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EXPERIMENTAL INVESTIGATION OF TUBE SIDE HEAT TRANSFER ENHANCEMENT USING WIRE COIL INSERT

S. Biswas* and B. Salam

Department of Mechanical Engineering, Chittagong University of Engineering and Technology, Chittagong-4349, Bangladesh

Abstract: Enhancement of heat transfer by inserting different types of turbulators in to the tube is one of the conventional passive augmentation methods. An experimental investigation was carried for measuring tube-side heat transfer coefficient, friction factor, heat transfer enhancement efficiency of air for turbulent flow in a circular tube fitted with wire coil insert. A copper tube having 27 mm inside diameter, 30 mm outside diameter and of which length of 285 mm was used as the test section. A stainless steel coil was used as an insert where the wire diameter (e) was 2.8 mm and coil diameter (d) was 24 mm. In the experiment, the dimensionless pitch (p_l/d_i) was considered to be 0.71 and the dimensionless wire diameter (e/d_i) was considered to be 0.11. A uniform heat flux condition was created by wrapping Nichrome wire around the test section and fiber glass over the wire. Outer surface temperatures of the tube were measured at 6 different points of the test section by K-type thermocouples. Two thermometers were used for measuring the bulk temperatures. Reynolds numbers were varied from 8317 to 17821 in the experiment with heat flux variation from 271 to 610 W/m^2 for smooth tube, and 284 to 929 W/m^2 for tube with insert. Nusselt numbers obtained from smooth tube were compared with Gnielinski [1] correlation and errors were found to be in the range of ±20% with r.m.s. value of 16.5%. At comparable Reynolds number, Nusselt numbers in tube with wire coil insert were enhanced by 1.5 to 2.3 times at the cost of increase of friction factors by 3 to 3.5 times compared to that of smooth tube. Heat transfer enhancement efficiencies were found to be in the range of 1.3 to 2.6 and increased with the increase of Reynolds number.

Keywords: *Heat transfer enhancement, wire coil insert, heat flux, Nusselt number*

NOMENCLATURE

A = Area of the heated region of tube (m^2)
C_p = Specific heat of air at constant pressure (J/kg.K)
d_i = Tube inner diameter (m)
d_o = Tube outer diameter (m)
e = wire diameter (mm)
f = Friction factor (-)
h = Heat transfer coefficient (W/m^2.K)
k = Thermal conductivity of air (W/m^2.K)
k_c = Thermal conductivity of tube material (W/m^2.K)
L = Effective tube length (m)
M = Mass flow rate of air (kg/s)
Nu_i = Experimental Nusselt number with insert (-)
Nu_p = Experimental Nusselt number for smooth tube (-)
Nu_{prd} = Predicted Nusselt number (-)
Nu_{th} = Theoretical Nusselt number from Gnielinski [1] correlation (-)
Pr = Prandtl number (-)
Q = Heat transfer rate (W)
q = Heat flux (W/m^2)
q_i = Heat flux for tube with insert (W/m^2)
q_p = Heat flux for smooth tube (W/m^2)
p_l = pitch (mm)
P_i = Inlet pressure (N/m^2)
P_o = Outlet pressure (N/m^2)
P_m =Pumping power (W)
ΔP = Pressure drop (N/m^2)
$\dot{Q}$ = Volume flow rate (m^3/s)
Re = Reynolds number (-)
T_b = Bulk temperature (°C)
$T_{b,i}$ = Bulk temperature with insert (°C)
$T_{b,p}$ = Bulk temperature for smooth tube (°C)

* **Corresponding author:** sbiswas@cuet.ac.bd

T_i = Air inlet temperature (°C)
T_o = Air outlet temperature (°C)
T_{wi} = Tube inner surface temperature (°C)
$T_{wi,\,i}$ = Tube inner surface temperature with insert (°C)
$T_{wi,\,p}$ = Tube inner surface temperature for smooth tube (°C)
T_{wo} = Tube outer surface temperature (°C)
$T_{wo,\,i}$ = Tube outer surface temperature with insert (°C)
V = Mean velocity (m/s)
ρ = Density of air (kg/m^3)
μ = Dynamic viscosity of air (kg/m.s)
η = Heat transfer enhancement efficiency (-)

1. INTRODUCTION

Heat transfer enhancement provides a simple and economical method of improving the thermal performance of heat exchangers. The use of such a technique results in a reduced surface area requirement and consequently a smaller heat exchanger and reduced equipment cost. The augmentation of the heat transfer by inserting different type turbulators [2,3] into the channels is the conventional passive enhancement method. The wire coil insert is one of the common heat transfer devices. These tube inserts enhance the convective heat transfer by interrupting the boundary layer development and raising the degree of turbulence or by increasing the heat transfer area or by generating the swirl flow. Garcia et al. [3] experimentally studied the thermohydraulic behavior of a round tube with helical-wire coil inserts in laminar, transition and turbulent regimes at different Prandtl numbers. In the investigation, they used water and water-propylene glycol mixtures (R3) as the test fluid at different temperatures. The authors tested six wire coils within a geometrical range of helical pitch 1.17 < p/d <2.68 and wire diameter 0.07 < e/d <0.10. They showed that within the transition region, by fitting the wire coils inside a smooth tube heat exchanger, the heat transfer rate increased up to 200%. P. Naphon [4]presented the heat transfer characteristics and pressure drop results of horizontal concentric tubes with coil-wire insert. They initiated that the coil-wire insert had significant effect on the enhancement of heat transfer especially on laminar flow region and showed that the heat transfer enhancement decreased with the increase of Reynolds number. An experimental investigation had been carried out by Behabadi et al. [5] to study the heat transfer augmentation and pressure drop by wire coil inserts at the time of heating the engine oil inside a horizontal tube. They developed two empirical correlations within an error band of ± 20%. Gunes et al. [6] experimentally investigated the heat transfer and pressure drop in a tube with coiled wire inserts placed separately from the tube wall in turbulent flow regime. Where they used three different pitch ratio of the wire coil insert. An experimental study in a coiled wire inserted tube in turbulent flow regime was carried out by Gunes et al. [8]. They used equilateral triangle cross sectioned wire coil inserts and showed that low Reynolds number regime were the best operating regime for wire coil insert. Bhuiya et al. [9] conducted an experimental study to investigate the heat transfer enhancement for turbulent flow in a tube by means of triple tape inserts. The present work focuses on understanding the heat transfer enhancement due to the wire coil insert. Data are compared with smooth tube heat transfer and friction factor values.

2. MATERIALS AND METHODS

The experimental apparatus, shown in Fig. 1 consists of a heated test section, air and power supply system, instrumentation to measure flow rate and temperature. The tube was a smooth circular tube made of copper (kc = 379 W/mK) having 27 mm inside diameter, 30 mm outside diameter and 305 mm long, of which length of 285 mm was used as the test section. The wire coil insert used in the experiment was fabricated by bending a 2.8 mm diameter stainless steel wire into a coil of which the dimensionless wire diameter (e/d_i) was 0.11 and dimensionless pitch (p_l/d_i) was 0.71.

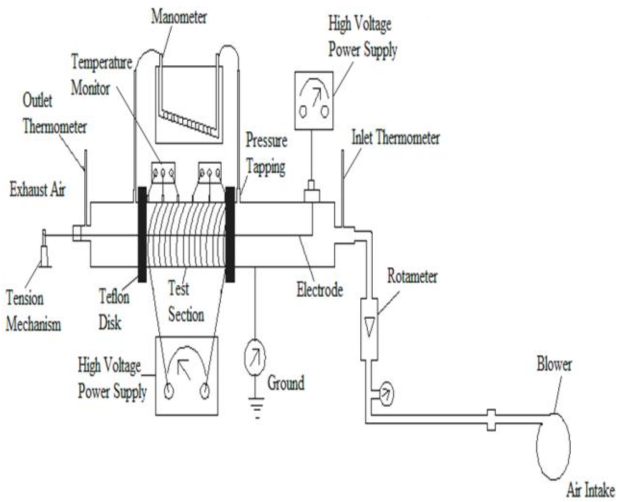

Fig. 1: Schematic diagram of experimental setup.

Fig. 2 shows the schematic diagram of a wire coil insert and Fig. 3 shows the photograph of the wire coil insert. To measure the surface temperature of the tube wall, six K- type thermocouples were used, which were placed 40 mm apart from each other and the distance of thermocouple from the entrance and exit part of the test section were about 25 mm. In order to prevent the heat flow in the longitudinal direction and to prevent leakage of air from the upstream and downstream, Teflon disks (density = 2150 kg/m^3, k = 0.35 W/mK) had been used as the joining of tube in such a manner so that upstream and downstream tubes were fully thermally insulated/ separated from the test section. The plain tube was covered with mica tape in two layers to electrically isolate the tube. A layer of glass fiber was put on the mica sheet. Nichrome wire (of resistance 31 ohm) used as an electric heater was spirally wound uniformly on the outer surface of the test section with spacing of approximately 2 mm. Then again mica sheet, glass fiber, teflon tape and heat insulating tape were sequentially put over the nichrome wire heater coil. The heating wire was connected to 220 Volt main. These protected the radial heat losses and also worked as an electrical insulation. The inlet and outlet air temperatures of test section were measured using thermometers inserted in the small holes made in the inlet and outlet of the test section and sealed to prevent any leakage. As the heat was added uniformly, the bulk temperature T_b of the fluid was assumed to vary linearly along the length of the test section. To break the boundary layer and to measure the mixing fluid temperature at the outlet section, a tee joint was adjusted at the end of the outlet section, of which one branch is closed by inserting a wooden block and at the outlet section the thermometer was placed at the other branch, 90° bend of the tee joint. And that part was open to the environment. So when flow came out through the downstream section, the boundary layers of the fluid flow broke at 90° bend of the tee joint and fully mixed with the wall of the tube. An inclined manometer was used at the test section to measure the air pressure drop. Two pressure taps were drilled at 365 mm intervals across the test section with the first tap position 40 cm downstream from the inlet of the test section and the other was placed at 40 cm upstream from the outlet of the test section. An air flow meter

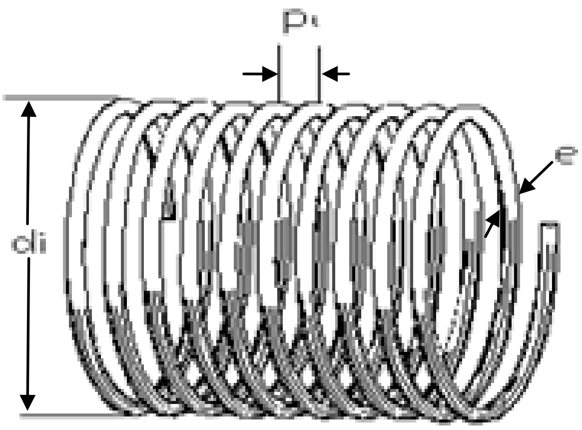

Fig. 2: Sketch of a helical wire coil fitted inside a smooth tube.

Fig. 3: Photograph of the wire coil insert.

was used to measure the volumetric flow rate of the experiment. Its measuring range was 100-500 NL/min. The air supply was done by a blower. The blower was both suction and discharge type. The speed range of the blower was 0-16000 rpm. The maximum flow range of the blower was 3 m^3/min. The blower speed was controlled by a regulator provided in the system. It was powered by 220 Volt AC supply. After switching on the heating power; sufficient time was given to attain the steady state condition. In each run, data were taken for air flow rate, inlet temperature, outlet temperature and tube outer surface temperatures, pressure drop wall of the tube.

3. DATA REDUCTION

The measured data were reduced using the following procedure:

The heat transfer rate by the heater to air was calculated by measuring heat added to the air. Heat added to air was calculated by,

$$Q = mC_p (T_o - T_i) \tag{1}$$

Heat transfer coefficient was calculated from,

$$h = Q / A (\mathrm{T_{wi}} - T_b) = q / (\mathrm{T_{wi}} - T_b) \tag{2}$$

where, $$A = \pi d_i L \tag{3}$$

The bulk temperature was obtained from the average of air inlet and outlet temperatures,

$$T_b = (T_i + T_o) / 2 \tag{4}$$

Tube inner surface temperature was calculated from one dimensional radial conduction equation,

$$\mathrm{T_{wi}} = \mathrm{T_{wo}} - Q.\frac{\ln(d_o/d_i)}{2\pi kL} \tag{5}$$

Tube outer surface temperature was calculated from the average of six local tube outer surface temperatures,

$$\mathrm{T_{wo}} = \frac{1}{6}\sum_{i=1}^{6} \mathrm{T_{wo,i}} \tag{6}$$

Theoretical Nusselt number was calculated from Gnielinski [1] correlation and ‘f’ from Petukhov [12] relation.

$$\mathrm{Nu_{th}} = \frac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7(f/8)^{1/2}(\mathrm{Pr}^{2/3}-1)} \tag{7}$$

where,

$$f = (0.79 \ln \mathrm{Re} - 1.64)^{-2} \tag{8}$$

$$3000 \le \mathrm{Re} \le 5\times10^6$$

$$\mathrm{Re} = \rho V d_i/\mu \tag{9}$$

$$\mathrm{Pr} = \mu C_p/k \tag{10}$$

$$\mathrm{Nu} = hd_i/k \tag{11}$$

Pressure drop across the test section, is given as:

$$\Delta P = P_i - P \tag{12}$$

Here fiction factor, f can be calculated from

$$f = \frac{\Delta P}{(L/d_i)(\rho V^2/2)} \tag{13}$$

Pumping power, P_m can be defined as

$$P_m = \frac{\Delta P \times m}{\rho} \tag{14}$$

To assess the performance of heat exchanging devices with insert it is necessary to evaluate heat transfer enhancement efficiency (η). This efficiency is calculated using constant pumping power [13]. For constant pumping power,

$$\dot{Q}_i \times \Delta p_i = \dot{Q}_p \times \Delta p_p \tag{15}$$

Now considering the assumptions from Webb and Kim [13] and from Eqs. (13) and (15)

$$f_i.\mathrm{Re}_i^3 = f_p.\mathrm{Re}_p^3 \tag{16}$$

At constant pumping power the heat transfer enhancement efficiency,

$$\eta = \left|\frac{\mathrm{Nu}_i}{\mathrm{Nu}_p}\right|_{PP} \tag{17}$$

4. RESULTS AND DISCUSSION

Before undertaking the experiments using the tube equipped with tube inserts, the heat transfer data were taken in a plain tube. In order to evaluate the validity of the set up, the plain tube experimental data were compared with Gnielinski [1] correlation. Fig. 4 shows that the experimental data of Nusselt number for smooth tube fall within the range of ± 20% of the Gnielinski [1] value with the r.m.s value of error 16.5%. Salam et al. [10] compared experimental Nusselt number for smooth tube with those calculated from Gnielinski [1] correlation and error were found within -24.7% and -5.6% with r.m.s value of error 20.3%. Salam et al. [11] found the error within -13% and + 18% with r.m.s value of error 12% when they compared the data with Dittus and Boelter [14].

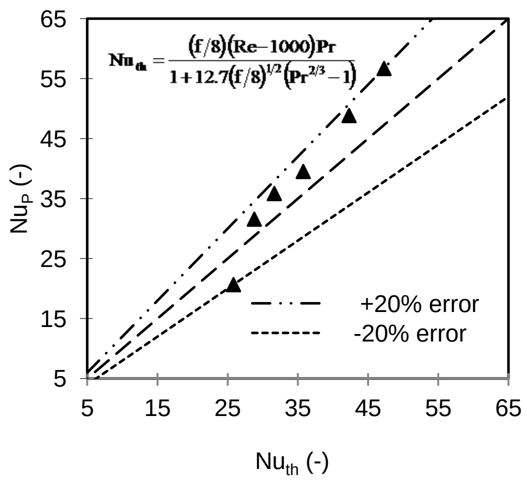

Fig. 4: Comparison between experimental and theoretical Nusselt number.

The developed correlations for the smooth tube are given as follow:

$$\mathrm{Nu_p} = 0.000449\ \mathrm{Re_p}^{1.1943}.\ \mathrm{Pr}^{-0.4} \tag{18}$$

$$f_p = 152\ \mathrm{Re_p}^{-0.94899} \tag{19}$$

Nusselt numbers for the plain tube and the tube with wire coil insert are shown in Fig. 5. It is seen that, Nusselt number increased with the increase of Reynolds number and wire coil insert gave higher values of Nusselt number than those for plain tube. Reynolds number was calculated based on inner diameter of the tube. For plain tube $\mathrm{Nu_p}$ increased from 20.7 to 56.7 with the increase of Reynolds number from 8317 to 17821.

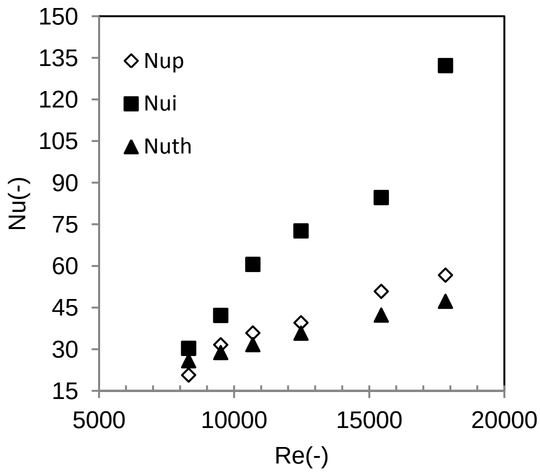

Fig. 5: The variation of Nusselt number with Reynolds number

For tube with insert Nusselt number Nui increased from 30.3 to 132.2 with the increase of Reynolds number from 8317 to 17821. Therefore, at comparable Reynolds numbers, Nusselt number for tube with wire coil insert varied from 1.5 to 2.3 folds in comparison to the plain tube. The characteristics of the Fig. 5 are similar to the experimental result of Garcia et al. [4]. Fig. 6 shows the heat flux variation for plain tube and tube with wire coil insert. It is seen that, heat fluxes increased with the increase of Reynolds number and insert gave higher heat flux than those for plain tube. The insert contributed to continuous interruption of the development of boundary layer as well as to generate the secondary flow and increased mixing resulting in higher heat transfer coefficients as compared to the tube without insert. In Fig. 7 the comparison of different temperature for insert has been shown. The temperature difference between wall and bulk fluid significantly decreased for tube with insert, Fig. 8. Fig. 8 also shows the variation in temperature difference (wall and bulk) for plain tube and the tube with wire coil insert. From the Fig. 7 and Fig. 8 it is clear that the plain tube has the highest temperature difference. Keeping the other parameters fixed, the smallest temperature difference indicates the highest heat transfer coefficient as heat transfer rate per unit area is assumed constant through the test section. The blockage of the flow due to the presence of the insert increases the flow velocity and improves the mixing of the flow (Bhuiya et al. [9]). So, the temperature difference for the insert was lower than that of the plain tube. In this case, considerable circumferential temperature difference was determined at the tube wall, which indicated that the flow was affected by the buoyancy forces (Garcia et al. [4]). Average of 52% enhancement of heat flux was observed for tube with insert than that of plain tube. The Fig. 9 shows that, the friction factor tends to decrease with the increasing Reynolds number. In this experimental study, it is observed that the friction factor for tube with wire coil insert varies from 3 to 3.5 times than that of the plain tube at comparable Reynolds numbers. The presence of small vortices behind the insert may be responsible for increase of the friction factor. For the tube with insert pumping power increased as the Nusselt number increases. The pumping power required for the tube with wire coil insert varies from 3 to 3.5 times compared to that of the plain tube.

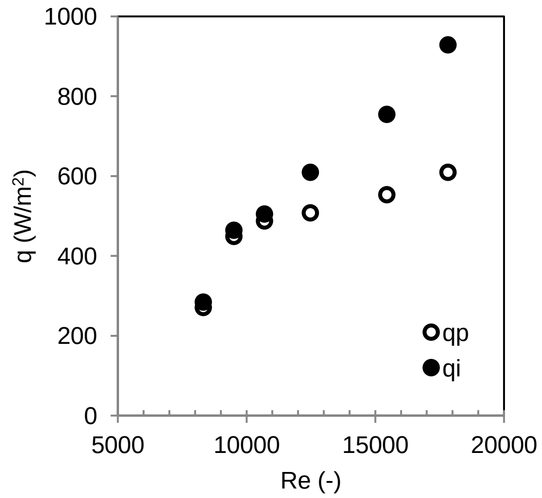

Fig. 6: The variation of heat flux with Reynolds number.

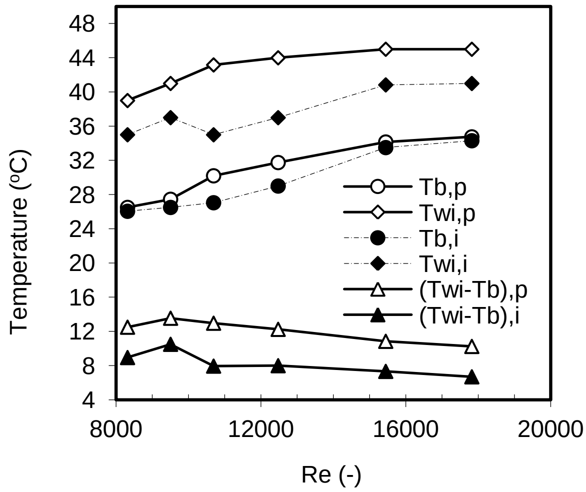

Fig. 7: The variation of temperature with Reynolds number.

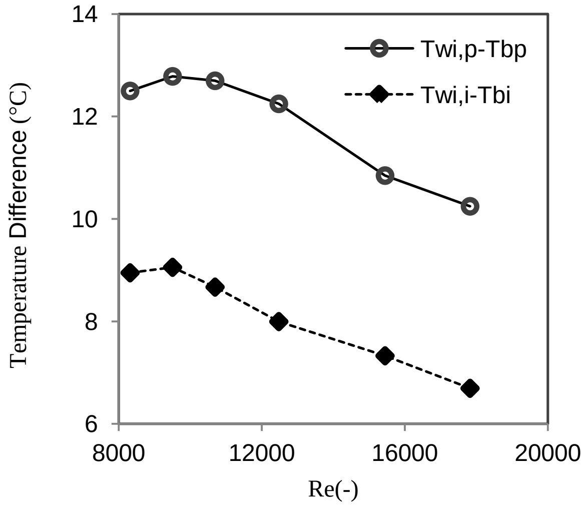

Fig. 8: Reynolds number vs. Temperature difference.

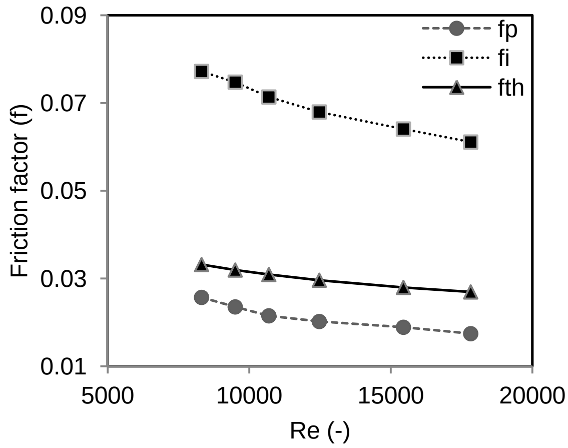

Fig. 9: The variation of friction factor with Reynolds Number.

4.1 Heat transfer enhancement efficiency (η):

The experimental values of Nusselt number and friction factor for tube fitted with insert are correlated as:

$$Nu_i = 0.273.\ Re_i^{\,0.728}.\ Pr^{\,0.55}.\ (e/d_i)^{0,\,1253}.\ (p_l/d_i)^{2.256} \tag{20}$$

$$f_i = 5.769.\ Re_i^{\,-0.158}.\ (e/d_i)^{0.9498}.\ (p_l/d_i)^{2.487} \tag{21}$$

The empirical correlations are fitted with the experimental data as shown in the Fig. 10. The errors between Nusselt numbers for experimental and predicted values for tube with insert were found to be in the range of -26% to 85% with r.m.s. value of

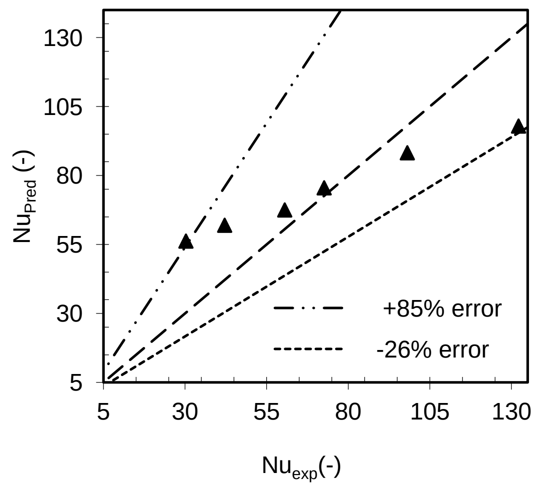

Fig. 10: Comparison between experimental and predicted Nusselt number.

41.8%. For friction factor these errors were found to be from -5% to 7% with r.m.s. value 4%.
From Eqs. (16), (19) and (21)

$$Re_p = 0.2029.\ Re_i^{1.386}\ (e/d_i)^{0.463}.\ (p_l/d_i)^{1.21257} \quad (22)$$

From Eqns. (18) and (22),

$$Nu_p = 6.68373\times10^{-5}\ Re_i^{1.65}\ Pr^{-0.4}\ (e/d_i)^{0.553}.\ (p_l/d_i)^{1.448} \quad (23)$$

From the Fig. 11 it is observed that the heat transfer enhancement efficiency increase provided by wire coil insert is remarkable and tends to decrease with the increase of Reynolds number. Heat transfer enhancement efficiencies were found to be in the range of 2.6 to 1.3 and decreased with the increase of Reynolds number. The result was as same as Gunes et al. [7,8] and Garcia et al. [3]. A considerable heat transfer enhancement efficiency 1.3-2.5 was presented by Gunes et al. [6] by using wire coil insert. In order to accurately determine the heat transfer enhancement efficiency by the wire coil insert the proposed correlation was of the form:

$$\eta = 4084.5437.Re_i^{-0.927}.Pr^{0.95}.\ (e/d_i)^{-0.43}.\ (p_l/d_i)^{-1.17506} \quad (24)$$

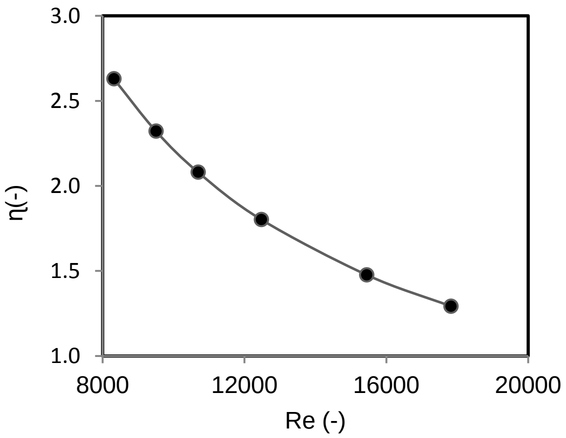

Fig. 11: Variation of heat transfer enhancement efficiency with Reynolds number for wire coil insert.

5. CONCLUSIONS

An experimental study was carried out to investigate the tube side heat transfer coefficient of air for turbulent flow in a circular tube fitted with wire coil insert. The experimental results show that the Nusselt number and heat flux increased with increasing Reynolds number. The Nusselt number for tube with wire coil insert varied from 1.5 to 2.3 folds in comparison to the plain tube. At equal Reynolds number the heat transfer coefficient for tube with wire coil insert increased from 1.5 to 2.3 times compare to that of plain tube. The friction factor for tubes with wire coil insert varied from 3 to 3.5 times than that of the plain tube at comparable Reynolds numbers and at constant pumping power, the heat transfer enhancement efficiencies were found to be in the range of 2.6 to 1.3 and decreased with the increase of Reynolds number.

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