Heat Augmentation Using Non-metallic Insert in Forced Convection

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Abstract: *Investigations have been carried out over decades to enhance the heat transfer using active, passive and compound heat augmentation techniques. The present work includes passive heat transfer augmentation technique using non-metallic flow divider type inserts with air as working fluid. Whenever inserts are used for heat transfer enhancement, along with increase in heat transfer rate, the pressure drop also increases. This increase in pressure drop increases the pumping cost. Hence heat augmentation technique should optimize between the benefits of heat transfer rate and increased pressure drop. Initially, for validation of setup, experiments were carried out for plain tube. The results i.e. Nusselt number and friction factor obtained were validated against those obtained from empirical correlation. Secondly, Experimental investigations were carried out using flow divider type non-metallic inserts with varying the Reynolds number and heat input. The Nusselt number is found to increase to 112 and 187 percent for "Differential Annular Type of Insert" and "Non- metallic Acrylic Twisted Tape Insert" respectivelythan that of plain tube. Overall enhancement efficiency for different Reynolds number was above unity for both the inserts.*

Keywords: *Forced Convection, Flow Divider Insert, Heat Augmentation, Nusselt Number, Reynolds Number.*

I. Introduction

Heat transfer processes in industrial, Automotive and domestic application involves the conversion, transfer and utilization of energy. The Enhancement of heat transfer in such several application can significantly improve the thermal performance of heat based equipment's (for instance, heat exchanger) are called as heat augmentation techniques. Heat augmentation techniques, used to increase the convective heat transfer coefficient by increasing the turbulence of the fluid. This technique also leads to increase in pressure drop and hence the pumping cost. Hence the augmentation technique should optimize between the heat transfer rate and pressure drop. Heat transfer augmentation techniques are broadly classified into three categories:

1) Active techniques,
2) Passive techniques and
3) Compound techniques.

Active methods require some external power input for the improvement of heat transfer and have not shown much prospective owing to intricacy in design. While, Passive methods do not need any external power input and the supplementary power needed to augment the heat transfer is taken from fluid itself, which ultimately leads to drop in fluid pressure. Compound methods are hybrid methods in which both active and passive methods are used in combination. Passive and active techniques that are used for heat augmentation of laminar flow have been discussed in detail by Bargles[1].These are effective but involve complicated design and hence have limited applications [1-3]. Heat augmentation can further increased by perforating the twisted tape with different shapes with different arrangement [3]. These techniques includes twisted tape [3,6]. Use of alternate clockwise and counterclockwise twisted tapes with different twist ratio increases the Nusselt number over a range of Reynolds number with uniform flux condition. Square cut circular ring [4], coiled tube [5], rough surfaces [8], additives to the fluid [8], extended surfaces [10] have been used successfully to enhance heat transfer.

According to the first technique, insert acts as a turbulence promoter, producing the helical flow of fluid which induces the drive forces to move the heated fluid from boundary surface to core flow. In second technique, by inducing the heat transfer surface area and redevelopment of boundary layer the thermal performance of system is increased. Insertion of twisted tube is the simple passive technique for enhancement of heat transfer coefficient on the tube side of heat exchanger. Although typical twisted tapes offer better heat transfer enhancement, the tapes leads to high pressure drop penalty which limits thermal performance of systems. Therefore, various modified twisted tapes have been designed to reducepressure loss by introducing spaces or gaps to twisted tapes. Overall enhancement ratio is used to indicate the thermal performance of the

enhancement technique over a plain tube with constant pumping power. Dittus-Boelter and Petukhov correlations of forced convection for plain tube were used to validate the setup.

II. Experimental Setup

The photograph of experimental setup is depicted in Fig.1.Experimental setup consists of assembly of channel flow passages including blower, orifice meter, wattmeter, test section, Nichrome band heater. The test section is the G.I.pipe with inner diameter of 27.5mm and 500mm length. Band type Nichrome heater is used for heating the pipe (test section). Wattmeter was used to adjust the heater input to the test section. Test section is insulated using asbestos insulation. Two Copper-Constantan (k-type) thermocouple are placed at inlet and outlet section to measure air temperature and 5 thermocouple soldered to the test section. For measuring the pressure head loss across the test section, tapping are provided at inlet and outlet of test section. This head loss is measured usingdigital manometer. Velocity of air is calculated from the discharge obtained from manometer attached across orifice plate. Control valve is used to adjust the flow rate of air. Before experimentation thermocouples and orifice plate were calibrated.
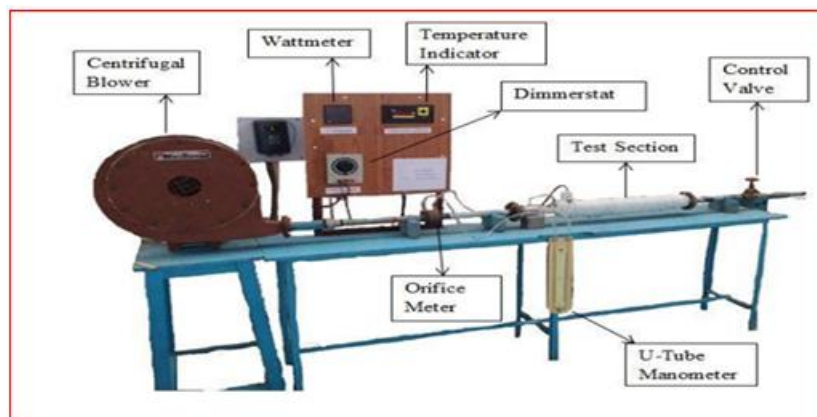
Fig. 1(a)
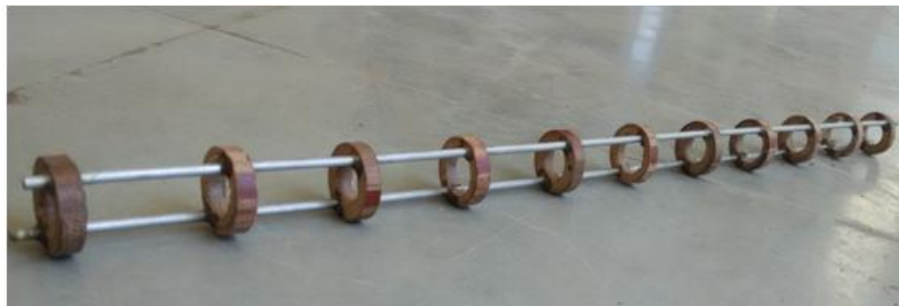
Fig.1 (b)

Fig.1 (c)
Fig.1. a) Photograph of experimental setup **b)** Bakelite Differential Annular Insert **c)** Twisted tape insert

III. Experimental Procedure

Experimentation is carried out in two stages.

3.1 Validation of Test Section (Plain Tube) using correlations

Supply is given to the blower motor and the valve is opened slightly. The blower was started and the airflow rate was adjusted by operating the control valve, so that the desired difference in manometer level (20mm, 40 mm, 80 mm, 100 mm, 120 mm) was obtained. Start the Nichrome bend heater wound on the test section by giving desired heat input by adjusting the dimmer stat. Thermocouples 2 to 6 are fixed on the test surface to measure surface temperature and thermocouples 1 and 7 are fixed inside the pipe to measure inlet air temperature and outlet air temperature respectively. The readings of the thermocouples are observed every 5 minutes until they show constant values. The variation of tube wall temperatures (T2, T3, T4, T5 and T6) were observed until the constant value was attained and then the outlet air temperature (T7) was observed until the temperature didn't deviate over fifteen to twenty minutes. Steady state temperatures from T1 to T7 are tabulated**.** Under steady state condition, the readings of all the seven thermocouples are recorded for heater input 50W, 75W, 100W. The experiment is repeated for different openings of the valve, thus varying the airflow rate. For each test run, temperatures, volumetric flow rate and pressure drop of bulk air at steady state conditions were noted down. The fluid properties were calculated as the average between the inlet and the outlet temperature. It took 90 minutes to reach steady state conditions. Experiment was carried out at constant heat flux conditions and constant heat input at different mass flow rates. The Reynolds number of the air was varied from 6000 to 12000 during the experimentation. The various characteristics of the flow, Nusselt number and the Reynolds number were based on the average tube wall temperatures and the inlet and outlet air temperatures were plotted. Nusselt numbers and friction factor calculated from the experimental data for plain tube are compared with the correlation recommended by Dittus–Boelter and Blasius respectively.

3.2 Experimental investigation with non-metallic flow divider inserts.

Test section is disconnected from the flange. The **B**akelitedifferentialinsert is inserted into the tube axially. The experiment is carried out at similar conditions to the plain tube experiment and the test section is heated to the preset value, which is adjusted using a dimmer stat on the instrumentation panel. It took 150 minutes to reach steady state conditions. The various characteristics of the flow, Nusselt number and the Reynolds number were based on the average tube wall temperatures and the inlet and outlet air temperatures were plotted.

IV. Mathematical model

Mean Bulk Temperature,

$$T_{mb} = \frac{T1 + T7}{2} (^{o}C) \tag{1}$$

Mean Surface Temperature,

$$T_S = \frac{T_2 + T_3 + T_4 + T_5 + T_6}{2} (^{o}C) \tag{2}$$

Properties of air, Cp, μ, k and Pr are calculated at bulk mean temperature i.e. at $T_{mb.}$. Equivalent Column Height of Air,

$$h_a = \frac{h_w \rho_w}{\rho_a} (m) \tag{3}$$

Volume flow rate

$$Q_d = \frac{A * A_0 * \sqrt{2 * g * h_a}}{\sqrt{A^2_0 - A^Z}} \quad (m^3/s) \tag{4}$$

Mean Velocity of Air through Pipe

$$U = \frac{Q_d}{A} (m) \tag{5}$$

Nusselt number from Dittus-Boelter correlation is given as,

$$(Nu)_{DB} = 0.023 \times Re^{0.8} \times Pr^{0.4} \tag{6}$$

$$\text{Where, Reynolds Number (Re)} = \frac{4 * \dot{m}}{\pi * D * \mu} \tag{7}$$

$$\text{Prandtl Number (Pr)} = \frac{\mu * C_p}{K} \tag{8}$$

Nusselt number from experimental data can be calculated as,

$$(Nu)_{exp} = \frac{h \times D_h}{k} \tag{9}$$

Where, Convective heat transfer coefficient,

$$(h) = \frac{Q_t}{A_S * (T_S - T_{mb})} \quad (W/m^2K) \tag{10}$$

Hydraulic diameter, $\quad (D_h) = \frac{4 \times A_{fr}}{wetted\ perimeter}(m)$ (11)
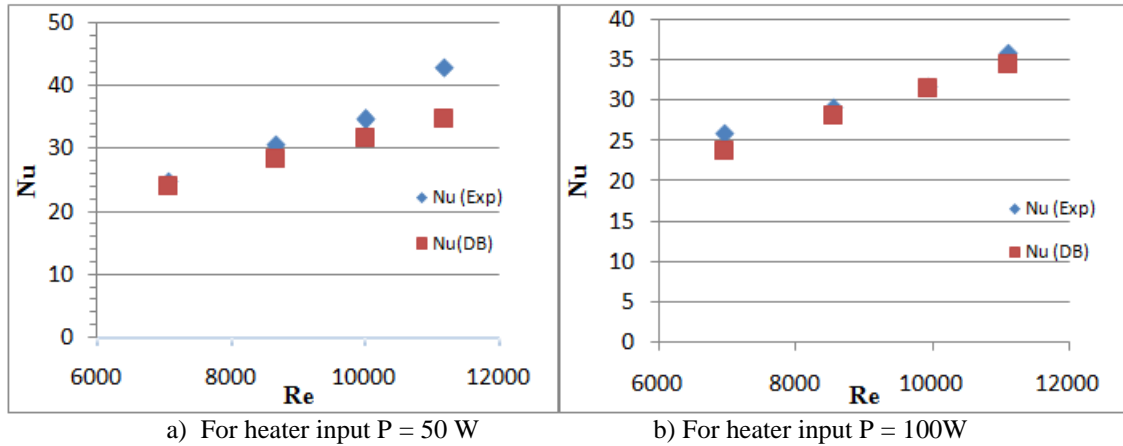
Friction factor Blasius correlation for plain tube,
$$f = 0.316 \times Re^{-0.25} \tag{12}$$

Friction factor from Petukhov Correlation for plain tube,

$$f = (0.79 \times ln\ Re - 1.64)^{-2} \tag{13}$$

Friction factor obtained from experimental data can be calculated as,
$$f = \frac{\Delta p_{exp}}{\left(\frac{L}{D}\right) \times \left(\frac{\rho_a \times U_i^2}{2}\right)} \tag{14}$$

V. Results And Discussion

A} Differential Annular Insert

a) For heater input P = 50 W   b) For heater input P = 100W

Nusselt number and friction factor obtained are compared in order to validate the experimental setup. It is observed that, the experimental data are in good agreement with the correlation values, i.e., Dittus – Boelter Correlation and with Blasius correlation as shown in Fig.2. Experimental investigations were conducted with **B**akelite differential annular insert which revealed that, the thermal performance of tube fitted with insert was better than that of plain tube. Results obtained from plain tube and with inserts are as shown in fig.3. Nusselt number and heat transfer coefficient increased for differential annular insert. Maximum increase in Nusselt number is 112% more than plain tube.
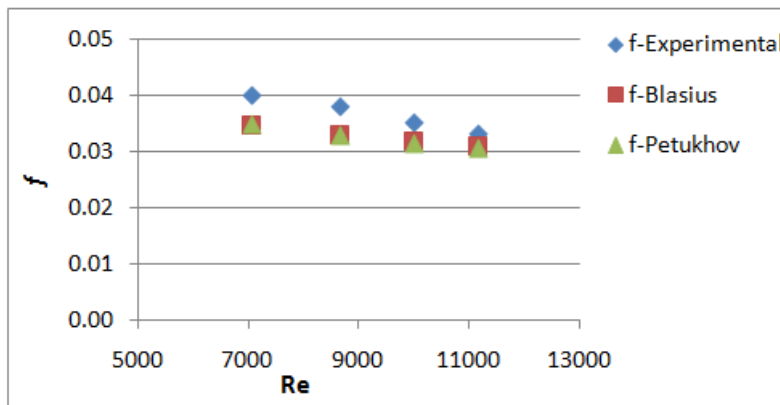
c) Friction factor vesues Re
Fig.2. Validation of setup (plain tube)
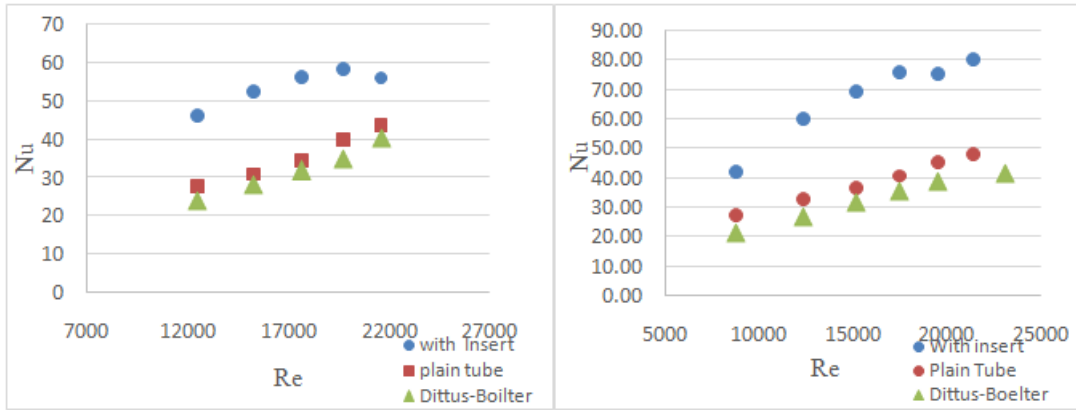
a) For heater input P = 50W   b) For heater input P = 100W
Fig.3 Comparison of Nusselt number with and without insert
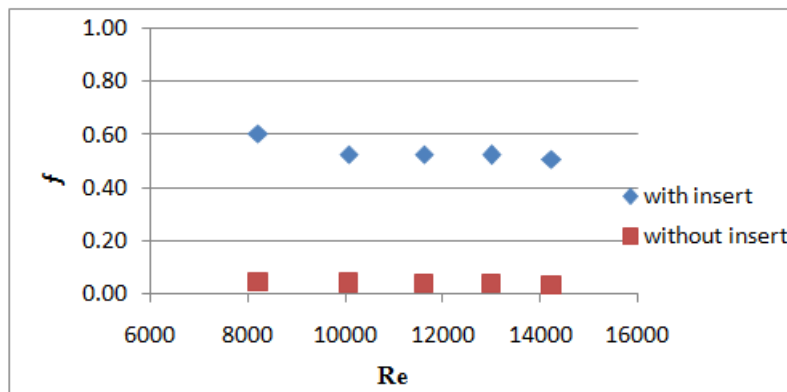
Fig.4 Variation of friction factor with Reynolds number

The friction factor, which is high at lower Reynolds number, tends to reduce with the increase of Reynolds number as shown. This is due to the fact that, *f is* inversely proportional to square of velocity for a given pipe. So, friction factor decreases due to increased velocity associated with increase in mass flow rate as shown in Fig.4. Overall enhancement ratio and enhancement efficiency are the important terms used to compare the thermal performance of the insert. The overall enhancement ratio (η) is useful to evaluate the quality of heat transfer enhancement obtained over plain tube at constant pumping power. The value of OER is above unityfor different flow rate of air and heater input and maximum value of OER is 1.12 as shown in Fig.5.

$$\eta = \frac{[\frac{Nu_i}{Nu}]}{[\frac{f_i}{f}]^{\frac{1}{3}}} \qquad\qquad (15)$$

Where, Nu_i = Nusselt number with insert
f_i = frictionfactor with insert

Enhancement efficiency (ψ) indicates thermal performance of heat augmentation technique. Enhancementefficiency increased with increase in Reynolds number. Maximum value of enhancement efficiency obtained is 2.39. Fig.5 shows the variation of enhancement ratio with Reynolds number.

$$\psi = \frac{h_i}{h} \qquad\qquad (16)$$

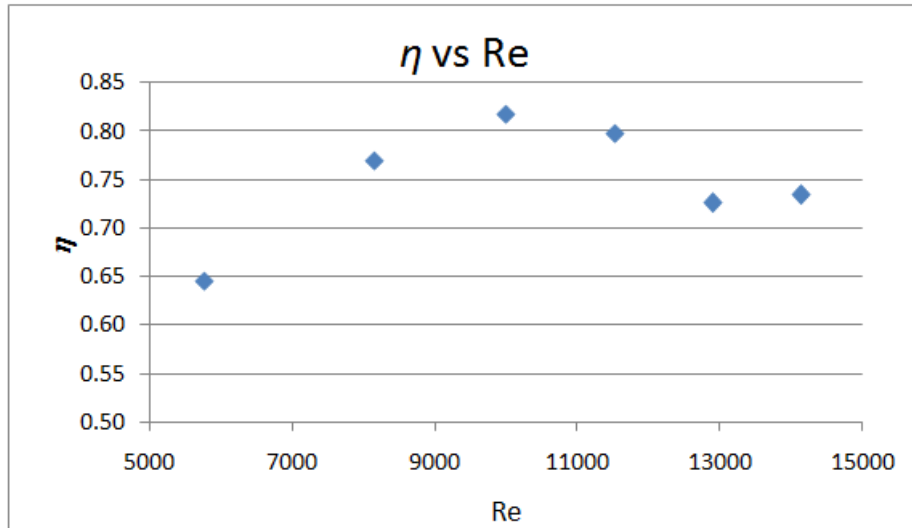Where, h_i = convective heat transfer coefficient with insert

Fig.5. Variation of Overall Enhancement ratio for different Reynolds number

B} Non metallic Acrylic Twisted Tape Insert

Similarly, Nusselt number and friction factor obtained are compared in order to validate the experimental setup.It is observed that, the experimental data are in good agreement with the correlation values, i.e., Dittus – Boelter Correlation and with Blasius correlation as shown in Fig.6a and 6b.Also in the case of Twisted tape insert the friction factor is high at lower Reynolds number and shows decreasing trend with the increase of Reynolds number as shown in the Fig 7. This is because of it's inverse proportion to square of velocity for a given pipe. Therefore, friction factor shows a decrease due to increasein velocity associated with increase in mass flow. Similarly from equation 15,the value of OER is above unityfor different flow rate of air and heater input and maximum value of OER is 1.58 as shown in Fig.8.Enhancement efficiency is found to increases with increase in Reynolds number. Maximum value of enhancement efficiency (equation 16) obtained is 1.87. Fig. 8 shows the variation of enhancement ratio with Reynolds number.
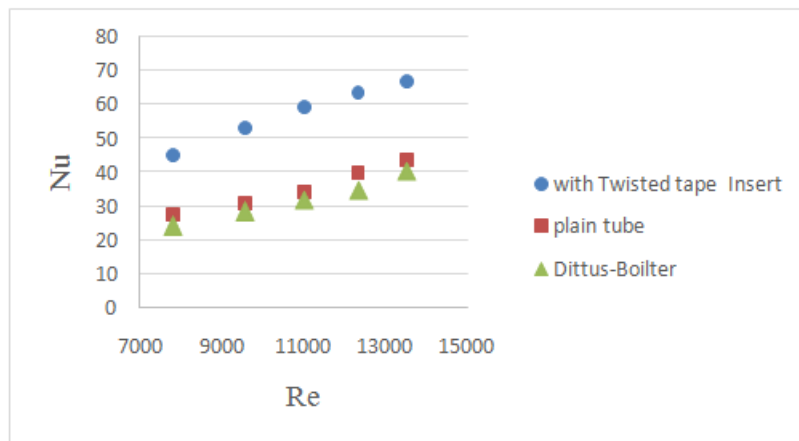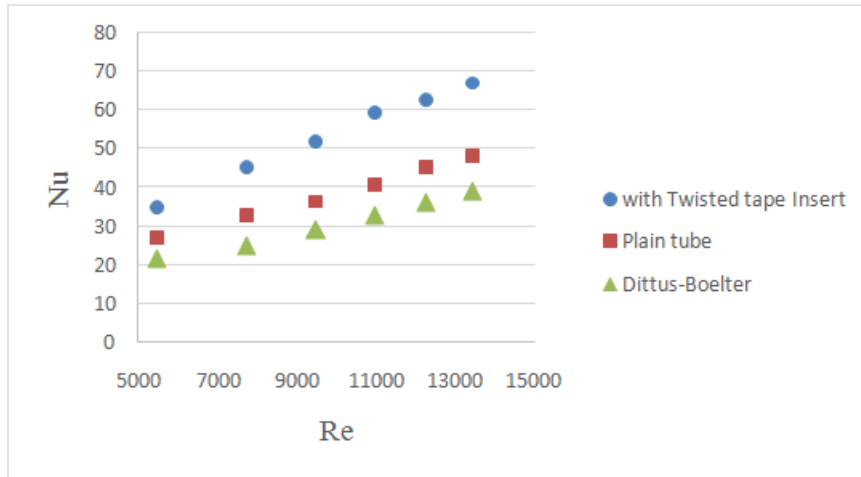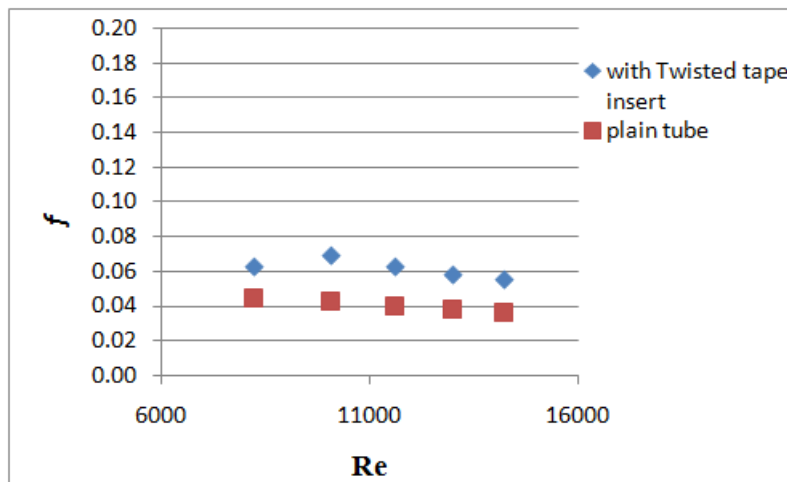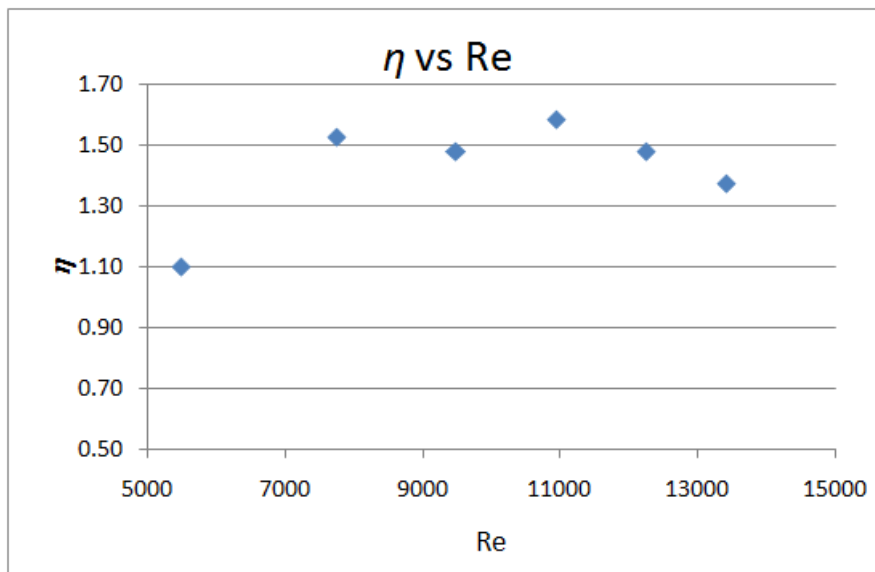

Fig.(6a) For heater input P = 50W

Fig.(6b) For heater input P = 75W
Fig.(6a, 6b) Comparison of Nusselt number with and without insert

Fig.7 Variation of friction factor with Reynolds number

Fig.8. Variation of Overall Enhancement ratio for different Reynolds number

VI. Conclusion

For plain tube, the experimental data is in good agreement with correlations value i.e. Dittus-Boelter and Petukhov correlation. Experimental data of the present work is reasonably agreed within 10% for Nusselt number from Dittus Boelter correlation and 13 % for friction factor from Blasiuscorrelation. For Differential Annular Type insert maximum increase in Nusselt number is 112% and Overall Enhancement ratio ranges from 0.54 to 1.12. The friction factor which is directly proportional to pressure drop is found to be in range of 0.5 to 0.6.For Nonmetallic Acrylic Twisted tape Insert maximum increase in Nusselt number is 187% and Overall Enhancement ratio ranges from 1.10 to 1.58. The friction factor which is directly proportional to pressure drop is found to be in range of 0.06 to 0.08.

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