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Turbulent Heat Transfer and Pressure Drop in Tube Fitted with Square-cut Twisted Tape

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Abstract Heat transfer, friction factor and thermal enhancement factor characteristics of a double pipe heat exchanger fitted with square-cut twisted tapes (STT) and plain twisted tapes (PTT) are investigated experimentally using the water as working fluid. The tapes (STT and PTT) have three twist ratios ($y = 2.0$, 4.4 and 6.0) and the Reynolds number ranges from 2000 to 12000. The experimental results reveal that heat transfer rate, friction factor and thermal enhancement factor in the tube equipped with STT are significantly higher than those fitted with PTT. The additional disturbance and secondary flow in the vicinity of the tube wall generated by STT are higher compared to that induced by the PTT is referred as the reason for better performance. Over the range considered, the Nusselt number, friction factor and thermal enhancement factor in a tube with STT are respectively, 1.03 to 1.14, 1.05 to 1.25 and 1.02 to 1.06 times of those in tube with PTT. An empirical correlation is also formulated to match with experimental data of Nusselt number and friction factor for STT and PTT.
Keywords heat transfer enhancement, friction factor, plain twisted tape, square-cut twisted tape, secondary flow

1 INTRODUCTION

The technique of improving the performance of heat transfer system is referred to as heat transfer augmentation or intensification. This leads to reduced size and cost of heat exchanger. Heat transfer augmentation techniques are widely used in areas such as heat recovery process, air conditioning and refrigeration systems, and chemical reactors. Passive and active methods of heat transfer augmentation techniques have been discussed in detail by Webb and Hyun Kim [1]. Insertion of twisted tapes in tubes is an effective passive heat transfer augmentation technique. The inserted twisted tape generates a swirling flow with higher extent of turbulence, and induces effective cross mixing and an efficient redevelopment of the thermal/hydrodynamic boundary layer which is the major influencing factor for heat transfer enhancement. The past investigations suggested that heat transfer enhancement by twisted tape strongly depends on their shape. The proper design of the twisted tapes provides an increase of heat transfer rate with a reasonable pressure drop, resulting in effective energy savings.

Dewan *et al*. [2] reviewed that passive techniques particularly twisted tape and wire coil insert are economical heat transfer augmentation tools. Wang and Sunden [3] reported that twisted tape is more effective than wire coil insert, if no pressure drop penalty is considered. Eiamsa-ard *et al*. [4–6] experimentally investigated the heat transfer and friction factor characteristics in a double pipe heat exchanger fitted with full length tape, spaced twisted tape, helical screw tape with and without core rod, and forward and backward arrangement of louvered strips. The results revealed that both heat transfer coefficient and friction factor increased with the decrease in free space ratio [4] for a tube with spaced twisted tape. A tube fitted with helical tape without a rod provided a higher heat transfer rate in comparison with the helical tape with a rod [5]. The louvered strip with backward arrangement led to better overall enhancement ratios than that of forward arrangement [6]. Heat transfer characteristics study was conducted by Naphon [7] for twisted tape insert by varying the mass flow rates and inlet temperature of hot water. Chang *et al*. [8, 9] investigated the heat transfer and friction factor characteristics of the tubes fitted with broken and serrated twisted tape inserts at different twist ratios. The results show that similar trends of Nusselt number and Fanning friction factor increased as the twist ratio decreased. Eiamsa-ard *et al*. [10, 11] presented the effect of delta-winglet twisted tape (DWT) and peripherally-cut twisted tape (PT) on the heat transfer enhancement in a heat exchanger tube. The results show that DWT [10] generated the additional flow disturbance in the tape edge region, leading to superior heat transfer and thermal performance factor in comparison with the typical one. Peripherally-cut twisted tape (PT) [11] generates higher turbulence intensity of fluid in the vicinity of the tube wall compared to that induced by the typical one, PTs also yield better performance mainly in the laminar regime than that of the turbulent regime. Chang *et al*. [12] fabricated the spiky twisted tape for the use of single phase and two phase flows. The results show that the performance of the spiky tape is better for two phase flow system in comparison to the single phase flow system. Chiu and Jang [13] compared the thermal-hydraulic characteristics of straight tape with and without holes and twisted tapes both numerically and experimentally.

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Promvonge and Eiamsa-ard [14] reported the effect of combined conical-ring and twisted tape insert fitted in a tube on heat transfer enhancement and flow friction, and found that the tube fitted with the conical ring and twisted tape provides higher enhancement efficiency than that of the conical ring alone. Bilen *et al*. [15] studied the effect of heat transfer and friction characteristics of a fully developed turbulent air flow in different grooved tubes compared with a plain tube. Sivashanmugam and Suresh [16–19] experimentally investigated that the heat transfer and friction factor characteristics of a circular tube fitted with helical screw tapes and regularly spaced helical screw tape inserts with different twist ratios for both laminar and turbulent flows and found a significant increase in the heat transfer rate [16, 17] with full length helical screw tape insert and decrease in friction factor than the full length helical twist insert [18, 19]. Hasim *et al*. [20] used compound enhancement *i.e*. twisted tape with helically ribbed tube for heat transfer enhancement based on non dimensional clearance. Krishna *et al*. [21] introduced the straight full twist in a circular tube to enhance the heat transfer rate and reduce the pressure drop in both laminar and turbulent flows. Bharadwaj *et al*. [22] described the effect of spirally grooved tube with twisted tape insert for laminar to turbulent region and found that heat transfer rate increased considerably in laminar and moderately in turbulent range of Reynolds number. Eiamsa-ard *et al*. [23] used the short length twisted tape for heat transfer enhancement in a circular tube. However the results exposed that heat transfer enhancement for the short length twisted tape is lower than that of the full length twisted tape. Eiamsa-ard *et al*. [24] used twisted tapes in forms of twin counter twisted tapes (CTs) and twin co-twisted tapes (COTs) and found that the CTs offered high thermal performance factor. Thianpong *et al*. [25] experimentally studied the heat transfer enhancement and friction factor of a dimpled tube fitted with twisted tape insert and reported that the heat transfer rate of a dimpled tube fitted with twisted tapes, are higher than the dimpled tube acting alone and the plain tube. More information about heat transfer by means of twisted tapes fitted in a circular tube were viewed in other reports [26–30].

Based on the available literature, it was pointed out that the modification on plain twisted tapes (PTT) *i.e*. small cuts on the tape [8–12], for example broken tape or spiky, serrated tape, delta-winglet tape (DWT) and peripherally-cut tape (PT), gave assurance for enhancement of both heat transfer rate and thermal enhancement factor. The reason behind the high thermal enhancement factor is that those small gaps bring pressure drop in the system to a reasonable level.

The present work reports the experimental work on heat transfer rate and friction factor characteristics of a double pipe heat exchanger fitted with PTT and square-cut twisted tapes (STT) for twist ratios 2.0, 4.4 and 6.0 with Reynolds number between 2000 and 12000. The modified twisted tapes comprise the square-cut with dimensions of 8 mm width and 8 mm depth alternately in the peripheral region of the tape. This type of tape is believed to perform in the same manner as mentioned in the literature for the case of broken or spiky tape, delta-winglet tape and peripherally-cut tapes. The experimental results obtained for the tube fitted with STT were also compared with those for the tube with PTT and the plain tube.

2 EXPERIMENTAL

Schematic diagram of the experimental set-up is shown in Fig. 1. It consists of two concentric tubes in which hot water flows through the inner tube (copper tube, $d_{\mathrm{i}} = 25$ mm, $L = 2000$ mm) and cold water flows in counter flow through annulus. The outer tube is insulated with asbestos rope and glass wool to minimize the heat loss to surroundings. Two calibrated crystal rotameters having flow ranges of 0−20 L·min^{-1} with ±0.1 L·min^{-1} accuracy are used to measure the flow rates of cold and hot water. Seven RTD Pt 100 type temperature sensors with ±0.1 ºC accuracy measure the inlet and outlet temperatures of the hot and cold water. Twisted tapes are made up of aluminum strips of thickness 1.5 mm and width 23.5 mm [26]. The twist ratio (y) is defined [2–4, 26] as the value of one length of twist (or) pitch length to diameter. In the experimentation, PTT and STT with twist ratios of 2.0, 4.4 and 6.0 are used. Geometries of STT and PTT are shown in Figs. 2 and 3.

Square-cut twisted tapes (Fig. 3) with square-cut dimensions of 8 mm depth and 8 mm width on both top and bottom alternately in the peripheral region of the tape were used to increase the disturbance near the walls of the test section. The water is heated using 3 kW water heaters and the desired temperature was controlled by temperature controller. The inlet temperatures at the hot and cold water sides were kept constant at 54 °C and 30 °C, respectively. The cold water was constantly flowed at 0.166 kg·s^{-1} whereas the hot water flow rate was adjusted from 0.033 kg·s^{-1} to 0.12 kg·s^{-1}. As steady state conditions were reached, the inlet and outlet temperatures of hot and cold water were recorded and pressure drop was measured using U-tube manometer (manometric fluid-carbon tetrachloride) for the case of plain tube. Thereafter, the experiment was repeated for PTT and STT. Details of experimental set up, all twisted tape inserts and operating conditions are summarized in Table. 1. Experimental uncertainties of Nusselt number, Reynolds number and friction factor were calculated using ANSI/ASME standard [32] to be ±5%, ±8%, and ±10%, respectively.

3 DATA REDUCTION

The data reduction [4, 25] of the measured results is summarized as follows.
Heat transferred to the cold water in the test section

$$Q_{\mathrm{c}} = m_{\mathrm{c}} c_p \left(T_{\mathrm{c2}} - T_{\mathrm{c1}}\right) \tag{1}$$

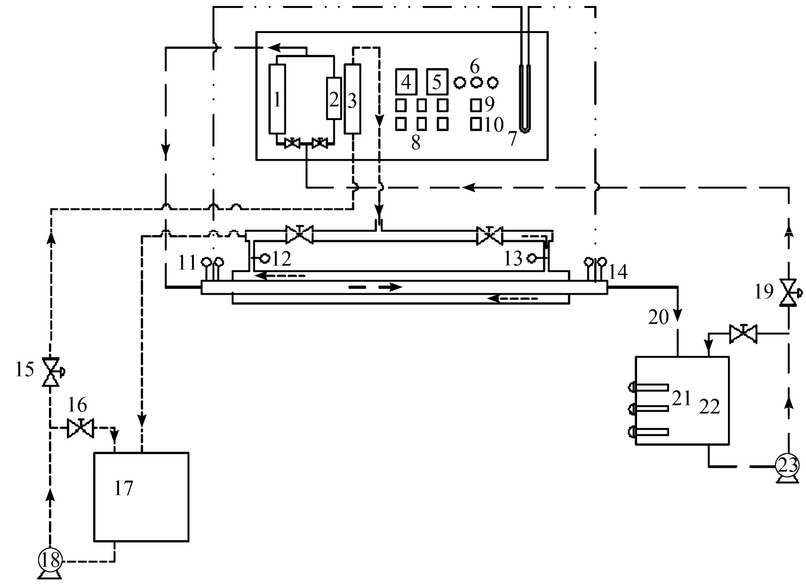
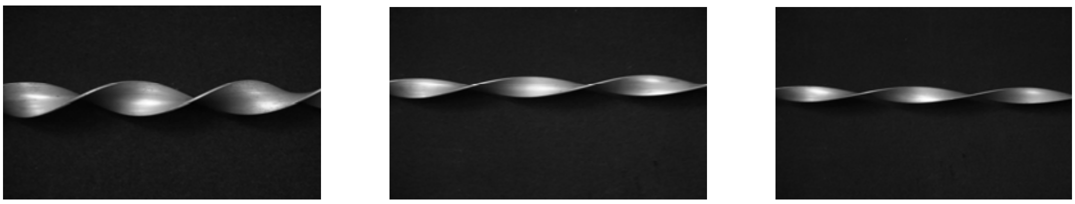
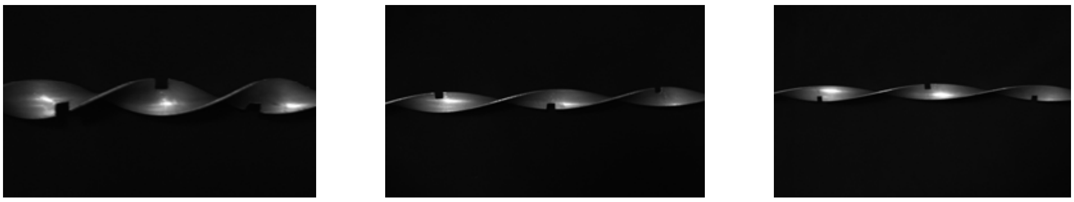


Figure 1 Schematic diagram of the experimental set-up
1—hot water rotameter (0–20 L·min^{-1}); 2—hot water rotameter (0–2 L·min^{-1}); 3—cold water rotameter (0–20 L·min^{-1}); 4—temperature indicator; 5—temperature controller; 6—heater indicators; 7—U-tube manometer; 8—swiches; 9—ammeter; 10—voltmeter; 11—RTD-hot water inlet; 12—RTD-cold water inlet; 13—RTD-cold water inlet; 14—RTD-hot water outlet; 15—cold water control valve; 16—cold water bypass valve; 17—cold water sump; 18—cold water pump; 19—hot water control valve; 20—hot water bypass valve; 21—heater; 22—hot water tank; 23—hot water pump
— — — hot water line; ---- cold water line; — · · — hot water pressure line



(a) $y=2.0$ (b) $y=4.4$ (c) $y=6.0$

Figure 2 Geometries plain twisted tapes (PTT)



(a) $y=2.0$ (b) $y=4.4$ (c) $y=6.0$

Figure 3 Geometries square-cut twisted tapes (STT)

Heat transferred from the hot water in the test section

$$Q_{\mathrm{h}}=m_{\mathrm{h}}c_{p}\left(T_{\mathrm{h1}}-T_{\mathrm{h2}}\right) \tag{2}$$

The percentage of heat loss [30] between hot and cold water side in the present heat exchanger can be given as

$$\varepsilon=\frac{Q_{\mathrm{h}}-Q_{\mathrm{c}}}{Q_{\mathrm{c}}}\times 100\% \tag{3}$$

The heat loss was found out to be 2% to 8% [5, 6] due to thermal loss from the test section to surroundings and it was reasonably small. Thus, the average heat transfer rate for hot and cold water side was taken for internal convective heat transfer coefficient calculation:

$$Q_{\mathrm{avg}}=\frac{Q_{\mathrm{c}}+Q_{\mathrm{h}}}{2} \tag{4}$$

The over all heat transfer coefficient is

$$U=\frac{Q_{\mathrm{avg}}}{A_{\mathrm{i}}\Delta T_{\mathrm{lm}}} \tag{5}$$

where, $A_{\mathrm{i}}=\pi d_{\mathrm{i}}L$

Table 1 Details of experimentation

Parameter	Value
Experimental set-up	
inner tube inner diameter (plain tube) d_i/mm	25.0
inner tube outer diameter (plain tube) d_o/mm	28.5
outer tube inner diameter (plain tube) d_s/mm	54.5
test tube length/mm	2000
material of inner tube	copper
material of outer tube	galvanized iron
Twisted tape	
tape width/mm	23.5
tape pitch length (H, 180º)/mm	50, 110 and 150
twist ratio ($y = H/d_i$)	2.0, 4.4 and 6.0
tape thickness/mm	1.5
material	aluminium
width of square cut/mm	8
depth of square cut/mm	8
Test conditions	
Reynolds number, Re	2000–12000
type of flow in inner tube and annulus	turbulent
hot water inlet temperature/ºC	54
cold water inlet temperature/ºC	30

The tube side heat transfer coefficient (h_i) was determined by neglecting the conduction thermal resistance of copper tube wall:

$$\frac{1}{U}=\frac{1}{h_i}+\frac{1}{h_a} \tag{6}$$

where the annulus side heat transfer coefficient (h_a) was estimated using the correlation of Dittus-Boelter equation [33]:

$$Nu_a=\frac{h_a D_h}{k}=0.023Re^{0.8}Pr^{0.3} \tag{7}$$

where $D_h = d_s - d_o$. Thus,

$$Nu_i=\frac{h_i d_i}{k} \tag{8}$$

Friction factor is defined as

$$f=\frac{\Delta p}{\left(\frac{L}{d_i}\right)\left(\frac{\rho u^2}{2}\right)} \tag{9}$$

4 RESULTS AND DISCUSSION

In this section, the heat transfer and friction factor characteristics and thermal enhancement factor of a heat exchanger double pipe heat exchanger fitted with PTT and STT are presented. The experiments are performed using twisted tapes (PTT and STT) with three twist ratios $y = 2.0$, 4.4 and 6.0 in the range of Reynolds number between 2000 and 12000.

4.1 Validation of plain tube experimental results

The variation of Nusselt number with Reynolds number for plain tube is shown in Fig. 4. The experimental data are matching with the plain tube forced convection correlations of Dittus-Boelter [1930, Eq. (10) and Gnielinski (1976, Eq. (11)] [31] with the discrepancy of ±8.4% and ±5.0% respectively for the Nusselt number.

$$Nu=0.023Re^{0.8}Pr^{0.3} \tag{10}$$

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

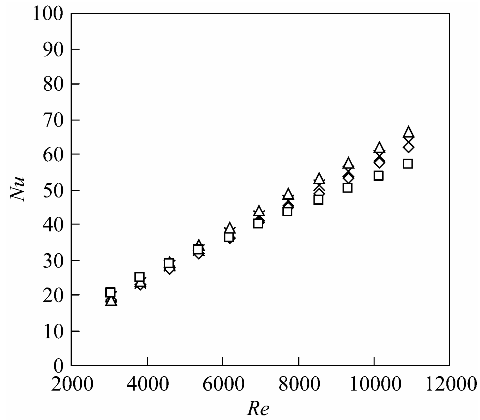


Figure 4 Validation of plain tube experimental data for Nusselt number
× plain tube experimental data; ◇ plain tube correlation; △ Gnielinski equation; □ Dittus-Boelter equation

The variation of friction factor with Reynolds number for plain tube is shown in Fig. 5. The data obtained in this work is compared with Blasius Eq. (12)

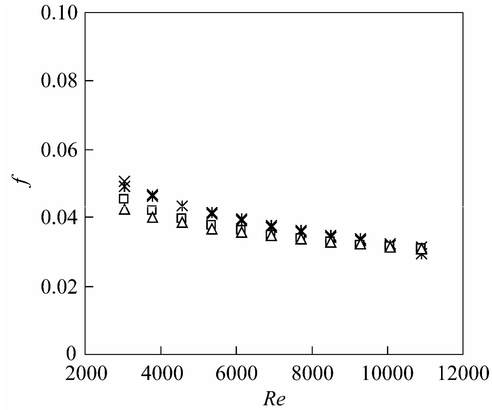


Figure 5 Validation of plain tube experimental data for friction factor
⁎ plain tube experimental data; × plain tube correlation; □ Petukhov equation; △ Blasius equation

and first Petukhov Eq. (13) with the deviation of ±10% and ±8% respectively:

$$f = 0.316 Re^{-0.25} \tag{12}$$

$$f = \left(0.790 \ln Re - 1.64\right)^{-2} \tag{13}$$

In addition, the experimental data of the plain tube are correlated with Nusselt number and friction factor as follows:

$$Nu_{\mathrm{p}} = 0.00595 Re^{0.95} Pr^{0.33} \tag{14}$$

$$f_{\mathrm{p}} = 0.255 Re^{-0.374} \tag{15}$$

Eqs. (14) and (15) are found to represent the experimental data within ±4% for Nusselt number and ±6% for friction factor shown in Figs. 4 and 5 respectively.

Correlations Eqs. (14) and (15) are used to evaluate the thermal enhancement factor associated by PTT and STT in Section 4.5.

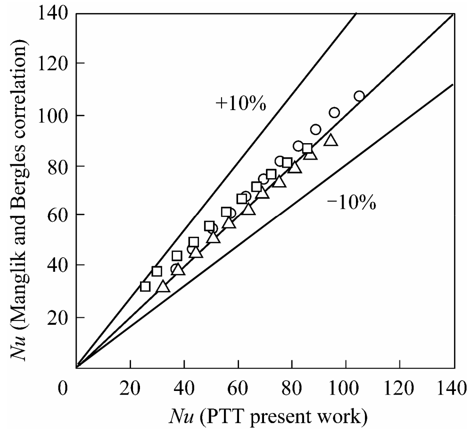
4.2 Validation of plain twisted tape experimental results

The tube fitted with PTT experimental results are validated using the correlations developed by Manglik and Bergles [27] which yielded maximum deviation of ±20% [10, 11, 26, 29] for both Nusselt number and friction factor respectively. Our data on Nusselt number and friction factor of a tube fitted with PTT are compared with the correlations by Manglik and Bergles [27] for the twist ratios of $y = 2.0$, 4.4 and 6.0 as demonstrated in Fig. 6 (a, b). Apparently, present results reasonably agree well with the available correlations within ±10% for Nusselt number and ±20% for friction factor respectively.

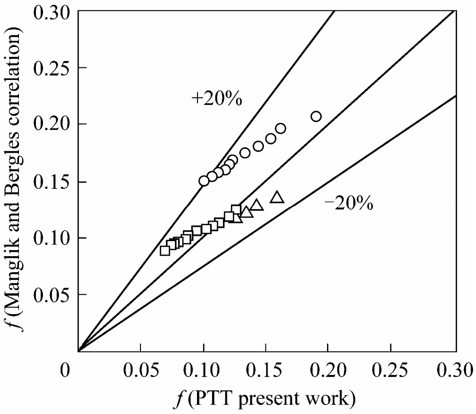
4.3 Effect of square-cut twisted tape on heat transfer

Variation of Nusselt number with Reynolds number in the tube fitted with STT, the tube fitted with PTT and also the plain tube are presented in Fig. 7. It is observed that for all cases, the Nusselt number increases with the increasing Reynolds number. As expected, PTT heat transfer rates are higher than those from the plain tube fitted without twisted tape. The lower twist ratio ($y = 2.0$) heat transfer rate [4, 7] is higher than those from higher ones ($y = 4.4$ and 6.0) due to increase in turbulent intensity and flow length across the range of Reynolds number. Mean Nusselt numbers for PTT with twist ratios, $y = 2.0$, 4.4 and 6.0 are respectively, 1.67, 1.50 and 1.33 times better than that for the plain tube.

Nusselt number (Fig. 7) in the tube with STT is higher than those in the plain tube and tube with PTT insert over the range of Reynolds number 2000–12000. STT provides an additional disturbance to the fluid in the vicinity of the tube wall and vorticity behind the cuts and thus leads to a higher heat transfer enhancement in comparison with plain tube and PTT. From the literature [8–12], it was found that the reason for heat transfer enhancement due to the promotion of fluid mixing and turbulence intensity [8, 9, 12], synergy effect of vortex circulation together with secondary flow in addition with the main swirl flow [10] and in-



(a) Nusselt number



(b) Friction factor

Figure 6 Validation of plain twisted tapes
○ PTT; △ PTT 4.4; □ PTT 6.0

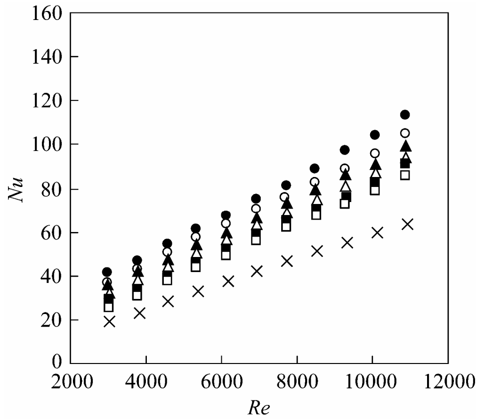


Figure 7 Square-cut (STT) and plain twisted tapes (PTT): Nusselt number *vs*. Reynolds number
● STT 2.0; ○ PTT 2.0; ▲ STT 4.4; △ PTT 4.4; ■ STT 6.0; □ PTT 6.0; × plain tube

crease in turbulence near the wall surface of the tube [11] using the broken, serrated, spikes, delta winglet and peripherally-cut twisted tapes respectively. This mechanism is believed to be existed in the tube fitted with the square-cut twisted tape as well. In the present study, mean Nusselt number for tube equipped with STT of twist ratios of 2.0, 4.4 and 6.0, are respectively 1.81, 1.60 and 1.40 times of that plain tube and 1.08, 1.067 and 1.055 times of that for the tube equipped with PTT.

4.4 Effect of square-cut twisted tape on friction factor

Variation of friction factor with Reynolds number in the tube fitted with STT, PTT and also the plain tube are presented in Fig. 8. It shows that the friction factor decreases continuously with Reynolds number, and friction factor for lower twist ratio ($y = 2.0$) is significantly larger than that of higher twist ratios ($y = 4.4$, 6.0) due to stronger swirl flow in the tube. Over the range studied, the mean friction factor for the PTT with twist ratios, $y = 2.0$, 4.4 and 6.0 are respectively, 3.48, 2.92 and 2.45 times higher than that for the plain tube.

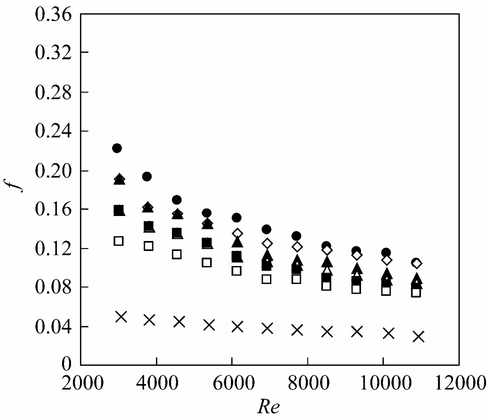


Figure 8 Square-cut and plain twisted tapes: friction factor *vs*. Reynolds number
● STT 2.0; ◇ PTT 2.0; ▲ STT 4.4; △ PTT 4.4; ■ STT 6.0; □ PTT 6.0; × plain tube

Figure 8 shows that STT yields higher pressure drop than those in the plain tube as well as the tube fitted with PTT. This is because the additional disturbance increases the tangential contact between secondary flow and the wall surface of the tube. Mean friction factor for the STT with twist ratios of 2.0, 4.4 and 6.0 are respectively, 3.81, 3.27 and 2.83 times of that for the plain tube and 1.09, 1.12 and 1.15 times of that for the tube with PTT insert.

The correlations for Nusselt number and friction factor are developed to the present experimental results respectively for a plain tube fitted with PTT and STT are as follows.

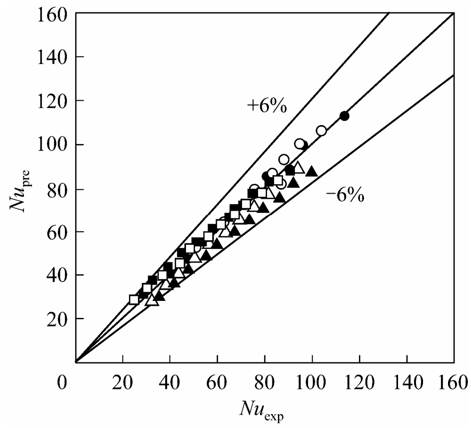
$$Nu_{\mathrm{PTT}} = 0.027 Re^{0.862} Pr^{0.33} y^{-0.215} \tag{16}$$

$$f_{\mathrm{PTT}} = 2.642 Re^{-0.474} y^{-0.302} \tag{17}$$

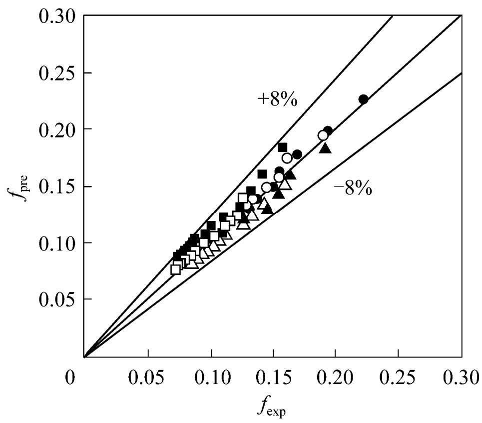
$$Nu_{\mathrm{STT}} = 0.041 Re^{0.826} Pr^{0.33} y^{-0.228} \tag{18}$$

$$f_{\mathrm{STT}} = 6.936 Re^{-0.579} y^{-0.259} \tag{19}$$

The predicted values agree with experimental data within ±6% and ±8% respectively, for Nusselt number and friction factor shown in Fig. 9 (a, b).



(a) Nusselt number



(b) Friction factor

Figure 9 Comparison between predicted and experimental results
● STT 2.0; ○ PTT 2.0; ▲ STT 4.4; △ PTT 4.4; ■ STT 6.0; □ PTT 6.0

4.5 Thermal enhancement factor for PTT and STT

According to the past literature studies [10, 11], a comparison of heat transfer coefficients in a plain tube (p) and the tube fitted with turbulator (t) were made at the same pumping power since it is relevant to operation cost.

For constant pumping power,

$$(\dot{V}\Delta P)_{\mathrm{p}} = (\dot{V}\Delta P)_{\mathrm{t}} \tag{20}$$

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

$$\left(f\,Re^{3}\right)_{\mathrm{p}} = \left(f\,Re^{3}\right)_{\mathrm{t}} \tag{21}$$

The thermal enhancement factor (η) at equal pumping power is defined as the ratio of the convective heat transfer coefficient of the tube with turbulator to that of the plain tube. It can be expressed as

$$\eta = \left|\frac{h_t}{h_p}\right|_{pp} \tag{22}$$

Using Eqs. (15), (17), (19) and (21), the Reynolds number for the plain tube $(Re)_p$ is written as the function of Reynolds number $(Re)_t$ for the tube with turbulator for PTT and STT:

$$\left(Re_p\right)_{PTT} = 2.436 Re_t^{0.962} y^{-0.115} \tag{23}$$

$$\left(Re_p\right)_{STT} = 3.518 Re_t^{0.922} y^{-0.098} \tag{24}$$

Employing Eqs. (14), (16), (18) and (22), the thermal enhancement factor for the PTT and STT can be written as

$$\eta_{PTT} = \left|\frac{h_t}{h_p}\right|_{pp} = 1.95 Re_t^{-0.052} y^{-0.106} \tag{25}$$

$$\eta_{STT} = \left|\frac{h_t}{h_p}\right|_{pp} = 2.07 Re_t^{-0.049} y^{-0.135} \tag{26}$$

Thermal enhancement factor (η) for PTT and STT at different twist ratios $y = 2.0$, 4.4 and 6.0 calculated from Eqs. (25) and (26) respectively for PTT and STT are presented in Fig. 10.

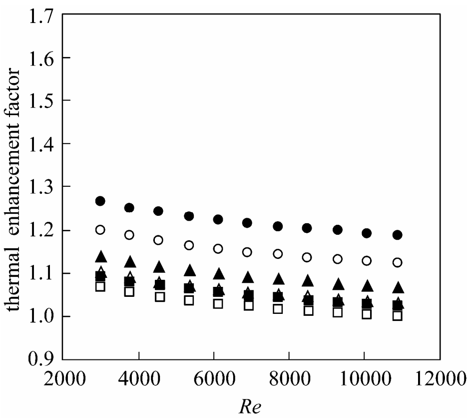


Figure 10 Thermal enhancement factors *vs*. Reynolds number for tubes with PTT and STT
● STT 2.0; ○ PTT 2.0; ▲ STT 4.4; △ PTT 4.4; ■ STT 6.0; □ PTT 6.0

At the same Reynolds number, the thermal enhancement factors for STT are found to be greater than those for the PTT. The thermal enhancement factor for all twisted tapes tends to decrease with the increasing Reynolds number. With the use of PTTs, thermal enhancement factors are in a range between, 1.12 to 1.2, 1.03 to 1.10 and 1.0 to 1.07 respectively for the twist ratios $y = 2.0$, 4.4 and 6.0. On the other hand the use of STTs offered thermal enhancement factors in a range between 1.19 to 1.27, 1.07 to 1.14 and 1.02 to 1.09 respectively for the twist ratios $y = 2.0$, 4.4 and 6.0. The above data indicates that the use of STTs gave more efficient heat transfer enhancement than the application of PTT. The varieties tube inserts were taken to compare the thermal enhancement factor are shown in Table 2. Fig. 11 shows that comparison of the STT with helical screw tape with and without core rod [5], louvered strip with forward and backward arrangement [6], broken twisted tape [8], delta winglet twisted tape [10], peripherally-cut twisted tape [11] and twin co and counter twisted tapes [24].

It was observed from the present work that STT thermal enhancement factor was comparatively low with helical screw tape without core rod (helical tape—WOCR), louvered strip with forward (LSF) and backward (LSB) arrangement, broken tape and twin counter twisted tapes (CT). This is mainly attributed to the intensity of turbulence which is higher in those tapes compared with that of present STT. The performance of STT was competent with the other tapes of delta winglet (DWT), peripheral-cut (PT) and twin co-twisted tapes (COT).

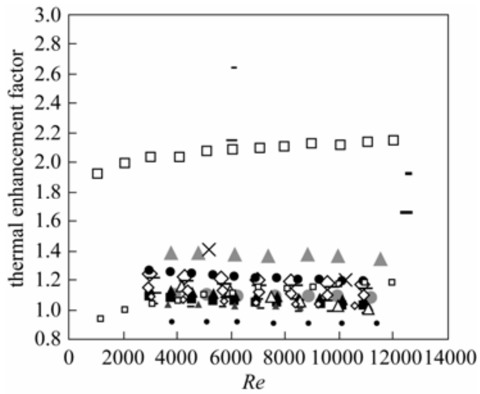


Figure 11 Comparisons of present thermal enhancement factors with the varieties of heat transfer enhancement devices
▲STT 4.4; ●ST 2.0; △ PT 3.0; ▲ CT 2.5; ▴ CT 4.0; ● COT 2.5; • COT 4.0; ■ STT 6.0; ◇ O-DWT 3.0; ◇ O-DWT 4.0; ◇ O-DWT 5.0; — S-DWT 3.0; – S-DWT 4.0; - S-DWT 5.0; × broken tape $y = 2.0$; - LSB; – LSF; □ helicel tape—WOCR; □ helical tape—WCR

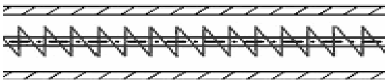
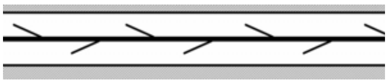
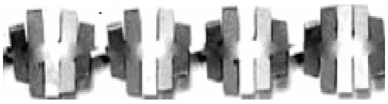
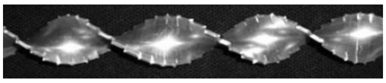
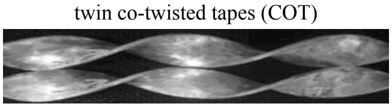
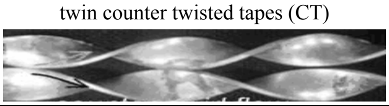
5 CONCLUSIONS

Experimental investigations of heat transfer, friction factor and thermal enhancement factor of circular tube fitted with PTT and STT in turbulent regimes (2000<Re<12000) for twist ratios of 2.0, 4.4 and 6.0 are studied. The conclusions drawn from the present study are as follows.

(1) The values of Nusselt number and friction factor values for the tube with square-cut twisted tapes (STT) are noticeably higher than that of plain tube and also tube equipped with plain twisted tapes (PTT).

(2) Over the range of Reynolds number considered, thermal enhancement factors in the tube equipped

Table 2 Varieties of heat transfer enhancement devices

No.	Insert configuration	Description	Reference
1	helical screw tape with core rod 	helical screw tape with core rod (WCR) fitted in a double pipe heat exchanger with loose fit arrangement	[5]
2	helical screw tape without core rod 	helical screw tape without core rod (WOCR) fitted in a double pipe heat exchanger with loose fit arrangement	[5]
3	louvered strip with forward (LSF) 	LSF fitted in a double pipe heat exchanger with different angles of 15°, 25° and 30°	[6]
4	louvered strip with backward (LSB) 	LSB fitted in a double pipe heat exchanger with different angles of 15°, 25° and 30°	[6]
5	broken twisted tape insert 	broken twisted tape insert fitted in a uniformly heated tube with different twist ratios of $y=1$, 1.5, 2 and 2.5	[8]
6	delta-winglet twisted tape (DWT) 	delta-winglet twisted tape (DWT)fitted in a uniformly heated tube with different twist ratios of $y=3.0$, 4.0 and 5.0	[10]
7	peripherally-cut twisted tape (PT) 	peripherally-cut twisted tape (PT) insert fitted in a uniformly heated tube with twist ratio of $y=3.0$	[11]
8	twin co-twisted tapes (COT) 	twin co-twisted tapes (COT) insert fitted in a uniformly heated tube with twist ratio of $y=2.5$, 3.0, 3.5 and 4.0	[24]
9	twin counter twisted tapes (CT) 	twin counter twisted tapes (CT) insert fitted in a uniformly heated tube with twist ratio of $y=2.5$, 3.0, 3.5 and 4.0	[24]

with the STT and PTT are found to be around 1.02 to 1.27 and 1.0 to 1.2, respectively. The thermal enhancement factors for all the cases are more than unity indicates that the effect of heat transfer enhancement due to the enhancing tool is more dominant than the effect of rising friction factor and *vice-versa*.

(3) The empirical correlations for the Nusselt number, friction factor and the thermal enhancement factor for PTT and STT are developed and it was reasonably fitted with the experimental data.

(4) The square-cut twisted tapes offered better heat transfer enhancement than the plain twisted tapes. Therefore, STT can be used in place of PTT to reduce the size of heat exchanger.

NOMENCLATURE

A	area, m^2
c_p	specific heat, J·kg^{-1}·K^{-1}
D_h	hydraulic diameter, m
d	tube diameter, m
d_s	inner diameter of outer tube, m
f	friction factor
H	pitch length based on 180°
h	heat transfer coefficient, W·m^{-2}·K^{-1}
k	thermal conductivity, W·m^{-1}·K^{-1}
L	tube length, m
m	mass flow rate, kg·s^{-1}
Nu	nusselt number
ΔP	pressure drop
Pr	Prandtl number
Q	heat transfer rate, W
Re	Reynolds number
T_c	cold water temperature, °C
T_h	hot water temperature, °C
ΔT_{lm}	logarithmic mean temperature difference, °C
U	overall heat transfer coefficient, W·m^{-2}·K^{-1}
u	velocity, m·s^{-1}
y	twist ratio
η	thermal enhancement factor
μ	dynamic viscosity, kg·m^{-1}·s^{-1}

ρ density, $kg \cdot m^{-3}$

Subscripts

a annulus
avg average
c cold water
h hot water
i inner
lm logarithmic mean temperature
o outer
p plain tube
pp pumping power
t turbulator
1 inlet
2 outlet

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