Heat transfer enhancement by tapered twisted tape inserts

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Abstract

The effects of inserted tapered twisted tapes, their taper angle and twist ratio on heat transfer rate, pressure drop and thermal performance factor characteristics have been reported. The experiments were carried out by using the tapered twisted tapes with 4 different taper angles which $\theta = 0.0^\circ$ (typical twisted tape), 0.3$^\circ$, 0.6$^\circ$ and 0.9$^\circ$. At each taper angle, the tapered twisted tapes were twisted at three different twist ratios ($y/w$) of 3.5, 4.0 and 4.5. All tapes were tested under turbulent flow regime for Reynolds numbers between 6000 and 20,000. A twist ratio is defined as the ratio of twist length ($y$) to twisted tape width at the large end ($W$). The plain tube was also tested for comparison. Heat transfer enhancement and friction loss increased with decreasing taper angle and twist ratio. Thermal performance factor tended to increase with increasing taper angle and decreasing twist ratio. For the present range, the tube with the tape with taper angle ($\theta$) of 0.9$^\circ$ and twist ratio ($y/W$) of 3.5 yielded the maximum thermal performance factor of 1.05 at Reynolds number ($Re$) of 6000.

1. Introduction

Swirl generators have been extensively applied for convective heat transfer enhancement in several engineering and industrial applications such as: solar air/water heater, shell and tube heat exchanger, air conditioning, refrigeration, gas cooled nuclear reactor, chemical reactor, chemical and petrochemical industries, etc [1–3]. A twisted tape insert is one of the promising swirl generators for enhancing heat transfer for both laminar and turbulent flows. Heat transfer enhancement by twisted tape insert is attributed to its promoting the transverse mixing and producing swirl flow or vortex inside a heat exchanger leading to an efficient disruption of thermal boundary layer and breaking down the viscous sub-layer. In addition, a twisted tape is easily installed in an existing plain tube heat exchanger and cost-competitive.

Twisted tapes with various geometries were proposed and utilized in research works. Salman et al. [4] reported the effect of the quadrant-cut twisted tapes at different cut depths ($w = 0.5, 1.0$, and 1.5 cm) on the heat transfer enhancement characteristics in a circular tube. Their results indicated that heat transfer coefficient significantly increased with decreasing cut depth. Salman et al. [5] also studied the behaviors of heat transfer and friction factor in a tube equipped with twisted tape inserts with different alternative angles ($\theta = 30^\circ$, 60$^\circ$ and 90$^\circ$). For their studied range, heat transfer coefficient increased with increasing alternative angle. Murugesan et al. [6,7] examined the effect of square-cut and V-cut twisted tape inserts on the heat transfer enhancement, pressure drop and thermal performance characteristics. Zhang et al. [8] compared heat transfer in a converging-diverging tubes fitted with and without regularly-spaced twisted tape. Their results revealed that the tube with regularly-spaced tapes yielded considerably higher heat transfer coefficient than the one without twisted tape. Patil and Babu [9] employed twisted tapes with different twist ratios in a square duct and found that heat transfer and pressure loss were insignificantly affected by changing tape twist ratio. Vazifeshenas and Delavar [10] invented center-cleared twisted tapes for heat transfer enhancement. They found that the tube with center-cleared twisted tapes yielded considerably higher heat transfer coefficient than the plain tube alone. Liu and Bai [11] reported the formation and development of helical vortices by short twisted tapes and mentioned that vortex intensity increased with increasing swirl intensity.

In some research studies, tape inserts were in form of helical screw tape inserts. Most inserts in this form gave high heat transfer coefficients and thermal performance factors [12,13]. Sivashanmugam and Suresh [12] employed typical helical screw tapes with different at twist lengths ($y = 1.95, 2.93, 3.91$ and 4.89) for heat transfer enhancement in a circular tube. Their results found that heat transfer coefficient increased with decreasing twist length.

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Abbreviation: TTT: tapered twisted tape; TT: typical twisted tape.
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The comparative studies of heat transfer enhancement by modified twisted tapes and typical twisted tape were also reported. Rahimi et al. [14] compared heat transfer augmentation by modified twisted tapes (perforated, notched and jagged twisted tapes) with that by the typical one. Their results found that jagged twisted tape gave better heat transfer while perforated and notched twisted tapes gave poorer heat transfer than the typical one. Wongcharee and Eiamsa-ard [15] studied the effect of tape shape (triangle, rectangle and trapezoid twisted tapes) on the heat transfer enhancement and found that coefficient characteristics. Under similar conditions, the trapezoid tape gave the highest heat transfer coefficient. Eiamsa-ard et al. [16] modified the twisted tape by twisting a straight tape to form a twisted tape then bending the twisted tape into a helical shape to form "helically twisted tape". The helically twisted tape was subjected to a comparative test with a typical helical tape. The experimental results showed that the helically twisted tape gave lower heat transfer coefficient but higher thermal performance than the typical one. Recently, Jaisankar et al. [13] invented helical twisted tape in left-right arrangement. The tape induced bidirectional swirl flow, giving better heat transfer than the typical helical twisted tape which induced unidirectional swirl flow.

Eiamsa-ard et al. [17] further modified helically twisted tape by varying tape number (single, dual and triple-helical twisted tapes). The experimental results showed that the dual and triple-helical tapes yielded higher heat transfer coefficient than the single one. Bas and Ozceyhan [18] reported the heat transfer and pressure drop in a tube equipped with twisted tape inserts at different clearance ratios and twist ratios, by using Taguchi method. Their results indicated that heat transfer coefficient increased with the decreases of twist ratio and clearance ratio. Beigzadeh et al. [19] applied the hybrid model, including back propagation network and genetic algorithm, to predict the thermal and flow characteristics in a channel equipped with multiple twisted tapes while the multi-objective optimization with genetic algorithm was applied for the optimization.

For better heat transfer enhancement, twisted tapes were applied together with other heat transfer enhancement techniques [20–23]. Bhattacharyya and Saha [20] equipped centre-cleared twisted-tapes with a circular duct and found that the combined devices yielded significantly higher heat transfer enhancement than the circular duct alone. Similarly, Promvonge et al. [21] reported that the use of helical-ribbed tube equipped with twin twisted tapes resulted in noticeably higher heat transfer coefficient the use of the ribbed tube alone. Again, Promvonge et al. [22] examined the heat transfer characteristics of a square duct equipped with combined wigglet vortex generators and twisted tapes. Their results showed that the duct with combined wigglet vortex generators and twisted tapes gave higher heat transfer coefficient and thermal performance than the one with only twisted tapes. Khoshvaght-Alibadi and Eskandari [23] combined the effects of conductive and convective heat transfer enhancement by using Cu-water nanofluids as the working fluids in the tube equipped with twist tapes. Their experiments were carried out using the non-uniform twist tapes with different twist lengths (low–high, high–low, low–high–low, and high–low–high). They reported that the simultaneous use of the nanofluid and twisted tapes resulted in superior heat transfer than the individual use of a single technique. Among the studied tapes, the one with low to high twist lengths gave the maximum heat transfer coefficient. In the similar way, Chougule and Sahu [24] studied the heat transfer coefficient of tube equipped with helical twisted tape using Al2O3/water and CNT/water nanofluids at different particle volume fractions as the testing fluids. Their results showed that the use of the helical tape together with the nanofluids resulted in better heat transfer than the use of the tape or the nanofluids alone.

Based on the above literature review, the heat transfer enhancement by twisted tapes is strongly dependent on tape geometry. In common, a thermal performance factor as an overall heat transfer enhancement is dependent on both heat transfer coefficient and friction loss. The good heat transfer enhancement device should give a reasonable trade-off between the increased heat transfer coefficient and friction loss. The present investigation focuses on the developments of the twisted tape geometry with aims to reduce a pressure drop and increase a thermal performance factor. The tapered tapes proposed in the present work are expected to cause lower friction loss than a typical twisted tape, attributed to their smaller cross-section areas in axial direction. The effects of taper angles (θ = 0°, 0.5°, 0.6° and 0.9°) and twist ratios (γ/W = 3.5, 4.0 and 4.5) on the thermal enhancement factor are studied. The experiments are carried out in a turbulent regime for Reynolds numbers between 6000 and 20,000, under a uniformly heated wall condition. The experimental results of the heat transfer rate (Nu), pressure drop (f) and thermal performance obtained by using tapered twisted tapes were compared with those by the use of typical twisted tape inserts.
2. Experimental facility

2.1. Tapered twisted tapes

The details of the tapered and typical (or classic) twisted tapes applied in the present work are demonstrated in Fig. 1. All tapered twisted tapes (T-TTs) were made of aluminum strip with tape thickness (t) of 1.0 mm. The typical twisted tapes possessed a constant tape width (W) of 20 mm. On the other hand, tape width of tapered twisted tapes (T-TT) gradually reduced along the flow passage. To form tapered twisted tapes (T-TT) the edges of tapes were trimmed prior to a twisting process. The edges of tapes were trimmed with different trimmed widths, related to the taper angles as shown in the figure. The studied tapered twisted tapes possessed four various taper angles (θ = 0.0° (typical twisted tape), 0.3°, 0.6° and 0.9°) and three different twist ratios, y/W = 3.5, 4.0 and 4.5, respectively.

2.2. Test procedure

Experiments were performed in an open-loop experimental facility (as schematic diagram shown in Fig. 2). The system consisted of a circular copper tube, a high-pressure blower, an electrical power supply unit and measurement instruments and recorders. The copper tube had 64 mm inner diameter (D) and 67 mm outer diameter (D_o), 1.5 mm thickness (t), and 3900 mm length (L) which was divided into three parts: a calm section (2000 mm), a test section or heating section (1500 mm) and an exit (400 mm). The test section was heated by continually winding flexible electrical wire connected to the electrical power supply unit. The power supply unit was controlled via a variac transformer to obtain a constant heat flux condition by keeping the output current below 3 amps. The outer surface of the test tube was well insulated to minimize convection and radiation heat losses to surroundings. In addition, necessary precautions were taken to prevent leakages from the system. Wood bars with low thermal conductivity, were fitted around the copper tube to function as thermal barriers at the inlet and exit of the test section. The detailed description of experimental set up was previously performed by Ref. [25].

The uncertainty of the Nusselt number and friction factor data can be expressed as follows [26,27].

2.2.1. Nusselt number

\[
\frac{\Delta Nu}{Nu} = \left[ \left( \frac{\partial}{\partial t} (Nu) \Delta h \right)^2 + \left( \frac{\partial}{\partial t} (Nu) \Delta D \right)^2 + \left( \frac{\partial}{\partial t} (Nu) \Delta k \right)^2 \right]^{0.5}
\]

where \( h = \frac{q}{w - t_0} \)

\[
\frac{\Delta h}{h} = \frac{\left[ \left( \frac{\partial}{\partial t} \Delta T \right)^2 + \left( \frac{\partial}{\partial t} \Delta T_w \right)^2 + \left( \frac{\partial}{\partial t} \Delta T_b \right)^2 \right]^{0.5}}{\left( \frac{\partial}{\partial t} \Delta T \right)^2 + \left( \frac{\partial}{\partial t} \Delta T_w \right)^2 + \left( \frac{\partial}{\partial t} \Delta T_b \right)^2}
\]

where \( q = \frac{D}{mC_p(T_{bo} - T_{bi})} \)

2.2.2. Friction factor

\[
\frac{\Delta f}{f} = \left[ \left( \frac{\partial}{\partial t} (\Delta P) \right)^2 + \left( \frac{\partial}{\partial t} (\Delta D) \right)^2 + \left( \frac{\partial}{\partial t} (\Delta k) \right)^2 \right]^{0.5}
\]

where \( \frac{\Delta (\Delta P)}{\Delta P} = \frac{\Delta h}{h} \) and \( \frac{\Delta Re}{Re} = \left[ \left( \frac{\Delta m}{m} \right)^2 + \left( \frac{\Delta D}{D} \right)^2 \right]^{0.5} \)

The uncertainties of non-dimensional parameters were within ±6%, ±5% and ±8% for Reynolds number, Nusselt number and friction factor, respectively. The uncertainties of axial velocity, pressure and temperature measurements were within ±7%, ±5% and ±0.5%, respectively. The experimental results were reproducible within these uncertainty ranges.

3. Data reduction

In the present experiments, the inner surface of the test tube was maintained under uniform wall heat flux condition. The inlet fluid temperature (T_i) of the test tube was kept constant at 26 °C. During the test, air in the test tube received heat (Q_{air}) from the electrical heat wire mainly via the convective heat transfer mechanism. Thereby, the Q_{air} is assumed to be equal to the convective heat transfer within the tube wall which can be expressed as:

\[
Q_{air} = Q_{conv}
\]

The heat gained by air in term of enthalpy change can be written as:

\[
Q_{air} = mC_{p, air}(T_o - T_i)
\]
During the test, the heat equilibrium test performed that the heat received by the air \( Q_{\text{air}} \), was within 5% lower than the heat supplied by electrical heating \( Q_{\text{el}} = IV \), this is due to the heat loss from the test section
\[
\frac{\left(Q_{\text{el}} - Q_{\text{air}}\right) \times 100}{Q_{\text{el}}} < 5\% 
\]

The average value of heat absorbed by the fluid is taken for internal convective heat transfer coefficient calculation as:
\[
Q_{\text{conv}} = hA\left(\bar{T}_w - T_b\right) 
\]

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

The mean inner wall surface temperature \( \bar{T}_w \) of the test tube is calculated from surface temperatures of 15 stations located between the inlet and the exit of the test tube, using the following equation:
\[
\bar{T}_w = \frac{\sum T_w}{15} 
\]

where \( T_w \) is the local wall temperature, evaluated at the inner wall surface of the test tube.

The mean heat transfer coefficient can be determined by:
\[
Q_{\text{air}} = Q_{\text{conv}} = \dot{m}C_{p,\text{air}}(T_o - T_i) = hA\left(\bar{T}_w - T_b\right) 
\]

The mean tube-side convection heat transfer coefficient \( (h) \) can be determined using the following equation:
\[
h = \frac{\dot{m}C_{p,\text{air}}(T_o - T_i)}{A(\bar{T}_w - T_b)} 
\]

It should be noted that, the thickness of tubes (0.5 mm) is negligible, conduction resistance is small in comparison with the convective resistance. The mean Nusselt number based on the diameter of the inner tube can be evaluated as:
\[
Nu = \frac{hD}{k} 
\]

The pressure loss \( \Delta P \) of the turbulent flow in the test section can be directly measured. The friction factor in term of pressure loss across the test section length \( (L) \) calculated from a difference in the level of a manometer liquid is acquired under an isothermal flow condition. Substituting the pressure loss value into the following equation yields the Darcy friction factor \( (f) \) as:
\[
f = \frac{\Delta P}{\left(L/D\right)(\rho U^2/2)} 
\]

The Nusselt number enhancement index is defined as the ratio of Nusselt number of the system with a heat transfer enhancement device (here is the tube equipped with tapered twisted tape: T-TT) that without a heat transfer enhancement device (here is a plain tube, p).
\[
N = \frac{Nu}{Nu_p} 
\]

Similarly, the friction factor enhancement index is defined as:
\[
F = \frac{f}{f_p} 
\]

The mechanical energy consumption factor in term of a thermal performance factor is defined as the ratio of convective heat transfer coefficient of the system with a heat transfer enhancement device to that without a heat transfer enhancement device. The ratio is based on the operation at an identical pumping power. Based on the definition mentioned above, the thermal performance factor can be expressed in terms of Nusselt number
enhancement index and friction factor enhancement index as 

$$\eta = \left( \frac{\text{Nu}}{\text{Nu}_p} \right) \left( \frac{f}{f_p} \right)^{1/2}$$

(14)

The flow regime is indicated by Reynolds number value.

$$Re = \frac{\rho UD}{\mu}$$

(15)

In the present study, the studied Reynolds numbers are between 6000 and 20,000. Air is used as the testing fluid which is assumed to be incompressible. The thermos-physical properties of testing fluid (air) are assumed to be temperature independent. The specific heat at constant pressure ($C_p$), thermal conductivity ($k$), density ($\rho$) and dynamic viscosity ($\mu$) of air are based on the bulk flow temperature which is the mean value of the inlet and outlet temperatures ($T_0 = (T_i + T_o)/2$).

4. Confirmation of the plain tube

In order to ensure the accuracy and reliability of the experimental setup, the experimental results (Nusselt numbers and friction factors) of the present plain tube data were compared with those achieved from the standard correlations [28]. Experimental Nusselt numbers were compared with those calculated from Dittus-Boelter correlation (Dittus-Boelter, 1930) and Petukhov correlation (Petukhov, 1970) as shown in Eqs. (16) and (17), respectively, while the experimental friction factor were compared with those from Petukhov correlation as shown in Eq. (18). The comparisons between the experimental results of the present plain tube and those from the standard correlations for the fully developed turbulent flow are demonstrated in Fig. 3(a and b). Apparently, the Nusselt numbers of the present plain tube deviated from those of Dittus-Boelter and Petukhov correlations within ±8% and ±7.4%, respectively, while experimental friction factor results deviated from those of within ±9.2%. The comparisons indicate that the present experimental data accord well with the correlations, confirming the reliability of the present experimental facility and method.

4.1. Nusselt number correlations

Dittus-Boelter correlation:

$$\text{Nu} = 0.023Re^{0.8}Pr^{1/3}$$

for $Re > 1 \times 10^4$

(16)

Petukhov correlation:

$$\text{Nu} = \frac{(f/8)RePr}{1.07 + 12.7(f/8)^{1/2}}(Pr^{2/3} - 1)$$

for $10^4 < Re < 5 \times 10^6$

(17)

4.2. Friction factor correlations

Petukhov correlation:

$$f = (0.79InRe - 1.64)^{-2}$$

for $3000 \leq Re \leq 5 \times 10^6$

(18)

5. Results and Discussion

The experimental results of heat transfer (Nusselt number, $Nu$), pressure drop (friction factor, $f$) and thermal performance factor ($\eta$) in a tube with tapered twisted tapes (T-TTs) are reported in the present section. The effects of taper angles ($\theta = 0.0^\circ$ (typical twisted tape), $0.3^\circ$, $0.6^\circ$ and $0.9^\circ$) and twist ratios ($y/W = 3.5$, $4.0$ and $4.5$) of the tapered tapes in a turbulent region of Reynolds number, $Re = 6000–20,000$ are described. The results of the plain tube are also provided for evaluation.

5.1. Effect of the taper angle and twist ratio on heat transfer

The variations of heat transfer in term of Nusselt number ($Nu$) with Reynolds number ($Re$) and Nusselt number enhancement index ($Nu/Nu_p$) with Reynolds number of the studied cases are demonstrated in Fig. 4(a and b), respectively. Nusselt number increased with the rise of Reynolds number for all cases. At a given Reynolds number the tubes with tapered twisted tapes (T-TTs) yielded higher Nusselt number than the one without tape (the plain tube). The superior heat transfer in the tubes with tapered twisted tapes (T-TTs) is attributed to 2 important factors (1) the swirl flow generated by the inserted tapes leads to a stronger turbulence intensity and (2) the decrease of cross-sectional area of a flow passage due to the twisted tape blockage, leads to an increase of fluid flow velocity. At similar conditions, heat transfer tended to decrease with the increase of taper angle. This can be explained that as the taper angle increases, the swirl intensity induced by a tapered twisted tape become weaker, resulting in poorer fluid mixing between wall and core regions. The experimental results in Fig. 4 showed that the tubes with tapered twisted tapes (T-TTs) with $\theta = 0.0^\circ$ (the typical twisted tape), $0.3^\circ$, $0.6^\circ$, $0.9^\circ$.
0.6° and 0.9° gave mean Nusselt numbers around 1.43, 1.40, 1.38 and 1.33 times of that given by the plain tube. In other words, the tapered twisted tape (T-TT) with θ = 0.0° offered higher heat transfer than those with θ = 0.3°, 0.6° and 0.9° by around 2.09%, 3.49% and 6.99%, respectively.

Fig. 4(a) also shows the effect of twist ratio (y/W) on the heat transfer (Nu). Nusselts number tended to increase with decreasing twist ratio or twist length (y). The twisted tape with smaller twist ratio possesses more twist numbers, thus induces more consistent swirl flow with stronger swirl intensity. This leads to higher turbulent intensity and thus better heat transfer. For the studied range, the use of the tapered twisted tapes (T-TTs) with twist ratio y/W = 3.5 gave higher Nusselt number than the ones with y/W = 4.0 and 4.5 by around 3.75%, and 9.99%, respectively.

The results in Fig. 4(b) also reveal that as Reynolds number decreases, heat transfer enhancement by inserting tapered twisted tape (T-TT) becomes more significant indicated by the increase of Nusselt number enhancement index (Nu/Nu_p). These results related to the dominant destruction of the thermal/velocity boundary layer in laminar regime in which the thermal boundary layer is relatively thick. It is noteworthy that the slope of Nu/Nu_p ratio–Reynolds number curve is dependent on enhancement device.

5.2. Effect of the taper angles and twist ratio on friction factor

The variations of pressure drop in term of friction factor (f) with Reynolds number (Re) and friction factor enhancement index (ff_p) with Reynolds number of the studied cases are demonstrated in Fig. 5(a) and (b), respectively. In all cases, friction factor and friction factor enhancement index tended to decrease with increasing Reynolds number (Re). For the studied range, the use of the tubes equipped with tapered twisted tapes (T-TTs) caused higher friction factor than that of the plain tube by 154.6–293%. This is attributed to the dissipation of dynamic pressure of the fluid due to the very high viscosity loss near the tube wall and the interaction of the

![Fig. 5. Effect of taper angle and twist ratio on friction factor (f).](image-url)
pressure forces with inertial forces in the boundary layer caused by the tapered twisted tapes (T-TTs).

For the tubes with tapered twisted tapes (T-TTs), friction factor decreased as taper angle increased. The friction factors caused by the tapered twisted tape (T-TT) with \( \theta = 0.9^\circ \) were lower than those caused by the ones with \( \theta = 0.0^\circ, 0.3^\circ, \) and \( 0.6^\circ \) by around 41.97\%, 30.6\% and 14.94\% respectively. This is due to the lower flow hindrance caused by the smaller surface area of the tape with larger taper angle. In addition, the friction factors of the tubes with the tapered twisted tape (T-TT) at taper angle of \( \theta = 0.0^\circ, 0.3^\circ, 0.6^\circ \) and \( 0.9^\circ \) are as high as 2.93, 2.69, 2.37 and 2.06 times of the plain tube, respectively.

Fig. 5 also shows that the friction factor tended to increase with decreasing twist ratio \( (y/W) \) for all Reynolds number studied. The use of the tapes with twist ratio \( (y/W) \) of 4.0 and 4.5, resulted in favorable reduction of friction loss as comparison with the one with the smallest twist ratio of 3.5, due to the lower flow resistance caused by the weaker swirl intensity. In the present range, the mean friction factors caused by the tubes with the tapered twisted tapes (T-TTs) at twist ratios \( (y/W) \) of 3.5, 4.0 and 4.5 were around 2.93, 2.73 and 2.57 times of that of the plain tube.

5.3. Effect of the taper angles and twist ratio on thermal performance

One major aim of the present report is to evaluate the heat transfer enhancement in term of thermal performance at the equal pumping power (power input). The effect of tapered twisted tape (T-TT) swirl generator on the thermal performance factor is depicted in Fig. 6. Apparently, thermal performance factor tended to decrease with increasing Reynolds number. This indicates that the use of tapered twisted tapes (T-TTs) is more favorable at lower Reynolds number.

At a given Reynolds number, thermal performance factor increased with increasing taper angle. As reported in Sections 5.1 and 5.2, a tapered twisted tape (T-TT) with larger taper angle gave lower Nusselt number and friction factor. Thus, the superior thermal performance factor is due to the dominant effect of the lower friction factor (see Eq. (14)). This signifies that the reduction of friction loss by a proper design of tapered twisted tape (T-TT) insert is extremely important for optimizing heat transfer enhancement condition. For the present range, the thermal performance factors achieved by the use of the tapered twisted tapes (T-TTs) with \( \theta = 0.0^\circ, 0.3^\circ, 0.6^\circ \) and \( 0.9^\circ \) were found to be 0.89 to 0.99, 0.91 to 1.01, 0.93 to 1.03 and 0.94 to 1.05, respectively, depending upon Reynolds number and twist ratio. The comparison shows that the tapered twisted tapes (T-TTs) with taper angle \( (\theta) \) of \( 0.9^\circ \) offered 4.49\%, 3.76\% and 1.58\% higher mean thermal performance factor than those of \( \theta = 0.0^\circ, 0.3^\circ, \) and \( 0.6^\circ \), respectively.

Fig. 6 also shows that thermal performance factor increased with decreasing tape twist ratio. Evidently, the tapered twisted tapes (T-TTs) with \( y/W = 3.5 \) gave 4.97\%, 4.67\% and 4.62\% higher mean thermal performance factor than those with \( y/W = 4.0 \) and 4.5, respectively. For the present range, the tube with the tape with taper angle \( (\theta) \) of \( 0.9^\circ \) and twist ratio \( (y/W) \) of 3.5 yielded the maximum thermal performance factor of 1.05 at Reynolds number of 6000. The thermal performance factor result suggests the favorable tape geometry for heat transfer enhancement is the small twist ratio and large taper angle, while the favorable condition is low Reynolds number.

5.4. Empirical correlations

The statistical correlations for Nusselt number \( (Nu) \), friction factor \( (f) \) and thermal performance factor \( (\eta) \) of the tube with tapered twisted tapes (T-TTs) were developed, using experimental data with the aid of least square regression analysis, as a function of tape geometry (taper angle and twist ratio) and flow (Reynolds number). The resultant correlations are expressed in Eqs. (19)–(21). The predicted data from the resultant correlations of the \( Nu_{\text{pred}} \) and \( \eta_{\text{pred}} \) are plotted against experimental data of the \( Nu_{\text{exp}}, f_{\text{exp}} \) and \( \eta_{\text{exp}} \) in Figs. 7–9 (a–c). The deviations of the
predicted data from the experimental ones were within ±5% for Nusselt number, ±5% for friction factor and ±10% for thermal performance.

\[ Nu = 0.076Re^{0.75}Pr^{0.4}(y/W)^{-0.39}(1 + \theta)^{-0.1} \]  

(19)

\[ f = 16.559Re^{-0.49}(y/W)^{-0.51}(1 + \theta)^{-0.53} \]  

(20)

\[ \eta = 1.871Re^{-0.04}(y/W)^{-0.22}(1 + \theta)^{-0.08} \]  

(21)

5.5. Comparison with other tape inserts

For benchmarking, the thermal performance factor of the tapered twisted tape (T-TT) with the best performance \((y/w = 3.5\) and \(\theta = 0.9^\circ\)) in the present study were compared with those of the other tapes in the previously published papers. The other tapes subjected to the comparison include two regularly spaced tapes (regularly spaced short-length twisted tape by Wang et al. [2] and regularly spaced twisted tape by Eiamsa-ard et al. [30]) and three modified twisted tape (serrated twisted tape by Chang et al. [29], notched twisted tape by Rahimi et al. [14] and perforated twisted tape by Rahimi et al. [14], as shown in Fig. 10. The comparison of thermal performance factors shown in Fig. 11, indicates that the T-TT gave superior thermal performance to the notched twisted tape [14], perforated twisted tape [14], serrated twisted tape [29], and regularly spaced twisted tape [30]. The better thermal performance of the T-TT is primarily attributed to the lower pressure loss. On the other hand, the T-TT gave slightly lower thermal performance factor than the regularly spaced short-length twisted tape [2] at the low Reynolds number. However, the opposite result is found at high Reynolds number (e.g. 15,000), since the T-TT possesses more effective heat transfer enhancement with low pressure loss \((\eta/f_p = 1.54-2.26)\) at such a high Reynolds number. In addition, the thermal performance of the tapered twisted tape (T-TT) is better than that of the typical twisted tape (TT) depending on the twist ratio, as demonstrated in Fig. 6.

[Fig. 8. Comparison of experimental data with correlation prediction in term of friction factor.]

[Fig. 9. Comparison of experimental data with correlation prediction in term of thermal performance factor.]

[Wang et al. [2] regularly spaced short-length twisted tape]

[Eiamsa-ard et al. [30] regularly spaced twisted tape]

[Chang et al. [29] serrated twisted tape]

[Rahimi et al. [14] perforated twisted tape]

[Rahimi et al. [14] notched twisted tape]

[Fig. 10. Photograph and picture view of the previously modified twisted tape.]
6. Conclusions

An experimental investigation of thermal and friction characteristics in a tube equipped with tapered twisted tube (T-TT) under uniform wall heat flux conditions were carried out. The effects of taper angle ($\theta = 0.0^\circ$, $0.3^\circ$, $0.6^\circ$ and $0.9^\circ$) and twist ratio ($y/W = 3.5$, 4.0 and 4.5) were also studied. The experimental results are reported in terms of heat transfer coefficient ($Nu$), friction factor ($f$) and thermal performance factor ($\eta$). The major findings can be summarized as follows:

(1) The use of the tubes equipped with tapered twisted tapes (T-TTs) resulted in better heat transfer than the use of the plain tube.

(2) Heat transfer enhancement and friction loss increased with decreasing taper angle. The tapered twisted tube (T-TT) with $\theta=0.9^\circ$ gave lower mean Nusselt number than the ones with $\theta=0.0^\circ$, $0.3^\circ$ and $0.6^\circ$ by around 2.09%, 3.49% and 6.99%, respectively. This can be explained that as the taper angle increases, the swirl intensity induced by a twisted tapered tube become weaker, resulting in poorer fluid mixing between wall and core regions. The mean friction factor caused by the tapes with $\theta=0.9^\circ$ was lower than those of the ones with $\theta=0.0^\circ$, $0.3^\circ$ and $0.6^\circ$ by around 41.97%, 30.6% and 14.94%, respectively.

(3) Heat transfer enhancement and friction loss increased with decreasing twist ratio. The tapered twisted tube (T-TT) with $y/W=4.5$ gave lower mean Nusselt number than the ones with $y/W=3.5$ and 4.0 by around 10.98% and 6.05%, respectively. It can be explained that the twisted tube with smaller twist ratio possesses more twist numbers, thus induces more consistent swirl flow with stronger swirl intensity. This leads to higher turbulent intensity and thus better heat transfer. It is also seen that the mean friction factor caused by the tapes with $y/W=4.5$ was lower than those of the ones with $y/W=3.5$ and 4.0 around 15.7% and 6.22%, respectively.

(4) Thermal performance factor tended to increase with increasing taper angle and decreasing twist ratio. For the present range, the tube with the tape with taper angle ($\theta$) of 0.9$^\circ$ and twist ratio ($y/W$) of 3.5 yielded the maximum thermal performance factor of 1.05 at Reynolds number of 6000.

References


