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Heat Transfer and Friction Factor in a Tube Equipped with U-cut Twisted Tape Insert

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Abstract

Experimental investigation of heat transfer and friction factor characteristics of circular tube fitted with plain twisted tapes (PTT) and U–cut twisted tapes (UTT) with twist ratios y = 2.0, 4. 4 and 6.0 were studied. The experimental data obtained from plain tube and PTT were verified with the standard correlation to ensure the validation of experimental results. The experimental results reveal that heat transfer rate, friction factor and thermal enhancement factor in the tube equipped with UTT significantly higher than those in the tube fitted with PTT and plain tube. The additional disturbance and secondary flow in the vicinity of the tube wall generated by the UTT compared to that induced by the PTT is referred as the reason for the enhancement. Subsequently an empirical correlation is also formulated to match with experimental results with ± 6% and ± 5%, variation respectively for Nusselt number and friction factor.

Keywords: heat transfer enhancement; friction factor; plain twisted tape; U –cut twisted tape; secondary flo

Nomenclature

A	Area	: [m^2]
C_p	Specific heat	: [J/kg K]
d	Tube diameter,	: [m]
d_e	U-cut depth	: mm
D_h	Hydraulic diameter	: [m]
f	Friction factor	
h	Heat transfer coefficient	: [W/m^2 K]
H	Pitch length based on 180°	: [m]
k	Thermal conductivity	: [W/m K]
L	Tube length	: [m]
m	Mass flow rate	: [kg/s]
Nu	Nusselt number	
ΔP	Pressure drop	
Pr	Prandtl number	
Q	Heat transfer rate	: [W]
Re	Reynolds number	
T_h	Hot water temperature	: °C
T_c	Cold water temperature	: °C
ΔT_{lm}	Logarithmic mean temperature difference	
U	Overall heat transfer coefficient	: [W/m^2 K]
u	Velocity	: [m/s]
W	Twisted tape width	: mm
w	U- cut width	: mm
y	Twist ratio	
ρ	Density	: [kg/m^3]
μ	Dynamic viscosity	: [kg/m-s]
η	Thermal enhancement factor	

Subscripts

avg	: Average
a	: Annulus
c	: Cold water
h	: Hot water
i	: Inner
lm	: Logarithmic mean temperature
o	: Outer
p	: Plain tube
t	: Turbulator
1	: Inlet
2	: Outlet

Abbreviations

PTT	: Plain twisted tape
UTT	: U –cut twisted tape

1. Introduction

Heat transfer augmentation techniques are widely used in areas such as heat recovery process, air conditioning and refrigeration systems, and chemical reactors. Passive and active methods of heat transfer augmentation techniques have been discussed [1]. The passive techniques particularly twisted tape and wire coil insert are economical heat transfer augmentation tools [2]. The heat transfer and friction factor characteristics in a circular tube fitted with different tube inserts were experimentally investigated and correlations for Nusselt number and

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friction factor were proposed [3-7]. The heat transfer and friction factor characteristics were experimentally compared between smooth twisted tape with broken and serrated twisted tape inserts [8 & 9]. More information about heat transfer by means of twisted tapes fitted in a circular tube can be viewed in other reports [10-14].

Based on the available literature, it was pointed out that the modification on PTT i.e. small cuts on the tape [6-9], for example delta –winglet tape (DWT), peripherally –cut tape (PT), broken tape and serrated tape gave assurance for enhancement of both heat transfer rate and thermal enhancement factor. The reason behind the high thermal enhancement factor is that those small gaps bring pressure drop in the system to the reasonable level.

The present work reports the experimental work on heat transfer rate and friction factor characteristics of tube in tube heat exchanger fitted with PTT and UTT for twist ratios 2.0, 4.4 and 6.0 with Reynolds number between 2000 and 12000. The modified twisted tapes comprise U – cut alternately in the peripheral region of the tape. This type of tape is believed to perform in same manner as mentioned in the literature for the case of broken or spiky tape, delta –winglet tape and peripherally –cut tapes. The experimental results obtained for the tube fitted with UTT were also compared with those for the tube fitted with PTT and the plain tube.

2. Experimental details

Schematic diagram of the experimental set-up is shown in figure 1. It consists of two concentric tubes in which hot water flows through the inner tube (Copper tube, d_i = 25 mm, L = 2000 mm) and cold water flows in counter flow through annulus (GI pipe, d_i = 54.5 mm). The outer tube is insulated with asbestos rope and glass wool to minimize the heat loss with the surroundings (Insulation thickness = 10 mm). Two calibrated crystal rotameters having flow ranges of 0-20 l min-1 with ±0.1 l min-1 accuracy are used to measure the cold and hot water flow rates. Seven RTD Pt 100 type temperature sensors with ±0.1ºC accuracy are used to measure the inlet and outlet temperature of the hot and cold water.

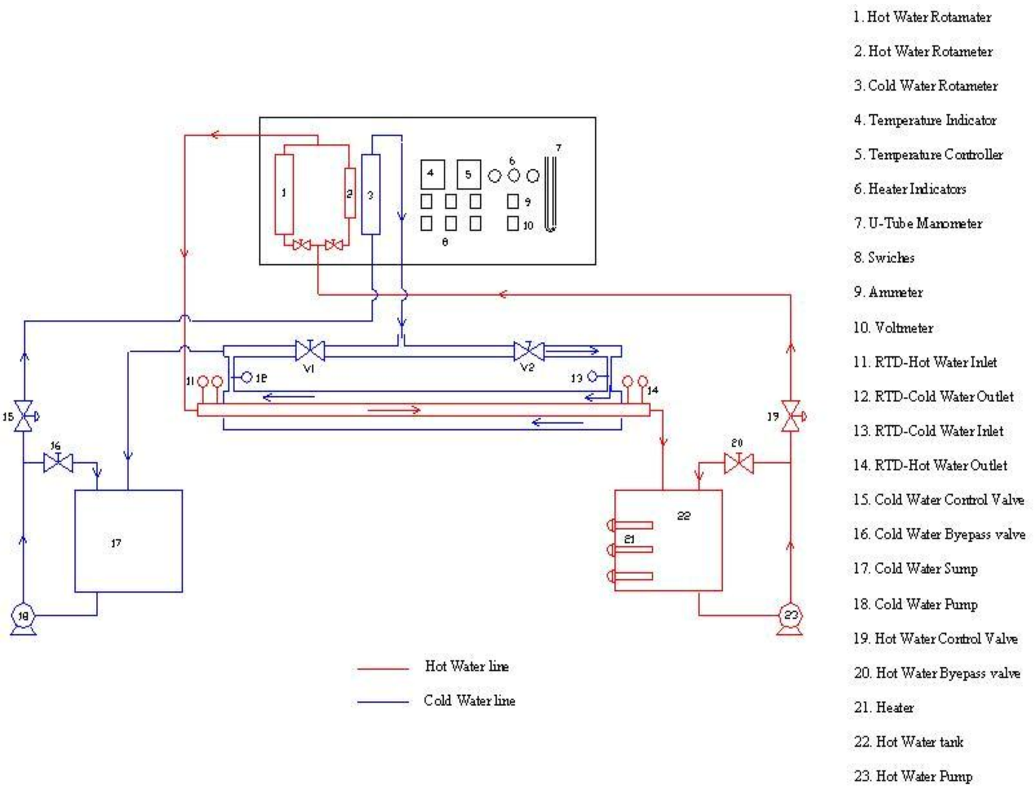


Figure 1: Schematic diagram of the experimental set-up.

Twisted tapes are made up of aluminum strips [11] of thickness 1.5 mm and width 23.5 mm. The twist ratio (y) is defined [2, 3 & 11] by ratio between one length of twist (or) pitch length (H = 50, 110, and 150 mm) to diameter. In the experimentation PTT and UTT with twist ratios 2.0, 4.4 and 6.0 are used. Geometries of the PTT and UTT are shown in figure 2(a-b). U -cut twisted tapes (figure 2b) with U –cut (8 mm depth and 8mm width) alternately in the peripheral region of the tape to increase the disturbance near the walls of the test section. The water is heated using 3 kW water heaters and the desired temperature is controlled by temperature controller. The inlet temperatures at the hot and cold water sides were kept constant at 54ºC and 30ºC, respectively. The cold water was constantly flowed at 0.166 kg s-1 whereas the hot water flow rate was adjusted from 0.033 kg s-1 to 0.12 kg s-1.

As steady state conditions were reached, the inlet and outlet temperatures of hot and cold water were recorded and pressure drop was measured using U tube manometer (manometric fluid –Carbon tetra chloride) for the case of plain tube. Thereafter, the experiment was repeated for PTT and UTT.

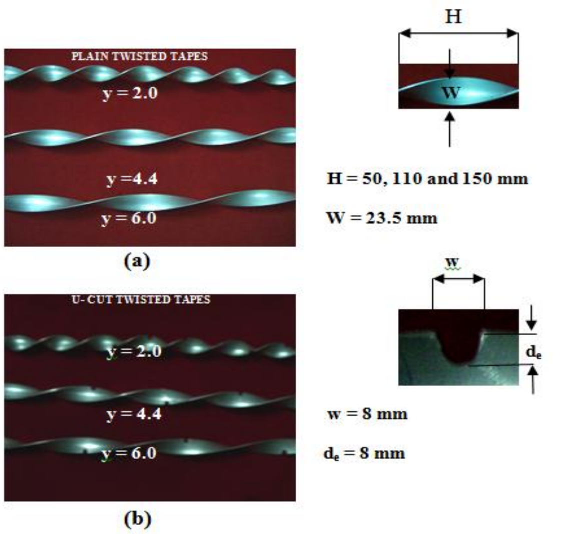


Figure 2: Geometries of twisted tapes with twist ratios y= 2.0, 4.4 and 6.0 (a) PTT (b) UTT.

3. Data reduction

The data reduction [3,10] of the measured results is summarized as follows:

Heat transferred to the cold water in the test section:

$$Q_c = m_c C_p (T_{c2} - T_{c1}) \tag{1}$$

Heat transferred from the hot water in the test section

$$Q_h = m_h C_p (T_{h1} - T_{h2}) \tag{2}$$

The error [14] in the present heat exchanger can be given as:

$$\%error = \left(\frac{Q_h - Q_c}{Q_c} \right) \times 100\% \tag{3}$$

The percentage of error was found out to be 2% to 8%. Thus it was concluded that the heat loss to the surroundings was reasonably small.

The average heat transfer rate for hot and cold water side

$$Q_{avg} = \frac{Q_c + Q_h}{2} \tag{4}$$

The over all heat transfer coefficient

$$U = \frac{Q_{avg}}{A_i \Delta T_{lm}} \tag{5}$$

Where,

$$A_i = \pi d_i L$$

The tube side heat transfer coefficient (h_i)

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_a} \tag{6}$$

Where the annulus side heat transfer coefficient (h_a) is estimated using the correlation of Dittus -Boelter equation:

$$Nu_a = \frac{h_a D_h}{k} = 0.023\, Re^{0.8}\, Pr^{0.4} \tag{7}$$

Where, D_h = Hydraulic diameter = $d_i - d_o$

Thus,

$$Nu_i = \frac{h_i d_i}{k} \tag{8}$$

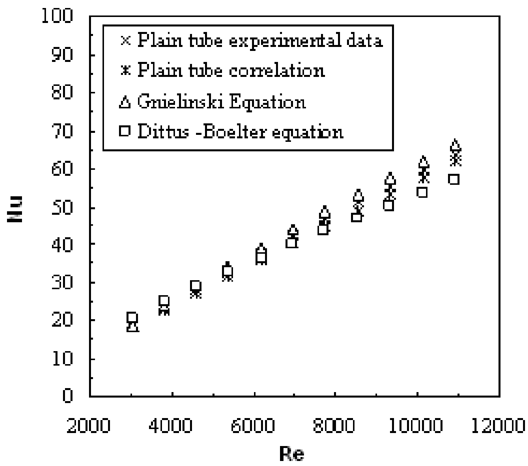
Friction factor

$$f = \frac{\Delta p}{\left[\frac{L}{d_i} \right] \left[\frac{\rho u^2}{2} \right]} \tag{9}$$

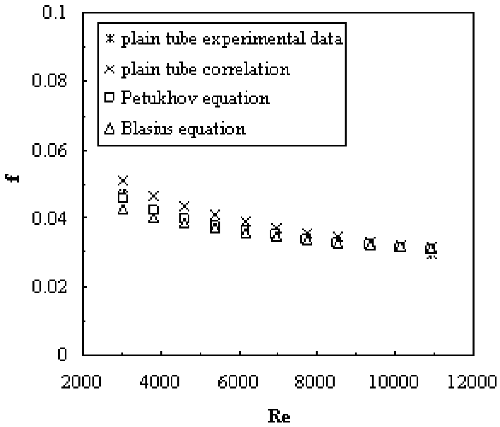
4. Results and Discussion

4.1. Plain tube data:

The variation of Nusselt number with Reynolds number for plain tube is shown figure 3a.



(a)



(b)

Figure 3: Validation of Plain tube: (a) Nusselt number and (b) friction factor.

The experimental data are matching with the plain tube forced convection correlations [15] of Dittus–Boelter (1930) (10) and Gnielinski (1976) equation (11) with the discrepancy of ± 8% and ± 5.0% respectively for the Nusselt number.

$$Nu = 0.023\,Re^{0.8}\,Pr^{0.3} \tag{10}$$

$$Nu = \frac{\left(\frac{f}{8}\right)\left(Re - 1000\right)\,Pr}{1 + 12.7\left(\frac{f}{8}\right)^{0.5}\left(Pr^{\frac{2}{3}} - 1\right)} \tag{11}$$

The variation of friction factor with Reynolds number for plain tube is shown in figure 3b. The data obtained by the experiment is compared with Blasius (12) and Petukhov (1970) equation (13) with the deviation of ± 10% and ± 8% respectively for the friction factor.

$$f = 0.0791Re^{-0.25} \tag{12}$$

$$f = \left(0.790\ln Re - 1.64\right)^{-2} \tag{13}$$

In addition, the experimental results of the plain tube are correlated and the equation for Nusselt number (14) and friction factor (15) as follows:

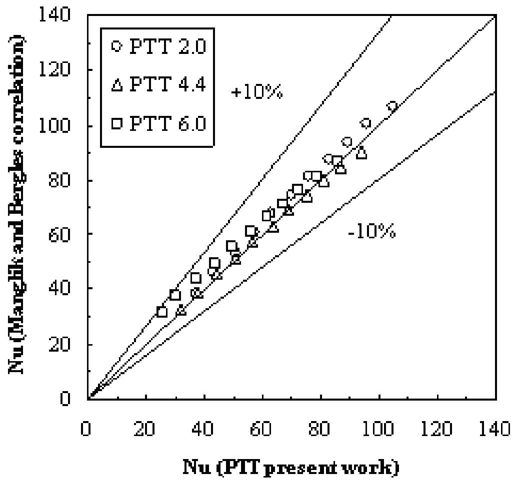
$$Nu = 0.00595\,Re^{0.95}\,Pr^{0.33} \tag{14}$$

$$f = 0.255\,Re^{-0.374} \tag{15}$$

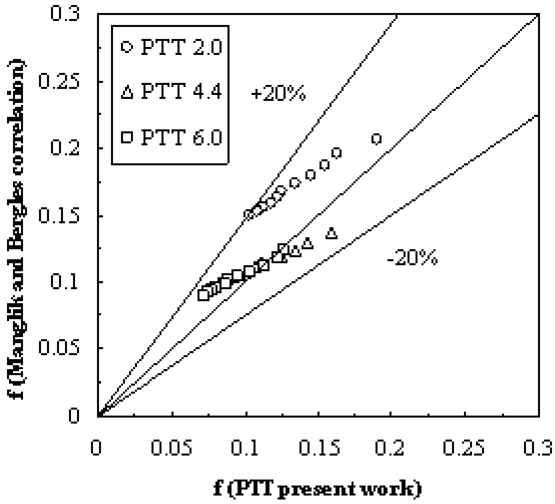
The equations (14) and (15) are found to represent the experimental data within ±4% for Nusselt number and ±6% for friction factor deviation is shown in figure. 3(a-b). These correlations (14) & (15) are useful to evaluate the thermal enhancement factor associated by PTT and UTT.

4.2. Plain twisted tape (PTT) data:

The tube fitted with PTT experimental results are validated using the correlations developed by Manglik and Bergles [12] which yields maximum deviation of ± 20 % [6, 7, 11 & 13] for both Nusselt number and friction factor respectively. Nusselt number and friction factor of a tube fitted with PTT are compared with the results obtained from the correlations by Manglik and Bergles [12] for the twist ratios of y = 2.0, 4.4 and 6.0 as demonstrated in figure 4(a-b). Apparently, present results reasonably agree well with the available correlations with in ± 10% for Nusselt number and ± 20% friction factor respectively.



(a)



(b)

Figure 4: Validation of Plain twisted tapes (a) Nusselt number and (b) friction factor.

4.3. Effect of U –cut twisted tape (UTT) on heat transfer:

Variation of Nusselt number with Reynolds number in the tube fitted with UTT, PTT and also the plain tube are presented in figure 5. It is observed that for all cases, Nusselt number increases with increasing Reynolds number. As expected, PTT heat transfer rates are higher than those from the plain tube fitted without twisted tape and also lower twist ratio (y =2.0) heat transfer rate is higher than those from higher ones (y = 4.4 and 6.0) due to increase in turbulent intensity and flow length across the range of Reynolds number. Mean Nusselt numbers for PTT with twist ratios, y =2.0, 4.4 & 6.0 are respectively, 1.67, 1.50 and 1.32 times better than that for the plain tube.

Nusselt number (figure 5) in the tube with UTT is higher than those in the plain tube and tube with PTT insert over the range of Reynolds number 2000-12000. UTT provides an additional disturbance to the fluid in the vicinity of the tube wall and vorticity behind the cuts and thus leads to a higher heat transfer enhancement in comparison with plain tube and PTT. In the range of the present experiments considered, mean Nusselt numbers for tube equipped with UTT of twist ratios of 2.0, 4.4 and 6.0, are respectively 1.81, 1.61 and 1.40 times of that plain tube and 1.08, 1.07 and 1.06 times of that for the tube equipped with PTT.

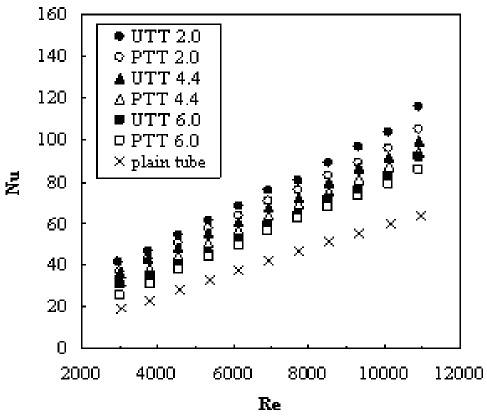


Figure 5: U–Cut (UTT) and Plain twisted tapes (PTT): Nusselt number Vs Reynolds number.

4.4. Effect of U–cut twisted tape (UTT) on friction factor:

Variation of friction factor with Reynolds number in the tube fitted with UTT, the tube fitted with PTT and also the plain tube are depicted in figure 6. It shows that friction factor continues to decrease with Reynolds number and friction factor for lower twist ratio (y = 2.0) is significantly more than that of higher twist ratios (y = 4.4 & 6.0) due to stronger swirl flow in the tube. Over range studied, the mean friction factor for the PTT with twist ratios, y =2.0, 4.4 and 6.0 are respectively, 3.48, 2.92 and 2.45 times higher than that for the plain tube.

It shows (figure 6) that UTT yields higher pressure drop those in the plain tube as well as the tube fitted with PTT. This is because of additional disturbance increases the tangential contact between secondary flow and the wall surface of the tube. Mean friction factor for the UTT with twist ratios of 2.0, 4.4 and 6.0 are respectively, 3.82, 3.28 and 2.8 times of that for the plain tube and 1.09, 1.12 and 1.16 times of that for the tube with PTT insert.

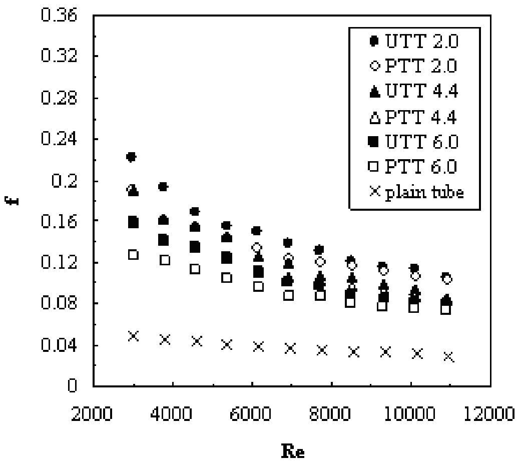


Figure 6: U–Cut (UTT) and Plain twisted tapes (PTT): Nusselt number Vs friction factor.

The following correlations for Nusselt number and friction factor developed for the present experimental results respectively for a plain tube fitted with PTT equation (16, 17) and UTT equation (18, 19) .

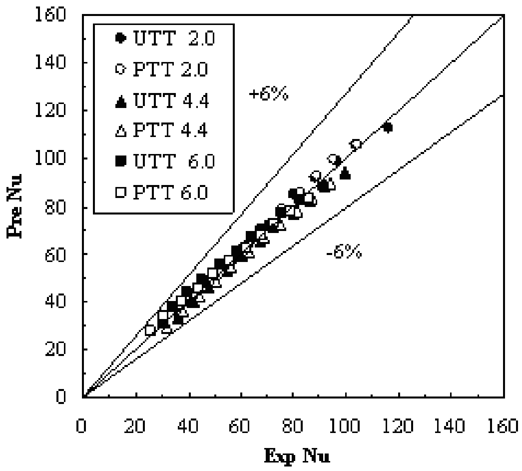
$$Nu = 0.027\,Re^{0.862}\,Pr^{0.33}\,y^{-0.215} \tag{16}$$

$$f = 2.642\,Re^{-0.474}\,y^{-0.302} \tag{17}$$

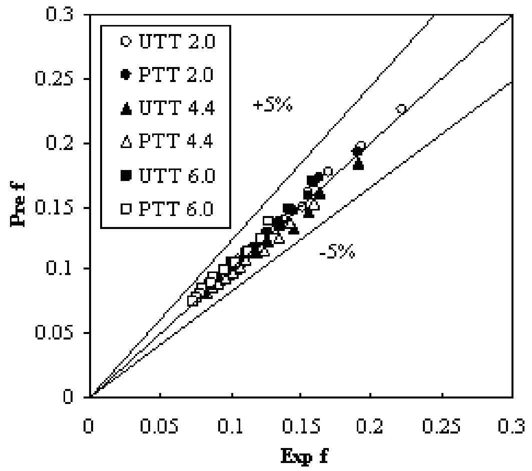
$$Nu = 0.044\,Re^{0.817}\,Pr^{0.33}\,y^{-0.224} \tag{18}$$

$$f = 6.705\,Re^{-0.575}\,y^{-0.257} \tag{19}$$

The fitted values are agreeing with experimental data within ± 6%, ± 5% respectively, for both Nusselt number and friction factor shown in figure 7 (a-b).



(a)



(b)

Figure 7: Comparison between predicted and experimental results (a) Nusselt number and (b) friction factor.

4.5. Thermal enhancement factor for PTT and UTT:

According to the literature studies [6-7] a comparison of heat transfer coefficients in a plain tube (p) and the tube fitted with turbulator (t) was made at the same pumping power since it is relevant to operation cost.

For constant pumping power

$$(\dot{V}\,\Delta P)_p = (\dot{V}\,\Delta P)_t \tag{20}$$

Where the relationship between friction factor and Reynolds number can drawn as below

$$(f\,Re^3)_p = (f\,Re^3)_t \tag{21}$$

Thermal enhancement factor (η) at equal pumping power is defined as ratio of the convective heat transfer coefficient of the tube with turbulator to that of the plain tube which can be expressed as

$$\eta = \left|\frac{h_t}{h_p}\right|_{pp} \tag{22}$$

Using equations (15), (17), (19) and (21), the Reynolds number for the plain tube $(Re)_p$ is written as the function of Reynolds number $(Re)_t$ for the tube with turbulator for PTT equation (23), UTT equation (24)

$$Re_p = 2.436\,Re_t^{0.962}\,y^{-0.115} \tag{23}$$

$$Re_p = 3.473\,Re_t^{0.924}\,y^{-0.098} \tag{24}$$

Employing equations (14), (16), (18) and (22), the enhancement efficiency for the plain twisted tape and U-cut twisted tape can be written as

$$\eta_{(PTT)} = \left|\frac{h_t}{h_p}\right|_{pp} = 1.95\,Re_t^{-0.052}\,y^{-0.106} \tag{25}$$

$$\eta_{(UTT)} = \left|\frac{h_t}{h_P}\right|_{PP} = 2.27\,Re_t^{-0.060}\,y^{-0.131} \tag{26}$$

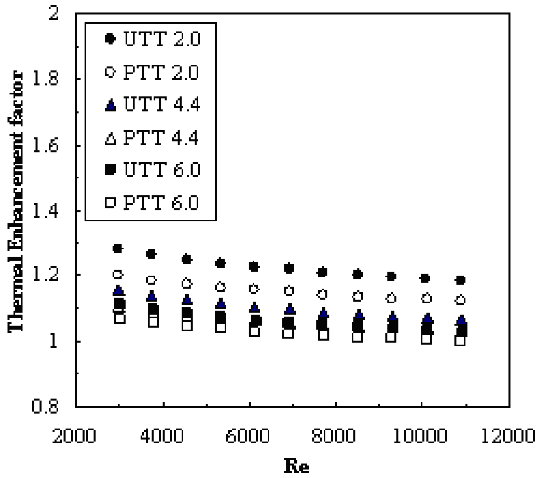


Figure 8: Thermal enhancement factor Vs Reynolds number for tube with PTT and UTT.

Thermal enhancement factor for PTT and UTT at different twist ratios y = 2.0, 4.4 and 6.0 calculated from equations (25) and (26) respectively for PTT and UTT are presented in figure 8.

At the same Reynolds number, the thermal enhancement factors for UTT are found to be greater than those for the PTT. The thermal enhancement factor for all twisted tapes tends to decrease with increasing Reynolds number. With the use of PTTs, thermal enhancement factors were in a range between, 1.12 - 1.2, 1.03 -1.10 and 1.0 - 1.06 respectively for the twist ratios y = 2.0, 4.4 and 6.0. On the other hand the use of UTTs offered thermal enhancement factors in a range between 1.19 - 1.28, 1.07 - 1.16 and 1.03 - 1.11 respectively for the twist ratios y = 2.0, 4.4 and 6.0. The above data indicates that the use of UTTs gave more efficient heat transfer enhancement than the application of PTT.

5. Conclusion

Experimental investigations of heat transfer, friction factor and thermal enhancement factor of circular tube fitted with PTT and UTT in turbulent regimes (2000<Re<12000) for twist ratios 2.0, 4.4 and 6.0 are described in the present report. The conclusions can be drawn as follows:

- The Nusselt number and friction factor values for the tube with UTT are noticeably higher than that of plain tube and also tube equipped with PTT.
- Over the range of Reynolds number considered average thermal enhancement factors in the tube equipped with PTT are found 1.15, 1.06, and 1.02 and tube equipped with UTT are 1.22, 1.10 and 1.06 respectively for twist ratios y =2.0, 4.4 and 6.0. The thermal enhancement factors for all the cases are more than unity indicates that the effect of heat transfer enhancement due to the enhancing tool is

more dominant than the effect of rising friction factor and vice versa

- The empirical correlations for the Nusselt number, friction factor and the thermal enhancement factor for PTT and UTT are developed and it was reasonably fitted with the experimental data.
- The UTT offered better heat transfer enhancement than that PTT therefore UTT can be used in place of PTT to reduce the size of heat exchanger.

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