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Thermal Analysis of a Helical Coil Heat Exchanger

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Abstract— ***In the present work the convective heat transfer coefficients of a helical coil heat exchanger are investigated experimentally. Three helical coils of different curvature ratio and pitch are arranged horizontally in a shell and are tested for counter flow arrangement. Hot water is made to flow through the helical coil and the cold water through the shell. The tube side and shell side flow rates were altered and appropriate instruments were used to measure the flow rates and temperatures of both the fluids. Tube side and shell side convective heat transfer coefficients were calculated using Wilson plots. Based on the curvature ratio and pitch ratio separate empirical correlations are proposed for tube side and shell side for 75 test runs. The calculated convective heat transfer coefficients of tube side and shell side those obtained from the consideration of curvature ratio and pitch ratio are compared and were found in good agreement.***

Keywords— Helical coil tube, Heat transfer coefficient, Heat exchanger, Dean Number, Nusselt Number

I. INTRODUCTION

Heat exchangers have significant applications in refrigeration & air-conditioning systems, heat recovery processes, chemical reactors, food processing, power engineering and other energy intensive industries. The design of an efficient heat exchanger had always been significant to equipment designers.

Due to their compact structure and high heat transfer coefficient, helical coils as one of the passive heat transfer enhancement technique is widely used in various industrial applications.The centrifugal force induced due to the curvature of the tube results in the secondary flow development which enhances the heat transfer. Numerous studies have been carried out by researchers to investigate the fluid flow and heat transfer characteristics in the coiled tubes. Dravid et al. [1] in the fully developed region and the thermal entrance region studied the effect of secondary flow on laminar flow heat transfer in helically coiled tubes. The results obtained from predictions were validated with those obtained from experiments in the range in which they overlapped. Patankar et al. [2] studied the effect of the Dean number on friction factor and heat transfer in the developing and fully developed regions of helically coiled pipes. Good agreements were obtained from comparisons between the developing and fully developed velocity profiles, the wall temperature for the case of axially uniform heat flux with an isothermal periphery obtained from calculation and those obtained from experiments. In the model mentioned above, the effects of the torsion and the Prandtl number were not taken into account. Rennie et al [3] investigated performance of a double pipe helical heat exchanger. The overall heat transfer coefficients were calculated and heat transfer coefficients in the inner tube and annulus were determined using Wilson plots. Results revealed that there was significant increase in Nusselt number in the entrance region and also heat transfer rates in counter flow configuration. Ghorbani et al [4] experimentally investigated the mixed convection in helically coiled heat exchanger for various Reynolds numbers, Rayleigh's number, various tube-to-coil diameter ratios and different dimensionless coil pitch for both laminar and turbulent flow inside coil. The mass flow rate of tube side to shell side ratio (Rm) was found to be effective on the axial temperature profile of heat exchanger .The results indicated that for Rm greater than unity, the temperature profiles were of quadratic form from bottom to top of the heat exchanger. The profiles were linear for Rm close to unity and when the mass flow rate ratio was considerably less than unity, the temperature profile were of the logarithmic form. Moawed [5] experimentally studied the forced convection heat transfer from a constant heat flux helical coil tube. The experiments were carried for the following range of parameters, $7.086 \le D/do \le 16.142$,$1.81 \le P/do \le 3.205$ and $6.6 \times 10^2 \le Re \le 2.3 \times 10^3$. A correlation of the average Nusselt number to describe the forced convection for the range of; $6.6 \times 10^2 < Re < 2.3 \times 10^3$; $7.086 < D/d_o < 16.142$ and $1.81 < P/d_o < 3.205$, was obtained as follows: $Nu_m = 0.0345\, Re^{0.48} (D/d_o)^{0.914} (p/d_o)^{0.281}$ (1)

The study revealed that the key design parameter ie D/do and P/do have a significant effect on the average heat transfer coefficient. Jamshidi et al [6] experimentally analyzed heat transfer enhancement in shell and coiled tube heat exchangers having copper tubes of 9 mm inner diameter and 12.7 mm outer diameter with 10 turn. Wilson plot was used to determine the convective heat transfer coefficients and Taguchi method was used to investigate the effect of fluid flow and geometrical parameters on heat transfer rate. Optimum condition was found for the desired parameters in the range of $0.0813 < Dc < 0.116$, $13 < Pc < 18$, tube and shell flow rates from 1- 4 LPM. Increase in overall heat transfer coefficient was a function of shell side Reynolds number. In addition, as coil pitch increases, tube side Nusselt number decreases and shell side Nusselt number increases. Xin and Ebadian [7] cited by [6] considered the effects of the Prandtl number and geometric parameters on the local and average convective heat transfer characteristics in helical pipes. Five helical pipes with different torsion and curvature ratios were tested with three different working fluids. The results showed that for the laminar flow region the peripheral Nusselt number changed significantly as the Prandtl and the Dean numbers increased. Based on the present data, new empirical correlations for the average fully developed flow for $20 < De < 2000$, $0.7 < Pr < 175$, $0.0267 < d/D < 0.0884$ and for $5 \times 10^3 < Re < 10^5$, $0.7 < Pr < 5$, $0.0267 < d/D < 0.0884$ were respectively obtained as "equation 2" and "equation 3": $Nu = (2.153 + 0.318\, De^{0.643})\, Pr^{0.177}$ (2)

$$Nu = 0.00619 Re^{0.92} Pr^{0.4} [1+3.455(d/D)] \quad (3)$$

Heat transfer and flow characteristics in helically coiled copper tubes with various curvature ratios (0.125, 0.0862 and 0.05) and different coils pitch was predicted by Beigzadeh and Rahimi [8] using artificial neural networks (ANNs). The predicted Nusselt number from ANNs was compared with experimental results as well as those obtained from correlation developed by Xin and Ebadian [7]. The ANN predicted friction factor was compared with measured values and those obtained from empirical correlation proposed by Ito [9]. In both cases, the superior performance of developed neural network was proved. Kumbhare et al. [10] reported an experimental work for heat recovery system in helical coils of circular and square cross section in laminar flow regime. Tube of di = 10 mm, do = 12 mm, Dc = 178 mm were tested. Wilson plot technique was used to determine the overall heat transfer coefficients under various operating conditions. The experimental results revealed that the heat transfer characteristics for coil of square cross section were higher than coil of circular cross section for tube side Reynolds number. . Hashemi and Behabadi [11] experimentally investigated heat transfer and pressure drop characteristics of CuO base oil nanofluid flow for Reynolds number 10-150 in a horizontal helically coiled copper tube of curvature ratio 0.044 and coil pitch 55 mm under constant heat flux condition. Salimpour [12] studied the heat transfer characteristics of temperature dependent-property engine oil flow inside, shell and coiled tube heat exchanger. Coils of different pitches in counter flow configuration at different oil temperatures were tested. The available empirical correlation for constant property assumption has a remarkable deviation from the experimental data especially in high Dean number region. The result revealed that the coil side heat transfer coefficient of the coiled tubes with larger pitches are less than those with smaller pitches also the effect of pitch on Nusselt number is more discernible in high temperatures. Shokouhmand et al [13] carried out investigation to study coil side and shell side heat transfer coefficients for coils of different pitches and curvature ratios for both parallel flow and counter flow configurations. The study revealed that the shell side heat transfer coefficients of the coils with large pitches are more than that the coils with smaller pitches, moreover the higher shell side Nusselt numbers are obtained in counter flow configuration than that of parallel flow configuration. Salimpour [14] tested three coils of different pitches for counter flow configuration. The empirical correlation for constant temperature boundary condition was found in good agreement with the experimental data in low Dean number region. After examining the existing literature it was found that there are very few investigations on the convective heat transfer coefficients of this kind of helical coil heat exchanger considering the geometrical effects of curvature ratio and pitch ratio. In the present work tube side and shell side convective heat transfer coefficients were calculated using Wilson plots. The calculated convective heat transfer coefficients of tube side and shell side those obtained from the consideration of curvature ratio and pitch ratio are compared.

A. Nomenclature

A	surface area of coiled tube, m^2
b	coil pitch, m
C	constant of Eq. (9)
Pr	Prandtl number, = μCp/k
Rc	curvature radius, m
C1,..., C12	unknown constants
d	coiled tube diameter, m
D	shell diameter, m
Dh	shell-side hydraulic diameter, m
De	Dean number, = Re $(d/2Rc)^{0.5}$
h	averaged convective heat transfer coefficient, $W/m^2\ {}^oC$
k	thermal conductivity, $W/m^2\ {}^oC$
L	Stretched length of coiled tube, m
n	exponent in Eq. (9)
N	number of data points
Nu	Nusselt number
Re	Reynolds number
U	overall heat transfer coefficient, $W/m^2\ {}^oC$
v	fluid velocity, m/s

Greek letters

δ	curvature ratio, = d/2Rc
γ	pitch ratio = b/2πRc
μ	viscosity, kg/m s
ρ	density, kg/m^3

Subscripts

i	inside condition
o	outside condition

II. GEOMETRY OF HELICAL COIL TUBE HEAT EXCHANGER

The helically coiled tube arranged in a shell is shown in Fig. 1 where d is the diameter of the coiled tube, D is the inner diameter of the shell, Rc is the curvature radius and b is the coil pitch. The curvature ratio δ is ratio of tube diameter to coil diameter. The pitch ratio is b/2πRc.

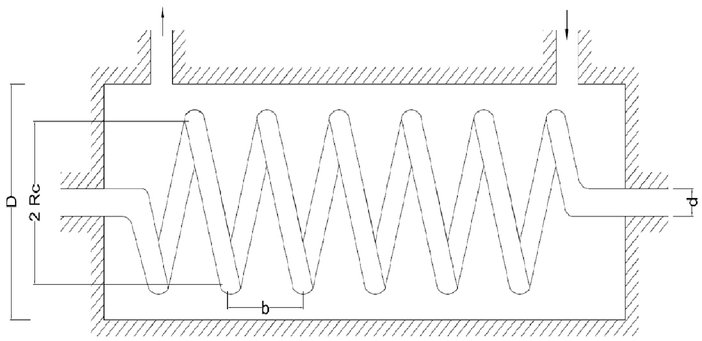

Fig. 1 Schematic of helical coil heat exchanger

III. EXPERIMENTAL SET UP

The schematic diagram of the experimental set up is as shown in Fig.2. The experimental set up consists of a shell in which the helical coil copper tube is placed through which hot water is made to flow with the help of a centrifugal pump. To ensure maximum heat transfer the copper helical coil is fully immersed in the cold water flowing through the shell, the inlet and outlet are so placed as shown in Fig. 1.The shell is well insulated so as to avoid the heat loss to the surrounding. The main components in the set up include centrifugal pump, heating element, cold water storage tank and hot water storage tank. The heat exchanger which includes the helical copper tube and insulated shell is perfectly sealed so as to avoid the leakage of hot water flowing through tube and cold water flowing through shell in a counter flow manner. The characteristic dimensions of the heat exchanger are as given in Table 1.

TABLE 1
CHARACTERISTIC DIMENSIONS OF THE HELICAL COIL HEAT EXCHANGER

Coil	di (mm)	do (mm)	Dc (mm)	L (mm)	b (mm)	δ	N
1	11.7	12.7	90	5715	18	0.13	18
2	11.7	12.7	105	6000	15	0.111	17
3	11.7	12.7	115	6000	24	0.101	15
Characteristic Shell length = 0.5m Shell Diameter =15.24cm							

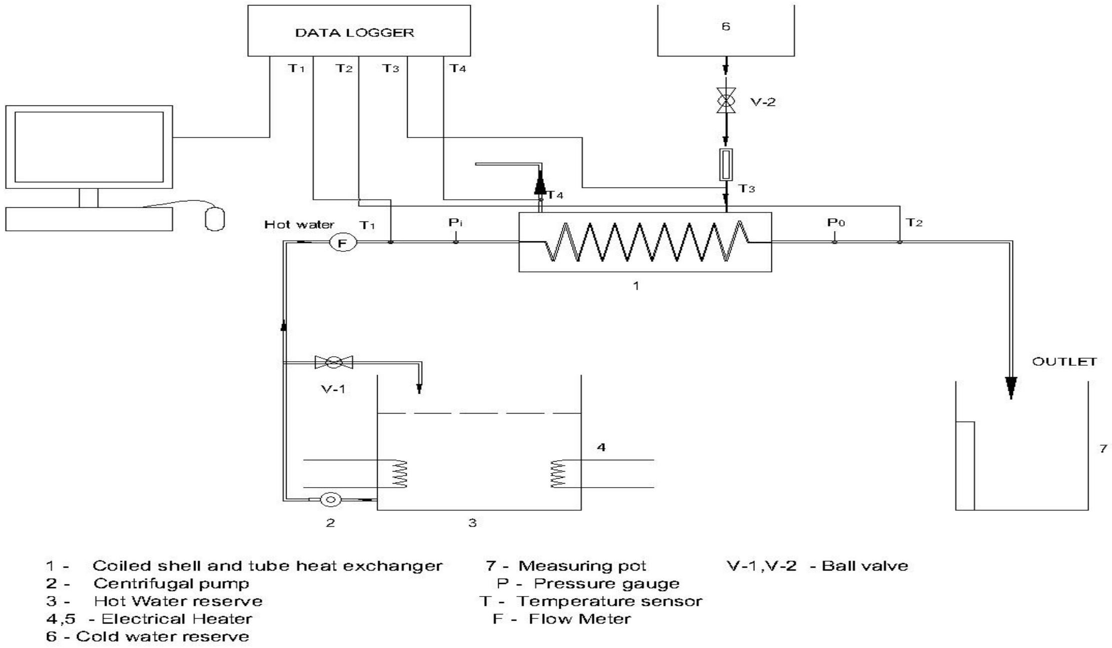

Fig.2 Schematic of Experimental setup of helical coil heat exchanger

The water in the storage tank is heated using a heating element, as the water reaches to a prescribed temperature the centrifugal pump circulates the hot water through the helical coil. The ball valves are used to control the flow rate of hot and cold water. A calibrated rotameter was used to measure the shell side cold water flow rate while for the tube side hot water flow rate a calibrated vane type flow meter is used and data was recorded using a data logger system. The inlet and outlet temperatures of hot and cold water were recorded using calibrated LM 35 temperature sensor through the data logger system. Pressure gauges were used to measure the pressure loss in the helical coil tube. The tube and shell side thermo-physical properties of water were assessed at their mean temperatures.

IV. DATA COLLECTION AND ANALYSIS

For the counter flow configuration a wide range of flow rates both in the tube side and shell side is covered. The range of parameters is given in Table.2.Total 75 test runs were carried out on all the three helical coil heat exchangers.

The overall heat transfer coefficient is calculated as

$$Uo = \frac{\text{Qavg}}{Ao\ \text{LMTD}} \tag{4}$$

Tube side heat transfer is given by

$$Q_h = m_h\, Cp_h (Th_i - Th_o) \tag{5}$$

Shell side heat transfer is given by

$$Q_c = m_c\, Cp_c (Tc_o - Tc_i) \tag{6}$$

Therefore, Average heat transfer is given by

$$\text{Qavg} = (Q_h + Q_c)/2 \tag{7}$$

The shell side convective heat transfer coefficient h_o and tube side convective heat transfer coefficient h_i were calculated using Wilson plots as described by Rose [15] and Jose [16].The convective heat transfer coefficients can be calculated with the use of overall temperature difference and the rate of heat transfer without the requirement of wall temperature. Wilson plot method is used to avoid the disturbance of flow pattern and heat transfer while attempting to measure wall temperatures. Wilson plots are generated by calculating the convective heat transfer coefficients for a number of test runs where one fluid flow is kept constant and the other is varied. In the present work the flow in the shell is kept constant and the flow in the tube is varied for the five different flow rates. The overall heat transfer coefficient is related to the inner and outer heat transfer coefficients as

$$\frac{1}{Uo} = \frac{Ao}{Aihi} + \frac{Ao\ln\left(\frac{do}{di}\right)}{2\pi kl} + \frac{1}{ho} \tag{8}$$

where di and d_o are inner and outer diameters of the helical tube respectively. After calculating the overall heat transfer coefficient, the unknown parameters are the convective heat transfer coefficients i.e. h_i and h_o. The mass flow rate in the shell is kept constant; it is assumed that the shell side heat transfer coefficient i.e. h_o is constant. The tube side convective heat transfer coefficient is assumed to behave in the following manner with the fluid velocity in the helical tube, V_i:

$$h_i = CV_i^n \tag{9}$$

Substituting Eq. (9) into Eq. (8) the values of constant C and exponent n are determined through curve fitting. The tube side and shell side heat transfer coefficient could then be calculated. Thus similar procedure is followed for each shell side flow rate, coil diameter and tube diameter, which resulted in 15 Wilson plots and 75 tube side heat transfer coefficients.

TABLE II
RANGE OF OPERATING PARAMETERS

Parameters	Range
Tube side water flow rate	0.037-0.111 Kg/sec
Shell side water flow rate	0.0143-0.0283 Kg/sec
Tube inlet temperature	48.8°C
Tube outlet temperature	38-45.4°C
Shell inlet temperature	18.5-35.3°C
Shell outlet temperature	35.1-43.5°C

V. RESULTS AND DISCUSSION

Fig. 3 represents the tube side Nusselt number versus Dean Number for helical coil heat exchanger with different coil pitches.

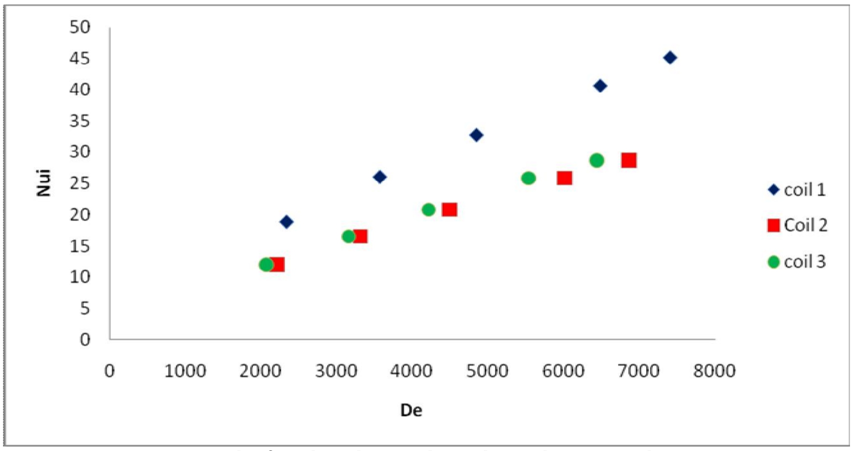

Fig.3 Plot for tube side Nusselt number with Dean number

Fig. 4 represents the tube side Nusselt number versus Dean number for helical coil heat exchanger with different coil curvatures.

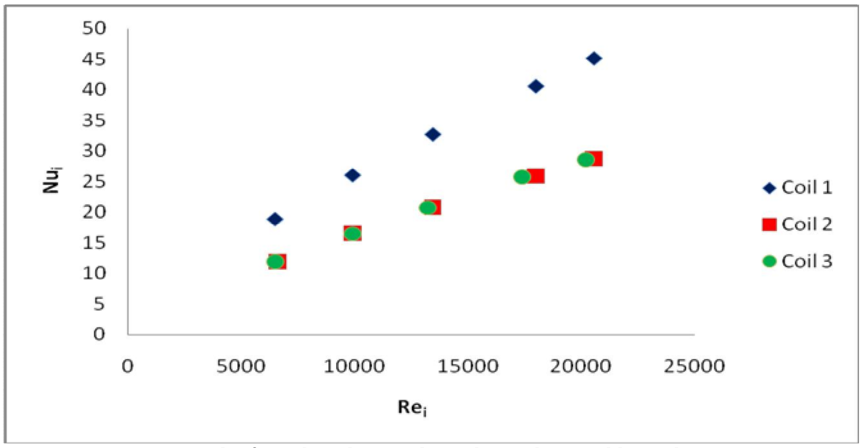

Fig.4 Plot for Tube side Nusselt number with Reynolds number

Fig. 3 and Fig. 4 illustrates that coil 1exhibits higher Nusselt number than coil 2 and coil 3 considering pitch and curvature separately for all the test runs. Fig. 5 represents the variation of shell side Nusselt number with Reynolds number which is calculated based on the hydraulic diameter of the shell for both curvature and pitch considerations. From the figure it is seen that geometric parameters of coil 1 are exhibiting good heat transfer characteristics as compared to the other two. Further on the shell side, Nusselt number decreases with increase in Reynolds number i.e. with increase in mass flow rate.

Thus it is concluded that the coil geometry is an important parameter since the heat transfer rate is dependent on the curvature and pitch.

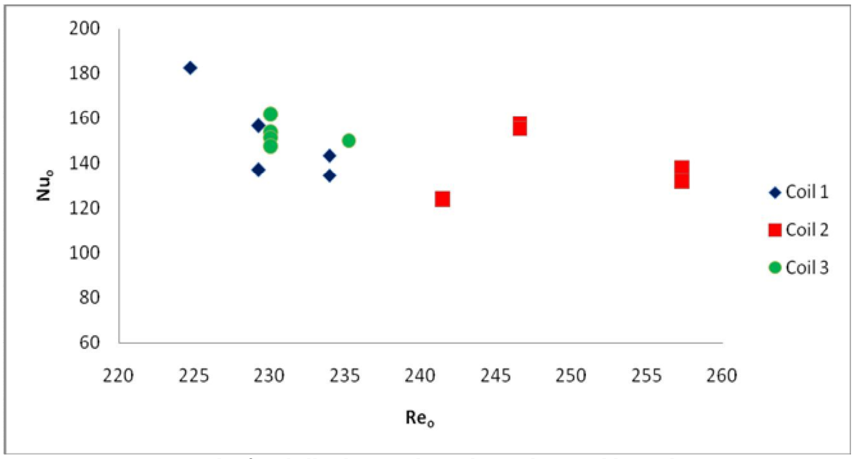

Fig.5 Plot for Shell side Nusselt number with Reynolds number

VI. PROPOSING CORRELATIONS TO PREDICT TUBE AND SHELL SIDES NUSSELT NUMBER

The majority of the published research papers are on heat transfer for tube side Nusselt number based on either constant wall temperature or constant heat flux boundary conditions. These two types of boundary conditions are often encountered in helical coil heat exchangers. In this experimentation a third boundary condition is used which is neither a constant wall temperature nor constant heat flux. Therefore the existing correlations do not conform to the results of present study and hence new correlations were developed to predict inner and outer Nusselt number considering pitch ratio and curvature ratio separately.

A. Pitch Ratio Consideration

Based on the experimental results the Nusselt number, Dean number and Prandtl number were determined and a correlation was established using least square regression analysis. For this purpose, the following functional relationship is assumed:

$$Nu_i = C1 \times De^{C2} \times Pr_i^{C3} \times \gamma^{C4} \tag{10}$$

Taking logarithm of Eq. (10), and introducing the data of Nu_i and corresponding De, Pr_i and γ into it, an error function can be obtained as

$$E\,(C_1, C2, C3, C4) = \sum_{j=1}^{N} \left\{ \ln(Nu_i)_j - \left[\ln C_1 + C_2 \ln De_j + C_3 \ln Pr_j + C_4 \ln \gamma_j \right] \right\}^2 \tag{11}$$

The above error function was minimized using least square regression analysis and the following correlation was obtained with average error of 0.18441%.

$$(Nu_i) = 8.58E-01\left\{\left(Re_i\sqrt{\frac{d_i}{D_c}}\right)^{0.7202}\left(\frac{\mu.C_p}{K}\right)^{-1.8224}\left(\frac{b}{\pi.D_c}\right)^{0.0119}\right\} \tag{12}$$

Fig. 6 shows the comparison between the modeled Nusselt numbers obtained by the proposed correlation, Eq. (12), and the experimental Nusselt numbers.

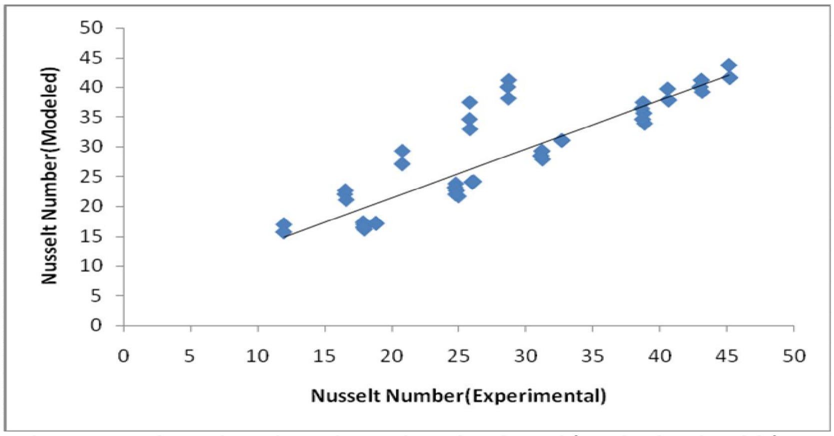

Fig.6 Comparison of Tube side experimental Nusselt number with Nusselt number obtained from developed model for ½ inch tube considering pitch ratio.

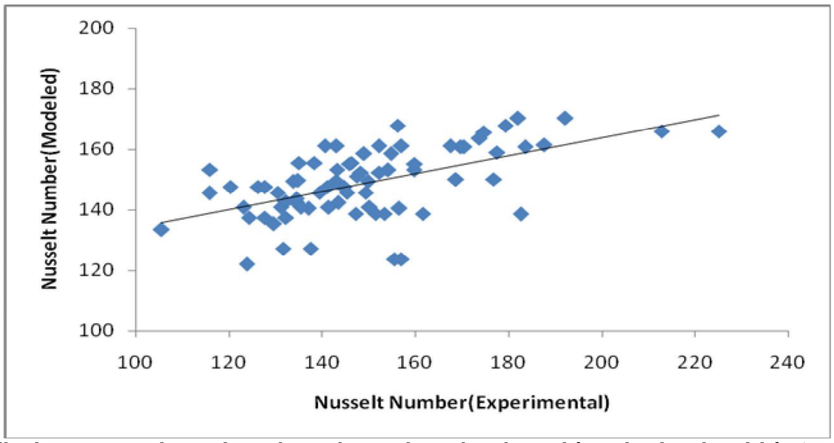

Fig.7. Comparison of Shell side experimental Nusselt number with Nusselt number obtained from developed model for ½ inch tube considering pitch ratio.

In the present study also a correlation was developed by considering pitch ratio to predict the shell side Nusselt number implementing similar treatment that was applied to tube side Nusselt number with an average error of 0.60196%.

$$(Nu_o) = 154.8103527\left\{(Re_o)^{0.2427}\left(\frac{\mu.C_p}{K}\right)^{-0.3721}\left(\frac{b}{\pi.D_c}\right)^{0.2982}\right\} \tag{13}$$

The comparison between the modeled Nusselt number by the proposed correlation, Eq. (13), and the experimental Nusselt number is given in Fig.7.

It is evident from Figs. 6 and 7 that the proposed correlations are found to be in good agreement with the experimental data.

B. Curvature Ratio Consideration

Based on the experimental results the Nusselt number, Reynolds number and Prandtl number were determined and a correlation was established using least square regression analysis. For this purpose, the following functional relationship is assumed:

$$Nu_i = C9 \times Re_i^{C10} \times Pr_i^{C11} \times \delta^{C12} \quad (14)$$

Taking logarithm of Eq.(14), and introducing the data of Nu_i and corresponding Re_i, Pr_i and δ into it, an error function can be obtained as

$$E\,(C_1,C_2,C_3,C_4) = \sum_{j=1}^{N}\left\{\ln(Nu_i)_j - \left[\ln C_1 + C_2 \, ln Re_j + C_3 \, ln Pr_j + C_4 \, ln \delta_j\right]\right\}^2 \quad (15)$$

The above error function was minimized using least square regression analysis and the following correlation was obtained with average error of 0.14657%.

$$(Nu_i) = 31.90803061 \left\{ (Re_i)^{0.6542} \left(\frac{\mu . C_p}{K}\right)^{-3.1737} \left(\frac{d_i}{D_c}\right)^{0.8986} \right\} \quad (16)$$

Fig. 8. Shows the comparison between the modeled Nusselt numbers obtained by the proposed correlation, Eq. (16), and the experimental Nusselt numbers.

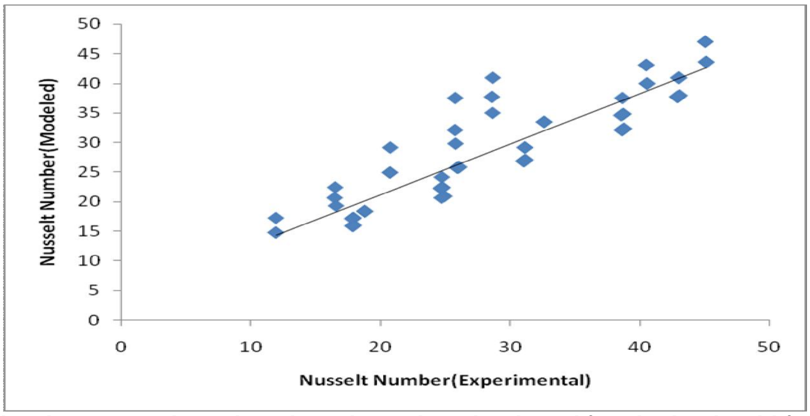

Fig.8. Comparison of Tube side experimental Nusselt number with Nusselt number obtained from developed model for ½ inch tube considering curvature ratio.

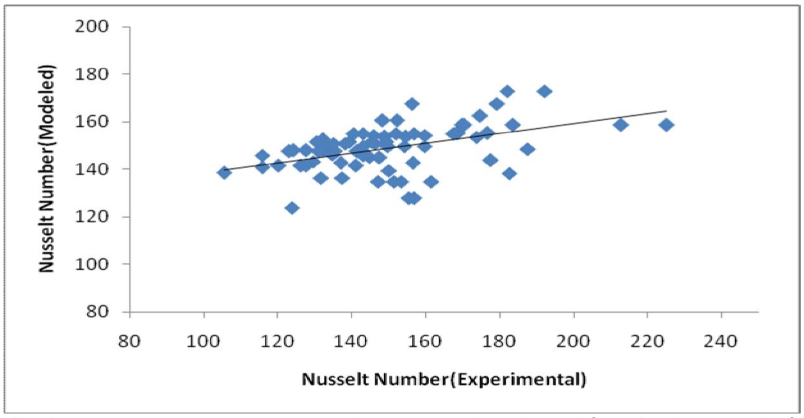

Fig.9. Comparison of Shell side experimental Nusselt number with Nusselt number obtained from developed model for ½ inch tube considering curvature ratio.

In the present study a correlation was developed by considering curvature ratio to predict the shell side Nusselt number implementing similar treatment that was applied to tube side Nusselt number with an average error of 0.69838%.

$$(Nu_o) = 272.8977783 \left\{ (Re_o)^{0.1905} \left(\frac{\mu . C_p}{K}\right)^{-1.1936} \left(\frac{d_i}{D_c}\right)^{-0.1101} \right\} \quad (17)$$

The comparison between the modelled Nusselt number by the proposed correlation, Eq.(17), and the experimental Nusselt number is given in Fig.9.It is evident from Figs. 8 and 9that the proposed correlations are found to be in good agreement with the experimental data.

VII. CONCLUSION

An experimental analysis was carried out to study heat transfer coefficients considering pitch ratio and curvature ratio of a helical coil heat exchanger. Helical coil heat exchangers of three different coil pitches and three different curvatures were investigated for counter flow configuration. From the outcomes of the present study, it is found that the shell side heat transfer coefficients are larger than the tube side heat transfer coefficients considering the pitch ratio and curvature ratio. It is also found that maximum heat transfer characteristics are exhibited by coil 1 as compared to other two coils. At last based on the results of this study, four correlations were developed to predict the tube side and shell side convective heat transfer coefficients of the helical coil heat exchangers.

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