

Experimental Analysis of Augmentation in Heat Transfer Coefficient Using Twisted Tape with Semi-Circular Cut Insert

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Abstract: Research is going on to investigate the level of heat transfer enhancement that can be achieved by forced convection in which water is flow inside horizontal pipe. The use of semi-circular cut inserts generate turbulence and superimposed vortex motion (swirl flow) causing a thinner boundary layer and consequently resulting in higher heat transfer coefficient with relatively low flow resistance. The study is focused on Reynolds number in turbulent flow. In the proposed work the influence of semi-circular cut twisted tape on pressure drop, Nusselt number (Nu) and friction factor (f) are experimentally determined. The twisted tapes with semi-circular cut are used as free generators. The test are conducting using the twisted tape with two different twist ratio for Reynolds number (Re) ranges between 4000 to 9000 under uniform heat flux conditions. The experiment using the semi-circular cut twisted tape and with plain twisted tape along with smooth tube to perform under similar operation test condition for comparison. The experimental results indicate that the tube with the various insert provide considerable improvement of the heat transfer rate over the smooth tube. The experimental results demonstrate that friction factor increases with decreasing twist ratio with Reynolds number respectively.

Keywords: Augmentation, Heat Transfer, Inserts, Turbulent Flow

1. Introduction

Heat transfer augmentations are frequently encountered in many industrial fields like power generation, air-conditioning and petrochemical engineering. Researchers throughout the world have made great efforts to develop various heat transfer augmentation techniques, classified distinctly into two categories: passive and active. The passive technique, due to no consumption of any additional energy, is generally more popular in practice. Utilizations of various kinds of tube inserts, such as twisted tape, helical screwed tape, coiled wire, louvered strip, porous media insert and conical rings are typical representatives of the passive technique. Tube inserts activate and intensify the swirl flow in the tube, reducing the thickness of the thermal boundary layer. However, the flow resistance is also increased as the tube inserts reduce the cross-sectional area of fluid pathway. Researching and optimizing the thermo-hydraulic performance of tube flow with tube inserts has gained continuing attentions in related scientific and industrial fields. The purpose is to seek for a maximized heat transfer augmentation with flow resistance being controlled as low as possible.

Tubes with twisted tape insert have been widely used as the continuous swirl flow devices for augmentation the heat transfer rate in heat exchanger tubes and applied in many engineering applications. Insertion of twisted tape in a tube provides a simple passive technique for enhancing the convective heat transfer by producing swirl into the bulk flow and by disrupting the boundary layer at the tube surface. It has been explained that such tapes induce turbulence and superimposed vortex motion (swirl flow) causing a thinner boundary layer and consequently

resulting in higher heat transfer coefficients. However, the increase in friction is seemed to be the penalty of the technique. Thus, tube with twisted tape insert is frequently used in heat exchanger systems because of its low cost, less maintenance and compact.

2. Literature Review

In the past decades, many researchers have investigated the effect of geometry of twisted -tapes on heat transfer and friction factor in a circular or rectangular tube in both experimental and numerical studies. Saha et al. [1] experimentally investigated friction and heat transfer characteristics of laminar swirl flow through a circular tube fitted with regularly spaced twisted-tape elements. It has been observed that pinching of tapes in place rather than connecting the tape elements with rods is better proposition from the thermo hydraulic performance point of view. Reducing the tape widths is worse than the full width tapes in the tape rod assembly. Smith et al. [2, 3] presented the experimental investigation on helical screw-tape with/without core-rod inserts and with alternate clockwise and counter-clockwise arrangement. The heat transfer rate obtained by using the tape without core -rod is found to be better than that by one with core-rod around 25–60% while the friction is around 50% lower. For alternate clockwise and counter-clockwise twisted tape inserts, it was observed that in the presence of novel alternate C–CC twisted-tapes, the periodic change of swirl direction and also the strong collision of the recombined streams behind the changing location, lead to superior chaotic mixing and to better heat transfer, compared with the typical twisted-tape. Sivashanmugam et al. [4, 5] experimentally studied on heat transfer characteristics of

turbulent flow through a circular tube fitted with regularly spaced helical screw-tape inserts and laminar flow through same tube with right and left helical screw-tape inserts. For regularly spaced helical screw-tape inserts, it was observed that heat transfer coefficient increases with the twist ratio and friction factor also increases with the twist ratio, whereas the heat transfer coefficient enhancement for right-left helical screw inserts is higher than that for straight helical twist for a given twist ratio. Zhang et al. [6] numerically studied on heat transfer and friction factor characteristics of a tube fitted with helical screw-tape without core-rod inserts. A three-dimensional turbulence analysis of heat transfer and fluid flow is performed by numerical simulation. Its results show that the average overall heat transfer coefficients in circular plain tubes are enhanced with helical screw-tape of different widths by as much as 212~351% at a constant tube-side temperature and the friction factor are enhanced by as much as 33% to 1020%. Physical quantity synergy analysis is performed to investigate the mechanism of heat transfer enhancement.

Ferroni et al. [7] reported the experimental data for investigated physically separated, multiple, short length twisted tapes. It was observed that yields pressure drops at least 50% lower than those due to the most well-known twisted-tape design, i.e., full-length tapes. Murugesan et al. [8] experimentally studied that the heat transfer and pressure drop characteristics in a circular tube fitted with and without V-cut twisted tape insert. It concluded that the V-cut twisted tape offered a higher heat transfer rate, friction factor and thermal performance factor compared to the plain twisted tape Thianpong et al. [9] has reported an experimental investigation on heat transfer and pressure drop characteristics of turbulent flow in a heating tube equipped with perforated twisted tapes with parallel wings (PTT) for Reynolds number between 5500 and 20500. Compared to the plain tube, the tubes with PTT and TT yielded heat transfer enhancement up to 208% and 190%, respectively. Naphon et al. [10, 11] experimentally investigated that heat transfer enhancement and pressure drop of the horizontal concentric tube with twisted wires brush inserts and effect of coil-wire inserts separately. It has been observed that the twisted wire brushes insert have a large effect on the enhancement of heat transfer and pressure drops also increase. Coil-wire insert has significant effect on the enhancement of heat transfer especially on laminar flow region. Promvong et al. [12] experimentally investigated the heat transfer augmentation in a helical-ribbed tube with double twisted tape inserts. To create vortex flows inside the tube the arrangement is used. It observed that the compound enhancement devices of the helical-ribbed tube and the twin twisted tapes show a considerable improvement of heat transfer rate and thermal performance relative to the smooth tube and the helical-ribbed tube acting alone, on the basis of twist ratios.

Smith et al. [13] experimentally investigated turbulent flow heat transfer and pressure loss in a double pipe heat exchanger with louvered strip inserts. Results confirmed that the use of louvered strips leads to a higher heat transfer rate over the plain tube. Increases in average

Nusselt number and friction loss for the inclined forward louvered strip were 284% and 413% while those for the backward louvered strip were 263% and 233% over the plain tube, respectively. Fan et al. [14] did parametric study on turbulent heat transfer and flow characteristics in a circular tube fitted with louvered strip inserts. Results show that the Nusselt number is augmented by 2.75–4.05 times ($Nu = 108.71-423.87$) as that of the smooth tube. Wenbin et al. [15] experimentally studied on heat transfer and friction factor characteristics of turbulent flow through a circular tube with small pipe inserts. It observed that the tube fitted with small pipe inserts can achieve a high heat transfer with a lower increase in friction factor. The friction factor is less than most of those recorded in other literature and is only 1.24–1.87times that of the smooth tube.

3. Experimental Details

The photograph of experimental setup is shown in Figure 1. It consists of a test section in which water flows from inlet to outlet. In the experimentation plain twisted tapes (PTT) with twist ratios 3.5 and 5.3 are used, which had shown in Figure 2(a). Semi-circular cut twisted tapes (STT) with twist ratios 3.5 and 5.3 have different semi-circular cut radii of 5mm and 10mm are shown in Figure 2(b). Semi-circular cuts are introduced in the PTT on both top and bottom alternately in the peripheral region with different dimensions of cut radius to improve the fluid mixing near the walls of the test section. Six thermocouples were soldered at six equally spaced points on test section to measure the surface temperature and two thermocouples were placed at inlet and outlet stream to measure stream inlet and outlet temperature. Heaters were wrapped around the test section in order to maintain constant heat flux. A U-tube manometer was used to measure the pressure drop across the tube. The inlet water flow rate was adjusted from 0.07 kg s^{-1} to 0.15 kg s^{-1} . As steady state conditions were reached, the inlet and outlet temperatures of water, surface temperatures of test section tube and the pressure drop were recorded for the case of smooth tube, PTT and STT. The details of experimental set up, all twisted tape inserts and operating conditions are summarized in Table 1.

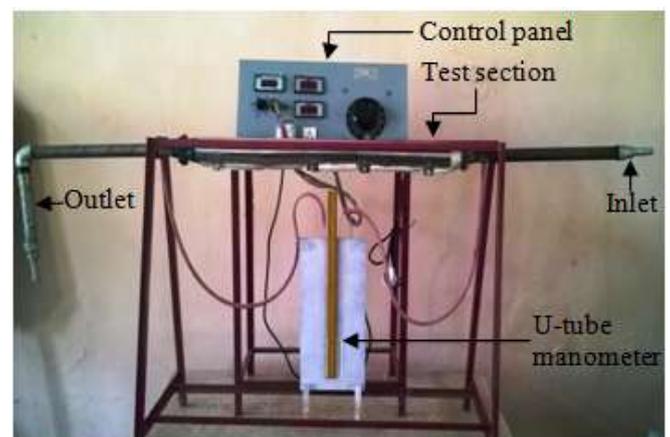


Figure 1: Photograph of the experimental set-up

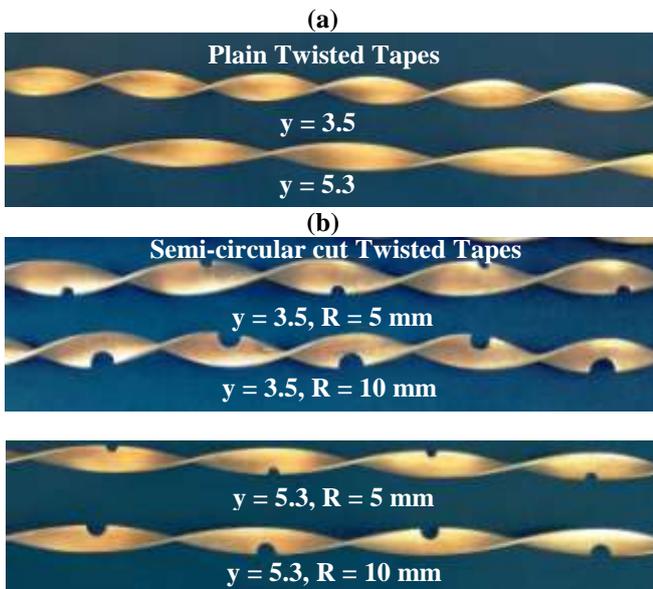


Figure 2: Geometries (a) plain twisted tapes (b) Semi-circular cut twisted tapes.

4. Heat Transfer Analysis

The data reduction of the measured results is summarized as follows:

Heat added to water was calculated by

$$Q = mc_p(T_{out} - T_{in}) \quad (1)$$

Heat transfer coefficient was calculated from,

$$h = \frac{q}{(T_{wi} - T_b)} \quad (2)$$

Heat flux,

$$q = \frac{Q}{A} \quad (3)$$

Where,

$$A = \pi d_i L \quad (4)$$

Bulk mean temperature

$$T_b = \frac{(T_{in} + T_{out})}{2} \quad (5)$$

Average outer wall temperature of tube

$$T_{wo} = \sum_{i=1}^6 T_{wo,i} / 6 \quad (6)$$

Average inner wall temperature of tube

$$T_{wi} = T_{wo} - Q \frac{\ln(\frac{d_o}{d_i})}{2\pi k_w L} \quad (7)$$

Experimentally Nusselt number was calculated from,

$$Nu = \frac{hd_i}{k} \quad (8)$$

Theoretically Nusselt number was calculated from Gnielinski 1976 correlation,

$$Nu_{th} = \frac{(\frac{f}{8})(Re-1000)Pr}{1+12.7(f/8)^{1/2}(Pr^{2/3}-1)} \quad (9)$$

Theoretically friction factor was calculated from Petukhov 1970,

$$f_{th} = (0.790 \ln Re - 1.64)^{-2} \quad (10)$$

$$Re = \frac{\rho U_m d_i}{\mu} \quad (11)$$

$$Pr = \frac{\mu C_p}{k} \quad (12)$$

Mean water velocity was obtained from,

$$U_m = \frac{m}{A_f} \quad (13)$$

Flow area was obtained from,

$$A_f = \frac{\pi}{4} d_i^2 \quad (14)$$

Experimentally friction factor was calculated from,

$$f_{exp} = \frac{\Delta p}{(L/d_i)(\rho U_m^2/2)} \quad (15)$$

Table 1: Technical details of experimental set up and test conditions.

(A) Experimental set-up	
(a) Test tube inner diameter, (di)	28 mm
(b) Test tube outer diameter, (do)	32 mm
(c) Test tube length	800 mm
(d) Material of test tube	Copper
(e) Insulation material	Glass wool
(f) Temperature measurements	K-type thermocouples
(g) Flow measurements	Beaker and stopwatch
(h) Pressure measurements	U-tube manometer
(i) Heater capacity	2 KW
(B) Twisted tape	
(a) Material	Aluminum
(b) Tape width, (W)	25 mm
(c) Tape thickness	3 mm
(d) Tape pitch length, (H)	100 and 150 mm
(e) Twist ratio, (y = H/di)	3.5 and 5.3
(f) Radius of cut, (R)	5 mm and 10 mm
(g) Configurations used	(i) Y=3.5 and R=5 (ii) Y=5.3 and R=5 (iii) Y=3.5 and R=10 (iv) Y=5.3 and R=10
(C) Test conditions	
(a) Reynolds number, (Re)	4000 to 9000
(b) Type of flow in tube	Turbulent
(c) Water inlet temperature	27 °C

5. Results and Discussion

As displayed in Figure 3(a-b), the data obtained from the present work are found to be in well agreement with the standard correlations of Gnielinski (1976) equation [9] for Nusselt number, Petukhov (1970) equation [10] for friction factor of smooth tube. The deviations of the present data from the above equations fall due to experimental errors.

From Figure 4, Nu increases with increase in Re for semi-circular cut twisted tape insert. Heat transfer rate increases with decrease in twist ratio (γ) and increase in cut radius (R). Maximum Nu is obtained for STT 3.5, R=10mm. And from Figure 5, it observed that friction factor (f) decreases with increase in Re for insert. At maximum Re, Friction factor decreases for STT 3.5, R=10mm compare with STT 3.5, R=5mm. Comparing with smooth tube and tube with PTT, it observed that f more but for increasing in cut radius, it reduces simultaneously.

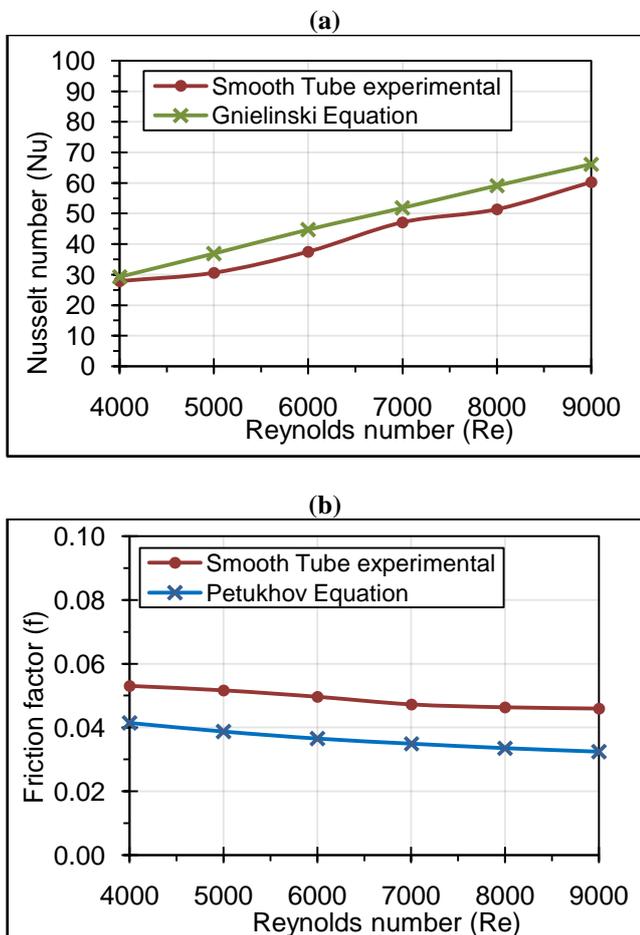


Figure 3: Validation of smooth tube experimental data: (a) Nusselt number and (b) friction factor

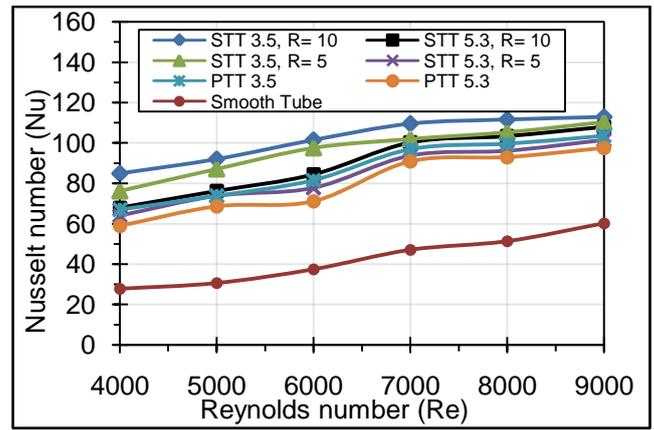


Figure 4: Nusselt number Vs Reynolds number.

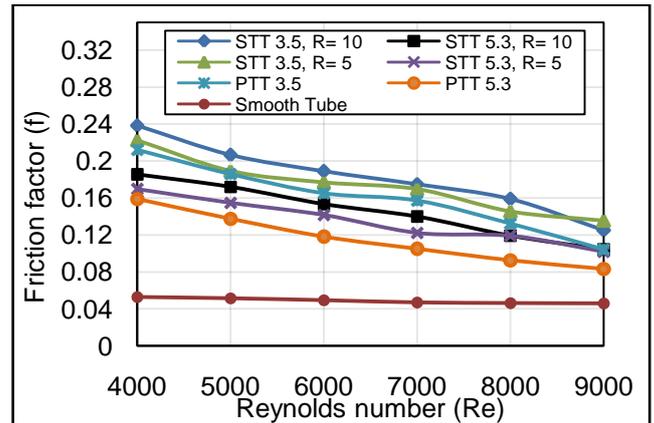


Figure 5: Friction factor Vs Reynolds number.

6. Conclusions

Experimental investigations of heat transfer and friction factor characteristics of circular tube fitted with plain twisted tape and semi-circular cut twisted tape for twist ratios 3.5 and 5.3 have been presented. The semi-circular cut tapes with different cut radii were also tested. The conclusion can be drawn as follows:

- Tubes fitted with semi-circular cut twisted tape inserts affect the heat transfer coefficient and friction factor. The geometry of the semi-circular cut twisted tape inserts makes it possible for water to flow easily through the pipe. This leads to a mixing of water with different temperatures and velocities. Mixing increases the temperature gradient of the thermal boundary layer and causes uniformity in fluid temperature. This enhances heat transfer.
- The semi-circular cut twisted tape offered a higher heat transfer rate and friction factor compared to the smooth tube and plain twisted tape.
- Nusselt number increases with decrease in twist ratio along with increase in cut radius. Friction factor decreases with increase of Reynolds number with increase in cut radius.

Nomenclature

A	Area of the heated region of tube, m ²
A _f	Flow area, m ²
C _p	Specific heat of water, J/kgK
d	Tube diameter, m
t	thickness of copper tube, m
H	Tape pitch length, m
h	Heat transfer coefficient, W/m ² K
k	Thermal conductivity of water, W/mK
k _w	Thermal conductivity of tube material, W/mK
L	Effective tube length, m
m	Mass flow rate of water, Kg/s
Q	Heat transfer rate, W
q	Heat flux, W/m ²
T _b	Bulk temperature, °C
T _{in}	Water inlet temperature, °C
T _{out}	Water outlet temperature, °C
T _{wi}	Tube inner surface temperature, °C
T _{wo}	Tube outer surface temperature, °C
U _m	Mean velocity, m/s
Nu	Nusselt number
f	Friction factor
Δp	Drop in pressure, N/m ²
g	Acceleration due to gravity, m/s ²
Pr	Prandtl number
Re	Reynolds number
ρ	Density of water, kg/m ³
μ	Dynamic viscosity of water, kg/m.s
Subscripts	
i	Inner
o	Outer
th	Theoretical
Abbreviations	
PTT	Plain twisted tape
STT	Semi-circular cut twisted tape

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