# Experimental Study on Enhancement Of Thermal Performance Of Wire Wound Tube In Tube Helical Coil Heat Exchanger

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## ABSTRACT

**Present work experimentally investigates the hydrodynamic and heat transfer analysis of three different geometries of the tube in tube helical coil. This study was conducted over a range of Reynolds numbers from 2500 to 6700 using cold water in annulus side. The experiments were carried out in counter flow configuration with hot water in tube side and cold water in annulus side. Each patterned coils were fabricated by bending 3.5 m straight mild steel tube having 10 mm ID and 12 mm OD, and stainless steel tube having 23 mm ID and 25 mm OD in four active helical turns having zero pitch with coil diameter 270mm. The mild steel wires of 1.5 mm were wound on outer side of inner tube having 6 mm and 10 mm pitch distance. The annulus side Nusselt number and friction factor were determined. The pressure drop and overall heat transfer coefficient is calculated at annulus side for different rate conditions. The results show that the 6 mm wire wound tube in tube helical coil have more overall heat transfer coefficient than that of 10 mm and plain tube helical coil.**

***Keywords -*** Tube in tube helical coil, wire wound, Nusselt number, Reynolds number,

## I. INTRODUCTION

The thermal performance of heat exchanger can be improved by two techniques. One is passive technique which is not required external power source and another is active technique which is required external power source. Tube in tube helical coil heat exchanger (TTHC) and coil wire inserts it has a passive heat transfer enhancement techniques and are most widely used tubes in several heat transfer application, for example heat recovery system, air conditioning and refrigeration system, chemical reactors etc.

The curvature of helical tube produces secondary flow patterns perpendicular to the main axial flow direction. Typically, fluid in the core of the tube moves towards the outer wall, then returns to the inner portion of tube. The secondary flow enhances heat transfer rates as it reduces the temperature gradient across the cross section of the tube.

Fig.1 shows a sketch of a wire coil inserted in close contact with the inner tube wall, where p stands for helical pitch, e for the wire-diameter and d is the tube inner diameter [12]. The tubeside flow pattern is modified by the presence of a helically coiled wire as follows:

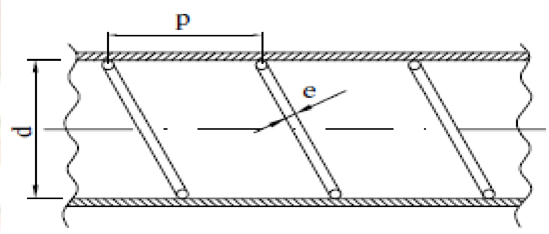


Figure1. Sketch of a helical-wire-coil fitted inside a smooth tube [1].

- If the wire coil acts as a swirl flow generator, a helical flow at the periphery is produced. This rotating flow is superimposed upon the axially directed central core flow and causes centrifugal forces. In most of liquids, where density decreases with temperature, centrifugal forces produce a movement of the heated fluid from the boundary layer towards the tube axis, which produces a heat transfer augmentation [1].
- If the wire coil acts as a turbulence promoter, flow turbulence level is increased by a separation and reattachment mechanism. Besides, whenever wire coils are in contact with the tube wall, they act as roughness elements and disturb the existing laminar sub layer [1].

Depending on flow conditions and wire coil geometry, the heat transfer rate will increase through one or both of the mechanisms mentioned earlier [1].

Alberto Garcıa et. al [1] present experimental study on three wire coils of different pitch inserted in a smooth tube in laminar and transition regimes. Isothermal pressure drop tests and heat transfer experiments under uniform heat flux conditions have been carried out. The friction factor increases lie between 5% and 40% in the fully laminar region. The transition from laminar flow to turbulent flow is continuous, without the instabilities and the pressure drop fluctuations that a smooth tube presents. For Reynolds numbers between 200 and 1000, wire coils remarkably increase heat transfer. At Reynolds numbers above Re =1000–1300, transition from laminar to turbulent flow takes place. At Reynolds number around 1000, wire inserts increase the heat transfer coefficient up to eight times with respect to the smooth tube.

Prabhanjan et.al [2] studied the relative advantage of using a helical coiled heat exchanger over a straight tube heat exchanger for heating liquids. They reported that the particular difference in the study was the boundary conditions for the helical coil, and results showed that the heat transfer coefficient was affected by the geometry of the heat exchanger and the temperature of the water affects both heat exchangers. It was also reported that the helical coil heat exchanger increased the heat transfer coefficient when compared to a similarly dimensioned straight tube heat exchanger.

Timothy J. Rennie et. al [3] performed experiment on two different sized double pipe helical coil heat exchanger for both parallel flow and counter flow configuration. They observed that there were a small difference between overall heat transfer coefficient for parallel and counter flow configuration but the heat transfer were much larger in counter flow configuration due to larger average temperature difference between the two fluids.

Vimal Kumar et. al [4] numerically modeled Tube in Tube Helical Coil (TTHC) Heat Exchanger for fluid flow and heat transfer characteristics at different fluid flow rates in inner as well as outer tube. New empirical correlation was developed for hydrodynamic and heat transfer prediction in the outer tube of Tube in Tube Helical Coil (TTHC) Heat Exchanger. It was observed that when the inner coil tube flow rates increases then the overall heat transfer coefficient is increases at constant wall temperature at that time the overall heat transfer coefficient were observed for different flow rates in the annulus region for a constant flow rates in inner coiled tube. it was also observed that while increasing the operating pressure in the inner tube , the result is rise in overall heat transfer coefficient . Also the heat transfer in inner and outer tube of Tube in Tube Helical Coil (TTHC) Heat Exchanger was higher.

Pioson Naphon et. al [5] had carried out test on experimental setup of shell and tube helical heat exchanger with wound iron wire. That experimental result show that heat transfer rate increases with increase in Reynolds Number. Also it shows that at the same hot water Reynolds Number, due to higher mixing of hot water at the boundary layer, the heat transfer rates obtained with the tube with coil wire inserted are higher than those without coil wire inserted. In addition, heat transfer rates at H=3.18 are higher than those at H=5.08.Same above mentioned result is for heat transfer coefficient also. Friction factor decreases with hot water Reynolds Number. As expected, the friction factor obtained from the tube with coil wire insert is significantly higher than that without coil wire insert.

Vimal Kumar et. al [6] studied on Pressure drop and heat transfer study in tube-in-tube helical heat exchanger It was observed that the overall heat transfer coefficient and friction factor increases with increase in the inner-coiled tube Dean number for a constant flow rate in the annulus region.

In the present work the performance of a tube in tube helical coil heat exchanger with externally wire wound on inner tube for a water–water countercurrent flow system is studied experimentally. We were studied the effect of various flow rate of water on heat transfer and hydrodynamics in the tube as well as in the annulus. In the present work four turns of coils with three different geometries were considered and also baffles introduce in the annulus area of the coil-in-coil heat exchanger. These features study are being reported first time, which is not considered in the previous literature. The experimental results of three geometries were compared with each other.

## II. EXPERIMENTAL SETUP

The experimental set up (Fig. 2) of tube in tube helical coil with externally wire wound on inner tube heat exchanger was reported and this topic presents the detailed information of apparatus, working procedure, operating parameters and geometries of helical coils.

The heat exchanger was constructed from mild steel and SS tubing. The outer tube of the heat exchanger had an outer diameter of 25 mm; The inner tube had an outer diameter of 12 mm. The end connections were constructed from standard mild steel tees and reducers. Each coil had a radius of curvature (measured from the center of the inner tube) of 135 mm. Washers were provided at two ends of the coil, which then held the inner tube in place. One baffle were placed in between the inner tube and outer tube after two turns of the outer coil for place the inner tube eccentric and to increasing the turbulence of annulus flow water.

Helical wire coils of 1.5 mm diameter fitted outside of an inner tube at different pitch have been experimentally studied the characterize their thermohydraulic behavior in laminar, transition and turbulent flow. Two wire coils of same diameter were wound on two helical tubes by taking the pitch distance 6 mm and 10 mm.

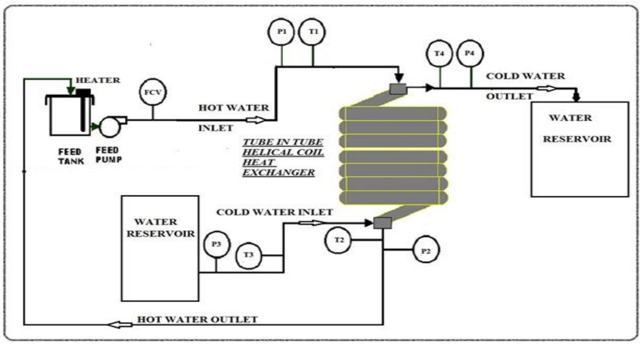


Figure 2 Experimental setup of tube in tube helical coil heat exchanger

***Experimental Apparatus***

It consist of cold water tank, hot water tank, temperature sensors, end connections, flow control valve, temperature display, pressure gauge, thermostatic water heater, tube in tube helical coil at different pitch of wire wound. The two FCV were used to controlled tube and annulus side mass flow rate. Temperature sensors (Thermocouple) were inserted into mild steel end connectors to measure inlet and outlet temperatures of both the fluids. Temperature data was recorded using data acquisition/switch unit. The end connectors brazed at inlet-outlet of coiled tube and threaded portion of connector fixed on treaded 'T' connector. The four pressure gauges (Range-0 to 4.2 Kg/cm$^2$) were used for measure the pressure drop at annulus cold water and inlet side hot water of inner tube.

## III. EXPERIMENTAL PROCEDURE

Experiments were conducted under steady state conditions with hot water (inner-coiled tube side) and cold water (outer coiled tube side) as the working fluids. The flow rate in the inner-coiled tube was varied over a range of 50-200 LPH for a constant annulus side flow rate. Four flow levels were used in the annulus tube: 200, 300, 400, and 500 LPH. Temperature data was recorded at every fifteen min, this data taken after temperature stabilized. Similarly for the heat transfer study in the inner-coiled tube, the outer-coiled tube flow rate was kept constant at 200 LPH and variation in the inner-coiled flow rate was made. At the inlet of the outer

**Table 1** Range of operating parameters

| Parameters | Range |
|---|---|
| Inner tube side water flow rate | 0.014–0.055 kg/s |
| Annulus side water flow rate | 0.055–0.11 kg/s |
| Inner tube inlet temperature | 50–61 °C |
| Inner tube outlet temperature | 32–44 °C |
| Annulus side inlet temperature | 29–31 °C |
| Annulus side outlet temperature | 32–41 °C |

**Table 2** Characteristic dimensions of coiled tube heat exchangers

| Dimensional Parameters | Heat Exchanger |
|---|---|
| $d_i$, mm | 10 |
| $d_o$, mm | 12 |
| Di, mm | 23 |
| Do, mm | 25 |
| Coil or Curvature Radius, mm | 135 |
| Stretch Length, mm | 3392 |
| Wire diameter, mm | 1.5 |

tube of the heat exchanger the cooling water temperature was 29–31 ◦C, and it rises by 4–7 ◦C at the outlet of the outer tube. During the experiments the ambient temperature was 29–31◦C, therefore, there was not much heat loss from the outer wall of the heat exchanger. Similarly, the heat transfer study was carried out for different coils (Plain tube, 6 mm wire pitch and 10 mm pitch wire wound).

## IV. SAMPLE CALCULATION

***Calculation of inner tube side heat transfer coefficients***

In present investigation, the overall heat transfer coefficient, *Uo*, was calculated from the temperature data and the flow rates using the following equation

The overall heat transfer coefficient *Uo* was calculated with,

$$Uo = \frac{Q}{Ao\ \Delta T\ LMTD} \quad (1)$$

The overall heat transfer rates were based on surface area, Ao; heat transfer rate, Q and LMTD is the log mean temperature difference, based on the inlet temperature difference $\Delta T_1$, and outlet temperature difference $\Delta T_2$.

$$\text{LMTD} = \frac{(\Delta T_2 - \Delta T_1)}{\ln(\Delta T_2/\Delta T_1)} \quad (2)$$

The overall heat transfer coefficient can be related to the inner and outer heat transfer coefficients by the following equation [2].

$$\frac{1}{U_o} = \frac{A_o}{A_i h_i} + \frac{A_o \ln(d_o/d_i)}{2\pi KL} + \frac{1}{h_o} \quad (3)$$

Where, $d_i$ and $d_o$ are inner and outer diameters of the inner tube respectively. k is thermal conductivity of wall material and L is length of inner tube (stretch length) of heat exchanger. The $h_i$ and $h_o$ are convective heat transfer coefficient of inner and annulus side respectively, calculated from wiloson plot method as described by rose [20]. Wilson plots are used to calculating the heat transfer coefficients based on overall temperature difference and the rate of heat transfer, without the requirement of wall temperatures. Wilson plots are generated by calculating the overall heat transfer coefficients for a number of trials where one fluid flow is kept constant and the other is varied. . In this case, the flow in the annulus side of tube was kept constant and the flow in the inner tube was varied for the four different flow rates mentioned above.

## V. RESULT AND DISCUSSION

In present study, experimental studies are conducted for single-phase water to water heat transfer application. The tube in tube helical coil heat exchanger has been analyzed in terms of temperature variation and friction factor for changing the pitch distance of wire which is wound on outer side of inner tube. The results obtained from the experimental investigation of heat exchanger operated at various operating conditions are studied in detail and presented.

***Thermal Performance of inner tube***

As discussed in last section the procedure to find overall heat transfer coefficients for various geometries. The results based on the experimental data are as follows,

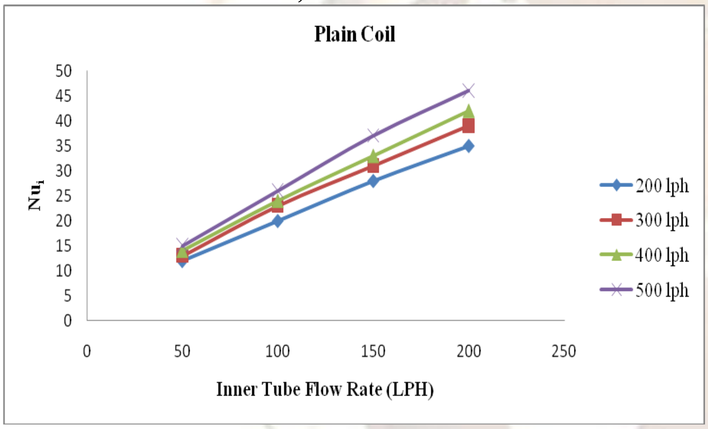


Fig. 3 Variation of inner tube flow rate with inner Nusselt Number at constant annulus flow rate for plain tube in tube helical coil heat exchanger

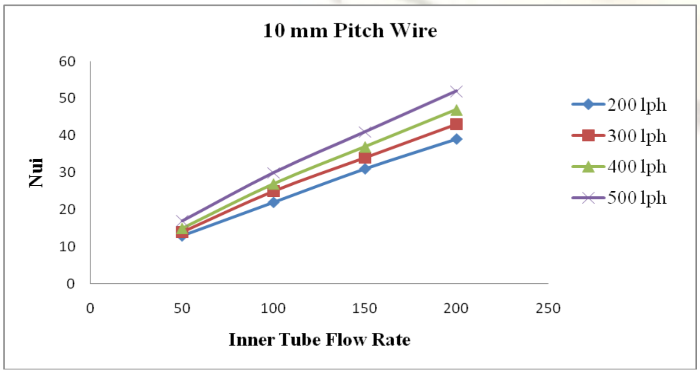


Fig. 4 Variation of inner tube flow rate with inner Nusselt Number at constant annulus flow rate for 10 mm pitch of wire wound of tube in tube helical coil heat exchanger

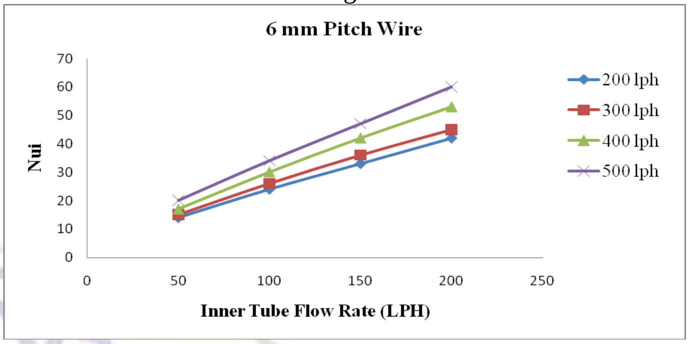


Fig. 5 Variation of inner tube flow rate with inner Nusselt Number at constant annulus flow rate for 6 mm pitch of wire wound of tube in tube helical coil heat exchanger

The Figure 3, 4, 5, represents, the Nusselt Number of inner tube at constant flow rate from annulus side was linearly increases with increasing flow rate of water through inner tube. Similarly the inner Nusselt Number was proportionally changed with variation of annulus side flow rate at same inner side flow rate.

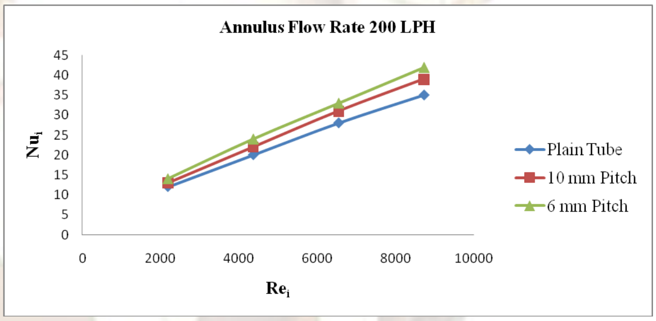


Fig. 6 Inner Reynold Number vs inner Nusselt Number in TTHC

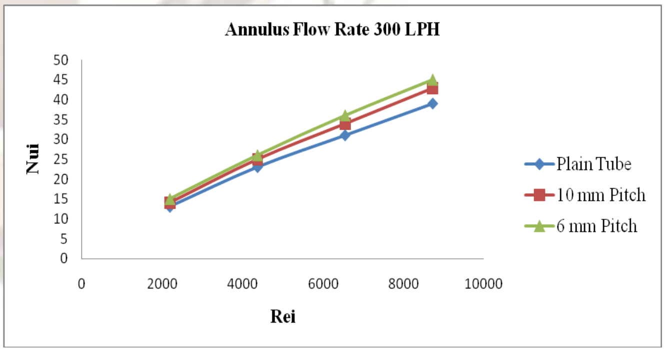


Fig. 7 Inner Reynolds Number vs inner Nusselt Number in TTHC

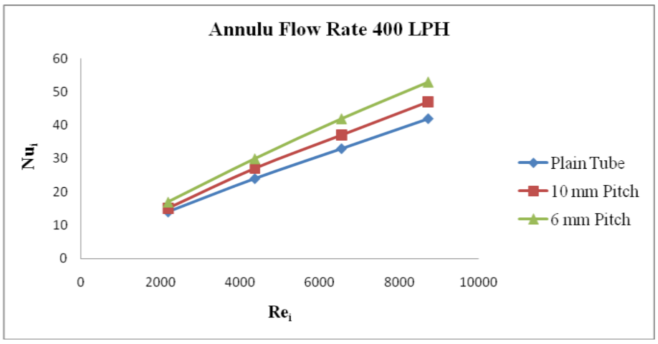


Fig. 8 Inner Reynold Number vs inner Nusselt Number in TTHC

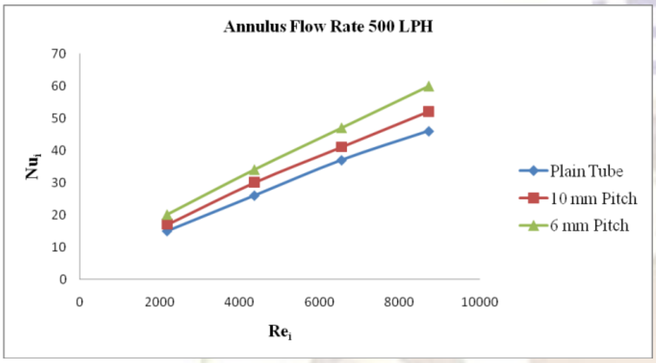


Fig. 9 Inner Reynold Number vs inner Nusselt Number in TTHC

The Fig. 6, 7, 8, 9 illustrates the comparison of fully developed Nusselt number in the inner-coiled tube. It was observed that the inner Nusselt Number increases with increase in the inner-coiled tube Reynold Number for a constant flow rate in the annulus region. Also observed that at same inner Reynold Number, the Nusselt Number of inner tube side increases by decreasing the pitch distance of wound wire from outer side of inner tube.

***Thermal Performance of inner tube***

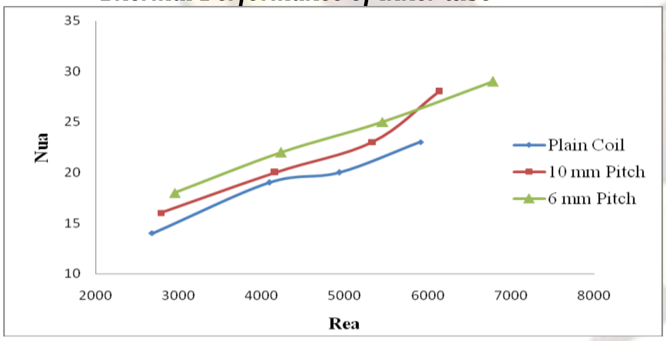


Fig. 10 Annulus Reynolds Number vs annulus Nusselt Number in TTHC

The Fig. 10 shows that, annulus side Nusselt Number increases with respect to increase in Reynolds Number. Also observed that at same annulus Reynolds Number, the Nusselt Number of annulus side increases by decreasing the pitch distance of wound wire from outer side of inner tube.

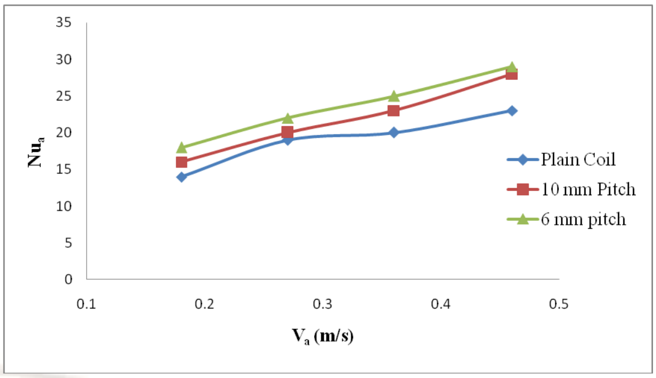


Fig. 11 Annulus Nusselt Number vs annulus velocity in TTHC

The Fig 11 represents the annulus side Nusselt number exhibit linear increase with increasing velocity of cold water through annulus side. Similarly the Nusselt number at same velocity is more as compare to plain tube also observed that at minimum pitch of wire coil, the Nusselt number was high.

The Fig 12 represents, the annulus side Reynolds Number was linearly increases with increasing velocity of cold water through annulus side. Similarly the Reynolds Number at same velocity is more as compare to plain tube also observed that at minimum pitch of wire coil, the Reynolds Number was high.

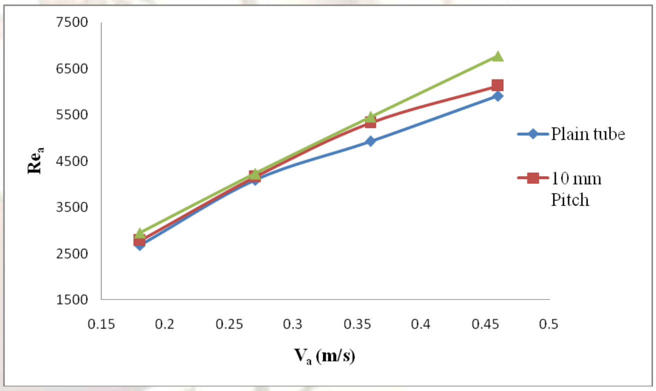


Fig.12 Annulus Reynold Number vs Annulus velocity in TTHC

***Friction Factor***

The annulus side friction factor can be calculated by using the formula,

$$f = \frac{\Delta P D_h}{2\rho v^2 L} \quad (4)$$

Where, L is the length of the heat exchanger and $D_h$ is the equivalent hydraulic diameter for the inner and outer tubes, respectively.

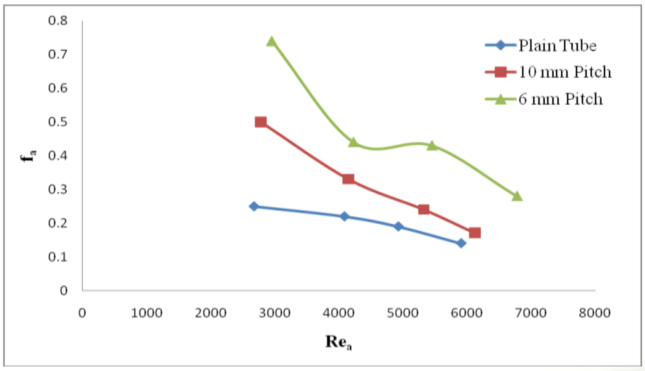


Fig.13 Annulus Reynolds Number vs Annulus Friction Factor in TTHC

It can be seen from the figure.13 that the pressure drop in the annulus section is higher. This may be due to friction generated by outer wall of the inner-coiled tube, inner wall of the outer-coiled tube, as well as baffle installed in the annulus region.

As expected, the friction factor obtained from the tube with coil-wire wound is significantly higher than that without coil-wire insert.

***Correlation for Friction Factor***

Based on the present data, correlations for the annulus-side pressure drop are proposed in term of the Reynolds number and the friction factor. The correlations are presented in Table 3:

**Table 3** Proposing correlations to predict annulus side friction

| **Wire Coil Geometry** | **Correlations** |
|---|---|
| For 6 mm Pitch | $f = \frac{3.1624 \times 10^{3}}{Re^{1.05}}$ Which$2900 \leq Re \leq 6800$ |
| For 10 mm Pitch | $f = \frac{4.9148 \times 10^{3}}{Re^{1.15}}$ which $2800 \leq Re \leq 6200$ |
| For Plain Tube | $f = \frac{33.91}{Re^{0.61}}$ which $2500 \leq Re \leq 6000$ |

## VI. CONCLUSION

Experimentally study of a wire wound tube-in-tube helical coiled heat exchanger was performed considering hot water in the inner tube at various flow rate conditions and with cooling water in the outer tube. The mass flow rates in the inner tube and in the annulus were both varied and the counter-current flow configurations were tested.

The experimentally obtained overall heat transfer coefficient (*U*o) for different values of flow rate in the inner-coiled tube and in the annulus region were reported. It was observed that the overall heat transfer coefficient increases with increase in the inner-coiled tube flow rate, for a constant flow rate in the annulus region. Similar trends in the variation of overall heat transfer coefficient were observed for different flow rates in the annulus region for a constant flow rate in the inner-coiled tube. It was also observed that when wire coils are compared with a smooth tube, at constant pumping power, an increase in heat transfer rate is obtained at Reynolds numbers below 6700. It was also observed that overall heat transfer coefficient is increases with minimum pitch distance of wire coils.

## VII. NOMENCLATURE

| | |
|---|---|
| A | Area of tube, ($m^2$) |
| b | Pitch, (mm) |
| di | Inner diameter of inside coiled tube,(mm) |
| do | Outer diameter of inside coiled tube,(mm) |
| Di | Inner diameter of outside coiled tube,(mm) |
| Do | Inner diameter of outside coiled tube,(mm) |
| D | Coil Diameter,(mm) |
| $D_h$ | Annulus side hydraulic diameter, (m) |
| $f_a$ | Annulus side friction Factor |
| FCV | Flow control valve |
| h | averaged convective heat transfer coefficient, ($W/m^2\,{}^oC$) |
| ΔP | Pressure Drop ,(Pa) |
| $\Delta T_1$ | temperature difference at outlet, (°C) |
| $\Delta T_2$ | temperature difference at outlet, (°C) |
| K | thermal conductivity, ($W/m^{2o}C$) |
| L | Stretch length of coiled tube,(mm) |
| LMTD | Log mean temperature Difference,(°c) |
| LPH | Liter per hour,(lph) |
| M | Mass flow rate, (kg/sec) |
| Nu | Nusselt number |
| Pr | Prandtl number, = $\mu C_P/k$, |
| PCD | Pitch circle diameter,(mm) |
| Q | Heat transfer,(watt) |
| $R_c$ | curvature radius,(mm) |
| Re | Reynolds number,(-) |
| $U_0$ | overall heat transfer coefficient, ($W/m^{20}C$) |
| $V$ | fluid velocity,( m/s) |

**Subscripts**

| | |
|---|---|
| i | Inner side |
| o | Outer side |
| a | Annulus side |

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