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AUGMENTATION OF HEAT TRANSFER USING COILED POROUS TWISTED TAPE INSERT

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Abstract-*Researchers are working on augmentation of heat transfer to find out an effective insert. The present analysis was carried out for enhancing tube side heat transfer performance with a coiled porous twisted tape insert of aluminum strip. The test section consisted of a circular tube of copper having 26.6 mm inside diameter, 470 mm length and air was used as the working fluid. Length of the coiled twisted tape was 450 mm, thickness was 1.5 mm, height was 20 mm and wire coil diameter was 20.1 mm. Four K- type thermocouples and two thermometers were used in the test section to measure temperatures. In this investigation, as Reynolds numbers ranging from 9317~18193, the flow was turbulent. Nusselt number varied from 26.95~50.85 for smooth tube and errors found +2.53%~-14.85% range comparing with* Gnielinski [1] equation. *Heat flux increased 1.54~1.83 times and performance found in the range of 1.78~2.26 after using insert.*

Keywords: Twisted tape insert, Heat transfer rate, Friction factor, Nusselt Number.

1. INTRODUCTION

Enhancement of heat transfer is used in many engineering applications such as heat exchanger, air conditioning, chemical reactor, process industries refrigeration systems etc. The goal of heat transfer augmentation is to reduce the size and cost of the heat exchanger. In order to produce more efficient heat transfer equipment, an increasing number of augmented surface are being produced commercially. Several options are available for enhancing heat transfer associated with internal flows. Enhancement may be achieved by increasing the convection coefficient and/or by increasing the convection surface area. The development of high performed thermal system has simulated interest in method to improve heat transfer rate. The study of improvement of heat transfer performance is referred to as heat transfer augmentation, enhancement & intensification. Heat transfer enhancement can improve the heat exchanger efficiency. In the turbulent flow, the fluid properties such as instantaneous velocity, temperature and pressure are subjected to fluctuations. This fluctuations increase the heat transfer rate and resistance to fluid flow. Augmentation of convective heat transfer in inner tube flows with twisted tape inserts is a well-acclaimed technique working in industrial purposes.

Burgles et *al.*[2] complied the variable literature on convection heat transfer. From the study, the augmentation techniques wereclassified into two categories, (i) active and (ii) passive. The active augmentation techniques had been studied widely requires the addition of external power to bring about the desires flow modification. Passive techniques are generally used for surface and geometrical modifications to the flow channel by incorporating inserts or additional devices. Passive augmentation examples are surface roughness displaced promoter and vortex generators. Surface roughness introduced through knurling, threading or forming of repeated ribs promotes enhancement through the disturbance of the viscous layer near to the surface. The existing flow mechanism was disturbed and heat transfer performance was improved, usually with an increase in the friction and pressure drop.Hsieh and Huang[3] studied an experimental investigation for heat transfer and pressure drop in horizontal tubes with/without longitudinal inserts. From the research,it was found that heat transfer enhancement as compared to a conventional bare tube at the same Reynolds number to be a factor of 16 at laminar flow, whereas friction factor increased of only 4.5. A new method was proposed by Sarmaet *al.*[4] to calculate heat transfer coefficients with twisted tape inserts in a tube in which the wall shear and the temperature gradients were properly reformed through friction coefficient correlation leading to heat transfer intensification from the tube wall. Hsieh et *al.*[5] conducted experimental studies on heat transfer and flow characteristics for turbulent flow of air in a horizontal circular tube with strip type inserts of longitudinal and crossed strip. They stated that friction factor rise due to inclusion of inserts was typically between 1.1 and 1.5 from low Reynolds number of 6500 to high Reynolds number of 19500 with respect to bare tube. Ahamed et *al*.[6] carried out an investigational analysis of heat transfer performance of porous twisted tape insert in a circular tube. Reynolds numbers were varied in the range of1.3 x 10^4 to 5.2 x 10^4 for tube with porous twisted tape insert, the average heat transfer coefficient was noticed2.60 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 smooth tube.Fahed and Chakroun[7] investigated the effect of tube-tape clearance on heat transfer under fully developed turbulent flow conditions in a horizontal isothermal tube. Bharatdwaj et*al.*[8]experimentally determined pressure drop and heat transfer characteristics of flow of water in a 75-start spirally grooved tube with twisted tape insert. Laminar to fully turbulent ranges of Reynolds numbers had been considered in the existing study. The grooves were clockwise with respect to the direction of flow. The heat transfer increased due to spiral grooves was further augmented by inserting twisted tapes having twist ratios 10.15, 7.95 and 3.4, while compared to smooth tube.Naphon[9] considered the heat transfer characteristics and the pressure drop in the horizontal double pipes with twisted tape insert. The results obtained from the tube with twisted insert were compared with those without twisted tape. Non-isothermal correlations based on the data gathered during this work for projecting the heat transfer coefficient and friction factor of the horizontal pipe with twisted taped insert, which was recommended. Project on heat transfer and friction factor characteristics of circular tube fitted with plain twisted tapes (PTT) and U–cut twisted tapes (UTT) with twist ratios 2.0, 4. 4 and 6.0 were considered by Murugesanet *al.*[10]. A significant increase in Nusselt number and friction factor were observed. Ray and Date[11] derived a relationship for the friction factor and Nusselt number for a square duct from the projected data. They matched the correlation for the friction factor with experimental data, which were found to be in reasonably good agreement with each other, for both laminar and turbulent flows. Sarada et *al.*[12] made an analysis with varying width twisted tape inserts ranging from 10 mm to 26 mm. The Reynolds number varied from 6000 to 13500. It was found that the increment of heat transfer with twisted tape inserts as compared to plain tube varied from 36~48% for width of 26 mm and 33~39% for width of 22 mm inserts. Kumar et *al*.[13] investigated the thermo hydraulic performance of twisted tape inserts in a large hydraulic diameter annulus. The thermo hydraulic performance in laminar flow with a twisted tape was better than the wire coil for the same helix angle and thickness ratio. Biswas and Salam[14] conducted an investigational study on heat transfer enhancement with a wire coil insert. In the experiment, Reynolds numbers were varied from 8317 to 17821 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 were compared with Gnielinski [1] correlation for smooth tube 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. Wang and Sunden [15] reported correlations for ethyl glycol and polybutene (Pr. No. 10000-70000).The research concluded by considering the overall enhancement ratio, twisted tape was effective for small Prandtl number fluids and wire coil was effective for high Prandtl number fluids. Hong et *al*. [16] considered Pressure drop and compound heat transfer characteristics of a converging-diverging tube with evenly spaced twisted-tapes (CD-T tube) experimentally. The results showed that the twisted-tape with twist ratio 4.72 and rotation angle 180° has the best performance among the four types of twisted-tapes presented in this paper. At Reynolds number ranging from 3400 to 20000, when space ratio 48.6, the heat transfer efficiency index, which increases as the Reynolds number increases, is 0.85 to 1.21 and 1.07 to 1.15 compared to that of a smooth circular tube and a CD tube without twisted-tape inserts, respectively. Afzal [17] studied the investigational research on tube side heat transfer co-efficient using porous cross strip insert. Reynolds numbers varied in the range of 1131 to 3643.Heat transfer rate, varied from 1.9 to 11.59.76W(for plain tube) and heat transfer rate varied from 2.96 to 24.405 W(with insert). Heat transfer co-efficient varied in the range of 5 to 13.52W/m^2.ºC (without insert) and 8.2 to 39.34 W/m^2.ºC (with insert). Nusselt numbers were diverse ranging in 4.74 to 12.76 (for plain tube) and 7.78 to 36.75(with insert).

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

The research work aims to develop the percentage of heat transfer rate by using insert. To determine the heat transfer co-efficient and pressure drop using insert and without insert and compare these results to notice the relative effectiveness. And another one is to define the heat transfer performance of the insert.

2. EXPERIMENTAL DETAILS

Convection is the transfer of potential energy like heat, by means of currents within a fluid. To enhance heat transfer rate, coiled porous twisted tape insert was used. This new investigation was carried out for measuring tube side heat transfer coefficient of air. The test section was a smooth circular tube made of copper having 26.6 mm inside diameter, 30 mm outside diameter and 475 mm large, of which 470 mm was used as the test section. Four K-type thermocouple were used in test section 10 cm apart from each other and started from 8.5 cm distance from entrance and finished before 8.5 cm from exit of heater length. In order to prevent leakage, Teflon tape was used for the joining of the tube and after that M-seal was used. Then the tube was wrapped at first with mica tape before wrapping with nichrome wire spirally wounded uniformly around the

tube. A constant heat flux condition was maintained by wrapping nichrome wire around the test section and fiber glass insulation over the wire. This was used to heat the test section. Outer surface temperature of the tube was measured at four points of the test section maintaining equal distance from one point to another point by K-type thermocouples. Two thermometers were used at the inlet and outlet section of the tube for measuring the bulk temperatures. Pressure drop was measured at inner side of the test section by using manometer. Open loop system of air was supplied. The rate of flow was measured with the help of orifice meter in the travelling path of inlet air. Two types of temperature were measured during the experiment. Air inlet temperature and air outlet temperature was measured. Then data was taken by K-type thermocouple in the test section to calculate the temperature inside the copper tube.
The coiled insert was made of galvanized iron wire displays in fig 2 and porous twisted tape was made of aluminum is shown in fig 1.Figure 3 shows the practical appearance of the coiled insert.

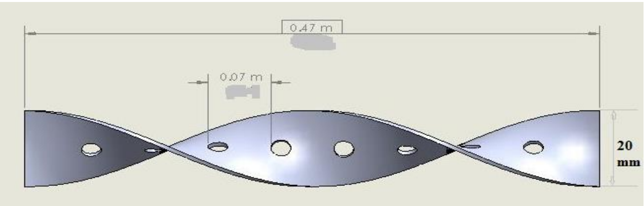

Fig.1: Designed of a porous twisted tape.

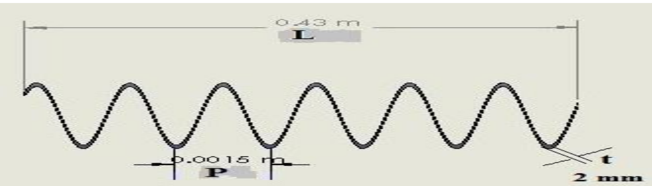

Fig.2: Designed of a wire coil

Fig.3: Photograph of the combination of porous twisted tape with wire coil

Figure 4 is the schematic diagram of experimental setup for heat transfer analysis. It consists of heating section, air blower, voltage regulator, thermocouple, manometer and thermometer. An air blower was used to supply the air through the tube. Air supply system was used as open loop system. The flow rate of the air was controlled with the help of a voltage regulator. The outlet section was open and air was entered to the test section through an orifice meter and a control valve. In the test section, air was heated by nicrom wire and surface temperature was measured by k-type thermocouples. Inlet and outlet temperature was measured by thermometer. Pressure drop was measured by a manometer. In each run of water into the tube, data were taken for water flow rate, temperatures and pressure drop readings in the prospective way.

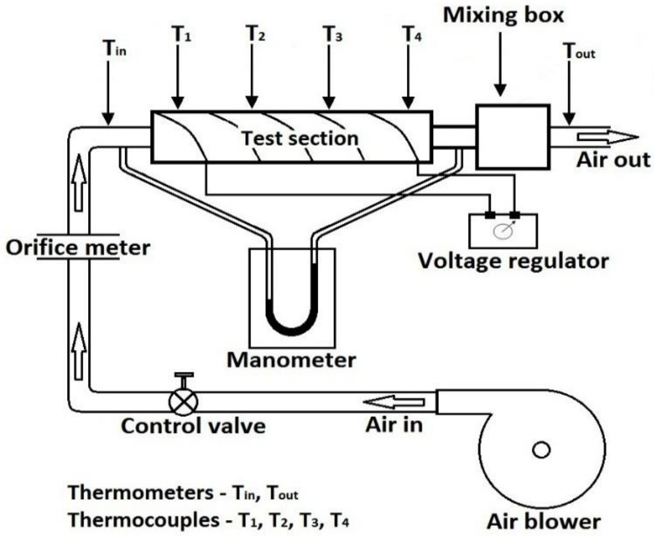

Fig.4: Experimental setup layouts

3. FORMULATIONS

The formulas used in this paper are important in the field of heat transfer. From the data originated in the project, aimed values were calculated using the following equations.

Outer surface area, $A_0 = \pi d_0 L$ (3.1)
Where, d_o is outer surface diameter, L is length.

Inner surface area, $A_s = \pi d_i L$ (3.2)
Where, d_i is outer surface diameter.

Heat transfer rate, $Q = mc_p(T_{out} - T_{in})$ (3.3)
Where, T_{out} and T_{in} are outlet and inlet temperature.

Cross sectional area, $A_x = \frac{\pi d_i^2}{4}$ (3.4)

Velocity, $v = \frac{m}{A_x}$; (3.5)
Where, m is the flow rate.

Reynolds Number, $Re_D = \frac{\rho V d_i}{\mu}$ (3.6)

Nusselt number, $Nu_{exp} = \frac{hd_i}{k}$(3.7)

Prandtl number, $Pr = \frac{\mu c_p}{k}$; (3.8)
μ and k at bulk temperature.

Convective heat transfer coefficient, $h = \frac{Q}{A(T_{wi} - T_b)}$(3.9)

Gnielinski equation, $Nu_D = \frac{\left(\frac{f}{8}\right)(Re_D - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}$; (3.10)
Where, $f = (0.790 \ln Re_D - 1.64)^{-2}$

The experimental friction co-efficient, $f_{exp} = \frac{2\Delta P d_i}{\rho L u_m^2}$; (3.11)
Where, p is the pressure drop across tapping.

Mean velocity, $u_m = \frac{Q}{A_f}$, (3.12)

Flow area, $A_f = \frac{\pi}{4} d_i^2$ (3.13)

Bulk temperature, $T_b = \frac{T_{in}+T_{out}}{2}$ (3.14)

Outer surface temperature, $T_{w0} = \sum_{i=1}^{5} T_{w0,i} / 5$ (3.15)

Inner surface temperature, $T_{wi} = T_{w0} - Q\frac{\ln(d_0-d_i)}{2\pi k_w L}$(3.16)

% of error = $\left(\frac{Nu_{exp}-Nu_D}{Nu_D}\right)100$(3.17)

Heat flux, $q = \frac{Q}{A_s}$ (3.18)

Pressure difference, $\Delta p = \Delta h \times \rho \times g \times 13.6$(3.19)

Efficiency $\eta = \left(\frac{Nu_{exp}}{Nu_D}\right)/\left(\left(\frac{f_{exp}}{f}\right)^{\frac{1}{3}}\right)$(3.20)

4. RESULTS ANALYSIS

It was very important to know the range of experimental error percentage, as to make sure that the analysis studied was standard. So, Nusselt numbers were calculated first by using Eq. (3.7) and compared with Gnielinski equation, Eq. (3.10). The comparison defined that the results within the acceptable error range of ±20%.The range found from+2.53 to -14.85and R. M. S. of error was +10.372.

Figure 5 represents the Nusselt numbers variation at different Reynolds numbers. Reynolds number was computed by Eq. (3.6). It is displayed that Nusselt number increase gradually with the increase of Reynolds number and relatively extreme Nusselt number obtained after using insert for 18192 value of Reynolds number. For a wide range of Reynolds number from 9317 to 18192, Nusselt number found for smooth tube ranging from 26.95 to 5.85

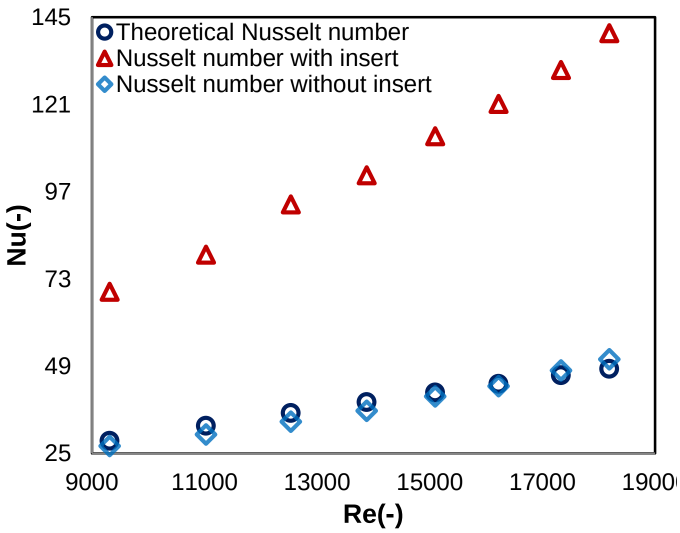

Fig.5: Variation of Nusselt number at different Reynolds number.

and for tube with insert ranging from 69.36 to 140.56. After comparing these results, it was found that the values of Nusselt number for tube with insert were increased 2.57 to 2.79 times from the smooth tube after using the insert.

The relation of heat flux and the Reynolds number is described by fig 6. Heat transfer rate was calculated by Eq. (3.3) and heat flux was computed by Eq. (3.18). With the increment of Reynolds number, heat transfer rate was also

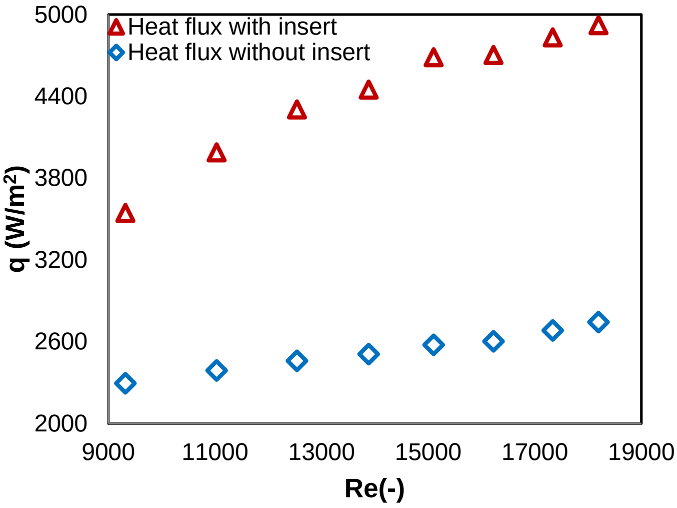

Fig.6: Variation of the heat flux at different Reynolds number.

increased. And for a wide range of Reynolds number heat flux varied from 2294 to 2743 W/m^2 for smooth tube and 3544 to 4922 W/m^2 for tube with insert. So, it was clear that the values for tube with insert were increased by 1.54 to 1.82 times comparing with smooth tube values.

Convective heat transfer coefficient and Reynolds number's relation was very important for the current experiment. Convective heat transfer coefficient was calculated by Eq. (3.9). When Reynolds number increased, the heat transfer coefficient was also increased significantly established in fig 7. The values of heat transfer coefficient for smooth tube increased from 28 to 51 kg/m^2 with Reynolds number ranging from 9317 to 18192 respectively and heat transfer coefficient for tube with insert varied from

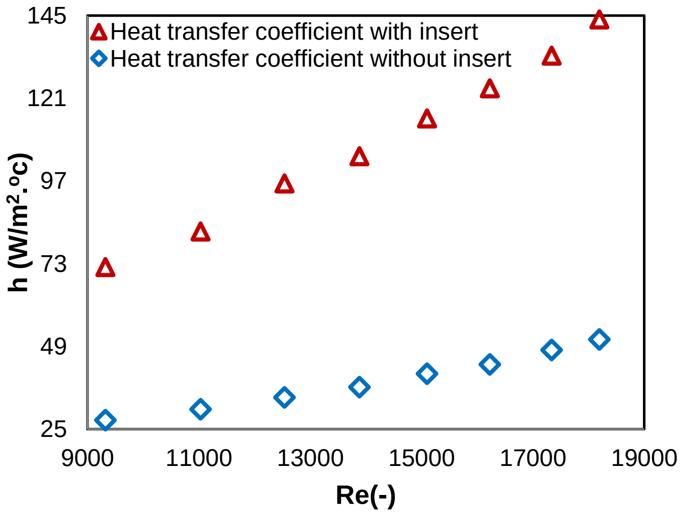

Fig.7: Variation of the heat transfer coefficient at different Reynolds number.

72 to 144 kg/m^2 range and a sharp increment was observed. Comparing the values of convection heat transfer coefficient for tube with insert and smooth tube, 2.61 to 2.83 times better result found for tube with coiled porous twisted tape insert.

Increment of flow rate made an effect to decrease in friction factor. Using Eq. (3.12), friction factor was computed and for smooth tube it varied in 0.04329 to 0.05828 ranges. After using coiled porous twisted tape insert, the range increased and changed from 0.05972 to 0.08741 ranges. When compared the data of friction factor

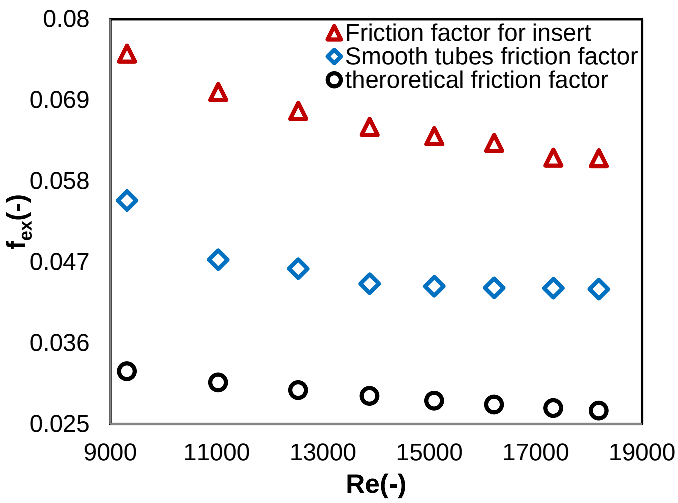

Fig.8: Variation of the friction factor at different Reynolds number.

with Reynolds number, it was found that with the increment of Reynolds number, friction factor decreased continuously plotted in fig 8. But, friction factor increases 1.36 to 1.48 times for using insert than the smooth tube, which causes power loss.

Figure 9 described that increasing the Reynolds number caused to increase convective heat transfer performance. For Reynolds number of 9317, heat transfer enhancement

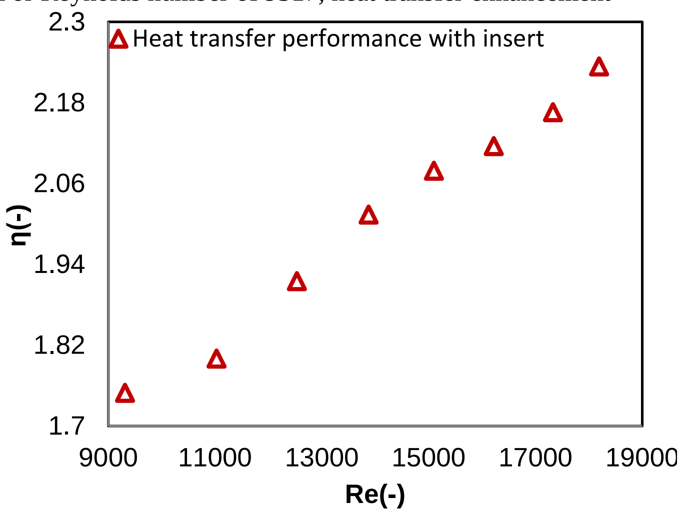

Fig.9: Variation of heat transfer enhancement efficiency at different Reynolds number.

efficiency was found 1.75 and for 18192, efficiency was increased to 2.23. The prime look of this experiment was on convective heat transfer enhancement efficiency of air which was calculated by Eq. (3.20).

5. CONCLUSION

The main purpose of the project was to augment the heat transfer. The application of single phase enhancement techniques is evaluated for tube side. Several passive techniques were identified as possibilities for tube enhancement. They did not rely on external power or activation. Therefore, these did not have any additional power costs. This study mainly focused on Reynolds number, Nusselt number, and especially on heat transfer coefficient. The present experimental study of tube side heat transfer enhancement with coiled porous twisted tape insert had made the simultaneous effects on Re, Nu and h. The Nusselt number increased 2.57 to 2.79 times in coiled porous twisted tape insert compared to smooth tube. The heat transfer coefficient increased2.61 to 2.83 times and the pressure drop increased1.36 to 1.48 times in coiled porous twisted tape insert. The most important factor, that the heat transfer performance was enhanced gradually with the increasing of velocity. The heat transfer enhancement efficiency was increased between 1.75 to 2.23.ranges. From the above discussions it is clear that heat transfer is increased using inserts but pressure drop also increased. So we have to optimize the results such a way where heat transfer is maximized with a minimum pressure drop. It is hope that this study will help different persons having interest in heat transfer related region.

Some limitations were occurred on this experiment, which are described here. Using porous twisted tape insert on the tube may cause high pressure drop. This will lead to high cost requirement. Another one is that, higher pressure may cause to failure of the tube strength and properties.

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7. NOMENCLATURE

Symbol	Meaning	Unit
A_o	Outer surface area of tube	(m^2)
A_s	Inner surface area of tube	(m^2)
A_x	Cross sectional area of tube	(m^2)
C_p	Specific heat	(J/kg.K)
d_o	Tube outer diameter	(m)
d_i	Tube inner diameter	(m)
d_c	Coil diameter	(m)
L	Tube length	(m)
t	Thickness	(m)
f	Friction factor	Dimensionless
q	Heat flux	(W/m^2)
h	Heat transfer coefficient	($W/m^2.K$)
k_w	Thermal conductivity	(W/m.K)
Q	Heat transfer rate	(W)
m	Mass flow rate	(kg/s)
Nu	Nusselt number	Dimensionless
ΔP	Pressure drop	Dimensionless
Pr	Prandtl number	Dimensionless
Re	Reynolds number	Dimensionless
T_o	Hot water temperature	(°C)
T_i	Cold water temperature	(°C)
T_1	Temperature of thermocouple 1	(°C)
T_2	Temperature of thermocouple 2	(°C)
T_3	Temperature of thermocouple 3	(°C)
T_4	Temperature of thermocouple 4	(°C)
T_b	Bulk temperature	(°C)
T_{wo}	Outer surface temperature	(°C)
T_{wi}	Inner surface temperature	(°C)
u_m	Mean velocity	(m/s)
V	Velocity	(m/s)
w	Twisted tape width	(m)
y	Twist ratio	Dimensionless
ρ	Density	(kg/m^3)
μ	Dynamic viscosity	(kg/m-s)
η	Thermal enhancement factor	Dimensionless