EXPERIMENTAL INVESTIGATION ON HEAT TRANSFER AND PRESSURE DROP OF CONICAL COIL HEAT EXCHANGER

by

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The heat transfer and pressure drop analysis of conical coil heat exchanger with various tube diameters, fluid flow rates, and cone angles is presented in this paper. Fifteen coils of cone angles 180° (horizontal spiral), 135°, 90°, 45°, and 0° (vertical helical) are fabricated and analysed with, same average coil diameter, and tube length, with three different tube diameters. The experimentation is carried out with hot and cold water of flow rate 10 to 100 L per hour (Reynolds range 500 to 5000), and 30 to 90 L per hour, respectively. The temperatures and pressure drop across the heat exchanger are recorded at different mass flow rates of cold and hot fluid. The various parameters: heat transfer coefficient, Nusselt number, effectiveness, and friction factor, are estimated using the temperature, mass flow rate, and pressure drop across the heat exchanger. The analysis indicates that, Nusselt number and friction factor are function of flow rate, tube diameter, cone angle, and curvature ratio. Increase in tube side flow rate increases Nusselt number, whereas it reduces with increase in shell side flow rate. Increase in cone angle and tube diameter, reduces Nusselt number. The effects of cone angle, tube diameter, and fluid flow rates on heat transfer and pressure drop characteristics are detailed in this paper. The empirical correlations are proposed to bring out the physics of the thermal aspects of the conical coil heat exchangers.

Key words**:** *helical, spiral, conical coil, heat exchanger, secondary flow*

Introduction

Heat exchangers are considered as an important engineering systems of energy generation and energy transformation in many industrial applications such as power plants, nuclear reactors, refrigeration and air-conditioning systems, heat recovery systems, chemical processing, and food industries. Extensive use of heat exchanger in industries necessitates not only performances, but also size of the heat exchanger. Hence, selection of proper heat transfer enhancement technique has a prime importance. In industrial applications, various techniques are used for heat transfer enhancement. These techniques are classified in two groups: active and passive techniques [1]. The techniques which require external forces for enhancement are known as active techniques like fluid vibration, electric field, and surface vibration, whereas passive techniques are the techniques which are due to special surface geometries or various tube inserts.

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Shell and coil tube configurations are the important passive techniques, frequently used in industry. Helical coiled configuration is very effective for heat transfer equipment such as heat exchangers and reactors because of enhanced heat transfer, compact structure, and accommodate large heat transfer surface area in a small space [2, 3].

Several studies have indicated that helically coiled tubes are superior to straight tubes when employed in heat transfer applications [4, 5]. The secondary developed in the curved helical coiled tubes due to centrifugal force observed in the fluid flowing, enhances the heat transfer in coiled tube heat exchanger. The intensity of secondaries [6-8] developed in the tube are the function of tube diameter, d, and coil diameter, D. For the smaller coil and tube diameter the intensity of secondaries developed is high. This increase in intensity of secondaries allows proper mixing of the fluid, which enhances heat transfer coefficient for the same flow rate. Increase in tube and coil diameter reduces the seconderies developed which reduces heat transfer coefficient [9].

In helical coil heat exchangers coil diameter remains same, hence the intensity of seconderies developed does not change which effects in constant heat transfer coefficient. In spiral coiled geometry the diameter of the coil continuously changes from innermost to the outermost section [10]. This indicates continuous alteration of the local heat transfer coefficient from innermost to the outermost section. This indicates, in case of helical coils the heat transfer coefficient is same throughout its dimensions, whereas in case of spiral coil heat exchanger overall heat transfer coefficient calculated with respect to mean diameter of the coil is slightly different than of local coil heat transfer coefficient.

In literature, numbers of correlations are available for helical coil which shows that Nusselt number, is a function of the Dean number, Prandtl number. Numerical studies of uniform wall heat flux for Dean number in the range of 80-1200 and curvature ratios, δ, of 1/10 to 1/100 for fully developed velocity and temperature field were performed by Kalb and Seader [11]. The analysis presented results in Nusselt number relationship:

$$\mathrm{Nu} = 0.836\,\mathrm{De}^{0.5}\mathrm{Pr}^{0.1} \quad \text{for} \quad \mathrm{De} \geq 80 \quad \text{and} \quad 0.7 < \mathrm{Pr} < 5 \tag{1}$$

Xin and Ebadian [12] has analysed the effect of Prandtl number and geometric parameters on the performance of on helical coil heat exchanger, which results in correlation:

$$\mathrm{Nu} = (2.153+0.318\mathrm{De}^{0.643})\,\mathrm{Pr}^{0.177} \tag{2}$$

for $0.7 < \mathrm{Pr} < 175$; $20 < \mathrm{De} < 1200$; $0.0267 < d/D < 0.0884$

Cengiz *et al.* [13] have provided the analysis of heat transfer and pressure drops in a heat exchanger with a helical pipe containing inside springs. The results indicated that the Nusselt number increased with decreasing pitch/wire diameter ratio. On the basis of the experimental data the empirical correlation was presented for Nusselt number:

$$\mathrm{Nu} = 0.055\,\mathrm{De}^{0.864}\,\mathrm{Pr}^{0.4} \quad \text{for} \quad 70 < \mathrm{De} < 1200; \quad 0.7 < \mathrm{Pr} < 5 \tag{3}$$

Experimental analysis of heat transfer enhancement in shell and helical coil heat exchanger was analysed by Jamshidi *et al.* [14]. Tube and shell side heat transfer coefficients are determined using Wilson plots. Experimental and Taguchi method are used to investigate the effect of fluid flow and geometrical parameters on heat transfer rate. The analysis indicate that the increase in coil diameter, coil pitch, and mass flow rate in shell and tube can increase the heat transfer rate in these types of heat exchangers.

An investigation on the shell-side flow and heat transfer performances of multilayer spiral-wound heat exchanger under turbulent flow was studied by experimentally and numerically by Lu *et al.* [15]. The heat exchanger was analysed by Wilson plot method and correlation for the shell-side Nusselt number outside the tube is obtained:

$$\mathrm{Nu} = 0.179\,\mathrm{Re}^{0.862} \quad \text{for} \quad 500 \le \mathrm{Re} \le 3500 \tag{4}$$

The third configuration in the coiled tube heat exchanger is the conical coil configuration, which is the intermediate configuration of helical and spirally coiled configurations. The conical coil configuration is shown in fig. 1.

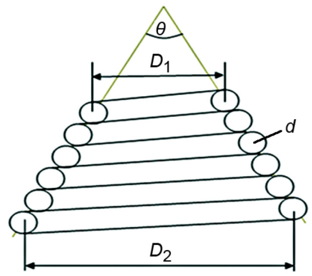

Figure 1. Conical coil used in heat exchanger; θ – cone angle, D_1 – minimum diameter, D_2 – maximum diameter, D_m – mean diameter D_m + $(D_1 + D_2)/2$, d – tube diameter

The conical coil is considered as spiral coiled when the coil cone angle is 180° and it is considered as helical coiled, when the coil cone angle is 0°.

In literature, it is observed that sufficient experimental and numerical data is available for the analysis of helical and spiral coil heat exchanger whereas conical coil configuration with respect to different cone angles is not explored thermodynamically and hydro-dynamically by the researchers for the heat transfer applications. This fact created the motivatation to undertake the current work. The objective of this work is to analyse the conical coil heat exchanger thermodynamically from spiral to helical coil configuration at different intermediate cone angles, spiral (180°), 135°, 90°, 45°, helical (0°). The analysis is extended to establish the relationship between Nusselt number (thermal parameter), Dean number (flow parameter), Prandtl number (fluid parameter), and δ (geometric parameter) from thermal analysis, and f (friction factor), Dean number (flow parameter), and δ (geometric parameter) form pressure drop analysis. In this paper the thermal and pressure drop analysis of conical coil configurations is presented.

Experimentation

Experimental set-up

For the experimentation the experimental set-up was developed as shown in fig. 2.

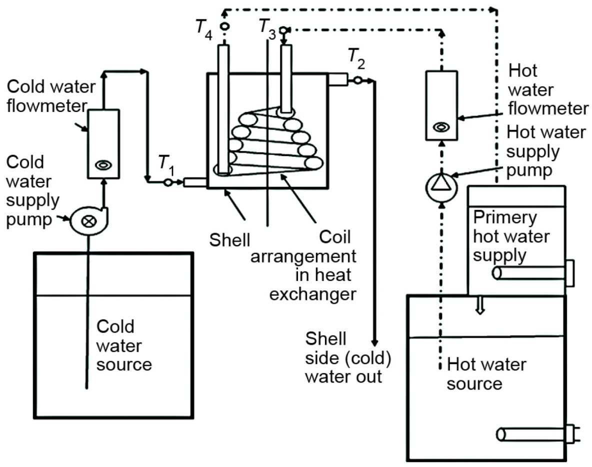

Figure 2. Experimental set-up; T_1 – cold water inlet, T_2 – cold water outlet, T_3 – hot water inlet, T_4 – hot water outlet

The experimental set-up consists of heat exchanger with conical coil. Conical coils are fabricated on the wooden formers of specified cone angles and by hot rolling process. The care has been taken for dimensional stability of the conical coils in fabrication process. The shell is designed to accommodate all coils, one by one, in the same shell. The specially designed closed loop constant temperature hot water supply system supplies hot water to the tube side of the heat exchanger by a constant discharge pump. Constant temperature cold water is supplied to shell side by pump (from separate storage of 500 L capacity installed and maintained at constant temperature in test run). Two rotameters (accuracy of ±1% of the maximum flow rate) are used to measure the cold and hot water flow through the heat exchanger. The constant temperature of inlet cold and hot water flowing through the heat exchanger are maintained with the help of thermostat controlled system. Heat exchanger is insulated by polyurethane foam layer of 6 mm thickness (designed on the basis of available data) to avoid the heat loss due to convection from outer surface. Temperature measurements are carried out using calibrated *k*-type thermocouples (Make-Kristake Instruments and transducers, with accuracy of 0.1 K) which are mounted at various locations. Pressure drop across the heat exchanger is measured with micro-manometer with an accuracy of 0.5 mm of water column. To reduce the heat loss, extensions of the heat exchanger coils are well insulated to ensure that the temperature measured should have minimum error. All properties were assessed at the mean bulk temperature of the fluids for both tubes as well as shell side fluid (average of inlet and outlet temperatures). The uncertainty in the experimental set-up are identified and taken into considerations for the uncertainty analysis.

Parameters for experimentation

For the experimentation and the analysis of the conical coiled heat exchanger the parameters considered are listed in tab. 1.

Table 1. Parameters considered for analysis

Parameter	Details
Total number of coils	15
Coil mean diameter [mm]	200
Coil pitch [mm]	Outer diameter of the tube
Coils	Helical (0°), 45°, 90°, 135°, and spiral (180°)
Tube size inner diameter × outer diameter [mm]	8 × 10, 10 × 12, 12 × 15
Tube side fluid [°C]	Hot water of temperature 70±1%
Shell side fluid [°C]	Cold water of temperature 22±1%
Hot water flow rate [L per hour]	10 to 100, Re = 500-5000, De = 120-1200)
Cold water flow rate [L per hour]	30 to 90

Experimentation and analysis

Thermal analysis

The experimentation is carried out by keeping the cold water flow rate to 30 L per hour and varying the hot water flow rate 10-100 L per hour by the interval of 5 L per hour. For each reading, steady-state is achieved with constant hot and cold water flow rate. The same procedure is repeated for the cold water flow rate of 45, 60, 75, and 90 L per hour. Totally about 125 test runs are recorded for each coil.

All properties were assessed at the mean bulk temperature of the fluids for both tubes as well as shell side fluid (average of inlet and outlet temperatures).

Heat transfer to the cold water in the heat exchanger is calculated by the equation:

$$q_c = m_c c_{pw} (t_{c,o} - t_{c,i}) \tag{5}$$

Heat transfer to the hot water in the heat exchanger is calculated by the equation:

$$q_h = m_h c_{pw} (t_{h,o} - t_{h,i}) \tag{6}$$

The average heat transfer rate used in the calculation is determined from the hot water-side and cold waterside:

$$q_{avg} = \frac{q_c + q_h}{2} \tag{7}$$

The overall heat transfer coefficient was calculated from the inlet and outlet temperatures:

$$U_{ov} = \frac{q_{avg}}{A_i \, LMTD} \tag{8}$$

The calculation of *LMTD* is considered as per the coulter flow condition [16]:

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln \dfrac{\Delta T_1}{\Delta T_2}} \tag{9}$$

where ΔT_1 is the temperature difference between inlet temperature of hot water and outlet temperature of cold water, and ΔT_2 is the temperature difference between outlet temperature of hot water and inlet temperature of cold water.

Heat transfer coefficients for the outer tube, h_o, and for the inner tube side, h_i, are calculated using traditional *Wilson plots* method [7]. Wilson plots allow the heat transfer coefficients to be calculated based on the overall temperature difference and the rate of heat transfer, without the requirement of wall temperatures. This method is chosen to avoid the disturbance of flow patterns and heat transfer while attempting to measure wall temperatures. The analysis is extended to evaluate the Nusselt number and furthermore to evaluate the relationship between Nusselt number and Reynolds number or Dean number (flow parameter), Prandtl number (fluid parameter), and δ (geometric parameter).

Pressure drop measurement

In order to evaluate f-Re relationship for the conical coil configuration, the pressure drop, Δp, and average velocity, U, are measured. The friction factor is defined [17]:

$$f = \frac{1}{2} \frac{\Delta p}{L} \frac{d_i}{\rho U^2} \tag{10}$$

where the average velocity is evaluated from U (Q/A_c) and nominal cross-sectional area A_c ($\pi d_i^2/4$). In order to ensure complete steady-state, 15-20 minutes is allowed to elapse when the flow rate was changed.

Uncertainty analysis

Least counts and the sensitivities of the measuring instruments used in the present investigation contribute the errors in the analysis. Coleman and Steele [18] and *Measurement uncertainty* ANSI/ASME [19] has proposed the uncertainty analysis for all experi-

Table 2. Uncertainty analysis

Quantity	Uncertainty [%]
Volume flow rate cold fluid	1.66
Volume flow rate hot fluid	2.50
Reynolds number	2.51
Dean number	2.54
LMTD	2.32
Heat transfer	3.94
Heat transfer coefficient	6.10
Nusselt number	6.10
Friction factor	5.32

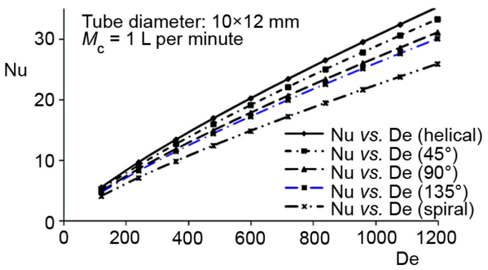

Figure 3. Variation of Nusselt number with Dean number

ments and by referring Kannadasan and Stelle [20], it was found that the experimental uncertainty is less than 7% for all the runs. The detailed uncertainty chart is as in tab. 2.

Result and discussion

Thermal analysis

The graphs represented in the results and discussions are of coil of size 10 × 12 mm and the shell side flow rate of 60 L per hour for the reference. The similar analysis is carried out for the coil size 8 × 10 and 12 × 15 mm, and shell side flow rate of 30, 45, and 90 L per hour for result confirmation.

Figure 3 shows the variation of Nusselt number with Dean number inside tube for different cone angle with same geometric parameters as same mean coil diameter, D_m, tube diameters, d_i and d_o, and same operating parameters as, flow rates, Q_h and Q_w, and constant inlet temperatures of both the fluids, t_{hi} and t_{ci}.

The analysis of fig. 3 indicates that Nusselt number increases with increase in inside Dean number. This indicates that, as flow rate inside tube increases the velocity increases, which increases the intensity of secondaries developed in the coiled tube [4], effects in increased Nusselt number. It is also observed that, for the same Dean number inside tube, Nusselt number is highest in helical coil, whereas for spiral coil it is lowest. In case of conical coils, Nusselt number decreases with increase in cone angle. In the analysis it is already proved that secondaries developed in the tube are function of coil diameter [5]. In helical coil heat exchanger the secondaries developed are of uniform intensity which shows proper mixing of fluid enhances the heat transfer. As cone angle increases from helical (0°) to spiral (180°) the intensity of secondaries developed are also considerably altered. In conical coil geometry, length of tube for below mean diameter is less than that of length of tube above. As cone angle increases the variation in lengths below and above mean diameter increases, which directly reduces the overall intensity of secondary flow developed in the coil. This reduction in intensity of secondaries reduces the Nusselt number. The second main reason may be due to compact structure of conical and spiral coil with smaller pitch. The minimum pitch in the coils may effects in shell side water confinement in the space between the successive coil rounds and semi-dead zones may observe around coil. As a result of this, in this region, the flow of shell side fluid may decelerate hence heat transfer coefficients and Nusselt number reduces [16]. This effect increases with increase in cone angle. As the fluid is flowing over tubes, the shell side flow observes streamline around the tubes effecting localised bulk heating of the water which reduces the temperature gradient in case of conical coil and furthermore it is increased in case of spiral coil. The effect of this may reduce temperature gradient, reducing Nusselt number for conical and furthermore to spiral coils. As the fluid is flowing over tubes the shell side fluid flow streamlined around the tubes effecting localised bulk heating of the water which reduces the temperature gradient in case of conical coil and furthermore in case of spiral coil. The effect of the reduced temperature gradient reduces the Nusselt number for conical coil and spiral coil.

Figure 4 shows variation of Nusselt number with Z (M_h/M_c) for same geometric and operating parameters. The figure indicates the analysis of tube size of 10 × 12 mm and cone angle of 90° for representation. The similar observations are recorded for all coils considered.

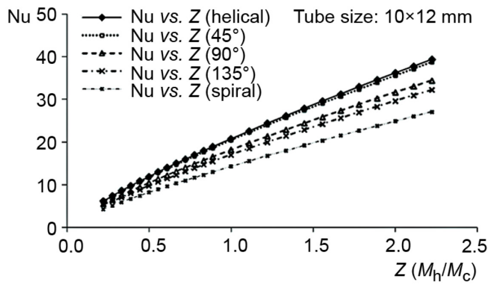

Figure 4. Variation of Nusselt number with Z (M_h/M_c)

The figure indicates that Nusselt number is a function of Z. This shows that Nusselt number is the directly proportional to tube side fluid, where as it is inversely proportional to shell side fluid. The figure also indicates that the slop of the graph reduces as cone angle changes from helical (0°) to spiral (180°)

Figure 5 shows variation of Nusselt number with cold water flow rate in the shell, Q_c, for constant flow rate inside the tube, same geometric and operating parameters. The figure indicates the analysis of tube size of 10 × 12 mm and cone angle of 90° for representation. The similar observations are recorded for all coils considered.

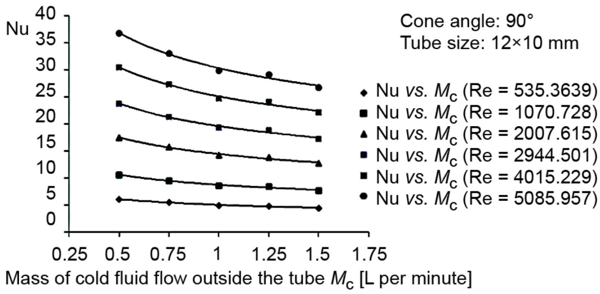

Figure 5. Variation of Nusselt number with cold water flow rate in shell, Q_c

Figure 5 indicates that Nusselt number inside the tube decreases with increase in shell side cold water flow rate, Q_c, for constant inside tube flow rate. As flow inside tube increases the graph shifted upward showing increase in Nusselt number. The figure also indicates that the slope of graphs increases with increase in Reynolds number inside the tube.

Figure 6 shows variation of Nusselt number with cone angle for constant Reynolds number inside the tube, for same geometric and operating parameters.

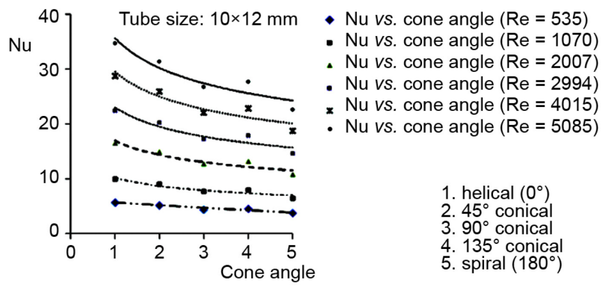

Figure 6. Variation of Nusselt number with cone angle, for constant Reynolds number inside the tube

The analysis shows that Nusselt number decreases from helical to conical, and as cone angle increases the Nusselt number decreases further more to spiral, for same Reynolds number. It is also observed that, Nusselt number is maximum for helical coil, where as it reduces with increase in cone angle up to spiral for a constant Reynolds number. It is also observed that, the variation in Nusselt number with respect to cone angle is grater for helical coil, whereas, as cone angle increases the variation reduces. This indicates that, the effect of secondary flow is predominant in helical coil, whereas as cone angle goes on increases the effect also reduces.

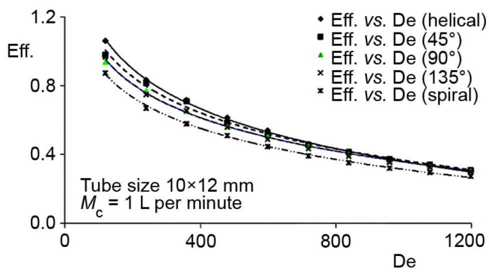

Figure 7. Variation of effectiveness with Dean number inside tube at different cone angles

The variation of effectiveness with Dean number inside tube at different cone angles is as shown in fig. 7, for same geometric and operating parameters.

Figure 7. indicates that the effectiveness of the heat exchanger decreases with increase in Dean number inside tube in all type of coils. From graph it is also evident that for lower inside tube flow rates, effectiveness is predominate as the slope of the graph is sufficiently high. But as flow rate increases the slope of the graph decreases, which indicates that at lower flow rates the coil type configurations are much more efficient. The analysis also indicates that the effectiveness of the helical coiled configuration is highest, whereas for spiral coiled configuration it is least, and for the conical coil it is in between the helical to spiral such that increase in cone angle reduces effectiveness.

From the analysis it is understood that effectiveness is the function of Z [20, 22], which is ratio of mass of hot water flowing in the tube to mass of cold water flowing in the shell. This can be correlated by using a simple power equation:

$$\varepsilon = a\,Z^b \tag{11}$$

where a and b are the constants and can be evaluated by regression analysis as shown in tab. 3. The shell-side water mass flow rate has favorable effect and tube side mass flow rate has adverse effect on effectiveness, ε, of the heat exchangers. Physically, more shell-side water flow means more heat extracted from the hot stream and therefore a greater temperature fall in that stream which translates into better effectiveness. On the contrary, more tube-side mass flow rate leads to less temperature fall in the hot stream and consequently lesser the effectiveness of the heat exchanger. The two mass flow rates, therefore act against each other with the same strength. Using the definition of the effectiveness, one can easily derive equations for predicting the tube-side and shell-side outlet temperature:

$$T_{\text{h,o}} = T_{\text{h,i}} - a\,Z^b\,(T_{\text{h,i}} - T_{\text{c,i}}) \tag{12}$$

$$\text{o} \tag{13}$$

Table 3. Constants for estimation of outlet temperature, eqs. (12) and (13)

Constants	Helical (0°)	45°	90°	135°	Spiral (180°)
a	0.5177	0.5047	0.4894	0.4624	0.4466
b	0.4114	0.4038	0.3833	0.3805	0.3804
R^2	0.9299	0.9042	0.9015	0.9217	0.9272

Figure 8 shows the variation of Nusselt number with inside tube fluid flow rate, Q_h, for coil of different tube diameter with same coil diameter and flow rates.

Figure 8 indicates the significant effect of tube diameter on the performance of the coiled tube type heat exchanger. For all type of tubes of different diameters, Nusselt number increases with increase in flow rate inside the tube, Q_h. The important observation from the figure is that, as tube diameter, d_i, increases, Nusselt number decreases for the same inside tube flow rate. This indicates that as tube diameter increases the curvature ratio, δ, increases which reduce the secondaries developed in the tube. The effect of reduction in secondaries reduces Nusselt number. Similar observations are recorded for all type of the coils considered for the analysis.

Figure 9 shows the variation of Nusselt number with outside tube fluid flow rate, Q_c, with coils of different tube diameter, d_i, with same mean coil diameter, D_m, and flow rates, Q_h.

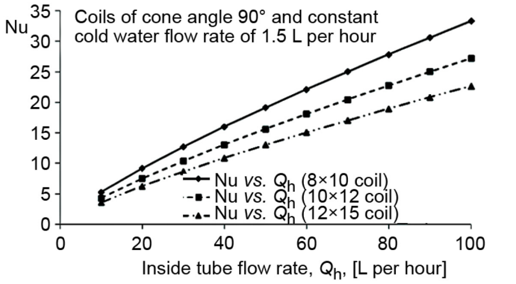

Figure 8. Variation of Nusselt number with inside tube fluid flow rate, Q_h, with coils of different tube diameter

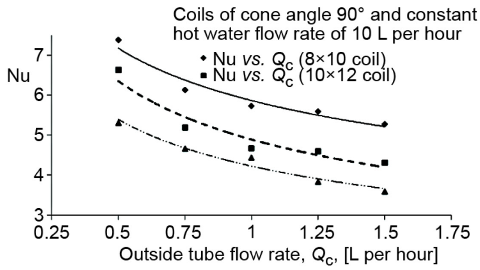

Figure 9. Variation of Nusselt number with outside tube fluid flow rate, Q_c, with coils of different tube diameter, d_i

The graph in the fig. 9 indicates that Nusselt number decreases with increase in outside tube fluid flow rate, Q_c, for all type of coils. The important observation recorded is that as tube diameter increases Nusselt number reduces for the same outside tube flow rate, Q_c, and inside tube flow rate, Q_h. This may be due to the compact structure of cone coil. This may be altered as pitch of the coil increased as helical coil heat exchanger [23]. Similar observations are recorded for the different cone angles from helical to spiral configuration.

Pressure drop analysis

In the experimentation the pressure drop analysis is carried out for all 15 tubes. The pressure drop analysis leads to analyse friction factor, *f*, which is the main characteristic of the hydrodynamic analysis. Figure 10 shows the variation of friction factor with Reynolds number inside the tube.

Figure 10 indicates that friction factor, *f*, is the function of Reynolds number (mass flow rate of the fluid flowing inside the tube) and it decreases with increase in Reynolds number. From graph it is also observed that helical coil is having the least value whereas as cone angle increases, friction factor increases for same Reynolds number. It is also observed that, the variation in friction factor with Reynolds number is sufficiently large at lower Reynolds number, and it reduces as Reynolds number increases. This indicates that at low Reynolds number secondaries are contributing more in friction factor, whereas as cone angle increases the variation is significant. As Reynolds number increases the variation of friction factor at various cone angle reduces.

Figure 11 shows the variation of friction factor, with Reynolds number at different tube diameters.

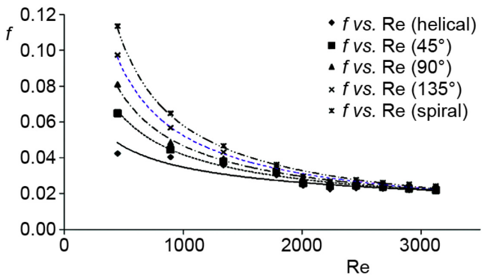

Figure 10. Variation of friction factor, *f*, with Reynolds number inside the tube

Figure 11 shows that friction factor is the function of tube diameter. For the same flow rate in bigger diameter tubes the velocity in the tube reduces which reduces the Reynolds number, increasing friction factor. As tube diameter increases for the same flow rate friction factor reduces. For the same Reynolds number smaller tube diameter has less friction factor whereas as tube diameter increases friction factor increases.

In the analysis of pressure drop it is observed that friction factor is a function of velocity Reynolds number, the geometric parameter (curvature ratio, δ), and cone angle of the coil. The friction factor is correlated with Dean number and δ by a power equation:

$$f = c_1 \,\mathrm{Re}^{c_2}\, \delta^{c_3} \tag{14}$$

where c_1, c_2, and c_3 are constants and can be evaluated by regression analysis. The values of constants are as in tab. 4.

Table 4. Constants for friction factor, *f*, correlation, eq. (14)

Constants	Helical (0°)	45°	90°	135°	Spiral (180°)
c_1	6.67	31.70	85.36	237.00	556.24
c_2	–0.4	–0.54	–0.62	–0.7	–0.767
c_3	0.879	1.03	1.15	1.28	1.39
R^2	0.8640	0.9644	0.9700	0.9801	0.9767

Figure 12 shows the comparison of friction factor, *vs.* Reynolds number The variation is ±15%. The equation coefficient, R^2, presented in tab. 3 are greater than 0.85, indicated good agreement of experimental and predicted values. Our correlation for helical coil are also in good agreement with correlations published in literature, (Ito [24], Shrinivsan [25], with variation up to ±13%).

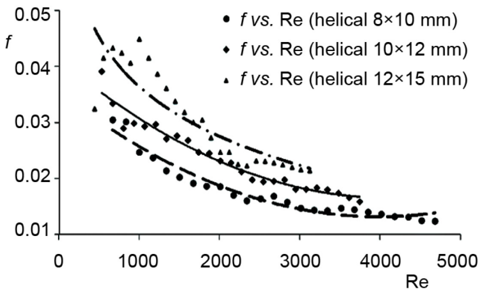

Figure 11. Variation of friction factor, *f*, with Reynolds number at different tube diameters

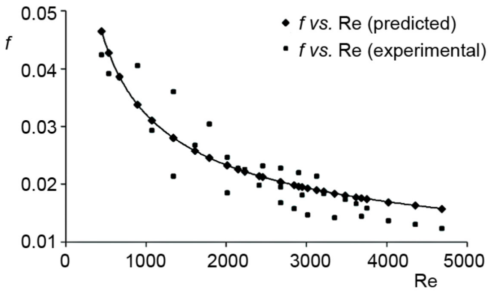

Figure 12. Comparison of friction factor, *f*, *vs.* Reynolds number for predicted and experimental values

Nusselt number correlation

The experimental analysis indicates that performance of the heat exchanger depends on flow rates, Q_h and Q_c, properties of the fluid and the geometric parameters as tube diameter, d_i, and coil diameter, D. The different parameters reveals in the form of Dean number, Prandtl number at bulk temperature, and the curvature ration, δ. The statistical analysis of the measured data of 15 different coils from helical to spiral for different cone angles is considered for finding the relation:

$$\mathrm{Nu} = a\, \mathrm{De}^b\, \mathrm{Pr}^c\, \delta^d \tag{15}$$

where *a, b, c,* and *d* are constants. The analysis is carried out with statistical tool by using least square criteria. The constants *a, b, c,* and *d* are calculated for different cone angles and given in tab. 5.

Table 5. Constants of the Nusselt number correlation, eq. (15)

Constants	Helical (0°)	45°	90°	135°	Spiral (180°)
a	0.57	0.56	0.54	0.52	0.50
b	0.8	0.8	0.8	0.8	0.8
c	0.6267	0.6022	0.5877	0.5625	0.5317
d	1.1565	1.1597	1.17396	1.1762	1.1794
R^2	0.9180	0.90017	0.9357	0.9432	0.9780

The comparison between the predicted Nusselt number from the proposed correlation and the experimental Nusselt number are given in fig. 13. The Nusselt number predicted from the correlation shows the variation of ±7% with the experimental Nusselt number. In the literature it is observed that many researchers have given the different correlations for helical coil in the range of experimentation. The comparison in various correlations given by researchers with the correlation derived from the experimental work for helical coil are presented in fig. 14. The figure indicates the good agreement with the available literature. There is no any correlation available in the literature for the conical coil as well as spiral coil heat exchanger for the range of experimentation.

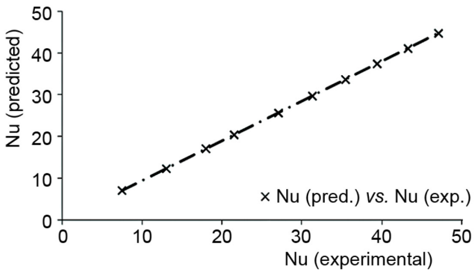

Figure 13. Variation of predicted *vs.* experimental Nusselt number

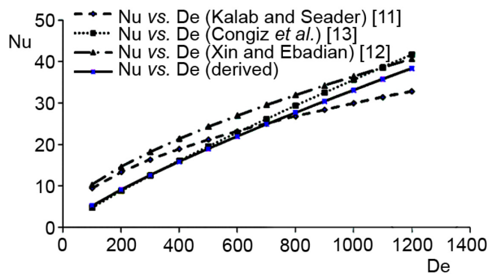

Figure 14. Comparison of various correlations with proposed for helical coil heat exchanger (variation of Nusselt number *vs.* Dean number)

Conclusions

The heat transfer and pressure drop characteristics of the conical coil heat exchanger with different cone angles, tube diameter, and flow rates are detailed in this paper. In this conical coil with cone angle of 0° (helical), 45°, 90°, 135°, and 180° (spiral) coils are analysed experimentally. The investigation leads to following conclusions for the present study:

- The Nusselt number increases with increase in Reynolds number inside tube, for constant outside (shell side) cold water flow. It is highest for helical coil, whereas least for spiral coil, and for conical coil it reduces with increase in cone angle.
- The Nusselt number is a function of *Z*, which shows that, it is directly proportional to hot water flow in the tube whereas inversely proportional to cold water flow outside the tube.

- Effectiveness, ε, of coiled tube heat exchanger is a function of Reynolds number inside tube and it reduces with increases in Reynolds number. The helical coil has a maximum ε, whereas minimum in case of spiral coil and in conical coil, as cone angle increases ε decreases from helical to spiral.
- Nusselt number is a function of tube diameter, d_i, and it decreases with increase in tube diameter for the same inside tube flow rate, Q_h, and outside tube flow rate, Q_c.
- The friction factor, f, is the function of Reynolds number and it reduces with increase in Reynolds number. Helical coil has a least value of f whereas it is maximum in case of spiral coil. As cone angle increases the f increases for the same Reynolds number. As Reynolds number increases the f decreases. The variation is significant in lower Reynolds number, whereas the variation reduces as Reynolds number increases.

The relationship between effectiveness, ε, and Z have derived for the experimental data and empirical correlations are presented for predicting the outlet temperature hot and cold side fluids with correlation coefficient, R^2, in each case above 0.90.

The friction factor, f, is correlated in terms of Reynolds number and δ, with power equations.

Empirical correlations for Nusselt number at different cone angles from helical to conical with different cone angles and spiral coil heat exchanger are developed and the correlation coefficient, R^2, in each case is above 0.90. The Nusselt number obtained for helical coil correlation is compared with the correlations from the literature, which shows good agreement [11-13], whereas correlations are not available in literature for comparison with different cone angles.

Nomenclature

$a, b, c, d, n, c_1, c_2, c_3$ – constants
A_c – cross section area of pipe, [m^2]
D – coil diameter, [m]
d – tube diameter, [m]
De – Dean number
Dm – mean coil diameter, [m]
f – friction factor
h – heat transfer coefficient, [Wm^{-2}K^{-1}]
k – thermal conductivity, [Wm^{-1}K^{-1}]
L – length of tube, [m]
M – mass flow rate, [kgs^{-1}]
Nu – Nusselt number
Pr – Prandtl number
Δp – pressure drop, [ ]
Q – flow rate, [Lph]
q – heat transfer rate, [W]
R – coil radius, [m]
Re – Reynolds number
t – temperature, [K]
U – overall heat transfer coefficient, [Wm^{-2}K^{-1}]
Z – mass flow rate ratio

Greek symbols

δ – curvature ratio
ε – effectiveness
θ – cone angle, [°]

Subscripts

avg – average
c – cold fluid
h – hot fluid
i – inside tube condition
o – outer tube condition
ov – overall
w – water

Abbreviation

LMTD – log mean temperature difference

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