Thermal performance evaluation of micro-fin tube with twisted tape inserts

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Abstract
The heat transfer coefficient and pressure drop characteristics of turbulent flow through micro-fin tubes fitted with twisted tape in co/counter-arrangement have been studied. In the experiments, water as the tested fluid is passed in a Reynolds number range of 7800 to 17,650. The results reveal that, the Nusselt numbers in the system with a micro-fin tube and twisted tape are considerably higher than those in the system without tape. Evidently, the tape with the smaller twist ratio gives higher heat transfer rate, friction factor as well as thermal performance factor than the one with larger twist ratio as a result of a larger contact surface area, and stronger turbulence intensity and thus better fluid mixing which leads to a thinner thermal boundary layer. The result reveals that for using the micro-fin tube with twisted tape, the increases in the mean Nusselt number and friction factor are, respectively, up to 181% and 3.86 times of the plain tube and the maximum thermal performance factor is 1.52. It is also obvious that the heat transfer and friction factor obtained from the micro-fin tube with twisted tape in counter-arrangement is higher than that from the micro-fin tube with twisted tape in co-arrangement.

Key words: Heat transfer enhancement, Micro-fin tube (MT), Twisted tape, Swirl flow, counter-arrangement(CT), co-arrangement(CoT)

1. Introduction

High performance heat exchangers are significantly desired, they offer not only the material and energy saving but also the device compactness. Several heat transfer enhancement tubes have been applied for this aspect (Eiamsa-ard et al. (2009), Eiamsa-ard et al. (2010), Kongkaipaiboon et al. (2010), Eiamsa-ard (2010)). Among promising ones, micro-fin tubes are most considered one. Micro-fin tubes have been proved to be practical for application in refrigeration and air-conditioning industries due to their effectiveness in enhancing evaporation and condensation heat transfer of the refrigerant and also their low production cost. Micro-fin tubes are also used for the single-phase heat transfer enhancement in many engineering applications, such as (1) in the sub-cooled region of air-cooled condensers, (2) in the superheated region of air-conditioning evaporators, and (3) in the heat exchangers of water chiller systems.

Micro-fin tubes in several geometries have been investigated and reported. Jensen and Vlakancic (1995) performed experimental studies the heat transfer and flow friction characteristics in micro-fin tubes with different fin numbers, fin heights and fin helix angles (Fig. 1). Han and Lee (2005a) carried out the heat transfer and friction loss of a micro-fin tube for condensation of refrigerants R134a, R22, and R410A. Han and Lee (2005b) have experimental investigated the single-phase heat transfer and fluid flow characteristics of micro-fin tube. They observed that the
micro-fin tube with higher relative roughness and smaller spiral angle provides better thermal performance than the one with larger spiral angle and smaller relative roughness. Wang et al. (1996) obtained seven commercially available micro-fin tubes and a smooth tube using water as the working fluid. The comparison between the results of the micro-fin tubes and smooth tube revealed that a heat transfer enhancement increased with the Reynolds number and eventually reached a maximum value at a critical Reynolds number. Chiou et al. (1995) reported that the single-phase heat transfer enhancement in two micro-fin tubes of double-pipe heat exchanger was successful. Al-Fahed et al. (1993) performed investigation of the heat transfer characteristics in the micro-fin tube with an outer diameter of 15.9 mm, a fin height of 0.3 mm and a fin helix angle of 18 degrees, for Reynolds numbers between 10,000 and 30,000. Their results revealed that the micro-fin tube gave heat transfer coefficient was 1.2-1.8 times and the isothermal friction factor 1.3-1.8 times of that achieved from a plain tube under similar conditions. The success of single-phase heat transfer enhancement in a micro-fin tube was also reported by Li et al. (2007). Swirl flow devices and modified surfaces have been widely employed in thermal engineering to enhance heat transfer. Colombo et al. (2012) studied on the flow boiling and convective condensation characteristics of oil-free R134a in micro-fin tubes. Akhavan-Behabadi et al. (2014) reported the convective heat transfer of the heat transfer oil-copper oxide nanofluid flow in horizontal micro-fin tubes. Their results found that the combination use of nanoparticles and the micro-fin tube leads to the heat transfer enhancement up to 2.3 times of the smooth tube. Derakhshan et al. (2015) performed the mixed convection heat transfer behaviors of multi walled carbon nano tube inside smooth and micro-fin tubes. It was found that the heat transfer coefficient increases slightly with the increase of particle weight concentration from 0% to 0.2% in micro-fin and smooth tubes under a given Grashof number. To extend the potential of heat transfer enhancement, twisted tape swirl generators have been compound with modified tubes such as corrugated tube, start spirally groove tube, tube having integral axial rib roughness, micro-fin tube and grooved tube, etc. Regarding to the previous studies, the combination of inserting swirl generators and roughening heat exchanger surfaces (compound augmentation technique) at appropriate conditions, offers higher heat transfer rate than the sum of the corresponding values achieved from the individual techniques. Therefore, the purpose of the present work is to investigation the compound effects of micro-fin tube fitted with twisted tape swirl generators.

Fig. 1 Micro-fin tubes(MFs) used in the previously published works: (a) MF of Al-Fahed et al. (1993), (b) MF of Bharadwaj et al. (2009), (c) MF of Nagarajan et al. (2010) and (d) MF of Eiamsa-ard and Wongcharee (2013).

The benefit of using micro-fin tube in common with the twisted tape, inspires this work to design twisted-tapes with different arrangements (co and counter swirl arrangements). The work has been carried out in order to find the optimum tradeoff between the increased heat transfer and the friction factor. The objective of this report is to investigate the combined effect of micro-fin tube together with a twisted-tape containing in a constant heat flux tube on heat transfer, flow friction, and thermal performance factor behaviors characteristics using water as the working fluid for Reynolds number ranging from 7800 to 17,650. In addition, the effects of twist ratio (y/W) on thermal performance were also conducted to investigate the characteristics of fluid flow and heat transfer associated by the twisted tapes.
2. Micro-fin tube with twisted tape

The micro-fin tube (MF) used in the present work made was of from copper, the photographs and details of the tube is shown in Fig. 2(a-d). The cross-section of fins is in trapezoidal shape. The important geometries of micro-fin tubes can be described as follows (1) fin pitch ($p$), (2) fin helix angle ($\alpha$), (3) fin apex angle ($\beta$), (4) fin height ($h_f$), (5) fin interval ($s$), (6) tube inner diameter ($d$), (7) tube outer diameter ($D$), (8) thickness ($t$), and (9) length ($L$), respectively. The micro-fin tube was wound with a heater wire along the test section to achieved constant heat flux condition during the experiments. Thermocouples were tapped on the micro-fin tube wall to measure the local wall surface temperatures. The micro-fin tube was well insulated by wrapping insulation tape around all exposed portions of the tube. Twisted tape was made of aluminum sheet that was 1.0 mm thick, 14.5 mm wide (W) and 1000 mm long. The twist ratio ($y/W$) of twisted tape was varied from 2.0 to 5.0 ($y$ is defined as the length with 180° rotation). The micro-fin tube with twisted tape were arranged in two different forms: (1) twisted tapes acted in the same direction for co-swirl (co-arrangement) and (2) twisted tapes acted in opposite directions for counter swirl (counter-arrangement).

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(a) micro-fin tube alone

(b) details of micro-fin tube

(c) micro-fin tube with tape inserts

(d) twisted tape at different twist ratios

Fig. 2 Photographs of micro-fin tube with twisted tape inserts.
3. Experimental facility and procedure

The experiment was carried out in an open-loop experimental facility as shown in Fig. 3. The heat transfer test section made of copper. The heat exchanger tube was insulated lengthwise using asbestos sleeves/disks to prevent heat transfer to the surroundings. The heat transfer test section was heated by continually winding flexible electrical wire. The electrical output power was controlled by a variac transformer to obtain a constant heat flux along the entire length of the test section and by keeping the current less than 9 amps. The inlet and outlet temperatures of the water were measured at certain points with a multi-channel temperature measurement unit (data logger) in conjunction with the RTD thermocouples. Fifteen thermocouples also were tapped on the local wall of the tube and the thermocouples were placed round the tube to measure the circumferential temperature variation, which was found to be negligible. The mean local wall temperature was determined by means of calculations based on the reading of the wall thermocouples.

Fig. 3 Schematic diagram of heat transfer apparatus.

In the apparatus setting above, the inlet bulk water at 25°C was drawn via a water pump to a rotameter and then a heat transfer test section. The storage tank which supplied water was maintained at a constant head. The volumetric water flow rate from the tank was adjusted by a control valve at the upstream of the rotameter. During the experiments, the bulk water was heated by an adjustable electrical heater wrapping along the test section. To obtain an adequately accurate average Nusselt number, wall temperatures were acquired at 15 stations between the entrance and exit of the test section. In the present work, the Reynolds number of water was varied from 7800 to 17,650. The local wall temperature, inlet and outlet water temperature, the pressure drop across the test section and water flow velocity were measured for heat transfer calculation of the heated tube with twisted tape inserts. All data (temperature, volumetric flow rate and pressure drop) were recorded at steady state conditions. The various characteristics of the flow, the Nusselts number, and the Reynolds numbers were based on the average of tube wall temperatures and inlet-outlet water temperatures. The average Nusselt numbers were calculated and discussed where all fluid properties were determined at the overall bulk mean temperature.
4. Data reduction

In the present work, the water is used as the test fluid and flowed through the plain tube and micro-fin tube with a constant wall heat flux condition. At a steady state condition, the heat absorbed by the cold water is assumed to be equal to the convective heat transfer from the test section which can be expressed as:

\[ Q_{\text{water}} = Q_{\text{conv}} \]  

where \[ Q_{\text{water}} = MC_{p,\text{water}}(T_o-T_i) \]  

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 the surroundings. Thus, only the heat transfer rate absorbed by the fluid is taken for internal convective heat transfer coefficient calculation.

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

\[ Q_{\text{conv}} = hA(T_w-T_h) \]  

where \[ T_h = \frac{(T_o+T_i)}{2} \] and \[ T_w = \frac{\sum T_w}{15} \]

in which \( T_w \) is the local inner wall surface temperature which is evaluated at the outer wall surface of the tube (thermocouples embedded in V-groove outer surfaces).

In case of a copper tube, the thermal resistance of the tube wall can be neglected and therefore the measured \( T_w \) can be approximated to be the same as the inner wall surface temperature. The average wall temperatures are calculated from 15 stations, lined equally between the inlet and the exit of the test pipe. The heating surface area \( A \) based on the hydraulic diameter \( D \) was used in all calculations for the micro-fin tube with/without twisted tapes.

The average heat transfer coefficient, \( h \) and the average Nusselt number, \( Nu \) are calculated from

\[ h = MC_{p,\text{water}}(T_o-T_i)/A(T_w-T_h) \]  

Three parameters of interest for the present case are: (1) friction factor, (2) Nusselt number, and (3) thermal performance factor (\( \eta \)).

\[ Nu = AD/k \]  

Uncertainty of Nusselt number can be determined by the following expression:

\[ \frac{\Delta Nu}{Nu} = \left[ \left( \frac{\Delta h}{h} \right)^2 + \left( \frac{\Delta C_{p,\text{water}}}{C_{p,\text{water}}} \right)^2 + \left( \frac{\Delta T_o}{T_o} \right)^2 \right]^{0.5} \]  

The Reynolds number is given by

\[ Re = \rho uD/\mu \]  

Uncertainty of Reynolds number can be determined by the following expression:

\[ \frac{\Delta Re}{Re} = \left[ \left( \frac{\Delta \rho}{\rho} \right)^2 + \left( \frac{\Delta u}{u} \right)^2 + \left( \frac{\Delta D}{D} \right)^2 \right]^{0.5} \]  

The friction factor, \( f \) is computed from

\[ f = \frac{\Delta P}{\left( \frac{L}{D} \right) \left( \frac{\rho u^2}{2} \right)} \]
Uncertainty of friction factor can be determined by the following expression:

$$\frac{\delta f}{f} = \left[ \left( \frac{\delta u}{u} \right)^2 + \left( \frac{\delta \rho}{\rho} \right)^2 + \left( \frac{\delta D}{D} \right)^2 + \left( \frac{\delta \ell}{\ell} \right)^2 + \left( \frac{\delta (\Delta P)}{\Delta P} \right)^2 \right]^{0.5} \quad (12)$$

The experimental results are reproducible within these uncertainty ranges. The maximum uncertainties of non-dimensional parameters are Reynolds number, Nusselt number and friction factor are 3.5%, 5.2% and 4.8%, respectively. The thermal performance factor ($\eta$) is defined as the ratio of the Nusselt number ratio to the friction factor ratio at the same pumping power:

$$\eta = \frac{Nu/N_{u,p}}{f/f_p}$$

5. Results and discussion

In this section, the heat transfer (Nusselt number, $Nu$), pressure drop (friction factor, $f$) and thermal performance factor results of the micro-fin tube fitted with twisted-tape are comparatively, reported with those of the plain tube alone. The results of the plain tube in the present work are compared with Dittus-Bolter correlation and Blasius correlation for the fully developed turbulent flow as seen in Fig. 4(a-b). It is demonstrated that the results of the present work reasonably agree well with the available correlations.

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Fig. 4 Verification of plain tube.

Fig. 5 Effect of MF equipped with twisted tapes on heat transfer (MT&CT: micro-fin tube fitted with twisted tape in counter-arrangement, MT&CoT: Micro-fin tube fitted with twisted tape in co-arrangement).
5.1 Heat transfer

Figure 5(a-b) shows the variation of Reynolds number with the Nusselt number for various twist ratios ($y/W = 2.0$, $3.0$, $4.0$ and $5.0$). It is seen that the Nusselt number increases with increasing Reynolds number and reduction of twist ratio. Micro-fin tube fitted with twisted tape provides a considerable increase in heat transfer rate over the plain tube and micro-fin tube alone. A close examination reveals that the heat transfer rate from using both micro-fin tube and twisted-tape is higher than that from using micro-fin tube alone. In general, the micro-fin tube is employed to create a re-circulating flow near the wall regime leading to redeveloping of thermal boundary layer while the twisted-tape swirl generator to generate a swirl flow about the core region. A combination of the micro-fin tube and the twisted-tape is used as a means of enhancing heat transfer by both of reverse flow and swirl flow in a tube. Heat transfer is increased remarkably if 1) the micro-fin tube is applied in conjunction with the twisted-tape, and 2) the twisted-tape ratio is small ($y/W = 2.0$) to produce a higher swirl flow. These phenomena result in a better mixing of the flow between the wall and the core regions. In addition, the effects of the co/counter swirl arrangements on the heat transfer rate are also presented. It is visible that the combined devices in counter-swirl arrangement provide the higher heat transfer rate than those of the co-swirl arrangement. From the experimental results, it can be interpreted that a counter-swirl flow causes higher swirl/turbulence in the radial direction than a co-swirl one due to a strong collision of cross-fluid streams with opposite directions induced by the combined tapes twisted in opposite directions. This leads to a better fluid mixing within the micro-fin tube and thus a thinner thermal boundary layer along the tube wall, resulting in superior convective heat transfer.

5.2 Friction factor

The effect of using the micro-fin tube in common with the twisted tape on the flow friction is presented in Fig. 6(a-b). In the figure, the average increases in friction factor of employing the micro-fin tube fitted with twisted-tape is higher than those the plain tube. The pressure loss mainly comes from 1) higher friction of increasing surface area because of the presence of the micro-fin tube and 2) the dissipation of the dynamical pressure of the fluid due to high viscous losses near the tube wall, and to the extra forces exerted by rotation or swirling flow. The higher friction loss induced by the tape with smaller twist ratio ($y/W$) can be attributed to the increasing flow blockage and flowing path length as well as the extra forces exerted by reverse flow. These factors are directly responsible for a larger dissipation of the dynamical pressure of the working fluid. Effects of the micro-fin tube in common with the twisted tape in co/counter swirl arrangement on friction factor are shown in the figure. Similar to the results reported above for Nusselt number, the friction factor generated by the combined tapes twisted in opposite direction (producing counter-swirl flow) are higher than those producing co-swirl flow.

5.3 Thermal performance factor

The variation between the thermal performance factor and Reynolds numbers is depicted in Fig. 7 for different twist ratios ($y/W$). In the figure, it is worth noting that the thermal performance factor reduces with the increase of...
Reynolds numbers and the decrease of twist ratio. The use of micro-fin tube in common with the twisted tape provides higher thermal performance factor than that of micro-fin tube alone. The thermal performance factor tends to reduce to unity at high Reynolds number for all cases. This indicates that the micro-fin tube in conjunction with the twisted tape is not feasible in terms of energy saving at higher Reynolds number. It is also obvious that the compound heat transfer enhancement devices in counter-swirl arrangement yields the higher values of thermal performance factors than that the co-swirl arrangement.

6 Conclusions

Convective heat transfer performance and pressure drop in turbulent flow through a micro-fin tube in common with a twisted tape at different twist ratios were experimentally studied under a constant wall heat flux condition. It is found that the smaller twist ratio is, the higher the heat transfer, friction factor and thermal performance factor for all Reynolds numbers. Nusselt numbers associated with the uses of a micro-fin tube in common with the twisted tape at \( y/W = 2.0 \) were higher than that of \( y/W = 3.0, 4.0 \) and \( 5.0 \) by around 4.7%, 10% and 16.4%, respectively. The maximum value of thermal performance factor obtained from using the \( y/W = 2.0, 3.0, 4.0 \) and \( 5.0 \), are found to be 1.34, 1.31, 1.24 and 1.2, for co-arrangement, and 1.52, 1.47, 1.41 and 1.38, for counter-arrangement respectively.

References


Eiamsa-ard, S. and Wongcharee, K., Heat transfer characteristics in micro-fin tube equipped with double twisted tapes: