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CHARACTERISTICS OF A CONICAL COIL HEAT EXCHANGER'S HEAT TRANSMISSION AND FLOW

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ABSTRACT

The current study focuses on experimental and numerical analyses of the fluid flow characteristics and heat transfer properties of a conical coil twin pipe heat exchanger. On the Nusselt number and overall heat transfer coefficient, the effects of the Dean number, curvature, and torsion are discussed. The results of the experiments are used to validate the numerical scheme. The Wilson plot is used to compute the overall heat transfer coefficient and to determine the heat transfer coefficient. The findings demonstrated that the inner Nusselt number is significantly influenced by the Dean numbers in the annulus and inner tube. The ratio of the mass flow rates of hot and cold fluids has an impact on the inner Nusselt number as well. The overall heat transfer coefficient is improved by using conical exchangers rather than helical ones. Conical coil heat exchangers have a slightly greater friction factor, though. There is an ideal cone angle, and the overall heat transfer coefficient changes with cone angle. It is suggested to estimate the Nusselt number using a correlation between the Prandtl and Dean numbers.

INTRODUCTION

Applications for heat exchangers include refrigeration, food processing, and heat recovery systems. Due to space limitations, the size of the heat exchanger is a crucial consideration in the majority of applications. By improving the heat transfer properties, the size of the heat exchanger can be decreased. Active and passive procedures are the two categories under which strategies for enhancing heat transfer characteristics are categorised. In the first scenario, vibration, an electric field, acoustic forces, injection, suction, etc. are employed to increase heat transfer, but in the second scenario, the geometry is changed, there are inserts in the flow path, and the heat transfer fluid is given additives. In comparison to straight pipes, helical coil and spiral coil heat exchangers have high heat transfer rates, particularly in the laminar regime. Heat transfer coefficient and friction factor are higher in a curved pipe than in a straight one due to the secondary flow motion caused by the curvature effect and the resulting centrifugal force. Additionally, the temperature and velocity fields are made more complicated by the torsion of helically coiled tubes. The spirally and helical coiled forms are combined in the conical coiled heat exchanger. Lin et al.'s (1997) investigation of three-dimensional laminar forced flow and heat transfer at the entry zone of helical pipes revealed that laminar flow at higher Reynolds numbers results in a weaker oscillation of the average friction factor than that at a lower Reynolds number. Ali (1998) suggested a correlation to measure the heat transfer coefficient in horizontally oriented, uniformly heated helical tubes placed in air. The benefit of a helically coiled heat exchanger over a straight tube heat exchanger for heating liquids was demonstrated by Prabhanjan et al. in 2002. The zones of transition and turbulent flow were used for the experiments. An experimental study on the heat transmission from helical coiled tubes submerged in water was conducted by Prabhajan et al. in 2004. As a function of Rayleigh number, a correlation for outside Nusselt number was proposed. Experimental research on steady state natural convection heat transfer from uniformly heated helicoidal pipes was done by Moawed (2005). The ratios of coil diameter to pipe diameter (D/do), pitch to pipe diameter (p/do), and length to pipe diameter (L/do) all rose, which led to a rise in the Nusselt number. A two pipe helical heat exchanger in the laminar regime was the subject of numerical research by Rennie and Raghavan in 2006. Both parallel flow and counter flow total heat transfer coefficients were estimated, and a link between annulus Nusselt number and modified Dean number was found. The heat transfer parameters of a tube-in-tube heat exchanger were investigated experimentally and numerically by Kumar et al. (2006) while operating in the counter-current mode. For a fixed flow rate in the annulus region, it was found that the overall heat transfer coefficient increases with an increase in the inner-coiled tube Dean number. In their experiments on a helical coil heat exchanger using air and water as the heat transfer fluids, Shokouhmand and Salimpour (2007) suggested a correlation for calculating the heat transfer coefficient. In order to comprehend forced laminar flow in rectangular coiled pipes with circular cross sections, Conte and Peng (2008) carried out numerical investigation. Smaller angle of straight tube inclination in the coil led to better heat transfer performance. To determine the impact of inner tube shape, Chen and Dung (2008) performed numerical research on double tube heat exchangers. Oval-cross section inner tubes demonstrated superior heat transfer properties. According to Xiaowen and Lee (2009), using a helical heat exchanger improved the coefficient of performance of a household water-cooled air conditioner. Mandal and Nigam (2009) looked at the fluid-to-fluid heat transfer properties of compressed air with turbulent flow in a two pipe helical heat exchanger. Correlations between the Nusselt number and friction factor in the inner and outer tubes were suggested. A correlation for the heat transfer coefficient for flow between concentric helical coils was developed by Kharat et al. in 2009. The numerical simulations of vertically oriented helical coils with turbulent water flow were done by Jayakumar et al. in 2010. They suggested correlations for estimating the average Nusselt number under the boundary conditions of constant wall temperature and constant wall heat flux. Experimental research on mixed convection heat transfer in a coil in shell heat exchanger was done by Ghorbani et al. (2010). It was discovered that as coil pitch increased, the convection heat transfer coefficient of the shell-side also increased. The outcomes showed that the ideal characteristic length is the comparable diameter of the shell. Using the refrigerant R134a as the heat transfer fluid, Chen et al. (2011) conducted critical heat flux studies on horizontal helically coiled tubes and found that the coil to diameter ratio is more significant than the length to diameter ratio. Experimental research on forced convection heat transfer from a constant heat flux helical coil tube was done by Moawed (2011). Ke et al. (2011) quantitatively examined the conical spiral heat exchanger's heat transfer characteristics and discovered that the cone angle and cross section have a significant impact on heat transfer while pitch has minimal bearing on enhancing heat transmission. According to another research, if the cross-sectional area is held constant, the circular section's heat transfer coefficient is greater than the elliptical section's. Ice slurries' flow and heat transmission characteristics in a helically-coiled pipe were investigated by Haruki and Horibe in 2013. The interaction of the centrifugal and buoyant forces caused by secondary flow significantly affects the flow resistance. The latent heat of the ice particles, on the other hand, had an impact on the heat transmission coefficient. In a 2013 study, Shinde and Dange compared cone-shaped coils against straightforward helical coils. The effectiveness and heat transfer rate of the conical-shaped heat exchanger are higher than those of the helical coil heat exchanger, according to the experiments, which were carried out for various flow rates in laminar and turbulent flow regimes. The thermal performance of the conical coil heat exchanger with a 90° conical coil heat exchanger was the subject of an experimental examination by Purandare et al. (2014). For forecasting Nusselt number in terms of Dean number and Prandtl number, empirical correlations were presented. Joshi and Anand (2015) examined the heat transfer properties of a conical helical coil heat exchanger utilised in a waste heat recovery system and discovered that the cone-shaped helical coil had heat transfer rates that are 1.18–1.38 times greater than those of a simple helical coil. In an experimental study on shell and conical coil heat exchangers, Purandare et al. (2015) discovered that the efficiency of conical coil heat exchangers decreases when the ratio of the mass flow rates of hot and cold fluids rises. According to Mahmoudi et al.'s (2017) investigation of the forced convection heat transfer and pressure drop in helically coiled pipes employing TiO_2/water nanofluid, Dean number significantly affects heat transfer for a given Reynolds number. Experimental research on the heat transfer properties of super-critical CO_2 in helically coiled tubes with continuous wall heat flow was conducted by Xu et al. (2018). Results showed that for supercritical CO_2, the dimensionless exergy destruction brought on by irreversible heat transfer is substantially more than that brought on by flow friction. An experimental examination of R-600a flow via a concentrically arranged, helically coiled capillary tube suction line exposed to ambient conditions was provided by Dubba and Kumar (2018). When helical coils were used instead of straight tubes, mass flow was reduced by 3% to 12%. Izadpanah et al. (2018) investigated the heat transfer by natural convection over a helically coiled heat exchanger's exterior surface inside a water tank. A power law correlation for the Nusselt number was shown after studying the impact of coil diameter, pitch, turns, and mass flow rate. Using nanofluids can enhance the thermal-hydraulic performance of conical heat exchangers, according to Jamshidi and Mosaffab's (2018) investigation on the energy extraction capacity of conical helical heat exchangers for geothermal applications. According to the literature review, there have been enough studies done on the heat transfer properties of helical and spiral coil heat exchangers. The features of conical coil tube in tube heat exchangers for heat transfer have only been the subject of a very small number of investigations. This study's major goal is to offer practical knowledge on the thermal and flow properties of double pipe conical coil heat exchangers. The conjugate heat transfer algorithm found in Fluent is used to calculate heat transfer in the numerical research. Experimental findings are used to validate the outcomes of numerical investigations. A correlation between the Nusselt number and the Dean number and Prandtl number is suggested in light of the numerous numerical investigations.

EXPERIMENTAL SETUP AND PROCEDURE

Conical coil tube in tube heat exchanger heat transfer and flow properties have been studied through experiments.

GEOMETRICAL DETAILS OF THE HEAT EXCHANGER

Figure 4.1 depicts the fundamental geometry of a conical coil pipe, while Figure 4.2 shows the cross-sectional features of the tube. The tube in tube conical coil heat exchanger employed for the current experimental work is shown in detail geometrically in Table 4.1.

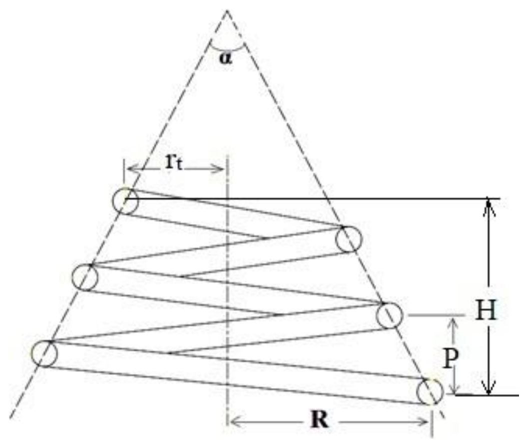


Figure 4.1 Basic geometry of a conical coil pipe

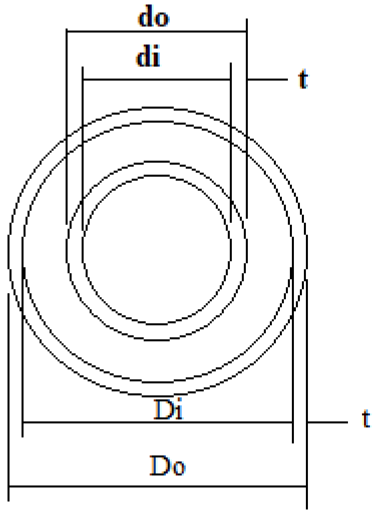


Figure 4.2 Tube cross-section

EXPERIMENTAL SETUP

Figure 4.3 displays the experimental setup's schematic diagram. The conical coil heat exchanger, which served as the test part, was built using copper tubing and regular copper connections. Numerical studies are used to determine the cone angle for the coil in the conical coil heat exchanger case. The geometry is chosen to use the cone angle with the best value, which is 72 degrees. In the conical coil heat exchanger, a coil with the same base curvature, pitch, and length as the helical coil is utilised for comparison. A 72° cone angle, 2.966 m long conical coil heat exchanger has 2.5 turns. Two layers of insulation are supplied on the exterior surface of the heat exchanger to prevent heat loss to the surroundings.Exchanger. Insulation made of polyurethane foam (PUF) is placed on top of an asbestos rope in the second layer. Hot water is cycled through the inner tube while cold water is passed through the annulus.The hot and cold fluids are pumped using centrifugal pumps, and valves are utilised to control the flow rate. Hot water is provided to the house using a 12 litre electric heater. Table 4.1 Dimensions of coil used in the present study

Symbols	Parameter	Dimensions
d_o	Outside diameter of inner tube	9.5 mm
d_i	Inside diameter of inner tube	7.9 mm
T	Wall thickness	0.8 mm
D_o	Outside diameter of outer tube	15.9 mm
D_i	Inside diameter of outer tube	14.3 mm
R	Base Radius of coil	235.9 mm

H	Height of the coil	119.6mm
r_t	Top radius of coil	149 mm
P	Axial pitch of coil	48 mm
α	Cone angle	72°
L	Total length of coil	2.966 m

Salient points in the heat exchanger are monitored for temperature using calibrated T type thermocouples. The thermocouples are accurate to within 0.5 °C. The flow rates are measured using flow metres, which have an accuracy of 1% of full scale. With a 0.5% reading accuracy, differential pressure gauges are used to measure the pressure drops in the inner and annulus tubes. Data about temperature is collected using a data gathering system. The estimated Nusselt number has a maximum uncertainty of 5.85%.Figure 4.4 depicts the several steps in the creation of the experimental instrument. The tube is wound on a metallic pattern with a conical contour to create the twin tube helical cone coil. Very fine salt powder was placed tightly into the inner and annular spaces before the tube was wound into a tube structure. The salt powder was bent and then removed with water. During the bending operation, care was taken to maintain the coil's round cross-section, and distortion was kept to a minimum.

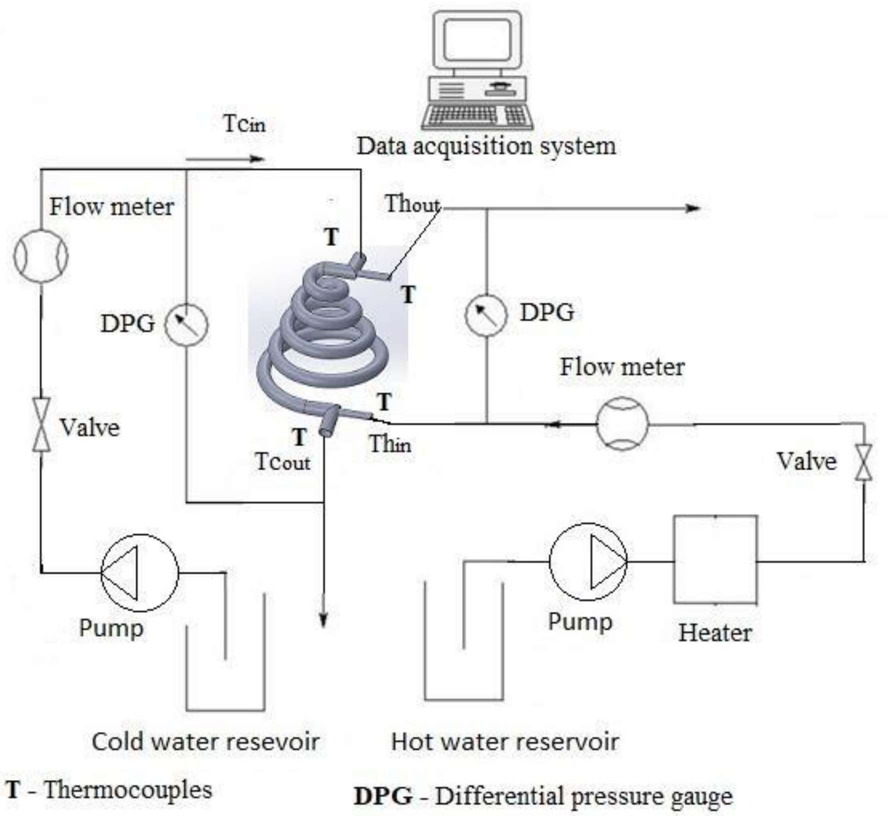


Figure 4.3 Experimental setup

Figure 4.4 Fabrication of conical coil heat exchanger

Additionally, attention was paid to bending with a steady pitch. The copper tube ends had the end connectors soldered on. Hot water for both coils can enter and escape with ease because to the provisions that have been created. Two layers of thermal insulation were added on the outside of the heat exchanger to stop heat from escaping to the surroundings. Insulation consisting of polyurethane foam (PUF) made up the first layer, and asbestos rope made up the second.

EXPERIMENTAL PROCEDURE

Conical coil tube in tube heat exchanger experiments are carried out in a counterflow configuration. Laminar flow was produced for both fluids by varying the flow rates in the annulus and the inner tube within reasonable bounds. This limit is determined using correlations that Ito (1969) suggested.

Critical Reynolds number for the helical coil is given by:

$$Re_{crit} = 20000\left(\frac{r}{R}\right)^{0.32} \quad - \qquad (4.1)$$

According to the aforementioned correlation, the critical flow rates for inner and annulus flows in this shape are 0.960 lpm and 4.23 lpm, respectively. The flow rates were adjusted in increments of 0.1 lpm from 0.1 to 1 lpm. We examined each and every combination of these flow rates in both the annulus and the inner tube. As a result, the inner Dean number fell between 97-972 (Reynolds number between 479-4799) while the annulus Dean number fell between 25 and 251 (Reynolds number between 159 and 1592). The hot fluid's input temperature is between 44 and 64 degrees Celsius, whereas the cold fluid's is between 26 and 28 degrees Celsius. The hot fluid's exit temperature is between 26 and 52 degrees Celsius, and the cold fluid's is between 29 and 60 degrees Celsius. Following the system's attainment of steady-state, measurements of the flow rates, pressure drop, and temperatures were taken to be used in the computations.

DATA REDUCTION

The heat transfer from the hot and cold fluids are calculated by applying theenergy balance equation, as shown below.The rate of heat transfer,

$$Q = m_h C_h (T_{h1} - T_{h2}) = m_c C_c (T_{c2} - T_{c1}) \qquad (4.2)$$

where m_h- mass flow rate of hot fluidm_c- mass

flow rate of hot coldC_h – specific heat of

hot fluid C_c – specific heat of cold fluid

T_{h1}- inlet temperature of hot fluid T_{h2}- outlet

temperature of hot fluid T_{c1}- inlet temperature

of cold fluid T_{c2}- outlet temperature of cold

fluid

Overall heat transfer coefficient

$$U_{io} = \frac{Q}{A_{io} LMTD} \quad (4.3)$$

Outside surface area of the inner tube,

$$A_{io} = \pi d_o L \quad (4.4)$$

Where L is the length of the tube and d_o is the outer diameter of the inner tube.

Log mean temperature difference,

$$LMTD = \frac{(\Delta T_2 - \Delta T_1)}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \quad (4.5)$$

Where, $\Delta T_1 = T_{h1} - T_{c2}$ and $\Delta T_2 = T_{h2} - T_{c1}$

Wilson plots are used to calculate the heat transfer coefficients at the inner and outer surfaces of the inner tube. By keeping the annulus flow rate constant, the flow of hot fluid in the inner tube is adjusted from 0.1 lpm to 1 lpm with a 0.1 lpm interval.

The overall heat transfer coefficient can be related to the inner and outer heat transfer coefficients by the following equation,

$$\frac{1}{U_O} = \frac{A_{io}}{A_{ii} h_i} + \frac{A_{io} \ln (d_o/d_i)}{2\pi k L} + \frac{1}{h_o} \quad (4.6)$$

Where A_{ii} and A_{io} are the inside and outside surface areas of the inner tube

d_i and d_o are the inside and outside diameters of the inner tubek is the

thermal conductivity of the copper tube

L is the length of the heat exchanger.

The only variables left in the equation above that are unknown after calculating the total heat transfer coefficient are the heat transfer coefficients. It is then assumed that the annulus heat transfer coefficient is constant by maintaining a constant mass flow rate in the annulus tube. According to the assumption, the inner heat transfer coefficient will behave as follows when the inner tube's fluid velocity, Vi,

$$h_i = C{V_i}^n \tag{4.7}$$

This is substituted in the above equation as shown below,

$$\frac{1}{U_O} = H + C^{-1}{V_i}^{-n} \tag{4.8}$$

where n is approximately taken as 0.8 for single-phase, forced convection problems,

$$H = \frac{A_{ii}}{A_{io}h_o} + \frac{A_{ii}\ln(d_o/d_i)}{2\pi kL} \tag{4.9}$$

The values of C and H were determined through curve fitting. C is obtained fromthe slope of the plot between $1/U_0$ and ${V_i}^{-n}$and H from the y-intercept. The values of h_i and h_o were then calculated. This procedure was repeated for all the ten inner flow rates.The inner and annulus Nusselt numbers were calculated from the above values of h_i and h_o by using the following relation,

The Nusselt number,

$$Nu = \frac{hd}{k_f} \tag{4.10}$$

Where h is the heat transfer coefficientd is the

characteristic dimension

k_f is the thermal conductivity of water

The Prandtl number was calculated by using the following relation,

$$Pr = \frac{\mu C_p}{k_f} \quad (4.11)$$

Where μ is the viscosity of the fluid.

Prandtl number shows the relative thickness of the velocity boundary layer to thethermal boundary layer.

Dean number,

$$De = Re\sqrt{\frac{r}{Rm}} \quad (4.12)$$

Where Rm is the mean coil radius.

The head loss in the inner tube and annulus is measured using a differential pressure gauge. From the pressure drop, the value of the friction factor is found out usingthe formula,

$$f = \frac{\Delta P}{\rho\left(\frac{V^2}{2}\right)\left(\frac{L}{d}\right)} \quad (4.13)$$

Where ΔP is the pressure drop between inlet and outlet, v is the mean velocity offlow, ρ is the density, L is the length, and d is the hydraulic diameter of the selected tube.

Based on the nature of the correlations available in the literature, a correlation ofthe following form is appropriate.

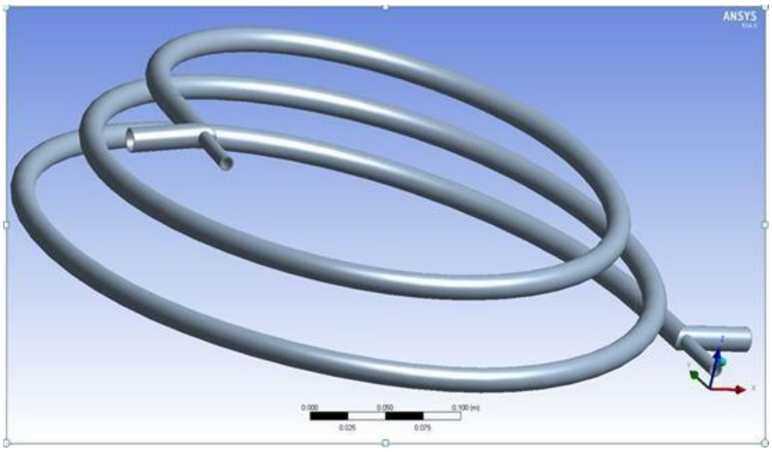
$$Nu = C\,(De)^x\,(Pr)^y \quad (4.14)$$

After linearization of the function and using the regression analysis (available inexcel software), the values of C, x and y are obtained.

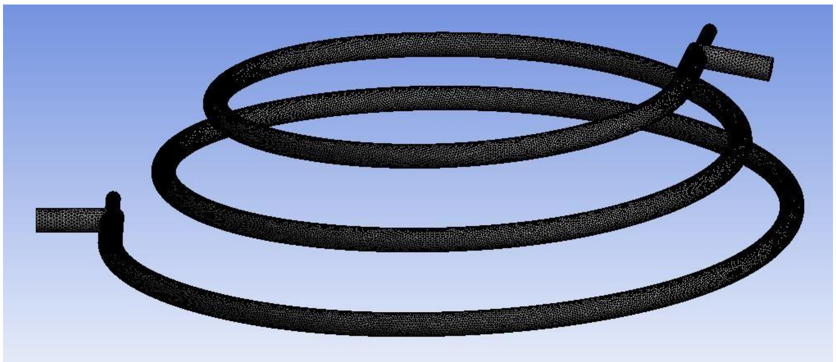
NUMERICAL SETUP

With the help of AutoCAD 14.0 and ANSYS 14.5 Design Modeller, the conical tube heat exchanger's 3D geometry is modelled. The ANSYS 14.5 workbench's meshing module was used to produce the mesh. A mesh that was first created was rather coarser. Tetra and hexahedral mixed cells, with triangular and quadrilateral faces at the borders, make up this mesh. The utilisation of structured hexahedral cells is carefully considered. By carefully constructing the mesh, especially close to the wall region, the goal is to minimise numerical diffusion. Later, a fine mesh is created, and all of the solid-fluid wall contacts are given inflation layers.

The geometry created for validation is shown in figure 4.5. The heat transfer and flowanalysis of the tube in the tube heat exchanger is carried out using Fluent 14.5



(a) Model of the conical coil heat exchanger



(b) Meshed geometry

Figure 4.5 Conical coil heat exchanger

Table 4.2 Major and minor diameters of elliptical cross-section

Coil	a (mm)	b (mm)	a/b ratio
Coil 1	5	3.1205	1.6023
Coil 2	4.5	3.46722	1.2979
Coil 3	4.2	3.71488	1.1306
Coil 4	3.95	3.95	1
Coil 5	3.1205	5	0.8845
Coil 6	3.4672	4.5	0.7705
Coil 7	3.7149	4.2	0.624

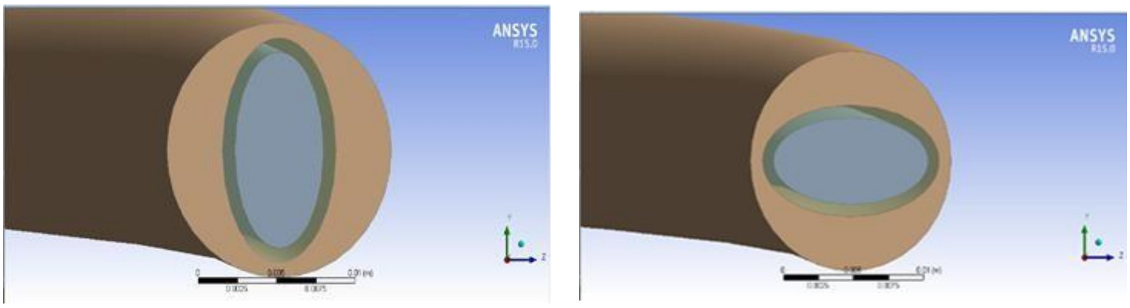
In order to investigate the heat transfer properties of a conical helical coil tube in tube heat exchanger, numerical analysis has been done by changing geometric parameters, namely the cross-section of the inner tube. By keeping the inner surface area of the elliptical tube constant, the inner pipe cross-section is made elliptical while the outside pipe cross-section remains round. By changing the major diameter (a) to minor diameter (b) ratios of the inner elliptical tube, much geometry can be produced. Figure 4.6 depicts the configuration of the inner elliptical tube inside the outer circular tube. With the exception of the inner tube properties, the dimensions of the helical coil heat exchanger with elliptical cross-section are the same as those mentioned before. Table 4.2 lists various major and minor diameter combinations that have been employed.

The following expression is used to determine the hydraulic diameter of an elliptical cross-section

$$H_d = \frac{4\pi ab}{\pi(3(a+b)-\sqrt{3(a+b)(a+3b)})} \quad (4.15)$$

Where a = semi-major axisb = semi-

minor axis



a/b < 1 **a/b > 1**

Figure 4.6 Configurations of the inner elliptical tube inside the circular tube

RESULTS AND DISCUSSION

A grid-independent research similar to the one described below was completed before the comprehensive numerical study.

GRID INDEPENDENT STUDY

The grid is iteratedly modified utilising edge sizing and inflation growth rates in a grid independence analysis (figure 4.7). This is done again until the tolerance between two refinements is roughly 1%.

VALIDATION OF THE NUMERICAL SCHEME

The impact of the inner Dean number on the rate of heat transmission is depicted in Figure 4.8. For comparison, the results of experimental research are also displayed. The results from experimental and numerical research are fairly in agreement. When the inner Dean number is raised from 97 to 972, the heat transfer rate for an annulus flow rate of 0.2 lpm increases by 187%, and for an annulus flow rate of 0.9 lpm, the increase is 668%. When the outer flow rate is large, the inner Dean number has a greater impact on the rate of heat transfer. The inner and outer surfaces of the inner tube's heat transfer coefficients have an impact on the rate of heat transmission. Even at high Dean numbers in the inner tube, low flow rates in the annulus cause the heat transfer coefficient at the outer surface of the inner to be low, which leads to low values for the overall heat transfer coefficient.

EFFECT OF DEAN NUMBER ON INNER NUSSELT NUMBER

The inner Nusselt number's experimental values as they relate to the Dean number are displayed in Figure 4.9. For comparison, the results of numerical investigations are also provided. The largest variance is only 9%, and the numerical results accord with the experimental data. Heat transfer coefficients are computed using the Wilson plot method using experimental data. When the Dean number changed from 97 to 972, the Inner Nusselt number increased by 85.7%. Purandare et al. (2015) report a similar pattern of trends. The secondary flow increases together with the tube side fluid velocity. The improved secondary flow guarantees that the fluid inside the tube is properly mixed, which raises the heat transfer coefficient.

EFFECT OF ANNULUS REYNOLDS NUMBER ON INNER NUSSELT NUMBER

Variations in the annulus flow rate have been used in research to examine the impact of annular Reynolds number on the inner Nusselt number. Figure 4.10 displays how the inner Nusselt number and inner Dean number change for different annulus Reynolds numbers. The temperature gradient across the fluids is lessened as the annulus mass flow rate and annular Reynolds number increase. The mean temperature of the hot fluid drops because the cold fluid may absorb more heat. As a result, the inner Nusselt number decreases as the annulus Reynolds number increases.

EFFECT OF ANNULUS DEAN NUMBER ON ANNULUS NUSSELT NUMBER.

The impact of the annulus Dean number on the annulus Nusselt number is depicted in Figure 4.11. The results of the numerical analyses are compared to the outcomes of the experiments. The greatest variance is 5%, and there is good agreement between the computational and experimental data. Dean number increases by 187% when the number of annuli is changed from 25 to 250. As the annulus' flow rate rises, centrifugal force also rises, enhancing both the secondary and main flows.

CONCLUSIONS

Convective heat transfer and flow characteristics of fully developed laminar flow in a conical coil heat exchanger in counter flow configuration are studied experimentally and numerically. Dean number and cone angle have strong influence on the heat transfer. The following conclusions are derived out of the studies (i) the inner Nusselt number is influenced by the inner and outer Dean Number (ii) ratios of mass flow rates of hot and cold fluids have significant effect on the heat transfer.

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