\pm 7\%$ deviation.

The most importance parameter for the heat exchanger design is the heat transfer enhancement efficiency (η). In the present work, the enhancement efficiency (η) considered under constant pumping power between the plain tube and the inserted tube can be made as follows:

$$(\dot{V}\Delta P)_p = (\dot{V}\Delta P)_s \quad (14)$$

where the relationship between friction factor and Reynolds number can be drawn as below

$$\left(fRe^3\right)_p = \left(fRe^3\right)_s. \quad (15)$$

The enhancement efficiency (η) defined as the ratio of the convective heat transfer coefficient of the inserted tube to that of the plain tube at a constant pumping power is

$$\eta = \frac{h_s}{h_p}\bigg|_{pp}, \quad (16)$$

Using Eqs. (11), (13) and (15), the Reynolds number for the plain tube (Re_p) is written as the function of Reynolds number for the tube with twisted tape insert (Re_s)

$$Re_p = 1.632\,Re_s^{0.979}LR^{0.071}. \quad (17)$$

Employing Eqs. (10), (12), (16) and (17), the enhancement efficiency for the twisted tape at different tape-length ratios can be written as

$$\eta = \frac{h_s}{h_p}\bigg|_{pp} = 1.82\,Re_s^{-0.068}LR^{0.067}. \quad (18)$$

The heat transfer enhancement efficiencies (η) for the inserted tube at various tape-length ratios ($LR=0.29$, 0.43, 0.57 and 1.0 (full-length tape)) calculated from Eqs. (17) and (18) are presented in Fig. 9. In the figure, it is worth noting that the enhancement efficiency decreases with the rise of Reynolds number and the reduction of tape-length ratios (LR). It is also seen that for all study cases the enhancement efficiency value is high at low Reynolds number. The maximum enhancement efficiency for using $LR = 0.29$, 0.43, 0.57 and 1.0 is found to be 0.95, 0.98, 1.00, and 1.04, respectively.

5. Conclusions

Influences of the tube with short-length twisted tape inserts at several tape-length ratios ($LR = 0.29$, 0.43, 0.57 and 1.0 (full-length tape)) on the heat transfer (Nu), friction factor (f) and enhancement efficiency (η) characteristics have been investigated experimentally. The following conclusions can be drawn in the present study:

(i) The presence of the tube with short-length twisted tape insert yields higher heat transfer rate (Nu) up to 1.16, 1.22 and 1.27 times of the plain tube, while friction factor up to 1.76, 1.88 and 1.99 times for using $LR = 0.29$, 0.43 and 0.57, respectively. The maximum heat transfer (Nu) and friction factor (f) is obtained for using the full-length tape. The Nu and f values of the short-length twisted tape insert with $LR = 0.29$, 0.43 and 0.57, respectively, are about 14%, 9.5%, and 6.7%; and 21%, 15.3%, and 10.5% lower than that of the full-length tape.

(ii) The heat transfer enhancement efficiency (η) is higher than unity only for the full-length tape insert at low Reynolds number. The maximum enhancement efficiencies of the short-length twisted tapes are around 0.95, 0.98 and 1.00 for $LR = 0.29$, 0.43 and 0.57, respectively.

(iii) The results of the heat transfer rate and pressure drop are formulated by the statistical correlations for heat transfer (Nu) and friction factor (f) against tape-length ratios (LR) and Reynolds number (Re). The correlations are validated with experimental data with $\pm 7\%$ deviation for both the heat transfer (Nu) and friction factor (f) values.

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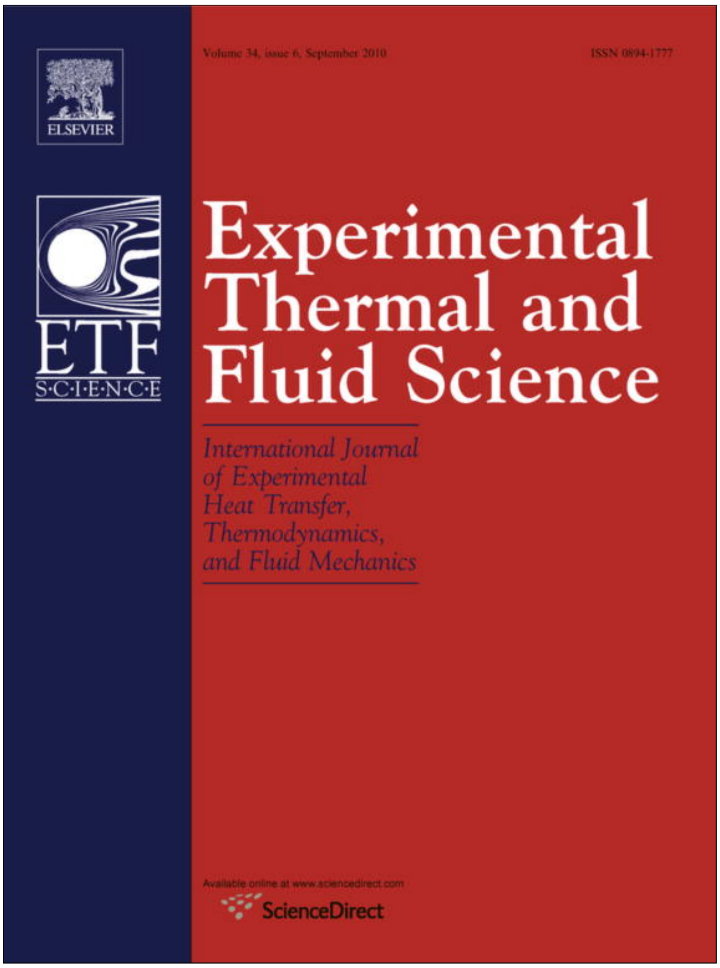
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Influences of peripherally-cut twisted tape insert on heat transfer and thermal performance characteristics in laminar and turbulent tube flows

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ABSTRACT

Effects of peripherally-cut twisted tape insert on heat transfer, friction loss and thermal performance factor characteristics in a round tube were investigated. Nine different peripherally-cut twisted tapes with constant twist ratio ($y/W = 3.0$) and different three tape depth ratios (DR = d/W = 0.11, 0.22 and 0.33), each with three different tape width ratios (WR = w/W = 0.11, 0.22 and 0.33) were tested. Besides, one typical twisted tape was also tested for comparison. The measurement of heat transfer rate was conducted under uniform heat flux condition while that of friction factor was performed under isothermal condition. Tests were performed with Reynolds number in a range from 1000 to 20,000, using water as a working fluid. The experimental results revealed that both heat transfer rate and friction factor in the tube equipped with the peripherally-cut twisted tapes were significantly higher than those in the tube fitted with the typical twisted tape and plain tube, especially in the laminar flow regime. The higher turbulence intensity of fluid in the vicinity of the tube wall generated by the peripherally-cut twisted tape compared to that induced by the typical twisted tape is referred as the main reason for achieved results. The obtained results also demonstrated that as the depth ratio increased and width ratio decreased, the heat transfer enhancement increased. Over the range investigated, the peripherally-cut twisted tape enhanced heat transfer rates in term of Nusselt numbers up to 2.6 times (turbulent regime) and 12.8 times (laminar regime) of that in the plain tube. These corresponded to the maximum performance factors of 1.29 (turbulent regime) and 4.88 (laminar regime).

1. Introduction

Insertion of twisted tape is one of the most popular passive heat transfer enhancement technique due to its low cost, ease of installation and low maintenance. Commonly, the twisted tape functions as the continuous swirl flow generator. The swirl flow induces the turbulence near the tube wall and increases the residence time of the fluid flow in the tube. The higher turbulence intensity of the fluid close to the tube wall associated with the twisted tape is responsible for an excellent fluid mixing and an efficient redevelopment of the thermal/hydrodynamic boundary layer which consequently results in the improvement of convective heat transfer [1–24]. In fact, swirl induces both desirable heat transfer enhancement and undesirable increase of shear stress and pressure drag in a tube. The latter escorts with the reduction of thermal performance factor which restricts the industrial applications of the twisted tape. Many previous research works signify that the heat transfer enhancing performances of twisted tapes strongly depend on their geometries. The proper design of twisted tape provides an increase of heat transfer rate with a reasonable pressure drop, resulting in effecting energy saving.

The use of twisted tape for heat transfer enhancement was early reported by Whitham [3]. Then, a vast number of research works in this field was reviewed by Dewan et al. [4]. Over the recent years, the research works on enhancing heat transfer by means of twisted tapes have been continuously released, emphasizing on the modification of twisted tape geometries to achieve the heat transfer enhancement with friction loss in a practical level.

Some attempts were made by formulating multiple twisted tapes [5,6]. Chang et al. [5] compared effects of the single, twin, and triple twisted tapes on heat transfer enhancement in a square duct. Due to periodically bursting swirls by the twin and triple twisted tapes, heat transfer was enhanced with the extended effective Reynolds number range. Among the twisted tape used, the triple twisted tape provided the highest performance factors for turbulent flows. Eiamsa-ard et al. [6] further modified twin twisted tapes in forms of twin counter twisted tapes (CTs) and twin co-twisted tapes (CoTs). The obtained results showed that the CTs offered better thermal performance factor which was as high as 1.39.

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Nomenclature

A	heat transfer surface area, m^2
C_p	specific heat of fluid, J kg^{-1} K^{-1}
D	inside diameter of the test tube, m
d	peripherally-tape depth, m
f	friction factor = $\Delta P/((L/D)(\rho U^2/2))$
h	heat transfer coefficient, W m^{-2} K^{-1}
I	current, A
k	thermal conductivity of fluid, W m^{-1} K^{-1}
L	length of the test section, m
M	mass flow rate, kg s^{-1}
Nu	Nusselt number = hD/k
P	pressure of flow in stationary tube, Pa
ΔP	pressure drop, Pa
Pr	Prandtl number = $\mu C_p/k$
Q	heat transfer rate, W
Re	Reynolds number = $\rho UD/\mu$
t	thickness of the test tube, m
T	temperature, K
$\tilde{T}$	mean temperature, K
U	mean axial flow velocity, m s^{-1}
V	voltage, V
w	peripherally-tape width, m
W	twisted tape width, m
y	twisted tape pitch, m

Greek symbols

ρ	fluid density, kg m^{-3}
δ	twisted tape thickness, m
μ	fluid dynamic viscosity, kg s^{-1} m^{-1}
η	thermal performance factor

Subscripts

b	bulk
c	convection
i	inlet
o	outlet
p	plain
s	surface
t	turbulator
w	water

Abbreviations

PT	peripherally-cut twisted tape
TT	typical twisted tape

Another approach of modification is to reduce friction factor of twisted tape with respect to that of the typical one. Eiamsa-ard et al. [7] modified the short length twisted tape for heat transfer improvement in a circular tube, however the results showed that the short-length twisted-tape gave lower heat transfer, friction factor and performance factor values than the full-length one, at similar operating conditions. Saha et al. [8] introduced regularly spaced twisted-tape elements connected by thin circular rods for heat enhancement in a circular tube fitted and found that at the constant pumping power, the regularly spaced twisted-tape performed significantly better than full-length twisted tapes at high Reynolds numbers, high twists, and small spaces. Eiamsa-ard et al. [9] numerically investigated effects of loose-fit twisted tape on heat transfer, friction loss and thermal performance factor compared with that of the tight-fit (typical) one. The results indicated that heat transfer rates associated by the loose-fit tape were considerably poorer than those obtained from the typical one, resulting in unattractive thermal performance factors. Similarly, the experimental and Computational Fluid Dynamics (CFD) results by Rahimi et al. [10] indicated that the twisted tape with spaces on the tapes (perforated and notched twisted tapes) provided lower heat transfer rate, friction factor as well as thermal performance factor compared to the typical one did. Recently, Jaisankar et al. [11,12] reported effect of twisted tape with full-length twist, twisted tapes with rod and spacer at the trailing edges on the heat transfer and friction factor characteristics in thermosyphon solar water heater. Their findings were that the twisted tape with full-length twist gave the highest heat transfer rate. However, the twisted tape with rod showed advantage over the twisted tape with full-length twist the by its less friction factor with a slight decrease of heat transfer rate.

On the other hand, many newly invented tapes have been modified from the typical twisted tape by focusing on improving fluid mixing and thus heat transfer rate rather than reducing friction loss. Chang et al. [13] invented serrated twisted tape by introducing the repeated ribs on both sides of the tape. Their results indicated that the modified twisted tape caused a large pressure drop penalty up to 80 times of that by the typical tape, resulted in a lower thermal performance factor in compared with that of proposed by the typical tape. Rahimi et al. [10] comparatively investigated the effects of the typical twisted tape and the modified twisted tape, namely jagged twisted tape and found that the jagged twisted tape offered higher heat transfer rate and thermal performance factor than the typical one, under the similar conditions. It was pointed out that the superior heat transfer enhancement by the jagged twisted tape over the typical one was due to the higher turbulence intensity of the fluid close to the tube wall. Eiamsa-ard et al. [14] presented the effect of the delta-winglet twisted tape on the heat transfer enhancement in a heat exchanger tube under uniform heat flux conditions. The modified tape generated the additional flow disturbance in the tape edge region, leading to superior heat transfer and thermal performance factor in comparison with the unmodified one.

Chang et al. [15,16] also fabricated broken or spiky twisted tape for the use in single-phase and two-phase flows. They convinced that the modified twisted tape produced shear layers downstream the spikes which interacted with the swirling mainstream in the tube fitted, resulting in the amplification of the turbulent intensity as well as the vorticity. The comparison demonstrated that the spiky tape provided a higher thermal performance factor than those given by the serrated twisted tape and the typical tape from Manglik and Bergles correlation. It was also discovered that performance of the spiky tape was even better for two-phase flow system in comparison to the single phase flow system. Chang and Yang [17] also described effects of spiky twisted tape on heat transfer enhancement for the cocurrent air–water slug flows in the vertical tube. Recently, Jaisankar et al. [18,19] invented the helical twisted tapes for the use in thermosyphon solar water heating system. More information about heat transfer enhancement by means of twisted tapes can be viewed in the other recent reports [20–24].

According to the above literature, it can be observed that the modified twisted tapes with small gaps on the tape, for example jagged tape [10], delta-winglet tape [14], and broken (or spiky) tape [15,16] show great promise for enhancing both heat transfer rate and thermal performance factor. The reason behind the high thermal performance factor is that those small gaps bring pressure drop in the system to the reasonable level.

The present work proposes the comprehensive investigation on heat transfer enhancement by means of peripherally-cut twisted tape. The modified tape comprises the small gaps in the peripheral region of the tape. This style of tape is believed to offer similar mechanism found for the promising twisted tapes (delta-winglet, spiky and jagged tapes) mentioned in the literature. This study focuses on the detailed examination of the heat transfer, friction factor and the thermal enhancement behaviors in the heat exchanger tubes fitted with the peripherally-cut twisted tape at different tape depth ratios (DR = d/W = 0.11, 0.22 and 0.33) and tape width ratios (WR = w/W = 0.11, 0.22 and 0.33). The measurement of heat transfer rate was conducted under uniform heat flux condition while that of friction factor was performed under isothermal condition with Reynolds number between 1000 and 20,000. The experiments results obtained for the tubes fitted with the peripherally-cut twisted tape were also compared with those for the tube fitted with the typical twisted tape and the plain tube.

2. Peripherally-cut twisted tape (PT)

Twisted tapes were made from the aluminum straight tape with tape length (L) of 1000 mm, tape thickness (δ) of 0.8 mm, and tape width (W) of 19 mm for the typical twisted tape (TT) and 18 mm for peripherally-cut twisted tape (PT). To obtain the PT, the TT was modified by peripheral cutting in order to generate additional turbulence in the vicinity of the tube wall. Geometries of the peripherally-cut twisted tape (PT) and the typical twisted tape (TT) are demonstrated in Fig. 1. Nine different PTs, included three tape depth ratios (DR = d/W = 0.11, 0.22 and 0.33) each with three different tape width ratios (WR = w/W = 0.11, 0.22 and 0.33) were used in the present work. Regarding to Fig. 1, the twist ratio "y/W" is defined as the ratio of the twist length (y, 180°/twist length) to the tape width (W), this ratio was kept constant at y/W = 3.0. Details of all twisted tape inserts and operating conditions are summarized in Table 1.

3. Details of experimental apparatus

In the experiments, the PT and TT were inserted into the tube as shown in Fig. 2. The tube was wound by electrical wire to maintain

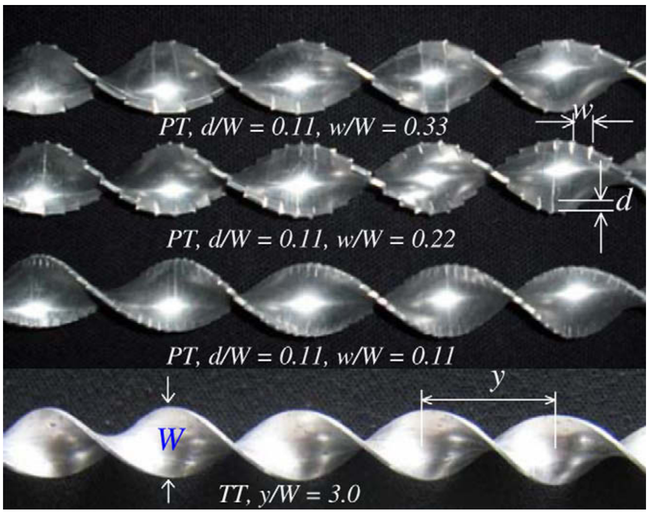

Fig. 1. Geometries of peripherally-cut twisted tapes (PTs) and typical twisted tape (TT).

Table 1
Details of the experimental set-up.

	TT	PT
Tape width (W)	19 mm	18 mm
Twist length (y)	57 mm	54 mm
Twist ratio (y/W)	3.0	3.0
Peripherally-cut width (w)	–	2, 4 and 6 mm
Peripherally-cut width ratio (w/W)	–	0.11, 0.22 and 0.33
Peripherally-cut depth (d)	–	2, 4 and 6 mm
Peripherally-cut depth ratio (d/W)	–	0.11, 0.22 and 0.33
Test tube diameter (D)	19 mm	19 mm
Test tube length (L)	1000 mm	1000 mm
Test tube thickness (t)	1.5 mm	1.5 mm
Reynolds number (Re)	1000–20,000	1000–20,000
Working fluid	Water	Water
Inlet temperature of water (T_i)	27 °C	27 °C

the tube at the constant heat flux condition. The tube was then well insulated to prevent heat loss. The schematic diagram of the experimental rig is presented in Fig. 3. The system consists of (1) a heating tube with insulator, (2) thermocouples for measuring the fluid and the tube wall temperatures, (3) a data logger connected with a PC, (4) a variac transformer for controlling the electrical power output to the test section, (5) an amp/volt meter, (6) a centrifugal water pump, (7) a water chiller, (8) two water tanks which positioned at the bottom at the top of the heating section, (9) a rotameter for measuring the volumetric water flow rate, (10) two pressure taps for measuring the pressure drop across the test section, (11) a mixing chamber located at the end of the heating section, (12) a calm section with the length of 1500 mm (or 83D) which connected to the entrance of the heating section for ensuring the fully developed flow of the entering fluid. More details of the experimental setup and procedure were described elsewhere [14]. In the experiments, the Reynolds numbers of the water was varied in the range between 1000 and 20,000. The Prandtl number values were calculated based on the bulk flow temperature ($T_b = (T_i + T_o)/2$). The inlet temperature (T_i) was maintained at around 27 °C while outlet temperature (T_o) was risen up to 37 °C, depending on the operating conditions. During the test, fifteen local temperatures of the tube wall were measured using type-K thermocouples while the inlet and outlet fluid temperatures were measured with the RTDs. All of the temperature data were recorded with the data logger. The pressure drop across the test length was measured with manometers using water as a working fluid. One tap of manometer was placed at 2.5D upstream the entrance of the test tube while another tap was located at the 6D downstream the exit of the test tube. To ensure the steady state for each condition, the period of around 25–50 min depending on the Reynolds number and tape shape was taken prior to the data record.

Experimental uncertainties of Nusselt number, Reynolds number and friction factor were calculated using ANSI/ASME standard [25]. The maximum uncertainties of Nusselt number, Reynolds number and friction factor were found to be ±8%, ±5%, and ±10%, respectively.

4. Data reduction

The heat transfer and pressure loss results in the tube with or without twisted tape insert were taken according to the experimental procedure described in previous section. During the test, water in the test section receives heat (Q_w) from the electrical heat wire mainly via the convective heat transfer mechanism. Thereby, the Q_w is assumed to be equal to the convective heat transfer within the test section which can be written as:

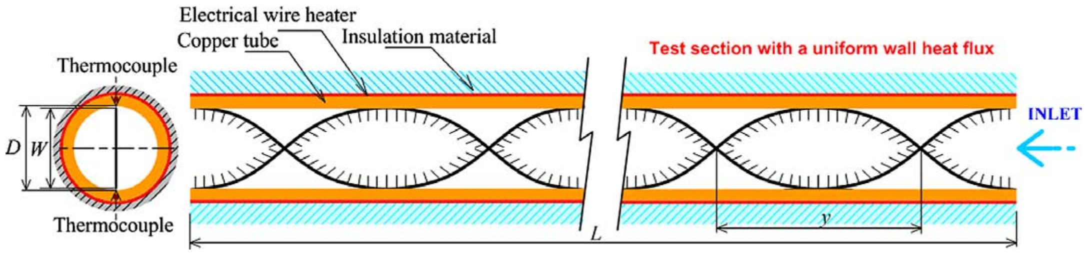

Fig. 2. Test tube with peripherally-cut twisted tape insert.

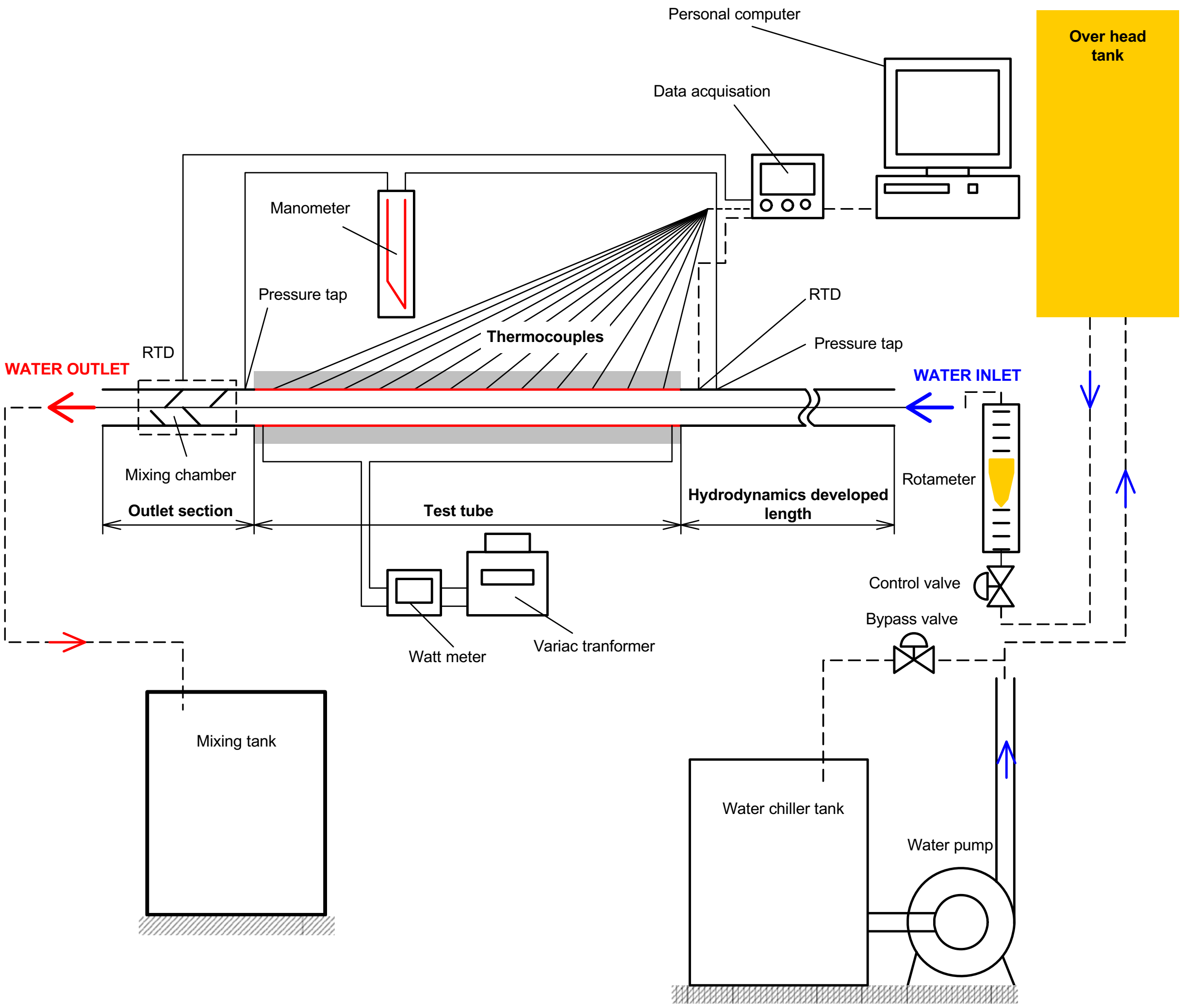

Fig. 3. Schematic diagram of experimental apparatus.

$$Q_w = Q_c \tag{1}$$

The heat gained by the water in term of enthalpy change can be expressed as:

$$Q_w = MC_{p,w}(T_o - T_i) \tag{2}$$

In the experiments, the heat equilibrium test showed that the heat supplied by electrical heating (Q_{VI} = IV) under uniform heat flux condition (UHF) is between 3% and 5% higher than that the heat received by the water (Q_w), this is due to the heat leak from the tube wall.

$$((Q_{VI} - Q_w) \times 100\%)/Q_{VI} \leqslant 5\% \tag{3}$$

The average value of heat absorbed by the fluid are taken for internal convective heat transfer coefficient calculation by the following equation:

$$Q_c = hA(\tilde{T}_s - T_b) \tag{4}$$

where A is the internal surface of the tube wall (πDL) and T_b is the mean bulk flow temperature ($T_b = (T_o + T_i)/2$).

The mean inner wall surface temperature ($\tilde{T}_s$) of the test tube is calculated from 15 stations of surface temperatures lined between the inlet and the exit of the test tube, using the following equation:

$$\tilde{T}_s = \sum T_s/15 \tag{5}$$

where T_s is the local wall temperature, evaluated at the inner wall surface of the test tube.

The mean heat transfer coefficient can be determined using equation shown below:

$$Q_w = Q_c = MC_{p,w}(T_o - T_i) = hA(\widetilde{T}_s - T_b)$$

The mean convective heat transfer coefficient (h) and the mean Nusselt number (Nu) are then estimated as follows:

$$h = (MC_{p,w}(T_o - T_i))/A(\widetilde{T}_s - T_b) \tag{6}$$

$$\mathrm{Nu} = hD/k \tag{7}$$

In the present work, the friction factor in term of pressure drop (Δp) across the test length (L) determined from a difference in the level of a manometer liquid (water) is acquired under an isothermal flow condition and can be expressed as:

$$f = \Delta P/((L/D)(\rho U^2/2)) \tag{8}$$

The thermal performance criteria (η) is defined as the ratio of the Nusselt number ratio to the friction factor ratio at the same pumping power [5,6,14,15,26,27]:

$$\eta = (\mathrm{Nu}_t/\mathrm{Nu}_p)/(f_t/f_p)^{1/3} \tag{9}$$

where Nu_t is the Nusselt number in the tube with PT insert, Nu_p is the Nusselt number in the plain tube, f_t is the friction factor in the tube with PT insert, and f_p is the friction factor in the plain tube. The flow regime can be defined from the Reynolds number.

$$\mathrm{Re} = \rho UD/\mu \tag{10}$$

All the fluid thermo-physical properties of the water are determined based on the mean bulk fluid temperature (T_b).

5. Results and discussion

The experimental results obtained in the present investigation (heat transfer rate, friction factor and thermal performance) are presented and discussed as follows.

5.1. Validation test of plain tube with/without twisted tape

Prior to the assessment of the peripherally-cut twisted tape performance, the tests using the plain tube with and without typical twisted tape (TT) were conducted over the targeting Reynolds numbers in turbulent regime, in order to verify the facility reliability. The experimental results included the mean heat transfer rate (Nu) and friction factor (f) for the plain tube with and without typical twisted tape (TT) insert are compared with those from the past correlations stated as follows:

Nusselt number for the plain tube given by Gnielinski [28]:

$$\mathrm{Nu} = \frac{\left(\frac{f}{8}\right)(\mathrm{Re} - 1000)\mathrm{Pr}}{1 + \left[12.7\left(\frac{f}{8}\right)^{0.5}\left(\mathrm{Pr}^{2/3} - 1\right)\right]} \tag{11}$$

Nusselt number for the plain tube fitted with TT by Manglik and Bergles for turbulent flow [1]:

$$\mathrm{Nu} = (1 + 0.769/(y/W))\left[0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}\left(\frac{\pi}{\pi - 4\delta/D}\right)^{0.8}\left(\frac{\pi + 2 - 2\delta/D}{\pi - 4\delta/D}\right)^{0.2}\right]\phi \tag{12}$$

where $\phi = (\mu_b/\mu_s)^{0.18} = (T_b/T_s)^{0.45}$

Friction factor for the plain tube by Blasius [28]:

$$f = 0.316\mathrm{Re}^{-0.25} \tag{13}$$

Friction factor for the plain tube fitted with TT by Manglik and Bergles for turbulent flow [1]:

$$f = \left(1 + 2.06\left(1 + (2(y/W)/\pi)^2\right)^{-0.74}\right) \times \left[0.079\mathrm{Re}^{-0.25}\left(\frac{\pi}{\pi - 4\delta/D}\right)^{1.75}\left(\frac{\pi + 2 - 2\delta/D}{\pi - 4\delta/D}\right)^{1.25}\right] \tag{14}$$

As displayed in Figs. 4 and 5a and b, the data obtained from the present work are found to be in excellent agreement with those from previous correlations for both Nusselt number and friction factor for turbulent flow. The deviations of the present data from the above equations fall within ±7.6% for Nusselt number and ±10.4% for friction factor in case of the plain tube, and within ±5.4% for Nusselt number and ±9.6% for friction factor in case of the tube fitted with typical twisted tape. In addition, the correlations for prediction of the Nusselt number and friction factor in the plain tube for turbulent flow are proposed as shown in Eqs. (15) and (16). These correlations are subjected for evaluating the heat transfer enhancement associated by the peripherally-cut twisted tape in Section 5.4.

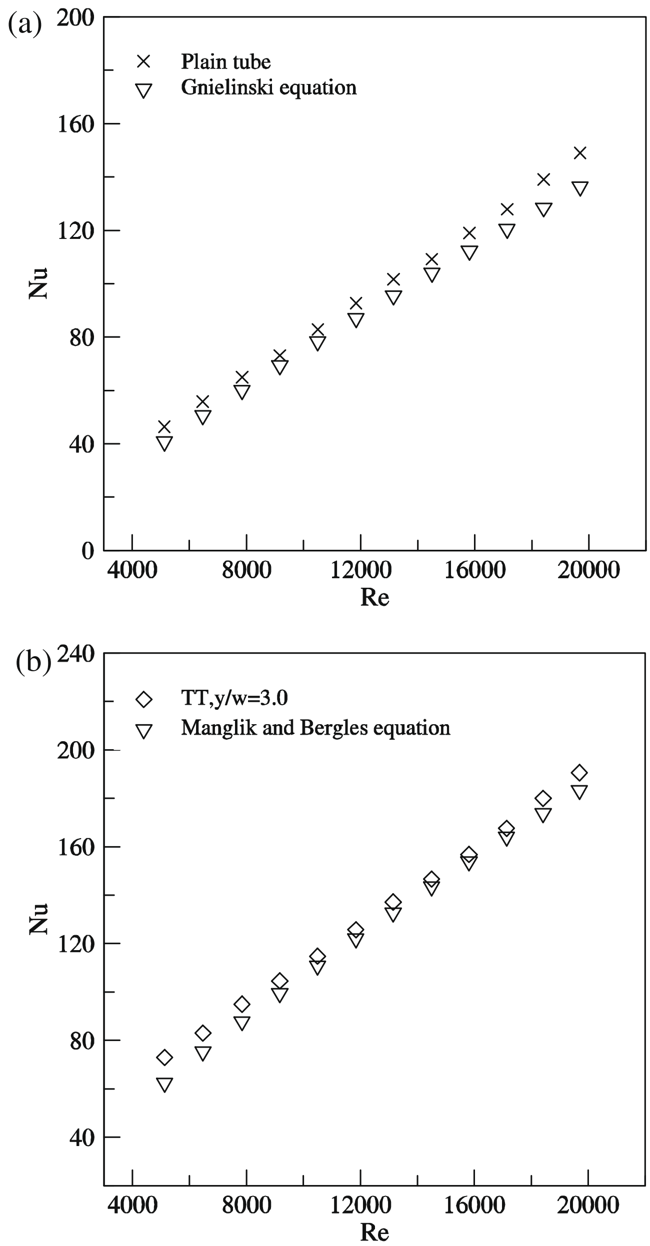

Fig. 4. Comparisons of experimental and predicted values of Nusselt numbers for (a) plain tube and (b) tube with typical twisted tape.

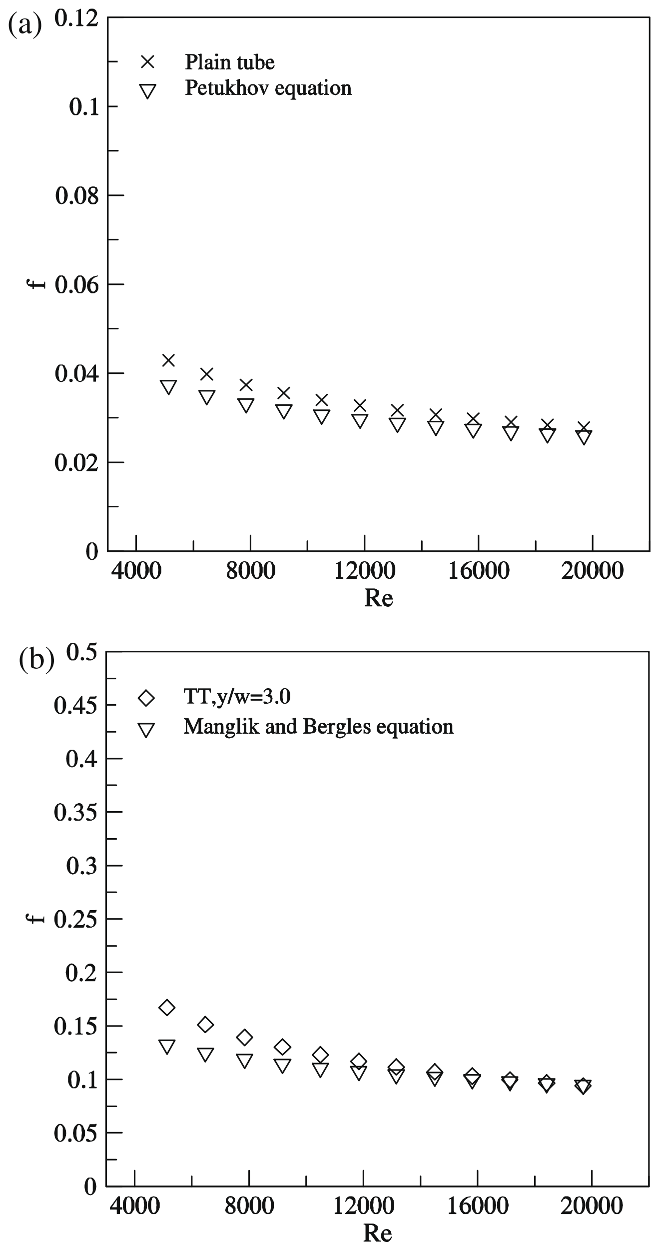

Fig. 5. Comparisons of experimental and predicted values of friction factors for (a) plain tube and (b) tube with typical twisted tape.

$$\mathrm{Nu} = 0.0141\mathrm{Re}^{0.864}\mathrm{Pr}^{0.4} \tag{15}$$

$$f = 0.68\mathrm{Re}^{-0.324} \tag{16}$$

5.2. Effect of peripherally-tape depth ratio (DR)

Effect of the peripherally-tape twisted tape on enhanced heat transfer in form of $\mathrm{Nu}_t/\mathrm{Nu}_p$ is displayed in Fig. 6. As shown, at the similar operating conditions, heat transfer rate in the tube with PT inserts were higher than that for the plain tube with and without typical twisted tape (TT) inserts for the whole range of Reynolds numbers tested. This can be explained by the fact that the peripherally-cut twisted tape provides a better mixing than a typical twisted tape. Besides the common swirl flow, it is presumed that the additional turbulence of fluid in the vicinity of the tube wall is also generated by the peripherally-cut twisted tape, removing the viscous sub-layer near tube wall and thus leading to a superior heat transfer improvement. In the past investigations, it was mentioned the jagged tape as well as the broken/spiky could induce fluid vorticity behind the jagged edges or the spikes

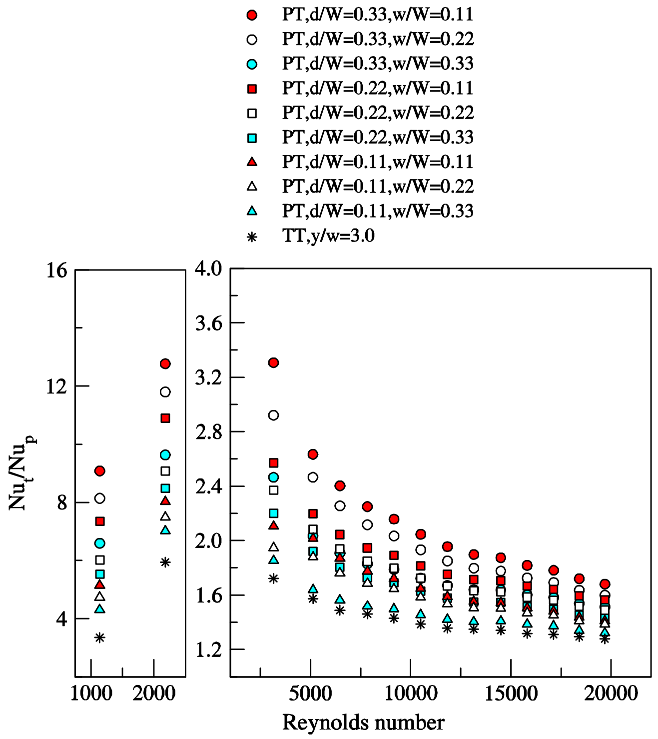

Fig. 6. Variation of $\mathrm{Nu}_t/\mathrm{Nu}_p$ with Reynolds number.

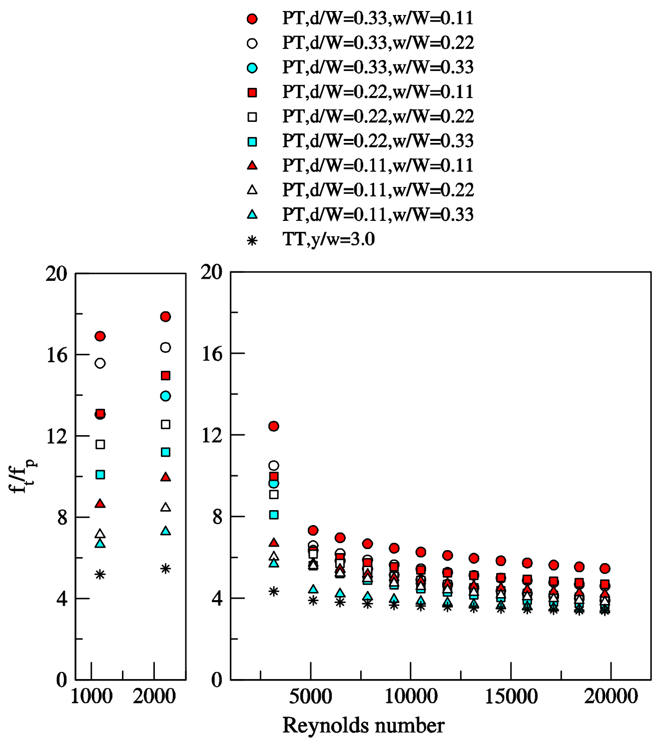

Fig. 7. Variation of f_t/f_p with Reynolds number.

[10,13,14] and the effect is favorable for the heat transfer enhancement. This mechanism is believed to be existed in the tube fitted the peripherally-cut twisted tape as well. Interestingly, the effect of disturbing the viscous sub-layer near tube wall is more prominent in laminar flow regime than that in the turbulent flow, since the thermal boundary layer in laminar regime is thicker. With respect to heat transfer rates in the plain tube, heat transfer rates in the tube with the peripherally-cut twisted tape were enhanced 12.8 times in laminar regime whereas only 2.6 times in turbulent regime. In addition, the heat transfer rate increased as the depth ratio (DR) increased. This implies that with the longer cut, the vorticity behind the cut is further promoted, resulting in the increasing turbulence and hence the additionally enhanced convective heat transfer coefficient.

Effect of the peripherally-cut twisted tape (PT) with different depth ratios (DR = 0.11, 0.22 and 0.33) on the friction loss in term of f_t/f_p is presented in Fig. 7. Similar to the heat transfer rate, the friction factors in the tube fitted by the peripherally-cut twisted tape were raised above those in the plain tube as well as the tube fitted with typical twisted tape. In general, the swirl/turbulence gives rise to substantial friction factor along the test section. The extent of turbulence caused by the peripherally-cut twisted tape as mentioned above is certainly accountable for the auxiliary of the increasing friction loss. It should be mentioned that in laminar regime friction ratio increased with increasing Reynolds number while the opposite trend was found in turbulent regime. These results accord with those reported in the earlier works [5,15]. It was also found that the friction factor increased with the rising depth ratio. Depending on the depth ratio, width ratio and Reynolds number, the friction factors in turbulent regime associated by the PTs were higher than those in the plain tube and the tube with the TT around 615% and 171%, respectively.

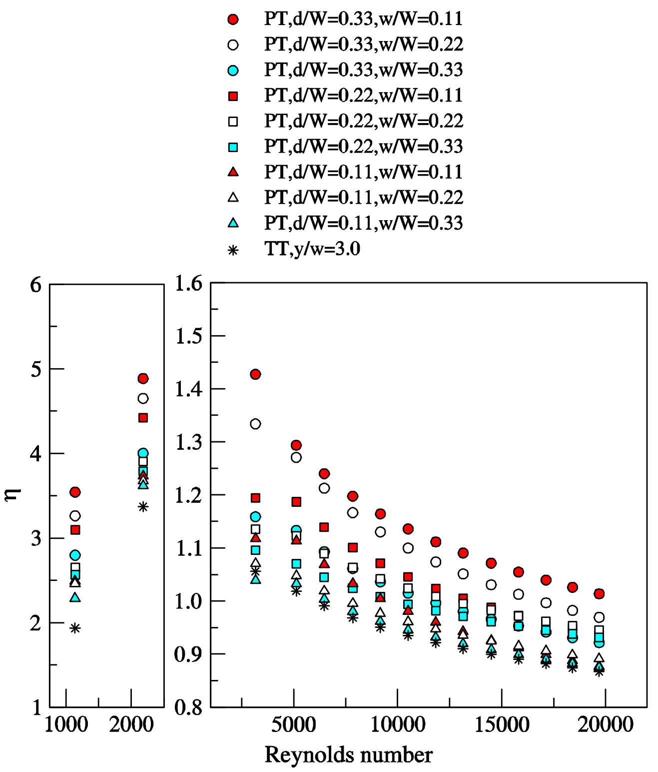

Fig. 8. Variation of thermal performance factor with Reynolds number.

5.3. Effect of peripherally-tape width ratio (WR)

The variation of Nusselt number with Reynolds number for the tube with peripherally-cut twisted tapes (PTs) with different peripherally-tape width ratios (WR = w/W = 0.11, 0.22 and 0.33) is presented in Fig. 6. Not surprisingly, at the same Reynolds number, one can observe that the peripherally-cut twisted tape with smaller tape width ratio provided higher Nusselt number than the one with a larger tape width ratio. As the tape width ratio decreases, the larger numbers of cuts appear on the tape, this directly promotes the vorticity behind the cut as well as the turbulence in locality of the tube wall, leading to a higher convective heat transfer. As found, the PT with the smallest tape width ratios (WR = 0.11) gave heat transfer rate around 5% and 28%, over those given by the tape with the WR = 0.22 and 0.33, respectively.

Fig. 7 shows that for a given Reynolds number, friction factor increased with the decreasing peripherally-tape width ratios (WR = 0.11, 0.22 and 0.33). This is attributed to the similar reason

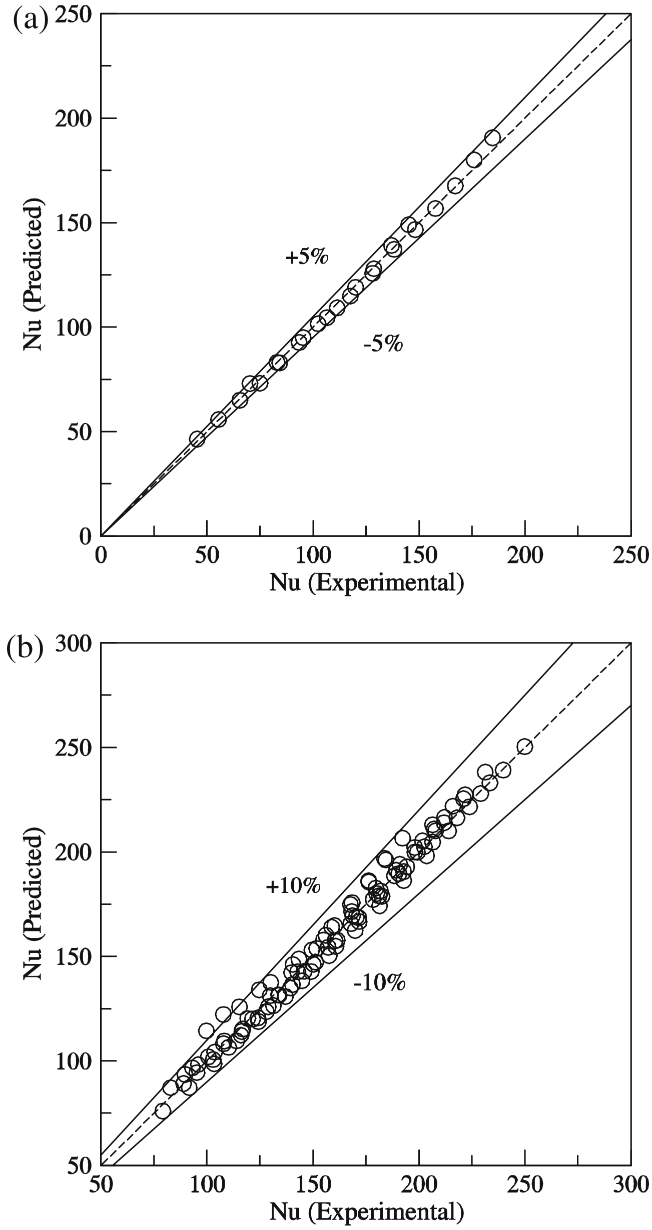

Fig. 9. Comparisons of experimental and predicted Nusselt numbers for (a) plain tube with/without TT and (b) tube with peripherally-cut twisted tape.

mentioned for the Nusselt number. The PTs with the smallest peripherally-tape width ratio (WR = 0.11) provided 3–24% and 13–40%, higher friction factors than those provided by the tapes with width ratios, WR = 0.22 and 0.33, respectively.

5.4. Performance criteria

Effect of the peripherally-cut twisted tape on the thermal performance factor characteristics is demonstrated in Fig. 8. The evaluation was made via Eq. (9) under the constant pumping power as described in Section 4 (data reduction), using $Nu_p = 48/11$ and $f_p = 64/Re$ as Nusselt number and friction factor of plain tube for laminar regime. With the use of PTs, thermal performance factors were in a range between 2.28 and 4.88 in laminar regime and around 0.88 and 1.29 in turbulent regime. On the other hand, the use of TT offered thermal performance factors in a range between 1.93 and 3.37 in laminar regime and around 0.87 and 1.01 in turbulent regime. This indicates that the applications of PTs give more efficient heat transfer enhancement both regimes compared to the use of TT. This may response to the requirement of heat transfer enhancement at high Reynolds number in the refrigerating systems and boilers [15]. In addition, the thermal performance increased with decreasing width-tape ratio (WR) and increasing depth-tape ratio (d/W). From the figure, the peripherally-cut tape with the smallest width ratio (WR = 0.11) provided the thermal performance higher than those offered by the tapes with WR = 0.22 and 0.33 of around 3% and 7%, respectively, for turbulent regime. On the other hand, the tape with the largest depth ratio (DR = 0.33) provided the thermal performance higher than those proposed by the tapes DR = 0.11 and 0.22 of around 12% and 6%, for turbulent regime. In addition, the influence of depth ratio was more dominant than that of the width ratio for all Reynolds numbers. Over the range investigated, the maximum thermal performance of 1.29 and 4.88 for turbulent and laminar regimes, was found with the peripherally-cut tape at the depth ratio, DR = 0.33 and WR = 0.11. It is noteworthy that only PT with $d/W = 0.33$ and $w/W = 0.11$ provided the thermal performance factor above unity for the whole range studied.

The empirical correlations for Nusselt number, friction factor and thermal performance factor in turbulent regime were developed from the experimental results as follows:

$$Nu = 0.244Re^{0.625}Pr^{0.4}\left(\frac{d}{W}\right)^{0.168}\left(\frac{w}{W}\right)^{-0.112} \quad (17)$$

$$f = 39.46Re^{-0.591}\left(\frac{d}{W}\right)^{0.195}\left(\frac{w}{W}\right)^{-0.201} \quad (18)$$

$$\eta = 4.509Re^{-0.152}\left(\frac{d}{W}\right)^{0.102}\left(\frac{w}{W}\right)^{-0.054} \quad (19)$$

The fitted values of Nusselt number, friction factor and thermal performance factor for turbulent flow obtained from Eqs. (17)–(19) are compared with the experimental values as shown in Figs. 9–11. For plain tube with or without typical twisted tape, the experimental data and the fitted values were in good agreement with deviations of within ±5 for Nusselt number and ±3 for friction factor. For the tube fitted with the peripherally-cut twisted tape, deviations of

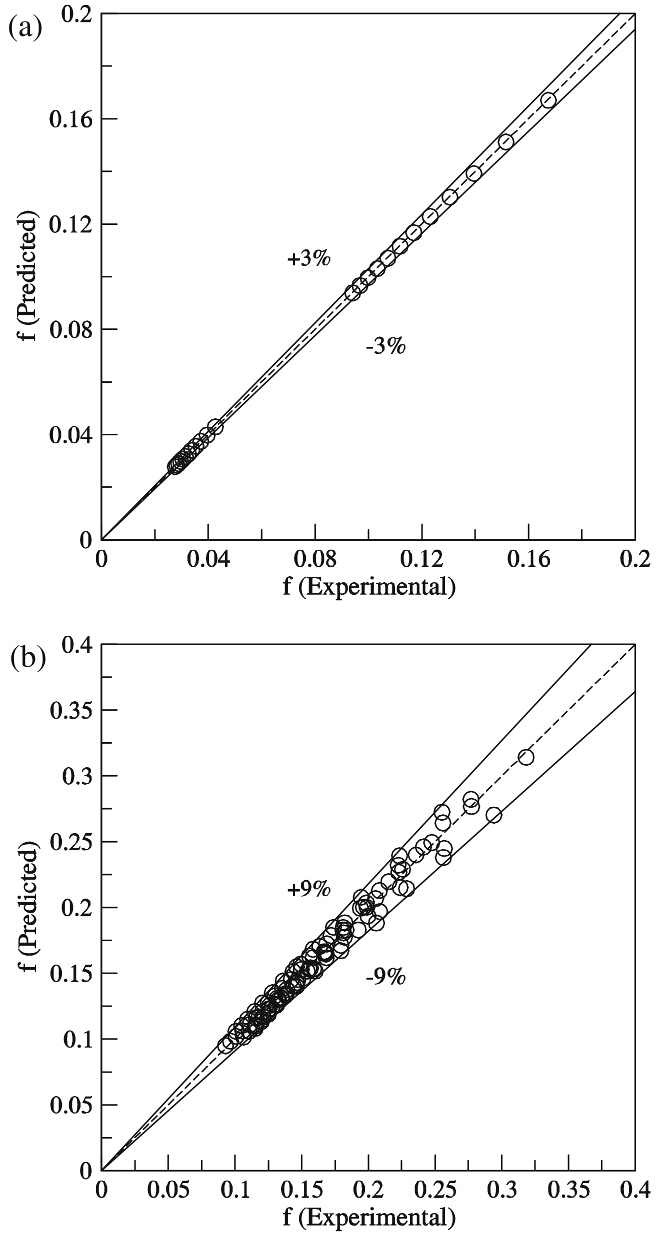

Fig. 10. Comparisons of experimental and predicted friction factors for (a) plain tube with/without TT and (b) tube with peripherally-cut twisted tape.

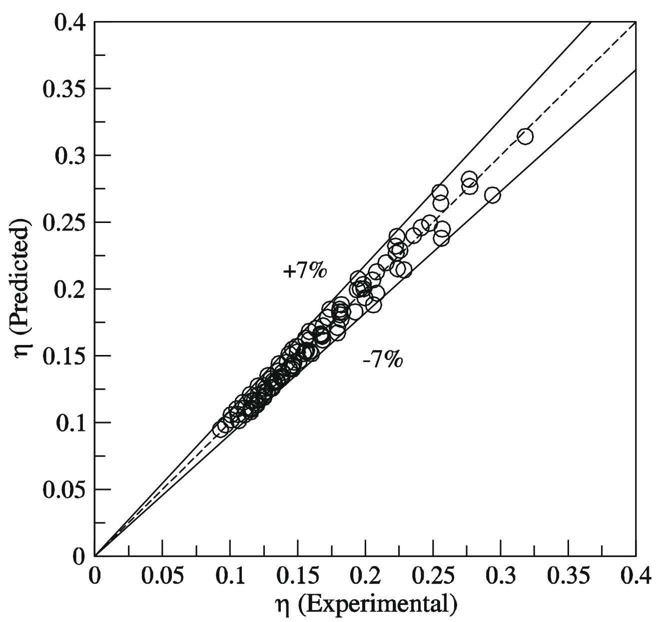

Fig. 11. Comparisons of experimental and predicted thermal performance factors for tube with peripherally-cut twisted tape.

experimental data from the fitted data were within ±10%, ±9% and ±7%, for Nusselt number, friction factor and thermal performance factor, respectively.

6. Conclusions

The effects of the peripherally-cut twisted tape on the heat transfer enhancement, friction and thermal performance factor behaviors in laminar and turbulent flow regimes ($1000 \leqslant Re \leqslant 20{,}000$) are described in the present paper. The peripherally-cut tapes with different tape depth ratios ($DR = d/W = 0.11$, 0.22 and 0.33) and tape width ratios ($WR = w/W = 0.11$, 0.22 and 0.33) were tested using the water as the working fluid. The conclusions can be drawn as follows:

- The peripherally-cut twisted tape offered higher heat transfer rate, friction factor and also thermal performance factor compared to the typical twisted tape. An additional turbulence of fluid in the vicinity of the tube wall and vorticity behind the cuts generated by the modified twisted tape are referred as the reason for a better heat transfer enhancement.
- Nusselt number, friction factor as well as thermal performance factor associated by the peripherally-cut twisted tape were found to be increased with increasing tape depth ratio (DR) and decreasing tape width ratio (WR).
- Over the range investigated, the maximum thermal performances of 1.29 for turbulent flow and 4.88 for laminar flow, were found with the use of the peripherally-cut tape at the depth ratio, DR = 0.33 and WR = 0.11.

Acknowledgements

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Thermohydraulic investigation of turbulent flow through a round tube equipped with twisted tapes consisting of centre wings and alternate-axes

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ABSTRACT

The effects of the twisted tapes consisting of centre wings and alternate-axes (WT-A) on thermohydraulic properties in a round tube, were investigated. The effects of other three types of twisted tapes including: (1) the twisted tape with wings alone (WT), (2) the twisted tape with alternate axes alone (T-A), and (3) the typical twisted tape (TT), were also studied for comparison. All twisted tapes used were twisted at constant twist length (y) of 57 mm, corresponding to a constant twist ratio (y/W) of 3.0. The wings were generated along the centre line of the tape with three different angles of attack, ($\beta = 43°$, $53°$ and $74°$). Test runs were conducted using water as a testing fluid with Reynolds number range between 5200 and 22,000. Under the similar condition, the heat transfer rate in the tube fitted with the WT-A was consistently higher than those in the tube equipped the WT, T-A and plain tube. It is also found that the heat transfer rate increased with increasing angle of attack. Over the range studied, the use of WT-A at $\beta = 74°$ was found to be the most effective for heat transfer enhancement, giving thermal performance factor of up to 1.4. Mean values of Nusselt number (Nu), friction factor (f), thermal performance factor (η) provided by the WT-A (at $\beta = 74°$) were respectively, 17.7%, 30.6% and 7.8% higher than those in the tube with WT (at $\beta = 74°$), 20.8%, 53% and 4.9% higher than those in the tube with T-A, and 62%, 123% and 24% higher than those in the tube with TT. The superior performance of the WT-A over those of the other tapes could be attributed to the combined effects of the following actions: (1) a common swirling flow by the twisted tape (2) a vortex generated by the wing (3) a strong collision of the recombined streams behind each alternate point. For a better understanding on flow phenomena, flow-visualization by smoke wire technique is also presented. In addition, the experimental correlations of Nusselt number, friction factor and thermal performance factor were also developed.

1. Introduction

Enhancement of heat transfer in a heat exchanger is widely applied in industries due to the need of more compact heat exchanger, a lower operating cost, energy saving as well as ecological benefit. Among many heat transfer enhancement techniques, utilization of twisted tape and deta-wing/delta-winglet vortex generators is a promising method. The approach possesses not only an effective heat transfer enhancement but also the advantage of a low cost and an ease of installation.

Twisted tapes have been extensively used as heat transfer enhancing devices in heat exchangers. The important effects induced by the tapes are: (1) swirl flow which improves fluid mixing, (2) helically twisting fluid motion which offers an effectively longer flow path, and (3) partitioning and blockage of the tube flow cross section which leads to a higher flow velocity [1]. All the effects mentioned above are directly responsible for the improvement of heat transfer within heat exchanger. The performances of twisted tape swirl generator have been intensively investigated by many researchers [1–14] for different tape geometries (e.g. broken, serrated, delta-wing, perforated, notched and jagged), working fluid types (e.g. water, air, oil, servotherm medium oil, turbine oil, ethylene glycol, nitrogen, and R134a), wall conditions (uniform heat flux and constant wall temperature) and flow regimes (laminar, transition and turbulent flows). Regarding to the results reported in the mentioned papers, it can be observed that the shape of twisted tape is an important key signifying the tape performance, for example serrated twisted tape [3], the modified tape with protrusive part could improve heat transfer rate significantly, however the tape caused an unavoidably dramatic increase of friction within system and this prohibits its practical applications. On

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Nomenclature

A	heat transfer surface area, m^2
C_p	specific heat of fluid, J kg^{-1} K^{-1}
D	inside diameter of the test tube, m
f	friction factor = $\Delta P/((L/D)(\rho U^2/2))$
h	heat transfer coefficient, W m^{-2} K^{-1}
I	current, A
k	thermal conductivity of fluid, W m^{-1} K^{-1}
L	length of the test section, m
M	mass flow rate, kg s^{-1}
Nu	Nusselt number = hD/k
P	pressure of flow in stationary tube, Pa
ΔP	pressure drop, Pa
Pr	Prandtl number = $\mu C_p/k$
Q	heat transfer rate, W
Re	Reynolds number = $\rho UD/\mu$
t	thickness of the test tube, m
T	temperature, K
$\tilde{T}_s$	mean temperature, K
U	mean axial flow velocity, m s^{-1}
V	voltage, V
W	twisted tape width, m
Y	twisted tape pitch, m

Greek symbols

β	angle of attack, degree
ρ	fluid density, kg m^{-3}
δ	twisted tape thickness, m
μ	fluid dynamic viscosity, kg s^{-1} m^{-1}
η	thermal performance factor

Subscripts

b	bulk
c	convection
i	inlet
o	outlet
p	plain
s	surface
t	turbulator
w	water

Abbreviations

TT	typical twisted tape
T-A	twisted tape with alternate-axes
WT	twisted tape with centre wings
WT-A	twisted tape with centre wings and alternate-axes

the other hand, the twisted tapes with space such as perforated, notched [2], and regularly-spaced [6] twisted tapes generated lower friction loss but their heat transfer enhancement and thus thermal performance factor are unattractive. Differently, the twisted tapes consisting of protrusive parts together with spaces on the tapes such as broken [4] and delta-wing twisted tapes [8] offered promising outcomes in both viewpoints of heat transfer enhancement and friction loss since the protrusion effectively improved mixing while the space brought down the friction to satisfactory level. This resulted in thermal performance factor above unity which is beneficial for energy saving and hence made the twisted tape suitable for industrial applications.

Delta-wing and delta-winglet vortex generators were used for heat transfer augmentation by many researchers [15–23]. Tiwari et al. [15] and Fiebig [16] described that vortices enhanced the mixing of fluid in the periphery region with that in the core region of the flow field by the swirling motion as a secondary flow. This effect led to the destabilization of primary flow and thermal boundary layer thinning, and thus heat transfer improvement. The numerical investigation by Biswas et al. [17] and the experimental finding by Valencia et al. [18] indicated that the proper use of vortex generators resulted in an effective heat transfer augmentation with low or moderate pressure drop penalty. Vortex generators can be incorporated into a plate, a fin surface or a channel with an attack angle by means of punching, embossing, stamping, or attachment process [19]. There are two types of vortices: transverse vortex and longitudinal vortex. The axis of a transverse vortex lies perpendicular to the flow direction while that of the longitudinal vortex (also called streamwise vortex) is parallel to the main flow. Practically, longitudinal vortex and transverse vortex are induced simultaneously and the dominance of the one over the another depends on the attack angle. The longitudinal vortex is governing at small attack angle whereas the transverse vortex becomes more influential with increasing attack angle [20]. This issue gains significance in design of delta-wing for specific applications [21]. It was also mentioned that a longitudinal vortex offered less friction loss and more efficient heat transfer than a transverse vortex [20,22]. Regarding to results by Fiebig [23], delta-wings were more efficient than rectangular wings, and punched wings were slightly better than mounted wings for the same vortex generator area.

Most of the delta-wing and delta-winglet vortex generator appear in a flat plate. In our previous report [8], the delta-winglet was located on the periphery of twisted tape. Our results indicated that under the similar operating conditions, the Nusselts numbers in the test tube equipped with peripheral delta-winglet twisted tape were considerably higher than those in the tube equipped with typical twisted tape. Besides, our evaluation revealed that the thermal performance factor in the tube with the modified twisted tape was enhanced up to 1.24. This points out that the presence of the winglet on twisted tape gives a promising result for heat transfer enhancement.

In present work, the modified twisted tape was alternatively designed by generating wings on the centre line of the twisted tape and then forming alternate axes (twisted tape with wings and alternate axes, WT-A). The modification has been assumed to induce the combined effects of the common swirl flow by the twisted tape, vortex by wings and also additional flow fluctuation by alternate axes, leading to excellent results for heat transfer enhancement. The effects of other three types of twisted tapes including (1) the twisted tapes with wings alone (WT), (2) the twisted tape with alternate axes alone (T-A), and (3) the typical twisted tape (TT), were also studied for comparison. The experiments were conducted to investigate the effects of twisted tapes used on the heat transfer rate, friction factor as well as thermal performance factor in a turbulent tube flow with Reynolds number between 5200 and 22,000 using water as a working fluid.

2. Wing twisted tape with/without alternate axis

All twisted tapes used in the present work, were made of aluminum strips with thickness (δ) of 0.8 mm, width (W) of 19 mm and length (L) of 1000 mm. The twist ratio y/W defined as ratio of twist length (y, 180°/ twist length) to tape width of twisted tapes (W), was kept constant at 3.0. All tapes were initially formed as typical twisted tapes. The twisted tape with centre wings (WT) was

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modified from the typical twisted tape (TT) by punching delta-wings on the TT along the centre line (one delta-wing per twist length). The adjacent wings were open to the different sides of the twisted tape (left to right and vice versa) as shown in Fig. 1a and b. The attack angle of the wing (β) was varied at 43°, 53° and 74°. Another modified twisted tape, twisted tape with

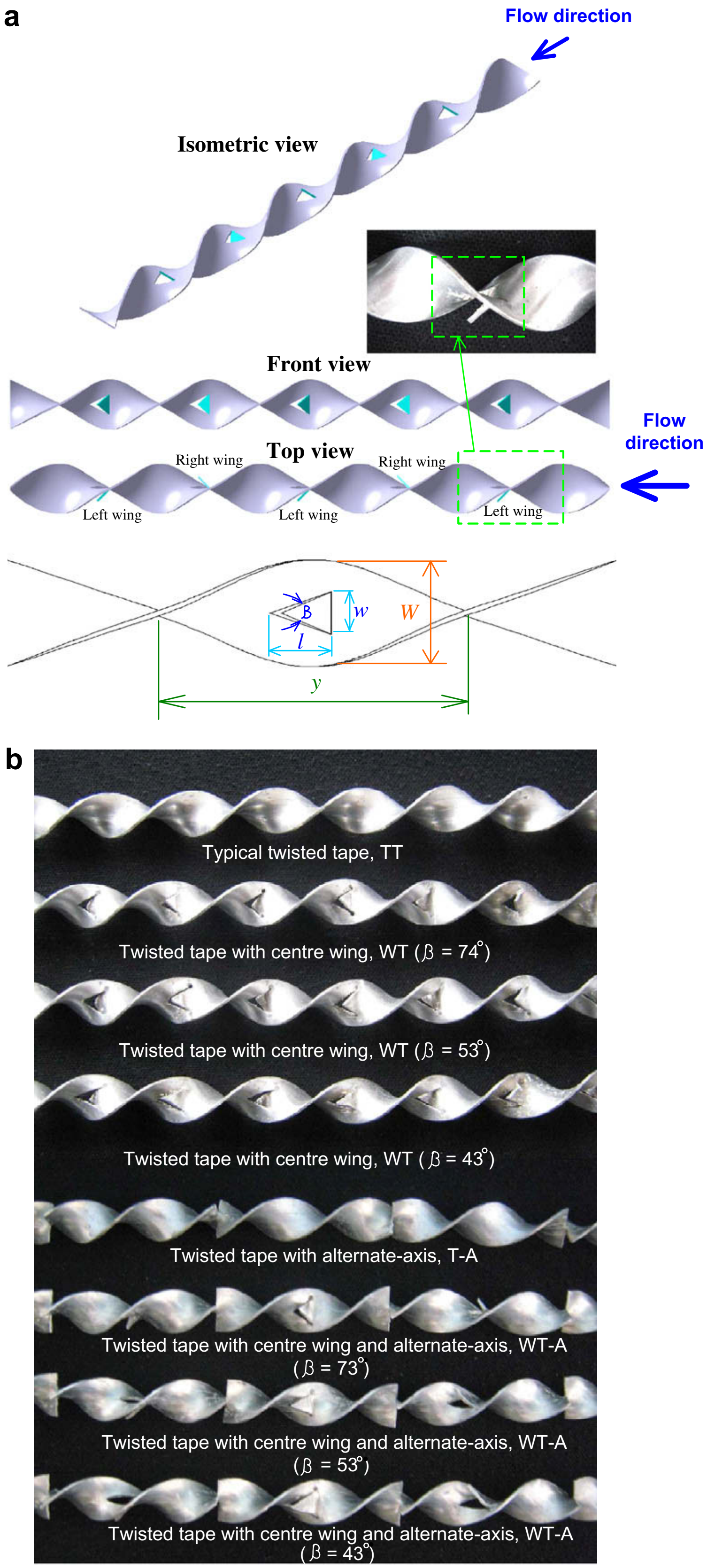

Fig. 1. (a)Geometry of twisted tapes. (b)Photograph of twisted tapes.

Table 1
The ranges of operation parameters.

Test tube conditions	
Reynolds number	5200–22,000
Inlet water temperature	28 °C
Inner tube diameter, D	19.5 mm
Outer tube diameter	22.5 mm
Tube thickness, t	1.5 mm
Test tube material	Copper
Twisted tape	
Twist ratio, y/W	3.0
Tape width, W	19 mm
Twist length, y	57 mm
Twist tape length, L	1000 mm
Tape thickness, δ	0.8 mm
Angle of attack, β	43°, 53° and 74° (for WT and WT-A)
Width of wing, w	6, 9 and 11.5 mm
Length of wing, l	9 mm
Tape material	Aluminum

alternate axes alone (T-A) was formed by cutting on both sides of the TT, with the depth of cut of 4 mm on each side. Subsequently, the tape was twisted to angle difference of 90° to produce an alternate axis. The distance between each pair of alternate points was set at two twist lengths. The last modified twisted tape, the twisted tape with centre wings and alternate axes (WT-A) was formed by creating wings on the TT (one wing per two twist lengths) and then generating alternate axis in the middle between each pair of the wings. Note that the consequence of generating the alternate points on the WT-A, made wings appeared on four sides of the tape (left–right–top–bottom). The photograph of all tapes used in the present work, including one TT, one T-A, three different WT and three different WT-A, is shown in Fig. 1b and their dimensions are also summarized in Table 1.

3. Experimental facility

3.1. Heat transfer set-up

An apparatus used in the present work, is shown in Fig. 2a and b. The heat transfer set-up facility consists of: (1) water reservoir (2) measuring and recording devices (3) a calm section and a test section which were made from copper. The length and inner diameter of the calm section were respectively, 1500 mm and 19.5 mm while those of the test section were respectively 1000 mm and 19.5 mm. The dimension of the calm section met a requirement for fully developed turbulent flow in which $L/D \approx 75$. In the experiment, a testing fluid (water) was continuously supplied from a water reservoir at ground level to an overhead tank. Water was directed from the overhead tank to the calm section and then the test section. The test tube was spirally wounded by electrical heating wire with a constant wire pitch on its outer surface. Due to the sufficient thermal insulation over the test section, only around 3–7% of total heat provided by heating wire was lost from test tube during the test. Heat supplied via the electrical heating wire, was controlled by the variac transformer. Local surface wall temperatures (T_s) at fifteen different axial stations were measured using K-type thermocouples while the inlet and outlet temperatures of cold water and hot water were measured using two RTDs. All temperature data were collected by data logger (Yokogawa: FX122-4-2) and directly transferred to a personal computer.

The inlet water (at temperature of about 28 °C) flow rate was set to the desired value varying from 4 to 16 L/min. This range corresponded to a range of Reynolds number from 5200 to 22,000. Pressure drop across the test section were measured via pressure taps which connected to manometer and located approximately 50 mm upstream and 150 mm downstream of the test section. Note that the pressure drop was measured under an isothermal flow condition without switching on the heater. All data were recorded after attaining steady state. More details of the experimental set-up are described elsewhere [8]. The test tube conditions are summarized in Table 1.

3.2. Flow-Visualization set-up

To gain a better understanding on flow phenomena induced by each twisted tape, the flow visualization was conducted. The smoke wire flow-visualization set-up is schematically illustrated in Fig. 2b. The facility consists of (1) a copper tungsten wire with diameter of 0.2 mm (2) a paraffin oil dropper (as smoke source), (3) a twisted tape insert, (4) a light bulb to lighten the test section, (5) a voltage supplier, (6) a digital camera (Nikon D80), (7) an inverter, (8) a hot wire anemometer, (9) a blower and test section made from acrylic tube, and (10) a personal computer. The twisted tape was made from the steel with width of 35 mm, and the tape was twisted at constant twist ratio (y/W) of 4.0. In the experiment, the paraffin oil was dropped on the copper tungsten wire. Then, heat was supplied to the oil through the voltage supplier to generate smoke and air was fed into the tunnel as a smoke carrier. The velocity of air in the tunnel was measured using a hot wire anemometer and air flow rate was kept constant, corresponding to the constant Reynolds number of 2500.

4. Data reduction

In the experiment, electrical power input (Q_{VI}) was supplied to the tube wall via heating wire:

$$Q_{VI} = V \times I \tag{1}$$

Due to heat loss from the test section to surrounding, the net heat transfer rate from the inner tube surface to water (Q_w) at the steady-state condition can be expressed as

$$Q_w = Q_{VI} - Q_{\text{loss}} = MC_{p,w}(T_o - T_i) \tag{2}$$

where Q_{loss} is the rate of heat loss, $M = \rho U(\pi D^2/4)$ is the mass flow rate of water through the test tube, T_i and T_o is the inlet and outlet temperatures of the working fluid, respectively. According to the heat balance at the steady state flow condition, Q_{loss} was found in between 3% and 7% of the input electrical power (Q_{VI}).

It is assumed that the rate of heat gained by water is equal to the rate of convective heat transfer (Q_c) as

$$Q_c = Q_w \tag{3}$$

The convective heat transfer is defined in term of the nominal inside surface area as

$$Q_c = hA(\tilde{T}_s - T_b) \tag{4}$$

where T_b is bulk flow temperature which was calculated from $T_b = (T_i + T_o)/2$. $\tilde{T}_s$ is the mean inside tube-wall temperature of the test section which was computed by taking the average temperatures of 15 axial stations as

$$\tilde{T}_s = \sum (T_{s1} + T_{s2} + \cdots + T_{s15})/15 \tag{5}$$

The mean heat transfer coefficient h was then evaluated from the following equation:

$$h = \mathrm{M}C_{p,w}(T_o - T_i)/A(\tilde{T}_s - T_b) \tag{6}$$

It should be mentioned that the water thermo physical properties were evaluated at the bulk flow temperature (T_b).

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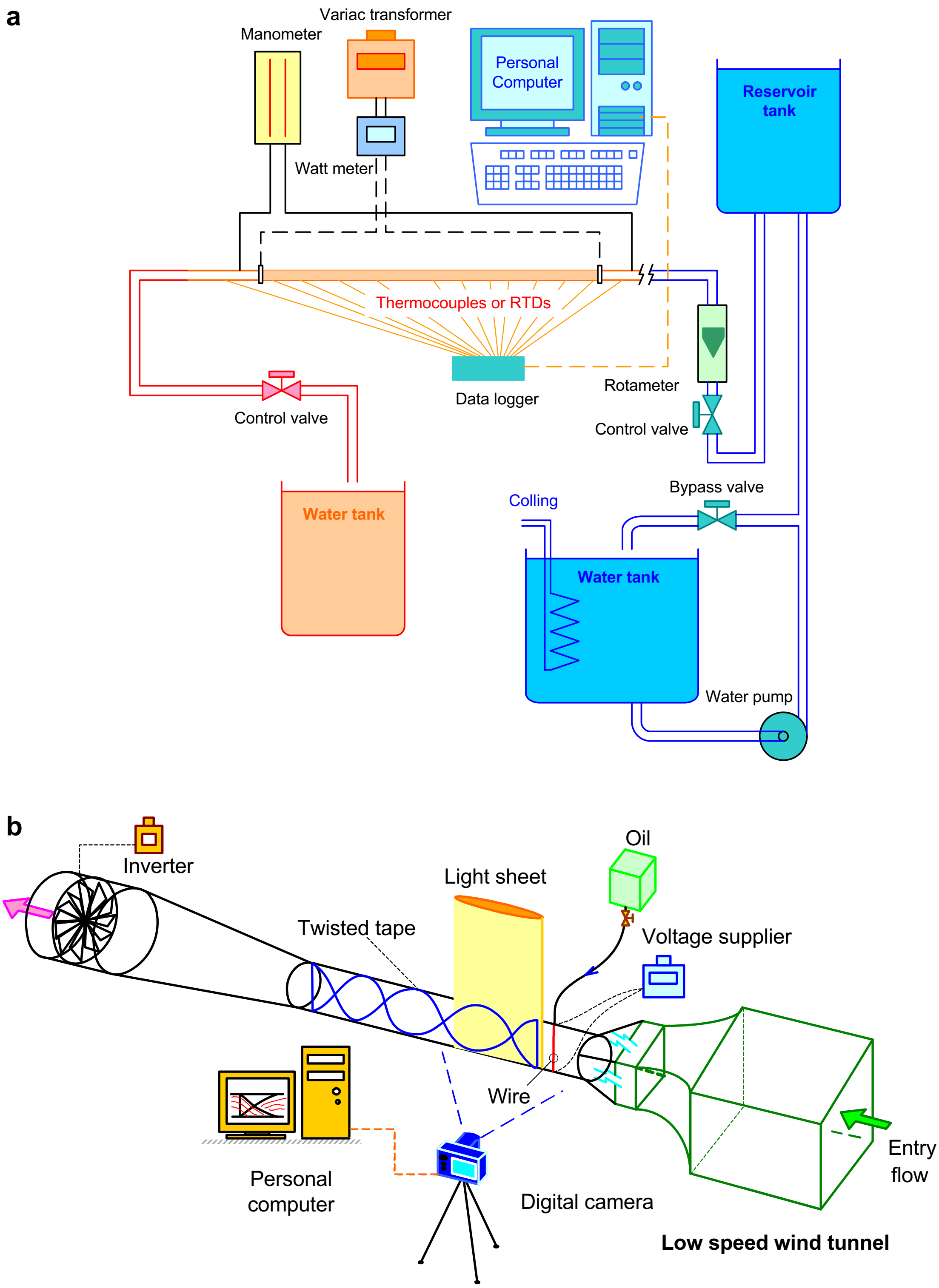

Fig. 2. Schematic diagram of experimental apparatus: (a) heat transfer set-up and (b) smoke wire set-up.

The heat transfer coefficient inside the test tube is reported in term of mean Nusselt number which can be written as

$$\mathrm{Nu} = hD/k \tag{7}$$

Another important parameter characterizing the fluid flow through the tube is the friction factor

$$f = 2(\Delta PD)/(\rho U^2 L)) \tag{8}$$

where ΔP is pressure drop across the test section which was evaluated under an isothermal flow condition. Reynolds number stated in the present work is based on the average flow inlet velocity and the tube inlet diameter as

$$\mathrm{Re} = UD/\nu \tag{9}$$

Finally, thermal performance factor at constant pumping power which is a factor for evaluating a potential of a twisted tape for actual application, can be expressed as

$$\eta = (\mathrm{Nu}_t/\mathrm{Nu}_p)/(f_t/f_p)^{1/3} \tag{10}$$

In the present experiment, the evaluations for experimental uncertainties of the present data reduction process were also performed by following the method described elsewhere [24]. The maximum uncertainties for Nu, Re and f were found at about 8%, 5% and 7%, respectively.

5. Experimental results

The experimental results including Nusselt number (Nu), friction factor (f) and the thermal performance factor (η) behaviors in the tube with different twisted tapes (TT, T-A, WT, and TW-A) are presented and discussed in the following subsections.

5.1. Validation test

To evaluate the reliability and precision the present experimental facility, Nusselt number and friction factor in the present plain tube were taken and then compared with those in the previously published correlations from the open literature including, Gnielinski correlation and Dittus and Boelter correlation for the Nusselt number and Petukhov correlation and Blasius correlation for the friction factor [25]. In addition, the Nusselt numbers and friction factors in the tube equipped with the typical twisted tape were also verified with those reported by Manglik and Bergles [1].

Fig. 3a–c presents the comparison between the measured data (Nusselt number, friction factor and thermal performance factor) and those calculated from the previous correlations [1,25]. For the plain tube, the present mean Nusselt number was respectively, 5% and 3% lower than those from Gnielinski equation and Dittus & Boelter equation while the present mean friction factor data was respectively 9% and 10% higher than those of Blasius equation and Petukhov equation. For the tube fitted with typical twisted tape, the present mean Nusselt number was 7% lower and friction factor was 9% higher than those from Manglik and Bergles equation. According to the comparative results mentioned above, the reliability of the present facility is considered acceptable. Results of the thermal performance factor (η) of the present plain tube fitted with typical twisted tape (TT) are compared with those of the

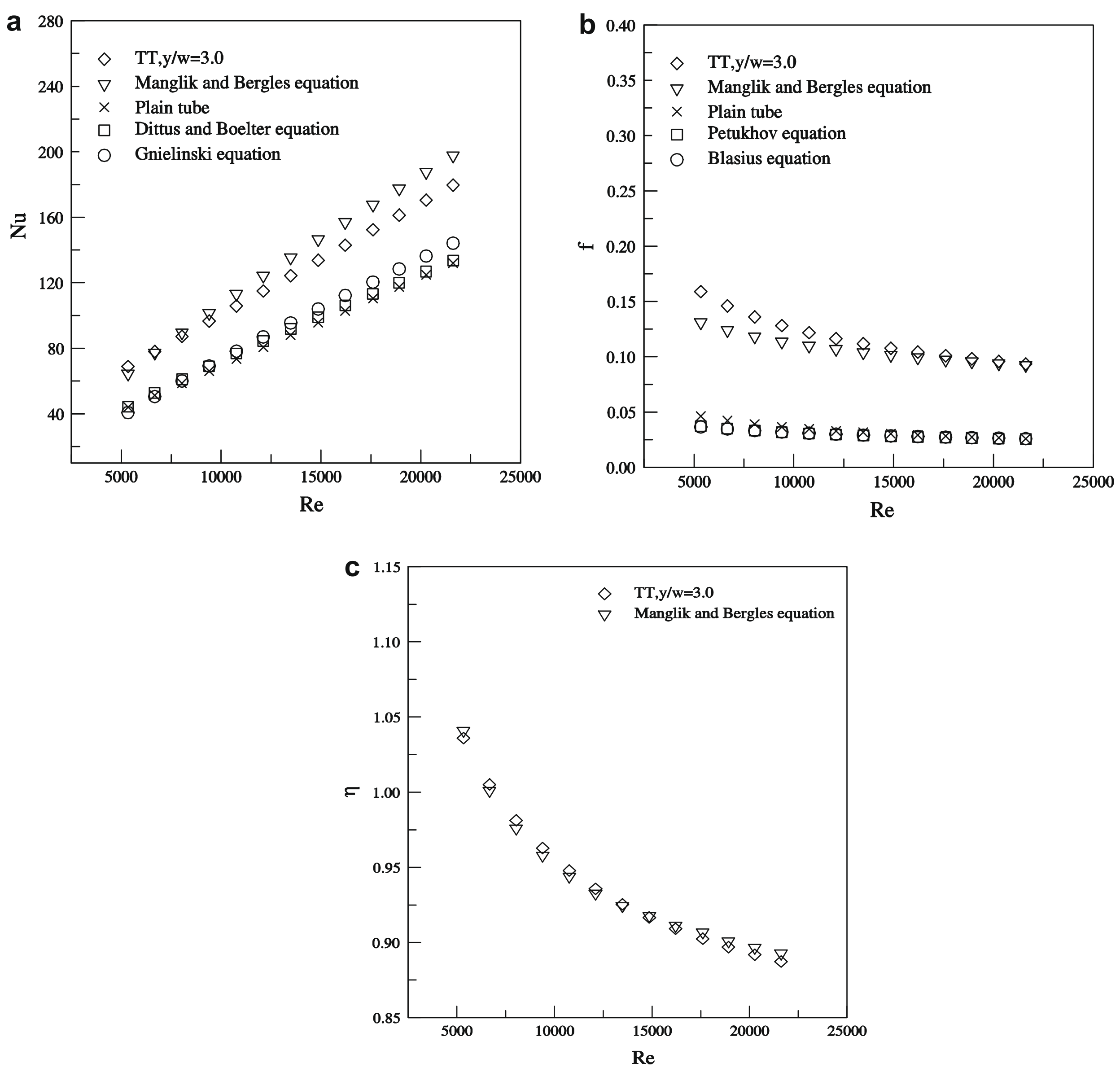

Fig. 3. Validation of the plain tube with/without typical twisted tape inserts for: (a) Nusselt number, (b) friction factor and (c) thermal performance factor.

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previous correlations [1,25] by using Eq. (10). For the previous studies, the values of the Nusselt number (Nu_t) and friction factor (f_t) of the tube fitted with TT were calculated from the Manglik and Bergles correlation while the Nusselt number (Nu_p) and friction factor (f_p) values of the plain tube were calculated using correlations of Gnielinski and Petukhov, respectively. It is obvious that the present results of thermal performance factor reasonably agree well with the previous work.

5.2. Effect of wing twisted tape

5.2.1. Heat transfer results

The comparison of Nusselt numbers of the tube with typical twisted tape (TT) and twisted tapes with centre wings (WT) are demonstrated along with those of the plain tube in Fig. 4. At the similar operating conditions, Nusselt numbers (Nu) in the tube with WT were found to be consistently higher than those in the tube with TT and plain tube. Depending on Reynolds number and angle of attack the Nusselt numbers in the tube with WT, were 16% to 52% and 58% to 138% higher than those in the tube with TT and the plain tube, respectively. The superior performance of the WT compared to that of the TT, can be attributed the combined effect of common swirl flow induce by the twisted tape and the vortex generated by the centre wings in the WT. This can be supported by the fact that the vortex provides the additional disturbance into the typical swirl stream generated by the twisted tape. This action directly improves heat transfer rate with respect to that offered by the swirl flow alone.

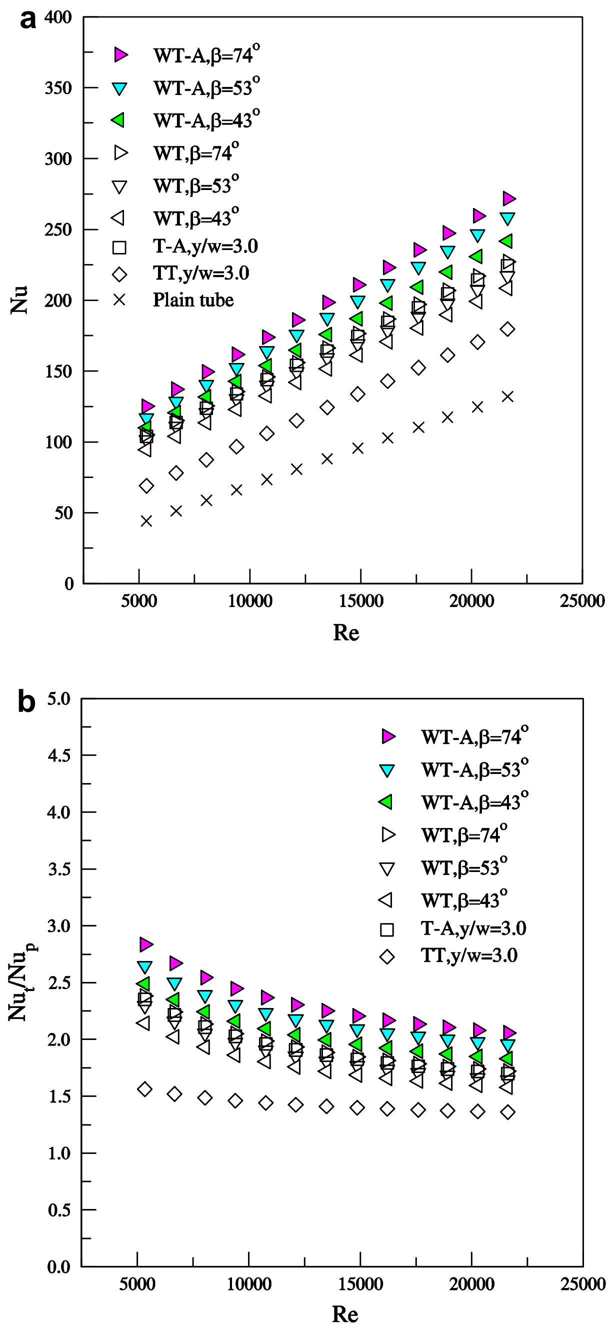

Fig. 4. Effect of twisted tape on Nusselt number: (Nu) and (Nu_t/Nu_p).

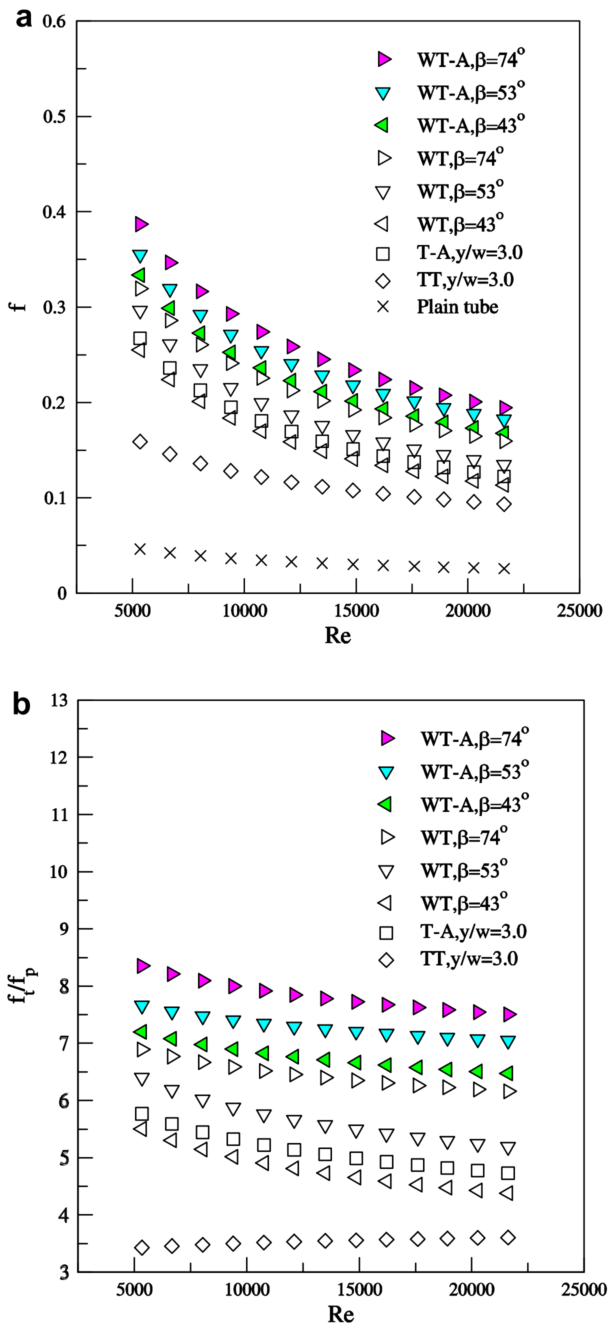

Fig. 5. Effect of twisted tape on friction factor: (f) and (f_t/f_p).

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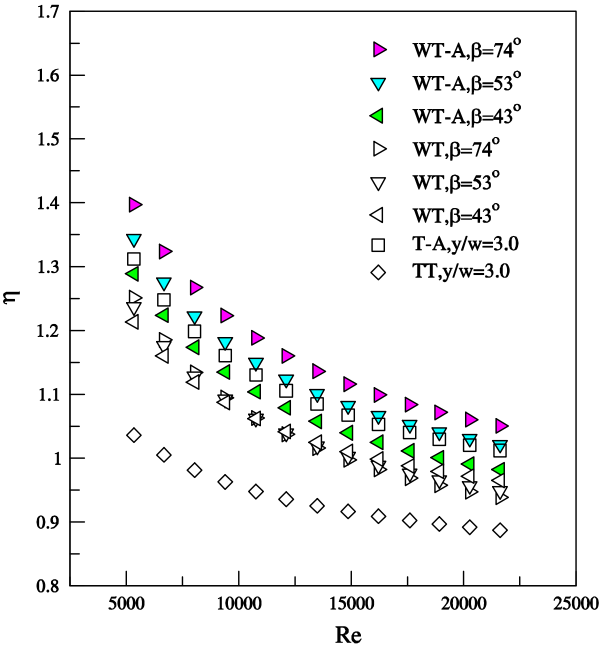

Fig. 6. Effect of twisted tape on thermal performance factor.

5.2.2. Friction factor results

The experimental results for friction factor in the tube fitted with the TT and the WT as well as a plain tube under an isothermal flow condition are shown in Fig. 5. Apparently, the friction factors in the tube fitted with the twisted tape consisting of centre wings (WT) were found to be higher than those in the tube with the typical twisted tape around 21.5–101% and higher than those in plain tube from 4.4 to 6.9 times. This is due to an extra blockage by the wings appears on the WT and the additional forces exerted by a vortex flow, giving rise to a more pronounced dissipation of the dynamical pressure of a working fluid.

5.3. Effect of angle of attack

5.3.1. Heat transfer results

The effect of angle of attack (β) on heat transfer characteristics in the tube fitted with the WT is also reported in Fig. 4. The result revealed that Nusselt number (Nu) increased with the increase of angle of attack. Over the range studied, the WT with the largest angle of attack ($\beta = 74°$) provided the highest heat transfer rate. The maximum increase in Nusselt number obtained in the tube with this tape was 11%, 4.7% and 138% compared to those in the tube with the WT at angles of attack, $\beta = 43°$ and 53° and the plain tube, respectively. This can be explained that the wing with the larger

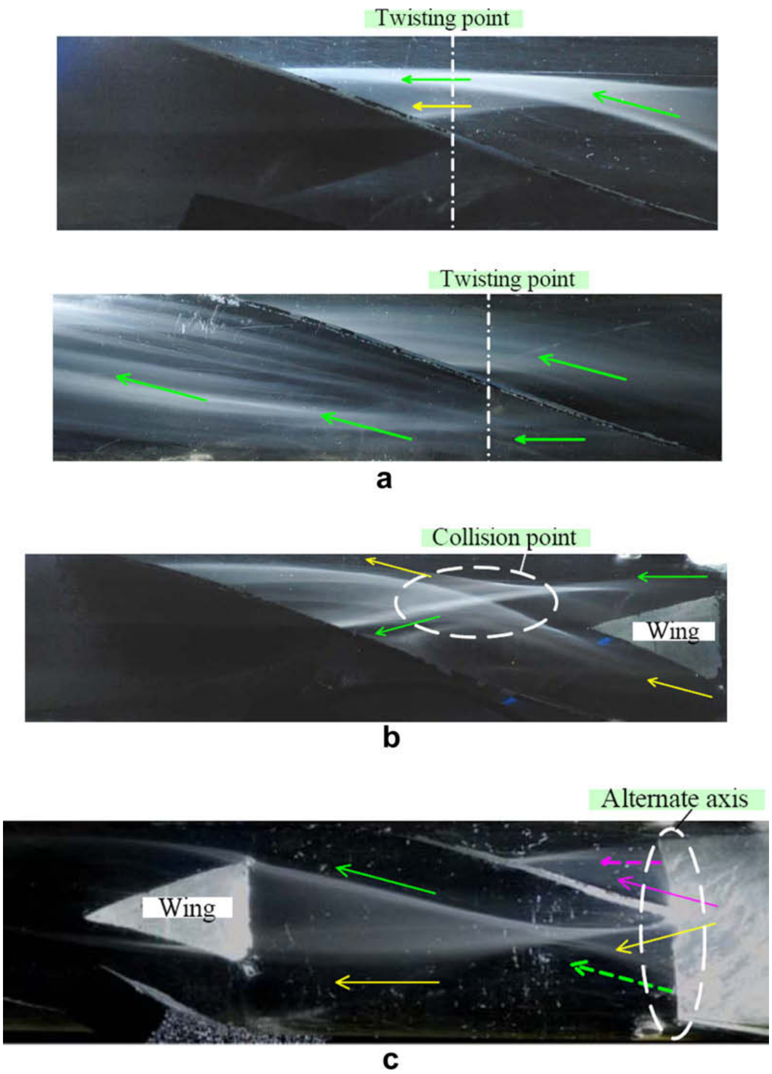

Fig. 7. Visualization of flow through tube with twisted tape inserts by smoke wire technique: (a) TT, (b) WT and (c) WT-A.

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attack angle gives the more efficient mixing which provides more disruption to a thermal boundary layer, leading to a better heat transfer.

5.3.2. Friction factor results

Fig. 5 presents the effect of angle of attack (β) on friction factor which was taken under an isothermal condition. As expected, under the similar conditions, friction factor increased with the increasing angle of attack. The mean friction factors in the tube with the WT at angles of attack, $\beta = 43°$, $53°$ and $74°$ were respectively 4.8, 5.6 and 6.4 times of those in the plain tube and 1.4, 1.6 and 1.8 times of those in the tube with the TT. This is simply attributed to a greater flow blockage by the centre wings at the larger angle of attack.

5.4. Effect of an alternate axis and a combination of wing and alternate axis

Determination from Figs. 4–6, it is found that under the similar conditions, the heat transfer coefficient, friction factor and thermal performance factor in the tube fitted with the WT-A were consistently higher than those in the tube equipped the WT, T-A and plain tube. The heat transfer enhancement (HTE) data is also presented in term of Nusselt number ratio (Nu_t/Nu_p) which is plotted against the Reynolds number as shown in Fig. 4b. Obviously, the HTE ratio is high at low Reynolds number and then decreased with increasing Reynolds number and became nearly constant at high Reynolds number. Fig. 5b shows the variation of the friction factor ratio (f_t/f_p) with the Reynolds number value for various twisted tapes. The trend with Reynolds number is found to be similar to that for the HTE ratio. Over the range investigated, WT-A with $\beta = 74°$ gave maximum thermal performance factor of 1.4 at Reynolds number of 5200. Mean values of Nusselt number (Nu), friction factor (f), thermal performance factor (η) provided by WT-A with $\beta = 74°$ were respectively, 17.7%, 30.6% and 7.8% higher than those in the tube with WT-A with the same angle of attack, 20.8%, 53% and 4.9% higher than those in the tube with T-A, and 62%, 123% and 24% higher than those in the tube with TT.

The visualization of flow by smoke wire technique through a tube with various twisted tape inserts, relating to their performance mentioned above, is presented in Fig. 7a–c. Evidently, the TT induced the swirl flow as depicted in Fig. 7a. The swirl flow is a basic flow commonly generated by any twisted tape. The photograph of fluid flowed through the tube with WT in Fig. 7b demonstrated that fluid stream was split into two streams and then recombined behind the wing, producing double longitudinal vortices. The flow behavior caused by the alternate axis in the WT-A is presented in Fig. 7c. According to the tape geometry, two streams of fluid (in front and back sides of tape) were split into four streams around the alternate axis. Then they recombined into two streams with the stream collision behind the alternate point. The flow caused by the centre wing induced by the WT-A is similar to that induced by the WT as shown in Fig. 7b.

The sketch of flow phenomena induced by each twisted tape is demonstrated in Fig. 8, which are: (1) the TT generates only single direction-swirl flow, (2) the WT provides a single-direction-swirl flow caused by the twisted tape together with vortices induced by the wings (3) the T-A generates double-direction-swirl flows and (4) a WT-A offers combined actions from both WT and T-A twisted tape. As mentioned in Section 5.2, the wings on WT appear on one side/twist ratio (left or right) of the tape while those on WT-A appear on two sides (left and right) as a consequence of the generating the alternate axis. Therefore, the additional flow disturbance by the WT-A over that by the WT is not only caused by the periodic change of swirl direction but also a better distribution of the wings in the tube. As described above, the WT-A thus gives a better fluid mixing and superior heat transfer rates as well as thermal performance factor over the other twisted tapes do. The thermal performance factor above unity offered by the WT-A indicates the potential of the WT-A in view point of pumping energy saving.

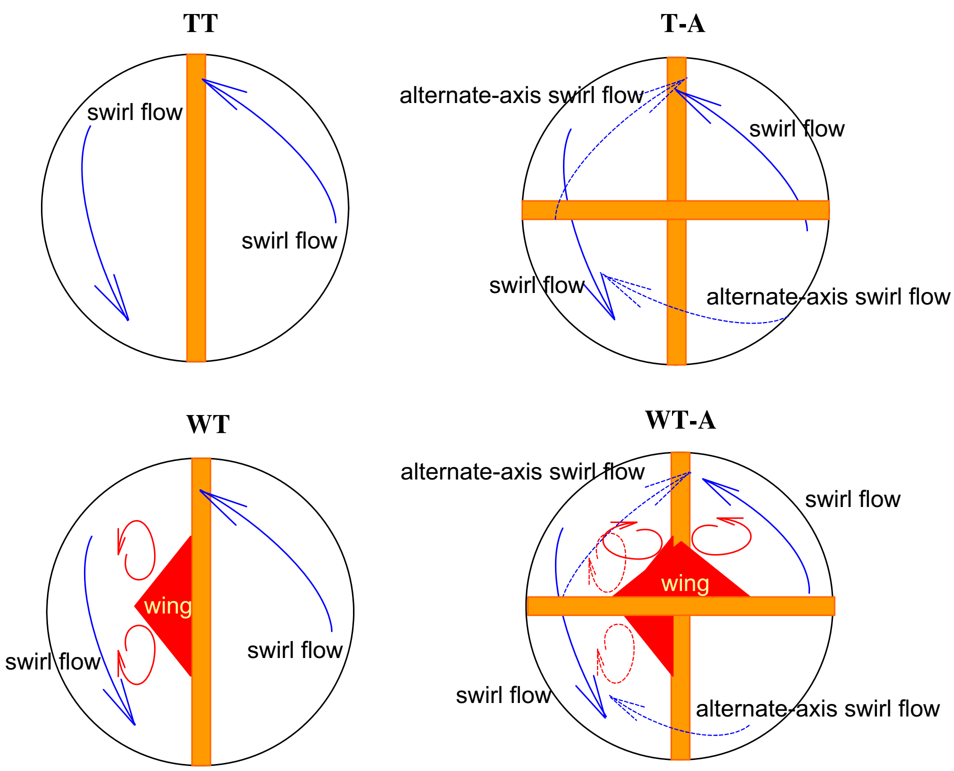

Fig. 8. Sketch of flow phenomena in the front view of the tube with various twisted tapes.

Table 2
Correlations of Nusselt number, friction factor and thermal performance factor.

Nusselt number	
Plain tube	$Nu = 0.025Re^{0.791}Pr^{0.4}$
Twisted tape with centre wings	$Nu = 0.232Re^{0.595}Pr^{0.4}(1+\tan\beta)^{0.202}$
Twisted tape with centre wings and alternate axes	$Nu = 0.385Re^{0.568}Pr^{0.4}(1+\tan\beta)^{0.129}$
Friction factor	
Plain tube	$f = 1.645Re^{-0.416}$
Twisted tape with centre wings	$f = 14.039Re^{-0.505}(1+\tan\beta)^{0.406}$
Twisted tape with centre wings and alternate axes	$f = 20.445Re^{-0.504}(1+\tan\beta)^{0.283}$
Thermal performance factor	
Twisted tape with centre wings	$\eta = 4.629Re^{-0.166}(1+\tan\beta)^{0.067}$
Twisted tape with centre wings and alternate axes	$\eta = 6.772Re^{-0.194}(1+\tan\beta)^{0.035}$

From the experimental results above, a least-squares regression was utilized to yield the equations summarized in Table 2. These equations are expressed in terms of Reynolds number (Re), Prandtl number (Pr), and angle of attack (β). Fig. 9a–c shows a comparison of experimental values of Nusselt number (Nu), friction factor (f) and thermal performance factor (η) with the predicted values, obtained from those equations. Apparently, the predicted data were in good agreement with the experimental data, within ±10% (WT) and ±6% (WT-A) for Nusselt number, ±10% (WT) and ±9% (WT-A) for friction factor and ±7% (WT) and ±4% (WT-A) for thermal performance, respectively.

6. Conclusions

The present study explored the effects of the wing and alternate axis of the modified twisted tapes on the heat transfer and fluid friction characteristics in a heat exchanger tube. For comparison, the twisted tapes used in the present work included one TT, one T-A, three different WTs and three different WTs-A. The angle of attack was varied at 43°, 53° and 74°. The main findings in the present work are concluded as follows.

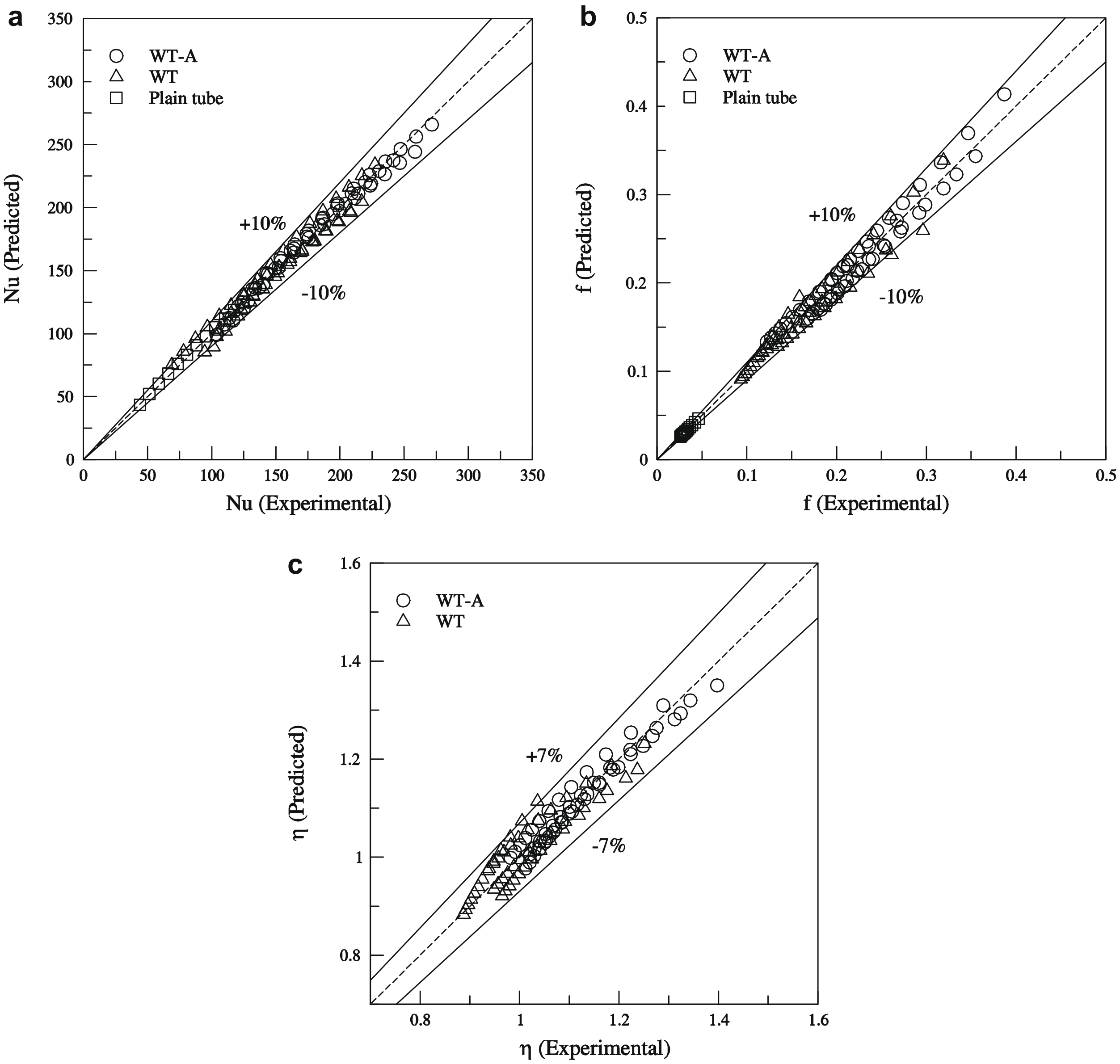

Fig. 9. Experimental data versus predicted data for: (a) Nusselt number and (b) friction factor and (c) thermal performance factor.

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1. The combined actions of the wing and alternate axis in the WT-A resulted in a better fluid mixing and thus heat transfer enhancement compared to those induced by wing alone (WT) or alternate axis alone (T-A). The results also revealed that Nusselt number (Nu) increased with the increasing angle of attack.
2. The WT-A generated higher pressure drop within the tube than WT, T-A and also TT, due to the flow disturbance caused by both the wing and the alternate axis and pressure drop became larger as the angle of attack increased.
3. The Nusselt number ratios (Nu_t/Nu_p) were in the range of 1.6–2.4 and 1.8–2.8 while the friction factor ratios (f_t/f_p) were in range of 4.4–6.9 and 6.48–8.4 for the WT and the WT-As, respectively.
4. The similar trend also found for the effects of the twisted tape form and the angle of attack on thermal performance factor (η) which is the consequential results from both heat transfer rate and pressure drop.
5. In the present range, the WT-A with the largest angler of attack ($\beta = 74°$) gave the highest Nusselt number (Nu), friction factor (f) as well as thermal performance factor (η). Mean values of these three parameters provided by the WT-A with ($\beta = 74°$ were respectively, 17.7%, 30.6% and 7.8% higher than those in the tube with WT, 20.8%, 53% and 4.9% higher than those in the tube with T-A, and 62%, 123% and 24% higher than those in the tube with TT.
6. The predicted data obtained from the developed correlations for Nusselt number, friction factor and thermal performance factor were in good agreement with the experimental data.

Acknowledgment

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Thermohydraulics of turbulent flow through a round tube by a peripherally-cut twisted tape with an alternate axis ☆

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ABSTRACT

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 has been experimentally investigated. Experiments were conducted using water as a testing fluid in a turbulent tube where the Reynolds number was varied from 5000 to 20,000. Peripherally-cut twisted tape (PT) and typical twisted tape (TT) were also tested in similar conditions, for comparison. Evidently, the heat transfer rates in the tube fitted with the PT-A, PT and TT are respectively enhanced up to 184%, 102% and 57% of that in the plain tube. In the present Reynolds number range tested, the PT-A, PT and TT offer the maximum thermal performances at constant pumping power of 1.25, 1.11 and 1.02, respectively. In addition, the correlations of the Nusselt number, friction factor and thermal performance were developed for the tube equipped with the PT-A in terms of peripherally-cut tape width ratio (w/W), Reynolds number (Re) and Prandtl number (Pr).

1. Introduction

Heat exchangers are widely used as the essential units in heat extraction and recovery systems in industries. The performance of heat exchanger can be substantially improved by a number of augmentation techniques. General goals of the augmentation are to reduce the size of heat exchanger required for specified heat duty, to upgrade the capacity of an existing heat exchanger, or to reduce the pumping power [1]. Among the different techniques which are effective at improving heat transfer rate in the tube-side of heat exchanger, the insertion of twisted tapes is prominent due to the ease of installation and low maintenance. In common, twisted tape imparts swirl to the essentially axial flow in the tube, leading to an excellent fluid mixing and thus efficient destruction of the boundary layer which is directly responsible for the improvement of heat transfer. In fact, the swirl promotes heat transfer rate with the resultant increase of friction loss in a tube. The latter escorts with the reduction of thermal performance factor which restricts the industrial applications of the twisted tape. Many previous research works have signified that the heat transfer enhancing performance of twisted tape is strongly dependent on their geometries. Optimization of the design is a challenging task to increase the rate of heat transfer while minimizing friction loss, which is beneficial for reducing the size of the heat exchanger and effecting energy savings [1–3].

Research works on heat transfer enhancement associated by swirl flow have been reported continuously for decades [4–13]. Dewan et al. [3] reviewed a vast number of research works on heat transfer enhancement by means of twisted tapes. After the review, the researches in this field over recent years have emphasized on modification of the twisted tape geometries for a superior heat transfer enhancement. Chang et al. [14] investigated heat transfer enhancement in the tubes fitted with serrated twisted tape. They concluded that the rib roughened twisted tape played the role of turbulator to enhance the turbulence intensity and induce the vortical flow cells behind each rib that was responsible for heat transfer augmentation up to 1.67 times of the heat transfer level in the tube fitted with typical twisted tape. However, this modified twisted tape caused a large pressure drop penalty and thus resulted in an unattractive thermal performance factor (mostly below unity). Chang et al. [15] also invented broken or spiky twisted tape for use in laminar and turbulent regimes. It was convinced that the shear layers downstream the spikes interacted with the swirling mainstream in the tube fitted with the twisted tape and thus considerably amplified the turbulent intensity as well as the vorticity. The consequences were that the heat transfer coefficients and thermal performance factors in the tube fitted with the broken twisted tape were, respectively, augmented up to 2.4 and 1.8 times of those in the tube fitted with the typical twisted tape. Rahimi et al. [16] reported experimental and computational fluid dynamics (CFD) investigation on the friction factor, Nusselt number and thermohydraulic performance of a tube equipped with the typical and three modified twisted tapes

Abbreviations: PT-A, peripherally-cut twisted tape with alternate axis; PT, peripherally-cut twisted tape; TT, typical twisted tape.
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Nomenclature

A	heat transfer surface area, m^2
C_p	specific heat of fluid, $J\ kg^{-1}\ K^{-1}$
D	inner diameter of the test tube, m
d	depth of peripheral cut, m
f	friction factor $= \Delta P/((L/D)(\rho U^2/2))$
h	heat transfer coefficient, $W\ m^{-2}\ K^{-1}$
I	current, A
k	thermal conductivity of fluid, $W\ m^{-1}\ K^{-1}$
L	length of the test section, m
M	mass flow rate, $kg\ s^{-1}$
Nu	Nusselt number $= hD/k$
P	pressure of flow in stationary tube, Pa
ΔP	pressure drop, Pa
Pr	Prandtl number $= \mu C_p/k$
Q	heat transfer rate, W
Re	Reynolds number $= \rho UD/\mu$
t	thickness of the test tube, m
T	temperature, K
$\tilde{T}$	mean temperature, K
U	mean axial flow velocity, $m\ s^{-1}$
V	voltage, V
w	width of peripheral cut, m
W	width of twisted tape, m
y	twist length, m

Greek symbols

ρ	fluid density, $kg\ m^{-3}$
δ	thickness of twisted tape, m
μ	fluid dynamic viscosity, $kg\ s^{-1}\ m^{-1}$
η	thermal performance factor

Subscripts

b	bulk
c	convection
i	inlet
o	outlet
p	plain
s	surface
t	turbulator
w	water

(perforated, notched, and jagged twisted tapes). Among these three styles, the jagged twisted tape offered maximum Nusselt number and thermal performance factor, which were as high as 31% and 22% over those provided by typical twisted tape. They pointed out that the superior heat transfer enhancement by the jagged twisted tape over the typical one was due to the higher turbulence intensity of the fluid close to the tube wall. Eiamsa-ard et al. [17] performed the effect of twin twisted tapes with counter/co-swirling flow on the heat transfer, friction factor and thermal performance characteristics. Eiamsa-ard et al. [18] also presented the effect of the twisted tape with alternate axis on the heat transfer enhancement. As found, the modified tape gave higher heat transfer rate and also thermal performance factor than a typical tape. This was attributed to the additional flow disturbance by strong collision of the recombined streams behind the alternate point. Recently, Eiamsa-ard et al. [19] comprehensively investigated on heat transfer enhancement by means of peripherally-cut twisted tapes (PT) with different tape depth ratios and tape width ratios. The results demonstrated that the Nusselt number, friction factor as well as thermal performance factor associated with the PT were found to be increased with increasing tape depth ratio (d/W) and decreasing tape width ratio (w/W). At the optimum depth and width ratio, the PT offered the maximum performance factors of 1.29 (turbulent regime) and 4.88 (laminar regime).

According to the above literature, it can be observed that the modified twisted tapes with small spaces appeared on the tapes, such as the broken (or spiky), peripherally-cut, jagged and alternate axis twisted tapes, hold a great promise for both enhancing the heat transfer rate and thermal performance factor. The reason behind a great thermal performance factor is that those small spaces bring friction loss in the system to the satisfactory level. This concept and our success in heat transfer enhancement by the uses of the peripheral cut and alternate axis twisted tapes [18,19] motivated the present research to design a new twisted tape, called a peripherally-cut twisted tape with alternate axis (PT-A). The combined actions from both peripheral cut and alternate axis of the PT-A were anticipated for a better performance over each action alone. For comparison, the experiments using a peripherally-cut twisted tape (PT), a typical twisted tape (TT) and also a plain tube were also conducted.

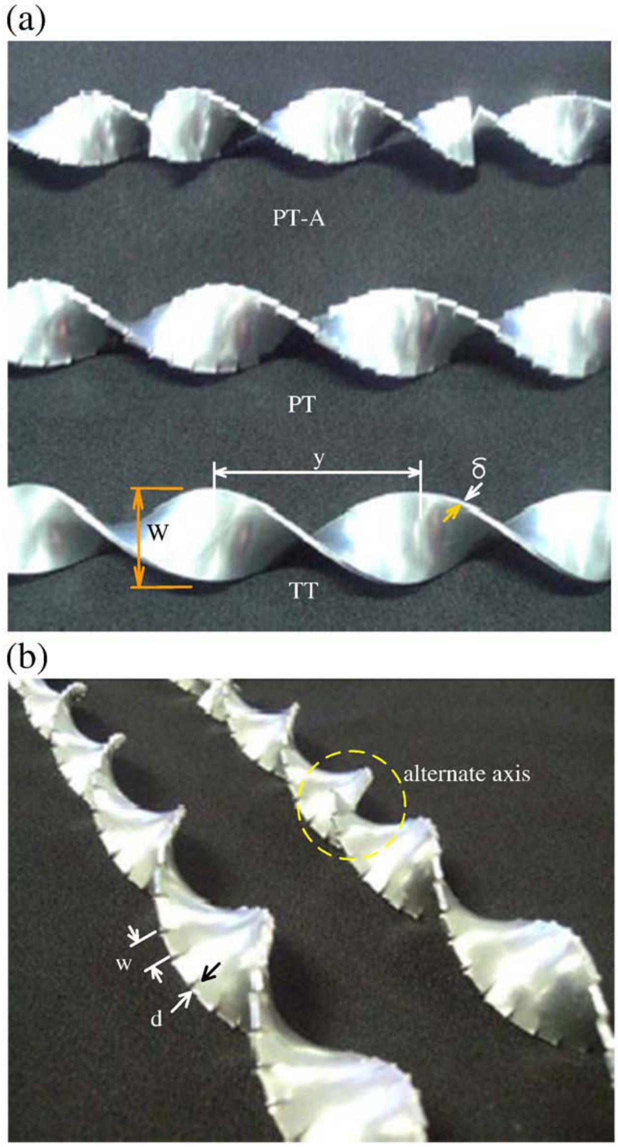

Fig. 1. Photographs and geometries of the various twisted tapes (TT, PT and PT-A) that are used in the present work: (a) font view and (b) isometric view.

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2. Peripherally-cut twisted tape with alternate axis (PT-A)

All twisted tapes used in the present work are made from an aluminum strip with thickness (δ) of 0.8 mm, width (W) of 19 mm and length (L) of 1000 mm. Initially, the typical twisted tape (TT) was fabricated by twisting the strip at constant twist ratio (y/W) of 3.0, where y is twist length per 180°. The mentioned twist ratio was selected with regard to the best thermal performance found in the previous study [14]. Then, three peripherally-cut twisted tapes with alternate axes (PTs-A) were formulated by cutting along the peripheral lines of the TTs with three different peripherally-cut width ratios ($w=2$ mm ($w/W=0.11$), 4 mm ($w/W=0.22$) and 6 mm ($w/W=0.33$)) while the peripherally-cut depth length was kept constant at $d=2$ mm ($d/W=0.11$). To form the alternate axis, the tape was cut on both sides at every twist length (180°). At each pair of the cuts, both sides of the tape were subsequently twisted simultaneously to angle difference of 90°. Three PTs with three different peripherally-cut width ratios ($w/W=0.11$, 0.22 and 0.33) were also prepared for comparative tests. The appearance of the tapes used in the present work is shown in Fig. 1. In the experiment, a twisted tape was equipped with the copper tube where water at temperature of around 28 °C was fed into the test section as the working fluid. More details of the experimental set-up can be found elsewhere [19] and Fig. 2.

3. Data reduction

In the experiment, the electrical power input (Q_{VI}) is supplied to the tube wall via heating wire:

$$Q_{VI} = V \times I \tag{1}$$

Due to heat loss from the test section to surrounding, the net heat transfer rate from the inner tube surface to working fluid, water (Q_w) at the steady-state condition can be expressed as

$$Q_w = Q_{VI} - Q_{loss} = MC_{p,w}(T_o - T_i) \tag{2}$$

where Q_{loss} is the rate of heat loss and $M=\rho AU$ is the mass flow rate of water through the test tube. According to heat balance at the steady-state flow condition, Q_{loss} was found in between 3 and 5% of the input electrical power (Q_{VI}).

The heat transfer from heating wire to water is mainly via convection. Therefore, it is assumed that the rate heat of gained by water is equal to the rate of convective heat transfer (Q_c) as

$$Q_c = Q_w \tag{3}$$

The convective heat transfer is defined in terms of the nominal inside surface area as

$$Q_c = hA\left(\tilde{T}_s - T_b\right) \tag{4}$$

where T_b is the bulk flow temperature and can be calculated from $T_b = (T_i + T_o)/2$. $\tilde{T}_s$ is the mean inner tube wall temperature of the test section which is computed by taking the average temperatures of 15 axial stations

$$\tilde{T}_s = \Sigma(T_{s1} + T_{s2} + \ldots T_{s15}) / 15 \tag{5}$$

The local heat transfer coefficient h is then evaluated from the following equation:

$$h = MC_{p,w}(T_o - T_i) / \left(\tilde{T}_s - T_b\right) \tag{6}$$

It should be mentioned that the water thermo physical properties are evaluated at the bulk flow temperature (T_b). The heat transfer rate inside the test tube is reported in terms of mean Nusselt number which can be written as

$$Nu = hD / k \tag{7}$$

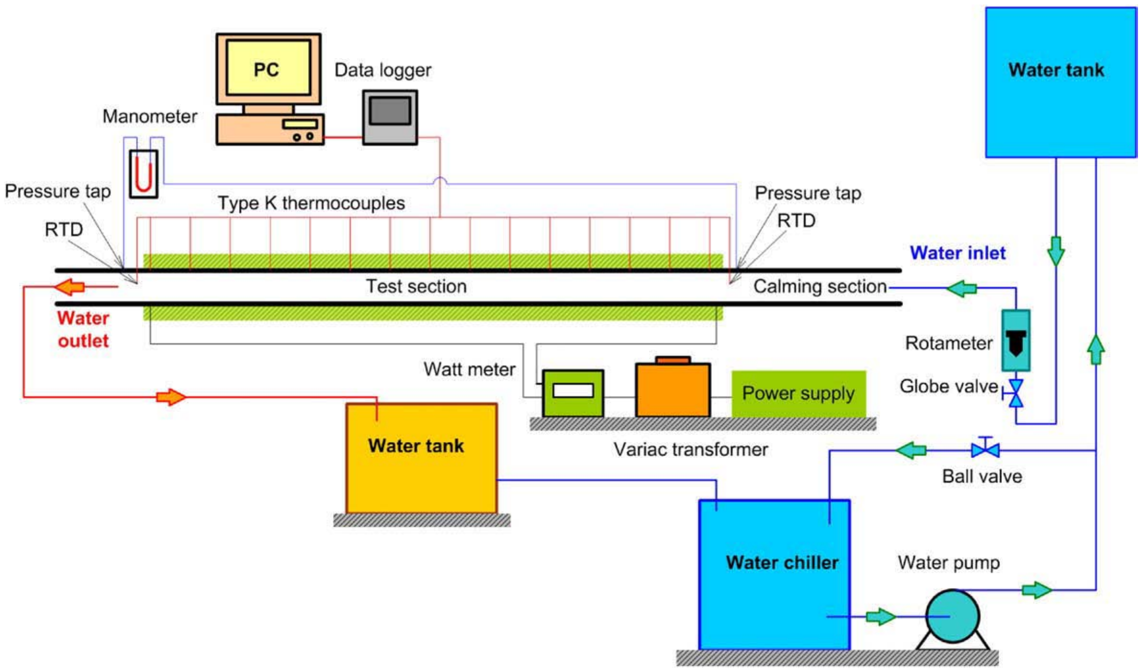

Fig. 2. A schematic diagram of heat transfer experimental facility.

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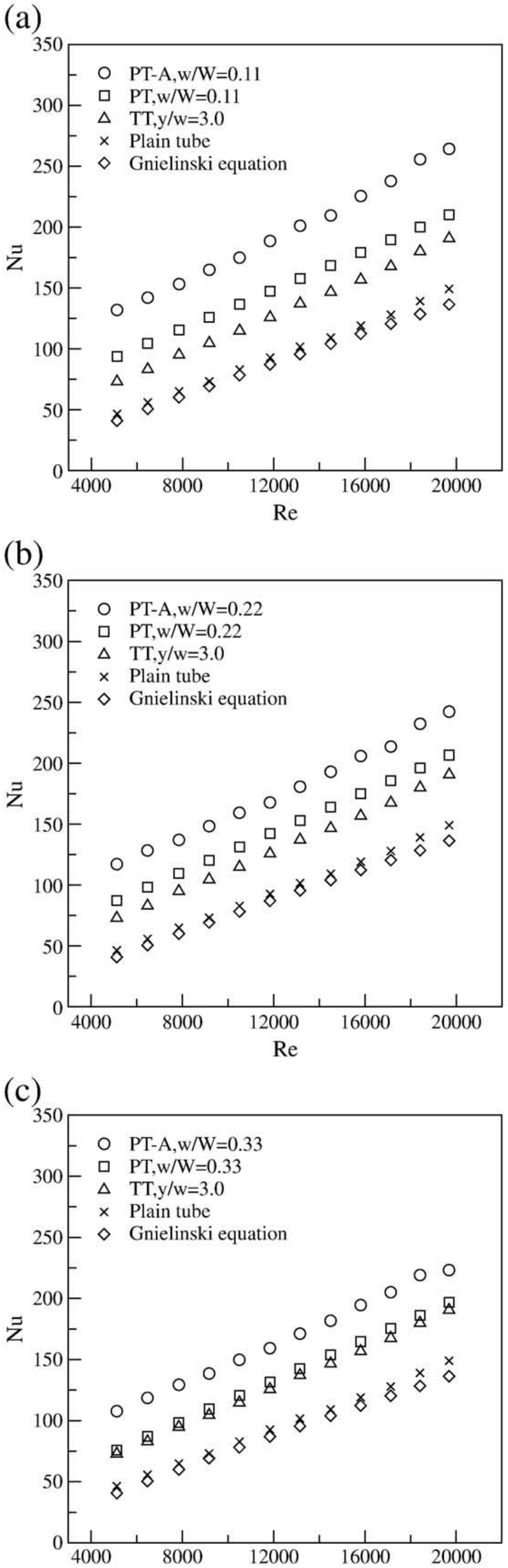

Fig. 3. Effect of twisted tapes on the heat transfer enhancement: (a) $w/W=0.11$, (b) $w/W=0.22$ and (c) $w/W=0.33$.

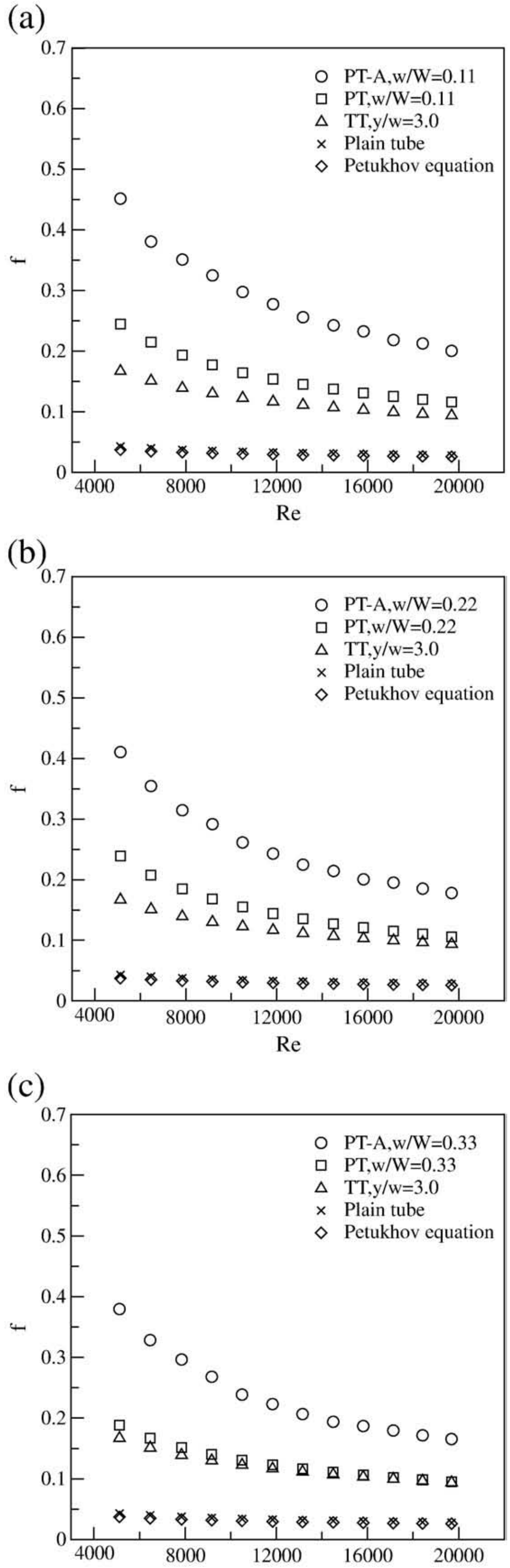

Fig. 4. Effect of twisted tapes on the friction factor: (a) $w/W=0.11$, (b) $w/W=0.22$ and (c) $w/W=0.33$.

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Another important parameter characterizing the fluid flow through the tube is the friction factor:

$$f = 2(\Delta PD) / \left(U^2 L\right) \tag{8}$$

where ΔP is the pressure drop across the test section evaluated under an isothermal flow condition and U is the mean flow velocity of the flowing fluid. Reynolds number stated in the present work is based on the average flow inlet velocity and the tube inlet diameter as

$$\mathrm{Re} = \rho UD / \mu \tag{9}$$

Finally, thermal performance factor at constant pumping power which indicates the potential of twisted tape for the industrial applications, is calculated using the following equation.

$$\eta = \left(Nu_\mathrm{t} / Nu_\mathrm{p}\right) \Big/ \left(f_\mathrm{t} / f_\mathrm{p}\right)^{1/3} \tag{10}$$

where index t denotes turbulator and index p denotes plain tube.

4. Experimental results

4.1. Validation test

Experiments began with the use of the plain tube, in order to verify the facility reliability. Experimental results including Nusselt number and friction factor have been compared with those from open literature. The reference correlations used for comparison recommended by Gnielinski (Nusselt number) and Petukhov (friction factor) [20] are given below.

$$Nu = \frac{\left(\frac{f}{8}\right)(\mathrm{Re}-1000)\mathrm{Pr}}{1 + \left[12.7\left(\frac{f}{8}\right)^{0.5}\left(\mathrm{Pr}^{2/3}-1\right)\right]} \tag{11}$$

$$f = (0.79\ln\mathrm{Re}-1.64)^{-2} \tag{12}$$

Figs. 3a and 4a show that the present results over the targeting Reynolds number are in satisfactory agreement with error limits of $\pm$7.6% for Nusselt number and $\pm$10.4% for friction factors. The deviations are attributed to the instrument uncertainties. It also can be observed that the Nusselt number increases almost linearly with the increasing Reynolds number and this is due to the rise of heat convection within the tube. In addition, the correlations for the present experimental results of the Nusselt number and friction factor for the plain tube are developed as follows:

$$Nu = 0.0141\mathrm{Re}^{0.864}\,\mathrm{Pr}^{0.4} \tag{13}$$

$$f = 0.68\mathrm{Re}^{-0.324}. \tag{14}$$

According to the above correlations, the mean deviations of predicted data from the actual Nusselt number from Eq. (13) and friction factor from Eq. (14) are 4% and 2%, respectively, as seen in Figs. 5a and 6a. Note that calculations are performed with the Prandtl numbers ranging from 5 to 5.5, depending on mean bulk temperatures (T_b).

4.2. Heat transfer and friction factor results

The results of heat transfer enhancement by the peripherally-cut twisted tape with alternate axis (PT-A) along with those by the typical twisted tape (TT) and the peripherally-cut twisted tape (PT) with respect to that of the plain tube are demonstrated in Fig. 3(a–c). Obviously, under similar conditions, the PT-A consistently provides greater heat transfer rates than the other tapes. Regarding to the data comparison, the heat transfer rates in the tube equipped with the PT-A are 13 to 38%, 17 to 81% and 50 to 184% greater than those in the tube fitted with the PT, TT and plain tube alone. This can be attributed to the combined effects of three actions by the PT-A including (1) a common swirling flow by the twisted tape (2) a periodic change of swirl direction by an alternate axis which leads to a strong collision of the recombined streams in the rear of each alternate axis and (3) a high turbulence intensity of fluid in the vicinity of the tube wall generated by the peripheral cut along the edge of the twisted tape. The observation of heat transfer rates in the tube fitted with PTs-A and PTs relative to those in the plain tube, indicates that the effects of the alternate axis and peripheral cut on heat transfer augmentation seems to be comparable.

The effect of the PT-A, PT as well as TT on the friction factor is depicted in Fig. 4(a–c). Trend found for all cases is that friction factor increases with decreasing Reynolds number. At the given Reynolds number, friction in the tube with twisted tape is always higher than that in the tube without the tape because of the swirling flow or turbulent

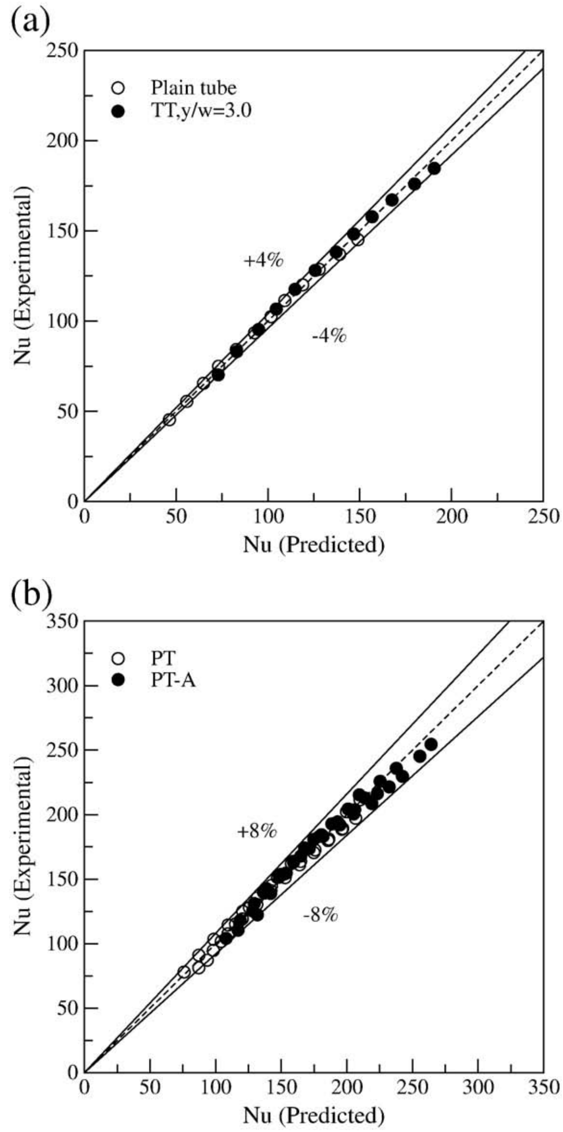

Fig. 5. Comparison of experimental and predicted values of Nusselt number for (a) plain tube and tube with TT and (b) tube with PT and PT-A.

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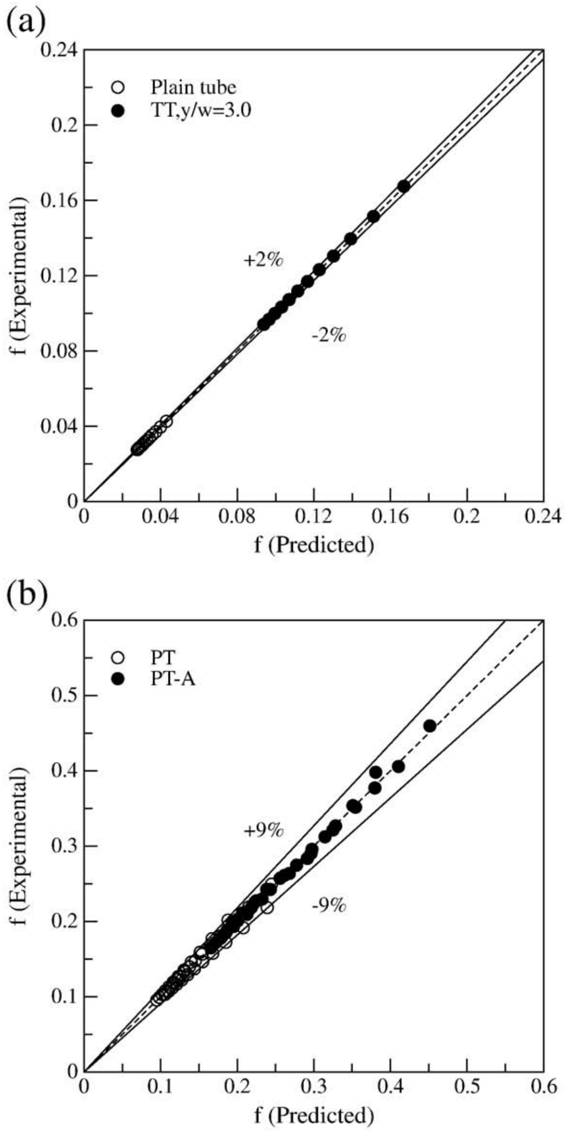

Fig. 6. Comparison of experimental and predicted values of friction factor for (a) plain tube and tube with TT and (b) tube with PT and PT-A.

flow generated by the twisted tape leading to the dissipation of dynamic pressure of the fluid at high viscosity loss near the tube wall and the interaction of the pressure forces with inertial forces in the boundary layer. The change of the flow pattern caused by the alternate axis of the PT-A leads to the increase of the flow resistance with higher pressure loss than those induced by the other tape inserts (TT and PT). Friction factor in the tube with the PT-A is increased up to 1.7, 2.1 and 7.7 times compared to those in the tube with PT, TT and plain tube, respectively. The Nusselt number and friction factor in the tube fitted with PT-A, PT and TT obtained in the present work are subjected to development of the correlations in terms of Reynolds number (Re), Prantl number (Pr) and peripherally-cut width ratio (in cases of PT-A and PT). The resultant equations are shown below.

For the tube fitted with the PT-A:

$$Nu = 0.422\mathrm{Re}^{0.544}\,\mathrm{Pr}^{0.4}\left(\frac{w}{W}\right)^{-0.148} \tag{15}$$

$$f = 59.08\mathrm{Re}^{-0.615}\left(\frac{w}{W}\right)^{-0.18} \tag{16}$$

For the tube fitted with the PT:

$$Nu = 0.126\mathrm{Re}^{0.658}\mathrm{Pr}^{0.4}\left(\frac{w}{W}\right)^{-0.101} \tag{17}$$

$$f = 18.82\mathrm{Re}^{-0.556}\left(\frac{w}{W}\right)^{-0.193} \tag{18}$$

For the tube fitted with the TT:

$$Nu = 0.076\mathrm{Re}^{0.718}\mathrm{Pr}^{0.4} \tag{19}$$

$$f = 6.42\mathrm{Re}^{-0.428} \tag{20}$$

The mean deviations of the predicted Nusselt number and friction factor from the actual data are 8% and 4% for the PT-A, 7% and 9% for the PT, and 4% and 2% for the TT as demonstrated in Figs. 5(a–b) to 6(a–b).

4.3. Thermal performance result

In the present work, a thermal performance factor is evaluated since the factor is an important parameter indicating the potential of a twisted tape for practical applications. The thermal evaluation is considered under constant pumping power for each twisted tape with respect to the case without twisted tape (plain tube). The thermal performance factors for PT-A, PT and TT are calculated via Eq. (10) and presented in Fig. 7(a–c). Apparently, a thermal performance factor decreases with increasing Reynolds number for all tape inserts. A larger pressure loss at a higher Reynolds number is responsible for the mentioned result. The obtained results showed that thermal performance factors are varied between 0.86 and 1.25 for PT-A, 0.88 and 1.11 for PT and, 0.87 and 1.02 for TT. Comparatively, the tube fitted with the PT-A gives about 5.6% and 7% higher mean thermal performance factors over the PT and TT, respectively. This suggests that the geometry of PT-A is more appropriate for practical use than the others in the view point of energy as well as operating cost savings. In addition, the correlations for thermal performance factors are developed based on experimental results over an investigated range of Reynolds number and width ratio (w/W). The predicted data show the maximum deviations of $\pm3\%$ for tube with PT-A, $\pm3\%$ for tube with PT and $\pm2\%$ for tube with TT, as seen in Fig. 8.

For the tube fitted with the PT-A:

$$\eta = 5.92\mathrm{Re}^{-0.205}\left(\frac{w}{W}\right)^{-0.082} \tag{21}$$

For the tube fitted with the PT:

$$\eta = 3.37\mathrm{Re}^{-0.14}\left(\frac{w}{W}\right)^{-0.028} \tag{22}$$

For the tube fitted with the TT:

$$\eta = 2.84\mathrm{Re}^{-0.12} \tag{23}$$

5. Conclusions

The effects of a peripherally-cut twisted tape with alternate axis (PT-A) on the heat transfer, flow friction and thermal performance factor characteristics were experimentally studied in a fully developed turbulent flow at Reynolds number ranges between 5000 and 20,000 using water as a testing fluid. The results achieved in the present work can be summarized as follows:

- Under the similar conditions, heat transfer rate, friction factor as well as thermal performance in a tube fitted with PT-A are consistently higher than those in the tube equipped with PT, TT and also in the plain tube.

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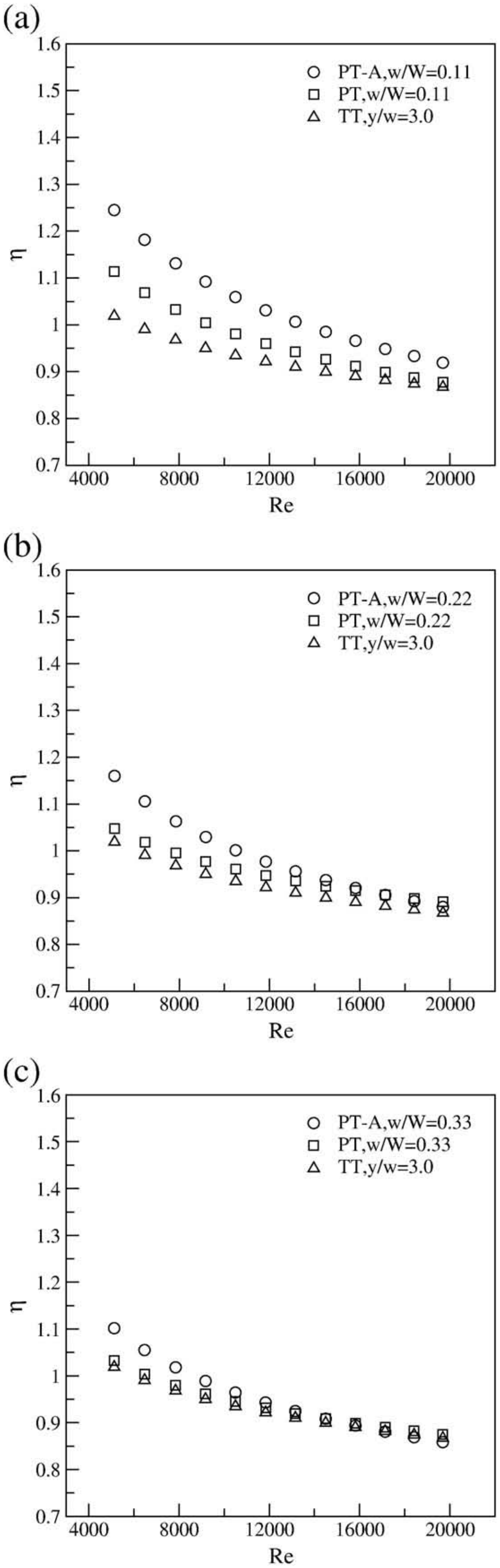

Fig. 7. Effect of twisted tapes on the thermal performance factor: (a) $w/W = 0.11$, (b) $w/W = 0.22$ and (c) $w/W = 0.33$.

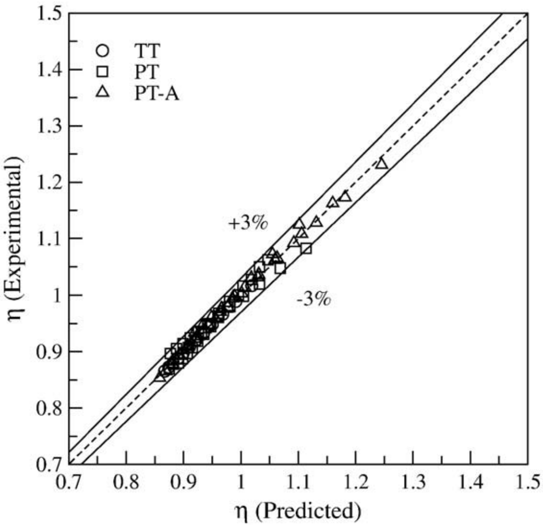

Fig. 8. Comparison of experimental and predicted values of thermal performance factor.

- The combined actions induced by the PT-A are responsible for the increases of heat transfer rate and friction factor of around 50 to 184% and 6 to 11 times, respectively compared to those in the plain tube.
- At constant pumping power, the thermal performance factors obtained in the present study are as high as 1.25 for PT-A, 1.11 for PT and 1.02 for TT. Over the range studied, the maximum thermal performance factor for each twisted tape insert is achieved at the lowest values of Reynolds number (Re = 5000).

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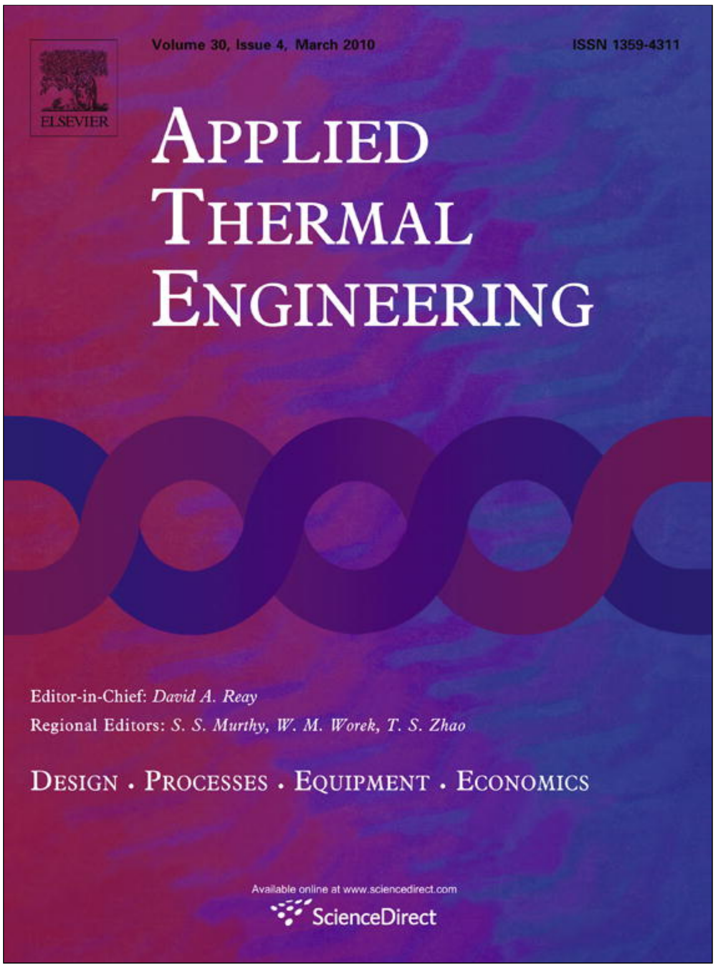
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Heat transfer enhancement in a tube using delta-winglet twisted tape inserts

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ABSTRACT

Heat transfer, flow friction and thermal performance factor characteristics in a tube fitted with delta-winglet twisted tape, using water as working fluid are investigated experimentally. Influences of the oblique delta-winglet twisted tape (*O-DWT*) and straight delta-winglet twisted tape (*S-DWT*) arrangements are also described. The experiments are conducted using the tapes with three twist ratios ($y/w = 3$, 4 and 5) and three depth of wing cut ratios ($DR = d/w = 0.11$, 0.21 and 0.32) over a Reynolds number range of 3000–27,000 in a uniform wall heat flux tube. The obtained results show that mean Nusselt number and mean friction factor in the tube with the delta-winglet twisted tape increase with decreasing twisted ratio (y/w) and increasing depth of wing cut ratio (DR). It is also observed that the *O-DWT* is more effective turbulator giving higher heat transfer coefficient than the *S-DWT*. Over the range considered, Nusselt number, friction factor and thermal performance factor in a tube with the *O-DWT* are, respectively, 1.04–1.64, 1.09–1.95, and 1.05–1.13 times of those in the tube with typical twisted tape (*TT*). Empirical correlations for predicting Nusselt number and friction factor have been employed. The predicted data are within ±10% for Nusselt number and ±10% for friction factor.

1. Introduction

High performance heat transfer system is of great importance in many industrial applications. Therefore, the heat transfer enhancement techniques are widely applied in heat exchangers, in order to improve heat transfer coefficient [1–4]. Passive heat transfer augmentation is method to enhance heat transfer without external power. Among the techniques used, insertion of twisted tape in a circular tube is one of the most effective approaches. The inserted twisted tape generates swirling flow and increases turbulence intensity which is major influencing factors for heat transfer enhancement. In fact, using twisted tape increases both desirable heat transfer rate and undesirable friction loss (pressure drop). An appropriate twisted tape modification is a challenge task as a proper design of twisted tape is a main key for heat transfer enhancement at a reasonable friction loss.

Numerous investigations using twisted tape with different geometrics for heat transfer enhancement have been released. Dewan et al. [5] overviewed vast numbers of research works of heat transfer enhancement using this type of the insert. Here, some of the literatures after the review paper are mentioned as follows. In the earlier papers of our research group, heat transfer and friction factor characteristics of a flow through the circular tube fitted with twisted tapes with various free space ratios [6], a conical-ring combined with a twisted tape insert [7], and helical tapes with and without a rod [8], were investigated. The results revealed that both heat transfer coefficient and friction factor increased with a decrease of the free space ratio [6], a tube fitted with the conical-ring and a twisted tape gave higher Nusselt number than that fitted with a simple conical-ring [7], and helical tape with a rod provided a higher heat transfer rate in comparison with the helical tape without a rod [8]. Promvonge [9] used wire coil in conjunction with twisted tape for heat transfer augmentation. The finding was that the combination of wire coil and twisted tape led to a double increase in heat transfer over the use of wire coil/twisted tape alone. It was also observed that, the twisted tape and wire coil with smaller twist and coil pitch ratio gave higher heat transfer rate than those with larger twist and coil pitch ratios. Chang et al. [10,11] investigated the heat transfer enhancement in the tubes fitted with serrated twisted tape and broken twisted tape at different twist ratios. The results of both papers showed similar trends, that the local Nusselt number and Fanning friction factor increased as the twist ratio decreased. Recently, Rahimi et al. [12] reported experimental and computational fluid dynamics (CFD) investigation on the friction factor, Nusselt number and thermal–hydraulic performance of a tube equipped with the typical and three modified twisted tapes (perforated, notched, and jagged twisted tape). Their results demonstrated that the Nusselt number

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Nomenclature

A	heat transfer surface area, m^2
C_p	specific heat of fluid, $J\ kg^{-1}\ K^{-1}$
D	inside diameter of the test tube, m
d	depth of wing cut, m
f	friction factor = $\Delta P/((L/D)(\rho U^2/2))$
h	heat transfer coefficient, $W\ m^{-2}\ K^{-1}$
I	current, A
k	thermal conductivity of fluid, $W\ m^{-1}\ K^{-1}$
L	length of the test section, m
M	mass flow rate, $kg\ s^{-1}$
Nu	Nusselt number = hD/k
P	pressure of flow in stationary tube, Pa
ΔP	pressure drop, Pa
Pr	Prandtl number = $\mu C_p/k$
Q	heat transfer rate, W
Re	Reynolds number = $\rho UD/\mu$
t	thickness of the test tube, m
T	temperature, K
$\tilde{T}$	mean temperature, K
U	average axial flow velocity, $m\ s^{-1}$
V	voltage, V
$\dot{V}$	volumetric flow rate, $m^{-3}\ s$
w	tape width, m
y	twisted tape pitch, m

Greek symbols

ρ	fluid density, $kg\ m^{-3}$
δ	twisted tape thickness, m
μ	fluid dynamic viscosity, $kg\ s^{-1}\ m^{-1}$
η	thermal performance factor

Subscripts

b	bulk
c	convection
i	inlet
o	outlet
p	plain
pp	pumping power
s	surface
t	turbulator
w	water

Abbreviations

O-DWT	oblique delta-winglet twisted tape
S-DWT	straight delta-winglet twisted tape
TT	typical twisted tape

and thermal performance of the jagged insert were higher than those of the others.

In the current work, the newly designed *DWTs* are used as heat transfer enhancement devices. It is assumed that, the wing part can enhance degree of turbulence near wall tube and thus, heat transfer rate. The experiments are performed using the tape with three different twist ratios (y/w = 3, 4 and 5) and three depth of wing cut ratios ($DR = d/w$ = 0.11, 0.21 and 0.32) with a Reynolds number range between 3000 and 27,000, where water is used as the working fluid. The effects of the oblique delta-winglet twisted tape (*O-DWT*) and straight delta-winglet twisted tape (*S-DWT*) arrangements on the heat transfer rate, friction factor and thermal enhancement factor characteristics in a round tube are also reported in this paper.

2. Delta-winglet twisted tape

All tapes used in the present work are made of aluminum strip with 0.8 mm thickness (δ) and 19.5 mm width (w). Firstly, aluminum strip was twisted to produce a typical twisted tape. A tape was subsequently modified to obtain the *DWT* by cutting at the edge of the tape with oblique shape and straight shape to produce an oblique delta-winglet twisted tape (*O-DWT*) and a straight delta-winglet twisted tape (*S-DWT*), respectively. Then the outer part of the cut was arranged to 90° (*degree*) relatively to the inner part, forming delta-winglet shape. The length between the two cuts was set equally to the pitch length (y). The *DWTs* were prepared with three different twist ratios (y/w) = 3, 4 and 5, and three different depth of wing cut ratios ($DR = d/w$ = 0.11, 0.21 and 0.32). The geometrical configurations of the delta-winglet twisted tape (*DWT*) inserts are presented in Fig. 1.

3. Experimental details

3.1. Experimental setup

Experiments were carried out in an open loop rig as shown in Fig. 2, using water as working fluid. The heating test tube is made of copper with thickness of 1.5 mm, diameter of 19.5 mm and length of 1000 mm. Cold water was continuously supplied from a water chiller (0.5 m^3) to an overhead water tank (at 3 m elevation) by 0.5 hp centrifugal water pump. The overhead water tank had a volume of 0.5 m^3. The constant head water level in overhead water tank was controlled by overflow discharged through the ball valve to the ground water tank (0.5 m^3). The water main line was 25.4 mm in diameter and the dimension then decreased to 19.5 mm at calming section. Calming section made of copper tube with length of 1500 mm was used to eliminate the entrance effect. The rubber bellows were also used to reduce the flow pulsations and vibration effect from the centrifugal water pump. One end of heating test tube was attached with the calming section, while the other end was attached with the mixing section where three baffles were assembled to the tube at a position of 150 mm ahead of the test tube exit, for efficient mixing of outgoing fluid.

The test tube was heated by continually winding flexible electrical wire to provide a uniform heat flux boundary condition. The heating tube was wound with ceramic beads coated electrical SWG Nichrome heating wire. The terminals of the Nichrome wire were connected to the Varaic transformer, by which heat flux of the tube wall was varied by adjusting the voltage (2–200 V) and current (9–15 A). The electrical output power was controlled via a Variac transformer to obtain a uniform heat flux (UHF) along the entire length of the test section. The K-type thermocouple beads were tapped along the local tube wall for 15 stations for monitoring the temperature of the surface tube wall. The heating tube and thermocouple were covered with insulation to minimize heat loss to surrounding. Calibrated RTD PT 100 type temperature sensors were used to measure the inlet (before the test section) and outgoing water temperature (behind the mixing section) of the heating test tube. At the entrance of the test tube, the volume flow rate of water was measured using rotameter. The water flow rate was adjusted by the globe valve which was located upstream of the rotameter. The experiments were carried out at the Reynolds number ranged between 3000 and 27,000. For each test run, temperature, volumetric flow rate and pressure drop of the bulk air were recorded at steady state conditions, in which the inlet water

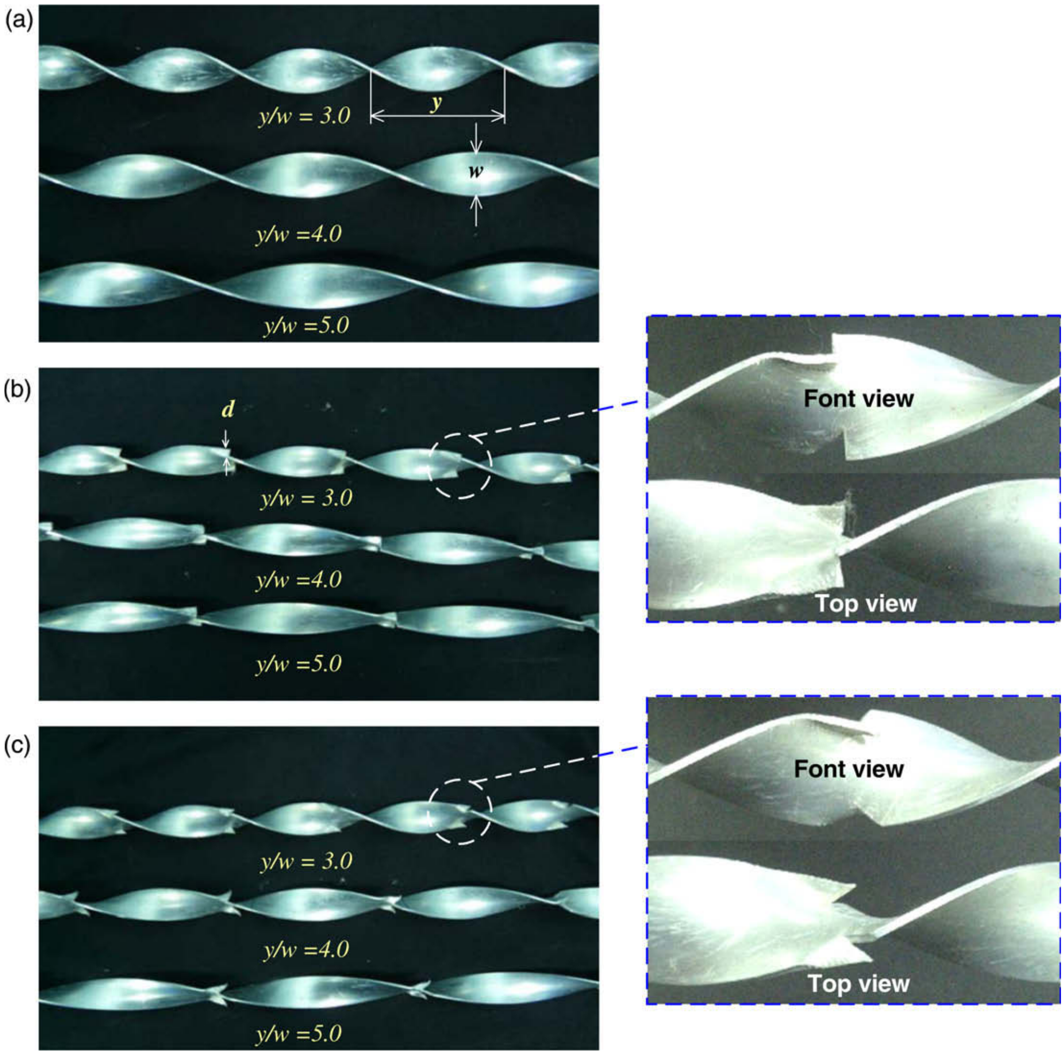

Fig. 1. Twisted tape vortex generator: (a) typical twisted tape (*TT*), (b) straight delta-winglet twisted tapes (*S-DWT*) and (c) oblique delta-winglet twisted tapes (*O-DWT*).

temperature was maintained at 27 °C and the volumetric flow rates were varied from 120 l/h to 1200 l/h.

3.2. Experimental procedure

In the experiments, water was pumped to the overhead water tank then discharged directly to the test tube. The heat flux at the test tube wall was set using Variac transformer and then, the system was allowed to reach steady state for each value of Reynolds number. The friction factor was calculated in term of pressure drop that measured by manometer under isothermal condition or without heating tube. In the experiment, the pressure taps were located around 50 mm upstream and 150 mm downstream from the test section. The length between both tapes was around 1200 mm. In the experiments, the tape was used as swirl generator to produce different swirl/turbulent intensity depends on the twist ratios ($y/w = 3$, 4 and 5), depth of wing cut ratios ($DR = d/w = 0.11$, 0.21 and 0.32) and Reynolds number. Effects of the oblique delta-winglet twisted tape (*O-DWT*) and straight delta-winglet twisted tape (*S-DWT*) arrangements are also examined.

4. Calculation of heat transfer and friction factor

The details of the calculation of heat transfer in term of Nusselt number and friction factor of the present study are described as follows. Heat transferred to the cold water is assumed to be equal to the heat loss from the test section which can be expressed as:

$$Q_w = Q_c \tag{1}$$

where

$$Q_w = MC_{p,w}(T_o - T_i) \tag{2}$$

and the heat balance between the heat input (Q_{VI}) and heat transfer rate of the water (Q_w) is less than 7%

$$(Q_{VI} - Q_w) \times 100\%/Q_{VI} < 7\% \tag{3}$$

The convection heat transfer from the test section can be written by

$$Q_c = hA(\tilde{T}_s - T_b) \tag{4}$$

where A is the internal surface of the tube wall (πDL) and T_b is the bulk/mean fluid (water) temperature ($T_b = (T_o + T_i)/2$) and $\tilde{T}_s$ is the mean wall temperature and evaluated at the outer wall surface of the test tube which can be written as

$$\tilde{T}_s = \sum T_s/15 \tag{5}$$

The mean wall temperature is calculated from 15 points of surface temperatures lined between the inlet and the exit of the test tube. The average heat transfer coefficient, h and the mean Nusselt number, Nu are estimated as follows:

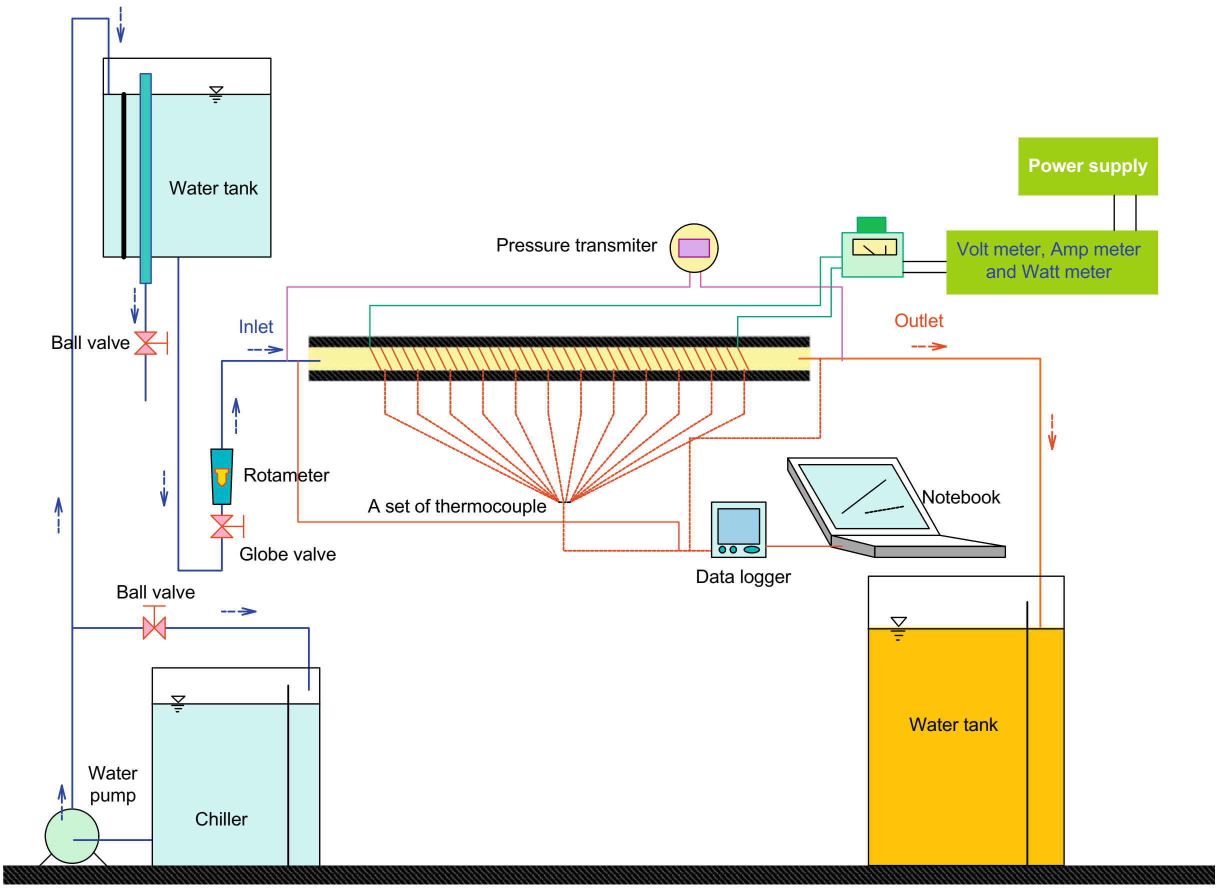

Fig. 2. Schematic diagram of the experimental heat transfer setup.

$$h = Q_w/A(\tilde{T}_s - T_b) \tag{6}$$

The heat transfer is measured by Nusselt number which can be obtained by

$$Nu = hD/k \tag{7}$$

The friction factor is calculated by pressure loss, Δp across the test length, L that defined as

$$f = \frac{\Delta P}{\left(\frac{L}{D}\right)\left(\rho \frac{U^2}{2}\right)} \tag{8}$$

where U is mean axial velocity of the test tube, ρ is the density of the working fluid and D is the inside diameter of the test tube. The situation of the flow can be defined from the Reynolds number that given by

$$Re = \rho UD/\mu \tag{9}$$

All of thermo-physical properties of the water are determined at the overall bulk fluid temperature (T_b). The uncertainties of the reduced data obtained experimentally are determined. Following the procedure based on ANSI/ASME [13] on reporting for the uncertainties in experimental measurements and results, the maximum uncertainty associated with the pressure, temperature, Reynolds number and Nusselt number are estimated to be ±4.8%, ±0.1%, ±8% and ±14%, respectively.

5. Experimental results and discussion

In this section, heat transfer rate, friction factor and thermal performance factor behaviors in the tubes fitted with the delta-winglet twisted tapes (oblique or straight delta-winglet twisted tape) for different twist ratios (y/w) and depth of wing cut ratios (DR) are reported. The results obtained by using plain tube and the tubes equipped with typical twisted tape are also presented for comparison.

5.1. Proof of plain tube and tube fitted with twisted tape

Heat transfer rate and flow friction characteristics in terms of Nusselt number and friction factor for the present plain tube are declared with standard correlations in Fig. 3a and b, for the proof of the plain tube. The data obtained from the experiments, shown in both figures, are in satisfactory agreement with the correlations. As seen in Fig. 3a, the Nusselt number data are within 7.4% error from Dittus and Boelter correlation ($Nu = 0.023Re^{0.8}Pr^{0.4}$) [14]. In Fig. 3b, the isothermal friction factors for the present plain tube are 12% higher than those obtained from the modified correlation of Blasius. In addition, the correlations of present experimental results for Nusselt number and friction factor are obtained as follows:

$$Nu = 0.028Re^{0.79}Pr^{0.4} \tag{10}$$

$$f = 0.448Re^{-0.275} \tag{11}$$

For the proof of the present typical twisted tapes, Nusselt number and friction factor of a tube fitted with the present typical twisted tapes are compared with the results obtained from the correlations by Maglik and Bergles [15] as demonstrated in Fig. 4a and b. Apparently, present results agree well with the available correlations within ±20% for Nusselt number and ±20% for friction factor.

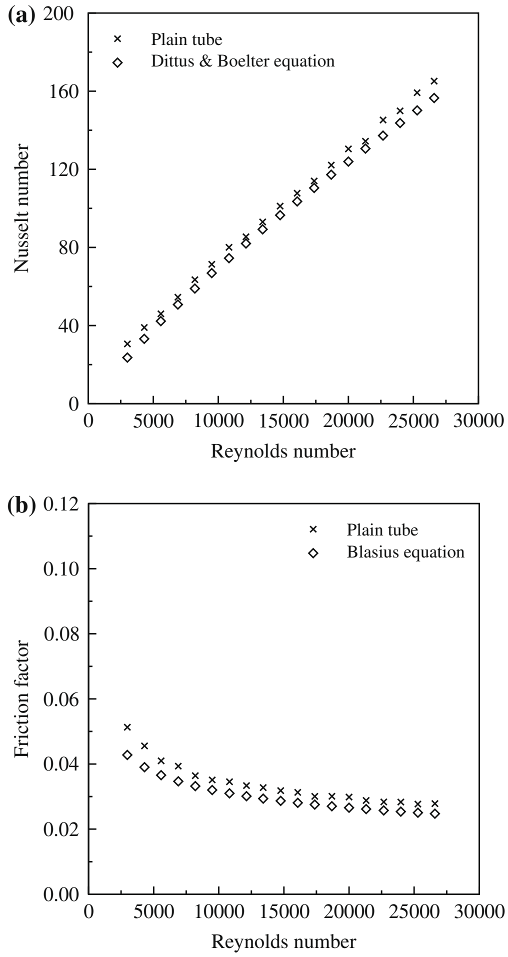

Fig. 3. Proof of the plain tube: (a) Nusselt number and (b) friction factor.

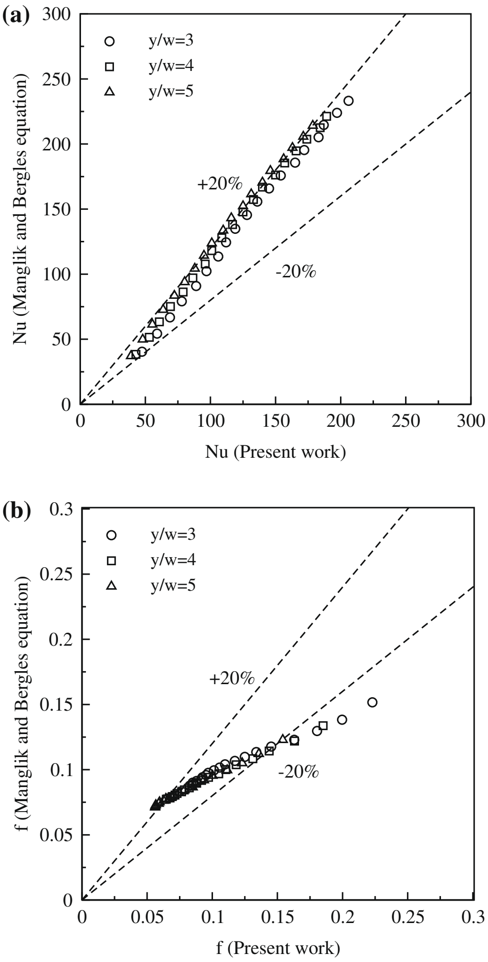

Fig. 4. Proof of the plain tube fitted with twisted tapes: (a) Nusselt number and (b) friction factor.

5.2. Effect of delta-winglet twisted tape (DWT)

Variation of Nusselt number with Reynolds number in the tube fitted with delta-winglet twisted tape (*DWT*), the tube fitted with typical twisted tape (*TT*) and also the plain tube are presented in Fig. 5a and b. For all experiments, Nusselt number increases with increasing Reynolds number. This is attributed to the increase of heat convection and also swirl flow intensity. Apparently, Nusselt numbers in the tube with both oblique and straight delta-winglet twisted tapes are higher than those in the plain tube and the tube with typical twisted tape insert over the considered Reynolds number range. Refers to the past investigations [16–19], typical delta-winglet tapes were applied as vortex generators to amplify turbulence intensities and produce secondary flows near the tube wall. Therefore, the high heat transfer rate in the tube with both oblique and straight delta-winglet twisted tapes (*O-DWT* and *S-DWT*) for the present work may be caused by the synergy effect of (1) vortex circulation together with secondary flow generated by the delta-winglet part and (2) main swirl flow produced by twisted tape. This effect results in superior heat transfer enhancement over that caused by the typical tape (*TT*) which induces only swirling flow. In the range of the present experiments, the Nusselt numbers for tube equipped with *DWT*, are respectively 1.1–2.55 and 1.02–1.64 times of those for plain tube and tube with typical twisted tape insert.

Friction factors are plotted against Reynolds numbers in Fig. 6a and b. Not surprisingly, the friction factors obtained from the tube with delta-winglet twisted tape (*DWT*) are considerably higher than those from the plain tube and also the tube with typical twisted tape insert (*TT*). This is a result of additional blockage and disturbance to the flowing stream generated by the delta-winglet part of *DWT* over those induced by typical twisted tape. Over the range considered, the friction factors for tube equipped with *DWT* are respectively 2.5–7.02 and 1.08–1.95 times of those for plain tube and tube with typical twisted tape insert.

5.3. Effect of twist ratio (y/w)

Fig. 5a and b shows variation of Nusselt number with Reynolds number in tube with oblique delta-winglet twisted tape (*O-DWT*) and straight delta-winglet twisted tape (*S-DWT*) for various twist ratios (y/w = 3, 4 and 5). At the given Reynolds number, the Nusselt number consistently increases with the decrease of twist ratio (y/w). This is due to the fact that, the tape with smaller twist ratio (y/w) induces stronger swirl/turbulent intensity, and also gives longer flowing path, leading to longer residence time and thus more efficient heat transfer compared to that with larger twist ratio (y/w). Mean Nusselt numbers for the *S-DWT* with twist ratio y/w of 3, 4 and 5 are respectively, 1.6, 1.4 and 1.23 times of that for the plain tube and 1.2, 1.16 and 1.1 times of that in the tube equipped with typical twisted tape (*TT*). With a similar trend, mean Nusselt numbers for the *O-DWT* with twist ratios y/w of 3, 4 and 5 are respectively, 1.66, 1.45 and 1.31 times of that for the plain tube and 1.25, 1.2 and 1.17 times of that for the tube with typical twisted tape insert. Depending on the Reynolds number, Nusselt

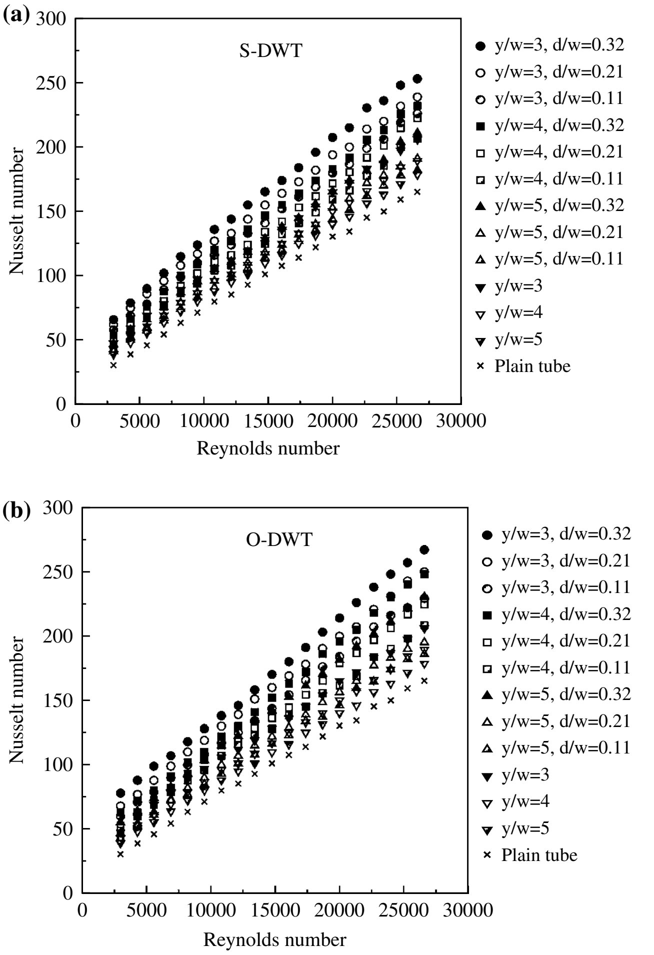

Fig. 5. Nusselt number versus Reynolds number for tubes with *DWT*, *TT*, and plain tube: (a) *S-DWT* and (b) *O-DWT*.

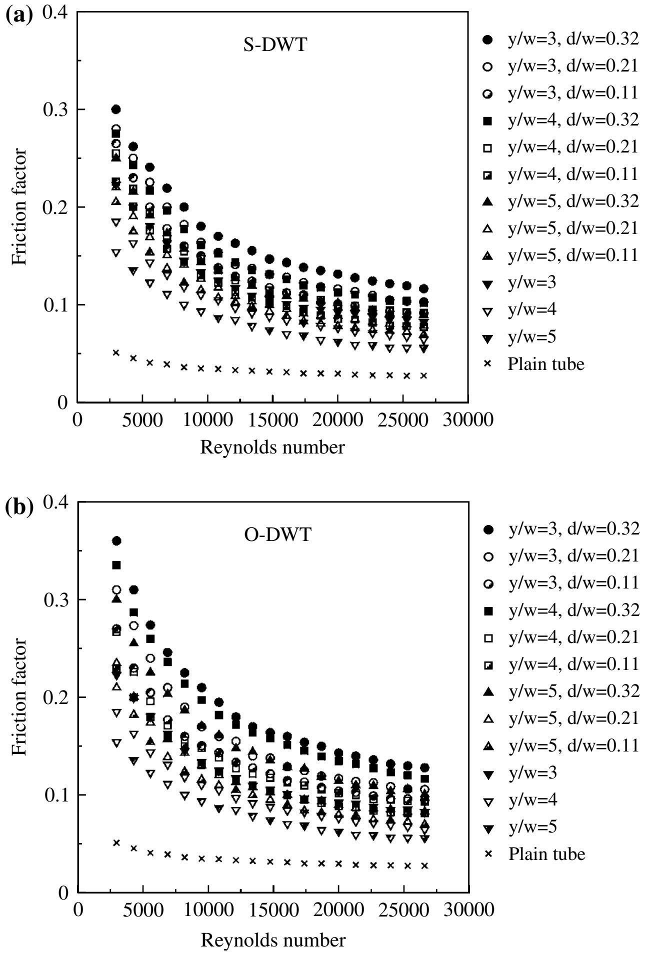

Fig. 6. Friction factor versus Reynolds number for tubes with *DWT*, *TT*, and plain tube: (a) *S-DWT* and (b) *O-DWT*.

numbers from using the *O-DWT* and *S-DWT* with twist ratio (y/w) of 3 are around 7–29% and 15.8–45.6% greater than those from the tapes with twist ratios (y/w) of 4 and 5, respectively.

Fig. 6a and b shows variation of friction factor with Reynolds number for oblique delta-winglet twisted tape (*O-DWT*) and straight delta-winglet twisted tape (*S-DWT*) at various twist ratios ($y/w = 3$, 4 and 5). As found, the friction factor increases with decreasing twist ratio (y/w). This is a consequence of the reasons mentioned for Nusselt number, in which twisted tape with shorter twist length provides longer flowing path, resulting in larger tangential contact between the flowing stream and tube surface. Over the range studied, mean friction factors for the *S-DWT* with twist ratios y/w of 3, 4 and 5 are respectively, 4.35, 3.86 and 3.4 times of that for the plain tube and 1.25, 1.34 and 1.4 times of that for the tube with typical twisted tape insert. For a tube with *O-DWT*, friction factors with twist ratios y/w of 3, 4 and 5 are respectively, 4.68, 4.18 and 3.67 times of that for the plain tube and 1.33, 1.45 and 1.52 times of that for the tube with typical twisted tape insert. In addition, friction factors for *O-DWT* and *S-DWT* with twist ratio (y/w) of 3 are approximately 3.5–22.8% and 17.9–37.8% over those for the tapes equipped with twist ratios (y/w) of 4 and 5, respectively.

5.4. Effect of depth of wing cut ratios (DR)

Effects of the of depth of wing cut ratios ($DR = d/w = 0.11$, 0.21 and 0.32) on the Nusselt number and friction factor are presented in Fig. 5a and b and Fig. 6a and b. Refer to Fig. 5a and b, for both tubes with *S-DWT* and *O-DWT*, Nusselt number increases with increasing wing cut ratios (*DR*). As the depth of wing cut increases, longer wing part protruded into flowing stream. Consequently, the flow is more seriously disturbed and degree of turbulence intensity is raised. This leads to more effective heat transfer enhancement. The results also show that mean Nusselt numbers for the *S-DWT* and *O-DWT* with the highest depth of wing cut ratios, $DR = 0.32$ are respectively, 1.25–2.16 and 1.35–2.55 times of that for the plain tube and 1.16–1.39 and 1.29–1.64 times of that for the tube with typical twisted tape. In addition, the average Nusselt number for employing the tapes with of depth of wing cut ratio (*DR*) of 0.32 is respectively, 17.3% and 9.7% higher than those for the tapes depth of wing cut ratios of 0.11 and 0.22.

Fig. 6a and b indicates that the tape insert with larger depth of wing cut ratio (*DR*) induces higher friction factor. This is caused by increasing extent of flow obstruction by the wing parts of the twisted tape. Mean friction factors for the *S-DWT* and *O-DWT* with the highest depth of wing cut ratio, $DR = 0.32$ are respectively, 3.22–5.85 and 3.62–7 times of that in the plain tube and 1.31–1.63 and 1.51–1.95 times of that in the tube with typical twisted tape. In addition, the average friction factor for employing the tape with depth of wing cut ratio, $DR = 0.11$ is found to be 10.9% and 25.2% lower than that for the tapes with $DR = 0.21$ and 0.32.

5.5. Effect of straight and oblique wings

Comparisons shown in Figs. 7 and 8 reveal that the oblique delta-winglet twisted tapes (*O-DWT*) give higher Nusselt number and

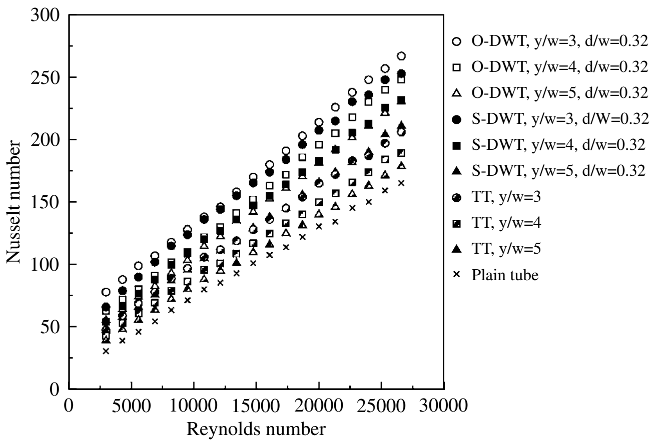

Fig. 7. Effect of straight and oblique wings on Nusselt number.

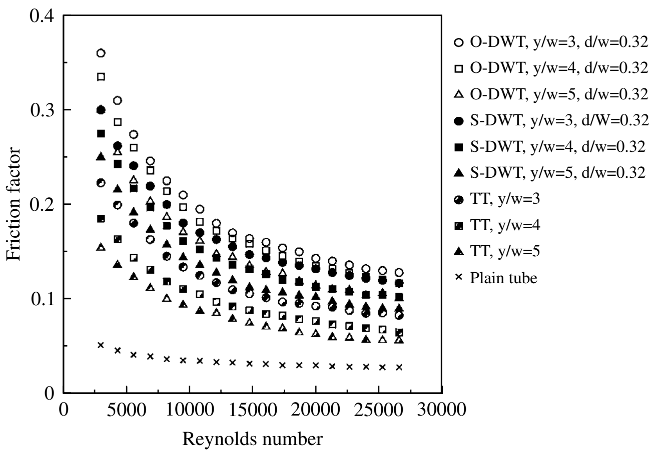

Fig. 8. Effect of straight and oblique wings on friction factor.

also friction factor than the straight delta-winglet twisted tapes (*S-DWT*), when other parameters are controlled. These results are directly related to geometries of both tape arrangements. As depicted in Fig. 1, with the same depth of wing cut, the *O-DWT* can be more effectively twisted, thus the tape possesses larger protruding part than the *S-DWT*. Consequently, greater heat transfer rate and also friction factor are obtained for the *O-DWT*. The calculated data show that the mean Nusselt number and friction factor for the *O-DWT* are respectively, 4.2% and 7.8% higher than those for the *S-DWT*.

5.6. Performance criteria

According to the previous studies [20–23], a comparison of heat transfer coefficients in plain tube (p) and tube fitted with turbulator (t) was made at the same pumping power since it is relevant to operation cost.

For constant pumping power:

$$(\dot{V}\Delta P)_p = (\dot{V}\Delta P)_t \tag{12}$$

Eq. (12) can be rewritten in terms of friction and Reynolds number as

$$(fRe^3)_p = (fRe^3)_t \tag{13}$$

Thermal performance factor (η) at equal pumping power is defined as ratio of the convective heat transfer coefficient of the tube with turbulator to that of the plain tube which can be expressed as:

$$\eta = \left.\frac{h_t}{h_p}\right|_{pp} \tag{14}$$

In general, the thermal performance factor above unity indicates that the effect of heat transfer enhancement due to the turbulator (or enhancing device) is more dominant than the effect of rising friction and vice versa. The thermal performance factors for the oblique delta-winglet twisted tapes (*O-DWT*) and straight delta-winglet twisted tapes (*S-DWT*) calculated from Eqs. (17) and (20) based on the same pumping power, are plotted versus Reynolds number in Figs. 9 (a–b) and 10. The performance factors in the tube equipped with typical twisted tape (*TT*) are also plotted for comparison. At the same Reynolds number, the performance factors for both *O-DWT* and *S-DWT* are found to be greater than those for the typical twisted tape (*TT*). The performance factors for all twisted tapes tend to decrease with increasing Reynolds number. This suggests that both *DWTs* are feasible in terms of energy saving at higher Reynolds numbers compared to typical twisted tape (*TT*) which is suitable only for low Reynolds number as suggested in the previous works [3,20–22].

Similar to the effects found for Nusselt number and friction factor, the performance factor increases with decreasing twist ratio (y/w) and increasing depth of wing cut (DR) for all given Reynolds numbers. In addition, the performance factor for the *O-DWT* is greater than that in the *S-DWT* tubes, for the similar operation test conditions. Over the range considered, the values of performance factors obtained from using *O-DWT* and *S-DWT* are about 0.92–

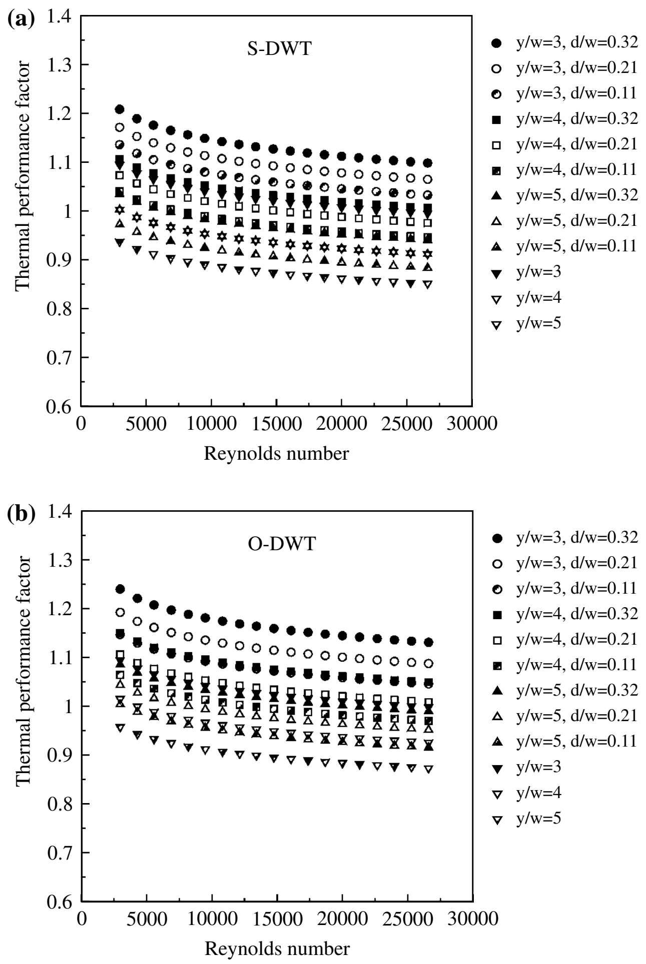

Fig. 9. Thermal performance factor versus Reynolds number for tubes with *DWT*, *TT*, and plain tube: (a) *S-DWT* and (b) *O-DWT*.

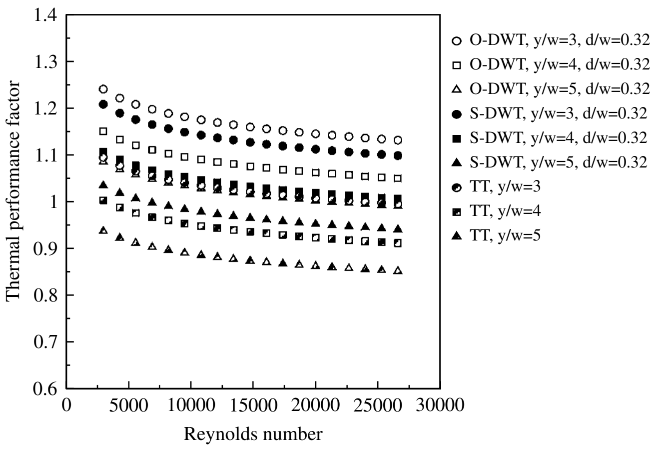

Fig. 10. Effect of straight and oblique wings on thermal performance factor.

1.24 and 0.88–1.21, respectively. The maximum thermal performance factor of 1.24 is achieved by using of *O-DWT* with $DR = 0.32$ and $y/w = 3$ at Reynolds number of 3000.

Empirical correlations for Nusselt number (Nu), friction factor (f), thermal performance factor (η) are developed for the tube with delta-winglet twisted tape inserts in the range of Reynolds number between 3000 and 27,000, $Pr = 4.91–5.57$, twist ratio ($y/w = 3$, 4 and 5), and depth of wing cut ratios ($DR = d/w = 0.11$, 0.21 and 0.32) as follows.

For oblique delta-winglet twisted tapes(O-DWT) :

$$Nu = 0.18Re^{0.67}Pr^{0.4}(y/w)^{-0.423}(1+(d/w))^{0.982} \quad (15)$$

$$f = 24.8Re^{-0.51}(y/w)^{-0.566}(1+(d/w))^{1.87} \quad (16)$$

$$\eta = 2.04Re^{-0.042}Pr^{0.4}(y/w)^{-0.261}(1+(d/w))^{0.45} \quad (17)$$

For straight delta-winglet twisted tapes(S-DWT) :

$$Nu = 0.184Re^{0.675}Pr^{0.4}(y/w)^{-0.465}(1+(d/w))^{0.76} \quad (18)$$

$$f = 21.7Re^{-0.45}(y/w)^{-0.564}(1+(d/w))^{1.41} \quad (19)$$

$$\eta = 2.164Re^{-0.0435}Pr^{0.4}(y/w)^{-0.304}(1+(d/w))^{0.356} \quad (20)$$

The experimental data and predicted data of the correlation by Eqs. (15), (16), (19), (20) are compared in Figs. 11 and 12. It appears that the mean absolute percentage deviation of the present experimental Nusselt number data is within 10% for Eq. (15) and 10% for

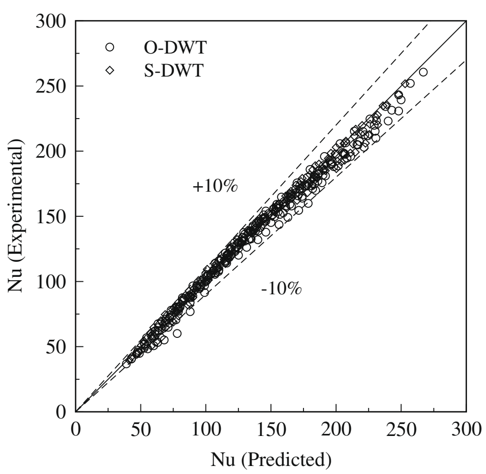

Fig. 11. Comparison of experimental Nusselt number and predicted results.

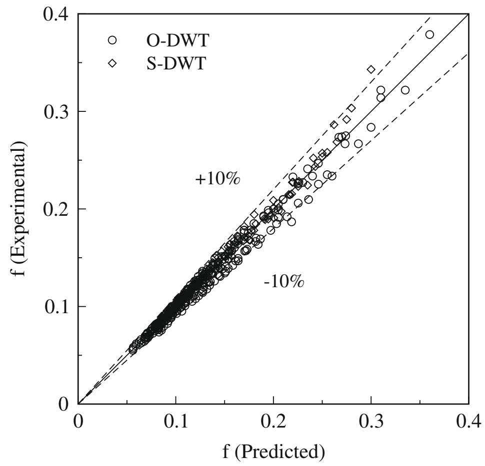

Fig. 12. Comparison of experimental friction factor and predicted results.

Eq. (18) from the values predicted by the present correlations while that of the friction factor is within 10% and 10% for Eqs. (16) and (19), respectively.

6. Conclusions

The enhancement of the heat transfer in a tube fitted with delta-winglet twisted tapes which act as swirl generator and turbulator is experimentally investigated. The values of Nusselts number and friction factor in the test tube equipped with delta-winglet twisted tape are noticeably higher than those in the plain tube and also tube equipped with typical twisted tape. Nusselt number and friction factor increase with decreasing of twist ratio (y/w) and increasing depth of wing cut ratio (DR) for all Reynolds numbers studied. In addition, the *O-DWT* gives higher Nusselt number and friction factor than that of the *S-DWT*. The thermal performance factor in the tube with *O-DWT* is greater than that with *S-DWT* and the factor increases with decreasing Reynolds number and increasing twist ratio. Over the range considered, the performance factor in the tubes equipped with the *O-DWT* and *S-DWT* are found to be around 0.92–1.24 and 0.88–1.21, respectively. The results of the present work agree well with the available correlations within ±10% in comparison with experimental data for the friction factor and within ±10% for the Nusselt number. In this study, several *DWT* tapes give the thermal performance factor higher than unity while the typical twisted tape provides lower unity. It is obvious that the *DWT* performs better heat transfer enhancement than that typical twisted tape. It indicates that the heat exchanger fitted with *DWT* is more compacted than one with the typical twisted tape. Again, the *DWT* can be replaced any of the *TT* efficiently to reduce the size of the heat exchanger.

Acknowledgements

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