Heat transfer enhancement in a tube using rectangular-cut twisted tape insert

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Abstract

An experimental investigation was carried for measuring tube-side heat transfer coefficient, friction factor, heat transfer enhancement efficiency of water for turbulent flow in a circular tube fitted with rectangular-cut twisted tape insert. A copper tube of 26.6 mm internal diameter and 30 mm outer diameter and 900 mm test length was used. A stainless steel rectangular-cut twisted tape insert of 5.25 twist ratio was inserted into the smooth tube. The rectangular cut had 8 mm depth and 14 mm width. A uniform heat flux condition was created by wrapping nichrome wire around the test section and fiber glass over the wire. Outer surface temperatures of the tube were measured at 5 different points of the test section by T-type thermocouples. Two thermometers were used for measuring the bulk temperatures. At the outlet section the thermometer was placed in a mixing box. The Reynolds numbers were varied in the range 10000-19000 with heat flux variation 14 to 22 kW/m² for smooth tube, and 23 to 40 kW/m² for tube with insert. Nusselt numbers obtained from smooth tube were compared with Gnielinski [1] correlation and errors were found to be in the range of -6% to -25% with r.m.s. value of 20%. At comparable Reynolds number, Nusselt numbers in tube with rectangular-cut twisted tape insert were enhanced by 2.3 to 2.9 times at the cost of increase of friction factors by 1.4 to 1.8 times compared to that of smooth tube. Heat transfer enhancement efficiencies were found to be in the range of 1.9 to 2.3 and increased with the increase of Reynolds number.

1. Introduction

To improve the performance of heat exchanging devices for reducing material cost and surface area and decreasing the difference for heat transfer thereby for reducing external irreversibility, lot of techniques have been used. Among different passive means to increase heat transfer coefficient, twisted tape inserts are promising. The secondary flow (swirl flow) generated by twisted tape effects fluid flow across the tape-partitioned tube, promotes greater mixing and higher heat transfer coefficients. Experimental investigation of heat transfer and friction factor characteristics in a double pipe heat exchanger fitted with regularly spaced twisted tape elements were studied by Eiamsa-ard et al., 2006. Heat transfer, friction factor and heat transfer enhancement efficiency characteristics in a circular tube fitted with conical-ring turbulators and a twisted-tape swirl generator have been investigated experimentally by Promvonge and Eiamsa-ard, 2007. Influences of insertion of wire coils in conjunction with twisted tapes on heat transfer and friction characteristics in a circular tube using air as the test fluid were experimentally investigated by Promvonge, 2008. Eiamsa-ard et al., 2009, experimentally

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Keywords: Heat transfer enhancement; rectangular–cut twisted tape insert; heat transfer enhancement efficiency.
investigated the influences of the tube with short-length twisted tape inserts on the heat transfer, friction factor, and heat transfer enhancement efficiency. Heat transfer, friction factor and heat transfer enhancement efficiency behaviors in a tube equipped with the combined devices between the twisted tape and constant/periodically varying wire coil pitch ratio were experimentally investigated by Eiamsa-ard et al., 2010a. Eiamsa-ard et al., 2010b experimentally determined the influences of twin-counter/co-twisted tapes on heat transfer rate, friction factor, and heat transfer enhancement efficiency. Heat transfer, flow friction and heat transfer enhancement efficiency characteristics in a tube fitted with delta-winglet twisted tape, using water as working fluid were investigated experimentally by Eiamsa-ard et al., 2010c. Murugesan et al., 2010, experimentally investigated heat transfer, friction factor, and heat transfer enhancement efficiency characteristics of a double pipe heat exchanger fitted with square-cut twisted tapes. Shabanian et al., 2011 reported the experimental and computational fluid dynamics modeling studies on heat transfer, friction factor and heat transfer enhancement efficiency of an air cooled heat exchanger equipped with classic and jagged twisted tape.
The scope of the present work is to experimentally investigate the tube side heat transfer and friction factor of a circular tube fitted with rectangular-cut twisted tape insert. Data are compared with smooth tube heat transfer and friction values and the values of heat transfer enhancement efficiency are reported.

2. Experimental set up

The schematic diagram of the experimental set up is shown in Fig 1. The test section was made from 914 mm of copper tube (26.6 mm ID and 30 mm OD), of which 900 mm was considered to be the test section. A stainless steel twisted tape was made by twisting 2 mm thick (δ) 20 mm width (w) straight strip. Tape pitch (y) of 105 mm was made which gave twist ratio (y/w) 5.25. The twisted tape was cut in the top region of 8 mm depth and 14 mm width rectangular cut to allow the flow from the both sides of the tape to mix at the cut regions. Figure 2 shows the twisted tape. The nichrome resistance wire was spirally wound uniformly on the outer surface of the test section to supply the heating power. Mica sheet was used between the tube and heating wire for electrical insulation. The heating wire was covered with mica sheet and fiber glass. The heating wire was connected to 220 Volt main. Five T-type (copper constantan) thermocouples were placed on five equally spaced points of the test section to measure the outer surface temperatures of the tube. Two thermometers were placed at the inlet and outlet of the tube to measure the inlet and outlet water temperatures respectively. To measure the outlet temperature, the thermometer was placed in a mixing chamber, which was thermally insulated to minimize the heat loss. A rotameter (Metric 24G, SS float) of 26 L/min capacity was provided to measure the water flow rate. A U-tube manometer was used to measure the pressure drop across the tube. The distance between two pressure tappings was 1180 mm.

Initially, water was taken in the tank and pumped to the test section through the rotameter. The flow rate of water was varied by the gate valve for different data and kept constant during the experiment. A minimum of 9.9 L/min was used and it was increased up to 18.9 L/min. After switching on the heating power the sufficient time was given to attain the steady state condition. In each run, data were taken for water flow rate, water inlet, outlet, tube outer surface temperatures and pressure drop readings.
3. Data reduction

Heat transfer rate by the heater to water was calculated by measuring heat added to the water. Heat added to water was calculated by,

\[ Q = m \cdot c_p \cdot (T_{\text{out}} - T_{\text{in}}) \]  

(1)

Heat transfer coefficient was calculated from,

\[ h = \frac{q}{(T_{\text{in}} - T_b)} \]  

(2)

and heat flux was obtained from,

\[ q = \frac{Q}{A} \]  

(3)

where, \( A = \pi d_i L \)  

(4)

The bulk temperature was obtained from the average of water inlet and outlet temperatures,

\[ T_b = \frac{T_{\text{in}} + T_{\text{out}}}{2} \]  

(5)

Tube inner surface temperature was calculated from one dimensional radial conduction equation,

\[ T_{wi} = T_{wo} - Q \cdot \frac{\ln(d_o/d_i)}{2 \pi k_w L} \]  

(6)

Tube outer surface temperature was calculated from the average of five local tube outer surface temperatures,

\[ T_{wo} = \frac{\sum_i T_{wo,i}}{5} \]  

(7)

Theoretical Nusselt number was calculated from Gnielinski, 1976, correlation,

\[ \text{Nu}_{th} = \frac{(f/8)(Re-1000)Pr}{1 + 12.7(f/8)^{0.5}(Pr^{2/3} - 1)} \]  

(8)

where from Petukhov, 1970,

\[ f = (0.79 \ln \text{Re} - 1.64)^2 \]  

(9)

\[ \text{Re} = \frac{n U_{\text{in}} d_i}{\mu} \]  

(10)

\[ \text{Pr} = \frac{\mu C_p}{k} \]  

(11)

\[ \text{Nu} = \frac{h d_i}{k} \]  

(12)

Mean water velocity was obtained from,

\[ U_{\text{in}} = \frac{m}{A_f} \]  

(13)

Flow area was obtained from,
Friction factor, $f$, can be calculated from

$$f = \frac{\Delta p}{(L_i/d_i)\left(\rho U_m^2/2\right)}$$

(15)

$\Delta p$ is the pressure drop across tappings. All the fluid properties were evaluated at bulk temperature.

4. Results and discussion

Heat transfer and friction data for the smooth tube were collected first. These data were taken to check the validity of the set up and measurement techniques over the range of Reynolds number 10000 to 19070. Figure 3 shows the comparison of experimental Nusselt number for smooth tube with those calculated from Gnielinski, 1976 correlation. Data fall within -24.7% and -5.6% of the Gnielinski, 1976 values with r.m.s. value of error 20.3%. Salam et al., 2010 used the same set up and errors were found within -13% and +18% with r.m.s. value of error 12% when they compared the data with Dittus and Boelter, 1930 values. Nusselt numbers for the smooth tube and the tube with twisted tape insert are shown in Fig. 3. It is seen that, Nusselt numbers increased with the increase of Reynolds number and twisted tape insert gave higher values of Nusselt number than those for smooth tube. For tube with twisted tape, Reynolds number was calculated based on inner diameter of the tube. For smooth tube Nusselt numbers, $N_u$, increased from 54 to 120 with the increase of Reynolds number from 10002 to 18811 respectively. For tube with rectangular-cut twisted tape insert for $Re = 10116$, Nusselt number, $N_{ue}$, was found to be 125 and for $Re = 19070$, $N_{ui}$ was increased to 309. At comparable Reynolds numbers, Nusselt numbers in tube with rectangular-cut twisted tape insert were enhanced by 2.3 to 2.9 times compared to those of smooth tube with the average enhancement of 2.6 times. Swirl flow generated by twisted tape was responsible for thinning the thermal boundary layer and increasing the mixing between core and tube wall flows, Eiamsa-ard et al., 2009. Rectangular-cut in the twisted tape was responsible for additional disturbances which increased the tangential contact between secondary flow and the wall surface of the tube, Murugesan et al., 2010. This made heat transfer coefficient higher through the flow.

The variation of heat flux with Reynolds number for smooth tube and tube with rectangular-cut twisted tape insert is shown in Fig. 4. It is found that with the increase of $Re$ heat fluxes increased and rectangular-cut twisted tape insert gave higher heat fluxes than those for smooth tube. Higher values of heat transfer coefficient was responsible for this enhancement, although temperature difference between wall and bulk fluid, $(T_{wi}-T_b)_{es}$, significantly decreased for tube with insert, Fig. 5. An average of 68% enhancement of heat flux was observed for tube with insert than that of smooth tube.

![Graph showing variation of Nusselt number with Reynolds number](image_url)
Figure 4 shows the variations of friction factor with Reynolds number. Friction factors for both smooth tube, $f_s$ and tube with insert, $f_e$ decreased with the increase of $Re$. And $f_e$ were found to be 39% to 80% higher than $f_s$. These higher values of $f_e$ than $f_s$ are due to the high viscous loss near the wall regions caused by swirl flow, Eiamsa-ard et al., 2010b. Figure 6 also shows the friction factor obtained from Petukhov relation, $f_{th}$. $f_{th}$ were found to be significantly lower than $f_s$. Entrance effect could be the reason for this.

5. Heat transfer enhancement efficiency ($\eta$)

To assess the performance of heat exchanging devices with insert it is necessary to evaluate heat transfer enhancement efficiency ($\eta$). This efficiency is calculated using constant pumping power (Web and Kim, 2005; Murugresan et al., 2010). For constant pumping power,
\[ \dot{Q}_s \times \Delta p_s = \dot{Q}_s \times \Delta p_s \]  

Now considering the assumptions from Web and Kim, 2005 and from Eqs. (15) and (16)

\[ f_s \cdot R_{e_s} = \text{Eq} \cdot R_{e_s} \]  

The experimental values of Nusselt number and friction factor for smooth tube can be correlated as

\[ \text{Nu}_s = 0.00053 \text{Re}_s^{1.1941} \cdot \text{Pr}^{0.3} \]  

\[ f_s = 147.2 \text{Re}_s^{-0.8305} \]  

The errors between Nusselt numbers for experimental and predicted values for smooth tube were found to be in the range of -5.2% to 7% with r.m.s. value of 4.1%. For friction factor these errors were found to be from -9.5% to 10.3% with r.m.s. value 6.9%.

Similarly the experimental values of Nusselt number and friction factor for tube fitted with insert are correlated as

\[ \text{Nu}_e = 0.00023 \text{Re}_e^{1.432} \cdot (y/w)^{-0.01} \]  

\[ f_e = 25.475 \text{Re}_e^{-1.0173} \cdot (y/w)^{2.4015} \]  

The errors between Nusselt numbers for experimental and predicted values for tube with insert were found to be in the range of -2.2% to 1.4% with r.m.s. value of 1.5%. For friction factor these errors were found to be from -4.4% to 5.3% with r.m.s. value 3.3%.

From Eqns. (17), (19) and (21)

\[ \text{Re}_s = 0.4455 \text{Re}_e^{0.9139} \cdot (y/w)^{1.107} \]  

Heat transfer enhancement efficiency,

\[ \eta = \frac{h_{\text{ex}}}{h_{\text{ip}}^{\text{PP}}} = \frac{\text{Nu}_{\text{ex}}}{\text{Nu}_{\text{ip}}^{\text{PP}}} \]  

From Eqns. (18) and (22),

\[ \text{Nu}_e = 0.0002 \text{Re}_e^{1.09} \cdot \text{Pr}^{0.3} \cdot (y/w)^{1.322} \]  

From Eqns. (20), (23), and (24)

\[ \eta = 1.2387 \text{Re}_e^{0.339} \cdot \text{Pr}^{0.3} \cdot (y/w)^{1.33} \]  

Heat transfer enhancement efficiency, \( \eta \), calculated using Eqn. (25) is shown in Fig. 7. It is quite obvious that with the increase of Reynolds number heat transfer enhancement efficiencies increased continuously. The value of \( \eta \) increased from 1.8 to 2.2 with the increase of Re from 10116 to 19070.

![Fig. 7. The variation of heat transfer enhancement efficiency with Reynolds number.](image)

6. Conclusions

An experimental investigation was carried out for measuring tube-side heat transfer coefficient, friction factor, heat transfer enhancement efficiency of water for turbulent flow in a circular tube fitted with rectangular-cut twisted tape insert. The results can be summarized as,

(a) The Nusselt number increased with the increase of Re. The experimental Nu, values fall within -6% and -25% of the Gnielinski, 1976 value (Nu). The experimental values were enhanced by 2.3 to 2.9 times compared to Nu, values.

(b) An average of 68% enhancement of heat flux was observed for tube with rectangular-cut twisted tape insert (q_e) than that of smooth tube (q_s).

(c) The experimental f_e values were found to be 39% to 80% higher than f_s values.
(d) The heat transfer enhancement efficiency ($\eta$) were found to be increased with Re, and $\eta$ values ranged between 1.9 and 2.3.

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