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Enhancement the Thermal performance of a Shell and Coil Heat exchanger with different Coil Geometries: Comparative experimental investigation

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Abstract. : The present paper study experimentally shell and coiled tube heat exchanger performance with two different configuration of copper coiled tubes at two different angles, 0° and 45°.Five heat exchangers of counter flow arrangement were fabricated with at different values of coil torsions (λ_m) and the same mean diameter of the coil (D_c), coiled tube diameter (d_t) and curvature ratio(δ). Experiments carried out at different flow rates of the tube side (60-268 lit/hr) and keeping the shell side flow rate at 60 lit/hr .The effects of mean coil torsion, tube side flow rate and the cone angle were investigated. Results showed that Nu of the tube side and coefficient of heat transfer increases by decreasing the mean coil torsion. At small and high values of Reynolds number (Re_t), the average increase in convection heat transfer coefficient (h_t), tube side Nusselt number (Nu_t) and the overall heat transfer coefficient (U_{ov}) is of 124% to 122%, 126% to 122% and 54% to 32% respectively, when the coil torsion decreases from 0.052 to 0.0202.Also, results illustrated that Nu_t decrease with the increase of cone angle from 0° to 45 °. For the two different angles (0° and 45°), the augmentation in Nu_t at λ_m=0.0202 and λ_m=0.052 is of 52 % to 49% and 81% to 77% respectively. Moreover, correlations for Nu_t at the two different cone angles are obtained.

Keywords: Conical and Helical Coil, Heat Exchanger, Coil Curvature, Coil Torsion.

Nomenclature

A	Area, m^2
De	Dean number
D_h	Hydraulic diameter, m
d	Tube diameter, m
h	Convection heat transfer coefficient, $W.m^{-2}.K^{-1}$.
K	Thermal conductivity, $W.m^{-1}.K^{-1}$
L	Length, m
$\dot{m}$	Mass flow rate, $Kg.s^{-1}$.
P_c	Coil pitch, m
q	Rate of heat transfer, W
R_{ov}	Overall thermal resistance
S	Coil spacing, m
U	Overall heat transfer coefficient, $W.m^{-2}.K^{-1}$

Greek symbols

δ	Dimensionless coil curvature ratio.
μ	Dynamic viscosity, $kg.m^{-1}.s^{-1}$
λ	Coil Torsion
γ	The heat balance deviation

Subscript

c	Cold
h	Hot
in	Inner
L	large
m	Mean
o	Outer
ov	Overall
out	Outlet
s	Small
sh	Shell
t	Tube

Abbreviations

CCT	Conical coiled tubes
HCT	Helical coiled tubes
HT10X	Heat transfer service unit
LMTD	Log mean temperature difference

1. INTRODUCTION

The importance of heat exchangers over the past years has increased extremely from the point of view of energy conversion, recovery, conservation, and effective usage of new sources of energy. Heat exchangers are devices used to transfer heat throw two or more fluid streams at different temperatures. The working fluids could be mixtures or single compounds. Heat exchangers are used in a broad variety of applications. These covers wide area of processes like power process, production, transportation, petroleum, refrigeration ,air-conditioning, heat recovery and alternate fuels .Conventional heat exchangers can substantially improve their performance by several augmentation methods .These methods can be categorized as passive methods, which don't need a direct application of external power source, or as active schemes, which needs an external source of power [1]. The heat exchanger of type shell and coil tube is one of the extreme important passive mechanism applied in the industry because of the massive heat transfer surface area and the compact structure.

White [2] used water and mineral oil to study the flow behavior in curved pipes in laminar conditions. He concluded that the starting point of turbulence did not depend on flow parameter (Re or De) alone. He also noted the more stable flow observed in curved pipes than that of flow in straight pipes and the resistance to flow in the curved pipe is a function of the De and the Re. He further concluded that the resistance to the flow was same for curved and straight tubes for De less than 11.6.

Koutsky and Adler [3] observed the flow field experimentally and numerically and suggested that the velocity profiles in both laminar and turbulent flow tends to flatter than in straight tubes. They also presented the effect of Pr and Re on the flow pattern and Nu. Mori and Nakayama [4] presented the fully developed flow for large De in a curved tube with a uniform heat flux. They analyzed the temperature and flow field both experimentally and analytically. Similar observations were made by McConalogue and Srivastava [5] in study of secondary flow characteristics for developed laminar flow that the maximum velocity shifted towards outer wall. This results in increased velocity and migration of secondary vortices towards the outer wall.

The analysis of steady laminar flow was studied by Topakoglu [6] with an incompressible viscous fluid approximating the stream function in curved pipes. The two parameters Re and curvature ratio (δ) defines the flow rate in the pipe.

Huttl and Friedrich [7-8] studied the curvature and torsion forces for turbulent flow conditions in coiled pipes. They used direct numerical simulations to show that the turbulent fluctuations in curved pipes were less than as straight tubes. They also showed that the curvature effect is much higher than that of effect of torsion on the axial velocity.

K. Yamamoto et al. [9-10] investigated the secondary flow structure with large torsion in a helical tube. They used the smoke visualization technique to study the fluid particle trajectory and validated numerically. In the analysis they observed that, at a constant De, in secondary flow, two counter-rotating vortices structure was transformed to a one-recirculation structure in a cross-section with increase of the torsion of the pipe. The increase in torsion changes the direction of line separating the two vortices from horizontal to vertical.

The velocity of flow in curved tubes measured by Park et al. [11] with the help of a laser photo chromic velocimetry method. Their analysis was performed at δ of 1:6 and Re of 250. They determined the shear stress at the wall, flow vorticity, and pressure field from the measured velocities.

The heat transfer in the coiled tube considering constant heat flux condition for both laminar and turbulent flow was first studied by Seban and McLaughlin [12] with the coils of curvature ratios (δ) of 1/17 and 1/104. Results revealed that in case of helical coils outer periphery had higher Nu than that of the inner, and as compared to straight tube, both inner and outer Nu values were substantially high under the same conditions.

Abo Elazm et al. [13] performed numerical and experimental comparison between the performance of ordinary helical coils and helical cone coils. The result indicated that the helical cone coils has some better heat transfer characteristics than the ordinary helical coils. Results illustrated that the coil geometry has an important effect on the exit temperature, also results revealed that the cone angle has an important effect on heat transfer characteristics.

Kurnia et al. [14] developed a 3D computational model to investigate the heat Transfer performance for different configurations of tubes heat exchangers (straight, conical, helical and in-plane spiral tubes) with square cross sections. Results revealed that in-plane spiral tubes achieve high heat transfer rates. Also results showed that straight tubes have a lower pressure drop compared to coiled tubes.

Tejas et al. [15] studied the design of CCTs and HCTs heat exchanger. The result indicated that the curvature ratio of CCTs heat exchanger increase and cause increase of De due to the conical geometry of the coil.

Akagawa et al. [16] measured the pressure drop across the HCTs with different curvature ratios (δ) of a two-phase (gas-liquid) flow and developed a correlation. They found 1.1 to 1.5 times higher friction factors (f) than the straight pipe.

The pressure drop oscillations were characterized by Guo et al., [17] in a closed loop steam generator system for two-phase flow of steam-water. They analysed the helical coils with the effect of inclination in single-phase and two-phase flow on friction factor (f). The friction factor (f) variation is considerably less in single phase. The effect was significant for the two-phase flow and depending on the inclination angle the friction factor increases up to 70%.

The effect of the pitch on the Nu was studied by Yang et al. [18] in the laminar flow of helicoidal pipes. The analysis shows that the temperature gradient in pipe increases with increased torsion and on the opposite side it decreases. Some related work can be found in [19-21].

According to the literature survey,it is necessary to study of the performance of shell and coiled tube heat exchanger with respect to different geometrical configuration. So,the present study aims to investigate experimentally the effect cone angle and coil torsion on the thermal performance of HCT and CCTheat exchangers with same curvature ratio over a broad domain of fluid-operating conditions. Present measurements were used to introduce experimental Correlations for Nu as a function of Re,λ_m, and Prandtl number.

2. EXPERIMENTAL SETUP

2.1Experimental setup description

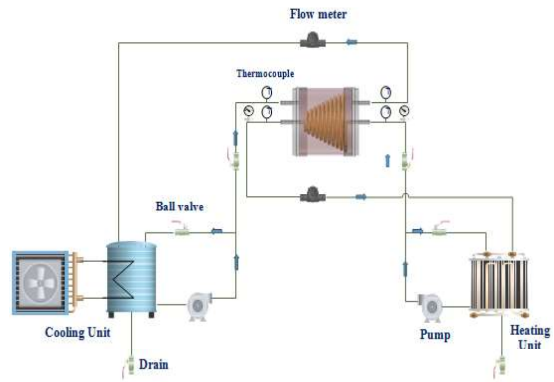


Fig.1 Schematic diagram of the experimental setup

The schematic diagram of the experimental setup illustrated in Fig.1. As shown the experimental setup consist of shell and coil heat exchanger (the test section),heating unit, cooling unit, valves, flow measuring devices, temperature measuring devices, pumps and connecting pipes .The description of the setup sections as follows.

2.1.1 Shell and coil heat exchanger

Five test sections of shell and coil heat exchangers used in the experimental work. Two of them are shell and HCT heat exchangers and three are shell and CCT heat exchanger. The characteristic dimensions of the five shell and coil heat exchanger used in experiments shown in Table 1.The coiled tubes used in the heat exchanger has been made from M-type copper tubes with a thermal conductivity of 401 $Wm^{-1}K^{-1}$.

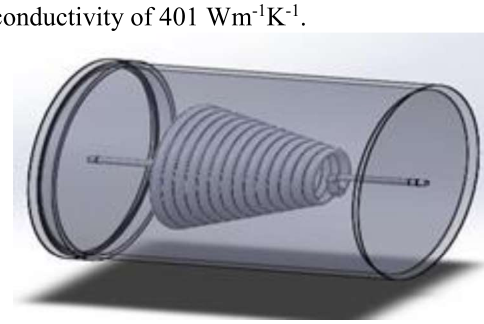


Fig.2Schematic diagram of Test sections

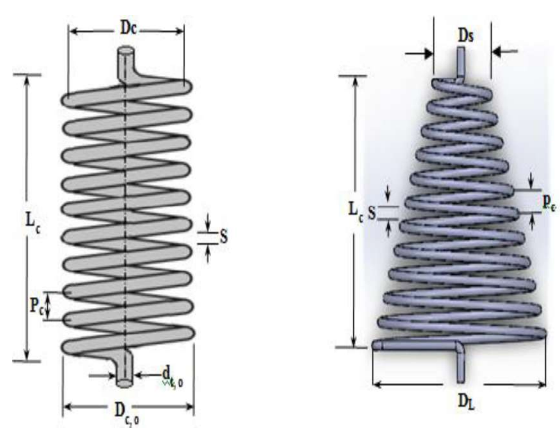


Fig.3 Schematic diagram of the coiled tubes

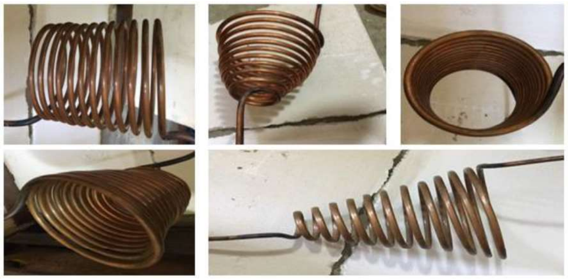


Fig.4 Photograph of the different geometries of the coiled

The coil curvature (δ) and coil torsion (λ) defined as follows:

$$\delta = \frac{d_{t,i}}{D_c} \quad (1)$$

$$D_e = Re_t \delta^{0.5} \quad (2)$$

$$\lambda = \frac{P_c}{\pi D_c} \quad (3)$$

The helical number (He) is an important dimensionless number in the field of helical coils which is determined using the following equation:

$$H_e = Re_t \left[\frac{\delta}{1+\lambda^2}\right]^{0.5} \quad (4)$$

According to equation (4), the corresponding helical number (He) for the studied copper tubes ranges between 713 and 3591.5.

Table 1. The characteristic dimensions of the five test sections used

Cone angle	D_c (mm)	S (mm)	P_c (mm)	L_c (mm)	$d_{t,I}$ (mm)	δ	λ_m
0°	200	0	12.7	152.4	11.28	0.0564	0.0202
0°		20	32.7	405.1			0.0520
45°		0	12.7	165.1			0.0202
45°		10	22.7	239.7			0.0361
45°		20	32.7	372.4			0.0520

Where $D_c = (D_s + D_L)/2$ for conical coiled tubes

2.1.2 Water loop

As shown in Fig. 1, there are two water loops; hot water loop and cold water loop. Two stainless steel tanks of 1.5 mm wall thickness used to hold the hot and coldwater insulated from outside to prevent any heat loss or gain. Electric heater (3 KW) putted at the bottom of the heating tank horizontally to achieve the wanted temperature while a Cooling unit (2.5 hp) used to remove the heat from water in the cooling tank .Thermostat controlled system used to keep the temperature at the inlet of hot and cold water constant. The hot and cold water circulated using two identical (0.5 hp) centrifugal pump with maximum capacity 20 lit/min and maximum head 15m.

2.1.3 Measurement devices

The temperatures at inlet and outlet of shell and tube sides measured using twelve calibrated(K-type) thermocouples .All the thermocouples connected to heat transfer service unit (HT10X)to display the output reading. Two identical water flow sensor used to measure the water flow of cold and water .The description of the devices used in experimental setup are presented in Table.2

Table.2 Technical description of sensors and devices

Device	Description
Thermocouple	• K-type • The measuring range for temperature: 0-1023 °C • Accuracy: ± 1.5 °C • Resolution: 0.25 °C.
HT10X (Heat transfer service unit)	• Readings are in the range 0 to 133°C with a resolution of 0.1°C.
Water flow sensor	• Working temperature: ≤80°C • Allowable pressure: Below water pressure 1.75Mpa • Accuracy: Between 2-60 L/Min ±3%.

3. EXPERIMENTAL PROCEDURE AND ANALYSIS

The hot water flows in the tube side, while the cold water flows in the shell side in a counter flow arrangement. The experimental procedure is as follow:

i. Firstly, the hot and cold water tanks filled with water from the water supply.
ii. The electric supply switched on to turn the pumps, the heater and the cooling unit.
iii. When the hot and cold water reach the desired temperatures, the centrifugal pumps circulate the hot water to the tube side and the cold water to the shell side.
iv. The flow rates of the hot and cold water controlled by ball valves and measured by water flow sensors.
v. The temperatures for both hot and cold water measured by the k-type thermocouples and the output reading displayed by HT10X.The range of operating conditions shown in Table 3.

Table.3The operating parameters of shell and coil heat exchangers

parameter	specification
Tube side inlet temperature	50 °C
Tube side flow rate	60 - 268 (lit/hr)
shell side inlet temperature	20 °C
shell side flow rate	60 (lit/hr)

The heat transfer rate of the hot water (q_h) determined using the following equation:

$$q_h = \dot{m}_h C_{p,h} \left(t_{in} - t_{out} \right)_h \quad (5)$$

Where $\dot{m}_h$ is the mass flow rate of the hot water and $C_{p,h}$ is the specific heat capacity.

The heat transfer rate of the cold water (q_c) determined using the following equation:

$$q_c = \dot{m}_c C_{p,c} \left(t_{out} - t_{in} \right)_c \quad (6)$$

Where $\dot{m}_c$ is the mass flow rate of cold water and $C_{p,c}$ is the specific heat capacity.

The average heat transfer rate determined by the following equation:

$$q_{avg.} = \frac{q_h + q_c}{2} \quad (7)$$

The heat balance deviation in percentage is

$$\gamma = \frac{|q_h - q_c|}{q_{avg.}} \times 100\ \% \quad (8)$$

LMTD (log mean temperature difference) determined by the following equation:

$$\mathrm{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} \quad (9)$$

where : $\Delta T_1 = \left(t_{in,h} - t_{out,c}\right)$

and $\Delta T_2 = \left(t_{out,h} - t_{in,c}\right)$

The overall heat transfer coefficient calculated using the following relation [22]:

$$U_{ov} = \frac{q_{avg.}}{A_o\ \mathrm{LMTD}} \quad (10)$$

Where: A_o is the outside surface area of the tube ($A_o = \Pi\, d_{t,o} L_t$).

The coefficients of heat transfer for the inner tube side (h_t) and for the outer tube side (h_o) determined based on the overall temperature difference and the heat transfer rate using traditional Wilson plot method[23, 24] as shown inFig.5.

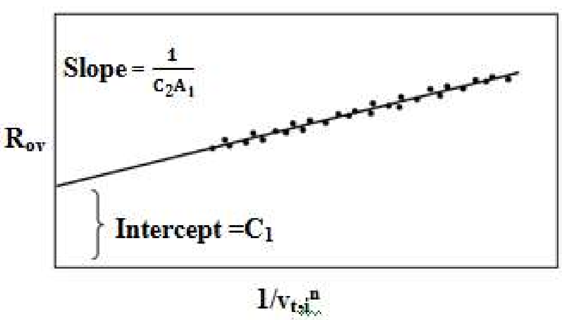


Fig.5 Wilson Plot [24]

Where $R_{ov} = 1/U_{ov}A_o$, $V_{t,I}$ is the velocity of hot fluid inside tube, A_i the inner surface area of the tube and, n, is exponent which is approx. 0.8.

The heat transfer coefficient of the inner tube side determined using the following equation:

$$h_t = C_2 V_{t,i}{}^n \quad (11)$$

The heat transfer coefficient of the outer tube side (h_{sh})determined using the following equation which relate the overall heat transfer coefficients to the inner and outer heat transfer coefficients [25]:

$$\frac{1}{U_o} = \frac{A_{t,o}}{A_{t,i} h_t} + \frac{A_{t,o} \ln(d_{t,o}/d_{t,i})}{2\,\pi\,K\,L_t} + \frac{1}{h_{sh}} \quad (12)$$

Where d_i and d_o are the inner and outer diameter of tube. After calculating hi and ho using Wilson plot method, now Nusselt number in tube and shell sides can be determined using the following equations:

$$Nu_t = \frac{h_t d_{t,i}}{K_t} \quad (13)$$

$$Nu_{sh} = \frac{h_{sh} D_h}{K_{sh}} \quad (14)$$

Where D_h is the hydraulic diameter of shell which is determined using the following equation [26]:

$$D_h = \frac{D_{sh,i}{}^2 L_{sh} - d_{t,o}{}^2 L_t}{D_{sh,i} L_{sh} + d_{t,o} L_t} \quad (15)$$

The Reynolds number of tube and shell sides can be written as follow:

$$Re_t = \frac{4\,\dot{m}_t}{\pi\, d_{t,i}\ \mu_{t,i}} \quad (16)$$

$$Re_{sh} = \frac{4\,\dot{m}_{sh}}{\pi\, D_{sh,h}\ \mu_{sh}} \quad (17)$$

The critical Re number is dependent on the value of the curvature ratio (δ). The relation for critical Reis given by [27].

$$R_{e_{crit}} = 2100\,[1 + 12(\delta)^{0.5}\,] \quad (18)$$

4. UNCERTAINTY ANALYSIS

The uncertainty is calculated based upon the root sum square combination of the effects of each of the individual inputs as introduced by Kline and McClintock [28].The measured data which contained quantifiable uncertainties was considered to be the temperature readings of the thermocouples, the dimensions of the heat exchanger and the mass flow rate of the coil and shell side flow meters. The uncertainty in the inside and outside tube diameters is ±0.01 mm according to the manufacturer which can be neglected. Also, according to thermodynamic tables the uncertainty of the physical properties of water (ρ, μ, k, C_p) was assumed to be ±0.1 %. For all of experiments, the uncertainty in Re_t, Nu_t and U_{ov} was ±2.5 %,± 6% and 2.1 % respectively.

5. RESULTS AND DISCUSSION

A group of experiments was carried out on five counter-flow heat exchangers illustrated in Table 1, which were fabricated with different coil torsions ($0.0202 \le \lambda_m \le 0.052$) and different two angles (0° and 45°) while have the same curvature ratio (δ=0.0564). The operating conditions are hold constant in the shell side while the operating conditions of HCT and CCT are varied as shown in Table 3.

5.1 THERMAL PERFORMANCE

5.1.1 Effect of coil torsion

Figures 6 to 8 indicate that decreasing λ_m leads to increase h_t, Nu_t and U_{ov} even at the same Re_t and $t_{t,i}$..At higher and lower Reynolds number, the average increase in h_t,Nu_t and U_{ov} is of 124% to 122%,126% to 122%and 54% to 32% respectively, when the coil torsion decrease from 0.052 to 0.0202 for $3006.1 \leq Re_t \leq 15123$.This may be due to the increase in rotational force as a result of increasing the coil torsion which reduce the intensity of secondary flow that created due to the centrifugal effect.

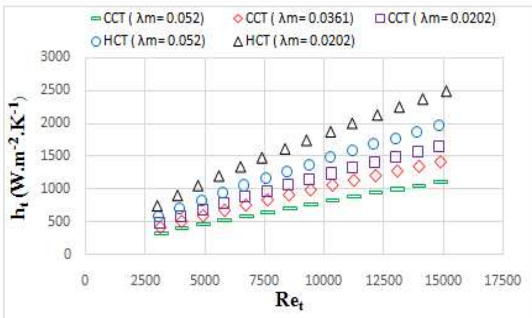


Fig.6 Variation of h_t with Re_t at different coil torsions

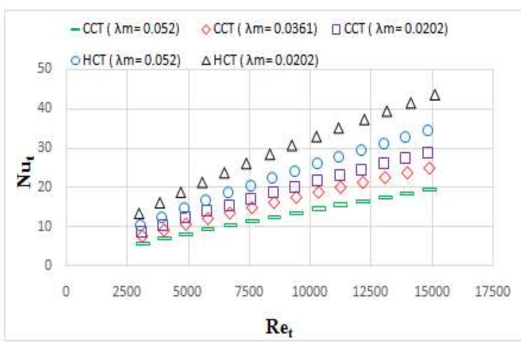


Fig.7 Variation of Nu_t with Re_t at different coil torsion

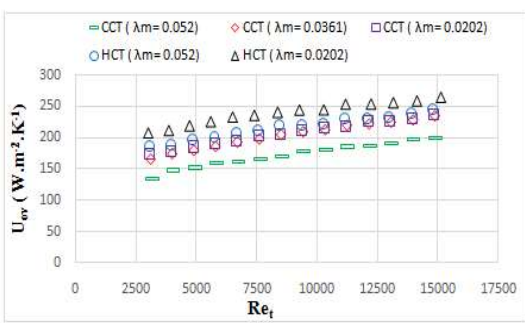
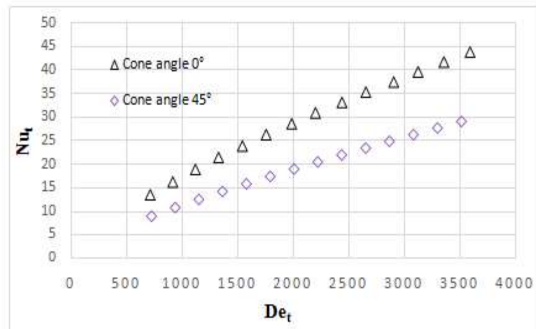


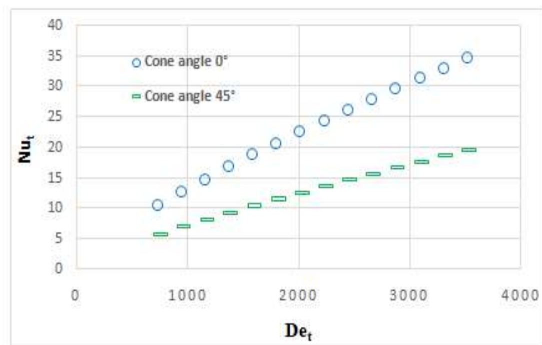
Fig.8 Variation of U_{ov} with Re_t at different coil torsions

5.1.2 Effect of cone angle.

Fig.9show that Nu_t decrease with an increase of cone angle form 0°(HCT) to 45 °(CCT) for $713 \leq De_t \leq 3591.5$.For the two different angles(0° and 45°), the augmentation in Nu_t at λ_m=0.0202 and λ_m=0.052 is of 52 % to 49% and 81% to 77%respectively. The values of Nu_t highest for helical coiled tubes and lowest for conical coiled tubes and this can be due to the change in the intensity of secondary flow which is a function of coil diameter. For the helical coiled tubes, the helical coil diameter is uniform throughout the length of the coil where a uniform intensity of secondary flow is obtained throughout the length of the coil and this keeps the uniform heat transfer per unit surface area. But the geometry of the conical coils make the intensity of secondary flow considerably altered where it is more in the upper part of the coil and less in the lower part. So ,Nu_t decrease with an increase of cone angle.



(a)



(b)

Fig.9 Variation of De_t with Nu_t at
a) λ_m=0.0202, b) λ_m=0.052

5.1.3 Effect of tube side flow rate

The effect of tube side flow rate on the thermal performance of HCT and CCT is studied at constant shell side flow (60 lit/hr) and $0.0202 \leq \lambda \leq 0.052$. At lower and higher tube side flow rate ,Nu_t increase with increase in tube side flow rate as shown in Fig.10. This can be due to the increase in the velocity of tube side fluid which leads to increase in the turbulence and secondary flow especially at high flow rates.

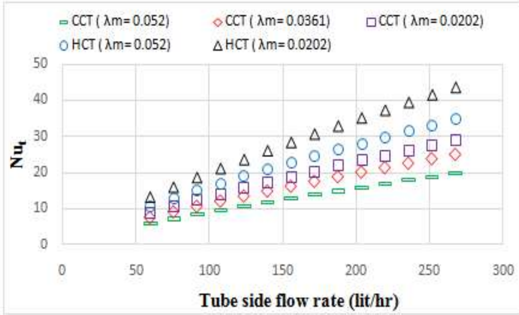


Fig.10 Variation of tube side flow rate with Nu_t

6. CORRELATION OF NUSSELT NUMBER

The correlations were developed to predict the tube side Nu using the experimental data of five different coils. The Nusselt number of these coiled tubes is correlated as a function of Re, Prandtl number and λ_m. The correlation for the conical coil tubes is expressed as;

$$Nu_t = 0.0035\ R_e^{0.7687} P_r^{0.0507} \lambda_m^{-0.409} \quad (19)$$

Equation (19) is applicable for $3070 \le Re_t \le 14890$, $3.68 \le Pr_t \le 4.03$ and $0.0202 \le \lambda_m \le 0.052$.

Also, the correlation for the helical coil is expressed as;

$$Nu_t = 0.00309\ R_e^{0.7945} P_r^{0.7457} \lambda_m^{-0.2437} \quad (20)$$

Equation (20) is applicable for $3006.1 \le Re_t \le 15123$, $3.61 \le Pr_t \le 4.12$ and $0.0202 \le \lambda_m \le 0.052$.

Fig.11 show the comparison between the correlated Nu_t and the experimental Nu_t and it is evident that the proposed correlations show good agreement with the present experimental data with maximum deviation of ±4%.

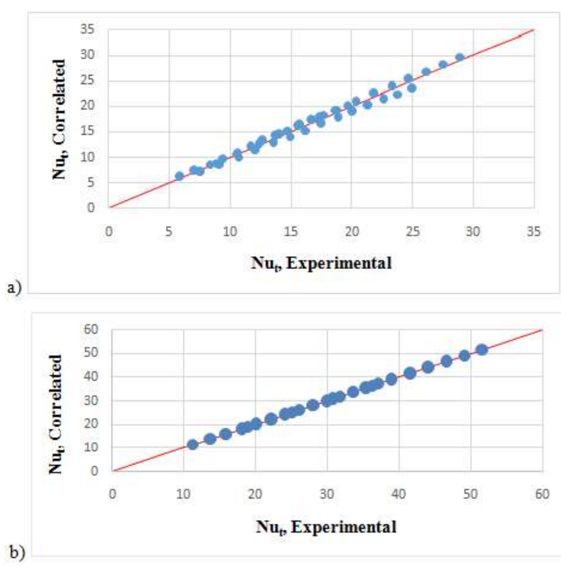


Fig.11 Comparison of experimental values for Nu_t with the Correlated values of Nu_t Proposed by correlation a) CCT, b) HCT.

The present experimental data of Nu_t was compared with the experimental data obtained by Purandare et al. [20] and Kalb et al. [29]. Fig.12 show the results of this comparison. It can be seen that the experimental results for heat transfer calculations are in compatible with previous studies within a deviation 7%.

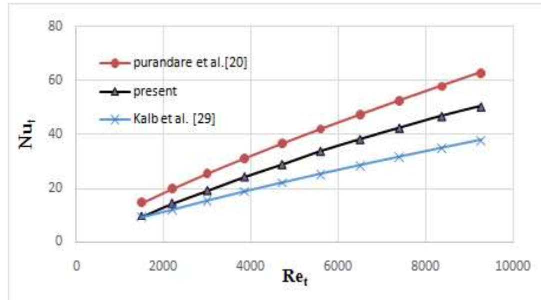


Fig.12Comparison of the experimental data with the previous studies for HCTs

7. CONCLUSION

The present work is done to investigate the heat transfer characteristics in HCT and CCT heat exchangers. The investigation ranges are $713 \le De_t \le 3591.5$, $3006.1 \le Re_t \le 15123$, $3.61 \le Pr_t \le 4.12$ and $0.0202 \le \lambda_m \le 0.052$. From the study, it could be conducted that:

- The dimensionless coil torsion has a slight effect on the thermal performance of shell and coiled tube heat exchanger. According to the data of the experimental data, reducing coil torsion increases U_{ov}, h_t and Nu_t
- At higher and lower Re, the average increase in h_t, Nu_t and U_{ov} is of 124% to 122%, 126% to 122% and 54% to 32% respectively, when λ_m decreases from 0.052 to 0.0202 for $3006.1 \le Re_t \le 15123$.
- The cone angle also has a slight effect on the heat transfer. Nu_t decrease with an increase of cone angle form 0° (HCT) to 45 ° (CCT). For the two different angles (0° and 45°), the augmentation in Nu_t at λ_m=0.0202 and λ_m=0.052 is of 52 % to 49% and 81% to 77% respectively. This can be due to the change in the intensity of secondary flow which is a function of coil diameter.
- Empirical correlations For Nu_t at different cone angles (0° and 45°) are developed and show good agreement with the present experimental data with maximum deviation of ±4 %.

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