(Invited Paper)

Turbulent Heat Transfer and Friction Factor in Circular Tube Fitted with Opposite/Parallel Rectangular Wing Twisted-Tapes

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ABSTRACT

The influences of rectangular wing twisted-tape inserts on heat transfer and friction factor behaviors in a heat exchanger tube are reported. The rectangular wing twisted tapes with a constant twist ratio (y/W), were arranged in two different forms: (1) opposite rectangular wing twisted-tape (OW-T) and (2) parallel rectangular wing twisted-tape (PW-T). The experiments were performed using water as a working fluid under a uniform wall heat flux condition, for the Reynolds number range between 5500 and 22,000. The plain tube and the tube with typical twisted tape (TT) were also tested at the same operating conditions, for comparison. The experimental results reveal that the heat transfer rate, friction factor and thermal performance factor of the OW-T and the PW-T are considerably higher than those of the TT. For the range examined the increases of mean Nusselt numbers in the tubes equipped with the OW-T, PW-T and TT, are, respectively, found to be 110%, 100% and 27.5% over the plain tube with no insert. The average thermal performance of the OW-T is higher than those of the TT and PW-T at 24.9% and 2.74%, respectively and the maximum thermal performance factors by the TT, PW-T and OW-T are 0.96, 1.36 and 1.4. In addition, the correlations of the Nusselt number, friction factor and thermal performance factor of the tubes with the OW-T and PW-T are also developed in terms of Reynolds number (Re) and Prandtl number (Pr).

Keywords: *Turbulent flow, Heat transfer enhancement, Twisted tape, Thermal performance factor*

1. INTRODUCTION

A high cost of energy and material has motivated several attempts to produce more efficient heat exchange equipment by improving convective heat transfer in the heat exchanger tubes. Heat transfer enhancement can be attained by an active technique which needs external power source or alternatively by a passive technique which does not need external power source. Both active and passive techniques have been applied to improve heat transfer in several areas such as nuclear reactors, chemical reactors and general heat exchangers. The principle of the passive technique involves either surface treatment, such as a coated surface, a rough surface and an extended surface or a flow manipulation such as a swirl flow or other modified flow. One of the most favorable passive techniques is generating swirl flow by insertion of a twisted tape because the tape is inexpensive and can be easily assembled to an existing system. The presence of twisted tape directs toward reducing the hydrodynamic or thermal boundary layer thickness, leading to the superior convective heat transfer. However, the desired heat transfer improvement is usually accompanied by an undesired increase of friction loss and thus a pumping power. The overall performance of twisted tape is usually evaluated from both heat transfer rate and friction factor in the form of thermal performance factor which depends strongly on the geometries and arrangements of the tapes. Therefore, to achieve a desired heat transfer rate in an existing heat exchanger at an economic pumping power, the design of twisted tape with a proper geometry (Table 1) and arrangement is necessary.

Many twisted tapes with different designs have been proposed [1] – [13]. Chang et al. [1], [2] studied the effect of the broken and serrated twisted tapes on the heat transfer and pressure drop characteristics in a circular tube. In spite of its superior heat transfer rate, the serrated twisted tape gave lower thermal performance factor than the broken one, owing to the prominent effect of its high friction factor. Rahimi et al. [3] proposed three different modified twisted tapes including perforated, notched, and jagged twisted tapes. Among the twisted tapes examined, the jagged twisted tape offered the highest thermal performance factor, due to its high heat transfer aided by the additional flow disturbance around the edge of the tape.

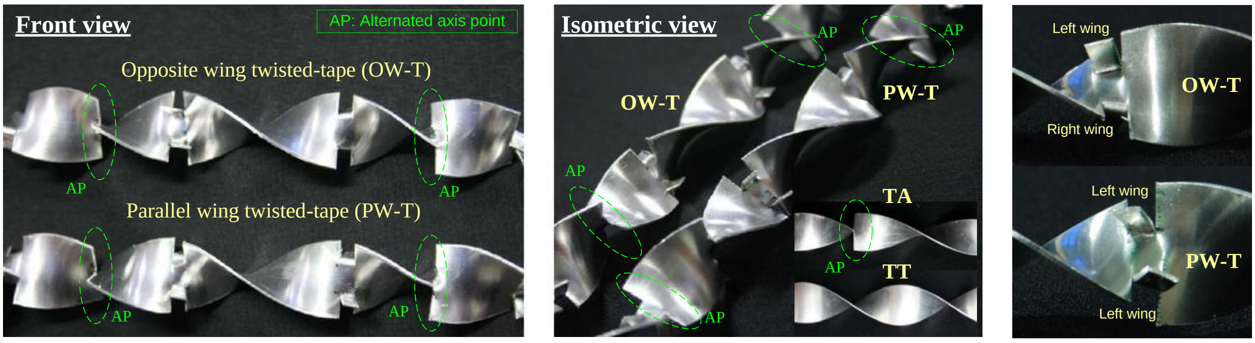


Fig. 1 *Geometry of rectangular wing twisted-tapes.*

Eiamsa-ard et al. [4], [5] introduced a loose-fit tape and a short length tape with prospect to reduce friction loss and improve thermal performance factor with respect to those of a typical tape. However, the modified twisted tape inefficiently enhanced heat transfer, resulting in a lower thermal performance factor than that given by the typical twisted tape. Alternatively, Eiamsa-ard et al. [6] – [11] employed the modified twisted tapes in different forms (clockwise-anti clockwise tape, peripherally-cut tape, delta-wing tape and dual/twin tapes) which provided an extra fluid mixing compared to that induced by a typical one. It was found that all modified twisted tapes successfully increased heat transfer rate and thermal performance above those of a typical tape. Murugesan et al. [12] investigated the heat transfer, friction factor and thermal enhancement factor characteristics in a double pipe heat exchanger fitted with a square-cut twisted tapes (STT) for additional disturbance and secondary flow in the vicinity of the tube wall. They observed that the heat transfer, friction factor and thermal enhancement factor of a tube with STT were respectively, 1.03 to 1.14, 1.05 to 1.25 and 1.02 to 1.06 times of those in a tube with typical twisted tape. Recently, Saha [13] reported that a full-length and a short-length twisted tape with oblique teeth in combination with axial corrugations showed marginal heat transfer improvements over a typical twisted tape. In the present, the heat transfer enhancement technique can improve the performance in many engineering system applications such as heat exchanger system, vortex tube, refrigeration, gas turbine, drying process, nuclear reactor, fuel cell, and combustor [14] – [21].

In the present work, the newly designed twisted-tapes are proposed. The twisted-tapes with a constant twist ratio (y/W) of 3.0 consisted of rectangular wings located on the edges of the tapes with two different arrangements: parallel and opposite bended wings. To generate an extra fluid mixing, alternate-axes were also formed on the tape. The tests using plain tube and tube with typical twisted-tape (TT) were also conducted, for comparison.

2. EXPERIMENTAL DETAILS

2.1 Twisted-tape

The *geometry of rectangular wing twisted-tape* used in the present study is shown in Fig. 1. All of the tapes used were made of aluminum strip with 0.8 mm thickness (t) and 20 mm width (W). The modified twisted-tapes were prepared by forming rectangular wings on the edges of tapes. As shown in the figure at every twist length (y, 180°), a pair of wings or alternate axis was formed (one pairs of wings followed by one alternate axis). All wings were bended to arrange the angle of attack to be 90 degree.

2.2 Experimental set-up

A schematic diagram of the apparatus with the basic components and fluid flow systems is presented in Fig. 2. The loop rig consisted of a 0.5 hp centrifugal water pump, Rota-meter for measurement of volumetric water flow rate, and the heat transfer test section. The copper test tube had an inside diameter of 19 mm, an outside diameter of 22 mm, thickness (δ) of 1.5 mm and length (L) of = 1000 mm. During the experiment, the test tube was heated by continually winding flexible electrical wires. The electrical output power was controlled by a variac transformer to obtain a constant heat flux along the entire length of the test section by keeping current to be less than 9 amps. The outer surface of the test tube was well insulated to minimize convective heat loss to surroundings, and necessary precautions were taken to prevent leakages from the system.

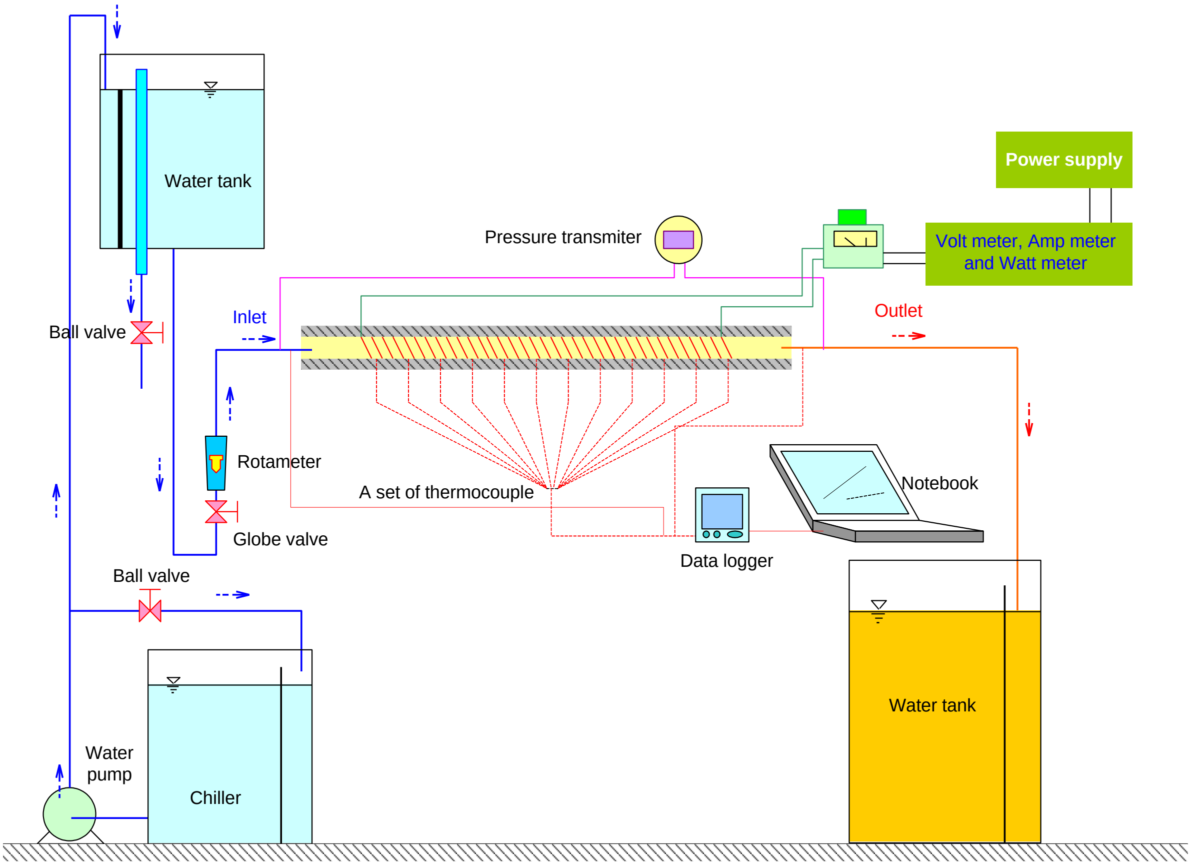


Fig. 2 *Schematic diagram of the experimental heat transfer set-up.*

2.3 Test condition and method

In the apparatus setting above, the inlet cold water at 27^{o}C was pumped through the rotameter and passed to the heat transfer test section. The pressure drop of the heat transfer test tube was measured with a pressure transducer. The volumetric water flow rate was varied via globe valve located upstream of the test tube. The inner and outer temperatures of the water were measured using a data logger unit in conjunction with the RTD PT 100 type temperature sensors. Fifteen thermocouples were tapped on the local wall of the tube and the thermocouples were placed round the plain tube to measure the circumferential temperature variation, which is found to be negligible. The mean local wall temperature was determined based on the reading of copper-constantan thermocouples. In each test run, the data of temperature, volumetric flow rate and pressure drop of the water were recorded at steady state condition in which the inlet water temperature was maintained at 27^{o}C. The Reynolds number of the water was varied from 5500 to 22,000.

Uncertainties of measurements are estimated based on ANSI/ASME. The uncertainties of axial velocity, pressure and temperature are found to be within ±7%, ±5% and ±0.5%, respectively. In addition, the uncertainties of non-dimensional parameters are ±5% for Reynolds number, ±10% for Nusselt number and ±12% for friction.

3. DATA REDUCTION

During the test, water in the test section receives heat (Q_{water}) from the electrical heat wire mainly via the convective heat transfer mechanism. Thereby, the Q_{water} is assumed to be equal to the convective heat transfer within the test section which can be written as equal to the convective heat transfer within the test section which can be written as

$$Q_{water} = Q_{convection} \quad (1)$$

Where

$$Q_{water} = \dot{m}C_{p,water}(T_o - T_i) = VI \quad (2)$$

The heat supplied by electrical winding in the test tube is found to be 4 to 8% higher than the heat absorbed by the fluid for thermal equilibrium test due to convection and radiation heat losses from the test section to surroundings. Thus, only the heat transfer rate absorbed by the fluid is taken for internal convective heat transfer coefficient calculation. The convection heat transfer rate from the test section can be written by:

$$Q_{convection} = hA(\tilde{T}_{water} - T_{bulk}) \quad (3)$$

Where

$$T_{bulk} = (T_{outlet} + T_{inlet})/2 \quad (4)$$

and

$$h = \dot{m}C_{p,water}(T_{outlet} - T_{inlet})/A(\tilde{T}_{wall} - T_{bulk}) \quad (5)$$

The average surface temperature $\tilde{T}_{wall}$ is calculated from 15 points of T_{wall} (the local surface temperatures at the outer wall of the inner tube) which lined between the inlet and the exit of the test tube.

$$\tilde{T}_{wall} = \Sigma T_{wall}/15 \quad (6)$$

The average Nusselt number, Nu is estimated as follows:

$$Nu = hD/k \quad (7)$$

The Reynolds number is given by

$$Re = UD/\nu \quad (8)$$

where U is mean water velocity of the tube and ν is a kinematic viscosity of the working fluid.

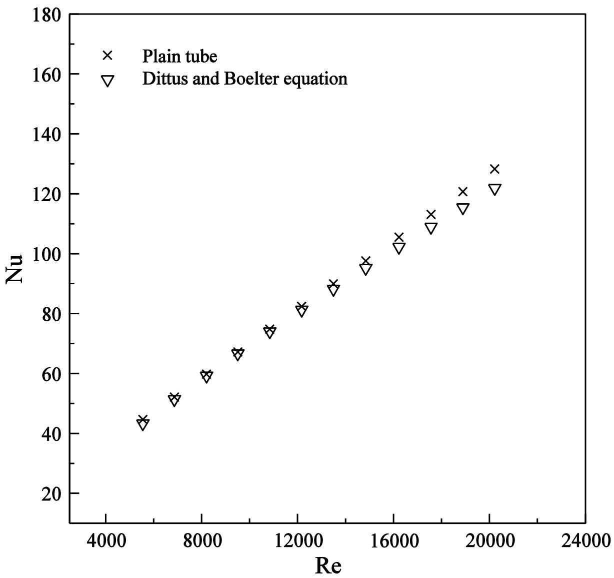
Friction factor, f can be written as:

$$f = \frac{\Delta P}{(L/D)(\rho U^2/2)} \quad (9)$$

All of thermo-physical properties of the water are determined at the overall bulk water temperature which obtained from equation (4).

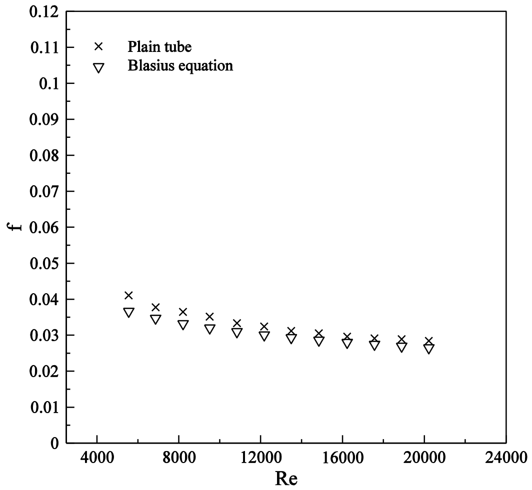
4. RESULTS AND DISCUSSION

The results of heat transfer (in term of Nusselt number), friction factor and thermal performance factor associated by the modified twisted tapes and typical one, are presented and discussed. Prior to the main experiments, the reliability of the experimental facility is evaluated by comparing the heat transfer and friction factor obtained from the present work with those achieved from the well known correlations [22], including Dittus-Boelter correlation for Nusselt number and Blasius correlation for friction factor as shown in Fig. 3(a) – (b).



(a) Nusselt number

Fig. 3 *Proof of the present plain tube and experimental facility.*



(b) friction factor

Fig. 3 *Proof of the present plain tube and experimental facility. (cont.)*

The comparison shows that the Nusselt number and friction factor of the present work agree well with those calculated from the available correlations within ±2.5% and ±7.9%, respectively.

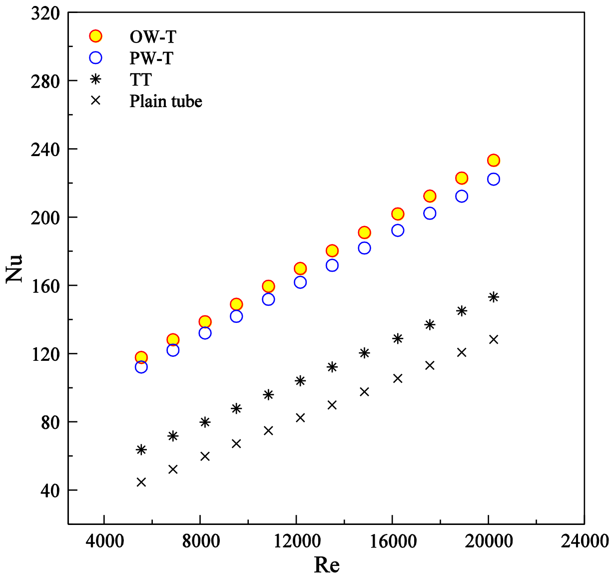


Fig. 4 *Nusselt number versus Reynolds number for the tubes with OW-T, PW-T, TT, and plain tube.*

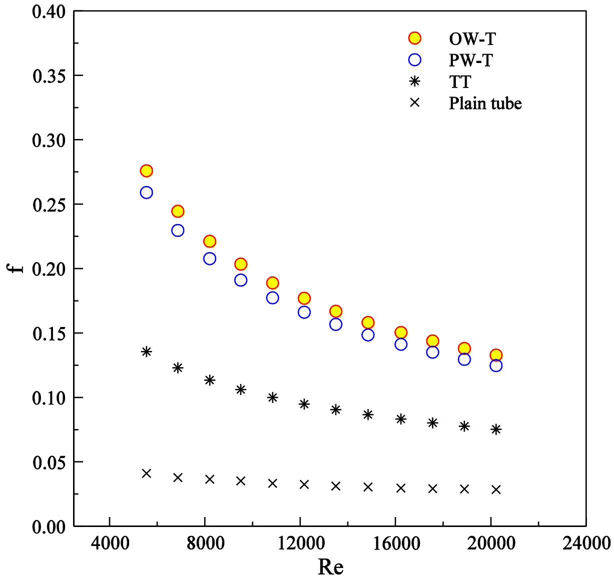


Fig. 5 *Friction factor versus Reynolds number for the tubes with OW-T, PW-T, TT, and plain tube.*

Fig. 4 displays the variation of Nusselt number with Reynolds number of the tubes with OW-T, PW-T and TT and also with the plain tube. The figure demonstrates that for all tubes, the Nusselt number consistently increases with increasing Reynolds number. This is attributed to the increases of turbulent intensity imparted to the flow and a better mixing of the fluid between the core and the tube wall region. At the same Reynolds number, the Nusselt numbers of all inserted tubes are considerably higher than that of the plain tube. It is also observable that the heat transfer improvement becomes prominent at low Reynolds number. This can be explained by the fact that at low Reynolds number, the thermal boundary is thicker, thus the disruption of the boundary by the twisted tape insert is more obvious. In addition, both the modified twisted tapes (OW-T and PW-T) possess considerably higher Nusselt number than the TT, owing to the additional flow disturbance acted by the wings and alternate axes on the tapes. At a similar operating condition, the Nusselt number in the tube with OW-T is slightly higher than that in the one with PW-T. This can be attributed to a better turbulence distribution on both sides of twisted tape induced by the opposite wings. For the range examined, the increases of mean Nusselt numbers in the tubes equipped with OW-T, PW-T and TT, respectively, are found to be 110%, 100% and 27.5% over the plain tube with no insert.

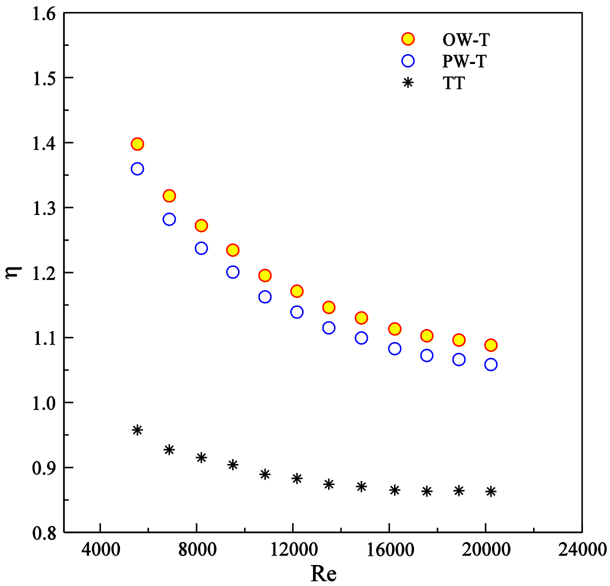


Fig. 6 *Thermal performance factor versus Reynolds number for the tubes with OW-T, PW-T and TT.*

Fig.5 portrays the variation of friction factor with Reynolds number values for all tubes. At a given Reynolds number, all inserted tubes possess higher friction factors than the plain tube with no insert, due to the flow obstruction of twisted tape as well as the increase of the flow contact of swirl flow compared to the common axial flow in the plain tube. Similar to Nusselt number, the friction factor of the OW-T is slightly higher than that of the PW-T, and friction factors of the OW-T and PW-T are considerably higher than that of the TT. This implies that both the OW-T and PW-T generate higher force against the tube wall than the TT, due to the extra forces exerted by wings and alternate axes. As shown in Fig.5, the friction factor obtained by the OW-T and the PW-T is higher than that by the TT, by the factors of 1.87 and 1.76 times, respectively.

With the help of experimental data, the empirical Nusselt number and friction factor correlations are derived for the tubes with opposite/parallel rectangular wing twisted-tapes as follows.

Parallel rectangular wing twisted-tape (PW-T):

$$Nu = 0.543\,Re^{0.536}\,Pr^{0.4} \tag{10}$$

$$f = 33.88\,Re^{-0.565} \tag{11}$$

Opposite rectangular wing twisted-tape (OW-T):

$$Nu = 0.57\,Re^{0.536}\,Pr^{0.4} \tag{12}$$

$$f = 36.08\,Re^{-0.565} \tag{13}$$

As preliminary design guidance to the selection of a technique, the thermal performance factor can be evaluated based on the power consumption per unit mass of fluid. The thermal performance criteria, η defined as the ratio of the heat transfer coefficient for the tube fitted with the opposite/parallel rectangular wing twisted-tape (h_t) to that for the plain tube (h_p) at a similar pumping power is written as

$$(f\,Re^3)_p = (f\,Re^3)_t \tag{14}$$

The thermal performance factor at a constant pumping power can be derived as follows:

$$\eta = \left.\frac{h_t}{h_p}\right|_{pp} = \left(\frac{Nu_t / Nu_p}{(f_t / f_p)^{1/3}}\right) \tag{15}$$

Fig. 6 depicts the variation of thermal performance factor (η) with Reynolds number for the tube inserted with the OW-T, PW-T and TT. For the whole range studied, the thermal performance factors of the OW-T, PW-T are consistently above unity and higher than those of the TT. This signifies the beneficial gain in view point of energy saving of using the tube with both the modified twisted tapes over the plain tube and the tube with TT. In the present range, the average thermal performance factor of the OW-T is higher than those of the TT and the PW-T at 24.9% and 2.74%, respectively. The maximum thermal performance factor of 1.4, given by the OW-T is found at the lowest Reynolds number.

The empirical correlations of thermal performance factor can be drawn as:

Parallel rectangular wing twisted-tape (PW-T):

$$\eta = 7.053\,Re^{-0.193} \tag{16}$$

Opposite rectangular wing twisted-tape (OW-T):

$$\eta = 7.251\,Re^{-0.193} \tag{17}$$

The validities of the correlations are verified by comparing the predicted data with the experimental data as shown in Figs. 7 - 9. Evidently, the predicted values are in good agreement with the experimental data within ±4% for Nusselt number, ±2% for friction factor and ±2% for thermal performance factor.

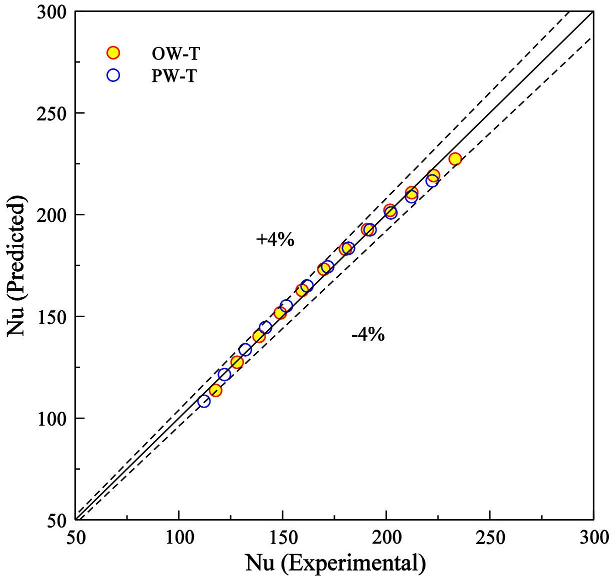


Fig. 7 *Comparison of experimental Nusselt number and predicted results.*

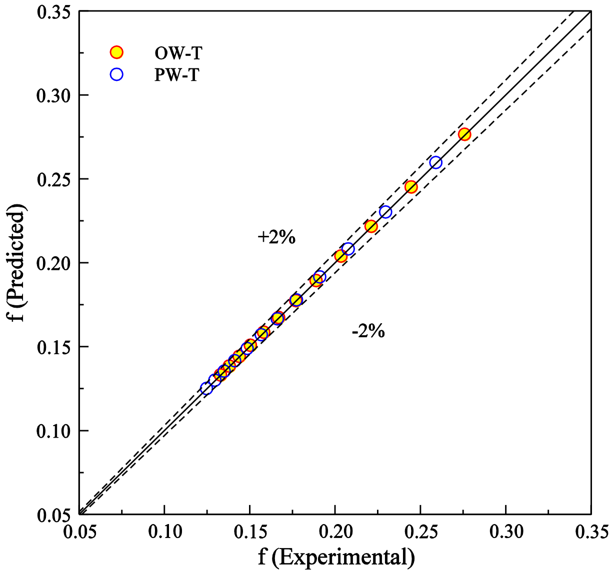


Fig. 8 *Comparison of experimental friction factor and predicted results.*

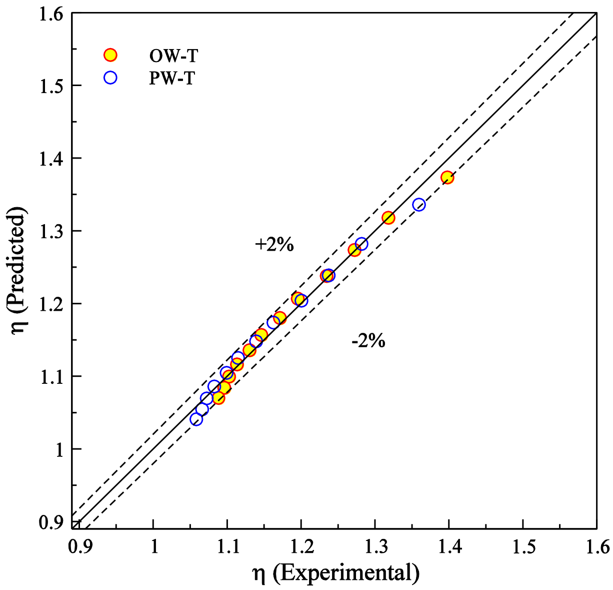


Fig. 9 *Comparison of experimental thermal performace factor and predicted results.*

5. CONCLUSIONS

The research work has been conducted to assess forced convection heat transfer and friction loss behaviors of turbulent flow through a heat exchanger tube inserted with the opposite/parallel rectangular wing twisted-tapes (OW-T/PW-T) as well as the typical twisted tape (TT). The work was performed under a uniform heat flux (UHF) condition for the Reynolds number range of 5000 to 20,200. At a similar operating condition, the OW-T gives the highest heat transfer rate, friction factor and thermal performance factor and the PW-T performs better than the TT. For the range examined, the mean Nusselt numbers increases of the tubes equipped with the TT, OW-T and PW-T, are found to be 27.5% 110% and 100% over the plain tube, respectively. In addition, the average thermal performance factor of the OW-T is higher than those of the TT and the PW-T at around 24.9% and 2.74%, respectively. The maximum thermal performance factor of 1.4, for the OW-T is found at low Reynolds number.

TABLE 1: Previous studies on twisted tapes

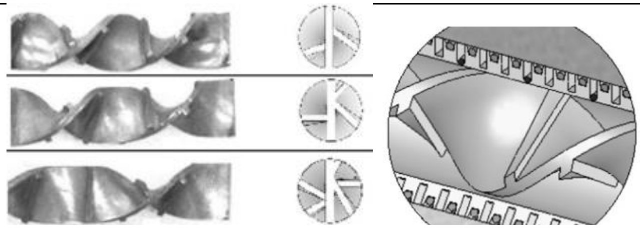
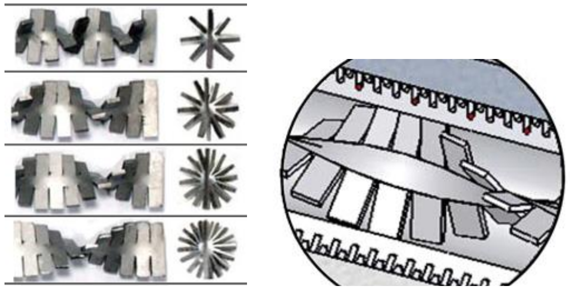
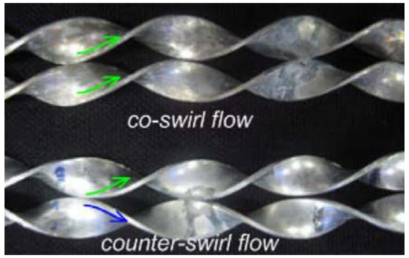
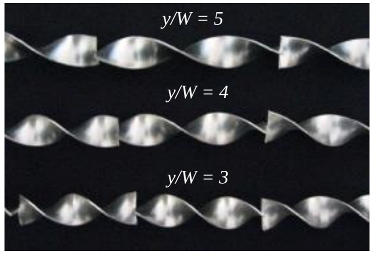
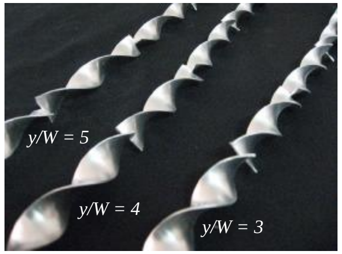
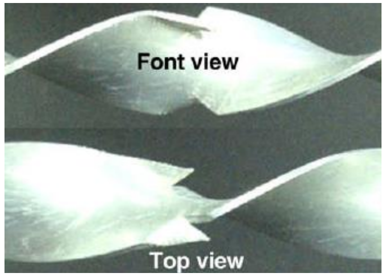
Author	Type of twisted tape	working fluid	Reynolds number	Nu_t/Nu_p
Chang et al. [1]		air	5000-25,000	2.5-4.8
Chang et al. [2]		air	1000-40,000	1.28-2.4
Rahimi et al. [3]		water	2950-11,800	1.96-2.49
Eiamsa-ard et al. [6]		water	3700-21,000	1.59-2.87

TABLE 1: Previous studies on twisted tape *(continued)*

Author	Type of twisted tape	working fluid	Reynolds number	Nu_t/Nu_p
Eiamsa-ard et al. [7]	 Font view of TA  Isometric view of TA	water	3000-27,000	1.52-1.91
Eiamsa-ard et al. [8]		water	1000-20,000	2.6-6.5
Eiamsa-ard et al. [9]		water	3000-27,000	1.35-2.55

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