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A Review on Heat Transfer of Fluids in Curved and Coiled Geometries

S. S. Pawar, Nikhil Shriyan*, AmeyVedpathak and Swapnil Mule

S.I.E.S Graduate School of Technology, Nerul, Navi Mumbai – 400706.

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ABSTRACT

The objective of the present study is to give a review on heat transfer characteristics of fluids in curved and straight tubes for various parameters and under different experimental conditions. The heat transfer can be increased by two techniques mainly like application of external forces or by modification in surface geometry of the fluid passages. Surface geometry modifications like bending of straight tubes into curved coils are effective and efficient method of heat transfer enhancement. In this paper, various correlations proposed based on experimental heat transfer data by earlier investigators are presented to support the enhancement in overall heat transfer coefficient in curved tube and in straight tube heat exchangers. However, comparison of experimental overall heat transfer coefficient in helical coil and straight tube heat exchangers are found to be limited in the present literature.

Introduction

A heat exchanger is a device used to transfer heat between two fluids. Design of heat exchanger is of utmost importance in industries due to its wide usage. They are widely used in many industries. Emphasis is laid on the compactness and the cost of the heat exchangers while designing industrial heat exchangers. Curved and coiled tube has a unique advantage of secondary flow pattern, generated due to the centrifugal forces. This secondary flow pattern generates dean vortices (two semicircular profiles) perpendicular to the direction of the flow. Thompson (1876) first noted that the centrifugal forces influence the flow in curved pipes. Curved tubes and passages are employed in many heat transfer applications. For example, helical and spiral coils are often used for transferring heat in mixing, storage, and reactor vessels, as well as in process heat exchangers. Curved heat transfer passages have been used in rocket engines and in jacketed vessels. Enhancement in heat transfer takes place in helical coiled configuration therefore the study of flow and heat transfer in curved tubes is very much required. The understanding of effect geometries like straight tube and helical coil on overall heat transfer coefficient is significant while selecting type of heat exchanger.

Review on experimental work in helical coil heat exchangers

Seban and McLaughlin [1] conducted experiments to study friction and heat transfer in laminar flow for oil as test fluid and turbulent flow for water in helical tubes. The helix diameters of the two coils were 4.93 and 30.1 inches. The coil to tube diameter ratios of the two coils was 17 and 104. The experiments were conducted for Reynolds number range of 12 to 65000. The inside diameter of the tubes was 0.290 inch with a wall thickness of 0.012 inch. Two helical coils were manufactured

For calculating the inner wall temperature, the concept of heat transfer through a hollow cylinder with uniform internal heat generation was used. For laminar flow, the range of Reynolds number was 12-5600 and Prandtl number varied from 100-657. Fluid temperatures were evaluated at the film temperature. The condition of constant heat flux was used and it was found that higher heat transfer coefficients were obtained on the outer surface than on the inner surface of the coil and both these values were greater than the values for straight pipes. It was observed that for the same Peclet number, the length for asymptotic behavior is less in helical coil than in straight tube. The developed correlation is:

For same test runs, heat transfer coefficients decreased normally until they increased at the last point of measurement. Seban and McLaughlin [1] reported that this behavior was due to the free convection effect which was caused because of the body forces arising from the curvilinear motion. For turbulent flow, the following correlation was developed

$$Nu(\mathrm{Pr})^{-1/3} = 0.644\left[\frac{f}{8}(\mathrm{Re})^2\frac{d}{x_1}\right]^{1/3} \tag{1}$$

$$(Nu)(\mathrm{Pr})^{-0.4} = 0.023(\mathrm{Re})^{0.85}(d/D_h)^{0.1} \tag{2}$$

While determining the heat transfer coefficients for turbulent flow, no evidence of thermal entry length was found and negligible longitudinal variation was found. Due to high heat transfer coefficient, the effects of circumferential conduction were found to be negligible.

Experiments were conducted by Rogers and Mayhew [2] for forced convection heat transfer. The working fluid in the helical coil was water and the coils were heated by steam from outside.

Three coils with D/d ratio of 10.8, 13.3, and 20.1 inches were tested. The mean diameters of the three coils were 4.005, 4.927, 7.474 inches with internal diameters of 0.3734, 0.3723, 0.3723 inches and length of the three coils were 8.975, 8.425, 8.823 feet respectively.

Tele:

E-mail addresses: shriyan.nikhil@siesgst.ac.in

All the three coils had the same pitch of 1.5 inches. A correlation was developed from the obtained experimental results as follows:

$$(Nu)_f = 0.021\,(\mathrm{Re})_f^{0.85}\,(\mathrm{Pr})_f^{0.4}(d/D)^{0.1} \quad (3)$$

The properties were evaluated at the film temperature. A correlation for the properties evaluated at the bulk temperature was also presented as follows:

$$(Nu)_b = 0.023\,(\mathrm{Re})_b^{0.85}(\mathrm{Pr})_b^{0.4}\left(d/D\right)^{0.1} \quad (4)$$

The results suggested a Re number exponent of 0.85 and d/D exponent of 0.1 but this data is limited to D/d less than 30. These correlations were developed for design purposes.

Heat transfer data for the condition of constant wall temperature conditions in different types of helical and spiral coiled pipes was obtained by Kubair et. al. [3]. Aqueous solution of glycerol was used as the working fluid. Steam was used as the heating fluid. The heat transfer coefficients were evaluated at the arithmetic mean temperature difference and correlated using Nusselt and Graetz number. The physical properties of the fluid were evaluated at the bulk temperature. It was observed that L/D ratio and the range of Prandtl number have an effect on the heat transfer rates. An empirical relation in terms of Gz number was proposed as

$$Nu = 1.75(Gz)^{1/3} \quad (5)$$

A new correlation for laminar heat transfer Nu number was given incorporating the results for all the tested helical coils and spiral tubes which was as follows:

$$Nu = [1.98 + 1.8(d/D)]Gz^{0.7}. \quad (6)$$

This equation holds good for 10 < Gz < 1000, 80 < Re < 6000 and 20 < Pr < 100. Experiments were also conducted for the flow on non-Newtonian fluid. The solutions were prepared by dissolving a known amount of CMC, CPM and sodium silicate powders in water. It was found that the heat transfer coefficients were very much higher as compared to Newtonian and Non-Newtonian fluids in straight pipes and Newtonian fluids in coiled pipes. It was found that the heat transfer coefficients increased as ‘n’ (flow index) decreased and the Nu number increased as the curvature ratio increased. The correlation given was

$$Nu = [1.98 + 1.8(d/D)]\left[\frac{3n+1^{0.7n}}{4n}\right]Gz^{0.7} \quad (7)$$

For 800 < Re < 9000, 0.4 < n < 2 and 10 < Gz < 1000, 10 <Pr< 100.

An experimental study on convective heat transfer through coiled tubes was carried out by Janssen and Hoogendoorn [4]. The Prandtl number range was from 10 to 500 and Reynolds number varied from 20 to 4000. Both the boundary conditions of uniform peripherally averaged heat flux and constant wall temperature were studied. Only laminar regime was covered with attention to both thermal entry as well as fully developed thermal region. Four different coils were tested and the liquids used were water glycerol mixtures. The oscillating nature of Nu number was observed and it was stated that due to the flow of fluid of higher temperature to the outer wall of the tube there occurs a sudden decrease in the heat transfer coefficients. Due to the thin thermal boundary layer at the outer wall the heat transfer in this region is affected to small temperature changes of the fluid. For the thermal entry region a correlation was obtained as

$$Nu = (0.32 + 3d/D)\mathrm{Re}^{0.5}\mathrm{Pr}^{0.33}\left(\frac{d}{z}\right)^{0.14+0.8d/D} \quad (8)$$

For 20 < Dn < 8.3×10^2, 30 < Pr < 450 and 0.01 < (d/D) < 0.08. For fully developed thermal region the peripherally averaged Nu number was given by the following correlation:

$$Nu = 0.43(\mathrm{Re}^2)^{0.26}\,\mathrm{Pr}^{1/6} \quad (9)$$

For 20 < Dn < 8.3×10^2, 30 < Pr < 450 and 0.01 < (d/D) < 0.08. Two asymptotic correlations were also derived as a function of Re, Pr and d/D which are as follows:

$$Nu = 0.9(\mathrm{Re}^2\,\mathrm{Pr})^{1/6} \quad (10)$$

For 20 < Dn < 100 and

$$Nu = 0.7\,\mathrm{Re}^{0.43}\,\mathrm{Pr}^{1/6}(d/D)^{0.07} \quad (11)$$

For 100 < Dn < 830.

It was concluded from above correlations that the effect of d/D was negligible for Dn number between 20 and 100 and very small for Dn > 100. A correlation was derived for Dn < 20 which was as follows:

$$Nu = 1.7(Dn^2\,\mathrm{Pr})^{1/6} \quad (12)$$

The peripherally averaged asymptotic Nusselt number for the fully developed thermal region can be described as a function of Dn^2Pr in case of small Dean Numbers only. For Dn > 20 the dependency of Nu number on d/D was not found to be described by Dean Number. The asymptotic Nu number was found to depend on $Pr^{1/6}$ rather than $Pr^{1/3}$. The dependency of Nu on $Pr^{1/3}$ was found only in the thermal entrance region.

Mujawar and Rao [5] carried out experiments for determining the isothermal pressure drop through coiled tubes for water and pseudoplastic polymer solutions. Coils of curvature ratios 0.0695, 0.0476, 0.0198, 0.0100 were tested. The ID of all the coils was 1.21 cm. Helix diameter of the coils were 17.4, 25.4, 61, 121 cm respectively with corresponding pitch values of 1.9, 2.6, 5.7, 13.6 cm. Sodium alginate (SA) and sodium carboxy methyl cellulose (SCMC) in the concentration range of 0.3 to 1.0% (w/w) were used as the test fluids. For the laminar flow of fluids in the four helical coils the following correlation was given:

$$\frac{f_c}{f_s} = 0.26(Dn)^{0.36} \quad (13)$$

This correlation was given for the range of 35 < Dn < 2200 and 0.0100 < (a/R) < 0.0695. For determining the friction factor a plot of wall shear stress v/s the average shear rate (8V/d) and a/R was plotted for the aqueous solution of 0.5% SA. For each test fluid similar curves were plotted and it was found that the enhancement in pressure drop is a function of a/R and the nature of the liquid. Thus the following generalized correlation was obtained for the wall shear stress:

$$\tau_{w_c} = \left(\frac{d\Delta p}{4L}\right) = K_c'\left(\frac{8V}{d}\right)^{n_c'}\left(\frac{a}{R}\right)^{m_c'} \quad (14)$$

The friction factor was given as:

$$f_c = \frac{16}{\mathrm{Re}*/(a/R)^{m_c'}} \quad (15)$$

The denominator in this equation was termed as a new dimensionless number called the M number. Therefore

$$M = \frac{\mathrm{Re}^*}{(a/R)^{m_c'}} \quad (16)$$

From this, the criterion for laminar flow in helical coils was established as M ≤ 2100. The dimensionless parameter, M, is very useful to characterize the hydrodynamics of any fluid, either Newtonian or power-law type, flowing through helical coils.

Manafzadeh [6] examined the oscillation of Nusselt number at the junction of a straight and helical tube. The helical coil used had ID of 12.57 mm and 1.65 mm wall thickness. A 3.05 m straight pipe was used which was connected to a vertical axis helical coil of length 6.22 m. The coil diameter was 236 mm with a pitch of 16.7 mm. Distilled water was used as the working fluid. It was found that in the region of the junction of the two tubes all the wall temperatures showed a decrease in value and this was because of the cold fluid in the core being pushed towards the wall which was due to the developing secondary flow. The Nusselt number initially increased at the junction. The transverse boundary layers begin to develop and the circumferential temperature begins to show variation. It was inferred that thin transverse boundary layers were formed during the initial development of secondary flow. A 3.2 m copper refrigeration tube of 4.8 mm diameter bent into helical coil of 0.07 m diameter was tested for constant temperature condition by immersing the tube in water bath. Then the same tube was bent in the shape of figure eight and tested under similar conditions. Increased value of Nu number for figure eight tube was seen. It was concluded that if the transverse boundary layers are kept thin by constantly reversing the direction of curvature a higher Nu can be obtained.

For the boundary condition of uniform heat flux Austen and Soliman [7] carried out experiments with two pairs of helical coil. Each pair had the same diametric ratio but the pitch ratio of each of the four coils was different. Water was used as the test fluid. The ID and OD of all the helical coils were 4.57 mm and 6.34 mm respectively. The tubes were heated by passing a dc electric current through the walls of the coils thus maintaining the condition of uniform heat flux input in the flow direction. The experiments were carried out for: 50 < Re < 7000, 3 < Pr < 6, 300 < Gr <5800. Heat transfer tests were performed to cover the laminar flow regime for each of the four coils. The helical axes of the coils were oriented vertically and the heat transfer tests were performed with water feeding from both top and bottom directions. The axial variation of the inner tube wall temperature showed the rapid development of the temperature field within a very short axial distance. Oscillations with decreasing amplitude were observed until the temperature field became fully developed. The amplitude of oscillations was found to increase with increase in Reynolds number and this behavior was the same as observed by Seban and McLaughlin [1] et al. Thus it was inferred that these oscillations were due to the secondary flow resulting from the centrifugal forces. The effect of pitch was found to be insignificant for same diametric ratio because pitch effects were more prominent for low Reynolds number. Similar to the values of temperature the Nusselt number also showed oscillations for high Reynolds number. When the direction of flow was changed from upward to downward heat transfer rate increased in high pitched coils at low Reynolds number. This was due to the effect of free convection heat transfer and this effect was limited to the coils of large pitch. As the Reynolds number increased this effect diminished. The variation of Nusselt number was also observed and a significant increase in Nu due to increasing h/D up to a certain Reynolds number was seen beyond which h/D has no effect.

Prasad et. al. [8] gave experimental data and correlations for heat transfer and pressure drop for shell and tube sides of a helical coil heat exchanger. Two coils with coil to tube diameter ratios of 17.24 and 34.9 were used. In the experiments hot fluid was passed through the helical coil and air was passed through an insulated duct surrounding the helical coil. The helical coil had a nominal diameter of 9.5 mm. The experiments were conducted for laminar and turbulent regime for Re range of 1780 to 59500 and for the shell side air the range was from 3.6×10^4 to 1.5×10^5. The readings were taken for fully developed thermal and fluid boundary layers. It was observed that for both the plots of friction factor and $NuPr^{1/3}$ against the Dn number, a change in slope was evident near Dn = 500 and this change was attributed to the velocity profiles becoming skewed due to increase in centrifugal forces. The following correlations were given for the design of heat exchanger: For laminar regime:

$$f = \frac{64}{\mathrm{Re}}\left\{1-\left[1-\left(\frac{B}{Dn}\right)^{0.45}\right]^{2.22}\right\}^{-1} \tag{17}$$

And

$$Nu = A\left[\frac{f}{8}\mathrm{Re}_d^2\right]^{1/3}\mathrm{Pr}^{1/3} \tag{18}$$

B = 11.6 for 10 <Dn< 500, B = 6 for 500 <Dn< 1500 and A = 0.25 for 200 <Dn< 500. For turbulent regime:

$$\frac{f}{f_s} = 1+0.18\left[\mathrm{Re}\left(\frac{d}{D}\right)^2\right]^{0.25} \tag{19}$$

For the shell side the correlations are as follows:

$$f = 0.74\,\mathrm{Re}_{Dh}^{-0.25} \tag{20}$$

For D/d = 17.24,

$$f = 1.24\,\mathrm{Re}_{Dh}^{-0.25} \tag{21}$$

For D/d = 34.9 and

$$Nu = 0.057\,\mathrm{Re}_{Dh}^{0.8} \tag{22}$$

For D/d = 17.24,

$$Nu = 0.110\,\mathrm{Re}_{Dh}^{0.8} \tag{23}$$

For D/d = 34.9. The experimental results agreed with those of Seban and McLaughlin[3] for A = 0.25.

Rao [9] conducted an experimental study, in which the fanning friction factor and Nusselt number was derived from the experimental setup for Newtonian as well as for power law fluids. Range of Reynolds number used for the study was 25,000 to 50,000, for Prandtl number 6 to 14 and curvature ratio from 10 to 26. Range of n for power fluids was 0.78 to 1. It was inferred that the entrance length required for the full development of fluid flow, in helical coil, for power law fluid was found to be same as that for Newtonian fluids. Author observed that the correlation given by Mishra and Gupta with actual Reynolds number overestimated the friction factor, therefore the authors gave a new correlation for friction factor using generalized Reynolds number:

$$f = n^{0.4}\{0.079(\mathrm{Re}^*)^{-0.25} + \left[\left(\frac{d}{D}\right)^{1.5}/14\right] \tag{24}$$

For 20,000 < Re* < 50,000, 0.76 ≤ n ≤ 1, 10 < [D/d] < 26. According to his results it could be said that the effect of secondary flow was weaker when value of D/d was increased. Schmidt's correlation for $\left(\frac{Nu_c}{Nu_S}\right)$ also gave overestimated values therefore the author gave a new correlation for it.

$$\frac{Nu_c}{Nu_s} = 1 + 1.48\left(\frac{d}{D}\right)^{1.15} \quad (25)$$

For the parameters 15,000 < Re < 63,000 and 10 < d/D < 30. The following correlation was given for power-law fluids as

$$\frac{Nu_c}{Nu_s} = \left[1 + 2.9\left(\frac{d}{D}\right)\right]^{1.15}\left[0.55 + 0.45n\right]^{-1.25} \quad (26)$$

For the parameters 9,000 < Re* < 55,000 and 10 < d/ D < 30. Rao [9] concluded that the helical coil heat exchanger is not of much use, compared to straight tube heat exchanger in turbulent regime except for its compactness.

Prabhanjan et. al. [10] performed an experimental study on heat transfer in helical coil heat exchanger using water as the working fluid. The experiment was conducted in turbulent regime with Reynolds numbers in the range of 12,000 to 27,000. The author gave correlations between Rayleigh's number and Nusselt number for different characteristic length:

$$Nu_o = 0.009759(Ra)^{0.3972} \quad (27)$$

For Ra (5×10^{14} - 3×10^{15}) and tube length as characteristics length.

$$Nu_o = 0.0749(Ra)^{0.3421} \quad (28)$$

For Ra (9×10^{9} - 4×10^{11}) and coil height length as characteristics length.

$$Nu_o = 2.0487(Ra)^{0.1768} \quad (29)$$

For Ra (2×10^{6} - 3×10^{9}) and normalized length as characteristics length.

Data of heat transfer of oil in a helically coiled heat exchanger was experimentally derived by Ali [11]. Range of parameters used were Pr number from 250 to 400, coil to tube diameter ratio used were from 10 to 30.Correlations for average Nusselt number with the help of Rayleigh number was obtained. Experimental study was focused on a high Prandtl number. The correlation between Nu and Ra was given by him as :

$$Nu = 0.287 Ra^{0.323} \quad (30)$$

For $7.35 \times 10^{11} \leq Ra \leq 5.5 \times 10^{14}$

$$Nu = 0.619 Ra^{0.3} \quad (31)$$

For the parameters $4.37\times10^{10} \leq Ra \leq 5.5\times10^{14}$; $10 \leq D/d_o \leq 30$. The classification of flow into laminar and transition regimes in this study is according to change in the heat transfer coefficient, which is increasing or decreasing with respect to area. Correlation in terms of Gr and Pr number was given as

$$Nu = 0.555 Gr^{0.301} \Pr^{0.314} \quad (32)$$

For the parameters $1\times10^{8} \leq Gr \leq 5\times10^{14}$; $4.4 \leq Pr \leq 345$.

Shokouhmand et. al. [12]. conducted an experimental study in which he used the Wilson plot method to determine the overall heat transfer coefficients. They concluded that, the shell-side heat transfer coefficients of the coils increased with an increase in pitch. It was noted that there was a minute difference between the shell-side Nusselt numbers of counter-flow configuration and parallel-flow configuration.

The heat transfer coefficients of shell and helically coiled tube heat exchangers were found experimentally by Salimpour [13]. A correlation between the tube-side Nusselt number, Dean number, Prandtl number, and dimensionless coil pitch was obtained

$$Nu_i = 0.152 De^{0.431} \Pr^{1.06} \gamma^{-0.277} \quad (33)$$

And another one using Reynolds number as:

$$Nu_o = 19.64 \operatorname{Re}_o^{0.513} \Pr^{0.129} \gamma^{0.938} \quad (34)$$

In the study by Ghorbani et. al. [15] an experiment of heat transfer in a helical coil heat exchanger was done, for a range of Reynolds and Rayleigh numbers, curvature ratio and dimensionless coil pitch in both laminar and turbulent regimes. The aim of the experiment was to observe the effect of various parameters of the setup over the modified effectiveness (ε^{l}) of helically coiled tube heat exchangers. Author found that the mass flow rate of tube-side to shell-side ratio (R_m) was directly affecting the temperature change along the heat exchanger. And temperature varied quadratically when Rm is greater than 1 and logarithamically when tending towards 1 and also they concluded that the LMTD decreased with increase in Rm.

Bandpy et. al. [16] experimentally studied the effect of coil pitch on the shell side heat transfer coefficient. The tests were conducted for laminar as well as turbulent regime. The tests were conducted for different flow rates and different inlet temperatures. With increase in pitch for same inside tube diameter, the heat transfer coefficient was found to increase. Then for different mass flow rates and different inlet temperatures two different coils of different pitch were tested. For all test runs the heat transfer in coil with larger pitch was found to be more than in coil with smaller pitch.

Moawed [17] used forced convection on outside surface of helical coil and conducted tests on it. A range of curvature ratio from 7 to 16.142 and pitch ratio from 1.81 to 3.205 were used. A range of Reynolds number from 6.6×10^{2} to 2.3×10^{3} was used. The higher Nu value was observed at higher curvature ratios, as the surface area increased and the centrifugal force increased it promoted better heat transfer. The increase of pitch provided more space for air to do convective heat transfer. The correlation obtained:

$$Nu = 0.0345 \operatorname{Re}^{0.48}\left(\frac{d_o}{D}\right)^{-0.914}\left(\frac{h}{d_o}\right)^{0.281} \quad (35)$$

For $6.6\times 10^{2} \leq Re \leq 2.3 \times 10^{3}$; $0.1411 \leq d_o/D \leq 0.06195$ and $1.81 \leq h/d_o \leq 3.205$.

Raboh et al. [18] carried out experimental studies on condensation heat transfer coefficient for steam flowing through helical coils and the effect of different parameters on the heat transfer coefficient. All tested helical coils have the same outer surface area (0.1 m^2). Five values of inner diameters of pipe were tested; 3.36, 4.95, 11.3, 14.48 and 17.65 mm with coil diameters as 100, 125, 150, 200, and 250 mm. Also the helical coil was formed at different coil pitches (20, 30, 40, and 50 mm). The tested helical coil orientations were vertical (90 degree) and inclined positions with different angles (30, 45, 60 degree). It was found that the heat flux increased as the tube inner diameter was reduced and reached a maximum value for inner diameter of 4.95 mm. Then it decreased with decreasing diameter for 3.36 mm due to capillary effects. The heat transfer increased with decreasing coil diameter and this was due to the air flowing quickly around the smaller coil producing more cooling. Increase in pitch also increased the heat transfer rate and the optimum orientation angle for the coil was obtained as 45^0. Considering all the above parameters a correlation for the Nu number was proposed as follow:

$$Nu = 8.25(\mathrm{Re})^{0.426}(D/d_i)^{-0.1023}(P/d_i)^{0.03245}(L/d_i)^{-0.5352} \quad (36)$$

For 40 < Re < 230, 8.5 < D/di < 74.4, 1.7 < P/di < 10.1 and
94.6 < L/di < 1,994.

Reddy et al. [19] determined experimentally the heat transfer coefficients using a helical coil in an agitated vessel. Two tubes of different length were tested and it was found that the smaller length coil showed an increasing trend compared to larger coil for same flow rate and heat input.

Sanchetti et al. [20] conducted experiments on helical coil setup and concluded that considering thermal and transport properties of the heat transport medium as constant gives inaccurate values of heat transfer coefficients.

Purandare et. al. [21] did parametric analysis of helical coil heat exchanger using the various correlations. Four different correlations obtained by Salimpour, Kalb et al., Xin et al. Roger et al. were used for heat transfer analysis. In the laminar regime Nu and hi were found to increase with increase in Re.

Rainieri et. al. [22] conducted an experimental study for enhancing convective heat transfer of highly viscous fluids in helical coiled corrugated tubes. Under the uniform heat flux boundary condition, two coiled tubes with a curvature ratio of 0.06, one with smooth wall and the other with spirally corrugated was subjected for experimentation. The smooth coiled tube had heat transfer capabilities 3.6 times the straight section whereas corrugated coiled tubes showed nearly 8 times increase compared to straight tubes. Author gave a critical Dean number value, above which the wall corrugation effect starts to become effective for helically coiled tubes. For the geometry used by the authors the critical Dean number value was about 120 and Reynolds number 500. But they couldn't give a generalized correlation for finding critical dean number for the corrugation effect to start. Yordam et. al. [13] performed similar work and concluded that the use of the helically coiled corrugated tube made it possible to handle the same heat load with a shorter length of the tube.

The objective of the study by Pimenta [23] was to carry out an experiment to obtain heat transfer coefficients for Newtonian and non-Newtonian fluids at constant wall temperature as boundary condition, in fully developed laminar flow inside a helical coil. Coil to tube diameter ratio of 38.18 and pitch 11.34 mm was used. The earlier works only concluded that, pseudoplastic fluids have lesser heat transfer capability than Newtonian fluids, while dilatant fluids have higher heat transfer capabilities. Therefore the author realized that the elastic effect on the heat transfer coefficients in helical coils was not explored in depth. Correlations were presented in terms of Nusselt, Dean, Peclet and Weissenberg number.

$$Nu_c = 0.486\left[\left(\frac{3n+1}{4n}\right)^{0.275n}\left(0.717+0.993\frac{d}{D}\right)Pe^{0.275}\right]\left(1+0.728Dn_{(g)}^{0.225}\right)(Wi+1)^{0.011(n-1)} \quad (37)$$

For the parameters of glycerol solutions (15 < Dn < 1020, 10 < Pr <352); for CMC and XG solutions with power of 0.34 and 0.90 (4 < De < 487, 17< Pr <203, 32 <Wi < 19700). Nusselt numbers of the CMC solutions higher than that of the Newtonian fluids for the same parameters. And that of the XG solutions were quite lower than those of the Newtonian solutions.

Experimental research on convective heat transfer in helically coiled tube heat exchanger was carried out by Pawar and Sunnapwar [24]. The study was done in isothermal and non-isothermal conditions for Newtonian as well as power fluids. The coil to tube diameter ratios used were 13.21, 15.625 and 18.18. Total 276 tests were conducted in laminar as well as in turbulent regimes.

Al-Jabair et al. [25] conducted experiments to determine the heat transfer coefficients of shell and helically coiled tube heat exchangers. Three heat exchangers with different coil pitches were tested for both parallel-flow and counter-flow arrangement. The tube diameter was found to have negligible influence on the shell-side heat transfer coefficient but the coil pitch was found to affect the heat transfer considerably and with increase in pitch the heat transfer was found to increase.

Vijay D. et al. [26] worked experimentally to investigate the hydrodynamic and heat transfer analysis of three different geometries of tube in tube helical coil. This study was conducted over a range of Re from 2,500 to 6,700 using cold water in annulus side. The experiments were carried out in counter flow configuration with hot water in tube side and cold water in annulus side. The results showed that the 6 mm pitch wire wound tube in the inner tube helical coil has more overall heat transfer coefficient than that of 10 mm and plain tube helical coil. They experimentally obtained overall heat transfer coefficient for different values of flow rate in the inner-coiled tube and in the annulus region were reported. It was observed that the overall heat transfer coefficient increases with increase in the inner-coiled tube flow rate, for a constant flow rate in the annulus region. Similar observations were made in the variation of overall heat transfer coefficient for different flow rates in the annulus region for a constant flow rate in the inner-coiled tube. It was also observed that when wire coils are compared with a smooth tube, at constant pumping power, an increase in heat transfer rate is obtained at Reynolds numbers below 6,700. It was also observed that overall heat transfer coefficient increases with minimum pitch distance of wire coils.

Andhare et al. [27] carried out experimental studies to study the heat transfer coefficient considering pitch and curvature ratio. Three helical coil heat exchangers of different pitch and different curvatures were tested for counter flow arrangement. The following correlations were developed considering pitch ratio:

$$Nu_i = 0.858\left\{\left(\mathrm{Re}_i\sqrt{\frac{d}{D}}\right)^{0.7202}(\mathrm{Pr})^{-1.8224}\left(\frac{h}{\pi d}\right)^{0.0119}\right\} \quad (38)$$

$$Nu_o = 154.8103527\left\{(\mathrm{Re}_0)^{0.2427}(\mathrm{Pr})^{-0.3721}\left(\frac{h}{\pi d}\right)^{0.2982}\right\} \quad (39)$$

The following correlations were developed considering curvature ratio:

$$Nu_i = 31.908030\left\{(\mathrm{Re}_i)^{0.6542}(\mathrm{Pr})^{-0.1737}(d/D)^{0.8986}\right\} \quad (40)$$

$$Nu_o = 272.8977\left\{(\mathrm{Re}_o)^{0.1905}(\mathrm{Pr})^{-1.1936}(d/D)^{-0.1101}\right\} \quad (41)$$

It was concluded that the shell side heat transfer coefficient are larger than the tube side heat transfer coefficients.

Pawar and Sunnapwar [28] did the correlating of Nusselt number with dimensionless number, '*M*', Prandtl number and coil curvature ratio using least-squares power law fit is presented in this paper. Experiments were conducted for similar parameters and under similar conditions as that in their earlier paper Pawar and Sunnapwar [24]. In 2013 they didn't consider the coil curvature ratio as a direct function of Nusselt number which is necessary for constant mass flow rate and increasing or decreasing values of coil curvature ratio. Hence, this present work is undertaken to develop innovative correlations including the effect of coil curvature ratio in case of *M* and *Gz* number correlations for Newtonian fluids under different experimental conditions, development of Nusselt number and friction factor correlations based on non-Newtonian fluids. Correlation of nusselt number, M number, curvature ratio and Prandtl number obtained was:

$$Nu = 0.0049(M)^{1.2593}\delta^{0.3343}\mathrm{Pr}^{0.4} \qquad (42)$$

Applicable for the range of: 850≤M ≤ 3348, 3.83 ≤ Pr ≤7.308 and 0.055 ≤d ≤ 0.0757. Friction factor calculation based on present experimental data for non-Newtonian fluids under isothermal condition

$$f = 3.6273(Dn_{(g)})^{-0.972} \qquad (43)$$

Applicable for the range of: $27 \le Dn(g) \le 126$, $1.2961 \le n_{c'} \le 1.4633$ and 0.055 ≤ d ≤ 0.0757.

Dev et al. [29] conducted an experimental study to investigate the effects of heat transfer in a helical coil heat exchanger used for simple vapour absorption refrigeration system in laminar regimes. According to their findings it was concluded that for simple vapour absorption refrigeration system the helical coil heat exchanger could be used, which could bear a pressure of 11 bars and temperature up to 130°C. Because of scanty space available in the system, helical coil heat exchangers are preferable. Secondary flow pattern, imparts higher shear stress and turbulence for an available pressure drop, which can in turn increase the film coefficients by 40%.

Swapnil et al. [30] analyzed the performance of helical coil heat exchanger, in counter flow arrangement, experimentally and commented on the variation of dimensionless numbers. They plotted various graphs such as h_i vs Re_i, De vs Re_i, Nu_i vs Re_i and Nu_i vs De at different flow rates. According to the results, as Reynolds number increased the Nusselt number also increased complimenting the fact that as velocity increases the heat transfer coefficient also increases. Dean number in coiled and curved geometries signifies the turbulence of the flow. Dean number vs. Nusselt number plot showed that the increase in one corresponds to increase in another, which complements the fact that with increase in turbulence the heat transfer also increases.

Tejas Patil and Atul Patil [31] presented data for new phenomenon of heating of water flowing through helical coil using induction heating provided by induction cooker. Three different coils differing only in the number of turns were used. Tests on each coil were conducted for ten different flow rates from 15 lphtp 95 lph. They concluded from experimentation that as number of turns of helical coil increases the water outlet temperature also increases at constant flow rates and for the same coil the outlet temperature decreased as the flow rate was increased keeping other parameters constant.

Kshirsagar et. al. [32] did the experimentation on wire wound helical heat exchanger with annular tubes. The inner helical tube was wounded by a helical wire to increase turbulence. For smaller pitch of the wounded wire higher heat transfer was obtained than for larger pitch wire. Heat transfer also increased with increasing flow rate in the inner tube at constant flow rate in annular region.

Balachandran [33] did an experimental investigation on helically coiled heat exchanger using a nanofluid. CuO-water nano fluid was used as the test fluid. It is inferred that nano fluid coolant can absorb heat better than water as coolant at low flow rate. After their experimentation it was concluded that the performance of the helical coil heat exchanger using nano fluid are comparatively higher than that of water as coolant and was better than straight tube heat exchanger in both the cases.

The objective of the research, done by Hardik et al. [34] was to study the influence of the curvature factor and the Reynolds number on local heat transfer coefficient in helical coil using water as the study fluid. Coil to tube diameter of range 13.1 to 67 mm were used for experimentation, with a constant pitch of 50mm in all coils. The flow parameters set for the experiment was- Reynolds number from 217 to 191000 and Prandtl number from 3 to 5.6. Use of infrared thermal imaging technique was done for the measurement of temperature along the circumferential direction of the tube. The inner side and outer side wall temperature was calculated at eleven different circumferential locations. Temperature distribution on the inner side was non-uniform while it was more uniform on the outer side. Centrifugal force moves cold fluid towards the outer side of the helical coil and hot fluid moves towards the inner side. Hence, wall temperature on the outer side is less than that of the inner side. This temperature difference decreases with the decrease in centrifugal force. Centrifugal force of the coil having high curvature (low coil to tube diameter ratio) is higher than the coil having less curvature. Hence, the temperature difference between inner side and outer side is higher in helical coil 1 (D/d = 13.1) and lower in helical coil (D/d = 67). Difference in inner side Nusselt number and outer side Nusselt number is higher in coil having small coil to tube diameter ratio (D/d = 13.1). Nusselt number distribution of outer side for smallest curvature helical coil (D/d = 67) is uniform along the circumference. Regression analysis of the measured overall averaged total Nusselt number in the present study is given by

$$Nu = 0.0456\left(\frac{d}{D}\right)^{0.16}\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \qquad (44)$$

The conclusions that maybe drawn from the present study is –The transition from laminar flow to turbulent flow in helical coils is smoother than straight tube. Total Nusselt number and Nusselt number for outer side of helical coil decrease (from 100% to 20%) with decrease in curvature ratio. However, on the inner side, the Nusselt number increases mildly (from 25% to 35%) with the increase of coil to tube diameter ratio (D/d).

Gore et al. [35] compared straight tube and helical coil heat exchanger experimentally and then compared it with the help of CFD to infer that helical coil gives far superior heat transfer rates.

Marode et. al. [36] did the thermal analysis validation for different shapes of heat transfer coils and compared it. They used a setup in which the outer fluid was forced through a shell and it flowed over the outside surface of the tubes two fluids were used namely water and Al_2O_3 Nano fluid. Different geometries used were circular tube, elliptical type, twisted type and coil type. Out of four different tubes the twisted type of tube gave high heat transfer coefficient.

Gurav [37] did the post experimental analysis of tube-in-tube helical coil heat exchanger. He concluded that increase in curvature ratio delays the transformation of laminar flow to turbulent regime, which in turn increases the critical Reynolds number. With an increase in the pitch of coil, the heat transfer rates become similar to those of straight tube heat exchanger, for same flow rate.

Pawar et. al. [38] conducted experiments on 2 set of helical coils with different curvature ratios of 0.1136 and 0.0833 of same length of 5m. They tested it for both laminar and turbulent regimes under isothermal boundary conditions. As the helix diameter was decreased the centrifugal force subsequently increased, in turn increasing the overall heat transfer coefficient.

Puttewar and Andhare [39] performed thermal evaluation of shell and helical coil heat exchanger and also provided the design procedure for the same. From thermal evaluation it was concluded that the effectiveness of heat exchanger increases with increase in the mass flow rate of hot fluid flowing through the coil for constant flow rate of cold water flowing through the shell. When the flow rate of cold water was less the outlet temperature of hot water was higher and for maximum flow rate of cold water the heat transferred to the cold water and the outside overall heat transfer coefficient were highest.

Korane et. al. [40] conducted experimental analysis of circular and square helical coil heat exchanger. Hot water was made to flow through the coils and cold water was made to flow through the shell. The inner and outer heat transfer coefficients were calculated and two correlations were proposed as follows:

$$Nu_i = 0.23 Dn^{0.735} \Pr^{0.22} (d/D)^{0.01} \tag{45}$$

For circular geometries

$$Nu_i = 0.27 Dn^{0.77} \Pr^{0.2} (d/D)^{0.014} \tag{46}$$

For square geometries

It was observed that the heat transfer rate in square coiled tube was higher than circular coiled tubes and the corner radius curvature effect was the reason for this increased heat transfer.

Comparison of heat transfer between helical coil and straight tube heat exchanger

The objective of the research by Prabhajan et. al. [41] was to verify and check comparative advantage of helically coiled heat exchanger over the straight tube heat exchanger in terms of heat transfer. They focused on fluid to fluid heat exchange instead of constant wall temperature / constant heat flux was the parameter for other most of the studies. Material for tubes was copper and setup consisted of helical coil having length 6.38 m 15.7 mm tube inner diameter, thickness of 1.2 mm, coil diameter 203 mm, having 10 numbers of turns. Thermocouples were inserted into the pipe by drilling small holes to get fluid temperatures; they were glued by using epoxy to keep them in place and to prevent leakage into or out of the coil. The copper straight tube having equivalent length consisting 17 mm inner diameter was inserted inside a 1287 mm in diameter MS pipe having a wall thickness of 6 mm the circular portion between pipes were filled with water which was heated by using heaters. After conducting experiments results showed that the heat transfer coefficient of a helical coil was greater than the similarly dimensioned straight tube heat exchanger. Also in both the exchangers had higher heat transfer coefficients when the bath temperature was increased which may occur due to the increased in buoyancy effects. Rise in the fluid temperature can be affected by coil geometry and by the flow rate. All tests were performed in the transitional and turbulent regions.

The main objective of the study by Coronel and Sandeep [42] was to determine the convective heat transfer coefficient in both helical and straight tube heat exchangers under turbulent flow conditions and compare them. The experiments were conducted on two helical coils having different curvature ratios (d/D) 0.114 and 0.078 and for straight tube heat exchanger at various flow rates (1.89×10^{-4} to 6.21×10^{-4} m^3/s) along with different outlet temperatures ranges from 92 to 149 °C. After conducting experiments results showed that the overall heat transfer coefficient (U) in the Helical coil heat exchanger was much higher as compared with straight tube heat exchanger also it was found that the U for the coil having larger curvature ratio (d/D) were higher and lesser with smaller curvature ratio. On the basis of the U they also determined the inner (hi) and outer (h_o) convective heat transfer coefficients. They developed the relation between inside convective heat transfer coefficient (hi) and Re, Pr and d/D

$$Nu_i = 0.0302 \left(\frac{d}{D} \right)^{0.009} \mathrm{Re}^{0.85} \Pr^{0.4} \tag{47}$$

For 5x104 < Re < 3×10^5 ; 2.0 < Pr < 3.5; 0.078 < d/D < 0.114.

Gurav [43] in did a comparative study of heat transfer inside the helical annular coil and straight tube annular coil heat exchanger. From the results he inferred that the development of secondary flow increases with an increase in curvature ratio. The heat transfer coefficient for helical tube-in-tube arrangement is approximately 10 to 20 times that of straight tube-in-tube arrangement.

Shirigire and Kumar [44] carried out the comparative study between helical coil heat exchanger and straight tube heat exchanger. The effectiveness of heat exchanger is greatly affected by hot water mass flow rate and cold water mass flow rate when cold water mass flow rate is kept constant and hot water is passed through the coil at different flow rate, as flow rate is increased the effectiveness decrease. Increase in cold water mass flow rate for steady hot water mass flow rate resulted in increase in effectiveness, for both the coil geometries. Overall heat transfer was increased with increased in hot water mass flow rate and cold water mass flow rate. The highest overall heat transfer coefficient was noted for cold water mass flow rate 100LPH and hot water mass flow rate 100 LPH, in helical coil counter flow. They commented that the use of helical coil heat exchanger was seen to increase the heat transfer coefficient compared to similarly straight tube heat exchanger.

Ankanna and Reddy [45] performed parametric analysis of straight and helical tube heat exchanger. Both the tubes were tested for parallel and counter flow configurations. The effect of various parameters on the effectiveness was observed. For parallel configuration the effectiveness of helical coil increased with increase in flow rate and then decreased after a certain value whereas for straight tube effectiveness increases gradually and then shows a sudden increase. Similar observations were made for counter flow arrangement but the effectiveness was higher. As the inlet temperature increased overall heat transfer coefficient increased sharply and then declined and attained a constant value for helical parallel flow arrangement. For helical counter flow arrangement the overall heat transfer obtained was less than parallel arrangement. For straight tube, overall heat transfer coefficient showed periodic variation. It was found that effectiveness of helical coil heat exchanger was always more than that of straight tube and higher values were observed for counter flow configuration.

Gavade et. al. [46] performed analysis by conducting the experiments on helical (parallel and counter flow) and straight (parallel and counter flow) tube. Based on the results obtained they concluded that the helical pipe have more surface area which allows the fluid to be in contact for greater period of time period so that that there is an enhanced heat transfer compared to that of straight pipe. With increase in the mass flow rate the temperature drop was found to decrease due to less resident time.

Namrata et. al. [47] carried out the comparative study between helical coil heat exchanger and straight tube heat exchanger. From study they concluded that use of helical coil heat exchanger in counter flow is most effective whereas straight tube heat exchanger with parallel flow arrangement is least effective. They also observed that the overall heat transfer coefficient increases with increase in hot water mass flow rate and cold water mass flow rate. The coil pitch is found to have significance only in the developing section of heat transfer. The torsional forces induced by the pitch causes oscillations in the Nusselt number. However, the average Nusselt number is not affected by the coil pitch. Unlike the flow through a straight pipe, the centrifugal force caused due to the curvature of the pipe causes heavier fluid (water-phase) to flow along the outer side of the pipe. High velocity and high temperature are also observed along the outer side. The torsion caused by pitch of the coil makes the flow unsymmetrical about the horizontal plane of coil. As the pitch is increased, higher velocity and higher temperature regions are on the bottom half of the pipe. Increase in pipe diameter, keeping the inlet velocity constant, causes higher heat transfer coefficient and lower pressure drop. This effect is due to the influence of secondary flows. As the PCD is increased, the centrifugal forces decreases and this causes reduction of heat transfer coefficient and pressure drop. The coil parameters, viz., PCD and pipe diameter and void fraction at inlet have significant effect on the heat transfer.

Sreejith et. al. [48] experimentally analysed and compared helical and straight tube heat exchanger. The tests were conducted for both parallel and counter flow arrangement pattern. It was found that the effectiveness and overall heat transfer coefficient of helical coil heat exchanger are higher when compared to that of the straight tube heat exchanger for all the inlet temperatures and different mass flow rates. It was concluded that the secondary flow due to the centrifugal force was responsible for this enhancement in the heat transfer.

Conclusion

From above literature survey, it is noticed that there are various correlations developed based on experimental heat transfer data generated using helical coil heat exchangers for different geometries and parameters for single phase flow. However, very limited experimental data for comparison of overall heat transfer coefficient in helical coil and straight tube heat exchangers is reported in the literature. For future work, more experiments are required to conduct in comparison of straight tube and helical coil heat exchangers for same length and same experimental conditions to understand more clearly the actual effect of these two geometries on heat transfer.

Nomenclature

a inner radius of tube (m)
c_p specific heat at constant pressure (J/kg K)
d inner diameter of coil (m)
d_o outer diameter of coil (m)
D diameter of coil (m)
Dn Dean Number
$Dn_{(g)}$ Generalized dean number
Dh hydraulic diameter (m)
h_i inside convective heat transfer coefficient (W/m^2K)
h_o outside convective heat transfer coefficient (W/m^2K)
h coil pitch (m)
f Friction factor
f_c Friction factor of coiled tube
f_s Friction factor of straight tube
k thermal conductivity (W/m^2K)
L_c length of coil (m)
$\dot{m}$ mass flow rate (kg/s)
Re Reynolds number (vd_i/v)
N number of coil turns
Nu Nusselt number
Pr Prandtl number (v/α)
Pe Peclet number
Q heat transferred to cold water (W)
R mean helical radius of the coil (m)
R_m mass flow rate of tube-side to shell-side ratio(kg/sec)
Gz Graetz number ($\dot{m}\, c_p)/(kb\, L$)
T temperature (◦C)
T temperature (◦C)
U_o overall outside heat transfer coefficient (W/m^2K)
U_i overall inside heat transfer coefficient (W/m^2K)
v water velocity inside the coil (m/s)
Wi Weissenberg number

Greek symbols

ρ density of test fluid (kg/m^3)
δ coil curvature ratio (a/R)
α thermal diffusivity ($m^{-2}s^{-1}$)
v kinematic viscosity ($m^{-2}s^{-1}$)

Subscripts

c coil
cr critical
i inner
o outer
s straight

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