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A critical review on Flow, Heat transfer Characteristics and Correlations of Helically Coiled Heat Exchanger

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Abstract –*Helically Coiled Heat Exchangers are widely used due to their compact design, high heat transfer coefficient and easy manufacture. There are two types of heat transfer enhancement techniques: active and passive techniques. The heat transfer enhancement due to use of Helical Coils falls in the category of passive heat transfer technique. This paper provides a literature review on Manufacturing, Heat transfer and flow characteristics of single-phase and two-phase flow in curved tubes. The second section of this paper focuses on the correlations developed for the calculation of heat transfer coefficients.Various factors influencing the heat transfer in Shell and Helical Coil Heat Exchangers are discussed. Variation in heat transfer coefficients and their reasons are discussed.*

Keywords –*Helical Coil, Two Phase, Heat transfer enhancement, Pressure drop, Coil pitch, Laminar, Turbulent, Flow Patterns*

I. INTRODUCTION

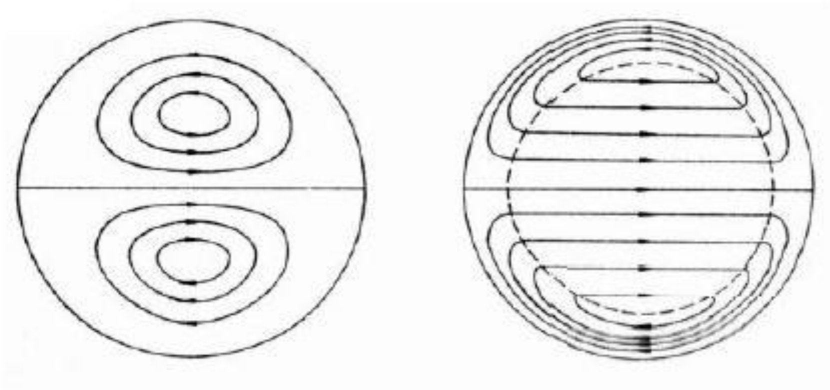
Heat exchangers are mainly used to transfer heat between two or more fluids having different temperatures. They are having applications in variety of fields, e.g. refrigeration and air conditioning systems, power engineering, petrochemical, pharmaceutical, aerospace, food processing industries and thermal processing plants. In any heat exchanger the aim of the designer is to always increase the heat transfer by some means.The heat transfer enhancement techniques can be grouped into two types: active and passive techniques. Active techniques make use of external forces such as electric field, acoustic, surface vibration. Passive techniques consist ofrough surface geometries, fluid additives, swirl flow devices, extended surfaces. Helical Coils are compact in structure and have high heat transfer coefficient, therefore theyare used as one of the passive heat transfer enhancement techniques and have large scope in various industries [1]. Secondary flow motion is induced by the coil curvature which creates turbulence and leads to high heat transfer coefficient. Numerical studies have been conducted to investigate the fluid flow and heat transfer characteristics existing in Helical coil for single phase and two phase flowand haveimportance in various fields.

II. PRELIMINARY RESEARCH ON HELICAL COIL

It has long been appreciated that the flow in helical pipe is considerably more complex than that in straight pipes. The earliest observation of this was done for open channel flow, where the effects of curvature are most evident and striking.

In 1911, Eustice [2] injected ink into water flowing through a coiled pipeand observed the existence of a secondary flow. The flow was found to change position continuously within the pipe, while exerting a 'scouring' action on the pipe walls. When a single filament of dye was introduced into the central plane, it split into two, leaving the central plane in opposite directions, forming a loop through the tube. With the increase in the flow velocitythe curvature of the filament was found to increase. Streamline flow occurs in straight pipe but in curved tube the dye eventually mixed throughout the tube diameter. This shows that the flow patterns occurring in curved tubes are significantly different over the straight tubes.

Dean [3] studied theoretically the flow in the curved pipe. The flow motion in coils was predicted to have two independent streamlines, in parallel planes going in opposite directions (Fig. 1). This figure shows the secondary flow which divides itself along the diameter of the tube. This constitutes two sets of distinct vortices. Superimposing the secondary flow on the motion along the tube, itwas shown that resulting flow of a fluid element corresponds to a skewed helical motion. These theoretical investigations of Dean were in complete agreement with dye injection experiments of Eustice.



De < 20 De > 100

Figure 1. Secondary flow for low and high Dean numbers

In 1929, Taylor [4] experimentally confirmed the presence of a secondary circulation in helical pipesby repeating Eustice's experiments. Coloured dye was slowly introduced through a small hole into the stream after one complete turn of the coil, so that the secondary circulation could be established. Eustice introduced the dye at the inlet section of coil so that the dye may mix with the flow while the secondary circulation was developing. It was found that the dye usually remains to one side of the tube. It first flowed inwards along the wall, until reaching the innermost point of the cross-section, then it left the wall and moved across the middle section and towards the wall again. The dye occasionally crossed to the other side, but this was explained to be due to imperfections in the uniformity of the helix. It was reported that, at a certain flow rate, the colour started to vibrate irregularly. Another important observation was that the colour retained its identity for at least one helix turn until the unsteadiness gave rise to diffusion by eddies with a rapid rise in flow resistance. After a further increase in the flow rate, turbulence was achieved and the colour dispersed.

III. HEAT TRANSFER IN HELICAL COIL

Patil et al. [5] described the procedure of designing helical coil heat exchanger and discussed advantages such as compactness and efficient use in laminar region. Equations are given to design Shell and Helical Coil and numerical example is undertaken.

Prabhanjan et al. [6]suggested the method forCoil fabrication.A comparison was done for heat transfer coefficient in case of double-pipe helical heat exchanger and straight pipe heat exchangers. After conducting experiments itwas found that the helical coil had a heat transfer coefficient of 1.16 and 1.43 times greater than for the straight pipe heat exchanger, at temperatures of 40 and 50 ºC.

Naphon et al. [7] studied the thermal performances and pressure drop of the helical-coil heat exchangers. Comparison was done first using helically crimped fins and secondly without using it. Two different coil diameters were selected for carrying out experimentation. Diameter of copper tube used in investigation was 9.5 mm and it consisted of thirteen turns. Hot and cold water were used as working fluids in the range of 0.10$\frac{kg}{s}$to 0.22$\frac{kg}{s}$ and 0.02$\frac{kg}{s}$ to 0.12$\frac{kg}{s}$, respectively. It was observed that as the mass flow rate of hot water increases, friction factor decreases.

Jayakumar et al. [8] carried out numerical investigation to find out the influence of coil parameters namely Pitch Circle Diameter, coil pitch and diameter of pipe on heat transfer. It has been seen that, in case of Helical pipe there occurs transition from laminar to turbulent flow regime at higher Reynolds number whereasin case of straight pipe transition takes place at lower Reynolds number. CFD analysis was undertakenand constant wall temperature boundary condition was selected for above analysis. Velocity contours were plotted at various planes along the length of the coil. It was found that when the pipe diameter is low, the Dean number is less and hence the secondary flow which depends on Dean numberis weak which leads to less mixing of the fluid. As the diameter of the coil increases the heat transfer at the outer surface is highest. The variation of heat transfer coefficient at different sections of coil was analyzed.

MATLAB software was used to carry out Multiple Regression analysis, the following correlation was generated for estimating the inner heat transfer coefficient.

$$\mathrm{Nu} = 0.025\, De^{0.9112}\, Pr^{0.4} \quad (1)$$

which is applicable to 2000 <De < 12,000.

Salimpour et al. [9] had carried out investigation with three coils having different coil pitches. Inner and outer heat transfer coefficients were computed and two correlations were proposed. Wilson plots were used for calculation of heat transfer coefficients for the shell-sideh_o and for the coiled tube-side h_i. The Wilson plot method is beneficial as it cancels the process of direct measurement of the surface temperature and hence removes the possibility of disturbanceof the fluid flow and heat transfer occurring due to measurement of those temperatures. It was seen that increasing the tube pitch leads to decrease in the inner Nusselt number.

Ghorbani et al. [10] had carried out experimental investigation on coil inside shell heat exchanger for the mixed convection heat transfer. Experiments were undertaken for laminar as well as turbulent flow inside coil. Calculations were performed at steady state. It was observed that as the mass flow rate increases, modified effectiveness decreases. Equation was developed to relate data of modified effectiveness with the mass flow rate ratio.

Zachar [11] studied natural convection heat transfer over the outer surface of helically coiled-tube heat exchangers.Geometrical parameters used in the analysis consist of curvature ratio 0.035−0.082.Flow parameters selected for the above analysis: Reynolds number $4000 \leq Re \leq 45000$. The results of the inner side heat transfer rate with existing empirical formulas were compared. Water has been chosen as the working fluid inside and outside of the coiled tube with Prandtl number range: $3 < Pr < 7$. Investigation was carried out for the outer side heat transfer rate along the helical tube axis. This was essential toevaluate the heat transfer at different location of the helical tube. It was found that the outer side heat transfer rate is slightly dependent on the flow rateat the inner portion of any helical tube. This dependence is because ofincreasing temperature differences between the tank working fluid temperature and the coil inlet temperature. It was concluded that as the mass flow rate inside is increased, it gives larger heat transfer rate on the outer surface of the helical coil along the axial direction. This takes place in case of heat flow direction from the tube working fluid to the Shell side when the coil inlet is located on top side of the Shell side.

Ebadian et al. [12] studied the combined effect of convection and thermal radiation heat transfer in three-dimensional laminar forced flow through a helical pipe. Helical pipe of finite pitch was selected and simulated with the CVFDM method. Thenumerical results with and without thermal radiationwere compared to study the effects of thermal radiation on the convective flow and heat transfer. On comparing it wasfound that the thermal radiation had no significant influence for flow and temperature fields especially in a fully developed region, moreover it furtherincreased the total heat transfer in the helical pipe. It was also found that there is no influence of thermal radiation on axial velocity and temperature fields when only the radiation participating medium is considered, but the buoyancy has more effects in temperature field than it has in velocity field.

IV. CORRELATIONS FOR HEAT TRANSFER

Kirpikov [13] tested coils with *D/d* ratios of 10, 13 and 18, two coils of *D/d* = 10 having markedly different *l/d* ratios (viz. 208 and 115). Water was heated in helical tubes by the steam which was used on the outside of the coil. Some unspecified allowances were made for the entry and exit sections; these involved a length coaxial with and a length perpendicular to the coil axis, and two bends, at each end of the coil. The heat-transfer coefficients were obtained using the wall to bulk temperature difference. Following correlation was proposed for the range:

$10^4 < Re < 4.5 \times 10^4$

$$(\mathrm{Nu})(\mathrm{Pr})^{-0.4} = 0.0456\left(\frac{\mathrm{d}}{\mathrm{D}}\right)(\mathrm{Re})^{0.8} \quad (2)$$

The properties were evaluated at the arithmetic mean of the bulk temperature of the fluid at inlet and outlet.

Rogers et al. [14] presented experimental results for forced convection heat transfer and friction factors. The experiments were carried out with water flowing through coils and these coils are heated by steam surrounding it. The range of Reynolds number covered was 3×10^3 to 5×10^4. Following correlation was presented:

$$(\mathrm{Nu}) = 0.023(\mathrm{Re})_{\mathrm{b}}^{0.85}(\mathrm{Pr})_{\mathrm{b}}^{0.4}\left(\frac{d}{D}\right)^{0.1}$$
(3)

Ali [15] carried out experimental study of turbulent natural convection heat transfer from vertical helical coil tubes to find average heat transfer coefficient. Experiment was carried out on 10 coils having different sizes. Nusselt number is correlated with Rayleigh number and the results are compared with available correlations for vertical plates and cylinders. Following correlations were presented:

$$\mathrm{Nu} = 0.685\,(\mathrm{Ra})^{0.295},\ 3 \times 10^{12} \leq \mathrm{Ra} \leq 8 \times 10^{14} \quad (4)$$

$$Nu = 0.0004\,(Ra)^{0.516},\ 6 \times 10^{12} \leq Ra \leq 1 \times 10^{14} \quad (5)$$

Ali [16] carried out experimental investigation on four helical coiled tubes under steady state laminar natural convection heat transfer from uniformly heated horizontal coils in the air. The purpose of the experiment was determination of average heat transfer coefficients including for constant heat flux. The experiment was carried out for the values of heat flux in the range of 500 – 5000 $\frac{W}{m^2}$. Nusselt number correlation was given with Rayleigh number for various values of heat fluxes using the coil tube diameter or horizontal coil axis distance as characteristic lengths. The developed correlations are:

$$Nu_d = 10824.2\ Ra_d^{-1.196},\ 340 \leq Ra_d \leq 645 \quad (6)$$

$$Nu_d = 187508\ Ra_d^{-1.526},\ 728 \leq Ra_d \leq 938 \quad (7)$$

Jaric et al. [17] carried out experimentation on three heat exchangers with concentric helical coils to find out the shell side heat transfer coefficient. Experiments were conducted with water flowing on the both sides of heat exchangers. The following correlation was presented in which Nusselt and Reynolds numbers are based on shell side hydraulic diameter.

$$Nu = 0.5 \times (Re)^{0.55} \times (Pr)^{1/3} \times \left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (8)$$

$$d_h = 4\,\frac{\text{Shell side volume}}{\text{Surface contact with fluid}} \quad (9)$$

The range of non dimensionless numbers used in the experiments were Re = 1000−9000 and Pr = 2.6−6.0, and the range of the hydraulic diameter was d_h = 9.1−18.3 mm.

Farhadi [18] carried out an experimental investigation to enhance the heat transfer rate in shell and coiled tube heat exchangers. Hot water flows in helical tube and cold water flows in the shell side. Tube and shell side heat transfer coefficients were determined using Wilson plots. Effect of fluid flow rate, coil diameter and coil pitch on heat transfer rate is determined by using experimental setup. Optimum condition was found by Taguchi method for the desired parameters in the range of $0.0813 < D < 0.116$, $13 < H < 18$, for the tube and shell flow rates of 1 to 4 LPM. It was found that the heat transfer rates increases with higher coil diameter, coil pitch and mass flow rate in shell and tube. The most important design parameters for coiled heat exchangers areshell side flow rate, coil diameter, tube side flow rate and coil pitch and was shown by Taguchi method.

V. FLOW PATTERNS IN HELICAL COIL

Peng et al. [19] carried out numerical simulation of steady laminar fluid flow in rectangular coiled pipes having circular cross section. Four rectangular coiled pipes at different Reynolds numbers and different straight tubes inclinations (9°, 15°, 30° and 45°) were taken. It was shown that temperature distributions are affected by the flow pattern in the coiled tubes. For pipes having 9° and 15° angles, temperature profile in bends is symmetric to horizontal centerline of the cross-section through the bends and because of this phenomenon better heat transfer performance is achieved as there is much uniform fluid distribution through the tube. In case of larger angles (30° and 45°) fluid accumulates on one side (top or bottom) of the wall and a very little fluid flows on the opposite side where very less heat transfer occurs from tube wall to the fluid. It was concluded that secondary motions in the bends and the flow redevelopment in the straight-tubes help fluids with different temperatures to mix and enhance heat transfer.

Xin et al. [20] had carried out an investigation on the single-phase and two-phase air–water flow pressure drop in annular helical pipes with horizontal and vertical orientations. The water side Reynolds number range selected for performing the experiments was from 210 to 23,000 and for air side the Reynolds number was selected from 30 to 30,000. A correlation was proposed for calculating friction factor for single-phase flow in laminar, transition and turbulent flow regime. It was observed that the Dean number range is approximately 800 to 4000 for transition from laminar to turbulent flow. Also in annular helical pipe there is no clear boundary between laminar flow and turbulent flow. The two-phase flow pressure drop multiplier in annular helical pipe was found to be dependent on the Lockhart–Martinelli parameter and the flow rate of air or water. As the pipe diameter decreases the effect of flow rate tend to decrease.

Soliman et al. [21] investigated the flow patterns occurring inside the helical coil during condensation. These flow patterns affect heat and momentum transfer. Gravitational and inertial forces acting on liquid film acting along condensation path plays an important role in heat transfer process. Liquid film thickness increases along the condensation path. As the influence of gravitational force increases alongwith the centrifugal forces along the condensation path, the liquid film loses its symmetry. There will be large layer of liquid at the bottom of tube. Various correlations suggested by different authors were tested in these flow regimes.

Liebenberg et al. [22] carried out investigation on two phase flow patterns inside coil (formation of slug, plug, bubbles). Video images of condensing R-134a were taken at mass fluxes of 300, 500, and 800 $\frac{kg}{m^2s}$ in a smooth tube. Froude rate parameter versus heat transfer coefficient graphs were plotted and then based on the graph experimental visual observations were made. It was observed that change in the slopes of the curves occurs at certain point .This change is due to the transition from a shear-dominated flow to a gravity-dominated flow. Flow patterns in smooth tubes and microfin tubes were compared at various vapour qualities. Higher heat transfer coefficients were observed in microfin tubes. This is due to delay in transition from annular to intermittent flow.

Wongwises et al. [23] carried out investigation on condensation of R-134a in a helically coiled heat exchanger. It was observed that pressure drop and heat transfer coefficients are higher for helically coiled tube heat exchanger as compared to the straight tube heat exchanger. It was shown that heat transfer coefficient increases with increase in average vapor quality.

Alhajeri et al. [24] investigated the condensation heat transfer of R-134a flowing through annular helical tubes. The experiment on R-134a was performed for mass flow flux range from 50 to 680 $\frac{kg}{m^2s}$. Effect on average heat transfer coefficient with Refrigerant mass flux was studied. Variation of Overall heat transfer coefficient with saturation temperature was studied.

VI. CONCLUSIONS

From the literature it was observed that:

[1] For helical coil tubes, the literature survey carried out in above paper indicates that various numerical and experimental work is carried out in the form of flow characteristics inside coil in case of single phase and two phase flow, but the investigation related to two-phase heat transfer characteristics have not been carried out onlarge scale.

[2] Combined effects of active and passive method on the heat transfer enhancement and pressure drop have to be carried out.

[3] Dependanceof heat transfer coefficient on different inclinations of Helical Coil has to be studied.

[4] Correlations are to be developed for the heat transfer enhancement due to use of nanofluid inside Helical Coil.

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