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Augmentation of heat transfer using longitudionally drilled turbulators

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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 longitudinally drilled twisted tape inserts with air as the working fluid. Experiments were carried out for plain tube with/without longitudinally drilled twisted tape insert at constant wall heat flux and different mass flow rates. The longitudinally drilled twisted tapes are of same pitch but three different gaps as 3,5 & 7 mm. 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 longitudinally drilled twisted tape inserts as compared to plain tube varied from 9% to 43 % for various inserts. Also the results are compared with the plane twisted tape insert.

Keywords— **longitudinally drilled twisted tape insert, twist ratio, Heat transfer Enhancement, Reynolds number**

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I. 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]. These type inserts generate swirling flow and cause improved fluid mixing between central region and the nearly wall region so, the heat transfer in tubes can be enhanced by fluid mixing. 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 Prandtl number. The contribution to thermal performance of the conventional, short-width and center-cleared twisted tapes was studied numerically [2]. 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 on 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 analyse 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. They made their analysis for both continuously placed twisted tape and twisted tape placed with various free spaced in circular tube. The heat transfer augmentation and pressure drop during condensation of HFC-134a in a horizontal tube fitted with twisted tapes were experimentally analysed [11]. Jaisankar et al. [12] experimentally examined the heat

transfer, friction factor and thermal performance caused by twisted tape for solar water heater. Twist ratio, space ratio, tape width, rod-diameter and phase angle effects on heat transfer and pressure drop were analysed experimentally in a circular tube fitted with regularly spaced twisted tape elements [13]. Naphon [14] also made experiments by using conventional twisted tape inserts in horizontal double pipe. Ferroni et al. [15] conducted some experiments in circular tube equipped with physically separated, multiple, short-length twisted tapes. Laminar convective heat transfer enhancement in twisted tape inserted tube was discussed experimentally by Sarma et al. [16]. In some studies, researchers focused the thermal effects of twisted tape inserts in modified tube instead of smooth tube, for example; Thianpong et al. [17] examined heat transfer enhancement in a dimpled tube with a twisted tape swirl generator inserted. They also presented the empirical correlations based on the experimental results of their study for prediction the Nusselt number and friction factor for Reynolds number from 12,000 to 44,000. Bharadwaj et al. [18] conducted experiments by using conventional type of twisted tapes to determine pressure drop and heat transfer characteristics of water in a 75-start spirally grooved tube. Some researchers [19] modified the conventional twisted tape geometries, for example; Murugesan et al. [20] used V-cut twisted tapes to analyse heat transfer and pressure drop in a circular tube.

II. EXPERIMENTAL SETUP

a The schematic diagram of experimental set-up is given in Fig.1. 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 (D1), 26.4 mm outer diameter (D2), and 2 mm thickness (t). The longitudinally drilled turbulators are tested in this experiment, with three different gaps between drilled holes as 3,5& 7 mm but have same twist ratio (y/D =8) & same horizontal distance as 30 mm. Also diameter of holes are same as 4 mm. They are fabricated from aluminium. Also one plane twisted tape made up of aluminium is tested. The schematic figure of the test tube with longitudinally drilled twisted tape turbulator is given in Fig.2. The longitudinally drilled twisted tape turbulator contained in the experimental study are shown in Fig. 3. 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 5000 to 9400 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.

III.DATA COLLECTION & ANALYSIS

The data reduction of the obtained results is summarized in the following procedures:

Figure 1: Experimental setup block diagram

A. *Heat Transfer Calculations*

Avg. Surface Temp., $T_s = (T_2+T_3+T_4+T_5+T_6+T_7)/6$ (1)

Avg. Temp of air, $T_b = (T_1+T_8)/2$ (2)

Air head, $ha = h_w \rho_w / \rho_a$ (3)

where,

ρ_w = Density of water = 1000 kg/m^3

Air volume flow rate, $Q_a = C_d * A_o \sqrt{(2 * g * h_{air})}$ (4)

where,

A_0= cross sectional area of orifice.

Mass flow rate, $m = Q_a * \rho_a$ (5)

Velocity of air, $V = Q_a/A$ (6)

where,

A= cross sectional area of pipe.

Heat carried out, $q = m * C_p * (T_8 - T_1)$ (7)

$h = Q/A(T_s - T_b)$ (8)

where,

h = heat transfer coefficient.

T_s = surface temperature

The Reynolds number for the fluid is defined by,

$Re = VD/\upsilon$ (9)

where,

V= velocity of the fluid.

ν = Kinematic viscosity of the fluid.

For internal flow conditions, if Reynolds number (Re) is greater than 4000 then the flow is said to be turbulent. After the flow is decided i.e. laminar or turbulent then the Nusselt number can be calculated. The theoretical Nusselt number is calculated below without considering friction which is theoretical Nusselt number and then calculated by considering friction which is experimental Nusselt number.

Nuth = 0.023*(Re)0.8*(Pr)0.4 (10)

This equation is called Dittus-Boelter equation.

$f_s = (1.82 \log_{10} Re^{-1.64})^{-2}$ (11)

Above equation is used to find friction factor and is called as Petukhov equation for smooth surface.

where,

f_s= Friction factor for smooth tube. Re= Reynolds number.

The actual pressure drop & friction factor is calculated with the help of tappings provided on both the ends of test pipe connected to U-tube manometer and the friction factor is calculated from the formula given below:

$$f = P/ (L/D)^2 (\rho_a V^2/2) \quad (12)$$

where,

P= pressure difference at both ends of test pipe.

L= length of test pipe.

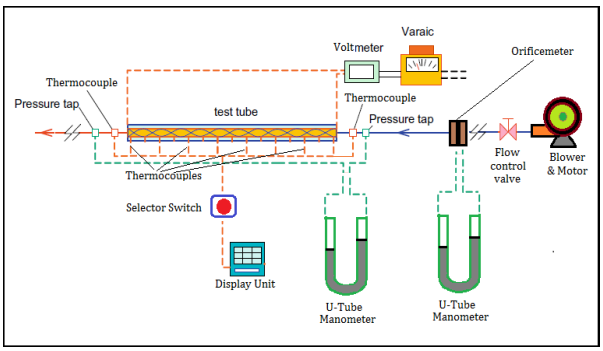
D= Inner diameter of pipe.

The experimental Nusselt number are calculated as given below:

$$Nu = hD/k \quad (13)$$

Nu= Nusselt number

h = heat transfer coefficient



k = thermal conductivity of fluid

D = diameter of test section

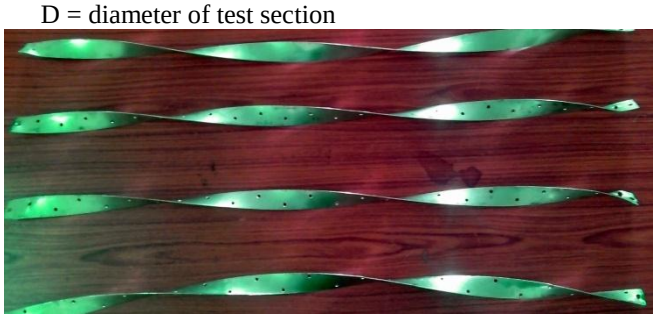


Figure 3: Actual view of twisted tape turbulators

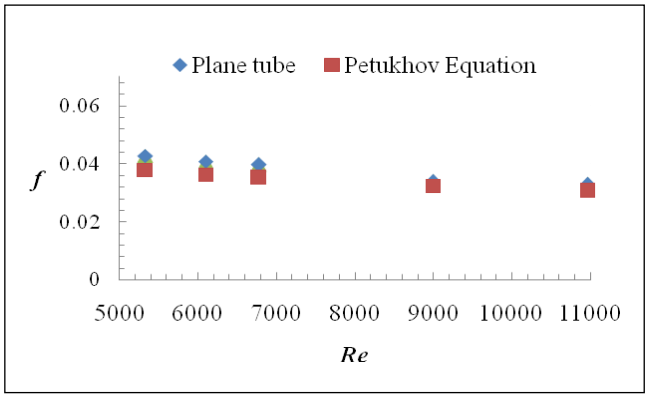


Figure 4: Validation results for friction factor

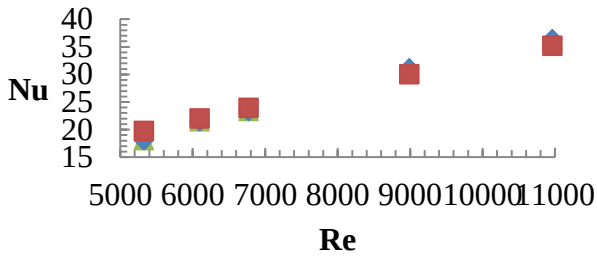


Figure 5: Validation results for Nusselt number

The overall enhancement efficiency is expressed as the ratio of the Nusselt number of an enhanced tube with corrugated twisted tape insert to that of a smooth tube, at a constant pumping power. This factor is introduced by Webb[19].

$$PEC = \eta = (Nu_{with} / Nu_{w/o}) / (f_{with} / f_{w/o})^{1/3} \quad (14)$$

B. *Validation experiments of plain tube*

In this study, experimental results of Nusselt number and friction factor for the plain tube are obtained and compared with equations of Dittus Boelter(10) and Petukhov (11) as mentioned above. The comparisons of friction factor and Nusselt number for the current plain tube with existing correlations are shown in Fig. 4 and 5, respectively. These figures show 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 current plain tube and preceding equations are nearly the same. Thus, this accuracy provides dependable results for heat transfer and friction factor in a tube with twisted tape inserts in this current study. The Reynolds number for validation test were ranged from 4500 to 10500 i.e. the range of Reynolds number used is for turbulent flow. Turbulent flow means Reynolds number greater than 4000. The results of the tests carried for performance inspection of present longitudinally drilled twisted tape turbulator are discussed next in results and discussion.

IV. RESULT & DISCUSSION

A. *Heat Transfer and Overall Enhancement Characteristics*

The variation of Nusselt number with Reynolds number for various longitudinally drilled twisted tape turbulator is shown in Figure 6. Highest Nusselt number was obtained for tape with gap 3mm. The Nusselt number for longitudinally drilled twisted tape turbulator varied from 18-62% 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 longitudinally drilled twisted tape

turbulator 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 gap 3 mm. It is evident from Figures 6, 7 and 8 that when a longitudinally drilled twisted tape turbulator 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 smaller wave-width for same twist ratio. 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 reduction in wave-width causes increment in Nusselt numbers as well as rise in pressure drop. From Figure 6, the percentage rise in Nusselt numbers for longitudinally drilled twisted tape turbulator compared to plain tube are about 24-62%, 20-57 % and 18-53% respectively for tape with wave-widths of 3, 5 and 7 mm respectively for twist ratio 8. 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 longitudinally drilled twisted tape turbulator 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 1.29 for longitudinally drilled twisted tape turbulator having gap of 7 mm with twist ratio equal to 8. It is seen in Figure 8 that, for tapes of of gaps 3, 5 and 7 mm curves are of decreasing order for a given pitch in the range of Reynolds number from 5900 to 9300.

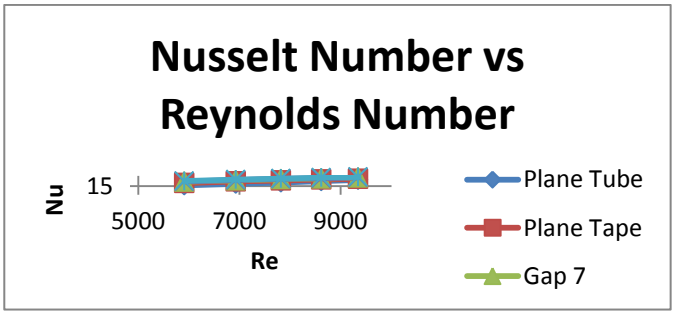


Figure 6: Variation of Nusselt number for different insert configurations

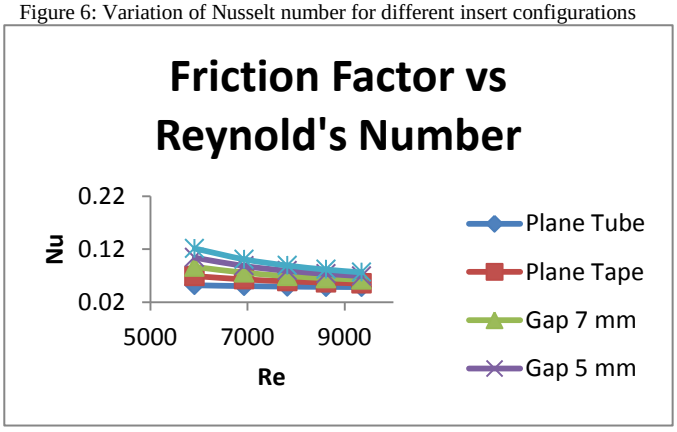


Fig.7 Variation of friction factor for different insert configurations

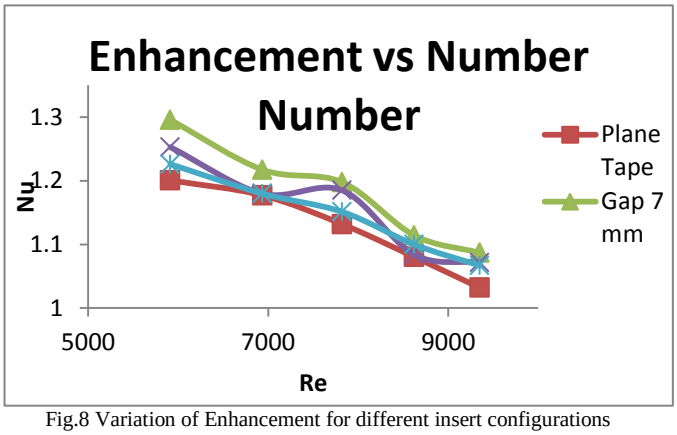


Fig.8 Variation of Enhancement for different insert configurations

V.CONCLUSION

potential of longitudinally drilled twisted tape turbulators 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 wave-width, Reynolds number on the heat transfer and overall enhancement ratio are studied.

The following conclusions can be drawn:

- The enhancement of heat transfer with longitudinally drilled twisted tape turbulators as compared to plain tube varied from 6 to 29% for gaps of 3 mm to 7 mm. This enhancement is mainly due to the centrifugal forces resulting from the spiral motion of the fluid.
- Reduction in tape width causes rise in Nusselt numbers as well as friction factors. The maximum friction factor rise was about 57% for 3mm compared to plain tube.
- The overall enhancement for the tubes with longitudinally drilled twisted tape turbulators is 1.29 for gap of 7 mm.
- Thus the enhanced performance can be achieved using longitudinally drilled twisted tape turbulators as compared to plane twisted tape. Thus, from the considerations of enhanced heat transfer and savings in pumping power longitudinally drilled twisted tape turbulators are seen to be attractive for enhancing turbulent flow heat transfer in a horizontal circular tube.

Future work may be extended to:

A. Change the tape material from Aluminium to Copper

B. Compound enhancement techniques maybe applied i.e., the tape inserts can be coupled with coil wire inserts for betterenhancement.

NOMENCLATURE

A_0 area of orifice, (m^2)
A test section inner tube area, ($\pi/4\ D^2$) (m^2)
C_p 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 longitudinally drilled twisted turbulator, (mm)
H/D twist ratio
f_{th} friction factor(theoretical) for plain tube
f friction factor(experimental) for plain tube
f_i friction factor obtained using tape inserts
h experimental convective heat transfer coefficient, (W/m^2K)
h_w manometer level difference,(m)
h_{air} equivalent height of air column, (m)
k thermal conductivity, (W/mK)
L length of test section, (m)
$\dot{m}$ mass flow rate of air, (Kg/sec)
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
p pitch, (m)
ΔP pressure drop across the test section, (Pa)
Q total heat transferred to air (W)
Re Reynolds number, (ρ V D/μ)
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)

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