Experimental Analysis of Heat Transfer and Friction Factor Characteristics in Turbulent Flow through a Tube Fitted with Screw Tape

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Abstract

The methods of augmentation heat transfer are widely utilized in many applications in the heating process to enable reduction in weight and size or increase the efficiency of heat exchangers. These methods are classified as passive, active and combination of (active & passive) these two techniques. For implementing active technique it is required external power such as electric or acoustic fields and surface vibration, whereas while adopting the passive techniques required special surface geometries, fluid additives, swirl/vortex flow devices, that is various types of tape inserts. The passive techniques are better the active techniques because the manufacturing process swirl formation inserts is simple and can be easily implemented in an existing heat exchanger. Moreover the passive techniques can play an important role in the intensification of heat transfer. According to working conditions a proper configuration of the insert is being selected .A Copper tube of 21 mm internal diameter and 25 mm outer diameter and 1000 mm test length was used. The heat transfer parameters was carried out by conducting an experimental study in tube with twisted tape and Screw tape with different diameters for Reynolds number range of 12,000 to 25000. The heat transfer coefficient, friction factor, Nusselt number, Thermal Performance Factor and enhancement efficiency of the twisted tape and Screw tape with different diameters have been obtained and compared with the plain tube. The comparisons showed that, compared with the twisted tape, the Screw tape with different diameters has further improved convective heat transfer performance by about 15 to 20 % and whereas lowered flow friction. The Screw tape with different diameters shows about 25% greater thermal performance than twisted tape at a cost of very low friction factor values.

Keywords- Heat transfer; twisted tape; Screw Tape with three different diameters; Nusselt number, friction factor, Thermal performance Parameter

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I. INTRODUCTION

Many researchers were investigated for augmentation of heat transfer rates inside circular tubes using wide range of inserts when turbulent flow is considered. The inserts studied included coil wire inserts, brush inserts, mesh inserts, strip inserts, twisted tape inserts right-left helical screw inserts etc. Heat exchangers have plays vital role in many industrial applications for example, coolant filtration systems, chilling oil units, heat recovery processes, food and dairy processes air conditioning and refrigeration systems etc. The design of heat exchanger needs exact analysis of heat transfer rate and pressure drop, for better performance and economic aspects of the equipment. Garc et al. [1] investigated and flow patterns in the tube with wire coil inserts in case of the laminar–transition–turbulent flow heat transfer enhancement. Hsieh et al. [2] experimentally studied the flow characteristics in a horizontal circular tube with strip-type inserts for the turbulent heat transfer domain. Bhuiya et al. [3] studied with inserts like twisted wire brush fitted in a circular tube the heat transfer performance and friction factor characteristics were investigated experimentally. It was achieved by 5 mm diameter of two twisted iron core-rods were winded by a 1 mm diameter of the copper wire over a. The densities are 100, 150, 200, and 250 wires per centimeter used for experimentation. For Reynolds number ranging from 7,200 to 50,200, and examining heat transfer and friction factor data in tubes. It is found that due to twisted wire brush inserts the heat transfer performance and friction factor characteristics in a tube for turbulent flow are affected by means of generation swirl. The twisted wire brush inserts have great significance on intensification of heat transfer rate with the penalty in increase in friction factor, which leads high power consumption. At the comparable Reynolds number the friction factor obtained for the tube with twisted wire brush inserts varied from 1.35 to 2.0 times than that of the plain tube values. With the increase of twisted wire densities it was found that the Nusselt number, thermal performance factor, and friction factor increased. At constant blower power for the tube with twisted wire brush inserts the thermal performance factor (η) obtained varied from 1.1 to 1.85 times than those of the plain tube values Bhuiya et al. [4] have done the experimental investigation on Nusselt number, friction factor and thermal performance factor inserts equipped in a circular tube with perforated twisted tape inserts with four different porosities of Rp = 1.6, 4.5, 8.9 and 14.7%. This study was emphasizing on heat transfer characteristics such as thermal performance factor Nusselt number, friction factor etc. They have conducted experiments using air as the working fluid under uniform wall heat flux boundary condition in a turbulent flow regime with Reynolds number ranging from 7200 to 49,800. Perforated twisted tape offers a higher heat transfer rate, friction factor and thermal performance factor as compared with the plain tube. With perforated twisted tape inserts the Nusselt number and thermal performance factor obtained from the tube was 340% and 59% higher than those of the plain tube values respectively. But friction factor increased by 360% hence pumping power also increased. In addition, among all the porosities i.e. 1.6%, 4.5%, 8.9%, and 14.7% the influence of porosity 4.5% was more effective than that of the other porosities of for the entire Reynolds number. The Thermal performance factor for all a configuration of perforated twisted tape inserts tends to decreasing with increasing Reynolds number. The thermal performance factor was more than one for all the cases which indicated that the effect of heat transfer enhancement due to the perforated twisted tape inserts was more dominant than the effect of the rising friction factor. M.M.K. Bhuiya [5] explored the effects on heat transfer and fluid friction characteristics in a heat exchanger tube by incorporating the inserts as the double counter twisted tapes. The counter-swirl flow generators in the test section produced by incorporating double counter twisted tapes in the test section. Double counter twisted tapes are used for performing experiments by using air as the testing fluid in a circular tube turbulent flow regime having Reynolds number from 6950 to 50,050. The experimental results demonstrated that with decreasing twist ratio the thermal enhancement efficiency Nusselt number and friction factor was increased. With increase in pressure drop the heat transfer rate was increased significantly in the tube fitted with double counter twisted tape. In general it was found that with decreasing twist ratio the heat transfer, friction factor and thermal enhancement efficiency increased. Furthermore, with increasing Reynolds number the Nusselt number increased the while for the case of friction factor and thermal enhancement efficiency the opposite trends were found i.e. it was decreased with increase in Reynolds number. At constant blower power the thermal enhancement efficiency for tubes with double counter twisted tapes achieved to be around 1.01 to 1.34. Bodius Salam et.al [8] have obtained results for friction factor, heat transfer enhancement efficiency, tube-side heat transfer coefficient of water for turbulent flow by carried an experimental investigation with an inserts as rectangular-cut twisted tape insert fitted in a circular tube. He showed variation of heat flux with Reynolds number between the smooth tube and tube with rectangular-cut twisted tape insert. The values of Nusselt numbers enhanced by 2.3 to 2.9 times in tube with rectangular-cut twisted tape insert were compared to those of smooth tube. The average enhancement was to found 2.6 times. For generation of swirl flow twisted tape was responsible because increasing the mixing between core and tube wall flows due to thinning the thermal boundary layer and, Eiamsa-ard et al., 2009. In addition the Rectangular-cut twisted tape makes additional disturbances because the tangential contact between secondary flow and the wall surface of the tube was increased, Murugesan et al., 2010. This made heat transfer coefficient higher through the flow. Pankaj N. Shrirao et.al [6] Experimental investigation of heat transfer rate and pressure drop which is measured in terms of friction factor such a heat transfer characteristics of horizontal circular pipe using pitch of internal threads of 100mm, 120mm and 160mm with air as the working fluid.

A. Nomenclature & Abbreviations

d_i	*Inside Diameter of tube (m)*	T_b	*Mean bulk temperature of fluid (air) (°C)*
D	*Outer Diameter of tube (m)*	C_p	*Specific heat of fluid (air) (J/KgK)*
d_o	*Diameter of orifice (m)*	Q	*Heat transfer rate (W)*
L	*Test tube section length (m)*	μ	*Viscosity of the fluid (air) (N s/m^2)*
A_f	*Flow area of test tube section (m^2)*	K	*Thermal conductivity of air (W/m k)*
A_s	*Surface Heat transfer area of tube (m^2)*	h	*Heat transfer coefficient for tube with insert (W/m^{2}k)*
a_1	*Cross-sectional area of test tube (inlet to orifice meter) (m^2)*	h_o	*Heat transfer coefficient for plain tube (W/m^{2}k)*
a_0	*Cross-sectional area of orifice (m^2)*	Re	*Reynolds Number (Dimensionless)*
C_d	*Coefficient of discharge of orifice meter*	Nu	*Nusselt Number for tube with insert (Dimensionless)*
ΔH_{wT}	*Height difference of water in manometer of test section (m)*	Nu_o	*Nusselt Number for plain tube (Dimensionless)*
ΔH_{wO}	*Height difference of water in manometer of orifice meter (m)*		
ρ_w	*Density of the water (kg/m^3)*	Pr	*Prandtl Number(Dimensionless)*
ρ_a	*Density of the air (kg/m^3)*	g	*acceleration due to gravity (m/s^2)*
$V^{\cdot}$	*Volume flow rate (m^3/sec)*	f	*Friction factor for the tube with inserts (Dimensionless)*
U_m	*Mean Velocity of air flow (m/s)*	f_o	*Friction factor for plain tube (Dimensionless)*
$m^{\cdot}$	*Mass flow rate (kg/sec)*	Δh_{en}	*Rate of Enthalpy change with insert (J/sec)*
T_i (T_1)	*Inlet temperature of fluid (air) (°C)*	Δh_{eno}	*Rate of Enthalpy change without insert (J/sec)*
T_o (T_6)	*Outlet temperature of fluid (air) (°C)*	TPF	*Thermal Performance Factor (Dimensionless)*
T_s	*Averaged Surface temperature of the Tube (°C)*	η_{en}	*Enhancement Efficiency (Dimensionless)*

P. Sivashanmugam et.al [7] experimental investigation on increase in heat transfer and increase in friction factor characteristics of circular tube fitted with inserts as right-left helical screw having equal in length, and unequal length of different twist ratio have been presented. For a given twist ratio the intensification of heat transfer coefficient of straight helical twist is lower than that for right-left helical screw inserts. Halit bas [9] Flow friction and heat transfer behavior in a circular tube used swirl generator as twisted tape inserted in the tube are investigated experimentally. From the tube wall the twisted tapes are inserted separately. For comparison Halit bas was the discussed results in the range of Reynolds number from 5132 to 24,989. He used seamless stainless steel test tube of grade SS304 and calming tube with 56 mm inner diameter (D1) and 60 mm outer diameter (D2). In this experiments the twisted tapes tested, with five different twist ratios y/D = 2.0, 2.5, 3.0, 3.5 and 4.0 and two clearance ratios i.e. c/D = 0.0178 and 0.0357. The heat transfer enhancement inversely proportional to Reynolds number within 15,000 and above 15,000 it is nearly constant these results for twist ratios are lower than 3. For c/D = 0.0178 the highest heat transfer enhancements are obtained as 1.756 and for c/D = 0.0357 as 1.744. Subhankar Saha, Sujoy Kumar Saha et.al [10] has been developed helical screw-tape insert in a circular duct having integral helical rib roughness. He had investigated by experimentally friction factor and Nusselt number data for laminar flow of viscous oil in that duct fitted with inserts as helical screw tape. The major finding of this experimental investigation was that with integral helical rib roughness, the helical screw-tape inserts in combination perform significantly better. By using helical screw-tapes the inertia force due to swirl flow generated enhances the boundary layer separation and reattachment with temperature which results as equally flattered the velocity profiles, caused by integral helical rib roughness and hence the enhancement. This research findings gives enhancement in Nusselt number was 150% as compared to plain circular duct. That is useful in designing tubes in parabolic trough solar collector carrying viscous oil. Therefore solar thermal electric power plants becomes environmentally sound and cost-effectively.

The above mentioned survey of past literature indicates that the numerous research articles were reported on intensification of heat transfer and pressure drop characteristics in tubes with various configurations of inserts as turbulence creator. Hence this study emphasis on Screw tapes as an inserts which is placed at central position by two supporting tie rods in the test section. Further those results were compared with plain tube as well as twisted tape insert. The scope of the present work is investigation of the tube side heat transfer characteristics under steady state heat flux such as heat transfer coefficients and friction factor of a circular tube fitted with screw tape insert with three different diameters. When screw tape placed in the tube at central position, then it breaks the thermal boundary layer at core Centre only hence the width of boundary layer reduces radially which helps for intensification of heat transfer coefficient. Hence improved in heat transfer rate with minimum pressure drop as compared to twisted tape. Furthermore in that decreasing core diameter of screw tape gives better than others.

In order make easy analysis of this present experimental study some assumptions were made. These were:

1) Heat transfer rate by the heater to fluid was calculated by considering forced convection only. Heat transfer by conduction through contacts between inner side of tube and inserts were neglected.
2) All the fluid properties were considered local at bulk temperature.

II. MATERIALS AND METHODS

In this experimental apparatus centrifugal blower unit fitted on the ground floor by using a circular piping and necessary pipe fittings, it was connected to the test tube which was mounted in horizontal orientation. Nichrome heating wire encloses the test section to a whole length of 1m; with a pitch distance of 5 mm. The test section was insulated with glass wool and metal cladding. Input to heater is given through dimmer stat for achieving steady state heat flux condition. Four thermocouples T2, T3, T4 and T5 at an equal distance of 15 cm from the origin of the heating zone were inserted on the walls of the tube and one thermocouple (T1) was placed in the air stream at the inlet section of test tube and other was placed in the air stream at the outlet section of test tube (T6) of the test section to measure the temperature of flowing air at respective section as shown in fig. 1.

The digital multipoint temperature indicator was used to display the temperature at various positions which was measured by thermocouples. The temperature measured by instrument is in ^{0}C. For measurement pressure drop across the test section a U- tube manometer measures the pressure drop filled with water. The piping system consists of a ball valve, which was used for controlling the airflow rate through it and volume flow rate of air through the system an orifice meter was used. The diameter of the orifice is .0125m and coefficient of discharge is 0.65. The two pressure tapings of the orifice meter are connected to a water U-tube manometer to indicate loss of pressure head across the test section. Display unit is a digital Multipoint temperature indicator used for indicating temperatures of six different locations. A variation in the flow rate of air was measured by difference in the fluid of levels of manometer. The velocity of airflow in the tube is measured with the help of orifice plate and the water manometer fitted on board.

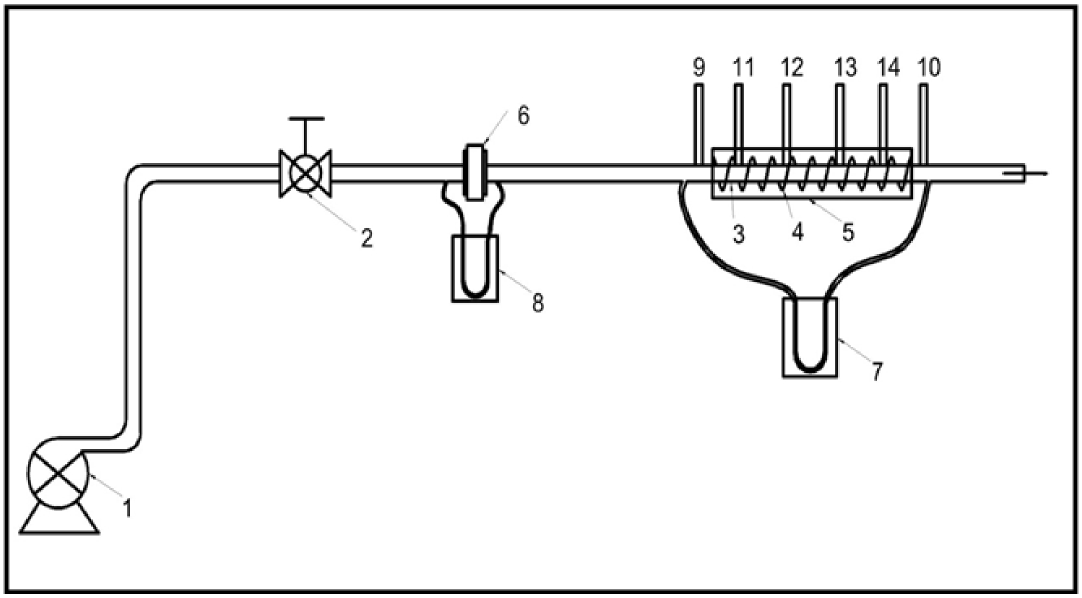

Fig. 1: Experimental Set up Layout

1) Blower, 2. Gate Valve 3. Test Tube section 4.Heating Coil, 5.Insulation 6.Orifice meter 7. U-tube Manometer for Test section (to measure pressure difference across test section) 8. U-tube Manometer for Orifice meter, 9&10 Inlet & Outlet Thermocouples, 11 To 14 Surface Thermocouples.

A. Materials

The experimental layout consists of PVC piping for vertical section from outlet of blower unit with necessary pipe fittings. Then it was connected to the ball valve of S.S. made through M.S. elbow and fittings. The inlet section of set up was consists of Stainless steel pipe, orifice plate made from S.S. and flange which was connected to the test tube made from copper through flange with gaskets nut bolt fittings. In this experimental investigation two types of inserts were used in that first one was twisted tape made from aluminium which was having only one configuration as shown in fig.2

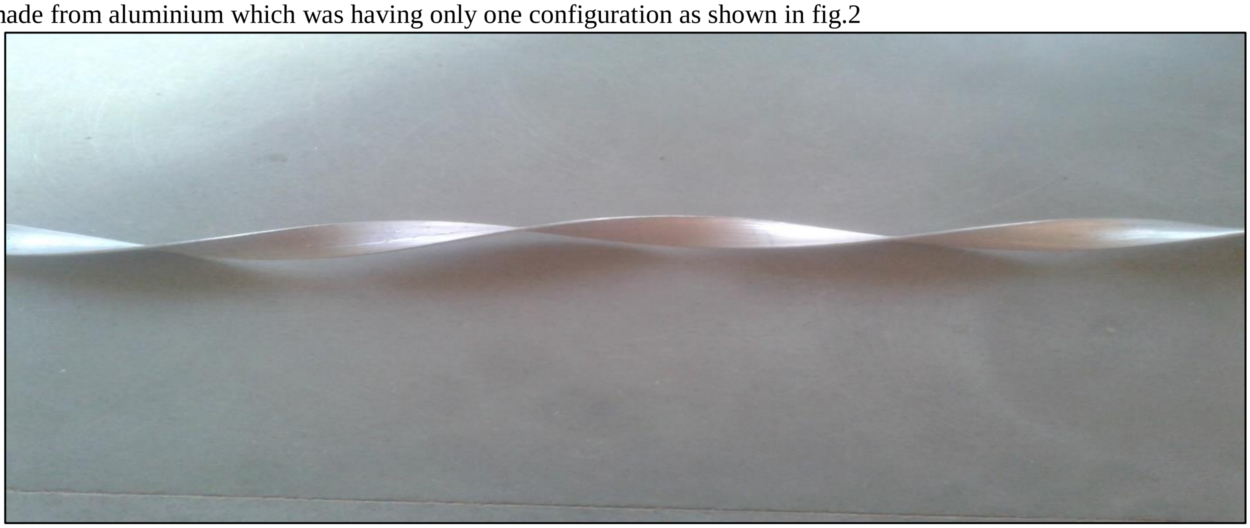

Fig. 2: Twisted Tape

The length of the twisted tape is equal to the length of test tube i.e. equal to 1 meter. The width of the twisted tape is 19 mm and pitch is 70 mm.

Second type of inserts was screw tape having three types of configurations as shown in fig.3.

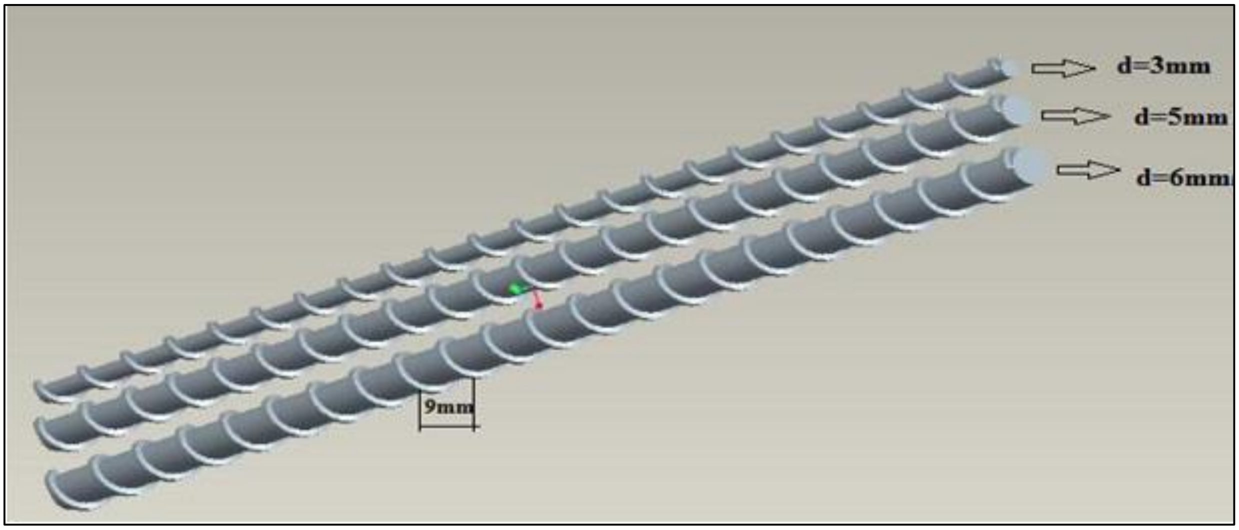

Fig. 3: Screw tape inserts (W= 8mm, p=9mm, t=1mm, d=3mm, 5mm, 6mm)

Screw tapes were made from M.S. and manufactured by using milling machine. In milling machine used rod of diameter (W= 8 mm) in two jaws. One jaw having circular motion from motor with the help of gear train. By using gear train we can adjust pitch (p= 9mm), thickness (t= 1mm) and height of tape from diameter of barrel rod (d= 3mm, 5mm, 6mm respectively). And second jaw is used to hold work piece.

B. Experimental Procedure

Switch on the motor of Centrifugal Blower first air was made to flow throughout system by full opening of ball valve. Then adjust the gate valve for controlling air flow rate. A heat input of 90 W was given to the flexi glass heater coil wound on the test tube by adjusting the dimmer stat. Thermocouples used for this was K-Type with an accuracy ±0.2%. In order to avoid the loss of heat energy to the surrounding the test tube was insulated. Thermocouples T_2 toT_5 were fixed on the test surface and thermocouples T_1 T_6 were fixed near the inlet and exit of the test tube section. The readings of the thermocouples were observed every 5 to 8 minutes until the steady state condition was achieved. The experiment was conducted for plain tube then repeated for test tubes different inserts with constant airflow rate. The fluid properties were calculated at the mean bulk temperature i.e. averaged of the inlet and the outlet temperatures. At constant heat flux input and constant mass flow rate the actual experiments were carried out. Experiments were carried out first on plain mild still horizontal circular tube then on horizontal circular tube with plain twisted tape of constant pitch and then on horizontal circular tube with twisted tape with elliptical hole of constant pitch, then on horizontal circular tube with screw tape varying diameters(d=3mm,5mm,6mm).

With Plain Tube: Initially, the experiment was carried out on plain circular tube without twisted tape. The working fluid air flows through the tube section with least resistance and at various mass flow rates such as 0.0036376, 0.0046898, 0.0052985, 0.0056891 and 0.0062864 kg/sec.

III. DATA REDUCTION

1) Average Surface Temperature

$$T_s = \frac{T_2 + T3 + T4 + T5}{4} \qquad 1$$

2) Mean bulk Temperature

$$T_b = \frac{Ti + To}{2} \qquad 2$$

3) Discharge of Air:

$$\dot{V} = C_d \frac{a_1 a_0 \sqrt{2g\Delta h_{wo} \frac{\rho_w}{\rho_a}}}{\sqrt{a_1^2 - a_0^2}} \qquad 3$$

Where

4) Cross sectional Area of tube:

$$a_1 = \frac{\pi}{4} d_i^2$$

5) Cross sectional Area of orifice:

$$a_0 = \frac{\pi}{4} d_0^2$$

$$\text{Velocity} \quad U_m = \frac{\dot{V}}{A_f} \qquad 4$$

Mass flow rate of air $\dot{m} = \dot{V} \times \rho_a$

6) Reynolds Number:

$$\mathrm{Re} = \frac{\rho_a U_m d_i}{\mu_a} \qquad 5$$

7) Rate of Heat Transfer:

$$Q = \dot{m} Cp (T_o - T_i) \qquad 6$$

Heat transfer coefficient:

$$h = \frac{Q}{A_s (T_s - T_b)} = \frac{\dot{m} Cp (T_o - T_i)}{A_s (T_s - T_b)} \qquad 7$$

Where A_s= Surface Area of Tube = $\pi d_i L$

8) Experimental Nusselt number:

$$Nu = \frac{h d_i}{k} \qquad 8$$

9) Prandtl number (Pr):

$$\mathrm{Pr} = \frac{\mu C_p}{k} \qquad 9$$

10) Theoretical Nusselt number by Dittus Boelter equation:

$$Nu_t = 0.023 \mathrm{Re}^{0.8} pr^{o.4} \qquad 10$$

11) Friction Factor:

$$f = \frac{\Delta P}{(L/d_i)(\rho_a U_m^2)/2} \qquad 11$$

12) Blasius equation for Theoretical Friction Factor of Turbulent Flow:

$$f_t = 0.3164 \mathrm{Re}^{-o.25} \qquad 12$$

13) Thermal performance factor:

$$TPF = \frac{\frac{Nu}{Nu_0}}{\left(\frac{f}{f_0}\right)^{1/3}} \qquad 13$$

14) Enhancement Efficiency:

$$\eta_{en} = \frac{h}{h_0} \qquad 14$$

IV. RESULTS AND DISCUSSIONS

A. Experimental Uncertainty

The Nusselt numbers which were obtained from experimental work which was called experimental values. Further these values were compared with Dittus-Boelter equation i.e. theoretical values. It is seen that the experimental results are in good agreement with afore mentioned studies and Nu (Experimental) is less than Nu (Dittus-Boelter). Actual heat carried away by air while passing through the test tube was nothing but the combination of convective and radiative heat transfers. As the heat transferred to fluid by only convection mode was considered while performing numerical calculations (Eq. 6), it can be expected that Nu experimental is less than Nu theoretical. The experimental uncertainty is found as 16.51% for Nusselt number.

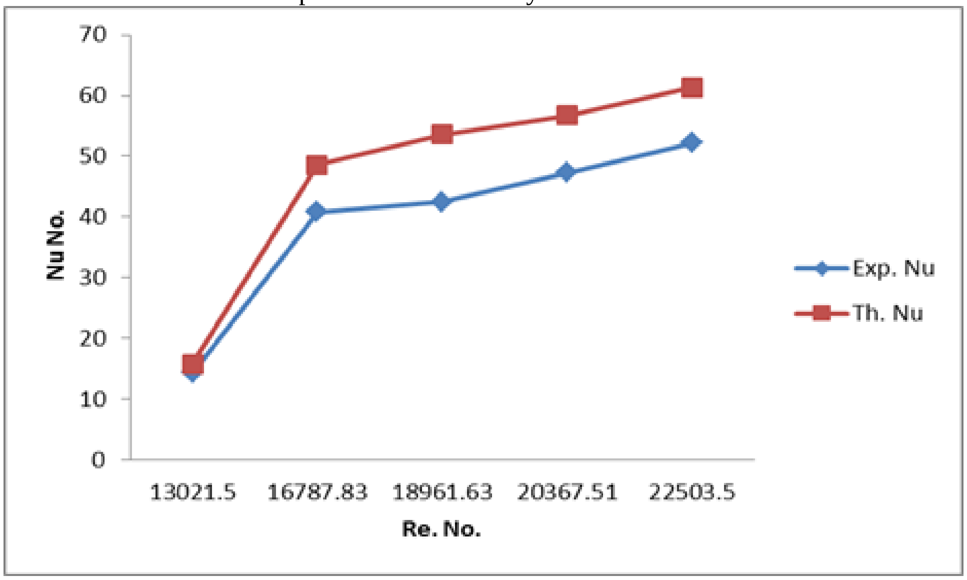

Fig. 4: Variation of Nusselt number for plain tube

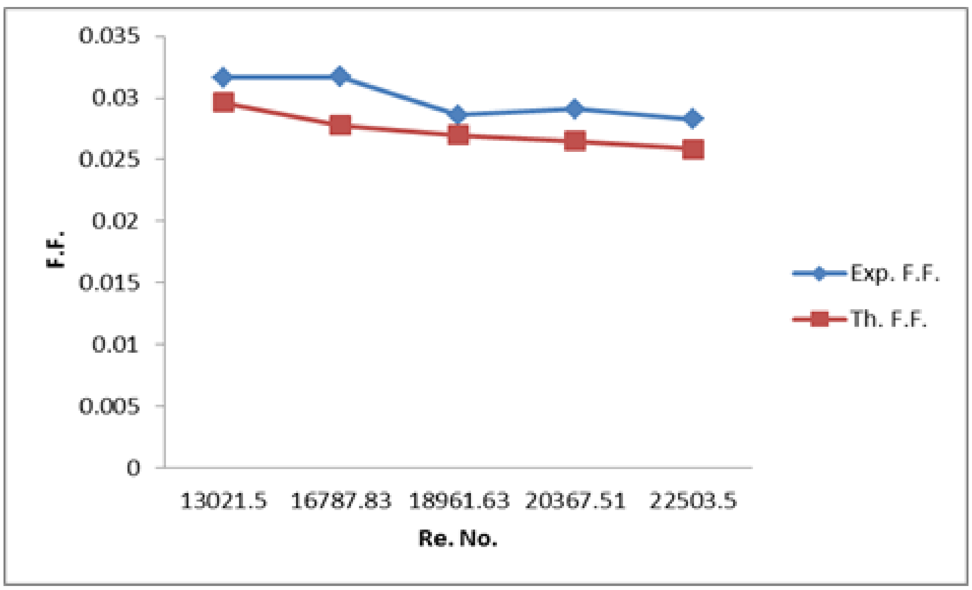

Fig. 5: Variation of Friction Factor for plain tube

From above Fig. 5 Friction factor is observed to reduce with increase in Reynolds number for plain tube. The friction factor obtained from experimental work was compared with the value obtained using Blasius equation (Theoretical) (Figure 5). It is seen that the experimental results were in good agreement with afore mentioned studies and Friction Factor (Experimental) is greater than theoretical Friction Factor (Blasius equation). Actual resistance to air flowing through the test section is higher. The experimental uncertainty is found as 8.44 % for Friction Factor

4.2 Heat transfer characteristics

Fig.6 shows that comparison of Nusselt Number and Reynolds number for Plain tube, with twisted tape insert and with different screw tape inserts of (d=3mm,5mm,6mm).

This study gives the best insert for using enhancement of heat transfer or increasing convective heat transfer coefficient of tube side at same Reynolds number. The effect of screw tape insert at (d=3) highest Nusselt number because rib helix angle and height of insert are plays important role in swirl flow generation in tube side flow, so it has been break the actual boundary layer and proper mixing of thermal and hydraulic boundary layer as compare to other inserts. Hence screw tape d=3mm is the best insert for the passive heat transfer techniques, and also has reduced the pumping power.

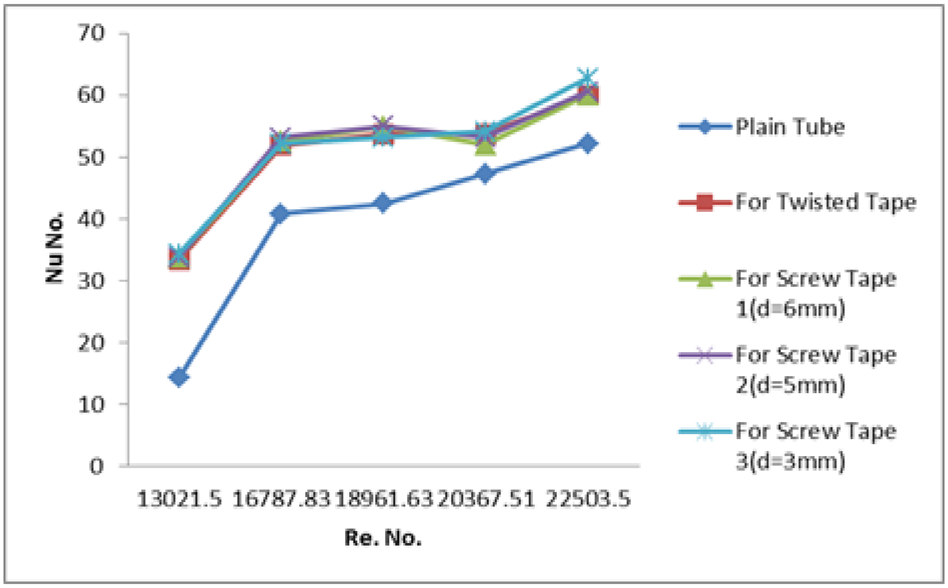

Fig. 6: The Variation of Nusselt Number with Reynolds Number.

B. Fluid Flow Characteristics

Fig 7 shows that comparison of Friction Factor and Reynolds number for plain tube with different tape inserts. This study gives the best insert for reducing pumping power because pumping power is heart of heat exchanger. Also to enhance the thermo hydraulic performance. The effects of twisted tape insert gives highest Friction Factor because twist angle and high width.

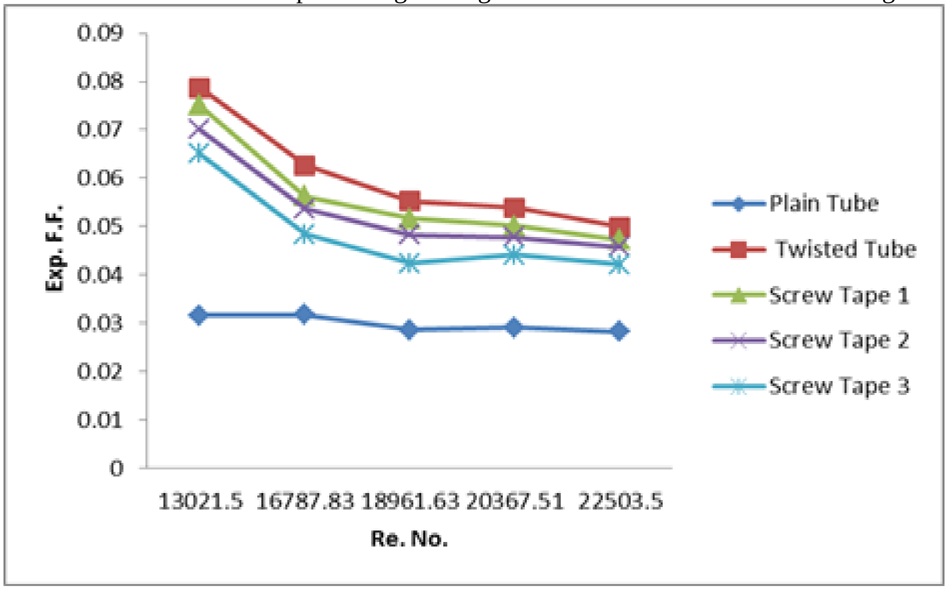

Fig. 7: The variation Friction Factor with Reynolds Number

Also surface roughness of inserts is plays important role to create a friction force in given flow domain. There is lot of effect on pressure difference between inlet and outlet section of the tube for a twisted tape and screw tape 1(d=6) insert hence increased pumping power as compared to screw tape insert (d=3mm and 5mm). It is concluded that screw tape 3 (d=3mm) gives less friction factor as compared to other types of inserts. Twisted tape shows highest friction factor among all the types of inserts which is to be considered to evaluate the performance of the inserts.

C. Thermal Performance

Fig 8 shows TPF decreasing with increasing Re no. for lower Re. no. furthers it is remains constant in between the value of 0.90 to 1.10. For Screw tape 3 (d=3mm) TPF never less than 1 therefore it is best insert as compared to types of insert.

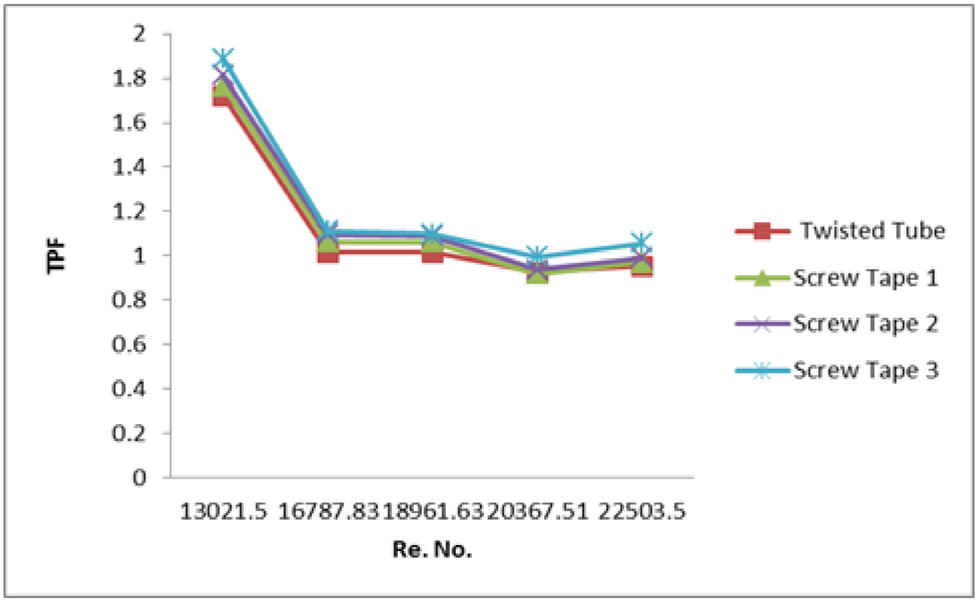

Fig. 8: Variation of TPF with Reynolds Number

D. Enhancement Efficiency

Fig.9 shows that enhancement efficiency at lower Re. No. is much higher than the higher Re. No. for higher values. Also for screw tape with lowest diameter gives best results.

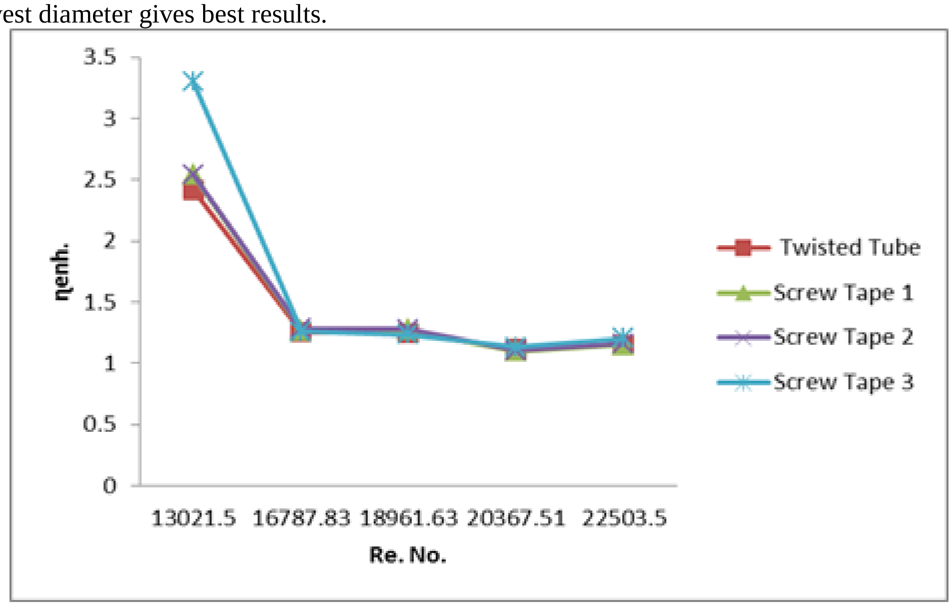

Fig. 9: Variation of Enhancement Efficiency with Reynolds Number

V. CONCLUSIONS

The effects of the augmentation techniques on the heat transfer enhancement and friction factor behaviors in turbulent flow regimes (12000<Re<25000) are described. The Twisted tape of Al and Screw tapes of M.S. in plain tube combined with different geometries (W= 8mm, d=3mm, 5mm, & 6mm H=9mm) at different fluid flow rates are tested using the atmospheric air as working fluid. The conclusions are drawn as follows:

1) The measurements are in good agreement with each other. The maximum error between the averaged experimental Nu number with numerically predicted is 16.51% , and that for friction factor is 8.11%
2) Nu no increases about 15 to 20 % in twisted tape, and by 16 to 25 % in Screw tape 1 & 2 and it slightly higher for screw tape 3
3) As the friction factor goes on decreasing as Re no increases. Experimentally for highest values of Re no it shows less deviation between all the types of inserts. Friction factor for twisted is high and that for screw tap 3 is low.
4) With the decrease of rib height or diameter of screw tape, Nusselt number increases with increase in Reynolds Number as compared to twisted tape.

5) For increase of the diameter of insert it shows that decrease in heat transfer coefficient and Nusselt number but at same time pressure drop also increases.
6) As Twist ratio increases greater the heat transfer and lower the friction factor.
7) From the above discussion we found that screw tape insert at (d=3mm) is the best insert for heat transfer augmentation.

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