International Journal of Engineering Research and General Science Volume 2, Issue 3, April-May 2014
ISSN 2091-2730

Fabrication and Analysis of Tube-In-Tube Helical Coil Heat Exchanger

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ABSTRACT – Conventional heat exchangers are large in size and heat transfer rate is also less and in conventional heat exchanger dead zone is produce which reduces the heat transfer rate and to create turbulence in conventional heat exchanger some external means is required and the fluid in conventional heat exchanger is not in continuous motion with each other. Tube in tube helical coil heat exchanger provides a compact shape with its geometry offering more fluid contact and eliminating the dead zone, increasing the turbulence and hence the heat transfer rate. An experimental setup is fabricated for the estimation of the heat transfer characteristics. A wire is wounded in the core to increase the turbulence in turn increases the heat transfer rate. The paper deals with the pitch variation of the internal wounded wire and its result on the heat transfer rate. The Reynolds number and Dean number in the annulus was compared to the numerical data. The experimental result was compared with the analytical result which confirmed the validation. This heat exchanger finds its application mostly in food industries and waste heat recovery.

Keywords—Tube-in-tube helical coil, Nusselt number, wire wound, Reynold number, Dean number, dead zone, efficiency .

1. INTRODUCTION

Several studies have indicated that helically coiled tubes are superior to straight tubes when employed in heat transfer applications. The centrifugal force due to the curvature of the tube results in the development of secondary flows (flows perpendicular to the axial direction) which assist in mixing the fluid and enhance the heat transfer. In straight tube heat exchangers there is little mixing in the laminar flow regime, thus the application of curved tubes in laminar flow heat exchange processes can be highly beneficial. These situations can arise in the food processing industry for the heating and cooling of either highly viscous liquid food, such as pastes or purees, or for products that are sensitive to high shear stresses. Another advantage to using helical coils over straight tubes is that the residence time spread is reduced, allowing helical coils to be used to reduce axial dispersion in tubular reactors.

The first attempt has been made by Dean to describe mathematically the flow in a coiled tube. A first approximation of the steady motion of incompressible fluid flowing through a coiled pipe with a circular cross-section is considered in his analysis. It was observed that the reduction in the rate of flow due to curvature depends on a single variable, K, which is equal to $2(Re)2r/R$, for low velocities and small r/R ratio. It was then continued for the study of Dean for the laminar flow of fluids with different viscosities through curved pipes with different curvature ratios (δ). The result shows that the onset of turbulence did not depend on the value of the *Re* or the *De*. It was concluded that the flow in curved pipes is more stable than flow in straight pipes. It was also studied the resistance to flow as a function of *De* and *Re*. There was no difference in flow resistance compared to a straight pipe for values of *De* less than 14.6.

Figure 1.1: Diagram of helical coil

Rough estimates can be made using either constant heat flux or constant wall temperature from the literature. The study of fluid-to-fluid heat transfer for this arrangement needs further investigation. The second difficulty is in estimating the area of the coil surface available to heat transfer. As can be seen in Figure, a solid baffle is placed at the core of the heat exchanger. In this configuration the baffle is needed so that the fluid will not flow straight through the shell with minimal interaction with the coil. This baffle changes the flow velocity around the coil and it is expected that there would be possible dead-zones in the area between the coils where the fluid would not be flowing. The heat would then have to conduct through the fluid in these zones, reducing the heat transfer effectiveness on the outside of the coil.

Figure 1.2 close-up of double pipe heat exchanger

Additionally, the recommendation for the calculation of the outside heat transfer coefficient is based on the flow over a bank of non-staggered circular tubes, which is another approximation to account for the complex geometry. Thus, the major drawbacks to this type of heat exchanger are the difficulty in predicting the heat transfer coefficients and the surface area available for heat transfer. These problems are brought on because of the lack of information in fluid-to-fluid helical heat exchangers and the poor predictability of the flow around the outside of the coil.

Nomenclatures:

A	surface area of tube (m^2)	C	constant in Eq. (4)
d	diameter of inner tube (m)	D	diameter of annulus (m)
De*	modified Dean number (dimensionless)	h	heat transfer coefficient (W/m^2 K)
k	thermal conductivity (W/m K)	L	length of heat exchanger (m)
LMTD	log-mean temperature difference (K or C)	q	heat transfer rate (J/s)
U	overall heat transfer coefficient (W/m^2 K)	ΔT1	temperature difference at inlet (K)
v	velocity (m/s)	ρ	density (kg/m^3)
ΔT2	temperature difference at outlet (K)	**ν**	dynamic viscosity (kg/ms)

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Subscripts

I	inside/inner	*hotin*	Hot fluid in
o	outside/outer	i	Inside/inner
c	Cold	*max*	Maximum
coldin	Cold fluid in	*min*	Minimum
cur	Curved tube	*o*	Outside/outer
h	Hot		

2. DIMENSIONAL AND OPRATING PARAMETERS:

Table 1: Characteristic dimensions of heat exchanger

Dimensional parameters	Heat Exchanger
di,mm	10
do,mm	12
Di,mm	23
Do,mm	25
Curvature Radius,mm	135
Stretch Length,mm	3992
Wire diameter,mm	1.5

Table 2: Range of parameters:

Parameters	Range
Inner tube flow rate	200-500LPH
Outer tube flow rate	50-200 LPH
Inner tube inlet temperature	28-30
Outer tube inlet temperature	58-62
Inner tube outlet temperature	30-40
Outer tube outlet temperature	35-46

2.1 METHODLOGY:

The heat exchangers were constructed from mild steel and stainless steel. The inner tube having outer diameter 12mm and inner 10mm was constructed from mild steel and outer tube of outer diameter 25mm and inner diameter 23mm was constructed from stainless steel. Mild steel wire is wounded on the inner tube which has pitch 6 mm and 10 mm on the heat exchangers. The curvature radius of the coil is 135 mm and the stretched length of the coil is 3992 mm. While the bending of tubes very fine sand filled in tube to maintain smoothness on inner surface and this washed with compressed air. The care is taken to preserve the circular cross section of the coil during the bending process. The end connections soldered at tube ends and two ends drawn from coiled tube at one position.

3. EXPERIMENTAL SETUP AND WORKING:

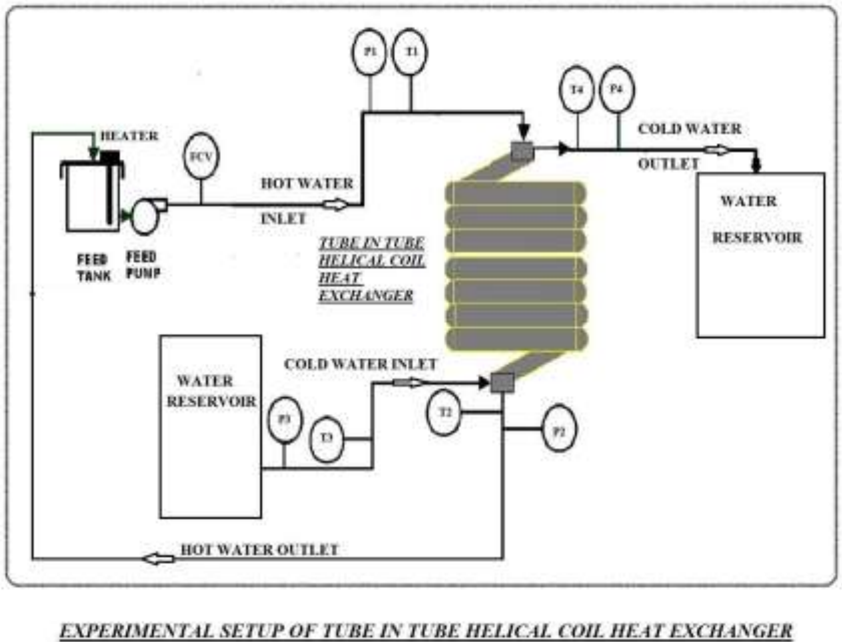

Figure 3.1: Experimental setup

Cold tap water was used for the fluid flowing in the annulus. The water in the annulus was circulated. The flow was controlled by a valve, allowing flows to be controlled and measured between 200 and 500 LPH. Hot water for the inner tube was heated in a tank with the thermostatic heater set at600 C. This water was circulated via pump. The flow rate for the inner tube was controlled by flow metering valve as described for the annulus flow. Flexible PVC tubing was used for all the connections. J-Type thermocouples were inserted into the flexible PVC tubing to measure the inlet and outlet temperatures for both fluids. Temperature data was recorded using a creative temperature indicator.

Figure 3.2: Actual setup

3.1 Experimental Study

A test run was completed on the apparatus. Once all of the components were in place, the system was checked thoroughly for leaks. After fixing the leaks, the apparatus was prepared for testing. The test run commenced with the apparatus being tested under laboratory conditions. Data was recorded every five minutes until the apparatus reached steady state. The hot temperatures fell as expected; the cold temperatures seemed to be more unpredictable in one instance rising six degrees in five minutes and then on the next reading falling three degrees. The apparatus took 120 minutes to reach steady state, which can vary based on operating conditions. Readings were taken until the three-hour mark; however, the data became inconsistent, so a steady state set was determined based on proximity of the readings.

Flow rates in the annulus and in the inner tube varied. The following five levels were used: 100, 200,300, 400, and 500 LPH. All possible combinations of these flow rates in both the annulus and the inner tube were tested. These were done for all the coils in counter flow configurations. Furthermore, three replicates were carried out every combination of flow rate, coil size and configuration. This resulted in a total of 50 trials. Temperature data was recorded every ten seconds. The data used in the calculations was synthesized only after the system had stabilized. Temperature measurements from the 120 s of the stable system were used, with temperature reading fluctuations within ±1.10C. All the thermocouples were constructed from the same roll of thermocouple wire thus carried out for the repeatability of temperature readings being high.

4. DATA COLLECTION AND ANALYSIS:

In present investigation work the heat transfer coefficient and heat transfer rates were determined based on the measured temperature data. The heat is flowing from inner tube side hot water to outer tube side cold water. The operating parameter range is given in table 2.

Mass flow rate of hot water (Kg/sec):

$$m_H = Q_{HOT}(LPH) \times \rho\ (Kg/m^3)$$

Mass flow rate of hot water (Kg/sec)

$$m_C = Q_{COLD}(LPH) \times \rho\ (Kg/m^3)$$

Velocity of hot fluid (m/sec)

$$V_H = \frac{QHOT}{1000 \times \text{Area}}$$

Heat transfer rate of hot water (J/sec)

$$q_H = m_H \times C_P \times \Delta t_{hot} \times 1000$$

Heat transfer rate of cold water (J/sec)

$$q_C = m_C \times C_P \times t_{cold} \times 1000$$

Average heat transfer rate

$$Q_{avg} = \frac{qH + qC}{2}$$

The heat transfer coefficient was calculated with,

$$Uo = \frac{q}{A \times LMTD}$$

The overall heat transfer surface area was determined based on the tube diameter and developed area of heat transfer which is A= $0.22272 m^2$, The total convective area of the tube keep constant for two geometry of coiled heat exchanger.

LMTD is the log mean temperature difference, based on the inlet temperature difference ΔT_1, and outlet temperature difference ΔT_2,

$$LMTD=\frac{(\Delta T1-\Delta T2)}{(\ln(\frac{\Delta T1}{\Delta T2}))}$$

The overall heat transfer coefficient can be related to the inner and outer heat transfer coefficients by the following equation,

$$]\frac{1}{U0}=\frac{A0}{Aihi}+\frac{A0\times\ln(\frac{do}{di})}{2\pi KL}+\frac{1}{ho}$$

Where di and do are inner and outer diameters of the tube respectively. k is thermal conductivity of wall material and L, length of tube (stretch length) of heat exchanger. After calculating overall heat transfer coefficient, only unknown variables are hi and ho convective heat transfer coefficient inner and outer side respectively, by keeping mass flow rate in annulus side is constant and tube side mass flow rate varying,

$$h_i=CV_i^n$$

Where Vi are the tube side fluid velocity m/sec., the values for the constant, C, and the exponent, n, were determined through curve fitting. The inner heat transfer could be calculated for both circular and square coil by using Wilson plot method. This procedure is repeated for tube side and annulus side for each mass flow rate on both helical coils.

The efficiency of the heat exchanger was calculated by,

$$\eta=\frac{1-e^{-\alpha}}{1-\frac{Cmin}{Cmax}e^{-\alpha}}$$

$$\eta=93.33\%.$$

The Reynolds number

$$Re=\frac{(\rho\times V\times D)}{\mu}.$$

Dean number,

$$D_e=\frac{\rho vD}{\mu}\ \ (\frac{D}{2R})^{\frac{1}{2}}.$$

Friction factor,

$$\frac{(\Delta P\times D)}{(2\times\rho\times V^2\times L)}.$$

5. RESULT AND DISSCUSION:

The experiment was 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.

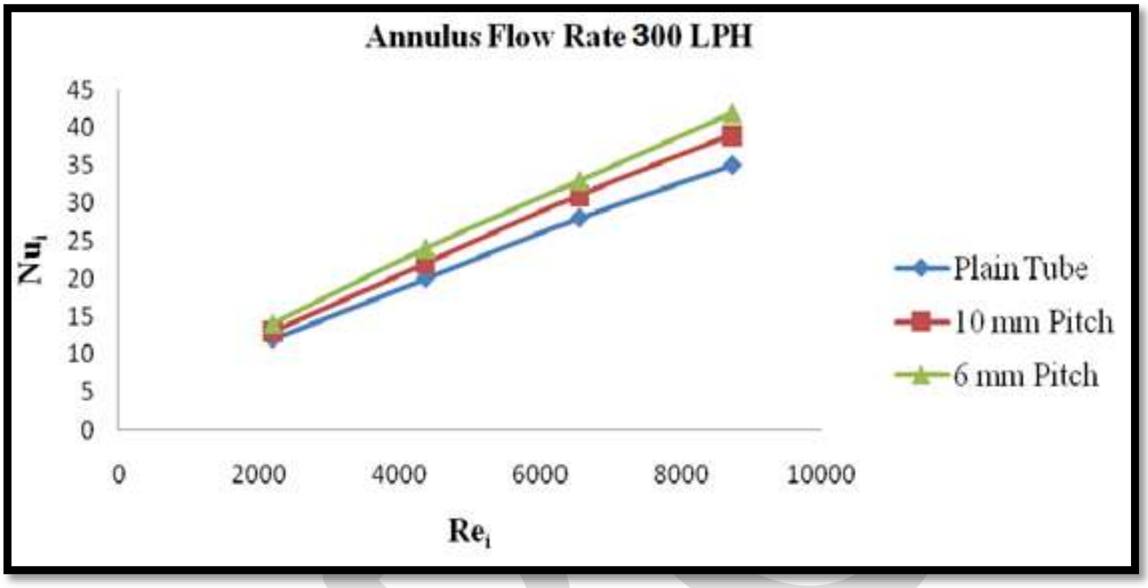

Figure 5.1: Inner Reynold Number vs inner Nusselt Number

Nusselt Number VS Reynolds Number (Annulus Area)

As the Reynolds number increases Nusselt number increases. A larger Nusselt number corresponds to more active convection the 10 mm pitch wire mesh tube place in the tube helical coil shows rapid increment after 5000 Re because of the decreasing friction factor.

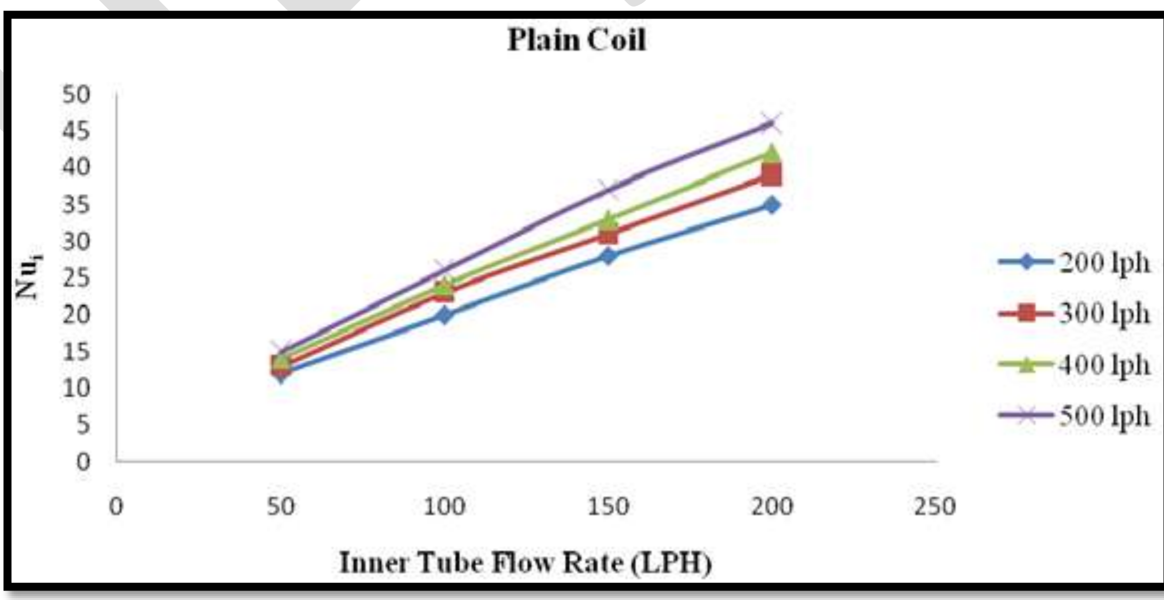

Figure 5.2: Variation of inner tube flow rate with inner Nusselt Number at constant annulus flow rate for plain tube in tube helical coil heat exchanger

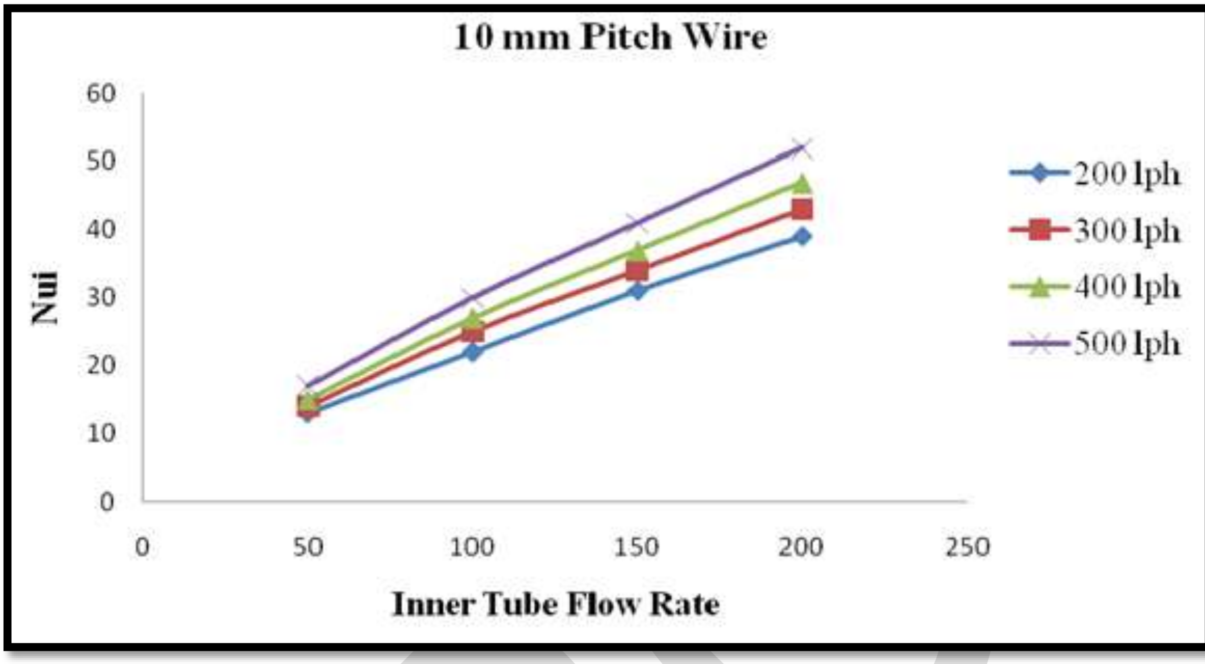

Figure 5.3: Variation of inner tube flow rate with inner Nusselt Number at constant annulus flow rate for 10 mm pitch of wire wound of tube in tube helical coil heat exchanger

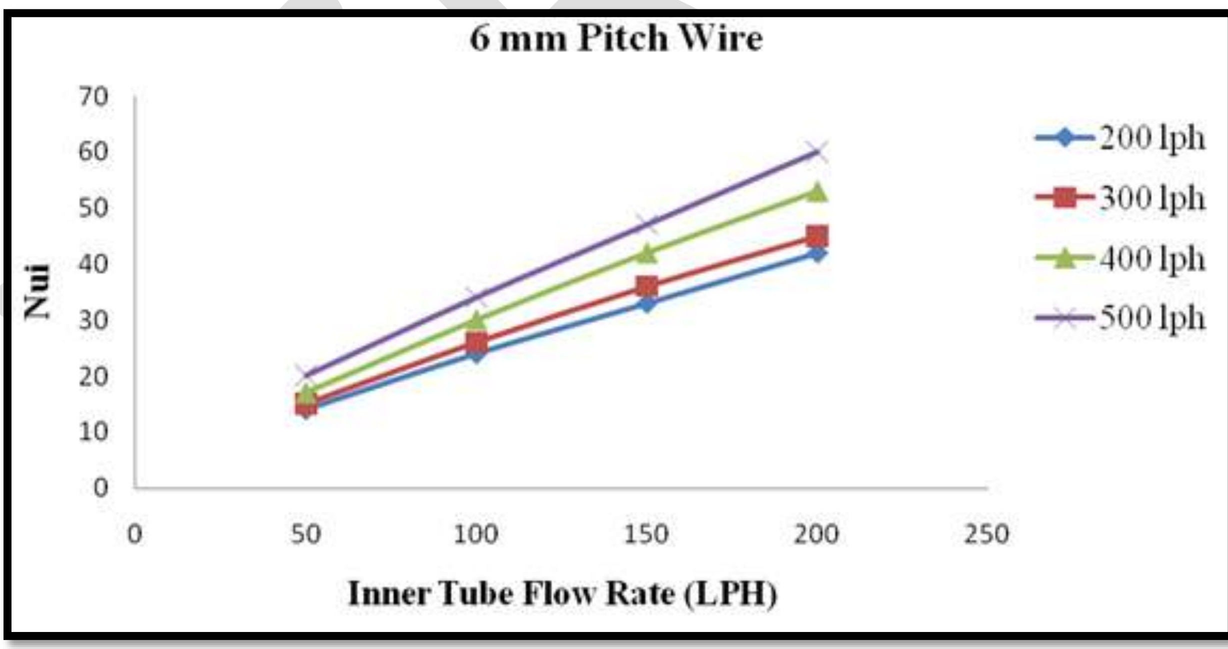

Figure 5.4: Variation of inner tube flow rate with inner Nusselt Number at constant annulus flow rate for 6 mm pitch of wire wound of tube in tube helical coil heat exchanger

The Nusselt Number of inner tube at constant flow rate from annulus side was linearly increasing 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.

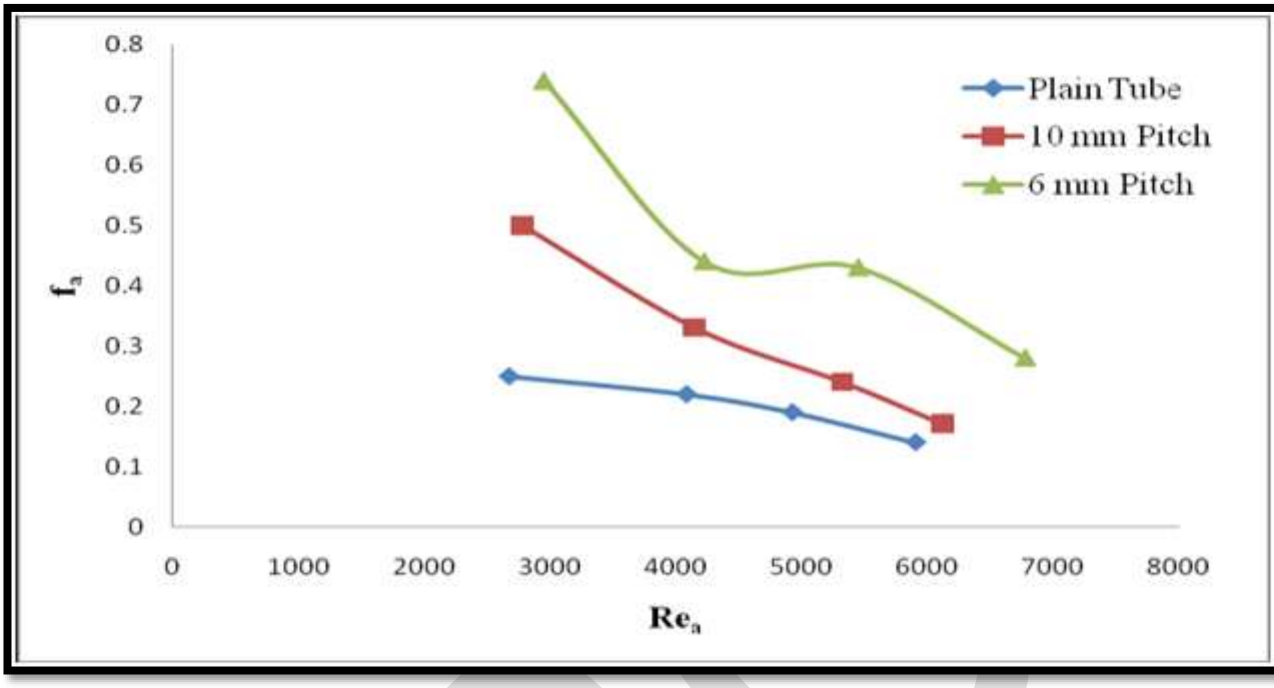

Figure 5.5: Annulus Reynolds Number vs Annulus Friction Factor

5.5 FRICTION FACTOR V/S R_A

It is observed from the figure.5.5, 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, as well as inner wall of the outer-coiled tube. As expected, the friction factor obtained from the tube with coil-wire wound is significantly higher than that without coil-wire insert.

ACKNOWLEDGMENT

This work is currently supported by Prof. Milind S. Rohokale, H.O.D.,Sinhgad Institute of Technology and Prof. Annasaheb Narode for their valuable input and encouragement.

CONCLUSION

Experimental 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, it was also observed that overall heat transfer coefficient is increases with minimum pitch distance of wire coils.

The efficiency of the tube-in-tube helical coil heat exchanger is 15-20% more as compared to the convention heat exchanger and the experimentally calculated efficiency is 93.33%.

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