Experimental investigation of heat transfer enhancement of a heat exchanger tube equipped with double-cut twisted tapes

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Abstract

An Experimental study has been performed to investigate the heat transfer characteristics and friction factors of fluid flow through a heat exchanger tube equipped with double-cut twisted tapes (DCTT) with different cut ratios. The rectangular cuts have aspect ratio values ranging from 0.25 to 0.90. This study is carried out under turbulent flow regime ($5000 \le Re \le 15{,}000$) and water is selected as the working fluid. Due to swirl flows, DCTTs lead to more fluid mixing between the tube wall and core regions, which has considerable effects on the heat transfer inside heat exchanger tubes. The results show that by increasing the cut ratio from 0.25 to 0.90, the Nusselt number (Nu) enhances up to 177.4%. The thermal performance varies from 1.63 to 1.44 for the cut ratio of 0.90. Three correlations based on the experimental data are developed to predict Nu, f and η as functions of design parameters under turbulent flow regimes. The experimental results show that using DCTTs is an effective method to improve the thermal performance of the heat exchangers.

Keywords: Double-cut twisted tape; heat exchanger; swirl flow; cut ratio; turbulent flow.

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1. Introduction

Nowadays, heat transfer enhancement methods play a vital role in designing different industrial systems such as heat exchangers, chemical processes, refrigeration, solar systems, etc. The main goals of these methods are to reduce the size and weight of the thermal system, reduce costs and improve the systems' performance. Several techniques have been applied to enhance heat transfer inside heat exchanger tubes such as wavy surfaces [1], spring tape inserts [2], grooved and ribbed geometries [3], vortex generators [4-6], angular cut wavy strips [7], louvered strips [8] and twisted tapes [9, 10].

Since the past decade, twisted tapes and swirl generators have been extensively applied in heat exchangers due to the low costs and simple installations. Mounting twisted tape causes swirl flows and enhances thermal boundary layer disturbances and fluid mixing between the tube wall and core regions. Stronger swirl flows improve the heat transfer of fluid flow through the heat exchanger tubes. The main design consideration for heat exchanger tubes is to minimize the pressure drop across the flow direction to improve the heat transfer performance by using optimized twisted tapes.

Various approaches have been explored to modify twisted tapes and improve the thermal performance of heat exchangers. Murugesan et al. [11] experimentally investigated heat transfer and friction factor of turbulent fluid flow through a circular tube equipped with square-cut twisted tapes. They found that using modified twisted tape offered better thermal performance compared to the conventional twisted tapes. The experimental results of He et al. [12] showed that the cross-hollow twisted tapes with hollow spaces of 6mm, improved heat transfer rate up to 5.7% compared with other tested geometries. Salam et al. [13] reported that the rectangular-cut twisted tape with a cut depth/width ratio of 0.57, provided an appropriate heat transfer enhancement in comparison with the plain tube and the tube fitted by conventional twisted tapes. Piriyarungrod et al. [14] experimentally evaluated heat transfer and pressure drops by using multiple twisted tapes. Their results revealed that among tested geometries, tubes fitted by six twisted tapes have the best thermal performance. They concluded that the multiple twisted tapes could enhance heat transfer up to 31% comparted with single twisted tape. Khoshvaght-Aliabadi and Eskandari [15] examined the thermal performance of nanofluid flows inside tubes fitted with twisted tapes with different twist lengths. It was observed that the heat transfer enhanced by using twisted tapes with low to high shape in comparison with the other ones. Nakhchi and Esfahani [16] numerically investigated nanofluid flow through the heat exchanger tube fitted by cross-cut twisted tapes with an alternate axis (CCTA). The cut-width/tape-width ratio (b/w) and the cut-length/tape-width ratio (s/w) varied from 0.7 to 0.9 and 2 to 2.5, respectively. The numerical results indicated that the CCTA with b/w=0.9 and s/w=2.5 had the best performance among tested geometries. Many other modified twisted tapes that were employed in the literature include V-cut twisted tape [17], trapezoidal-cut twisted tape [18], delta-winglet twisted tape [19], perforated twisted tape [20], transverse-cut twisted tapes [21], twisted tape with wire coil insert [22]. Jagged twisted tape [23], notched twisted tape [23], Centre-Trimmed twisted tape [24], serrated twisted tape [25], etc.

The techniques mentioned above have been focused on employing modified twisted tapes to augment the thermal performance in heat exchanger tubes. To the best of the authors' knowledge, however, the thermal performance of the tubes with the use of rectangular-cut twisted tapes has received less attention. A prior experimental investigation on rectangular-cut twisted tapes (RCT) [13] showed that a pipes equipped with RCTs offer excellent heat transfer enhancement under turbulent flow. However, the effects of cut ratios and the number of cuts on performance improvement were not considered in their studies. It is also necessary to find an appropriate arrangement of rectangular cuts to enhance the heat transfer inside heat exchanger tubes. This paper presents the results of an experimental investigation of the heat transfer and the friction factor of a heat exchanger tube using double-cut twisted tapes with four different cut ratios $\left(b/c = 0.25, 0.50, 0.75 \text{ and } 0.90\right)$. Experiments are performed under uniform wall heat flux for turbulent flow regime (5000 ≤ Re ≤ 15,000) using water as the working fluid. Based on the experimental results, new correlations are proposed for the Nusselt number, friction factor, and thermal performance for a better understanding of fluid flow through tubes fitted with double-cut twisted tapes.

2. Experimental setup

Fig. 1 shows the schematic diagram of the experimental setup. The experimental apparatus consists of a flow loop, heating elements, temperature measuring units, and a pressure drop measuring unit. The flow loop consists of a reservoir tank, heat exchanger cooling system, a ball valve for flow control, test section, and a rotameter for measuring the water flow rate within an accuracy of ±0.3%. The water inlet velocity varies from 0.19 to 0.57 m/s. The test section is made of a calming tube (3000 mm) for developing flow hydrodynamically, and a copper tube with a length of 840 mm, inner diameters $\left(D_i\right)$ of 22.6 mm and tube thickness $\left(t\right)$ of 4mm. Double-cut twisted tapes (DCTTs) are made of stainless steel with 20 mm width, tape pitch (y) of 105 mm and thickness $\left(\delta\right)$ of 2 mm. The twist ratio, which is defined as y/w [26, 27], is kept constant at 5.25.

The tube is uniformly heated by flexible silicone rubber heaters (SRFR-Series OMEGA) with a maximum power of 753.6 W. The power supply is controlled by a variac transformer to obtain constant heat flux along the entire length of the test section. The tube is carefully insulated with fiberglass with 50mm thickness to minimize the heat loss. Fifteen K-Type thermocouples are connected to the tube wall at equal distances. The distance between every two thermocouples is kept constant at 52.5 mm. Each thermocouple is fixed to the tube wall by using thermal glue (HC-131), and all of them are connected to a data logger. The water temperature at the inlet and outlet of the test section are measured with TT-T-30SLE high precision thermocouples $\left(\pm 0.1^{\circ}C\right)$. For the calibration of the thermocouples, a two-point calibration method (ice bath and boiling water) is used. Moreover, an analog mercury-based thermometer has been immersed in the reference temperature baths to ensure the accuracy of the calibration. The hot water at the outlet of the test section cooled down by passing it through a cooling unit. The measurements were performed after reaching steady-state conditions. The water is then transferred to the water tank by means of a pump. The pressure drop in the test section is measured by a digital pressure transducer. The heating pipe and the thermocouples were insulated by fiberglass to minimize the heat loss to the surroundings. The hot water at the outlet of the tube is cooled down by using a refrigeration cycle.

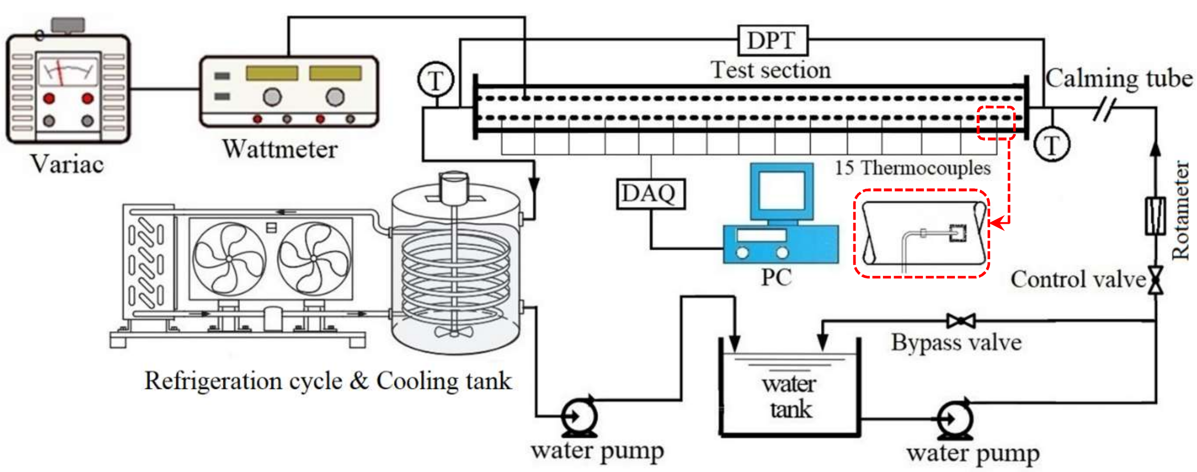
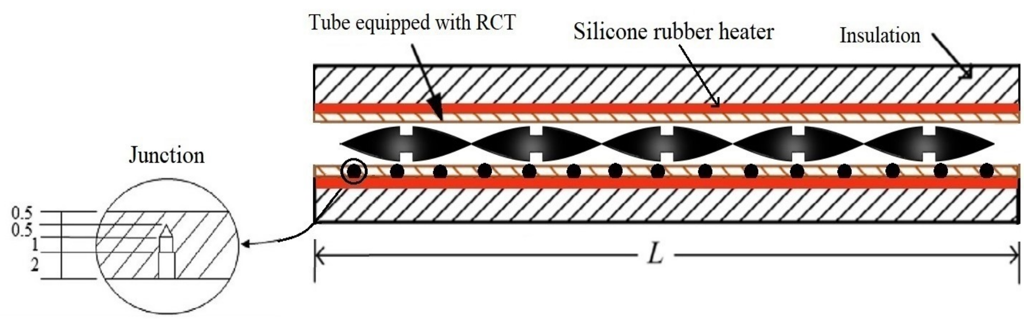
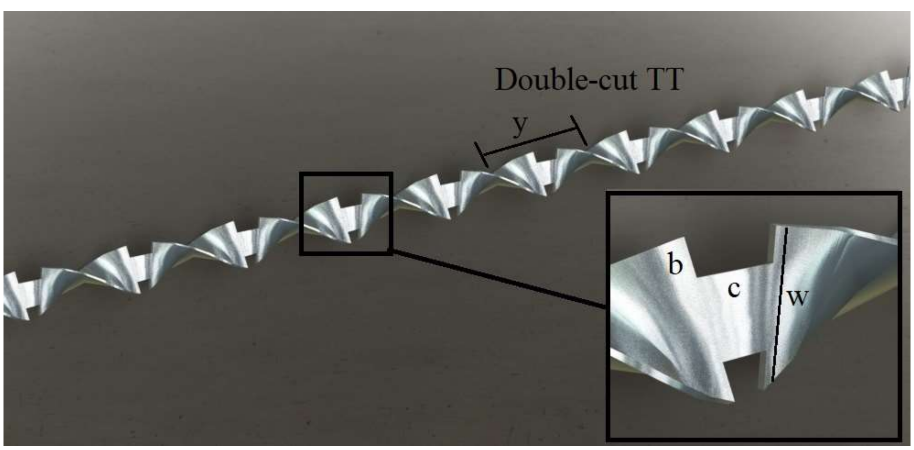


Fig. 1 Schematic of the experimental setup

The geometry details of the test section and double-cut twisted tapes are shown in Fig. 2. Fifteen K-Type thermocouples are silver-soldered to the test tube surface (embedded in V-groove tube surface) for measuring local wall temperatures. Reynolds number is defined based on the flow rate at the inlet of the test tube and varies from 5000 to 15,000. The thermo-physical properties are evaluated at the bulk fluid temperature. In the present study, the heat transfer and pressure drop of fluid flow in the presence of double-cut twisted tapes with different cut ratios are investigated. In this study, five different DCTTs with different cut ratios are designed and inserted into a center line of the heat exchanger tube. The geometric details of twisted tapes are shown in Table 1.



a) Test section



b) double-cut twisted tape (DCTT)

Fig. 2 Test section and double-cut twisted tapes used in the present work and their geometries

Table 1 Dimensions of the test section

Parameter	Symbol	Tube with DCTT insert
Tube length (mm)	L	840
Inner diameter (mm)	D_i	22.6
Outer diameter (mm)	D_o	26.6
Tube thickness (mm)	t	4
Tape thickness (mm)	δ	2
Tape width (mm)	w	20
Tape pitch (mm)	y	105
Twist ratio	y/w	5.25
Cut depth (mm)	b	2.5, 5, 7.5, 9
Cut width (mm)	c	10
Cut ratio	b/c	0.25, 0.5, 0.75, 0.9

3. Data reduction

Convective heat transfer rate (Q_c) is evaluated from the following equation:

$$Q_c = \dot{m} C_p (T_o - T_i) \tag{1}$$

where $T_o - T_i$ is the temperature difference between test section outlet and inlet, and $\dot{m}$ is the water flow rate. The total heating of the test section by heating elements (Q_{tot}) can be calculated from the output voltage (V) and current (I) of the power supply as:

$$Q_{tot} = VI = Q_c + Q_{loss} \tag{2}$$

The heat loss (Q_{loss}) to the environment can be evaluated as:

$$Q_{loss} = Q_{ins} + Q_{rad} \tag{3}$$

Subscripts "*ins*" and "*rad*" denote the insulation and radiation, respectively. The loss of the total heat flux due to radiation and natural convection from the insulated surface can be calculated as:

$$Q_{ins}+Q_{rad}=A_{ins}h_{ins}\left(T_{ins}-T_{\infty}\right)+\frac{\sigma\left(T_{ins}^{4}-T_{w}^{4}\right)}{\frac{1-\varepsilon_{ins}}{\varepsilon_{ins}A_{ins}}+\frac{1}{A_{ins}F_{ins-sw}}+\frac{1-\varepsilon_{sw}}{\varepsilon_{sw}A_{sw}}} \tag{4}$$

In the above equation, F_{ins-sw} is the view factor between the insulation and the surrounding walls (sw) which is equal to one in this study. $\sigma=5.67\times10^{-8}W/m^2K^{-4}$ is the Stefan–Boltzmann constant, $\varepsilon_{ins}\approx0.75$ and $\varepsilon_{sw}\approx0.86$ are the emissivity coefficients of the insulation and surrounding walls respectively, and A_{sw} is assumed to be infinity. The calculations show that $\left(Q_{loss}\right)/Q_{tot}\times100\%$ is less than 5.3% in extreme cases. The convective heat transfer rate is equal to the heat transfer gained by water through the test section under steady state conditions:

$$Q_c=\dot{m}C_p\left(T_o-T_i\right)=h\left(\overline{T}_w-T_b\right) \tag{5}$$

where $\overline{T}_w=\sum T_w/15$ is the average wall temperature and $T_b=\left(T_i+T_o\right)/2$ is the bulk temperature of the fluid across the test tube. Based on the above equations, the average heat transfer coefficient (h) can be calculated from $h=Q_c/A\left(\overline{T}_w-T_b\right)$ where A is the surface area of the test section. The Nusselt number can be expressed as $Nu=hD_i/k$. The friction factor can be calculated as:

$$f=\frac{D_i}{L}\frac{2\Delta P}{\rho u^2} \tag{6}$$

where u is the average velocity at the tube inlet, and ΔP is the pressure difference across the test section. The Reynolds number can be expressed as:

$$\mathrm{Re}=\frac{\rho u D_i}{\mu} \tag{7}$$

where μ is the dynamic viscosity of the working fluid flow inside heat exchanger tube equipped with DCTTs. All the thermo-physical properties of the working fluids are evaluated at the mean temperature. thermal performance of heat exchangers is an important parameter to analyze the performance of the systems in terms of both pressure drop and heat transfer enhancement. Thermal performance in this study is defined as [28]:

$$\eta = \frac{\left(Nu / Nu_s\right)}{\left(f / f_s\right)^{1/3}} \tag{8}$$

where s refers to the smooth tube without twisted tapes.

4. Uncertainty analysis

An uncertainty analysis was also performed for the measurement instruments as shown in Table 2. The measurement errors of instruments contain two different errors, an error which is provided by the manufacturer (calibration error) and a random estimation error.

Table 2 Equipment and their uncertainties.

Name of instrument	Range of instrument	Measurement	Uncertainty
K-Type thermocouple	-270 to1260 $^{\circ}C$	Wall temperatures	$\pm 0.1^{\circ}C$
TT-T-SLE thermocouple	-40 to 200 $^{\circ}C$	Bulk temperatures	$\pm 0.1^{\circ}C$
Rotameter	0 to 150 m^3/h	Water flow rate	±0.3%
Differential pressure transducer	0 to 5000Pa	Pressure drop	$\pm 2Pa$
Multi meter (Voltage& Ampere)	0-220V, 1-20A	Heating element	$\pm 0.5V$ $\pm 0.005A$
Analog Vernier	0 to 300 mm	Dimensions	$\pm 0.0005m$

Based on the methodology proposed by Moffat [29] the uncertainties of Re, Nu and f can be calculated as follows:

$$\frac{\delta \mathrm{Re}}{\mathrm{Re}} = \left[\left(\frac{\delta \rho}{\rho}\right)^2 + \left(\frac{\delta u}{u}\right)^2 + \left(\frac{\delta D_i}{D_i}\right)^2 + \left(\frac{\delta \mu}{\mu}\right)^2\right]^{0.5} \tag{9-a}$$

$$\frac{\delta \mathrm{Nu}}{\mathrm{Nu}}=\left[\left(\frac{\delta h}{h}\right)^2+\left(\frac{\delta D_i}{D_i}\right)^2+\left(\frac{\delta k}{k}\right)^2\right]^{0.5} \tag{9-b}$$

$$\frac{\delta f}{f}=\left[\left(\frac{\delta \Delta P}{\Delta P}\right)^2+\left(\frac{\delta D_i}{D_i}\right)^2+\left(\frac{\delta L}{L}\right)^2+\left(\frac{\delta \rho}{\rho}\right)^2+\left(2\frac{\delta u}{u}\right)^2\right]^{0.5} \tag{9-c}$$

The maximum uncertainties for calculated parameters are presented in Table 3.

Table 3 Uncertainties of calculated parameters

Parameter	Uncertainty
Reynolds number, (Re)	±3.4%
Nusselt number, (Nu)	±4.3%
Friction factor, (f)	±4.7%

5. Results and discussion

The Nusselt number and the friction factor for the tube without TT insert are compared with previous studies to validate the present experimental results. Fig. 3 shows that the plain tube results are in good agreement with data obtained from the Gnielinski and Blasius equations. The Nusselt number agrees within ±5.3% deviation from the correlation of Gnielinski, while the friction factors deviated within ±2.9% from the Blasius equation. The results show the reliability of the experimental system. The Gnielinski [30] and Blasius [31] equations frequently used in literature and valid for single-phase flow inside the plain tube are as follows:

$$Nu=0.012\left(\mathrm{Re}^{0.87}-280\right)Pr^{0.4}$$
$$\text{for } 1.5<Pr<500,\ 3\times10^3<\mathrm{Re}<10^5. \tag{10}$$

$$f=0.318\,\mathrm{Re}^{-0.25} \tag{11}$$

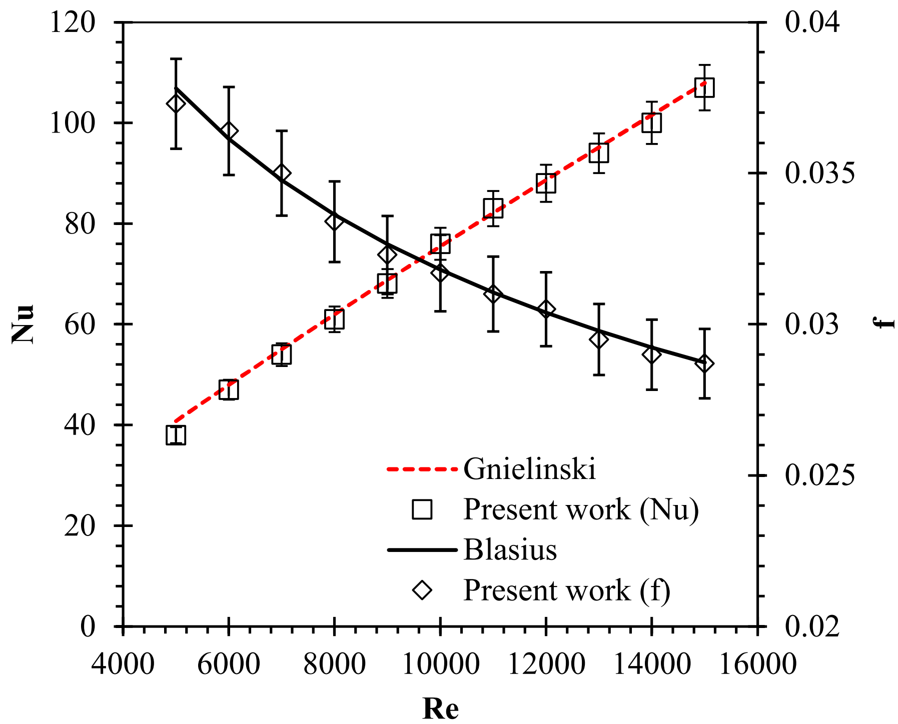


Fig. 3 Validation of experimental data

The effects of different cut ratios on the heat transfer enhancement inside the heat exchanger tube fitted with different DCTTs $(b/c = 0.25, 0.50, 0.75 \text{ and } 0.90)$ are presented in Fig. 4. The Nusselt number increases with the Reynolds number for all cases. As mentioned by Salam et al. [13], rectangular-cut twisted tapes can enhance swirl flows in the channel. The variations of Nusselt number can be explained by the fact that a higher flow disturbance is generated near the tube walls by rotational flow as the Reynolds number increases. This means that strong swirls increase the fluid mixing between the core and near the wall regions. The results show that the heat transfer performance inside tubes equipped by double-cut twisted tapes with larger cut ratios is far superior to the small cut ratio ones. It also should be pointed out that *b/c*=0.90 provides the best heat transfer performance between tested geometries, as shown in Fig. 4. The Nusselt number of DCTTs with *b/c*=0.25, 0.50, 0.75, 0.90 is 96.1%, 124.6%, 150.3% and 177.4%, respectively, higher than those of the plain tubes. This is because larger cuts create a better fluid mixing between the tube walls and the core region, and therefore, it greatly enhances the heat transfer performance.

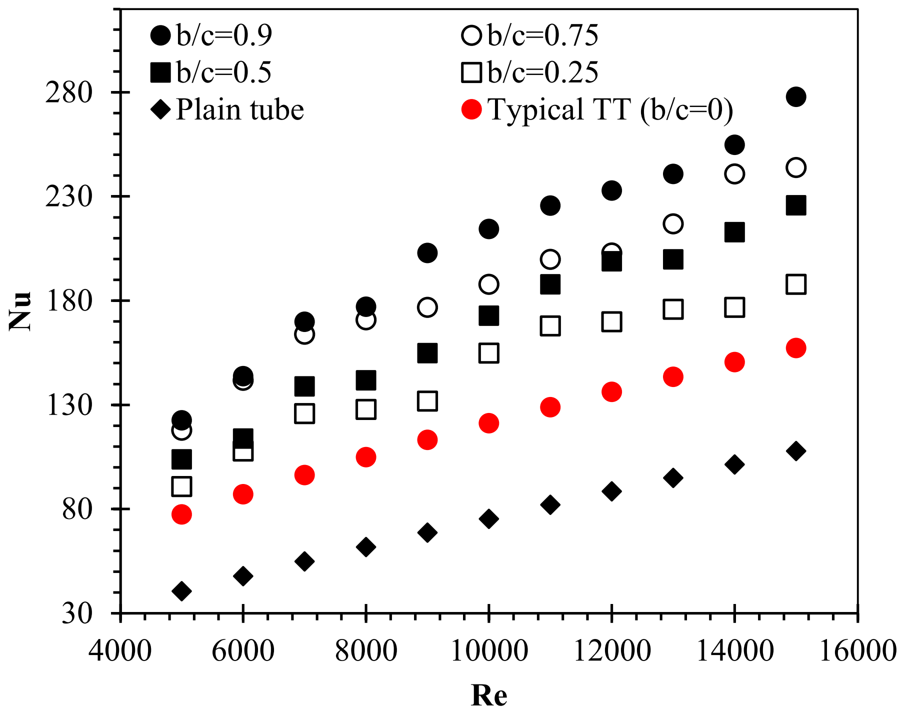


Fig 4. Variation in Nu with Reynolds number for tubes equipped with double-cut twisted tapes

Fig. 5 shows the variation of friction factor with Reynolds number for fluid flow inside heat exchanger tube fitted with DCTTs with different cut ratios. The results reveal that the f decreases with increasing Re from 5000 to 15000 for all cases. It can be observed that the friction factor of the fluid flow inside the tubes fitted by the DCTTs is increased by raising the cut ratio from 0.25 to 0.90, which is undesirable. This is mainly because of the higher pressure drop in the presence of twisted tapes with enlarged cuts. The main causes of higher flow resistance are: 1) long flow path between the tube walls and core region, 2) additional vortex generation in the presence of the DCTTs with larger cuts, and 3) strong turbulence and flow disturbance in the boundary layer region caused by swirl flows. Friction factor of the flows in the tubes with the DCTTs are increased up to 489% as compared to those in the plain tubes.

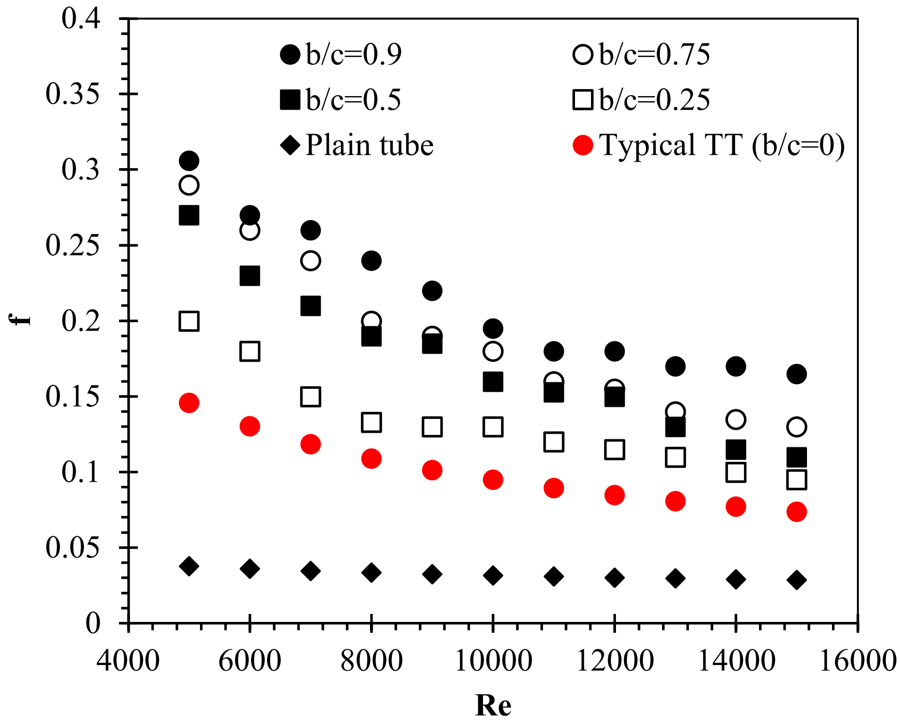


Fig 5. Variation in f with Reynolds number for tube equipped with the double-cut twisted tape

Fig. 6 presents the effect of Reynolds number and cut ratio of DCTTs on thermal performance factor (η) of the system. The thermal performance factor decreases with increasing Re number. It can be observed that η increases by increasing the cut ratio of DCTTs. This observation reflects that thermal performance becomes remarkable with the use of DCTTs with larger cut ratios due to stronger swirl flows. A similar trend for the thermal performance factor in this region was detected in the experimental study of He et al. [12] on cross hollow twisted tapes. The use of DCTT with b/c=0.25, 0.50, 0.75 and 0.90 (at $Re = 9000$) results in thermal performance of 1.16, 1.28, 1.33

and 1.44, respectively. It can be observed that the highest thermal performance of $(\eta = 1.63 \text{ at } \mathrm{Re} = 5000)$ is obtained from the DCTT with a cut ratio (b/c) of 0.90.

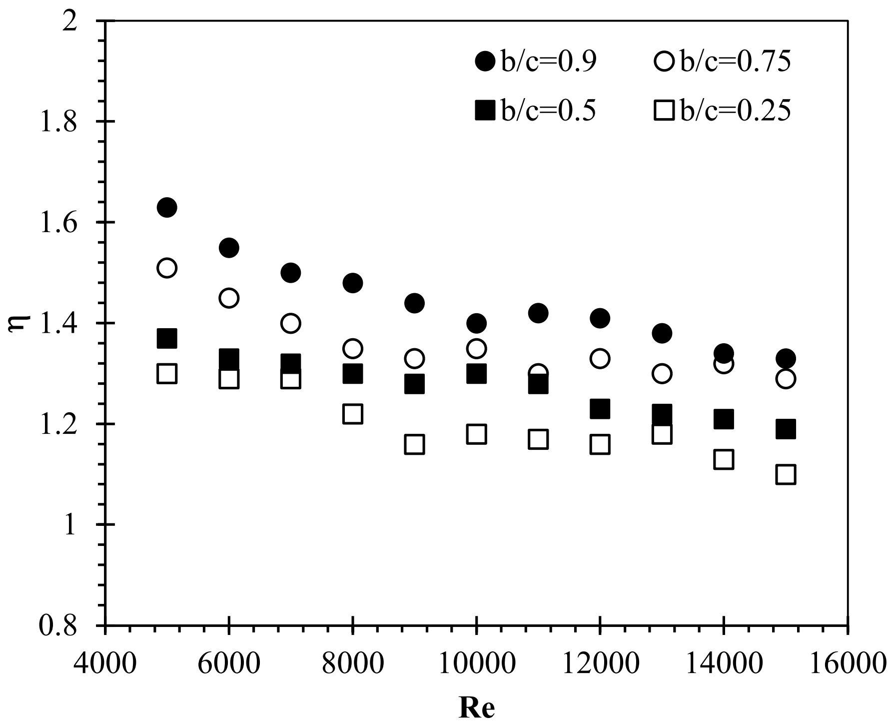


Fig. 6 Variation in η with the Reynolds number for tube equipped with double-cut twisted tape

The empirical correlations of the Nusselt number, the friction factor and thermal performance for heat exchanger tubes equipped with double-cut twisted tapes are also presented. The correlations are generated by curve fittings of experimental data. The results indicate that the Nusselt number is a function of cut ratio $(0 \le b/c \le 0.9)$, Reynolds number ($5000 \le \mathrm{Re} \le 15000$) and Prandtl number ($4.95 \le \mathrm{Pr} \le 6.51$). At the same time, the friction factor and thermal performance are affected by (*b/c*) and Reynolds number. Based on the mean average deviation equation ($MAD = \sum_{i=1} |x_i - \bar{x}| / n$), the mean and absolute deviations of the Nusselt number correlation are

1.21% and 4.75%, respectively. The mean absolute deviation of the experimental data is the average distance between each data point and the mean value.

$$Nu = 0.153\,\mathrm{Re}^{0.6345}\,Pr^{0.4}\left(1+b/c\right)^{0.831} \tag{12}$$

$$f = 28.19\,\mathrm{Re}^{-0.618}\left(1+b/c\right)^{1.127} \tag{13}$$

$$\eta = 4.75\,\mathrm{Re}^{-0.139}\left(1+b/c\right)^{0.172} \tag{14}$$

Figs. 7-9 presents comparisons between the current experimental data and the predicted Nu, f and η by using Eqs. (12-14) as functions of the Reynolds number, the Prandtl number and cut ratio of double-cut twisted tapes. Evidently, the correlations are capable to predict the Nu, *f* and η with maximum errors of ± 10%, 14%, 12% relative to the experimental data.

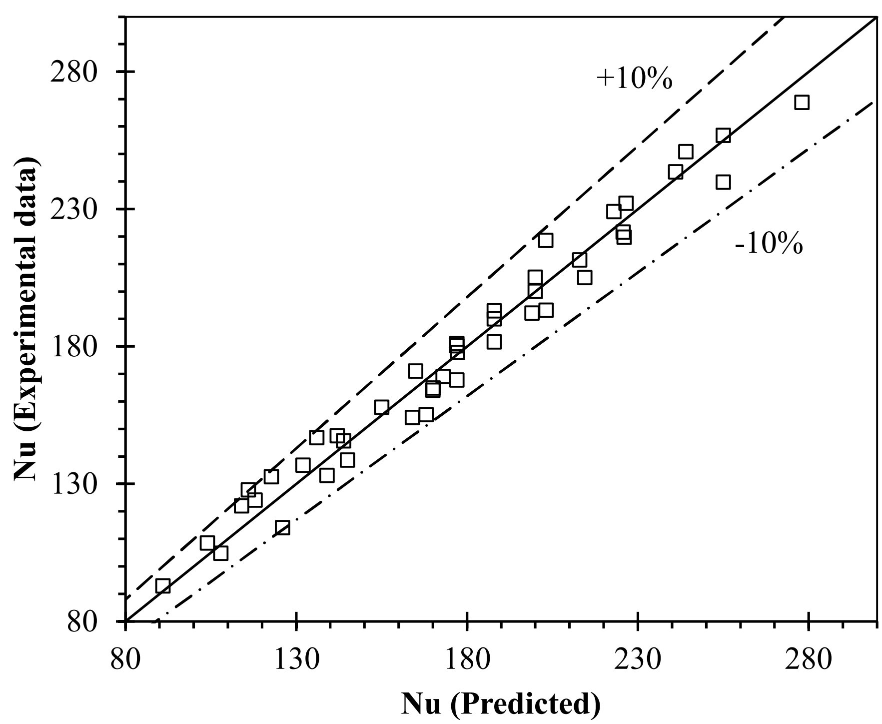


Fig. 7 Predicted Nusselt number (Nu) versus the experimental data.

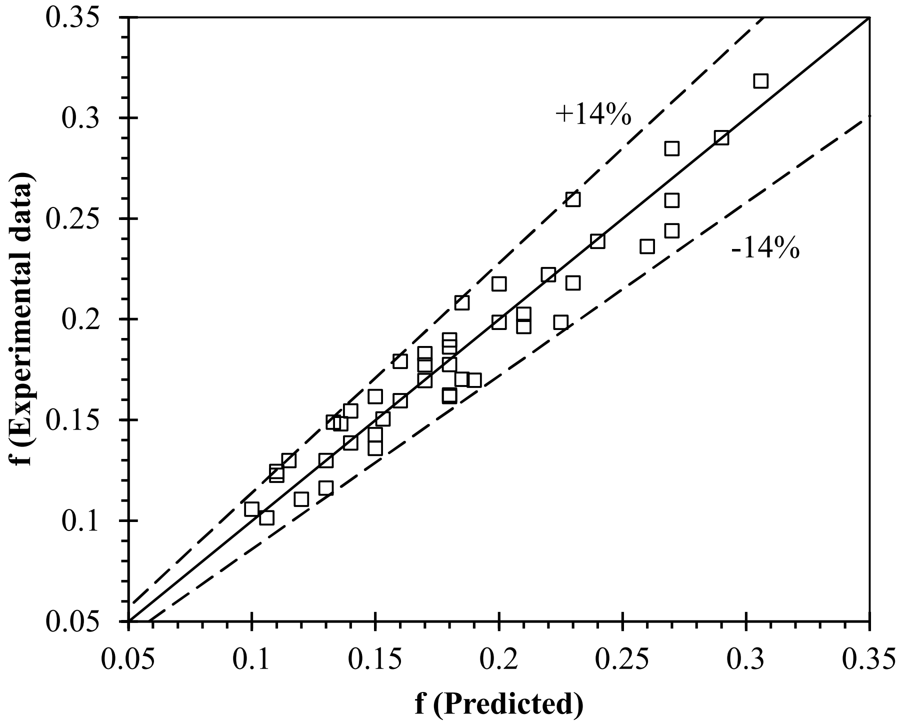


Fig. 8 Predicted friction factor (f) versus the experimental data.

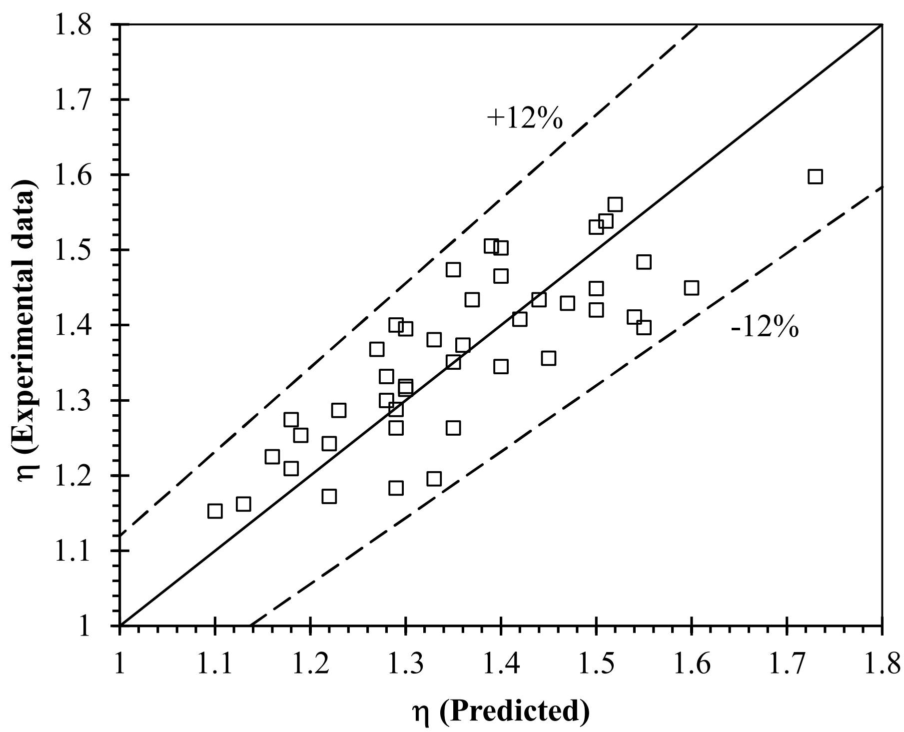


Fig. 9 Predicted thermal performance (η) versus the experimental data.

The proposed correlations (Eqs. 12-14) are able to predict the heat transfer and pressure drop inside heat exchanger tubes fitted with double rectangular-cut twisted tapes $\left(0.25 \le b/c \le 0.9\right)$, as well as typical twisted tapes without cuts (*b/c*=0). To validate the accuracy of the correlations with previous experimental correlations developed by other researchers for twisted tapes, the Nusselt number and friction factor are compared with proposed correlations of Maddah et al. [27] for the heat transfer coefficient (Eq. 15) and Eiamsa-ard et al. [19] for the friction factor (Eq. 16).

$$Nu = 0.056\,\mathrm{Re}^{0.72}\,\mathrm{Pr}^{0.4}\left(1+\pi\phi\right)^{2.75}\left(1+\frac{\pi}{2\left(y/w\right)}\right)^{1.1} GPR^{-0.75} \tag{15}$$

$$f = 24.8\,\mathrm{Re}^{-0.51}\left(y/w\right)^{-0.566}\left(1+\left(d/w\right)^{1.87}\right) \tag{16}$$

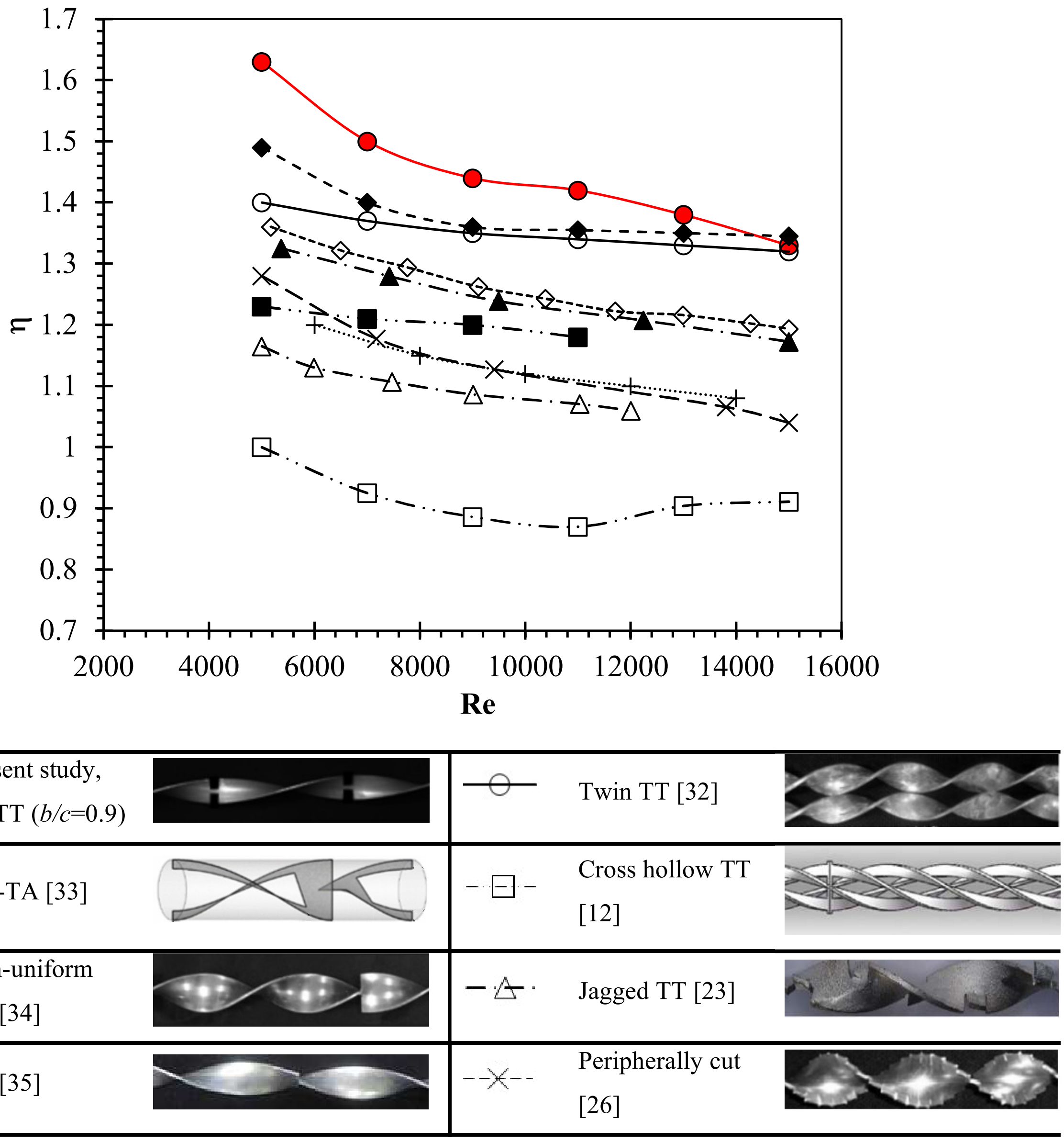
Where ϕ, *y/w*, *GPR*, and *d/w* are nanoparticles volume fraction, dimensionless twist ratio, geometrical progression ratio, and depth of wing cut ratio, respectively. By using *d/w*=0, *GPR* of 1, $\phi = 0$ (pure water), and non-dimensional twist ratio of *y/w*=5.25 in the above correlations, it would be possible to evaluate the heat transfer and friction loss inside tubes fitted by conventional twisted tapes without cuts. It should be pointed out that the working fluid in these two studies is water, and the design parameters are in the same range as the present study. The comparisons presented in table 4 illustrate that the heat transfer and friction factor predictions of the correlations developed in the present study are in excellent agreement with previous studies.

Table 4 Validation of the proposed correlations with previous studies for typical twisted tapes (*b/c*=0) for different Re numbers [19, 27].

	Nu			*f*		
Re	Present study Eq. (12)	Maddah et al. [27] Eq. (15)	Dev. %	Present study Eq. (13)	Eiamsa-ard et al.[19] Eq. (16)	Dev. %
7000	96.42	92.64	4.08	0.11852	0.10613	11.61
9000	113.34	111.02	2.08	0.10147	0.09336	8.65
11000	128.97	128.27	0.54	0.0896	0.08428	6.30
13000	143.60	144.67	0.74	0.08084	0.07739	4.54

15000	157.46	160.37	1.81	0.07401	0.07195	2.82

Fig. 10 shows a comparison between the thermal performances of the present study for the DCTT with *b/c*=0.9 with previous studies under same Reynolds number and Prandtl number. The thermal performance of the DCTT with *b/c*=0.9 in the present study is higher than those of the modified twisted tapes that were investigated in previous studies.



─■─	Square-cut TT [11]		┈┼┈	Multiple TT [14]	

Fig. 10 Comparison of thermal performance factor with previous experimental and numerical works

6. Conclusion

The effects of double-cut twisted tapes with cut ratios of 0.25, 0.5, 0.75 and 0.9 on the heat transfer and the friction factor of the fluid flow under turbulent flow regime through heat exchanger tubes are experimentally investigated and compared with conventional twisted tapes and plain tubes. The Reynolds number was in the range of 5000 to 15,000. The results show that:

- The Nusselt numbers of DCTTs with *b/c*=0.25, 0.50, 0.75, 0.90 are respectively 96.1%, 124.6%, 150.3% and 177.4% higher than those of the plain tubes.
- The best thermal performance of the fluid flow inside heat exchanger tubes is obtained by using a DCTT with b/c=0.90 $\left(\eta = 1.63 \text{ at } \mathrm{Re} = 5000\right)$.
- The friction factors of the tubes with DCTTs are increased 489% as compared with the plain tubes. This is mainly due to the long flow paths between the tube walls and core regions, more contacts between water flow and surfaces of DCTTs with larger cuts, strong turbulence, and flow disturbance in the boundary layer region.
- The thermal performance of the present improved twisted tape (DCTT) is higher than other modified twisted tapes previously investigated.
- The empirical correlations developed for heat transfer, the friction factor, and thermal performance are in a good agreement with experimental data.

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