EXPERIMENTAL STUDIES ON EFFECT OF BONDING THE TWISTED TAPE WITH PINS TO THE INNER SURFACE OF THE CIRCULAR TUBE

by

Swaminathan SELVAM [a*], ***Periyakaruppan THIYAGARAJAN*** [b], ***and Sivan SURESH*** [c]

[a] Department of Mechanical Engineering, Nehru Institute of Technology, Coimbatore, Tamil Nadu, India
[b] Department of Mechanical Engineering, Sri Krishna College of Engineering and Technology, Coimbatore, Tamil Nadu, India
[c] Department of Mechanical Engineering, National Institute of Technology, Trichy, Tamil Nadu, India

Original scientific paper
DOI: 10.2298/TSCI120807036S

Heat transfer and friction factor characteristics are studied for water flowing through the tube in tube heat exchanger for two different configurations, namely,(1) twisted tape with pins and (2) twisted tape with pins bonded to the inner surface of the test section. Experiments are carried out for different twist pitch to the width of the twisted tape ratios (y/w) for both configurations. But very few research works have been done based on the mentioned heat transfer enhancement techniques. This paper presents the effect of twisted tape with pins on the heat transfer for a fully developed turbulent flow. Experiments are conducted by maintaining constant wall temperature. The heat transfer rate and pressure drop is found to be high for configuration (2) of y/w=3.33 when compared to all other configurations. The empirical correlations developed for different twisted tape with pins results in ±7.28% deviation for Nusselt number and ±7.16% for friction factor.

Key words: *turbulent flow, bonding, twisted tape with pins, twist pitch, heat transfer enhancement, friction factor*

Introduction

Heat transfer can be increased by active and passive flow control techniques. In active flow control techniques, external energy is required to increase the heat transfer. Passive flow control techniques do not require external energy to increase the heat transfer. Some of the examples for passive flow control techniques are twisted tapes, wire brushes, fins and wire coils, *etc*. Twisted tape with pins falls under the category of passive flow control techniques.

Many researchers have proposed different passive techniques for enhancing heat transfer. Manglik and Bergles [1] reported experimental data for three different twisted tapes under uniform wall temperature boundary conditions. The experiments were conducted with water and ethylene glycol as working medium. From the experimental results the authors con-

* Corresponding author; e-mail: cuddaloreselvam@gmail.com

cluded that enhancement in heat transfer takes place due to thermally developed swirl flow, centrifugal forces due to the tape twist; longer flow path and tube blockage. Due to tube blockage there is increase in flow velocity and swirl mixing. Because of this effect there is increase in heat transfer. Manglik and Bergles [2] studied the effect of twisted tape inserts for water and ethylene glycol as working fluid. The authors concluded that the tube blockage and tape induced vortex mixing are the dominant phenomenon which resulted in increased heat transfer and pressure drop. Agarwal and Raja Rao [3] reported heat transfer augmentation for the flow of a viscous liquid in circular tubes using twisted tapes inserts under constant wall temperature and presented a correlation for isothermal friction factor and Nusselt number. Naphon [4] presented the heat transfer characteristics and the pressure drop of the horizontal double pipe with coil-wire insert. Naphon and Sriromruln [5] presented the heat transfer characteristics and the pressure drop of the horizontal double pipes with and without coiled wire insert. The results obtained from the micro-fin tube with coiled wire insert are compared with those obtained from the smooth and micro-fin tubes. Gul and Evin [6] reported heat transfer and friction characteristics in decaying turbulent swirl flow generated by a short helical tape placed at the entrance of the test section. They concluded that enhancement efficiency increases with increasing momentum ratio and decreases with decreasing momentum ratio and also decreases with increasing Reynolds number.

Eiamsa-ard and Promvonge [7] experimentally investigated the effects of insertion of a helical screw-tape with or without core-rod in a concentric double tube heat exchanger on heat transfer and flow friction characteristics. The experiments conducted for the Reynolds number ranging from 2000 to 12,000. Sivashanmugam and Suresh [8] reported heat transfer and friction factor characteristics of turbulent flow through a circular tube fitted with regularly spaced helical screw-tape inserts. Promvonge [9] experimentally studied the influences of insertion of wire coils in conjunction with twisted tape on heat transfer and turbulent flow friction characteristics in a uniform heat flux boundary condition. They all reported that the presence of wire coils together with twisted tape leads to a double increase in heat transfer over the use of wire coil/twisted tape alone. Sreenivasulu and Prasad [10] numerically studied the convective heat transfer for an annulus with inner cylinder wrapped with a helical wire. They reported numerical results for the variations in velocity, pressure, vorticity, turbulent kinetic energy, temperature, friction factor and Nusselt number for various annulus diameter ratios, pitch ratios, wire diameter ratios, and Reynolds number. Eiamsa-ard *et al*. [11] reported enhanced heat transfer and pressure loss by insertion of twisted tapes. Eiamsa-ard *et al*. [12] studied the effects of the twisted tape consisting of centre wings and alternate-axes on the thermo hydraulic properties in a round tube. Eiams7a-ard *et al*. [13] presented the effects of peripherally-cut twisted tape insert on heat transfer, friction loss and thermal performance factor characteristics in a round tube. Cui and Tian [14] reported three-dimensional numerical simulations and experiments on the heat transfer characteristics and the pressure drop of air flow in a circular tube with edge fold-twisted tape and classic spiral-twisted-tape inserts of the same twist ratio. They also compared the numerical results with the experiment results, with discrepancy for Nusselt number in a range of 1.6% to 3.6% and for friction factor in a range of 8.2% to 13.6%. Seemawute and Eiamsa-ard [15] presented the effect of peripherally-cut twisted tape with alternate axis on the fluid flow and heat transfer enhancement characteristics in a uniform heat flux circular tube. They conducted experiments using water as a testing fluid in a turbulent tube where the Reynolds number varied from 5000 to 20,000. Naphon and Suchana [16] experimentally investigated the heat transfer characteristics and the pressure drop of the horizontal concentric tube with twisted wire brush inserts. They concluded that the plain tube with twisted wire brushes insert have a large effect on the enhancement of heat transfer, and because of the twisted wire brush insert there is increase in pressure drop. Selvam *et al*. [17] experimentally studied the heat transfer and friction factor characteristics of turbulent flow through a circular tube fitted with wire coiled coil matrix turbulator inserts of various pitches with and without bonding, with water as working fluid.

It is clear from the cited literature that heat transfer increases for the tube fitted with various geometrical configurations of turbulator. However there is no information about the effect of bonding the twisted tape with pins to the inner surface of the test section. For the first time the authors have studied the effect of bonding the twisted tape with pins to the inner wall of the test tube and compared the results with plain tube. The main objective of this work is to study the effect of heat transfer rate and the pressure drop characteristics of the concentric tube heat exchanger (under constant wall temperature boundary condition when twisted tape with pins (TTP) is bonded to the inner surface of circular tube). All the experimental readings are taken for the turbulent flow condition with water as working fluid.

Details of turbulator

Details of TTP

Holes of 6 mm diameter are made on the copper sheet of 1.4 mm thickness and 21 mm width (w). The pitch distance between the drilled holes is 30 mm.The copper rod of 6 mm and length 21 mm is positioned in the holes. After placing the rod in correct position silver brazing is done between the rod and the copper sheet. The copper sheet is twisted to the pitch distance (y) of 70 mm. Thus the twisted tape with pins of y/w = 3.33 is fabricated. The distance between the rods is kept constant and only the pitch distance of the twist is varied. The mentioned fabrication procedure is repeated to produce the other twisted tape with pins of y/w = 4.29 and y/w = = 5.71.The configurations of the twisted tape with pins are given in figs.1(a) and 1(b).

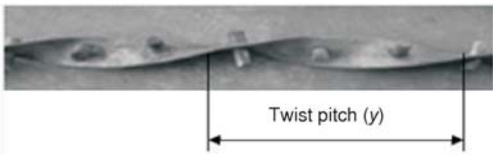

Figure 1(a). Twisted tape with pins

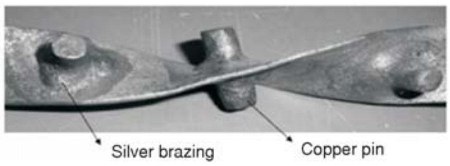

Figure 1(b). Silver brazing between pins and twisted tape

Details of bonded TTB

The twisted tape with pins is inserted into the inner tube of the concentric tube heat exchanger. After the insertion of twisted tape with pins there is an air gap between turbulator and the inner surface of the test section as shown in fig. 2. Due to this air gap there is decrease in heat transfer in radial direction because there is no metal to metal contact between turbulator and inside wall of test section. In order to avoid this air gap soldering wire is inserted inside the test tube along

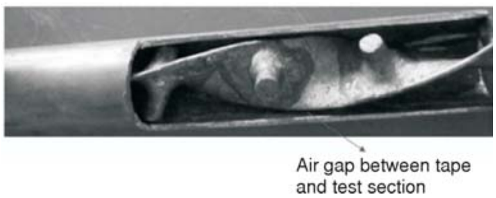

Figure 2. Air gap between TTP and wall of the test section

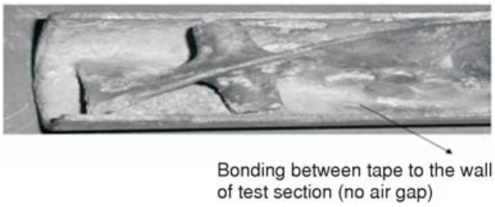

Figure 3. Bonding between the twisted tapes with pins to the inner surface of test section

with the twisted tape with pins. Then the test tube is immersed in the flux and heated in a furnace for about 600 °C. When the tube is maintained at this temperature the solder wire gets melted between the turbulator and the inner surface of the test tube. Then the test tube is taken from furnace and allowed to cool. At this point, the melted solder wire solidifies and bonding takes place between the twisted tape and the inner surface of the test section as shown in fig. 3. After this process the inner tube of the test section is inserted into the outer tube of diameter 50 mm and the both ends of the heat exchanger is closed with cover plate. Finally, all the thermocouples are positioned at various sections of the heat exchangers.In this manner, the concentric tube heat exchanger is fabricated for the TTP y/w = 3.33. The same procedure is repeated to fabricate the concentric tube heat exchanger for TTP of y/w = 4.29 and 5.71.

Experimental set-up

The schematic diagrams of the experimental set-up are shown in fig. 4(a) and 4(b). The details of the experimental set-up are same as explained in the previous paper [17].

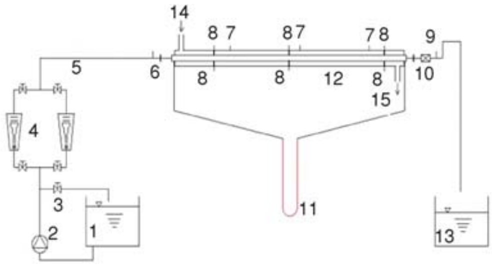

Figure 4(a). Experimental set-up
1 – Inlet water tank, 2 – Centrifugal pump, 3 – Ball valve, 4 – Rotometer, 5 – Calming section, 6 – Inlet water thermocouple, 7 – Steam temperature thermocouple, 8 – Wall temperature thermocouple, 9 – Outlet temperature thermocouple, 10 – Mixing chamber, 11 – U tube manometer, 12 – Test section, 12 – Test section, 13 – Water collection tank, 14 – Steam in, 15 – Steam + condensate out

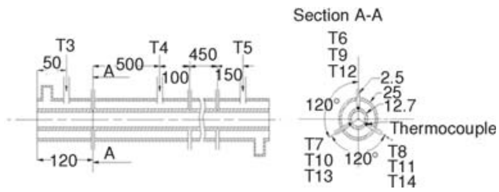

Figure 4(b). Thermocouples location
T3, T4 and T5 – thermocouples used to measure the saturation temperature of steam T6, T7, T8, T9, T10, T11, T12, T13, and T14 – thermocouples used to measure the wall temperature

Experimental procedure

Experimental procedure for TTP

After the leakage check of experimental set-up, the water is sent into the inner tube of the tube-in-tube heat exchanger by centrifugal pump and the steam produced by a diesel fired boiler, goes into the annulus of the test section. The steam pressure is set to 0.2 MN/m^2 to maintain constant wall temperature boundary condition and drain is opened to evacuate steam and condensate in the annulus. When the pressure drop of the manometer reaches steady-state, the temperature of water at inlet, outlet, and the tube wall temperatures are recorded. Then the water flow rate is set at a different value and the test is repeated. All the tests are completed for various y/w ratios of TTP.

Experimental procedure for bonded TTP

Heat exchanger with bonded twisted tape with pins (TTPB) of $y/w = 3.33$ is positioned in the supporting stand and then the calming section and mixing section are connected with the tube in tube heat exchanger. Pressure tap connection is provided at both the ends of the test section. After the mentioned procedure, the experimental set-up is ready to take readings. Experimental procedure is similar to the one explained in the subsection *Experimental procedure for TTP* for the twisted tape with pins. All the tests are completed for various y/w ratios of TTPB bonded with to the inner surface of the test section.

Experimental uncertainty

In this paper, estimation of uncertainty is done based on research paper by Moffat [18]. The uncertainty for a single measurement on the experimentally calculated result, for only that one measurement can be found using eq. (1):

$$\partial R_{\mathrm{X_i}} = \frac{\partial R}{\partial X_{\mathrm{i}}} \int X_{\mathrm{i}} \tag{1}$$

When more independent variables are used in the function R, the individual terms are combined by root-sum-square method, eq. (2):

$$\partial R = \sqrt{\sum_{i=1}^{N} \left(\frac{\partial R}{\partial X_{\mathrm{i}}} \partial X_{\mathrm{i}} \right)^2} \tag{2}$$

Equation (3) is used to consider the relative errors in the individual factors denoted by x_{n}:

$$w(k, h, D, f, v, \ldots) = \sqrt{(x_1)^2 + (x_{21})^2 + \cdots + (x_n)^2} \tag{3}$$

Reynolds number uncertainties can be calculated by combinations of eqs. (4) and (5). Equation (6) is the resultant equation after combining eqs. (4) and (5):

$$\mathrm{Re} = \frac{VD}{u} \tag{4}$$

$$w_{\mathrm{Re}} = \sqrt{\left(\frac{\partial \mathrm{Re}}{\partial V} W_{\mathrm{v}} \right)^2 + \left(\frac{\partial \mathrm{Re}}{\partial D} \right)^2 + \left(\frac{\partial \mathrm{Re}}{\partial u} W_{\mathrm{u}} \right)^2} \tag{5}$$

$$\frac{w_{\mathrm{Re}}}{\mathrm{Re}} = \sqrt{\left(\frac{w_{\mathrm{V}}}{V} \right)^2 + \left(\frac{w_{\mathrm{D}}}{D} \right)^2 + \left(\frac{w_{\mathrm{u}}}{u} \right)^2} \tag{6}$$

Each of the measured physical properties consists of non-dimensional parameters. The uncertainties of the non-di mensional parameters for each of the measured physical properties are given in tab.1. Maximum values of uncertainty calculations for Re, Nu, and f are 7.3%, 9.8%, and 13.2%, respectively.

Table1. Uncertainties values for the relevant variable

Variable	Uncertainty, [%]
Water density, ρ	1.4
Specific heat capacity of water, C_p	2.9
Diameter, D	1.2
Thermal conductivity of water, k	2.1
Dynamic viscosity of water, μ	2.6
Water flow rate, m	4.2
Pressure drop, ΔP	4.4

Friction factor calculation

The pressure drop is determined from the differences in the level of manometer fluid. The fully developed friction factor is calculated from the equation:

$$f = \frac{2\Delta P D_i}{\rho u_m^2 L} \tag{7}$$

where ΔP is the pressure drop over length L.

Data reduction equations

The average inside heat transfer coefficient and the mean Nusselt number for the plain and the different twisted tapes are evaluated as follows:

$$Q = \dot{m} C_p (T_o - T_i) = h_i A_i (\Delta T_i)_m \tag{8}$$

where

$$A_i = \pi D_i L \tag{9}$$

$$(\Delta T_i)_m = \frac{(\bar{T}_w - T_i) - (\bar{T}_w - T_o)}{\ln\left[\dfrac{\bar{T}_w - T_i}{\bar{T}_w - T_o}\right]} \tag{10}$$

$$\bar{T}_w = \frac{T_w}{9} \tag{11}$$

where T_w is the local wall temperature of the tube. To obtain the average wall temperatures, the temperature of the tube wall is measured at nine different positions. Other important reason of providing the thermocouple at nine different positions is to obtain the circumferential temperature variation, which was found to be negligible. It must be noted that the local wall temperature T_w is measured within the tube wall (at a position of 0.5 mm from the inner wall). Such a measurement technique for the wall temperature resulted in a maximum error of 2% in the Nusselt number because of negligible wall resistance. The average inside heat transfer coefficient and the mean Nusselt number are determined by:

$$h_i = \frac{Q}{A_i (\Delta T_i)_m} \tag{12}$$

$$\mathrm{Nu} = \frac{h_i D_i}{k} \tag{13}$$

All properties are evaluated at the bulk mean temperature.

Results and discussion

Plain tube data

The present experimental results on heat transfer and friction characteristics in a plain tube are first validated in terms of Nusselt number and friction factor. It is important to compare the experimental results obtained for the fully developed turbulent flow with the correlations from the literature. The Nusselt number and friction factor obtained from experiment on the plain tube are compared with the correlations of Sieder and Tate [19], Petukhov [20], and

Blassius correlations, respectively, found in the open literature for turbulent flow in circular tubes.

– Sieder and Tate correlation

$$\mathrm{Nu} = 0.027\,\mathrm{Pr}^{0.33}\,\mathrm{Re}^{0.8}\left(\frac{\mu_\mathrm{b}}{\mu_\mathrm{w}}\right)^{0.14} \tag{14}$$

– Petukhov correlation

$$Nu = \frac{\left(\dfrac{f}{8}\right)\mathrm{Re\,Pr}}{1.07 + 12.7\sqrt{\dfrac{f}{8}}(\mathrm{Pr}^{2/3} - 1)}\left(\frac{\mu_\mathrm{b}}{\mu_\mathrm{w}}\right)^{0.11} \tag{15}$$

where f is the friction factor and for plain tube it is given as [20]:

$$f = (1.82\log(\mathrm{Re}) - 1.64)^{-2} \tag{16}$$

– Blassius correlation

$$f = 0.3164(\mathrm{Re})^{-0.25} \tag{17}$$

Figure 5 shows the variation of Nusselt number obtained from experiment and Nusselt number estimated using Sieder and Tate with Reynolds number for the case of plain tube.

It is observed from fig. 5 that the Nusselt number estimated from experimental data lies within ±15% that of theoretical values calculated using Sieder and Tate and Petukhov correlation. Plain tube experimental correlation for Nusselt number and friction factor are obtained as given in eqs.18 and 19:

$$\mathrm{Nu} = 0.228\mathrm{Re}^{0.641}\mathrm{Pr}^{0.4} \tag{18}$$

$$f = 2.248\mathrm{Re}^{-0.45} \tag{19}$$

Figure 6 shows the variation of friction factor with Reynolds number. The experimental data matches with the Blassius and Petukhov correlation for plain tube with a discrepancy of less than ±7%.

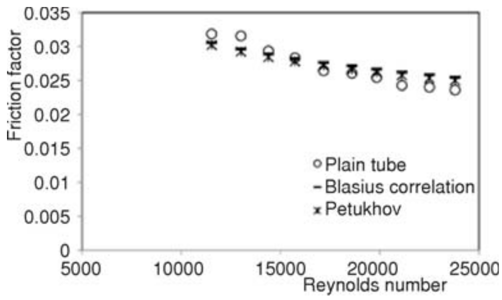

Figure 5. Data verification of Nusselt number for plain tube

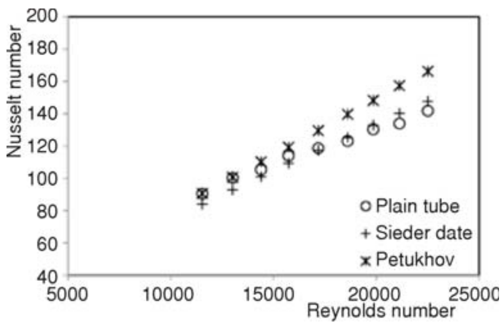

Figure 6. Data verification of friction factor for plain tube

Effect of y/w ratios on heat transfer enhancement for TTP and TTPB

Figure 7 shows the variation of Nusselt number with Reynolds number for the tube fitted with TTP of three different *y/w* ratios (3.33, 4.29, and 5.71). From fig. 7 it can be concluded that the Nusselt number for the tube fitted with TTP are higher than that of plain tube for a given Reynolds number. This is because the TTP interrupts the development of the boundary layer of

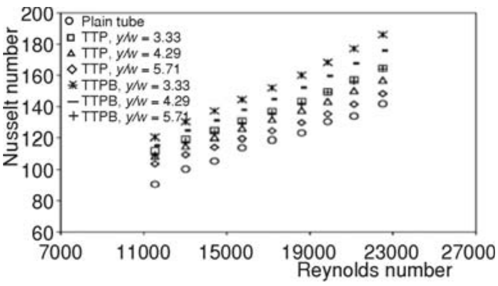

Figure 7. Nusselt number *vs*. Reynolds number for all TTP and TTPB configurations

the fluid flow near the wall of the test section. Hence it increases the average temperature of the fluid in the radial direction. Due to the larger contact surface area, the heat transfer rate increases. Also it creates the turbulence and whirling motion to the water which is flowing inside the test section. The whirling makes the flow to be highly turbulent, which leads to improved convection heat transfer. As the Reynolds number increases for a given *y/w* ratio of the TTP, the Nusselt number also increases, indicating enhanced heat transfer coefficient. As the pitch of the tape decreases, the intensity of swirl flow increases leading to higher heat transfer rate and the maximum being for the TTP of y/w = 3.33. Throughout the experimental results it is seen that the smaller *y/w* (3.33) yields the higher values of heat transfer of about 23.86% than plain tube. Similarly for y/w = 4.29 and 5.71 the enhancement are 19.9% and 14.4%, respectively.

The main objective of providing the TTP is to transfer the heat from the wall surface of the test tube to the centre core of the water and to disturb the boundary layer. When only TTP is used in the test section, there is air gap between the twisted tape with pins and the inner surface of the test section as shown in fig. 2. The water near the wall receives more heat because it is in direct contact with the wall surface. But the water at the centre of the tube receives less heat due to the fact the heat will be transferred to centre of the flow due to normal convective mode of heat transfer. Due to this, there is decrease in heat transfer when only TTP is used. In order to increase the temperature of water which is flowing at the centre of the tube, bonding is done between the twisted tape with pins and the inner surface of the test section. Due to this process, the air gap is filled with bonding due to soldering. Consequently, there will be metal to metal contact between the twisted tape and the wall of the test section. The TTPB disturbs the boundary layer near the wall of the test section and hence there is increase in convective heat transfer. Also the TTPB picks the heat from the wall and transfers the heat to the centre of the tube. So the temperature of the water at the centre of the tube also increases. Due to the above mentioned reason, there is increase in heat transfer for TTPB as shown in fig. 7. TTPB with smaller *y/w* (3.33) yields the higher values of heat transfer of about 32.9% than plain tube. Similarly for y/w = 4.29 and 5.7, the enhancement are 27.5% and 20.75%, respectively. From fig. 7 it can be seen that for TTPB with y/w = 3.33, the enhancement in heat transfer is very high when compared to other configurations of TTP and TTPB. The percentage increase in heat transfer of TTPB (y/w = 3.33) when compared to TTP of same y/w = 3.33 is 13.14%. Similarly for y/w = 4.29 and 5.71 of TTPB, the enhancement are 11.95% and 10.81%, respectively, when compared to TTP of same y/w = 4.29 and 5.71.

Effect of y/w ratios on friction factor for TTP and TTPB

Generally, the friction factor decreases conventionally with the increasing Reynolds number for different *y/w* ratios. From fig. 8 it can be seen that friction factor for the tube fitted with TTP is higher for a given Reynolds number. It indicates that friction factor for a given Reynolds number increases with the decreasing *y/w* ratio due to swirl flow generated by TTP and reaches the maximum for y/w = 3.33. From fig. 8, it can be seen that the friction factor for

y/w = 4.29 and 5.71 are less when compared with y/w = 3.33.This is due to less contact surface area of the turbulator.

From fig. 8 it can be seen that friction factor for the tube fitted with TTPB inserts is higher for a given Reynolds number when compared with plain tube. It indicates that friction factor for a given Reynolds number increases with the decreasing y/w ratio due to swirl flow generated by TTPB and reaches the maximum for y/w = = 3.33. From fig. 8 it can be seen that the friction factor for y/w = 4.29 and 5.71 are less when compared with y/w = 3.33.This is due to the less contact surface area of the TTPB with y/w ratio 4.29 and 5.71. From fig.8 it can be seen that the friction factor for TTPB is high when compared to the TTP for the same y/w = 3.33. The percentage increase in friction factor for TTPB of y/w = 3.33 is only 4% when compared to TTP of y/w = = 3.33. But the augmentation of heat transfer is 13.14% more than the increase in pressure drop for TTPB (y/w = 3.33) when compared to TTP of same y/w = 3.33.This is due to the increase in heat transfer in radial direction due to the bonding of the turbulator. The percentage increase in heat transfer is lower for TTP of y/w = 5.71 when compared to all the other TTP and TTPB configurations. The percentage increase in heat transfer for TTP (y/w = 5.71) is about 3.8%.This is due to less turbulence intensity created by the twisted tape with pins.

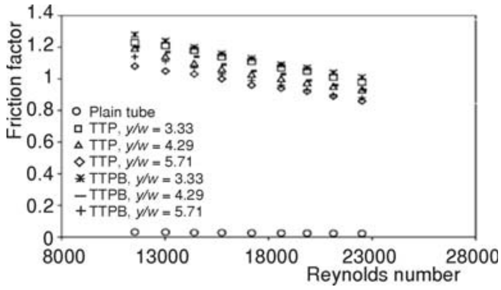

Figure 8. Friction factor *vs.* Reynolds number for all TTP and TTPB configurations

Empirical correlation for combined TTP and TTPB with different *y/w* ratios

The empirical correlation for combined TTP and TTPB with different y/w ratios is given by:

$$\mathrm{Nu} = 0.528\,\mathrm{Re}^{0.543}\,\mathrm{Pr}^{0.4}\left(\frac{y}{w}\right)^{-0.192} \tag{20}$$

$$f = 51.29\,\mathrm{Re}^{-0.36}\left(\frac{y}{w}\right)^{-0.259} \tag{21}$$

The fitted values of Nusselt number by eq. (20) and friction factor by eq. (21) are compared with the experimental values and are shown in figs. 9 and 10, respectively.

The correlated Nusselt number and friction factor results in maximum discrepancies of ±7.28 and ±7.16%, respectively, when compared with experimental results.

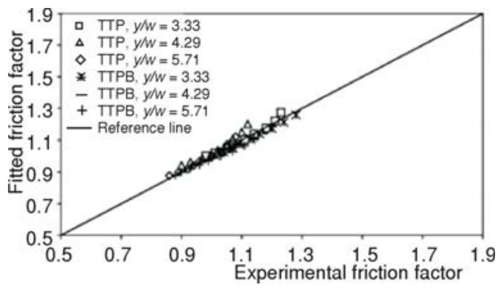

Figure 9. Comparisons of experimental and predicted Nusselt number for tube with TTP and TTPB configurations

Conclusions

For practical application of thermal systems operating under turbulent flow conditions twisted tape with pins are desirable in order to

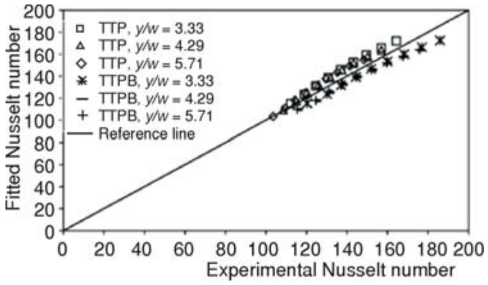

Figure 10. Comparisons of experimental and predicted friction factor for tube with TTP and TTPB configurations

increase the heat transfer rate. Experimental investigations of heat transfer and friction factor characteristics of a circular tube fitted with full-length twisted tape with pins of different *y/w* ratios have been studied for the turbulent regime, Re = 10000-23 000. Experimental data obtained were compared with those obtained from the theoretical data of plain tube. The following conclusions were drawn based on experimental and theoretical investigations.

- Nusselt number increases with the decrease of the ratio *y/w*.
- The friction factor also increases with the decreasing twist pitch.
- The empirical correlations developed relating pitch and Reynolds number were matching with the experimental data within ±7.28, and ±7.16% for Nusselt number and friction factor, respectively.
- The findings indicate that the use of bonded twisted tape with pins in the tube-in-tube heat exchanger enhances the heat transfer with considerable pressure drop.

Nomenclature

A_i	– inside surface area of test section area, [m^2]
C_p	– specific heat at constant pressure, [$kJkg^{-1}°C^{-1}$]
D_i	– inside diameter of test section, [mm]
f	– friction factor
h_i	– average convective heat transfer – coefficient, [$Wm^{-2}°C^{-1}$]
k	– thermal conductivity of fluid, [$Wm^{-1}°C^{-1}$]
L	– length of the test section, [m]
m	– mass flow rate of water, [kgs^{-1}]
Nu	– Nusselt number (= h_iD_i/k)
P	– pitch of the wire coil matrix turbulator, [mm]
ΔP	– pressure drop of fluid, [Nm^{-2}]
Q	– heat transfer rate, [KW]
Re	– Reynolds number based on internal diameter of the tube
T_b	– bulk mean temperature of water, [°C]
T_i	– inlet temperature of fluid, [°C]
T_o	– outlet temperature of fluid, [°C]
T_w	– local wall temperature, [°C]
$\overline{T}_w$	– average wall surface temperature in the test section, [°C]
u_m	– bulk average fluid velocity, [ms^{-1}]
y	– twisted tape pitch, [m]

Greek symbols

μ_b	– viscosity at bulk mean temperature of fluid [Nsm^{-2}]
μ_w	– viscosity at average wall surface temperature [Nsm^{-2}]
ρ	– density of fluid, [kgm^{-3}]

References

[1] Manglik, R. M., Bergles, A. E., Heat Transfer and Pressure Drop Correlations for Twisted-Tape Inserts in Isothermal Tubes, Part I. Laminar Flows, *ASME Journal of Heat Transfer, 115* (1993), 4, pp. 881-889
[2] Manglik, R. M., Bergles, A. E., Heat Transfer and Pressure Drop Correlations for Twisted-Tape Inserts in Isothermal Tubes, Part II, Transition and Turbulent Flows, *ASME Journal of Heat Transfer, 115* (1993), 4, pp. 890-896
[3] Agarwal, S. K., Raja Rao, M., Heat Transfer Augmentation for the Flow of a Viscous Liquid in Circular Tubes Using Twisted Tape Inserts, *International Journal of Heat and Mass Transfer, 39* (1996), 17, pp. 3547-3557
[4] Naphon, P., Effect of Coil-Wire Insert on Heat Transfer Enhancement and Pressure Drop of the Horizontal Concentric Tubes, *International Communications in Heat and Mass Transfer, 33* (2006), 6, pp.753-763

[5] Naphon, P., Sriromruln, P., Single-Phase Heat Transfer and Pressure Drop in the Micro-Fin Tubes with Coiled Wire Insert, *International Communications in Heat and Mass Transfer, 33* (2006), 2, pp. 176-183
[6] Gul, H., Evin, D., Heat Transfer Enhancement in Circular Tubes Using Helical Swirl Generator Insert at the Entrance, *International Journal of Thermal Sciences, 46* (2007), 12, pp. 1297-1303
[7] Eiamsa-ard, S., Promvonge, P., Heat Transfer Characteristics in a Tube Fitted with Helical Screw-Tape With/Without Core Rod Inserts, *International Communications in Heat and Mass Transfer, 34* (2007), 2, pp. 176-185
[8] Sivashanmugam, P., Suresh, S., Experimental Studies on Heat Transfer and Friction Factor Characteristics in Turbulent Flow through a Circular Tube Fitted with Helical Screw-Tape Inserts, *Journal of Chemical Engineering and Process, 46* (2007), 12, pp. 1292-1298
[9] Promvonge, P., Thermal Augmentation in Circular Tube with Twisted Tape and Wire Coil Turbulators, *Energy Conversion and Management, 49* (2008), 11, pp. 2949-2955
[10] Sreenivasulu, T., Prasad, B. V. S. S. S., Flow and Heat Transfer Characteristics in an Annulus Wrapped with a Helical Wire, *International Journal of Thermal Sciences, 48* (2009), 7, pp. 1377-1391
[11] Eiamsa-ard, S., *et al*., Thermal Characteristics in a Heat Exchanger Tube Fitted with Dual Twisted Tape Elements in Tandem, *International Communications in Heat and Mass Transfer, 37* (2010), 1, pp. 39-46
[12] Eiamsa-ard, S., *et al*., Thermohydraulic Investigation of Turbulent Flow through a Round Tube Equipped with Twisted Tape Consisting of Centre Wings and Alternate-Axes, *Experimental Thermal Fluid Science, 34* (2010), 8, pp. 1151-1161
[13] Eiamsa-ard, S., *et al.*, Influences of Peripherally-Cut Twisted Tape Insert on Heat Transfer and Thermal Performance Characteristics in Laminar and Turbulent Tube Flows, *Experimental Thermal Fluid Science, 34* (2010), 6, pp. 711-719
[14] Cui, Y.-Z., Tian, M.-C., Three-Dimensional Numerical Simulation of Thermal-Hydraulic Performance of a Circular Tube with Edge Fold-Twisted-Tape Inserts, *Journal of Hydrodynamics, 22* (2010), 5, pp. 662-670
[15] Seemawute, P., Eiamsa-ard, S., Thermohydraulics of Turbulent Flow through a Round Tube by a Peripherally-Cut Twisted Tape with an Alternate Axis, *International Communications in Heat and Mass Transfer, 37* (2010), 6, pp. 652-659
[16] Naphon, P., Suchana, T., Heat Transfer Enhancement and Pressure Drop of the Horizontal Concentric Tube with Twisted Wires Brush Inserts, *International Communications in Heat and Mass Transfer, 38* (2011), 2, pp. 236-241
[17] Selvam, S., *et al*., Experimental Studies on Effect of Wire Coiled Coil Matrix Turbulators with and without Bonding on the Wall of the Test Section of Concentric Tube Heat Exchanger, *Thermal Science, 16* (2012), 4, pp. 1151-1164
[18] Moffat, R. J., Describing the Uncertainties in Experimental Results, *Experimental Thermal Fluid Science, 1* (1988), 1, pp. 3-17
[19] Sieder, E. N., Tate, G. E., Heat Transfer and Pressure Drop of Liquids in Tubes, *Industrial and Engineering Chemistry, 28* (1936), 12, pp. 1429-1439
[20] Petukhov, B. S., Heat Transfer and Fraction in Turbulent Pipe Flow with Variable Physical Properties, in: Advances in Heat Transfer, Vol. 6. Academic Press, San Diego, Cal., USA, 1970, pp. 504-564

Paper submitted: August 7, 2012
Paper revised: February 27, 2013
Paper accepted: March 30, 2013