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FORCED CONVECTION HEAT TRANSFER PERFORMANCE OF POROUS TWISTED TAPE INSERT

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ABSTRACT

Heat transfer performance of porous twisted tape insert in a circular tube was experimentally investigated. Tube wall temperatures and pressure drops along the axial distance of the test section at steady state condition were measured for different flows having Reynolds number ranging from 1.4 x 10^4 to 5.2 x 10^4 for both the plain and the tube with porous twisted tape insert. Heat transfer coefficient, friction factor, and pumping power were calculated from the measured data. Heat transfer and fluid flow characteristics of the porous twisted tape inserted tube were explained from the measured and calculated values. Performance of the porous twisted tape inserted tube was also evaluated. The results showed for porous twisted tape inserted tube, the average heat transfer coefficient was 2.60 times higher, the heat flux was 1.55 times higher, the friction factor was 2.25 times higher and the pumping power was 2.0 times higher than those of plain tube values for similar flow conditions.

Keywords: Forced convection; Friction factor; Heat transfer coefficient; Porous twisted tape; Pumping Power.

NOMENCLATURE

A_{xp}	Cross- sectional area of the porous twisted tape inserted tube, m^2
T_{wx}	Wall temperature at any axial location x, ^{0}C
A_s	Cross sectional area of the plain tube, m^2
V	Mean velocity of air in the test section, m/s
C_p	Specific heat of air at constant pressure, J/(kg. ^{0}C)
V_i	Mean velocity of air at the inlet section, m/s
D_i	Inside diameter of tube, m
W_p	Wetted perimeter of the inserted tube, m
d_s	Pore diameter of the twisted tape, m
W_s	Wetted perimeter of the plain tube, m
f	Friction factor
x	Axial distance, m
f_i	Fanning friction factor
f_p	Fanning friction factor of the porous twisted tape inserted tube
μ	Dynamic viscosity, kg/(m.s)
f_s	Fanning friction factor of the plain tube
ρ	Density of air, kg/m^3
h	Average surface heat transfer co-efficient, W/(m^2.^{0}C)
ρ_b	Density at bulk mean temperature, kg/m^3
h_x	Surface local heat transfer co-efficient, W/(m^2.^{0}C)
k	Thermal conductivity of air, W/(m.^{0}C)
b	bulk
L	Length of the tube, m
i	inlet
m	Mass flow rate of the air, kg/s
o	outlet
P	Performance parameter
p	porous
ΔP	Pressure drop along the axial length, N/m^2
s	plain
P_m	Pumping power, W
w	wall
Q	Total heat transfer rate, W
x	local
q	Heat flux, W/m^2

q_p	Heat flux of the inserted tube, W/m^2
Nu	Nusselt number, = hD_i/k
q_s	Heat flux of the plain tube, W/m^2
Nu_p	Nusselt number of the porous twisted tape inserted tube
R	Variable of Uncertainty
Nu_s	Nusselt number of the plain tube
T	Room temperature, ^{0}C
Nu_{exp}	Experimental Nusselt number
T_b	Bulk fluid temperature, ^{0}C
Nu_{pre}	Predicted Nusselt number
T_{bx}	Bulk fluid temperature at any axial location x, ^{0}C
Pr	Prandtl number, = $\mu C_p/k$
T_i	Inlet temperature of air, ^{0}C
Re	Reynolds number, = $\rho VD/\mu$
T_o	Outlet temperature of air, ^{0}C
Re_p	Equivalent Reynolds number of the porous twisted tape inserted tube
T_w	Wall temperature of the tube, ^{0}C
Re_s	Equivalent Reynolds number of the plain tube

1. INTRODUCTION

Heat exchangers with the convective heat transfer of fluid inside the tubes are frequently used in many engineering applications. Enhancement of heat transfer intensity in all types of thermo technical apparatus is of great significance for industry. Besides the savings of primary energy, it also leads to a reduction in size and weight. Up to the present, several heat transfer enhancement techniques have been developed. Porous twisted tape is one of the most important members of enhancement techniques, which employed extensively in heat exchangers.

Ray and Date (2003) investigated experimentally correlations of heat transfer and flow frictions in a square duct with twisted-tape insert. Agarwal and Rao (1996) experimentally determined isothermal and non-isothermal friction factors and mean Nusselt numbers for uniform wall temperature during heating and cooling of Servotherm oil (Pr = 195 ~ 375) in a circular tube (Re = 70 ~ 4000) with twisted tape inserts (twist ratio, y = 2.41 ~ 4.84). Hseih *et al.* (2003) studied experimentally turbulent heat transfer and flow characteristics in a horizontal circular tube with strip-type inserts (longitudinal, LS and cross, CS inserts) using air as working fluid. Saha *et al.* (2001) experimentally studied the friction and heat transfer characteristics of laminar swirl flow through a circular tube fitted with regularly spaced twisted-tape elements. Sarkar *et al.* (2005) experimentally studied the heat transfer in turbulent flow through tube with wire-coil inserts. Ahmed *et al.* (2005) studied the heat transfer for turbulent flow through a circular tube with twisted tape inserts having different twist ratios. Eiasma–ard *et al.* (2006) experimentally investigated the heat transfer and friction factor characteristics in a double pipe heat exchanger fitted with full length tape, spaced twisted tape elements. Forced convection heat transfer performance of an internally finned tube was studied by Ashraful Islam and Mozumder (Islam and Mozumder, 2009). Sarada *et al.* (2009) experimental investigated the augmentation of turbulent flow heat transfer in a horizontal tube by means of mesh inserts with air as the working fluid. Naphon (2006) investigated heat transfer and the pressure drop characteristics in a horizontal double micro-fin tube with twisted tape inserts. Yadav (Yadav, 2008) investigated heat transfer and the pressure drop characteristics in a double pipe heat exchanger with full length twisted tape inserts. Experimental investigations of heat transfer and friction factor characteristics of circular tube fitted with full length twisted tape with trapezoidal-cut were studied for the Reynolds number range of 2000-12000 by Murugesan *et al.* (2009). Yadav (2009) investigated the effect of half length twisted-tape turbulators on heat transfer and pressure drop characteristics inside a double pipe u-bend heat exchanger.

So far very few research works have been reported in literature on heat transfer in turbulent flow through a tube with porous twisted tape insert. The porous twisted tape insert might be a good topic for heat transfer enhancement in a heat exchanger. The purposes of this research were to study heat transfer performance of porous twisted tape insert, to analyze the heat transfer and friction factor characteristics in a tube with porous twisted tape insert and to compare the results with the data of plain tube for augmentation of heat transfer.

2. DATA REDUCTION EQUATIONS

Mass flow rate was calculated by

$$m = \rho A_s V_i \qquad (1)$$

Where, V_i = Mean inlet velocity, (m/sec), A_s = Cross sectional area of tube, (m^2) and ρ = Density of air at room temperature, (Kg/m^3)

Total heat transfer rate was performed as

$$Q = mC_p(T_o - T_i) \quad (2)$$

Where, T_i = Inlet temperature of air (^{0}C), T_o= Outlet temperature of air (^{0}C)

The heat flux rate was calculated by

$$q = \frac{Q}{W \times L} \quad (3)$$

Where, Q = Heat transfer rate, (W), W_s = Wetted perimeter of the tube, (m), L = Length of the tube, (m)
The local bulk fluid temperature at any location, x was performed as

$$T_b(x) = T_i + \frac{q\,WX}{mC_p} \quad (4)$$

Local convective heat transfer co-efficient at location, x was obtained from

$$h_x = \frac{q}{T_{wx} - T_{bx}} \quad (5)$$

Where, T_{wx} = Wall Temperature at any axial location x, (^{0}C), T_{bx} = Bulk Temperature at any axial location x, (^{0}C).

Local Nusselt number at location, x was calculated from

$$Nu_x = \frac{h_x D_i}{k} \quad (6)$$

At constant pumping power, the performance parameter was obtained from

$$P = \frac{Nu_p}{Nu_s} \quad (7)$$

Where, Nu_p = Nusselt number of the tube with porous twisted tape insert, Nu_s = Nusselt number of the plain tube
The equivalent Reynolds number at constant pumping power was performed as

$$\mathrm{Re}_p = \mathrm{Re}_s \cdot \left(\frac{f_s}{f_p}\right)^{1/3} \quad (8)$$

Re_s is the Reynolds number of the plain tube, f_s = Fanning friction factor of the plain tube, f_p = Fanning friction factor of the porous twisted tape inserted tube
Fanning (Local) friction factor based on inside diameter was obtained by

$$f_i = \frac{\left(\frac{-\Delta P}{x}\right) D_i}{2\rho_b V^2} \quad (9)$$

Where, D_i = Inner diameter of the tube, (m)
Pumping power was calculated from

$$P_m = \frac{\Delta P \times m}{\rho_b} \quad (10)$$

Where, P_m = Pumping power, (Watt)

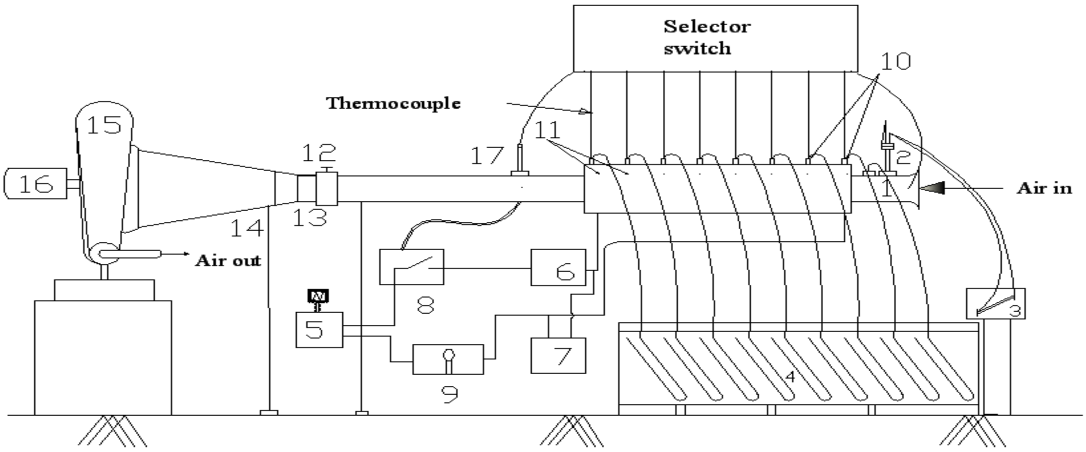
3. EXPERIMENTAL WORKS

3.1 Experimental Setup

The experimental facility consisted of an inlet section, a test section, an air supply system (Electric blower) and a heating arrangement. Fig. 1 shows the photograph of the experimental facility.

Fig. 1 Photograph of the experimental facility

Test section: The plain tube was made of brass and two halves of plain tube were clamped together by flanges at the ends having 70 mm inside diameter and 1500 mm length. The twist ratio of the porous twisted tape insert was 4.55. The pore diameter of the twisted tape was 5 mm. The schematic diagram of the experimental facility is shown in Fig. 2.



1. Inlet section
2. Traversing pitot tube
3. Inclined tube manometer
4. U-tube manometer
5. Variable voltage Transformer
6. Ammeter
7. Voltmeter
8. Temperature controller
9. Heater on off lamp
10. Pressure tappings
11. Thermocouples
12. Flow control valve
13. Flexible pipe
14. Diffuser
15. Blower
16. Motor
17. Traversing thermocouple

Fig. 2 Schematic diagram of the experimental facility

The mild steel porous twisted tape was made so that the tape might be fitted into the test section. The photograph of the porous twisted tape insert is shown in Fig. 3.

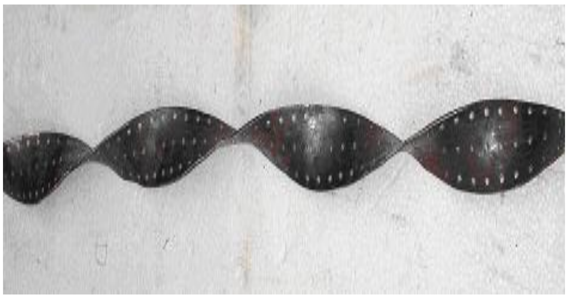


Fig. 3 Photograph of the porous twisted tape insert

The porous twisted tape was inserted into the plain tube to perform the experiment. The two halves of the plain tube were clamped together after inserting the tape axially in one half of the tube. A cross-sectional view of the tube with porous twisted tape insert is shown in Fig. 4.

In order to prevent leakage the metal putty was pressed into the joint of the tube. Then the plain tube in the test section was covered with mica sheet to isolate electrically. A layer of glass fiber was put on the mica sheet. Nichrome wire (of resistance 1.2 Ω/m) was used

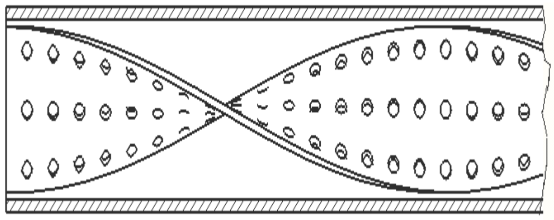


Fig. 4 Cross-section of the test section with porous twisted tape insert

as an electric heater and was spirally wounded uniformly with spacing of 16 mm around the tube. Then mica sheet, glass fiber tape, heat insulating tape, and asbestos tape were sequentially put over the heater coil of nichrome wire. These insulating materials helped to protect the radial heat losses. The test section was installed in the experimental facility with the help of the bolted flanges with gaskets of asbestos (thickness 3.5 mm) to prevent the heat flow in the longitudinal direction and to prevent the leakage of air.

Air supply system: The air was induced by an electrically driven centrifugal blower which was fitted down stream of the test section so that any disturbance produced by the blower would not affect the flow in the test section. The air entered through the inlet section containing traversing pitot tube and it passed successively through the heated test section and diffuser system. The gate valve (Butterfly type) was fitted ahead of the flexible duct to control the flow rate of air through the tube.

Heating arrangement: An electric heater (made of nichrome wire) was used to heat the test section at constant heat flux. The electric power was supplied to heater by a regulated alternating current source with a 5 kVA variable voltage transformer through a magnetic contactor and a temperature controller. The temperature controller was used to sense the outlet air temperature by providing signal for switching the heater off or on automatically. It was used to control the outlet air temperature at a specified limit. This was also used to protect excessive heating of the experimental facility. The electric heat input to nichrome wire was kept constant for all the experiments.

3.2 Measurement Systems

Three measurement systems namely flow measuring system, pressure measuring system and temperature measuring system were obtained to perform the experiment.

3.3 Experimental Procedure

At first the electric heater was switched on and allowed to heat for a few minutes and then the pump was switched on. The electric power was adjusted with the help of a regulatory transformer or variac. The flow rate of air through the test section was set to a desired value and kept constant with the help of the flow control valve. First the variations in wall temperatures at all locations were observed until constant values were attained at all the eight locations. Then the outlet bulk air temperature was monitored. The steady state condition was attained when the outlet air temperature did not fluctuate over some duration of time. At the steady state condition thermocouple readings were monitored with the help of the selector switch and then recorded. The manometer readings were observed and taken from the inclined tube manometers. The air flow rate was changed with the help of the flow control valve after each experimental run hence, changed the Reynolds number. Electrical power supply was kept constant for every change of the air flow rate. Different data were taken in the similar way in each experimental run at steady state condition.

4. RESULTS AND DISCUSSIONS

At first the temperature distributions along the tube wall for different flow conditions were presented which was followed by explanations of heat transfer characteristics and fluid flow characteristics. Finally, heat transfer performance was evaluated on the basis of constant pumping power.

4.1 Temperature Distribution

During each experimental run the wall temperatures along the axial locations of the test section were measured. Fig. 5 shows the variation of wall temperatures along the axial location of the test section of the plain tube at different Reynolds numbers and at constant heat flux condition. Fig. 5 indicates that at higher Reynolds number the wall temperature was lower.The axial variation of wall temperatures for the tube with porous twisted tape insert at different Reynolds numbers and at constant heat flux condition is shown in Fig. 6.

From the Fig. 6, it can be observe that the wall temperature increased along the axial distance of the test section for a given Reynolds number and reached its maximum at x/L= 0.577. Then the tube wall temperature dropped slightly at the downstream due to end effect. It might be noted that the wall temperature was lower at the entry and the exit partly because of the conduction losses (end effect). The end effect might be for the physical contact between the test section and the downstream unheated tube. Fig. 6 also reveals that the wall temperature decreased with the increase in Reynolds number for any axial position of the test section. Higher Reynolds number indicated a consequence of higher flow rate and ultimately caused better mixing of hot and cold fluid during its movement throughout the test section.

Fig. 7 shows the comparison of wall temperatures for both the plain tube and the tube with insert along the axial distance of the test section at different Reynolds numbers. It was shown from the Fig. 7 that the wall temperature of the tube with insert at any axial location was lower than that of the plain tube for a given Reynolds number. As the plain tube had lower wetted perimeter and less contact area with the working fluid compared to the inserted tube, its ability to transfer heat was low.

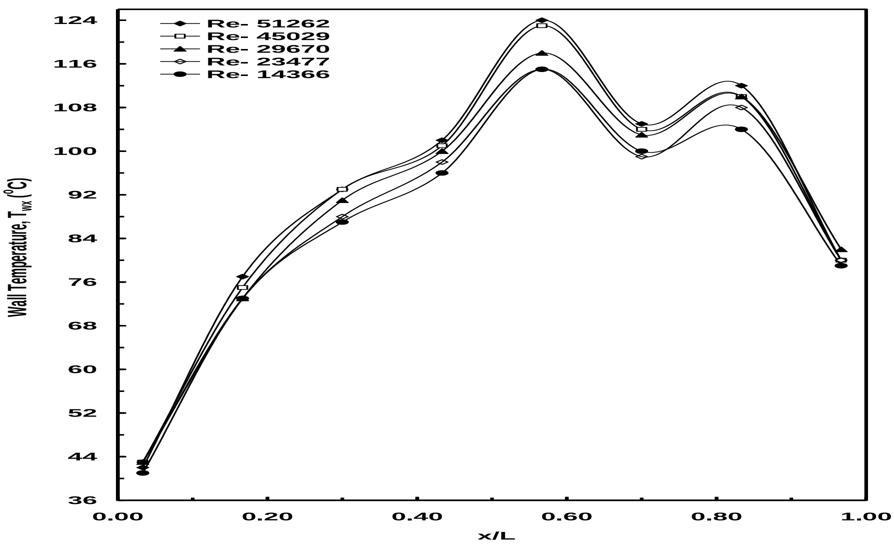


Fig. 5 Variation of wall temperatures along the axial location of the plain tube

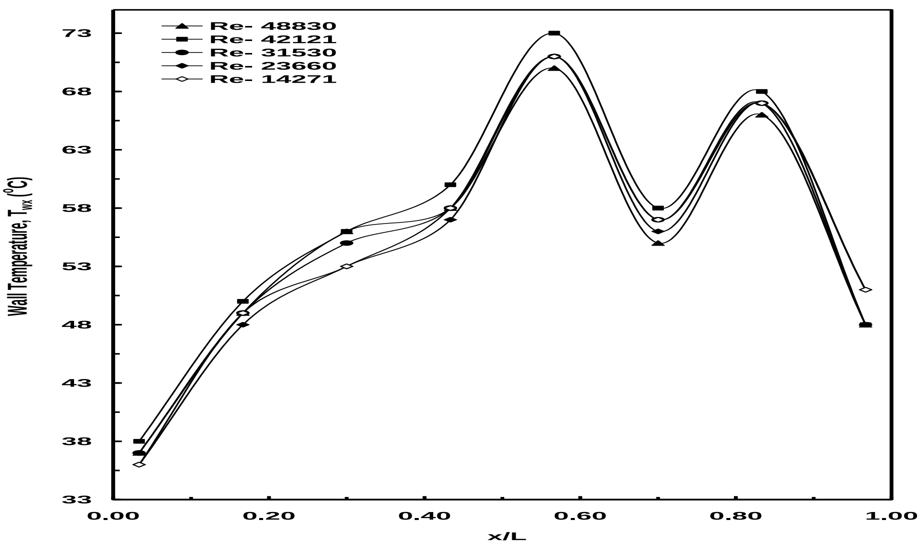


Fig. 6 Axial variation of wall temperatures of the inserted tube

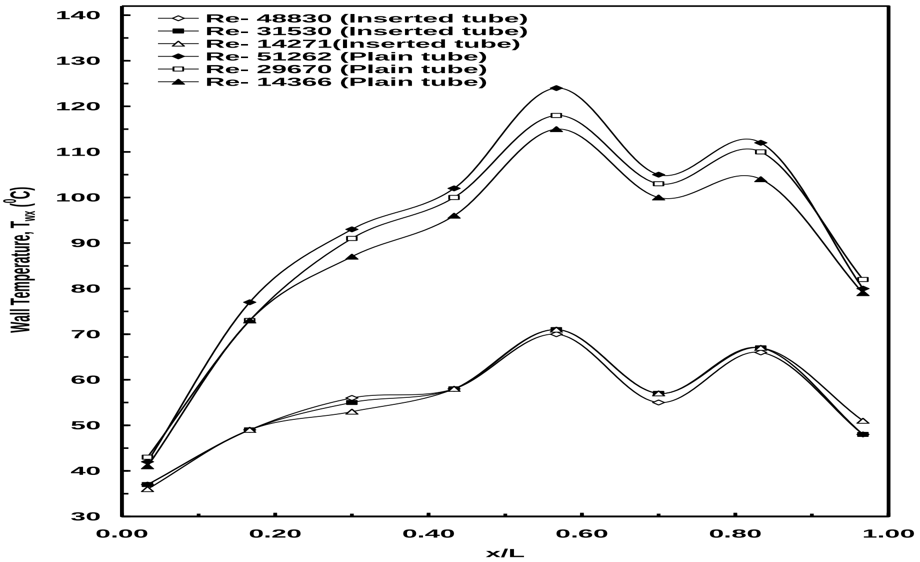


Fig. 7 Comparison of wall temperatures for both the plain tube and the inserted tube

The variation of bulk fluid temperatures along the axial position of the test section for both the plain tube and the tube with insert is represented in Fig. 8. The calculated bulk fluid temperatures based on constant heat flux condition along the axial location of the tube with and without porous twisted tape insert at different Reynolds numbers increased linearly. As air passed through the test section of heated tube along the length and took away heat from the tube wall.

At lower Reynolds number, the bulk fluid temperature was higher and at higher Reynolds number, the bulk fluid temperature was lower. The reason was that at lower Reynolds number there was less velocity of air which provided much time for sufficient heating of air in the test section, but at higher Reynolds number faster moving of air got insufficient time for being heated of air in the test section. It could be noted that total heat taken by turbulent flow of higher Reynolds number was obviously higher even though the bulk temperature was lower. It was clear from the Fig. 8 that the bulk fluid temperature of the plain tube was lower than that of the inserted tube.

Fig. 9 shows the axial variation in dimensionless temperature differences along the distance for both the plain tube and the tube with insert. From the Fig. 9, it was clear that the plain tube had the higher values of dimensionless temperature difference and the tube with insert had the lower values of dimensionless temperature difference. Lower values of dimensionless temperature difference indicated more heat was taken away from the tube surface.

4.2 Heat Transfer Characteristics

The comparison of local heat transfer coefficient along the axial distance of the test section for both the plain tube and the tube with insert at different Reynolds numbers is shown in Fig. 10. Fig. 10 indicated that the local heat transfer coefficient was higher at the entrance region of the test section as thermal boundary layer was developed with the entrance section at the leading edge. Then it gradually decreased up to a certain location. There was remarkable increased in the heat transfer coefficient at the exit of the test section for both the plain tube and the tube with insert because of lower radial temperature gradient was occurred due to end effect. It was clear from the Fig. 10 that at higher Reynolds number the local heat transfer coefficient was higher. This was expected because a higher flow rate resulted in increment of local heat transfer from the tube surface. It was shown from the Fig. 10 that the local heat transfer coefficient was lower in the plain tube than that of the tube with insert for any axial location of the test section at a particular Reynolds number.

As the temperature difference was lower in the inserted tube with compared to the plain tube, the heat transfer coefficient was higher in the inserted tube than that of the plain tube.

Fig. 11 exhibits the variation of average heat transfer coefficient with Reynolds number for both the plain tube and the tube with insert. The average heat transfer coefficient increased with the increase in Reynolds number for both the plain tube as well as for the tube with insert. It was illustrated from the Fig. 11 that at higher Reynolds number more heat was taken away from the tube surface as of intensive mixing of the fluid occurred in the tube hence, the average heat transfer coefficient was increased. However, as the flow rate was increased, a secondary flow and hence some turbulence was induced in the flowing fluid for the present of insert which increased the average heat transfer coefficient.

From the Fig. 11, it could be shown that the average heat transfer coefficient of the tube with insert was 2.25 to 2.60 times higher than that of the plain tube. The average heat transfer coefficient of the tube with wire-coil inserts (Sarkar *et al.*, 2005) varied from 1.2 to 2.0 folds with compared to the plain tube at the comparable Reynolds number.

The comparison of heat flux with Reynolds number for both the plain tube and the tube with porous twisted tape insert is represented in Fig. 12. It was shown from the Fig. 12 that the heat flux increased with the increase in Reynolds number.

The heat flux was higher of the tube with insert than that of the plain tube. The higher heat transfer rate was occured in the inner surface of the inserted tube than that of the plain tube.

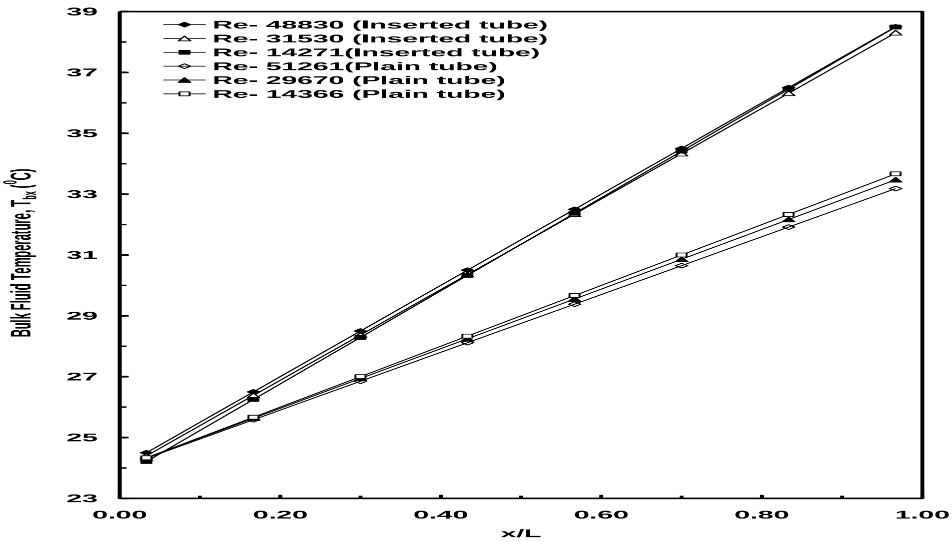


Fig. 8 Variation of bulk fluid temperatures for both the plain tube and the inserted tube

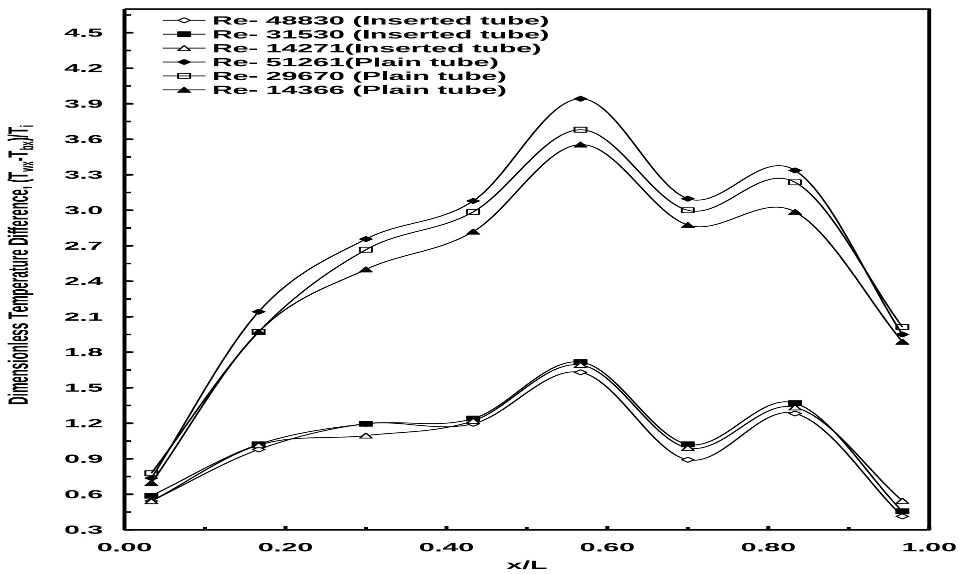


Fig. 9 Axial variation in dimensionless temperature differences for both the plain tube and the inserted tube

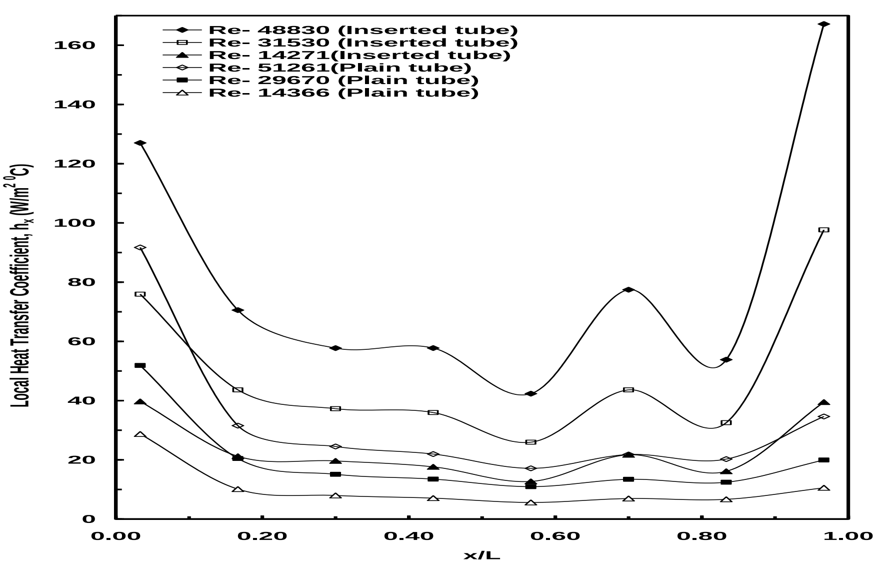


Fig. 10 Comparison of local heat transfer coefficient for both the plain tube and the inserted tube

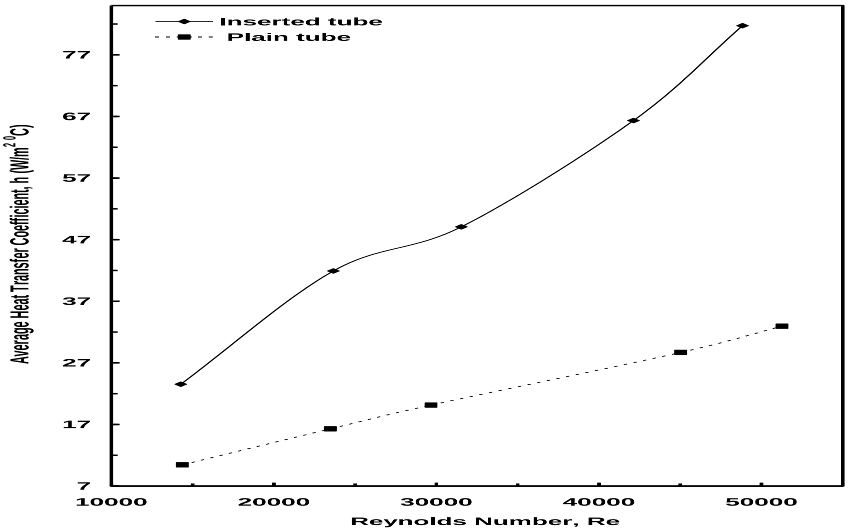


Fig. 11 Variation of average heat transfer coefficient with Reynolds number for both the plain tube and the inserted tube.

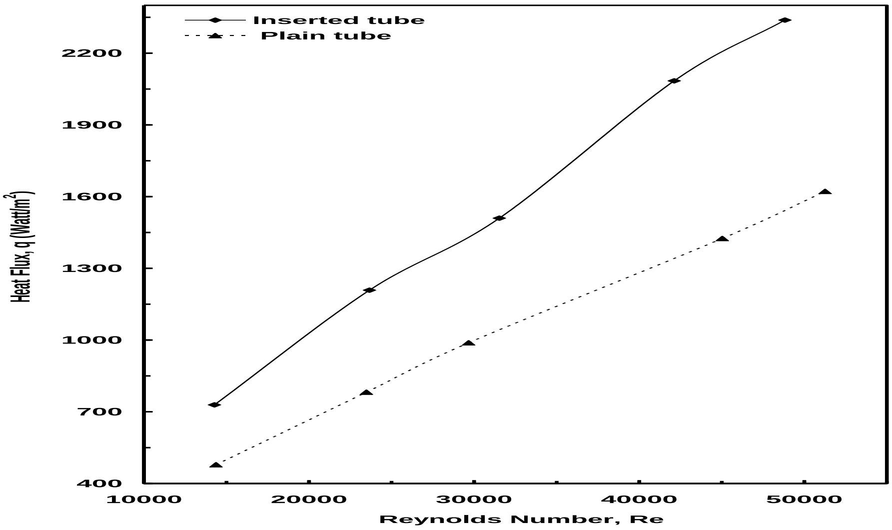


Fig. 12 Comparison of heat flux with Reynolds number for both the plain tube and the inserted tube

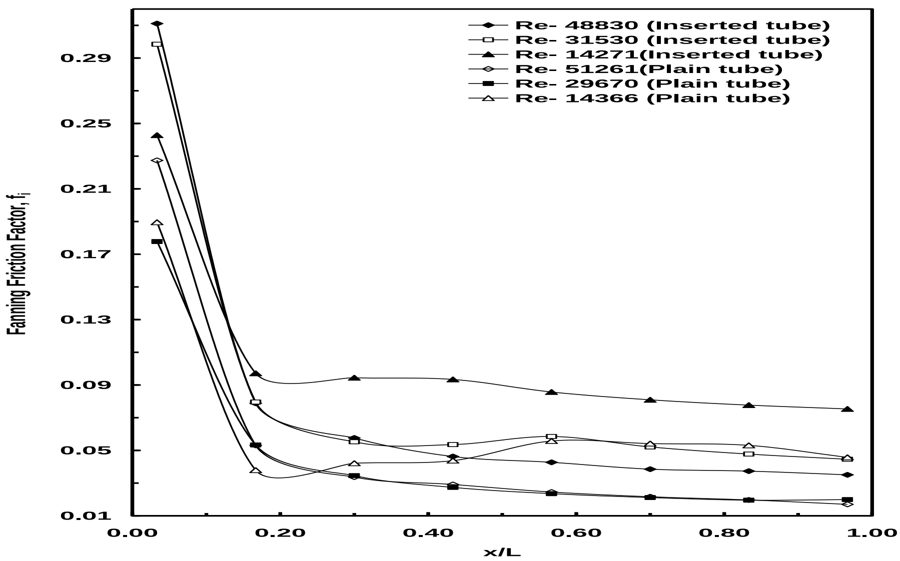


Fig. 13 Variation of fanning friction factor for both the plain tube and the inserted tube

4.3 Fluid Flow Characteristics

Fig. 13 shows the variation of fanning friction factor along the axial position for both the plain tube and the tube with insert at different Reynolds numbers. It could be shown from the Fig. 13 that the friction factor was higher near the entrance region then sharply fell up to a certain limit and finally it was almost constant. At the entrance region friction factor was higher as used of asbestos gasket between the inlet section and the test section. It was clear from the Fig. 13 that at lower Reynolds number the friction factor was higher. Fig. 13 showed that the friction factor was lower in the plain tube than that of the tube with insert for any axial location of the test section at a particular Reynolds number.

Tube with porous twisted tape insert created resistance to flow of the fluid through the tube. So, the friction factor was higher in the inserted tube than that of the plain tube.

The comparison of friction factor with Reynolds number for both the plain tube and the tube with porous twisted tape insert is exhibited in Fig. 14. The friction factor decreased with the increase in Reynolds number for both the plain tube as well as for the tube with insert. Fig. 14 indicated that at lower values of Reynolds number increase in the friction factor was comparatively high, and at higher values of Reynolds number it was low. This might be explained by the fact that at lower values of Reynolds number, corresponding to lower flow rate, air could pass all the pores and touched the tape and created high frictional forces.

It could be shown from the Fig. 14 that the friction factor of the tube with insert was 1.65 to 2.25 times higher than that of the plain tube at the comparable Reynolds number. The present of small vortices behind the wire, turbulence, increased surface area and the blockage created by the insert was responsible for higher friction factor. Rashid Sarkar *et al. (2005)* showed that the friction factor for tube with wire-coil inserts varied from 1.5 to 4.0 folds with compared to the plain tube at the comparable Reynolds number.

The variation of pumping power with Reynolds number for both the plain tube and the tube with insert is shown in Fig. 15. Pumping power increased with the increase in Reynolds number for both the plain tube as well as for the tube with insert.

It was shown from the Fig. 15 that the required pumping power of the tube with insert was higher than that of the plain tube at the comparable Reynolds number. The required pumping power of the tube with insert varied from 1.2 to 2.00 folds with compared to the plain tube. The required pumping power of the tube with wire-coil inserts (Sarkar *et al.*, 2005) varied from 1.25 to 2.5 folds in comparison to the plain tube.

4.4 Performance Evaluation

Fig. 16 shows the variation of performance parameter with pumping power of the tube with porous twisted tape insert. The performance parameter was evaluated on the basis of constant pumping power. Several performance criteria to evaluate the thermohydraulic performance of the enhance techniques have been proposed by Bergles *et al.* (1974) and Webb (1981). In this paper, the criterion three (R3) at constant pumping power outlined by Bergles was used and explained to quantify the benefits from the porous twisted tape inserted tube. Performance parameter was obtained of the tube with insert from Eq. (7) by using Eq. (8). It could be shown from the Fig. 16 that the performance parameter increased with the increase in pumping power of the tube with insert, but pumping power ranged from 1.07 to 1.76 watt the performance parameter slightly decreased. Performance parameter of the tube with insert varied from 1.85 to 2.0 folds with compared to the plain tube at the constant pumping power.

5. UNCERTAINTY ANALYSIS

The uncertainty analysis was performed by using the method of Kline and McClintock (1953). By this method the uncertainty of a variable R which was a function of independent variables x_1, x_2, x_2, --------------, x_n, could be estimated by taking root-sum-square of the contributions of individual variables. The individual uncertainties of different variables were measured in the present work and the uncertainties of the calculated quantities are provided in Table 1.

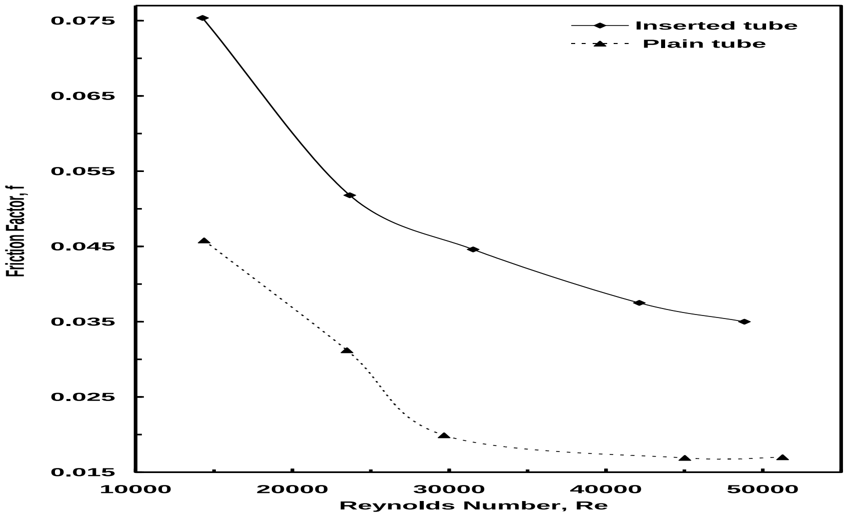


Fig. 14 Comparison of friction factor with Reynolds number for both the plain tube and the inserted tube

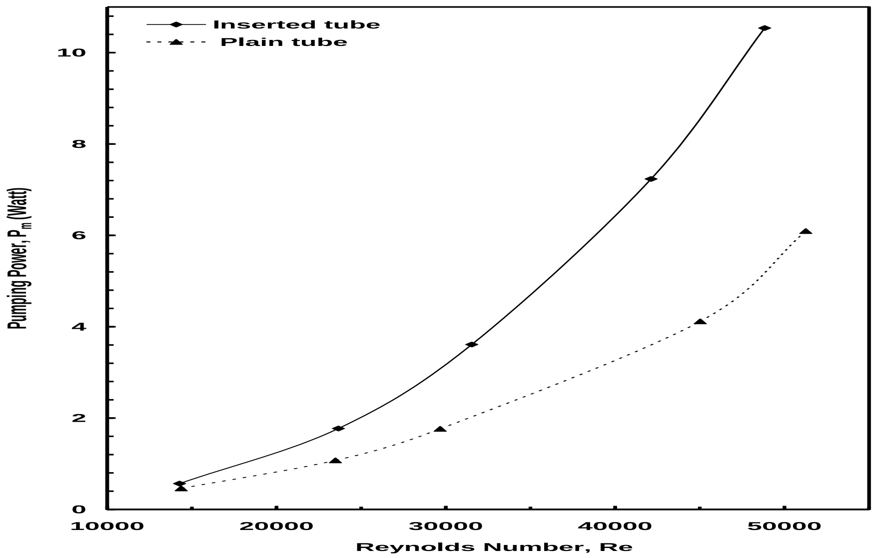


Fig. 15 Variation of pumping power with Reynolds number for both the plain tube and the inserted tube

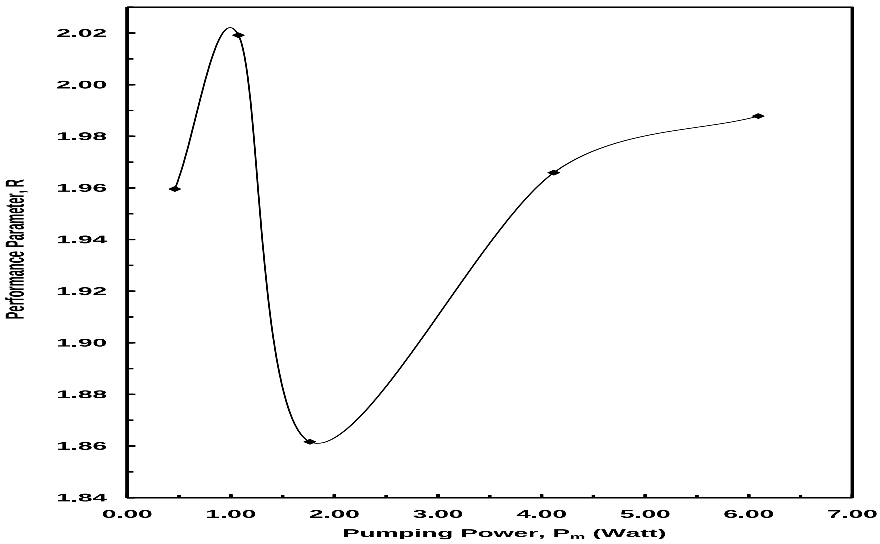


Fig. 16 Variation of performance parameter with pumping power

Table 1 Uncertainties of the different quantities

Measured Quantities	Uncertainty (%)	Measured Quantities	Uncertainty (%)	Calculated Quantities	Uncertainty (%)
Cross sectional area, A	0.520	Inlet temper., T_i	1.50	Friction factor, f_i	4.98
Atmospheric pressure, b	0.012	Outlet temper., T_o	1.50	Total heat transfer, Q	1.53
Velocity head, d	3.185	Wall temper., T_w	1.2	Velocity, V	1.59
Inside diameter, D_i	0.020	Axial distance, x	0.020	Nusselt number, Nu	4.46
Pressure drop, ΔP	3.83				

6. CONCLUSIONS

An experimental study was conducted to investigate the heat transfer performance of porous twisted tape insert. A significant enhancement in heat transfer was obtained without the scarifying of much additional pumping power. Heat transfer coefficient was higher in the entrance and exit regions of the test section. Average heat transfer coefficient varied from 2.25 to 2.60 folds, the heat flux varied from 1.45 to 1.55 folds of porous twisted tape inserted tube with compared to the plain tube values at the comparable Reynolds number. The friction factor was higher at the inlet of the test section and then dropped sharply towards the downstream up to the dimensionless distance of x/L = 0.2 and then became almost constant. The friction factor and the pumping power required of porous twisted tape inserted tube varied from 1.65 to 2.25 folds and 1.2 to 2.0 folds respectively, in comparison to the plain tube values at the comparable Reynolds number. The performance parameter was obtained as high as 2.0 times of porous twisted tape inserted tube with compared to the plain tube at constant pumping power.

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