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Experimental Study of Heat Transfer for Wavy Twisted Tape Insert of Various Pitches Placed in a Circular Tube

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Abstract:
The present work shows the results obtained from experimental investigations of the augmentation of turbulent flow heat transfer in a horizontal tube by means of wavy twisted tape inserts with air as the working fluid. Experiments were carried out for plain tube with/without wavy twisted tape insert at constant wall heat flux and different mass flow rates. The wavy twisted tapes are of same wave-width, but three different twist ratios as 8.33, 9.79 & 10.42. The Reynolds number varied from 4000 to 9500. Both heat transfer coefficient and pressure drop are calculated and the results are compared with those of plain tube. It was found that the enhancement of heat transfer with wavy twisted tape inserts as compared to plain tube varied from 17% to 45 % for various inserts. Also the results are compared with the plane twisted tape insert.

Key words: *Wavy twisted tape insert, Heat transfer Enhancement, wave-width, twist ratio*

1. Introduction

Among many techniques (both passive and active) investigated for augmentation of heat transfer rates inside circular tubes, a wide range of inserts have been utilized, particularly when turbulent flow is considered. A lot of methods are applied to increase thermal performance of heat transfer devices such as treated surfaces, rough surfaces, swirling flow devices, coiled tubes, and surface tension devices [1]. Twisted tape swirl turbulator is one of the commonly used passive types for heat transfer augmentation due to their advantages of steady performance, simple configurations and ease of installation [2]. Sarma et al. [3] gave generalized correlations to predict friction factor and convective heat transfer coefficient in a tube fitted with twisted tapes for a wide range of Reynolds number and the Prandtl number. Configuration optimization of regularly spaced short-length twisted tapes in a circular tube for turbulent heat transfer was carried out by Wang et al. [4] by using computational fluid dynamics (CFDs) modeling. Eiamsa-ard et al. [5] presented experimental study of convective heat transfer in a circular tube with short-length twisted tapes inserted under uniform heat flux. Akhavan-Behabadi et al. [6] conducted some experiments to analyze effects of twisted tapes on heat transfer enhancement and pressure drop in horizontal evaporators. They selected R-134a as working fluid. Heat transfer and friction factor of CuO/water nanofluid and water were experimentally investigated in circular tube equipped with modified twisted tapes has alternate axis [7–9]. Eiamsa-ard et al. [10] performed experimental works on heat transfer and friction factor characteristics in a double pipe heat exchanger fitted with twisted tape elements.

2. Experimental Set-Up

The schematic diagram of experimental set-up is given in Figure 1.

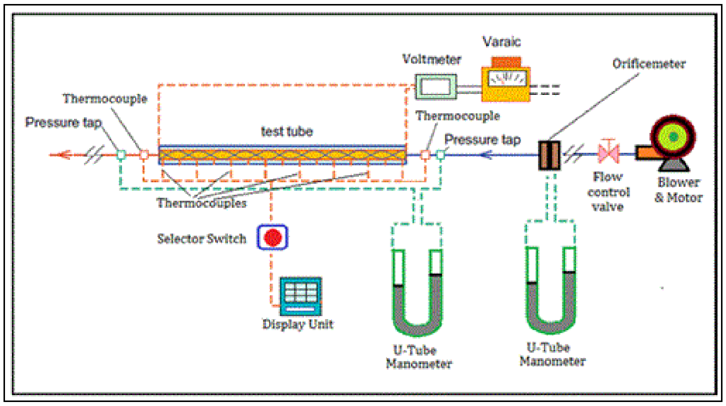


Figure 1:Experimental setup block diagram

The experimental facility includes a blower, an orificemeter to measure the volumetric flow rate, the heat transfer test tube (700 mm) . The MS test tube 26 mm inner diameter (D_1), 26.4 mm outer diameter (D_2), and 2 mm thickness (t). The wavy twisted tapes are tested in this experiment, with three different twist ratios as 8.33, 9.79 & 10.42 but have same wave-width as 13mm. They are fabricated from aluminium. Also one plane twisted tape made up of aluminium is tested. The schematic figure of the test tube with wavy twisted tape insert is given in Figure 2.

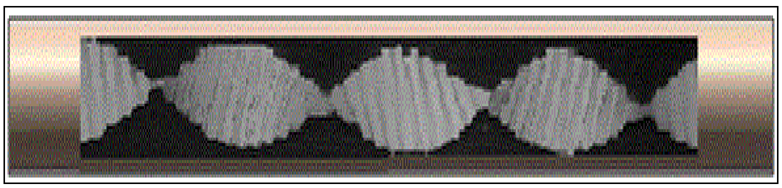


Figure 2: Schematic of test tube with wavy tape inserted

The wavy twisted tapes contained in the experimental study are shown in Figure 3.

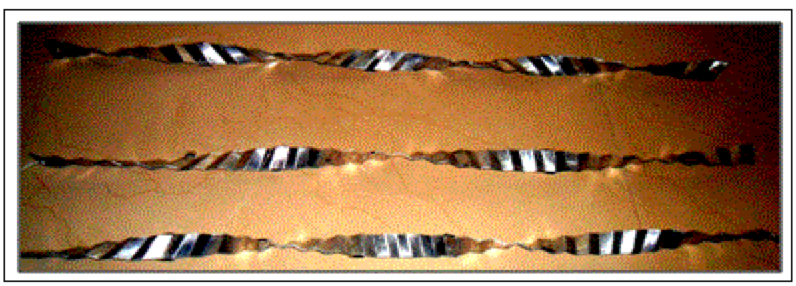


Figure 3: Actual view of wavy twisted tape insert

A 0.24 hp blower is used to force air through the test tube. Uniform heat flux is applied to external surface of the test tube by means of heating with electrical winding, whose output power is controlled by a variac transformer to supply constant heat flux along the entire section of the test tube. The outer surface of the test tube is well insulated with glass wool to reduce the convective heat loss to the surroundings. The external surface temperatures of the test tube wall are measured by 6 K-type thermocouples, which are placed on the outer wall of the test tube. Also, the inlet and outlet temperatures of the bulk air are measured by two K-type thermocouples at given points. An inclined manometer is used to measure pressure drop across the test tube. After air passes the test tube, it enters to the orificemeter for determining volumetric flow rate readings. For this purpose a separate U-tube manometer is placed across orificemeter. The volumetric flow rate of air supplied from the blower is controlled by varying control valve position. The experiments are conducted by varying the flow rate in terms of Reynolds numbers from 4181 to 9466 of the bulk air. The test tube is heated from the external surface during the experiments, and the data of temperatures, volumetric flow rate, pressure drop of the bulk air and electrical output are recorded after the system is approached to the steady state condition. The Nusselt number, Reynolds number, friction factor, heat transfer enhancement are calculated based on the average outer wall temperatures and the inlet and outlet air temperatures.

3. Data Collection and Analysis
The data reduction of the obtained results is summarized in the following procedures:

3.1. Heat Transfer Calculations

$T_s = (T_2+T_3+T_4+T_5+T_6+T_7)/6$ (1)

$T_b = (T_1+T_8)/2$ (2)

Equivalent height of air column, $h_{air} = (\rho_w * h_w)/\rho_a$ (3)

Discharge of air, $Q_a = Cd * Ao \sqrt{2 * g * h_{air}}$ (4)

Velocity of air flow, $V = Q_a / A$ (5)

Reynolds number, $Re = V D/\nu$ (6)

$Q = \dot{m} * C_p * (T_8 - T_1)$ (7)

$h = \frac{Q}{A\,(T_s - T_b)}$ (8)

$Nu = \frac{h D}{k}$ (9)

$f = \frac{\Delta P}{\frac{L}{D} * \frac{\rho a V^2}{2}}$ (10)

$\eta = \frac{(Nu_i/Nu)}{(f_i/f)^{0.333}}$ (11)

3.2. Validation Experiments of Plain Tube
In this study, experimental results of Nusselt number and friction factor for the plain tube are obtained and validated with equations of Dittus Boelter and Petukhov as given below;

$Nu_{th} = 0.023\, Re^{0.8}\ Pr^{0.4}$ (12)

$f_{th} = (1.82 * \log_{10} Re - 1.64)^{-2}$ (13)

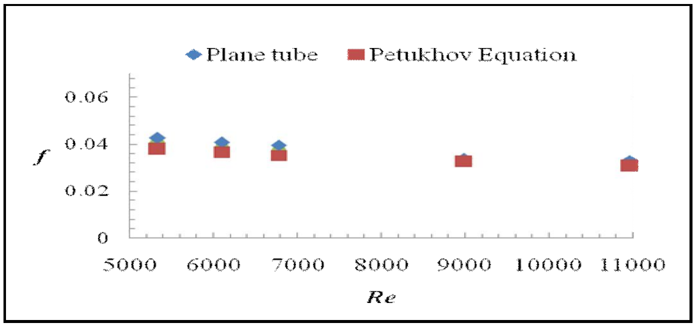


Figure 4: Validation results for friction factor

The comparisons of Nusselt number and friction factor for the present plain tube with existing correlations are shown in Figs. 4 and 5, respectively. These figures shows that validation experiments of heat transfer in terms of Nusselt number and friction factor for the plain tube are in good agreement with the results obtained from Dittus-Boelter and Petukhov equations. The results of present plain tube and previous equations are nearly the same. Thus, this accuracy provides reliable results for heat transfer and friction factor in a tube with twisted tape inserts in this present study.

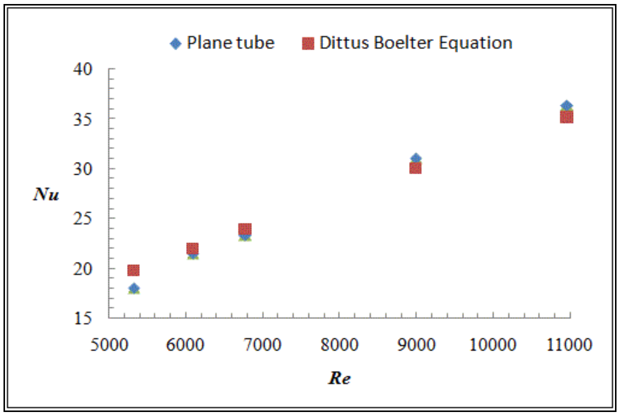
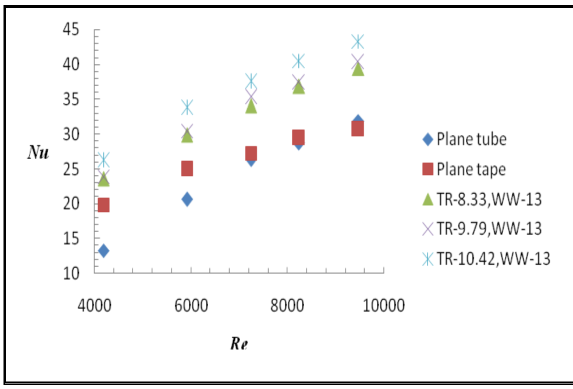


Figure 5: Validation results for Nusselt number



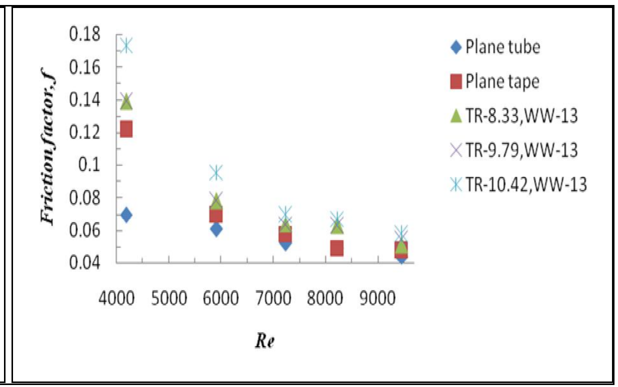


Figure 6: Variation of Nusselt number for different insert configurations
&
Figure 7: Variation of friction factor for different insert configurations

4. Results and Discussion

4.1. Heat Transfer and Overall Enhancement Characteristics

The variation of Nusselt number with Reynolds number for various wavy inserts is shown in Figure 6. Highest Nusselt number was obtained for tape with twist ratio of 10.42. The Nusselt number for wavy inserts varied from 35% to 77% compared to plain tube. This is due to strong turbulence intensity generated by corrugations on inserts leading to rapid mixing of the flow causing heat transfer enhancement. The variations of friction factor with Reynolds number for wavy tape inserts are presented in Figure 7. It is observed that the friction factor gradually reduced with rise in Reynolds number. It is observed to be maximum, for insert having twist ratio of 10.42. It is evident from Figures 6, 7 and 8 that when a wavy twisted tape is inserted into a plain tube there is a significant improvement in Nusselt number because of secondary flow, with greater enhancement being realized at lower Reynolds numbers and higher twist ratio keeping wave-width same.

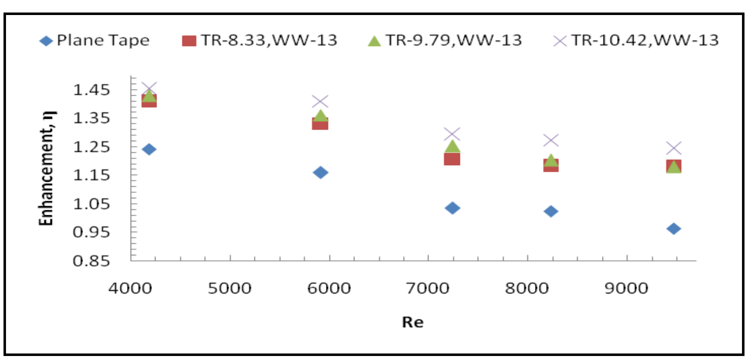


Figure 8: Variation of Enhancement for different insert configurations

This enhancement is mainly due to the centrifugal forces resulting from the spiral motion of the fluid and partly due to the tape acting as fin. It is observed that the rise in twist ratio causes increment in Nusselt numbers as well as rise in pressure drop. From Figure 6, the percentage rise in Nusselt numbers for wavy twisted tapes compared to plain tube are about 23-77%, 26-79% and 35-97% respectively for tape with twist ratio 8.33,9.79 & 10.42 respectively keeping wave-widths same as 13mm.The overall enhancement ratio is useful to evaluate the quality of heat transfer enhancement obtained over plain tube at constant pumping power. It is found to be more than unity for all the wavy twisted tape inserts used. Variations of overall enhancement ratio η against Reynolds number for various tapes are shown in figure 8. It is observed that overall enhancement tended to decrease gradually with the rise of Reynolds number for all twist ratios. The maximum value of overall enhancement is 1.45 for wavy twisted tape insert having wave-width of 13mm with twist ratio 10.42. It is seen in figure 8 that, for tapes of twist ratios 8.33, 9.79 & 10.42 curves are of decreasing order for a given wave-width in the range of Reynolds number from 4100 to 9400.

5. Conclusion

The study presents an experimental investigation of the potential of wavy twisted tape inserts to enhance the rate of heat transfer in a horizontal circular tube with inside diameter 26 mm with air as working fluid. The Reynolds number varied from 4100 to 9400. The effects of parameters such as modified twist ratio, Reynolds number on the heat transfer and overall enhancement ratio are studied.
The following conclusions can be drawn.

- The enhancement of heat transfer with wavy twisted tape inserts as compared to plain tube varied from 17 to 40% for twist ratio 8.33 and from 24 to 45% for twist ratio of 10.42. This enhancement is mainly due to the centrifugal forces resulting from the spiral motion of the fluid. Rise in twist ratio causes rise in Nusselt numbers as well as friction factors.
- The maximum friction factor rise was about 100% for twist ratio 8.33 and 150% for twist ratio 10.42 for wavy twisted tape inserts compared to plain tube.
- The overall enhancement for the tubes with wavy twisted tape inserts is 1.40 for twist ratio 8.33 and 1.45 for twist ratio 10.42 for wavy twisted tape insert.
- Thus the enhanced performance can be achieved using wavy twisted tapes as compared to plane twisted tape. Thus, from the considerations of enhanced heat transfer and savings in pumping power wavy-width tape inserts are seen to be attractive for enhancing turbulent flow heat transfer in a horizontal circular tube. Future work may be extended to:
- Change the tape material from Aluminium to Copper
- Compound enhancement techniques maybe applied i.e., the tape inserts can be coupled with coil wire inserts for better enhancement.

6. Nomenclature

A_0	area of orifice, (m^2)	A	test section inner tube area, ($\pi/4\ D^2$) m^2
Cp	specific heat of air, (J/kg K)	Q_a	air discharge through test section (m^3/sec)
D	Inner diameter of test section, (m)	H	pitch ,(mm)
w	width of wavy tape insert,(mm)	H/D	twist ratio
f_{th}	friction factor(theoretical) for plain tube	f	experimental friction factor (plain tube)
h	experimental convective heat transfer coefficient, (W/m^2K)	h_{air}	equivalent height of air column,(m)
f_i	friction factor obtained using tape inserts	h_w	manometer level difference,(m)
k	thermal conductivity, (W/mK)	L	length of test section, (m)
$\dot{m}$	mass flow rate of air, (Kg/sec)	p	pitch, (m)
Nu_i	Nusselt number (experimental) with tape inserts, (hD/k)	Nu	Nusselt number (experimental) for plain tube
Nu_{th}	Nusselt number for plain tube (theoretical)	Pr	Prandtl number
ΔP	pressure drop across the test section, (Pa)	Q	total heat transferred to air (W)
Re	Reynolds number, ($\rho VD/\mu$)	T_1, T_8	air temperature at inlet and outlet, (°K)
T_2, T_3, T_4, T_5	tube wall temperatures, (°K)		
T_s	average Surface temperature of the working fluid, (°K)	T_b	bulk temperature, (°K)
		V	air velocity through test section, (m/sec)
Greek symbols			
ν	Kinematic viscosity of air, (m^2/sec)	μ	dynamic viscosity, (kg/m.s)
η	Over all enhancement	ρ_w	density of water, (kg/m^3)
ρ_a	density of air (kg/m^3)		

Table 1

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