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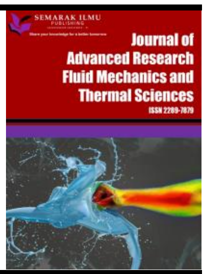

Suggestion of Correlations to Obtain Shell and Tube Side Nusselt Numbers in Coiled Heat Exchangers

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ABSTRACT

In coiled tube heat exchangers increase in curvature ratio and effective contact of shell fluid with coil surface led to better thermal characteristics. This work addresses study of thermal performance, shell and tube heat transfer coefficients (h_o and h_i). Also, mathematical correlations are predicted to obtain shell and tube Nusselt numbers (Nu_o and Nu_i). In the designing of compact coiled heat exchangers consisting of straight helical coil (SHC), conical coil (CC) and spiral coil knowledge of Nu_o and Nu_i is necessary whereas mathematical correlation to calculate Nu_o considering variations in shell geometry is not found. Five heat exchangers consisting of three cone coils (angle, Θ = 30°, 50° and 70°), spiral coil (Θ = 0°) and SHC (Θ = 90°) are tested. Using Wilson plot method h_o and h_i are obtained for wide range of tube side Reynolds numbers (Re_i = 3700 - 21000). It is observed that, highest h_o is obtained for Θ = 30°CC with better thermal performance. Lowest h_o is obtained for Θ = 90° SHC. Ratio of height of shell, H_s and diameter of shell, D_{so} is considered in prediction of single correlation as: $Nu_o = 16.55.Re_o^{0.55}.Pr_o^{0.4} *(H_s/D_{so})^{0.096}$. Comparison with experimental findings shows variation of 0-14%. Similarly two correlations are proposed as: $Nu_i = 0.0512.Re_i^{0.80}.Pr_i^{0.4}.CR_{ave}^{0.18}$ and $Nu_i = 0.0365.De_i^{0.815}.Pr_i^{0.4}.CR_{ave}^{-0.3}$. A fair agreement is found with existing researchers.

1. Introduction

Coiled tube heat exchangers are used in the field of heat recovery units, chemical industries, boilers, refrigeration and solar systems. In helical coils curvature ratio causes generation of secondary flow which further leads to increase in tube side heat transfer coefficients. Helical coils have better heat transfer coefficients than straight tubes and are best suitable where enough space is not available. In straight helical coils the curvature ratio remains constant along the vertical axis. In spiral coils the curvature ratio varies along the horizontal axis whereas in cone coils the curvature ratio increases from a base to apex. In the manual design of conical coil heat exchanger (CCHE), for known values of mass flow rates and inlet temperatures of both fluids, to achieve desired outlet

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temperature of one of the fluids the task is to calculate required surface area. This surface area is written as

$$A = \frac{Q}{U.LMTD} \tag{1}$$

where Q is the heat duty to be achieved and log mean temperature difference, LMTD is obtained from inlet and outlet temperatures. U is overall heat transfer coefficient and given as

$$\frac{1}{U_i.A_i} = \frac{1}{h_i.A_i} + R_t + \frac{1}{h_O.A_O} \tag{2}$$

Without the knowledge of tube side heat transfer coefficient (h_i) and shell side heat transfer coefficients (h_o), overall heat transfer coefficient U cannot be obtained. Due to which calculation of Area, A is not possible. Rose [1] discussed the traditional Wilson plot method to obtain inner and outer heat transfer coefficients. In this method measurement of wall temperature of coil is not necessary which otherwise may create disturbance to flow patterns. In Eq. (2), R_t as conductive thermal resistance of the tube can be calculated. As per the Wilson plot method, it is assumed that if tube side mass flow rate is kept constant, h_i remains constant. Further for this constant tube side mass flow rate, if mass flow rate of shell fluid is varied in the steps, h_o is directly proportional to shell side fluid velocity, v_s^n. Now h_i and h_o are written in Eq. (3) and Eq. (4).

$$\frac{1}{h_i.A_i} + R_t = C_1 \tag{3}$$

$$h_o = C_2.\, v_s^n \tag{4}$$

Eq. (2) can be written as

$$\frac{1}{U_i.A_i} = C_1 + \frac{1}{C_2.A_O.v_s^n} \tag{5}$$

If $1/U_iA_i$ is plotted against $1/v_s^n$ for variation in v_s, for proper selection of values of n, a straight line fit can be obtained through the data points of the plot. From this straight line fit, y intercept is obtained as C_1 and slope as m. From the values of C_1 and m, h_i and h_o are obtained as

$$h_i = \frac{1}{(A_i\,.(C_1 - R_t))} \tag{6}$$

$$C_2 = \frac{1}{m\,.\,A_0}\ ; \tag{7}$$

$$h_o = C_2\,.\,v_s^n \tag{8}$$

Rennie and Raghavan [2] experimentally studied two double pipe helical heat exchangers in which the size of the inner tube was varied. Heat transfer coefficients were obtained by using Wilson plot in which mass flow rate through tube was kept constant assuming h_i remained constant. In this way 60 h_i were obtained from 60 Wilson plots. Fernandez-Seara *et al.,* [3] presented review of application of Wilson plot method to obtain convective heat transfer coefficients in heat exchangers. It was concluded that, if the mass flow rate of cooling liquid was modified then change in overall heat transfer coefficients would be due to variation in thermal resistance of cooling liquid and other thermal resistances remained constant. Kumar *et al.,* [4] studied double pipe helical coil heat exchanger in which tube side and annulus side heat transfer coefficients were obtained by using Wilson plots. While using Wilson plots, h_i were obtained by keeping shell side mass flow rates constant, resulting h_o constant. In a similar way for calculation of h_o tube side mass flow rate kept constant and shell side mass flow rate was varied. Shokouhmand *et al.,* [5] studied three straight helical coil heat exchangers (SHCHE) and obtained convective heat transfer coefficients using Wilson plot technique. Mass flow rate in tube was kept constant and shell side mass flow rate was varied in five steps. This resulted in 30 Wilson plots, 30 h_i and 150 h_o. Hydraulic diameters were considered as a characteristic dimension in the calculation of Re_o. As Re_o was increased Nu_o also increased. It was also predicted that, for higher curvature ratios, higher tube side Nusselt no. were obtained. Salimpour [6] experimentally studied three straight helical coils keeping coil diameter fixed and varying tube diameter, pitch and length. Mass flow rate in the tube was kept constant and shell side mass flow rate was varied to get 15 Wilson plots. From these Wilson plots 15 h_i and 75 h_o were obtained. Also, dimensionless pitch, γ was considered to predict Nu correlations. Jamshidi *et al.,* [7] applied Wilson plot method to determine heat transfer coefficients for SHCHE. Shell side mass flow rate was kept constant and assumed that h_o remained constant. For three coil diameters, three pitches, three tube side mass flow rates and three shell side mass flow rates, the L_9 orthogonal array was used in Taguchi method. It was predicted as: a) U_o increased with increase in Nu_o b) as coil pitch was increased, Nu_i decreased but Nu_o increased. Purandare *et al.,* [8] experimentally investigated coiled heat exchangers. On the basis of variation in apex angle five types of coils were studied as: 45°, 90°, 135°, 0° (helical), 180° (spiral). All coils were accommodated one by one in the same shell. h_i and h_o were obtained using Wilson plot method. Effectiveness, tube side friction factor and Nu_i were obtained and correlations were proposed. Sheeba *et al.,* [9] experimentally and numerically investigated conical coil double pipe heat exchangers. Experimentation was carried out for cone apex angle of 72° and heat transfer coefficients were obtained by using Wilson method. In the application of Wilson plot, mass flow rate in inner tube was kept constant and assumed that variation in mass flow rate in annulus caused variation in h_o keeping h_i constant. Also, numerical simulation was carried for 72° cone apex angle and following correlation was predicted.

$$\mathrm{Nu_i} = 0.4565\,.\,\mathrm{De_i}^{0.7645}.\,\mathrm{Pr_i}^{-0.4786}\ \mathrm{For}\ 90 \ll \mathrm{De_i} \ll 1000\ \mathrm{and}\ 3 \ll \mathrm{Pr_i} \ll 4$$

Alklaibi *et al.,* [10] experimentally studied horizontal single SHCHE in which hot water was flowing through the shell. Ethylene glycol and water mixture based Fe_3O_4 nanofluid was flowing through the coil as a coolant. Wilson plot method was used to evaluate hi, by keeping shell side mass flow rate constant (h_o constant). Nu_i and friction factor correlations were proposed considering Dean number, nanoparticles concentration and Prandtl number. Patil *et al.,* [11] explained the design procedure of SHCHE to obtain required surface area for necessary heat duty. Nu-RePr correlations for calculation of h_i and h_o were presented. Following correlation was used to obtain shell side Nusselt No, Nu_o.

$\mathrm{Nu_o} = 0.6.\,\mathrm{Re_0^{0.5}}.\,\mathrm{Pr_0^{0.31}}$ For Re_o in the range of 50-10000.

Ali [12] experimentally studied natural convection in SHCHE and a constant temperature bath was maintained in the shell. Following Rogers and Mayhew [13] correlation for turbulent flow was used to obtain h_i.

$$\mathrm{Nu_i} = 0.023.\mathrm{Re_i^{0.85}}.\mathrm{Pr_i^{0.4}}.(\mathrm{d_i/D})^{0.1} \tag{9}$$

Also, height, H and length of coil, L_c and d_o are considered as characteristic dimensions in the calculation of Re_o. Further Nu_o correlations were developed. Kakaç *et al.,* [14] discussed Nu_i correlation given by Manlapaz- Churchill for laminar and turbulent flow in helical coils on the basis of constant temperature and constant heat flux boundary conditions. Also, for turbulent flow, Schmidt correlation was presented as

$$\frac{\mathrm{Nu_c}}{\mathrm{Nu_s}} = 1.0 + 3.6.[\left(1 - \frac{\mathrm{d_i}}{\mathrm{D}}\right)].\left(\frac{\mathrm{d_i}}{\mathrm{D}}\right)^{0.8} \text{ for } 2\text{ x}10^4 < Re_\mathrm{i} < 1.5\text{ x}10^5, 5 < {}^{\mathrm{d_i}}/_{\mathrm{D}} \leq 84$$

In above correlation Nu_c and Nu_s stands for Nusselt number for coil and straight tube respectively. Prabhanjan *et al.,* [15] experimentally studied the effect of water bath on straight helical coil (SHC). It was found out that, h_i of SHC is 1.16 and 1.43 times better than straight tube. Similarly, Naphon and Suwagrai [16] studied the effect of curvature ratio over heat transfer for spiral coils considering 3 curvature ratios. Constant wall temperature was considered on the outer surface. It was predicted that Nu_i obtained for spiral coils were found to be 1.49 times higher than Nu for a straight tube. Jayakumar *et al.,* [17] numerically and experimentally studied straight helical coils considering various boundary conditions. Schmidt, 1967 formulation was used to determine critical Renault number, Re_{crit}. Re_i above Re_{crit} is considered as transition of fluid from laminar to turbulent regime. While estimating following Nu_i correlation index of Pr_i was selected as 0.4 and this correlation is proposed as

$$\mathrm{Nu_i} = 0.025.\mathrm{De_i^{0.9112}}.\mathrm{Pr_i^{0.4}} \qquad \text{For } 2000 < \mathrm{De_i} < 12000$$

Salimpour [18] studied three straight helical coils with constant coil diameter. Hot oil flowing through coils was cooled by cold water flowing in the shells. For the comparison of the results, the following correlation proposed by Dravid *et al.,* [19] was considered.

$$\mathrm{Nu_i} = (0.65.\mathrm{De_i^{0.05}}+0.76).\mathrm{Pr_i^{0.175}}$$

Also, correlation was predicted for fluids having temperature dependent properties. Ghorbani *et al.,* [20,21] experimentally studied mixed convection in three straight coil heat exchangers considering a combination of two tube diameters, two coil diameters and two pitches. Shell side dimensions were kept constant for all heat exchangers. Nu_i for laminar and turbulent regimes were calculated by using the following Eq. (10) and Eq. (11) respectively.

$$\mathrm{Nu_i} = [\left(\frac{48}{11} + \frac{{}^{51}/_{11}}{1+\frac{1342}{\mathrm{Pr_i.He^2}}}\right)^3 + 1.816.\left(\frac{\mathrm{He}}{1+\frac{1.15}{\mathrm{Pr_i}}}\right)^{1.5}]^{0.333} \tag{10}$$

$$\mathrm{Nu_i} = 1 + 3.6.\left(1 - \frac{\mathrm{d_t}}{\mathrm{D_c}}\right).\left(\frac{\mathrm{d_t}}{\mathrm{D_c}}\right)^{0.8}.[0.0023.\mathrm{Re_i^{0.8}}.\mathrm{Pr_i^{0.4}}] \tag{11}$$

Two correlations were predicted to estimate Nu_o on the basis of D_{hx} and D_{eq}. Moawed [22] experimentally studied forced convection from the outside surface of straight helical coils. Constant heat flux was applied to the coil and air was forced to circulate over the outside surface of coils. Air side heat transfer coefficient was calculated as h_o = $Q/T_{sm} - T_\infty$ where T_{sm} and T_∞ were considered as surface temperature of coil and air stream temperature respectively. Following correlation was predicted.

$$\mathrm{Nu_o} = 0.0345.\,\mathrm{Re_o^{0.48}}.\,({}^{\mathrm{D}}/{}_{\mathrm{d_o}})^{0.914}.\,({}^{\mathrm{P}}/{}_{\mathrm{d_o}})^{0.281} \text{ for } 6.6 \text{ x } 10^2 \leq \mathrm{Re_o} \leq 2.3 \text{ x} 10^3$$

$$7.086 \leq {}^{\mathrm{D}}/{}_{\mathrm{d_o}} \leq 16.142 \text{ and } 1.81 \leq {}^{\mathrm{P}}/{}_{\mathrm{d_o}} \leq 3.205$$

Sobota [23] discussed various correlations proposed by researchers to calculate Nu_i. Out of these, Schmidt [24], Mori and Nakayama [25], Rogers and Mayhew [13], and Hewitt *et al.,* [26] are given below. These correlations do not involve terms which depend upon wall temperature. Schmidt [23] correlation for flow in laminar and turbulent regime

$$\mathrm{Nu_i} = 3.65 + 0.08.\left[1 + 0.8.\left(\frac{\mathrm{d_i}}{\mathrm{D}}\right)^{0.9}\right].\,\mathrm{Re_i}^{\left[0.5+0.2903\left(\frac{\mathrm{d_i}}{\mathrm{D}}\right)^{0.194}\right]}.\,\mathrm{Pr_i}^{1/3}$$
$$\text{for } 100 < \mathrm{Re_i} < \mathrm{Re_{crit}}$$

$$\mathrm{Nu_i} = 0.023.\left[1 + 14.8.\left(1 + \frac{\mathrm{d_i}}{\mathrm{D}}\right).\left(\frac{\mathrm{d_i}}{\mathrm{D}}\right)^{1/3}\right].\,\mathrm{Re_i}^{\left[0.8-0.22\left(\frac{\mathrm{d_i}}{\mathrm{D}}\right)^{0.1}\right]}.\,\mathrm{Pr_i}^{1/3} \tag{12}$$
$$\text{for } \mathrm{Re_{crit}} < \mathrm{Re_i} < 22000$$

Mori and Nakayama's [24] correlation

$$\mathrm{Nu_i} = \left[\frac{1}{41.0}\right].\,[1 + 0.061/(\left(\frac{\mathrm{d_i}}{\mathrm{D}}\right)^{2.5}.\ \mathrm{Re_i})^{0.167}].\left(\frac{\mathrm{d_i}}{\mathrm{D}}\right)^{\frac{1}{12}}.\,\mathrm{Pr_i}^{0.4}.\,\mathrm{Re_i}^{5/6} \tag{13}$$

Rogers and Mayhew [13], and Hewitt *et al.,* [26] correlation for turbulent flow

$$\mathrm{Nu_i} = (1 + 3.5.\left(\frac{\mathrm{d_i}}{\mathrm{D}}\right)).0.023\,.\,\mathrm{Re_i}^{0.80}.\,\mathrm{Pr_i}^{0.333} \tag{14}$$

Also, Schmidt [23] formula of Nu_o was discussed for tube and tube helical coil heat exchanger. It is given as

$$\mathrm{Nu_o} = \left(1 + 3.5\,.\left(\frac{\mathrm{D_{eq}}}{\mathrm{D}}\right)\right).\,[3.66\ + 1.2\,.\left(\frac{\mathrm{d_1}}{\mathrm{d_{2in}}}\right)^{-0.8} + 1.6.\,(\mathrm{Re_o}.\,\mathrm{Pr_o}.\left(\frac{\mathrm{D_{eq}}}{\mathrm{H_s}}\right))^{0.33}] \tag{15}$$

where d_1 is the outer diameter of the tube and d_{2in} is the inner diameter of annulus. Ke *et al.,* [27] performed numerical simulation of cone coil for variation of cone angle in the range of 55° to 85° keeping base diameter of cone as 250 mm. Three turns were fixed and accordingly the upper radius was varied in the range of 40 to 120 mm. It was predicted that, compared to cone angle helical pitch had a small effect over heat transfer. Elazm *et al.,* [28] investigated that, cone angle and geometry of conical coils had significant effect over coil exit temperature and heat transfer and predicted that, conical coils were found to be better than SHC. Flórez-Orrego *et al.,* [29] performed experimental and CFD study of single horizontal conical coils having zero space between turns. Experimentally h was calculated usingh $= \mathrm{Q/A}.\,\Delta\mathrm{T_{avg}}$. After calculating Nu_i correlation for cone coil was proposed as

$$\mathrm{Nu_i} = 0.00797.\,\mathrm{Re_i}^{0.82}.\,\mathrm{Pr_i}^{0.4} \text{ for } 4300 \ll \mathrm{Re_i} \ll 18600 \text{ and } 2 \ll \mathrm{Pr_i} \ll 6$$

Also, Nu_i obtained from experimental and numerical simulation was compared with Nu_i calculated using Seban and McLaughlin [30] and Xin and Ebadian's [31] correlations. Seban and McLaughlin's [30] correlation is presented below.

$$\mathrm{Nu_i} = 0.023\,.\,\mathrm{Re_i}^{0.80}.\,\mathrm{Pr_i}^{0.4}\left(\mathrm{Re_i}^{\frac{1}{20}}.\left(\frac{\mathrm{d_i}}{\mathrm{D}}\right)^{0.01}\right) \tag{16}$$
$$\text{for } 5000 \ll \mathrm{Re_i} \ll 10^5 \text{ and } \mathrm{Pr_i} = 5$$

Genić *et al.,* [32] experimentally investigated three coiled heat exchangers. Re_o was calculated on the basis of D_{hx} and the following correlation was developed for calculation of Nu_o.

$$\mathrm{Nu_o} = 0.50.\,\mathrm{Re_o^{0.55}}.\,\mathrm{Re_o^{0.55}}.\left({}^{\mu}/_{\mu_w}\right)^{0.14}$$

$$\text{for } 10^3 \le \mathrm{Re_o} \le 9\,\mathrm{x}10^3\,, 2.6\ \le \mathrm{Pr} \le 6 \text{ and } 9.1\ \le \mathrm{D}_{hx} \le 18.3$$

Pawar *et al.,* [33] experimentally studied two straight coiled heat exchangers with variation in coil diameter and number of turns with a bath of shell side fluid. For calculation of Nu_i, Rogers and Mayhew [13] correlation (Eq. (9)) was used. It was predicted that Nu_i, h_i and U_o were found to be higher for lower coil diameter. Alimoradi [34] studied SHCHE and correlation for effectiveness was predicted. Ghorbani *et al.,* [20,21] and Purandare *et al.,* [8] developed modified effectiveness correlations based on the ratio of $R_m = m_c/m_s$. In this type of correlation Alimoradi [34] included group of non-dimensional parameters as: (D/d_i), (D_s/d_i), (H_c/d_i), (H_s/d_i), (P/d_i) and (f/d_i). It was concluded that, for $R_m = 1$, effectiveness was found to be minimum. Jamshidi and Mosaffa [35] studied finned cone coil geothermal heat exchanger according to the environmental condition of City Tehran. According to Taguchi method 8 sets of heat exchangers were considered from two variations done in each coil diameter, pitch, angle, number of fins, fin ratio and Reynolds number. Chandrasekar and Kumar [36] experimentally studied heat transfer in single horizontal straight double helical coil using MWCNT/Water nanofluids. A constant heat flux condition was considered. Daghigh and Zandi [37] performed experimentation on CC, SHC and conical- cylindrical- spiral coils in which diameter and height of coils was varied keeping d_i and P constant. Heat transfer coefficient was calculated from the formulation containing Q, A and LMTD. Results were validated by using correlations suggested by Salimpour [7], Dravid *et al.,* [19], and Kalb and Seader [38]. It was predicted that, cylindrical-conical-spiral coil showed better heat transfer characteristics than other coils. Palanisamy and Kumar [39] experimentally studied single horizontal conical coil heat exchangers having cone angle of 8°. Multiwall carbon nanotube / water nanofluids were used. Q, h_i, Nu_i and ΔP_i were studied. Ali *et al.,* [40] performed numerical simulation to analyse double pipe conical coils. Heat transfer was studied through the annulus side for laminar and turbulent flow regimes and correlations were suggested for prediction of Nu and friction factor. Also, it was predicted that, as cone angle at apex varied from 0° to 90° friction factor and Nu increased by 15.51% and 37.7% respectively. Al-Salem *et al.,* [41] numerically investigated tube in tube conical coil heat exchanger considering apex of cone coil facing towards ground. It was predicted that, as cone angle at apex varied from 0°to 65° coefficient of exergy performance and exergy efficiency decreased by 11.1% and 32.56% respectively. Heyhat *et al.,* [42] experimentally studied conical tubes under usage of SiO_2/water nanofluids. Constant heat flux was provided at the outer surface. It was predicted that variation in cone apex angle was more effective than variation in coil pitch.

Maghrabie *et al.,* [43] investigated the effect of inclination angle of SHC from horizontal position (0°) to vertical position (90°). h_i was calculated using formulation between Q, A and ΔT. U_o, Nu_i and effectiveness were studied. Considering De_i and inclination angle, Θ_i equations were predicted to estimate Nu_i and ΔP_i. Sharma *et al.,* [44] numerically and experimentally investigated a conical shaped cavity receiver in solar power applications in which a cone coil was provided with coating of nano structured carbon florets (NCF). It was predicted that, for a cone angle of 50°and radial peach diameter ratio equal to 1, maximum efficiency was observed.

Chokphoemphun *et al.,* [45] experimentally studied SHCHE kept in the free board zone of a rice husk fluidized bed combustor. It was proposed that maximum heat transfer was observed for wrapping of steel wire at full length of tube instead of wrapping at two and four positions. Also, Missaoui *et al.*, [46] numerically studied helical coil immersed in heat pump water heater and investigated effect of pitch of coil and storage tank dimensions over heating process. Hasan *et al.,* [47] numerically analyzed SHC using water based nanofluids. Variation in geometrical parameters was studied as multiple head ribbed geometry with coil revolutions. For comparison of Nu_i, Manlapaz-Churchill's correlations of Nu_i (Eq. (10) and Eq. (11)) on the basis of constant temperature boundary conditions were used. Missaoui *et al.*, [48] numerically and experimentally compared normal coil with coil having variation in pitch and found out that average heat transfer coefficients of coil having variation in pitch was 16.7 % higher than normal coil. Omri *et al.,* [49] carried out experimentation to study enhancement of thermal performance of single straight helical coil using flow of nanofluids in laminar regime. Hot distilled water based CuO-GP (80 20%) hybrid nanofluid was flowing through the coil and cold water was allowed to flow in the shell. For the calculation of h_i, Seider-Tate correlation was used. Abdullah and Hussein [50] presented a review of nanofluids and its effect on thermal performance of coiled heat exchangers. Experimental and numerical analysis of straight helical and conical coils using nanofluids was reviewed. Missaoui [51] analysed three types of coils as straight helical coil with constant pitch, conical coil with constant pitch, straight helical coil with variable pitch and predicted that straight helical coil with variable pitch had better thermal performance.

In general, it is observed that approaches used to calculate h_i and h_o are: 1) use of Nu-RePr correlations. 2) For known value of experimentation is carried out and values of Q, LMTD and U are obtained. Then h_i is obtained from Nu-RePr correlations and h_o is further calculated. 3) Experimentation is carried out to know values of Q, LMTD and U. h_i and h_o are obtained using Wilson Plot technique. In comparison with cone coils and spiral coils the majority of work is contributed in the area of deriving Nu_i correlations for straight helical coils. Also, researchers have undertaken various operating conditions like shell side fluid bath, constant temperature and constant heat flux boundary conditions, fixed shell dimensions, applications of heat exchanger [8,12,14,16,17,20-22,33,35,37,39,42,45]. Very less amount of work is found in the area of application of Wilson plot to analyze forced convection in cone and spiral coils. Also, enough work is not found on calculation of h_o for spiral and conical coil heat exchangers. In addition to this for forced convection taking place in conical, spiral and straight helical coils derivation of single mathematical correlation to obtain Nu_o using Wilson plot method is not found. Few researchers developed Nu_i correlations but separately for SHC, CC and spiral coils. In addition, sufficient work is not found in the area of comparative analysis of straight, conical and spiral coils on the basis of variation in CR and geometry (by varying angle Θ).

Hence it is decided to use Wilson plot method to obtain h_i and h_o for straight helical, conical and spiral coil in which forced convection is taking place between coil and shell fluids. Additionally, while using Wilson plot method it is decided to keep mass flow rate of coil fluid constant and mass flow rates of shell fluid is varied in steps to get maximum number of h_o. To analyze this coil diameter of straight helical coil (Θ=90°) is considered equal to 0.07m. For spiral and cone coils smaller end diameter is considered as 0.07m and for fixed length of coils, variation in bigger end diameters is obtained for variation of angle Θ in the steps of 70°, 50°, 30° (CCs) and 0° (spiral coil). Due to this wide range of CR is achieved which is used to propose Nu_i correlations considering all coils at the same time. To have better comparisons tube side and shell side volumes are kept respectively same in all five heat exchangers. As a result of this variation in shell geometry is done which is considered in the derivation of single Nu_o correlation for all heat exchangers. These correlations may be used in the designing of coiled heat exchangers. Schematic diagram of conical coil heat exchangers is shown in Figure 1.

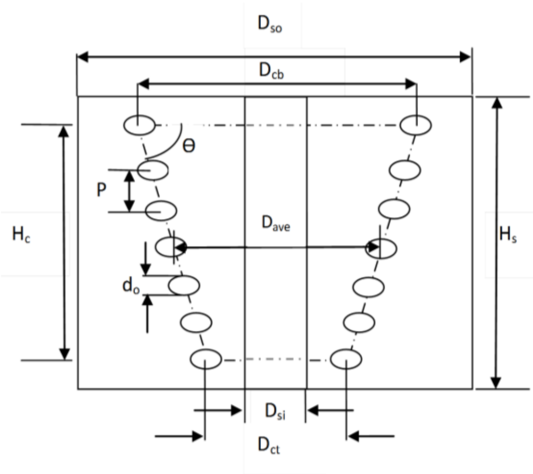


Fig. 1. Schematic diagram of conical coil heat exchanger

2. Wilson Plot Method and Heat Transfer Coefficients

In this study, hot water and cold water are forced through coil and shell respectively. It is decided to vary the mass flow rates of both the fluids into four steps. Mass flow rate of coil fluid is fixed at some specific value and mass flow rate of shell fluid is varied in four steps. After plotting Wilson plot for this case single h_i and four h_o are obtained. Thus, for the entire range of coil hot water and shell cold water mass flow rates 8 h_i and 32 h_o are obtained for parallel and counter flow arrangements. Here 05 heat exchangers are studied on the basis of variation CR and geometry. Thus, for all heat exchangers 40 Wilson plots are obtained. Also, most of the researchers have obtained Wilson plots in the range of 15-60. Thus 40 h_i and 160 h_o are obtained. During experimentation inlet-exit temperatures, mass flow rates of both fluids are recorded and energy balance equation is written as:

$$Q_{ch} = (mCp)_{ch}.(T_{hi} - T_{ho})\ ,\ Q_{ch} = Q_{sc} = (mCp)_{sc}.(T_{co} - T_{ci})$$

Q_{ch} and Q_{sc} are heat rejected by hot water and heat absorbed by cold water respectively. Also, T_{hi}, T_{ci}, and T_{ho}, T_{co} are temperatures of hot and cold water recorded at entrance and exit of coil and shell respectively. Readings were recorded when steady state is achieved and energy balance is observed for Q_{ch}/Q_{sc}=1 +0.5. Q_{ave} is calculated as [32]

$$Q_{ave} = \frac{Q_{ch}+Q_{sc}}{2}$$

Overall heat transfer coefficient U is calculated on the basis of LMTD.

$$Q_{ave} = U_{i/o} \,.\, A_{i/o} . LMTD$$

LMTD is given as: $LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\frac{\Delta T_1}{\Delta T_2}}$

For parallel flow, $\Delta T_1 = T_{hi} - T_{ci}$ and $\Delta T_2 = T_{ho} - T_{co}$

For counter flow, $\Delta T_1 = T_{hi} - T_{co}$ and $\Delta T_2 = T_{ho} - T_{ci}$

U is the overall heat transfer coefficient and is written as per Eq. (2). Wilson plot is obtained by plotting 1/UA vs. $1/v_s^n$. Where v_s is shell water velocity. Proper values of exponent, n are selected to get linear fit between data points in the plot. From this linear fit slope and y axis intercept are noted and h_i, h_o are obtained as per the Eq. (3) to Eq. (7). Curvature ratio is obtained considering D_{ave}. For SHC i.e. Θ = 90°, $D_{ave} = D_{cb} = D_{ct}$. For conical coils; $D_{ave} = D_{cb} + D_{ct}/2$. And for Θ = 0° HE, D_{ave} is obtained as average of inner and outer coil diameters of horizontal spiral coil. Formulations to calculate non-dimensional parameters are given below.

$$CR_{ave} = \frac{d_i}{D_{ave}}$$
$$Re_i = \frac{\rho_i \,.v_i \,.d_i}{\mu_i}; De_i = Re_i \,.\, CR_{ave}{}^{0.5}; Re_o = \frac{\rho_o . v_s \,.D_{eq}}{\mu_o} \tag{17}$$

For the calculation of shell side fluid velocity and Re_s, required D_{eq} is obtained using following formulation [11,32]

$$D_{eq} = ((D_{so}^2 - D_{si}^2). H_s - d_o^2 \,.\, L_c)/(d_o . L_c)$$

Tube side and shell side Nusselt numbers are given as

$Nu_i = \frac{h_i \,.\, d_i}{k_i}$ and $Nu_o = \frac{h_o . D_{eq}}{k_o}$

Schimidt's formula for Re_{crit} is preferred [52].

$$Re_{crit} = 2300. (1 + 8.6. (CR^{0.45})$$

Tube side flow having Re_i less than R_{crit} is considered as laminar and Re_i greater than R_{crit} is considered as flow in turbulent regime.

3. Experimentation

Test rig mainly consists of a hot and cold water tank, heat exchanger unit, temperature measurement systems, data collection and storage unit. Hot and cold water tanks are made up of steel and are provided with solenoid, float valves to maintain constant head of water during test run. Hot water is heated by the heater and temperature of hot water is maintained at set value by and automatic on-off of heater is done by thermostat. This temperature is displayed at section I1, shown in schematic diagram of the test rig, Figure 2. Insulations are provided to the hot water tank and heat exchanger units. Heat exchanger unit consists of a copper coil fitted in a steel shell. Piping arrangements are made on each heat exchanger unit to measure inlet-exit temperatures and pressure drops. RTD thermocouples are used for temperature measurement and are displayed at section I2. Cold water is forced into the shell from the top side only. Also, hot water is forced into the coil and its entry is done from top side or bottom side to have parallel flow and counter flow configurations. Steady state heat transfer between coil hot water and shell cold water is observed. After achieving steady state readings are recorded. Heat exchanger unit is replaced and again readings are taken for variation in mass flow rates of both fluids.

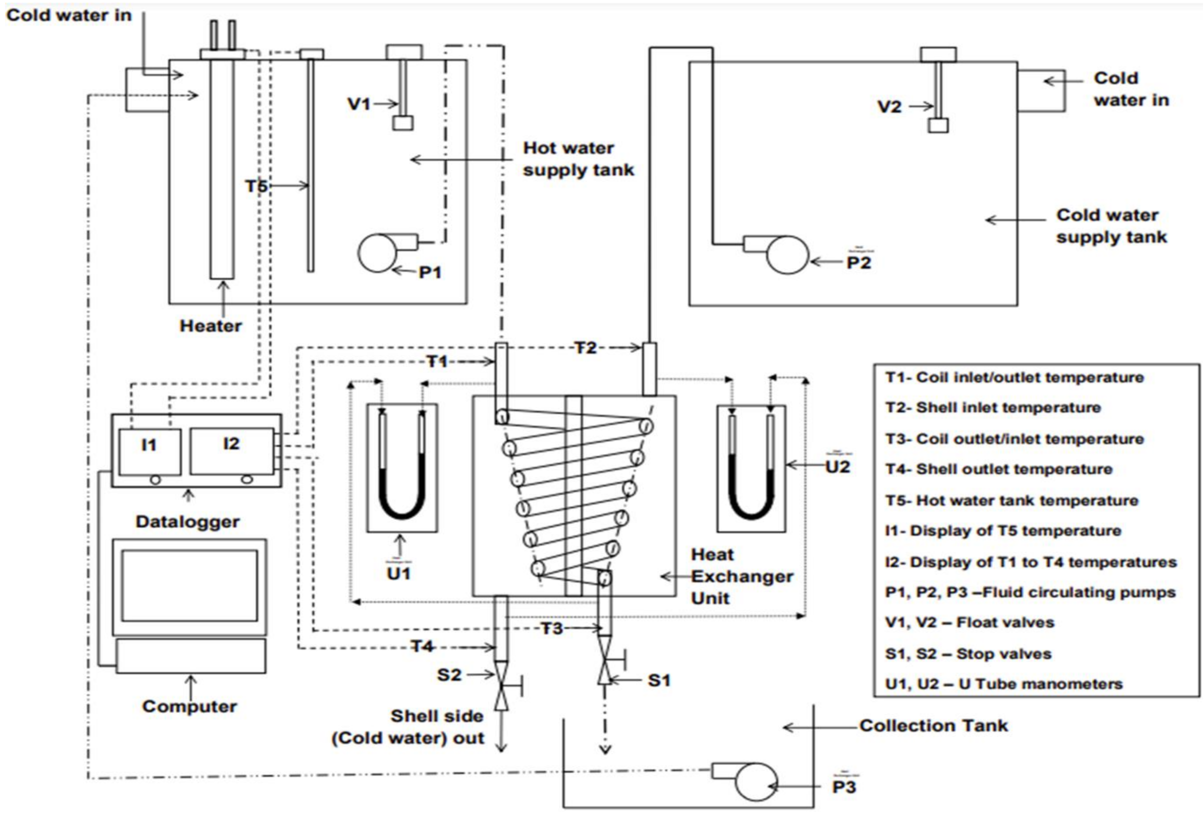


Fig. 2. Schematic diagram of Heat exchanger experimental set up

Copper coils of different shapes are made up by winding the copper tube on wooden blocks. Wooden blocks are converted into Cylindrical and conical shape as per the dimensions to produce straight helical, conical and spiral coils. Sand is filled into the copper tube to ensure a circular cross section. Other heat exchangers are manufactured by varying angle Ɵ = 70°, 50°, 30°, and 0° keeping smaller diameter D_{ct} = 0.07 m and length of coil fixed. As a result of this, the diameter of another end (D_{cb}) is increased. Additionally, the volume of the shell side is kept fixed and accordingly shell dimensions are obtained. For Ɵ = 90 ° SHCHE, D_{ave} becomes equal to D_{ct} = 0.07 m and it is close to D_{si} = 0.02 m. For Ɵ = 70°, 50°, 30°, and 0°, D_{cb} is increasing causing D_{ave} remains at the centre of shell. As a result of this, variation of coil curvature ratio takes place in the range of 0.14 to 0.048. Geometrical dimensions of the heat exchangers are shown in Table 1.

Table 1
Dimension of Heat Exchangers

Parameter	Dimension	Parameter	Dimension
Cone angle, Θ	90°, 70°, 50°, 0°, 30°	Average diameter, D_{ave}	0.07, 0.12, 0.16, 0.17, 0.21
Tube diameter, d_i, m	0.01	Curvature ratio, CR_{ave}	0.14, 0.083, 0.063, 0.059, 0.048
Top diameter, D_{ct}, m	0.07	Tube length, L_c, m	3.3
Pitch, P, m	0.018	Shell inner diameter, D_{si}, m	0.02
Base diameter, D_{cb}	0.07, 0.17, 0.25, 0.27,0.35	Shell outer diameter, D_{so}, m	0.24, 0.3, 0.33, 0.42, 0.38

Hot water flow into coils is considered in laminar and turbulent regimes. Reynolds numbers for coil fluid is in the range of 3700 to 21000. Accordingly fluid flow parameters are obtained and mentioned in Table 2. Method proposed by Kline and McClintock [53] is used to calculate uncertainties. Uncertainties involved in measurements and calculations of parameters are reported in Table 3. Experimental uncertainty is less than 4.5 % for all runs [8,10,20,21].

Table 2
Flow parameters used in heat exchangers

Parameter	Range
Mass flow rates (Coil- Hot water), m_{ch}, kg/s	0.02 – 0.1
Mass flow rates (Shell- Cold water), m_{sc}, kg/s	0.02- 0.1
Inlet temperature (Coil- Hot water), T_{ch}, °C	42 ±0.5
Inlet temperature (Shell- Cold water), T_{sc}, °C	27.5 ±0.5
Tube side Reynolds number, Re_i	3700-21000
Tube side Dean number, De_i	700-8000

Table 3
Uncertainty analysis

Parameter	Uncertainty (%)	Parameter	Uncertainty (%)
Mass flow rates	4.1	Heat transfer coefficients	4.3
Reynolds number	4.5	Nusselt number	4.3
Rate of heat transfer	4.3	Dean number	4.5

4. Results and Discussion

Mass flow rates of tube side hot water and shell side cold water are varied from 0.02 kg/s to 0.1 kg/s. All heat exchangers are analysed by examining cold water temperature difference (ΔT_{sc}), tube side, shell side heat transfer coefficients (h_i and h_o) and tube side, shell side heat Nusselt No. (Nu_i and Nu_o). After proposing mathematical correlations of Nu_i and Nu_o comparisons are done with results of existing researchers.

4.1 Coldwater Temperature Difference (ΔT_{sc}, °C)

In all five heat exchangers, tube side, shell side volumes are fixed and shell side volume is more than tube side volume. Also coil geometry is varying as straight helical coil, conical coil and spiral coils. Hence it is necessary to see how much rise is taking place in temperature of cold water, flowing through the shell. Variation in shell side cold water temperature difference ΔT_{sc} is plotted against mass flow rate of shell cold water, m_{sc} in Figure 3(a) to Figure 3(d) for laminar and turbulent flow regimes of coil hot water. Highest ΔT_{sc} are obtained for m_{sc} = 0.02 kg/s and decreases as mass flow rate increases to 0.1 kg/s. Also, mass flow rates of coil hot water, m_{ch} decreased from turbulent to laminar, ΔT_{sc} is found to be decreased. Highest ΔT_{sc} is obtained for m_{ch} = 0.1 kg/s and m_{sc} = 0.02 kg/s. It is observed that, as m_{sc} increases from 0.02 to 0.1 kg/s, exit temperature of cold water is decreased however rate of heat transfer increased. The rise in heat transfer is governed by the rise in heat capacity of cold water, $(mCp)_{sc}$. Highest ΔT_{sc} is found for ϴ = 30° CCHE. For ϴ = 0° HE, ΔT_{sc} is found close to ϴ = 30° CCHE followed by ϴ = 50° and 70° CCHEs. Lowest ΔT_{sc} is obtained for ϴ = 90° SHCHE. For ϴ = 90° SHCHE, coil diameter is very close to inner shell whereas for ϴ = 30° CCHE, coil top and bottom diameters are very close to inner and outer shell respectively. Due to this, the average diameter of conical coils lies at the centre of shell resulting in better contact of shell fluid with coil surface.

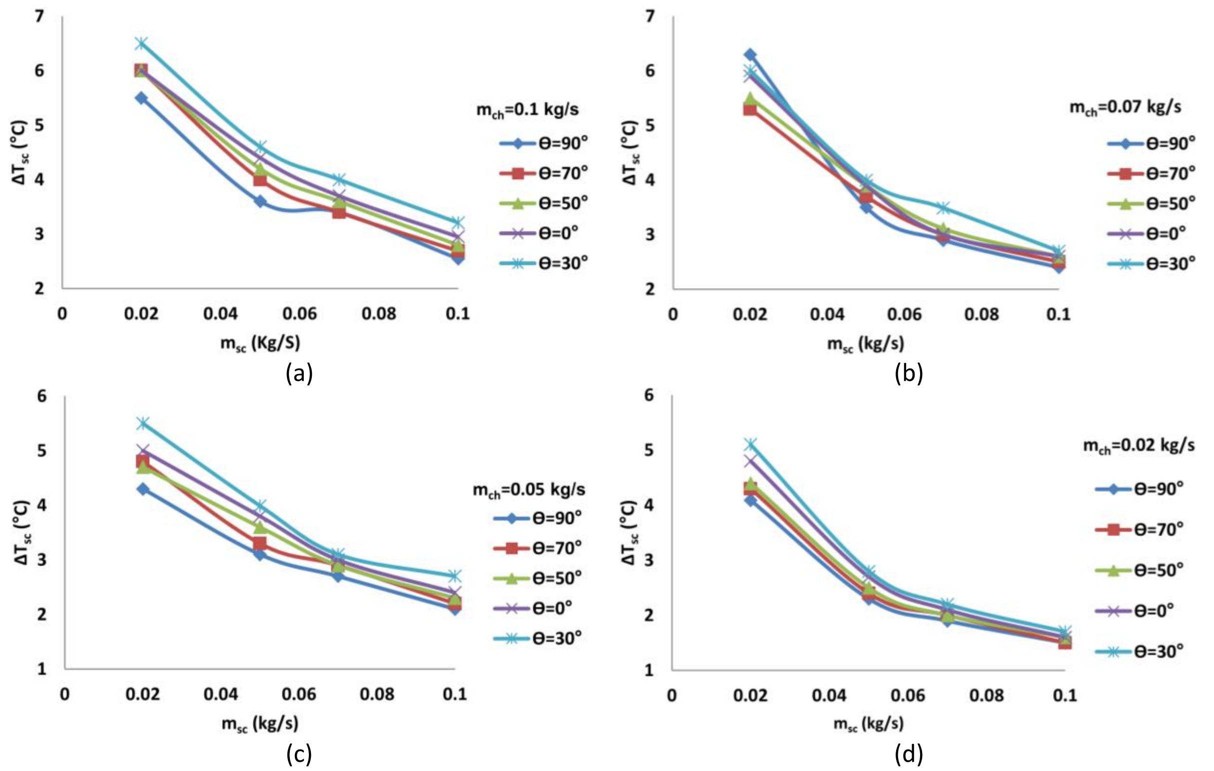


Fig. 3. Variation in temperature of shell cold water, ΔT_{sc} against shell cold water mass flow rate, m_{sc}

4.2 Rate of Heat Transfer, Q_{ave} (W)

Q_{ave} is average of heat rejected by hot water and heat absorbed by cold water. Q_{ave} is plotted against shell cold water mass flow rate, m_{sc}. Figure 4 shows that as m_{sc} increases in the range of 0.02 to 0.1 kg/s, Q_{ave} increases. Maximum Q_{ave} is obtained for coil hot water mass flow in turbulent regime. Maximum Q_{ave} is obtained for conical and spiral coils compared to straight helical coils. Among conical and spiral coils, maximum Q_{ave} is obtained for ϴ = 30° CCHE. In case of ϴ = 30° CCHE, as coil diameter decreases from base towards apex maximum variation of curvature is taking place from 0.048 to 0.14. In addition to this conical geometry and maximum slant edge length of 0.29 m is making maximum contact of shell water with coil surface. This aids in better heat transfer performance. This does not happen with other coils. Further in this study Wilson plot method is used to obtain values of h_i and h_o. These values are entered in the formulation of overall heat transfer coefficient, U and using the same LMTD, again rate of heat transfer is calculated. This rate of heat transfer is labelled as Q_{wp}. Then $((Q_{ave} - Q_{wp}/Q_{ave}).100)$ is obtained to check the relevance of values of h_i and h_o obtained from Wilson plots. It is found that for selected values of h_i and h_o, ((Q

- Q_{wp}/Q_{ave}).100) is found to be within 7%. These values of h_i and h_o are further used to obtain Nu_i and Nu_o.

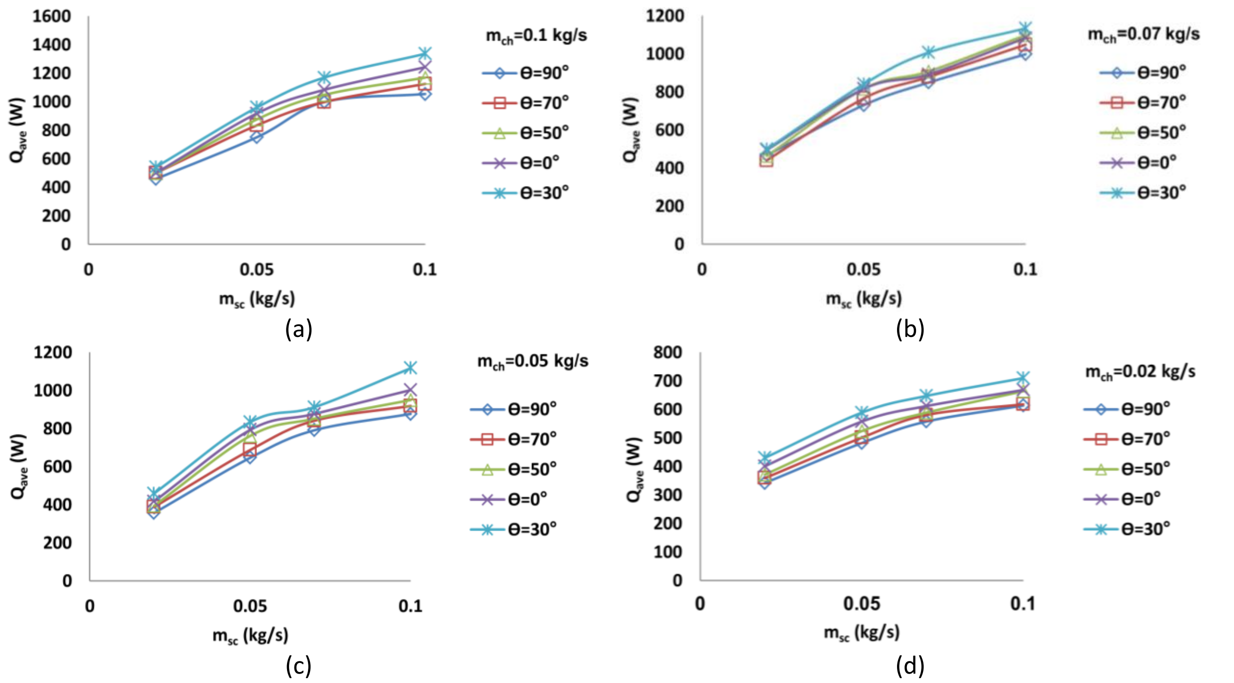


Fig. 4. Plot of rate of heat transfer, Q_{ave} against shell cold water mass flow rate, m_{sc}

4.3 Shell Side Heat Transfer Coefficients, h_o (W/m^2°C)

Shell side heat transfer coefficients, h_o are obtained using Wilson plot method discussed in section 2. Values of h_o are obtained according to formulae given in Eq. (2) to Eq. (8). In this study Θ is varied from 90° to 0°. As a result of this contact of shell fluid with coil surface is varied. Hence the shell side heat transfer coefficient h_o is compared for all heat exchangers. Effect of shell cold water mass flow rates, m_{sc} over h_o is illustrated in Figure 5. As m_{sc} increases, h_o is found to increase. Maximum h_o is obtained at m_{sc} = 0.1 kg/s for all heat exchangers. Highest h_o is obtained for Θ=30° CCHE and for Θ = 90° SHCHE, lowest h_o is obtained. For Θ = 90° SHCHE, coil diameter is 0.07 m and it is close to the inner shell diameter. This is causing less contact of shell water with the coil surface. But it has the highest h_i due to highest curvature ratio. On the contrary, for Θ = 30° CCHE, coil diameter decreases from D_{cb} = 0.35 m to D_{ct} = 0.07 m. As a result, maximum portion of the shell is occupied by the coil having a maximum slant edge length of 0.29 m with average diameter (D_{ave}) at the centre of the shell. This causes effective contact of shell fluid with coil. For Θ = 0°, 50° and 70° HEs D_{ave} is near to centre showing better h_o than Θ = 90° SHCHE. In addition to this h_o is plotted against m_{ch} and shown in Figure 6. Figure 6(c) and Figure 6(d) indicates that for lower Reynolds no. of shell side water (m_{sc} = 0.02 and 0.05 kg/s), when m_{ch} rises from 0.02 to 0.05 kg/s (laminar to transition regime) h_o is found to decrease for conical spiral and straight helical coils and in the turbulent regimes of hot water (m_{ch} = 0.7 and 0.1 kg/s), h_o is further increased. Also, for shell side water with higher Reynolds no. (m_{sc} = 0.07 and 0.1 kg/s), increment in m_{ch} from 0.02 to 0.05 kg/s (laminar to transition regime) h_o is found to decrease for Θ=30° and 0° HEs. And when m_{ch} increases beyond 0.05 kg/s (turbulent regime) h_o is increased. Specifically, for Θ=30°CCHE sagging in the curve is found at m_{ch} = 0.05kg/s and it is predominant when m_{sc} = 0.02 kg/s. But for remaining heat exchangers, at higher m_{sc}, h_o increases directly as m_{ch} increases from 0.02 to 0.1 kg/s.

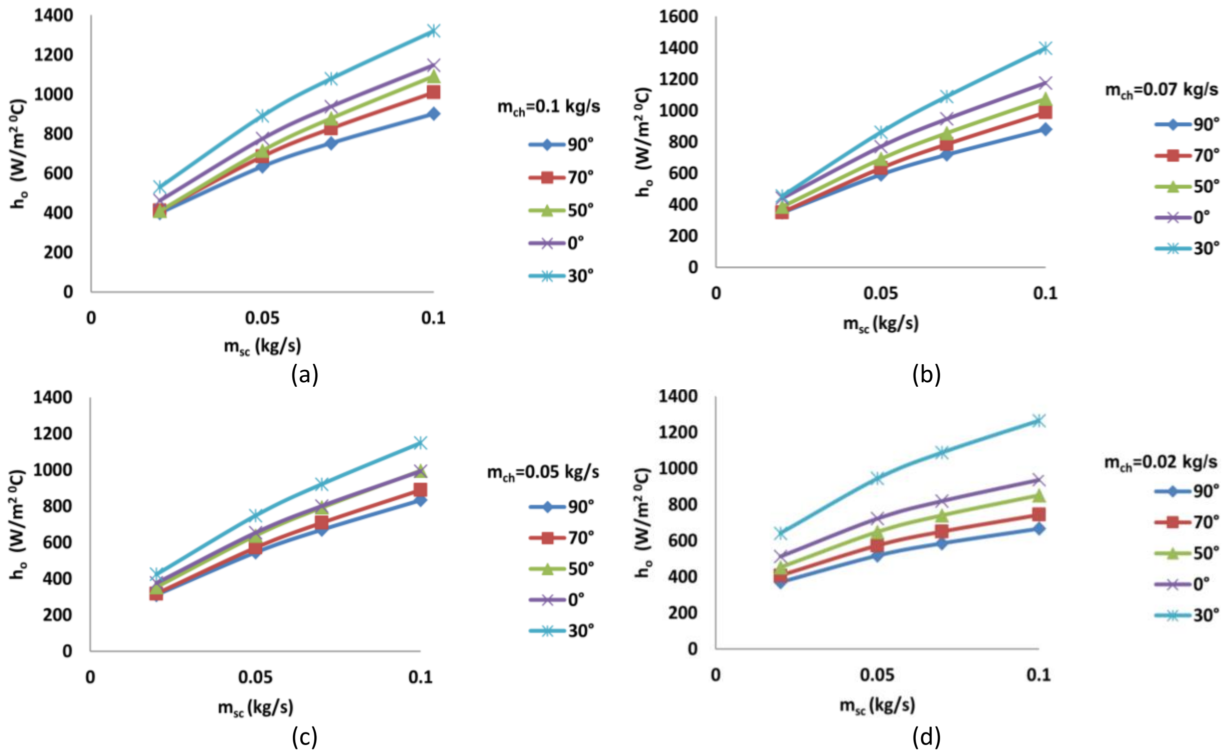


Fig. 5. Plots of h_o vs. m_{sc} for (a) m_{ch} = 0.1 kg/s, (b) m_{ch} = 0.07, kg/s, (c) m_{ch} = 0.05 kg/s, (d) m_{ch} = 0.02 kg/s

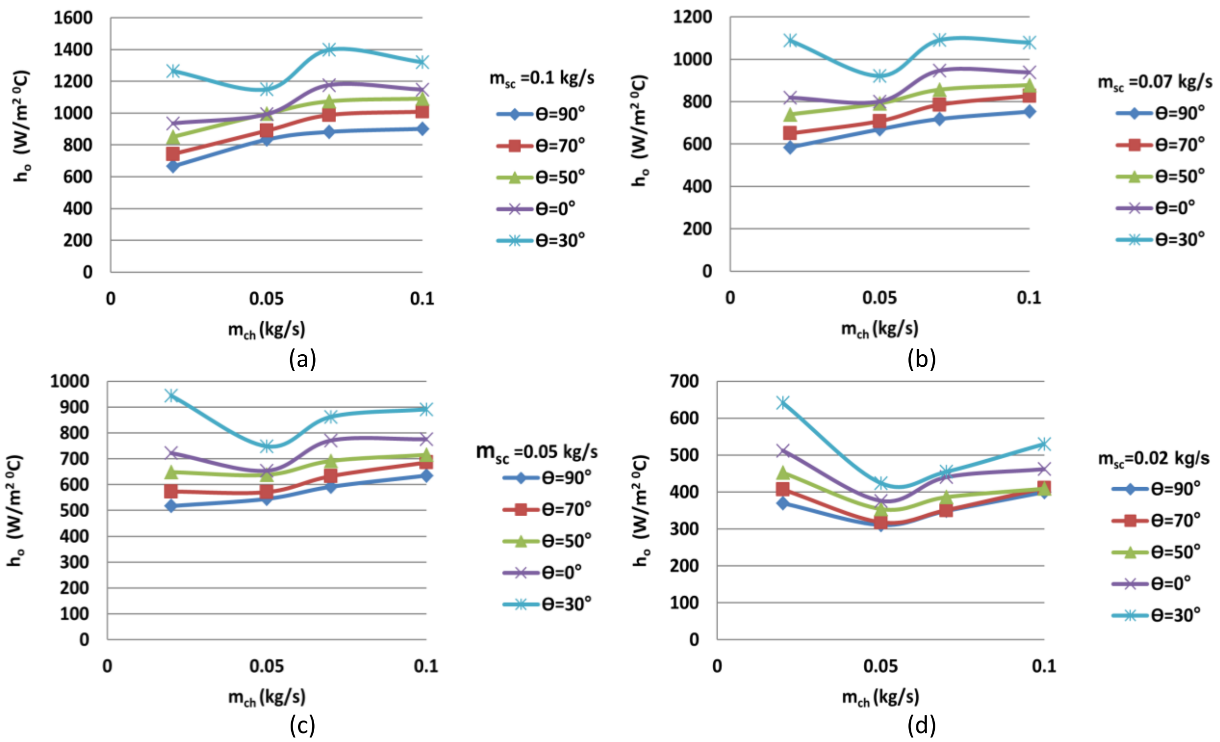


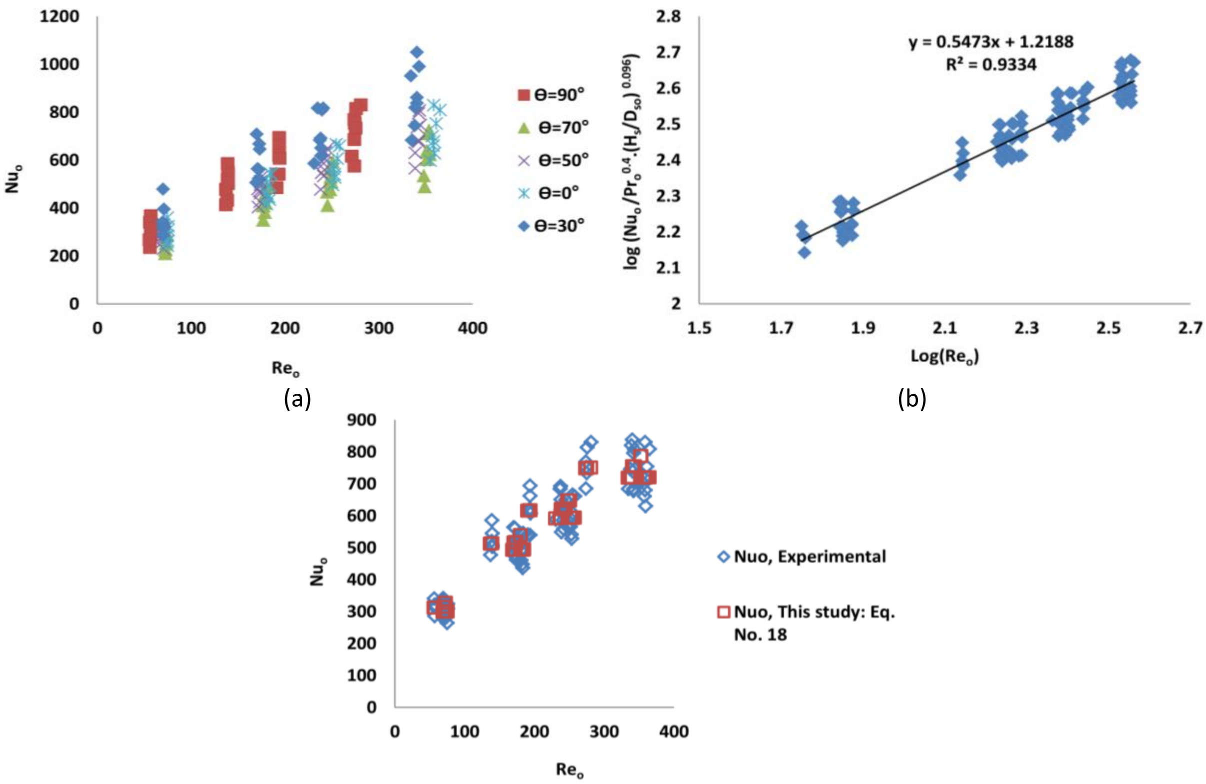
Fig. 6. Plots shell side heat transfer coefficients, h_o with respect to variation of coil side mass flow rate, m_{ch} for (a) m_{sc} = 0.1 kg/s, (b) m_{sc} = 0.07 kg/s, (c) m_{sc} = 0.05 kg/s, (d) m_{sc} = 0.02 kg/s

Looking at the formulation of U, h_i and h_o, if h_i is very much higher than h_o then small changes in h_o makes maximum effect on U compared to h_i. In CCHE h_i is very high compared to h_o and due to conical geometry increment in h_o is more compare to SHCHE. Further using D_{eq} as the characteristic dimension in the formulation shell side Reynolds number, Re_o and shell side Nusselt number, Nu_o

are obtained. Plot of Nu_o against Re_o is shown in Figure 7. As expected Nu_o is found to increase, when Re_o increases. In case of coiled heat exchangers for selected range of parameters mathematical correlations to obtain Nu_o are not found. Hence correlation to obtain Nu_o is proposed here. In this study shell dimensions are varied to accommodate conical coils, spiral coil and straight helical coil which results in variation in contact of shell fluid with coil surface. To consider its effect, ratio of height of heat exchanger, H_s to shell diameter D_{so} is considered as a non-dimensional parameter in the derivation of Nu_o correlation. To develop the correlation procedure discussed by Kern [54] is followed considering Re_o, Pr_o and H_s/D_{so} as non-dimensional parameters. Linear plot of the same is shown in Figure 7(b) in which R^2value is equal to 0.93. This correlation is given as Eq. (18).

$$\mathrm{Nu_o} = 16.55\,.\,\mathrm{Re_o}^{0.55}.\,\mathrm{Pr_0}^{0.4}.\left(\frac{\mathrm{H_s}}{\mathrm{D_{so}}}\right)^{0.096} \text{for } 50 \ll \mathrm{Re_0} \ll 400,\ 5 \ll \mathrm{Pr_o} \ll 6 \tag{18}$$

Comparison between Nu_o obtained from experimentation and Nu_o predicted using Eq. (18) is shown in Figure 7(c). Maximum 14 % variation is found.



(c)

Fig. 7. (a) Plot of Nu_o vs Re_o, (b) Plot for Nu_o equation (Eq. (18)), (c) Comparison of Nu_o

4.4 Tube Side Heat Transfer Coefficients, h_i (W/m^2°C)

To study the effect of variation of curvature ratio, CR in SHC, conical and spiral coils tube side heat transfer coefficients, h_i are discussed. In SHC, curvature ratio, CR remains constant along the axis whereas CR varies in CC and spiral coils. To understand the effect of variation of curvature ratio, h_i is discussed here. h_i is obtained using Wilson plot method (Eq. (2) to Eq. (6)). Figure 8 shows variation of h_i with hot water mass flow rate, m_{ch}. As expected, increase in m_{ch} causes increase in h_i. Highest h_i is found for Θ=90°SHCHE which is having maximum curvature ratio. Also, lowest h_i is obtained for Θ = 30°CCHE which is having lowest CR_{ave} as 0.048. Thus, curvature ratio increases, h_i increases. This is in agreement with the results of researchers. In addition to this tube side Nusselt numbers (Nu_i) are calculated. Plots of Nu_i with respect to tube side Reynolds number, Re_i and tube side Dean number, De_i are shown in Figure 9. It is found that, as Re_i and De_i increases, Nu_i also increases. To check the relevance of Nu_i, it is decided to compare Nu_i obtained for every heat exchanger with Nu_i predicted from the correlations suggested by Schmidt [23] (Eq. (12)), Roger and Mayhew [13] (Eq. (9)), Mori and Nakayama [24] (Eq. (13)), Roger and Mayhew [13] and Hewitt *et al.,* [26] (Eq. (14)) and Seban and McLaughlin [30] (Eq. (16)).

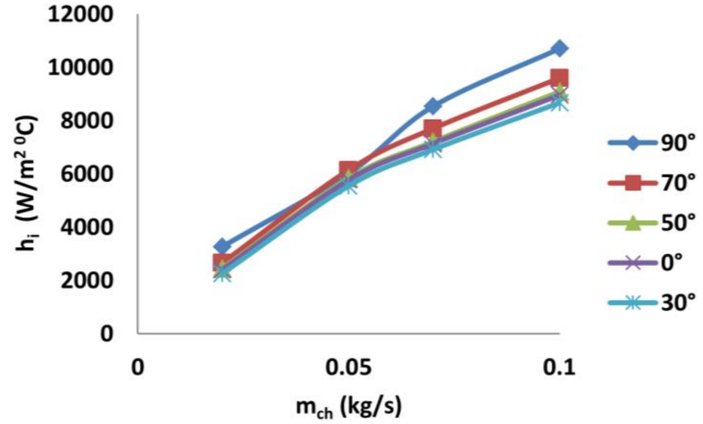


Fig. 8. Variation of h_i against tube side mass flow rate, m_{ch}

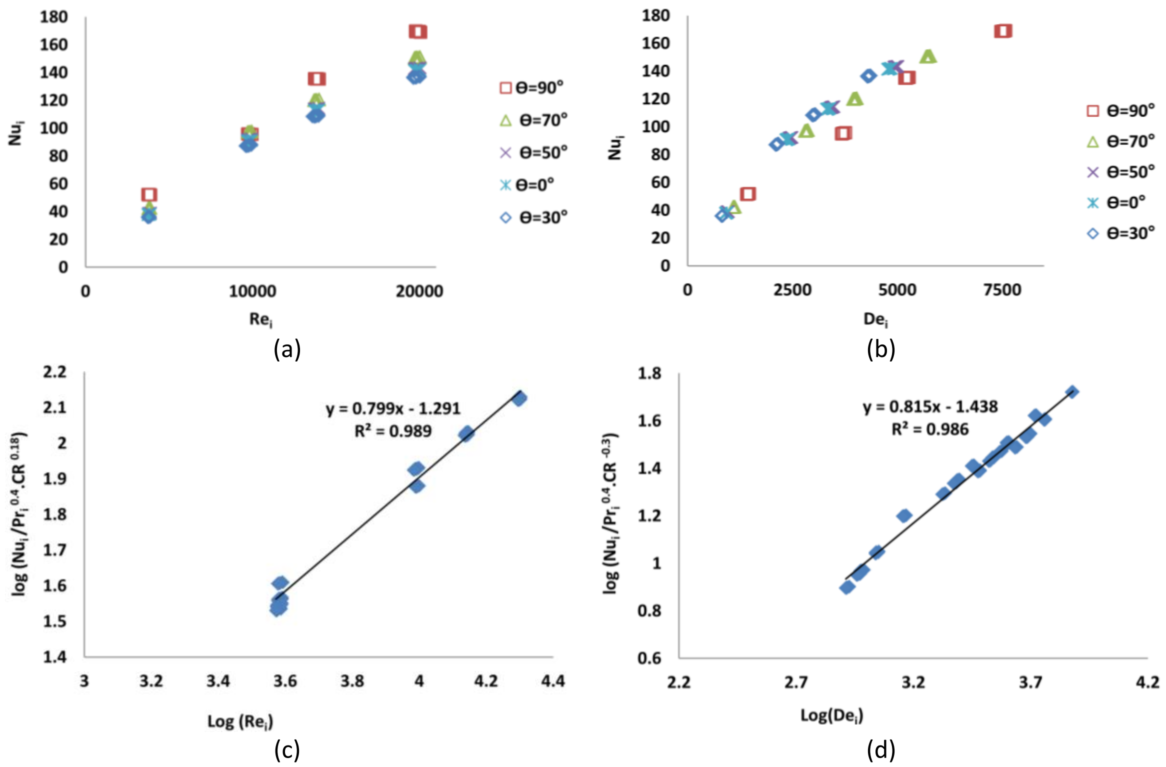


Fig. 9. (a) Plot of Nu_i vs. Re_i, (b) Plot of Nu_i vs. De_i, (c) and (d) Plots for Nu_i correlations (Eq. (19) and Eq. (20))

Nu_i obtained in this study for all heat exchangers are showing agreement with above mentioned researchers. % variation in Nu_i obtained in this study is 0-5%, 0-17%, 1-18 %, 1- 18 % respectively when compared with Schmidt [23] (Eq. (12)), Mori and Nakayama [24] (Eq. (13)), Rogers and

Mayhew [13] (Eq. (9)) and Seban and McLaughlin [30] (Eq. (16)) respectively. But in comparison with Rogers and Mayhew [13] and Hewitt *et al.,* [26] (Eq. (14)) % variation in Nu_i is found to be up to 23%.

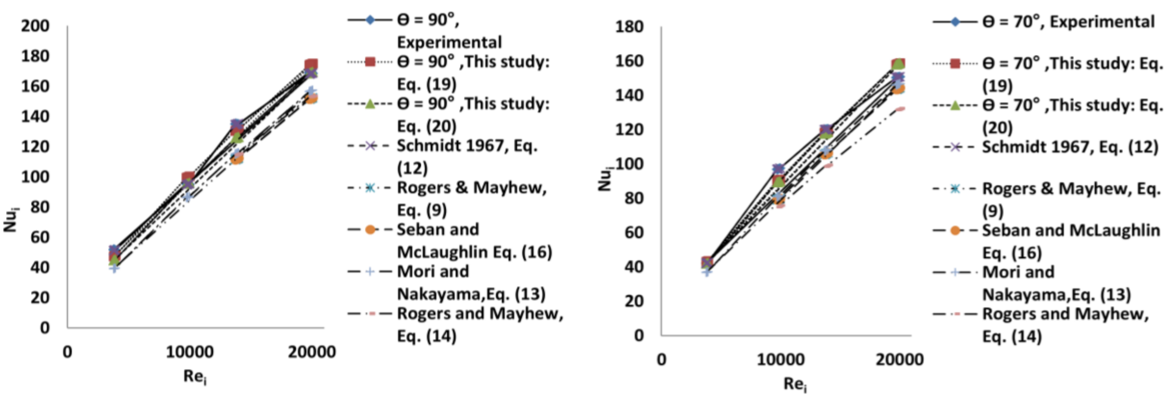
Further these calculated Nu_i values are used to suggest two correlations on the basis of Re_i and De_i. Effect of variation of curvature ratio in conical coils, spiral coil and straight helical coil is considered simultaneously selecting curvature ratio on the basis of average coil diameter. Also, two separate linear plots are obtained on the basis of Re_i and De_i. These plots are shown in Figure 9(c) and Figure 9(d). It shows values of R^2as 0.989 and 0.986 respectively. First correlation is expressed in Eq. (19) in which Re_i, Pr_i and curvature ratio are used as non-dimensional parameters.

$$\mathrm{Nu_i} = 0.0512\,.\,\mathrm{Re_i}^{0.80}.\,\mathrm{Pr_i}^{0.4}.\,\mathrm{CR_{ave}}^{0.18}$$
$$\text{for } 3700 \ll \mathrm{Re_i} \ll 21000, 4 \ll \mathrm{Pr_i} \ll 5 \text{ and } 0.04 \ll \mathrm{CR} \ll 0.15 \tag{19}$$

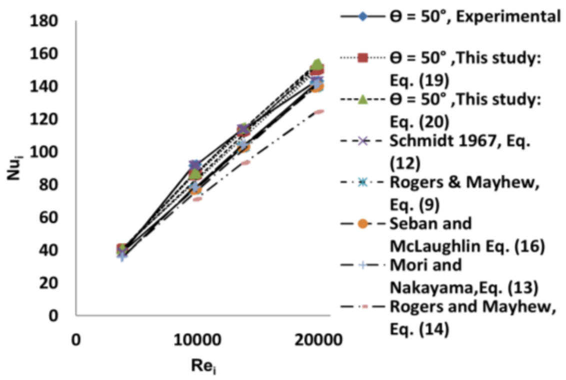
Second correlation is proposed using De_i, Pr_i and CR as:

$$\mathrm{Nu_i} = 0.0365\,.\,\mathrm{De_i}^{0.815}.\,\mathrm{Pr_i}^{0.4}.\,\mathrm{CR_{ave}}^{-0.3}$$
$$\text{for } 700 \ll \mathrm{De_i} \ll 8000, 4 \ll \mathrm{Pr_i} \ll 5 \text{ and } 0.04 \ll \mathrm{CR} \ll 0.15 \tag{20}$$

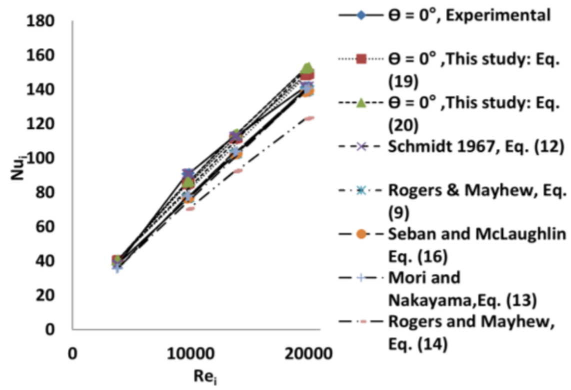
In addition to this, Nu_i obtained from correlations suggested in this study (Eq. (19) and Eq. (20)) are further compared with Nu_i predicted by Rogers and Mayhew [13] (Eq. (9)), Schmidt [23] (Eq. (12)), Mori and Nakayama [24] (Eq. (13)), Rogers and Mayhew [13] and Hewitt *et al.,* [26] (Eq. (14)) and Seban and McLaughlin [30] (Eq. (16)). Comparative plots for each heat exchanger are shown in Figure 10. Thus, comparison shows that maximum variations in Nu_i obtained from the Eq. (19) and Eq. (20) are 17 % and 13 % respectively. In a few cases this maximum variation in Nu_i reaches up to 20% when compared with Rogers and Mayhew [13] and Hewitt *et al.,* [26] (Eq. (14)). Thus, while designing coiled heat exchangers cone coils are selected preferably with an angle around 30°. For the minimum possible coil diameter at the smaller end of cone coil, a bigger diameter of coil is obtained according to space requirements. Using proposed correlations Nusselt numbers, heat transfer coefficients and overall heat transfer coefficients are obtained for coiled heat exchangers. This is used to decide area required to achieve the desired outlet temperature of either of the fluids.



(a) (b)



(c)



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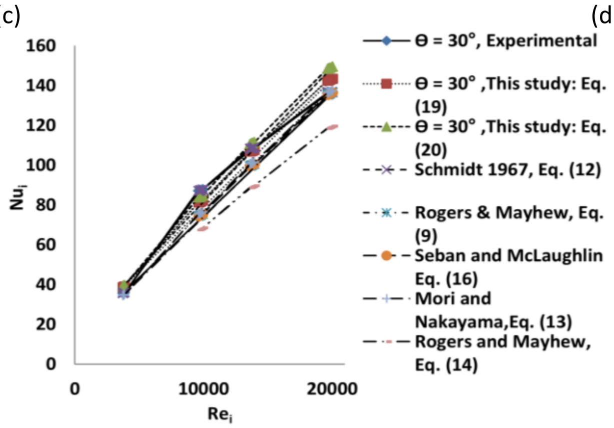

(e)

Fig. 10. Comparative plots of Nu_i vs. Re_i for all heat exchangers

5. Conclusion

In comparison with straight helical coils in conical and spiral coils, variation in coil diameter causes variation in curvature ratio and effective contact of shell fluid with coil surface. This is contributing to heat transfer. For designing these heat exchangers values of shell and tube side heat transfer coefficients (h_o and h_i) is necessary and can be obtained from shell and tube side Nusselt number (Nu_o and Nu_i) correlations. For cone and spiral coils enough information about this is not available. To analyze this heat exchangers consisting of three cone coil heat exchangers (angle Θ = 70°, 50° and 30° CCHE), spiral coil (Θ = 0° HE) and straight helical coil (Θ = 90° SHCHE) are studied. For tube side flow, Reynolds number, Re_i and Dean number, De_i varies from 3700-21000 and 700-8000 respectively. h_i and h_o are obtained using Wilson plot method.

i. Shell side heat transfer coefficient, h_o is found to increase as shell side mass flow rate is increased. Maximum h_o is obtained for Θ = 30° CCHE. Also, small variation in h_o causes significant effect over rate of heat transfer. It is achieved with conical coils due to conical geometry. Also, for flow of shell side water with higher Reynolds no. (m_{sc} = 0.07 and 0.1 kg/s), increment in the tube side hot water mass flow rate (m_{ch}) from 0.02 to 0.05 kg/s (laminar to transition regime) h_o is found to decrease for Θ = 30° and 0° HEs. And when m_{ch} increases beyond 0.05 kg/s (turbulent regime) h_o is increased. Considering new non-dimensional parameter as ratio of height of shell, H_s and diameter of shell, D_{so}, correlation to calculate Nu_o is proposed as: $Nu_o = 16.55.\ Re_o^{0.55}.Pr_o^{0.4}.(H_s/D_{so})^{0.096}$. A comparison of these values with experimental values shows 0-14 % variation.
ii. h_i is found to increase when Re_i and De_i are increased. Maximum h_i is obtained for Θ = 90° SHCCHE as it has maximum curvature ratio in the selected range. Further Nu_i are obtained. These are compared with Nu_i predicted by other researchers and agreement is

found. % variation in Nu_i obtained in this study is 0-5 %, 0-17 %, 1-18 %, 1-18 % respectively when compared with Nu_i predicted by Schmidt [23], Mori and Nakayama [24], Rogers and Mayhew [13] and Seban and McLaughlin [30] respectively. But in comparison with Nu_i proposed by Rogers and Mayhew [13] % variation in Nu_i is found to be up to 23%.

iii. Also, the first correlation is predicted as: $Nu_i = 0.0512.\ Re_i^{0.80}.Pr_i^{0.4}.CR_{ave}^{0.18}$. Comparison between Nu_i obtained from Wilson plot, Nu_i obtained using this correlation and Nu_i predicted from correlations suggested by above mentioned researcher's shows maximum variation up to 17 %.

iv. In addition, the second correlation is predicted as: $Nu_i = 0.0365.\ De_i^{0.815}.Pr_i^{0.4}.CR_{ave}^{-0.3}$. Similarly, comparison of Nu_i shows maximum variation up to 13 %.

v. Also, after selecting values of h_i and h_o from Wilson plots, to check the correctness, rate of heat transfer (Q_{wp}) is obtained using the same values of log mean temperature difference. Variation between Q_{ave} and Q_{wp} is found in the range of 0 to 7 %.

Thus, preferably conical coiled heat exchangers can be used. Nusselt numbers are obtained by using proposed correlations in this study and surface area is decided to achieve the desired outlet temperature of one of the fluids.

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Conflict of Interest
The authors declare no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

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