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Experimental Study of Double Pipe Helical Coil Heat Exchangers in the Laminar to Transitional Flow Regime

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Abstract— This study experimentally investigates various heat transfer characteristics, pressure drop, friction factor, Nusselt number, and Dean Number in a double-pipe helical coil heat exchanger. The study examines the hot water mass flow rates ranging from 0.02 to 0.12 kg/s and the cold-water mass flow rates ranging from 0.055 to 0.15 kg/s. The cold-water inlet temperature was maintained at 29°C, while the hot water inlet temperature varied between 65-75°C. Based on the hydraulic diameter of the annular space between the two tubes, the Reynolds number ranged from 1000-6000. All required parameters, such as the inlet and outlet temperatures of the inner coil and annulus fluids and the flow rate of fluids, were measured using appropriate instruments. Seventy-two test runs were conducted, from which the heat transfer coefficients of the inner coil side and annulus side were calculated. The calculated heat transfer coefficients of the inner coil were compared with available correlations from the literature, and a reasonable agreement was found.

Index Terms— Double pipe, helical coil, Heat transfer

I. Introduction

The demand for efficient utilization and recovery of heat has led to the development of different heat transfer enhancement techniques for heat exchangers [1-2]. These techniques aim to improve the heat exchanger's performance and reduce its size, making it increasingly important in energy conservation and recovery. In recent years, the focus on heat exchangers has grown due to environmental concerns, such as thermal, air, and water pollution, as well as the need for heat recovery. Heat exchangers are a critical piece of equipment in the chemical process industry, and heat transfer between flowing fluids is an essential physical process [3].

The demand for miniaturization and improved performance has led to the evolution of the helical coil heat exchanger [4], which uses coiled tubes of a helical shape as a passive enhancement technique. Fluid dynamics in these helical tubes are complex and have been an enduring topic of research. The curvature of the coil induces centrifugal forces on the moving fluid, forming a secondary flow and improved heat transfer coefficients. The fluid streams within the outer face of the pipe move faster than those within the inner face due to the curvature effect. This velocity difference sets up secondary flows whose pattern varies with the Dean number of the flow. Helically coiled tubes are used in various applications, including compact heat exchangers, condensers, and evaporators in the food, pharmaceutical, modern energy conversion and power utility systems, heating, ventilating, and air conditioning (HVAC) engineering, and chemical industries [5]. They have a wide range of uses, including installations, transportation, food processing, process industries, domestic applications, nuclear power plants, air conditioning, heat recovery, HVACs, refrigeration, and more [6].

Nomenclature

A	surface area (m2)
c	thermal capacity (J/kg K)
h	heat transfer coefficient (W/m2 K)
k	thermal conductivity (W/m K)
m	mass flow (kg/s)
Q	heat flow (W)
P	Pressure
De	Dean Number
Nu	Nusselt Number
Re	Reynolds number
δ	curvature ratio
θ	Angle
di	Inner Tube inside diameter
do	Inner Tube outside diameter
Di	Outer Tube inside diameter
Do	Outer Tube outside diameter
P	Pitch
L	Length of tubes
PCD	Pitch Circle Diameter
α	helix angle

In a study by Patankar et al. [7], a two-equation turbulence model was used in a numerical solution procedure to predict flow characteristics in a curved pipe. Both developing and fully developed flow regimes were examined and the predictions showed reasonable agreement with experimental data and computation results. Prabhanjan et al. [8] compared coiled tubes and straight tubes, and the results indicated that helical coil heat exchangers increase the heat transfer coefficient, and the temperature rise of fluid depends on the coil geometry and flow rate. Rennie [9] examined double-pipe helical heat exchangers numerically and

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experimentally without considering the effect of coiled tube pitch. The boundary condition used in this work differed from the conventional boundary conditions of constant wall temperature and constant heat flux. Shokouhmand et al. [10] experimentally investigated helical coil heat exchangers using the Wilson plot and presented correlations for inner and outer Nusselt numbers, which are as follows:

$$Nu_i = 0.112 De^{0.51} \gamma^{-0.37} Pr^{0.72} \quad (1)$$

$$Nu_o = 5.48 Re_o^{0.511} \gamma^{0.546} Pr^{0.226} \quad (2)$$

In a study by S.S. Pawar et al. (11), experimental investigations were conducted on heat transfer to Newtonian and non-Newtonian fluids in helical coils with laminar and turbulent flow. The study found that the overall heat transfer coefficient was higher for smaller helix diameters due to the significant effect of centrifugal force on the coil's secondary flow. The results also showed that water had a higher overall heat transfer coefficient and Nusselt numbers than glycerol-water mixture and non-Newtonian fluids like dilute aqueous polymer solutions of sodium carboxymethyl cellulose (SCMC) and sodium alginate (SA). Furthermore, both fluids' overall heat transfer coefficient and Nusselt numbers decreased as the helix diameter increased for the same flow rates.

In another study by José Fernández-Seara et al. (12), the performance of a vertical helical coil heat exchanger was described. The results indicated that the Nusselt number, calculated with the outer tube diameter as the characteristic length, improved by increasing the outer tube diameter. The heat transfer was independent of the other geometric parameters for a given inner heat exchanger area value.

Previous studies on helically coiled heat exchangers have focused on constant wall temperature or heat flux boundary conditions. They have mostly investigated single-coil helical and shell and tube heat exchangers. However, more investigational information is needed in the present literature on heat transfer coefficients, Nusselt number, and friction factor in the case of double-pipe helical heat exchangers, especially in the laminar-to-transition regime. These parameters are essential in the design of double-pipe helical coil heat exchangers.

This present study aims to investigate a double-pipe helical coil heat exchanger and determine the heat transfer characteristics by changing fluid flow rates in both the inner and outer tubes for a counter-flow arrangement in the laminar-to-transition regime.

II. EXPERIMENTAL CONFIGURATION

A. Geometry of double pipe helical coil:

Figure 1 illustrates a double-pipe helical coil commonly used in heat exchangers. The coil's diameter is the Pitch Circle Diameter (PCD), while the distance between two adjacent turns is called the pitch (P). The Curvature Ratio (δ) is the outer pipe diameter divided by the coil diameter (Do/PCD). The helix angle is the angle between the projection of one coil turn and a plane perpendicular to the axis. The heat exchanger coils were built from TYPE M copper tubing, and standard copper connections were used. A specialized bending machine wound the tubing using a unique wooden matrix to achieve the desired geometry with minimal cross-section distortion. Small holes were drilled into the outer coil and tapped with set screws to center the inner tube before soldering the end connections to ensure the inner coil was concentric with the outer coil. After soldering, the set screws were removed, and the holes closed to avoid disrupting the fluid flow in the annulus. The heat exchanger consisted of four turns, requiring a certain length of tubing [6].

$$L = N * (P^2 + (\pi * PCD)^2)^{0.5} \quad (3)$$

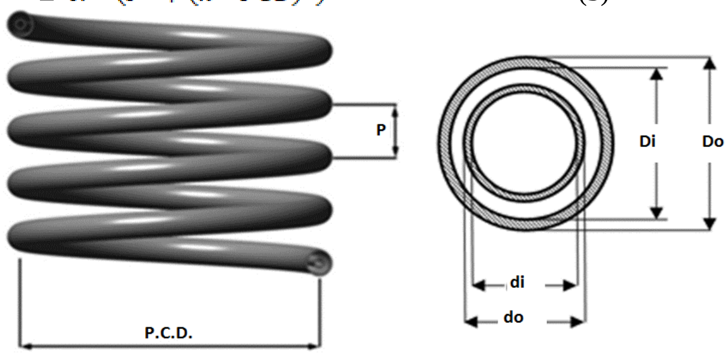

Fig. 1. Schematic layout of the test section.

TABLE I: CHARACTERISTIC DIMENSIONS OF DOUBLE PIPE HELICAL COIL HEAT EXCHANGERS

Coil No.	d_i (mm)	d_o(mm)	D_i(mm)	D_o (mm)	Pitch (mm)	PCD (mm)	Length (m)
Coil 1	11.43	12.7	26.69	28.57	45	240	3.021
Coil 2	14.45	15.87	32.80	34.925	67.5	360	4.531

B. Experimental apparatus

Figure 2 provides a schematic illustration, while Figure 3 displays a photographic representation of the experimental setup. The inner coil had straight entry and exit hydrodynamic lengths, and the orientation of the double-pipe helical coil heat exchanger was vertical. The test section's outer coil was insulated with glass wool and covered with a thin rope. The heat supply for heating water was produced by burning LPG from a cylinder and controlled by regulating the burner inlet pressure. The flame was intended to irradiate the coil's surface uniformly, and heating caused a linear enhancement in the bulk temperature across the heated length, verified using a Resistance Temperature Detector (RTD) attached to the pipe's outer wall [13].
The cold fluid entered the coil through the left side connection, and the hot fluid entered through the right side connection and discharged into a reservoir. A ball valve regulated the flow to the desired conditions, and the flow rate was measured using a calibrated rotameter at the inlet. Temperatures were measured using four K-type RTDs (PT 100) set at uniformly distanced locations to measure the coil surface and fluid temperature. Four additional RTDs were positioned at the heat exchanger's inlets and outlets to measure the hot and cold fluid temperatures. The ACCEL model, an 8/16 Channel Universal data logger with 16 analog input channels, was used to record all temperature

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measurements as a data acquisition device. All experiments were conducted under steady-state conditions. The data acquisition system scanned and stored data every 5 seconds, and the measured values were averaged over a period of 4 min [14]. The pressure of the cold and hot fluids were measured using a digital pressure gauge, and the values were available on the data logger. All measurements were calibrated before the tests. The pressure drop at the fluid's coil inlet and outlet and the temperature at each location in the outer pipe wall were recorded six times and averaged over time. In addition to the Rotameter, the mass flow rates were anticipated by recording the flow into a graduated vessel (750 ml) from the heat exchanger outlet as an additional measurement source.

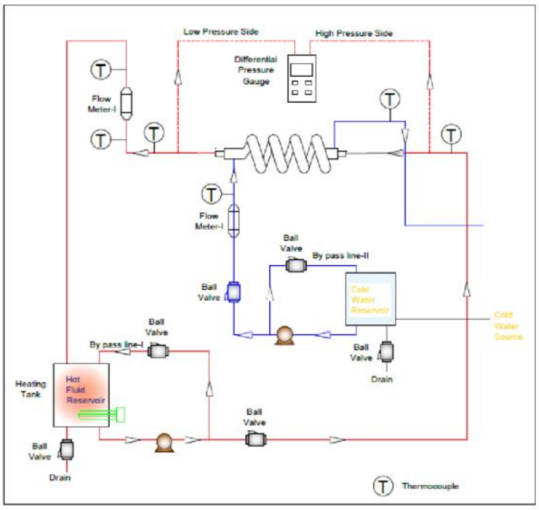

Fig. 2. Schematic diagram of the experimental apparatus

Fig. 3. Photo of the experimental apparatus

III. EXPERIMENTAL PROCEDURES AND DATA REDUCTION

A study was conducted to experimentally analyze the performance of a double-pipe helical heat exchanger. Two heat exchanger coils with different dimensions and counterflow configurations were tested. The cold water was supplied to the heat exchanger's annulus from a reservoir through a small pump. The reservoir was connected to the local water supply, and the cold water temperature was maintained at 29°C. The flow rate was adjusted using flow meters and ball valves to obtain the required flow rates in the primary lines while the remaining water was bypassed to the reservoir. The range of operating conditions is presented in Table II. The thermodynamic properties of water were obtained using Ansys fluent 14.5. The heat transfer coefficient, Nusselt number, and heat transfer rates were calculated based on the measured temperatures, flow rates, and pressure during the experiments.

TABLE II: RANGE OF OPERATING PARAMETERS

Parameters	Range
Inner tube water flow rate	0.020–0.12 kg/s
Annulus water flow rate	0.055–0.1500 kg/s
Inner tube inlet temperature	65–75 °C
Annulus inlet temperature	29 °C

A. *Uncertainty Analysis*

The precision of the measuring equipment and techniques used influences the accuracy of the experimental findings. Uncertainties can arise from various sources, such as the measuring instrument, operator, and environmental conditions. The method proposed by Kline and McClintock [15] is utilized to estimate the uncertainties. The apparent uncertainties related to the measurement of different parameters are considered. To verify the repeatability of the experiments, several runs for each test fluid are performed, producing consistent outcomes. The uncertainty is estimated by calculating the root mean square combination of the impacts of each input. The uncertainties in determining a result (WR+) due to independent variables are computed using the following equation:

$$W_R^+=\left(\left(\frac{\partial R^+}{\partial x_1}W_1\right)^2+\left(\frac{\partial R^+}{\partial x_2}W_2\right)^2+\cdots+\left(\frac{\partial R^+}{\partial x_n}W_n\right)^2\right)^{1/2} \quad (4)$$

TABLE III (A): UNCERTAINTY MEASUREMENT OF DIFFERENT FACTORS

Parameter	Unit	Comment
Hot fluid inlet temperature	^0C	± 0.5
Hot fluid outlet temperature	^0C	± 0.5
Cold fluid inlet temperature	^0C	± 0.5
Cold fluid inlet temperature	^0C	± 0.5
Uncertainty in the measurement of volume flow rate Water	Kg S^{-1}	± 0.3
Uncertainty in the measurement of pressure drop	KPa	0.136

TABLE III (B): UNCERTAINTIES IN CALCULATING A RESULT

Uncertainties in calculating a result	%	
Nusselt number	%	± 3
Reynolds number	%	± 3.2
Friction factor	%	± 2 to 3
Uncertainty in reading values of table	%	± 0.1 *to* 0.2

The analysis indicated that the uncertainties associated with the measurements ranged from 2 to 5% for different parameters. As a result, the correlations obtained from the experimental data would not be significantly affected. Deviations could be attributed to factors such as differences in tube roughness, errors in measurement, limitations of the

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equations employed, entrance effects, and variations in Reynolds numbers.

IV. RESULTS AND DISCUSSION

A. Calculation of heat transfer coefficients:

This study investigates the heat transfer characteristics and performs thermodynamic analyses of a double pipe helical coiled heat exchanger. To achieve this, previously published equations in the literature were taken into account [17]. The average heat transfer rate (Qave) in a double-pipe helical coil heat exchanger can be determined using the following equation:

$$Q_h = m_h C_{ph}\left(T_{h,in} - T_{h,out}\right) \quad (5)$$
$$Q_c = m_c C_{pc}\left(T_{c,out} - T_{c,in}\right) \quad (6)$$
$$Q_{ave} = \frac{1}{2}\left(Q_h + Q_c\right) \quad (7)$$

The experimental data provided all the necessary quantities except the heat transfer rate. The calculated heat transfer rate was then used to determine the overall heat transfer coefficient, Uexp, using the following equation:
Uexp = Qave / (A × ΔTlm)
where Qave is the average heat transfer rate, A is the heat transfer area, and ΔTlm is the logarithmic mean temperature difference.

$$U_{exp} = \frac{Q_{ave}}{A\Delta T_m} \quad (8)$$

$$\Delta T_{mLMDT} = \left(\frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{ln\left(\frac{T_{h,in} - T_{c,out}}{T_{h,out} - T_{c,in}}\right)}\right) \quad (9)$$

$$\frac{1}{U_{exp}} = \frac{1}{h_o} + \frac{A_o \, ln\left(\frac{d_o}{d_i}\right)}{h_o} + \frac{A_o}{A_i h_i} \quad (10)$$

Inner and outer tube heat transfer coefficient h_i and h_O were calculated by using a following relation as

$$h_i = \frac{Q_h}{A_i \, \Delta T_i} \qquad h_O = \frac{Q_c}{A_o \Delta T_o} \quad (11)$$

Where the subscripts i and o pertain to the inside and outside of the inner tube.

Figure 4 illustrates the overall heat transfer rate as a function of varying mass flows for the counter flow arrangement in coils 1 and 2 for flow in the annular region. The flow rate in the inner tube was kept constant while the annulus flow rate was varied. The trends observed are consistent with those expected for a fluid-to-fluid heat exchanger, with the overall heat transfer rate increasing as the annulus flow rate increases. For a fixed flow rate in the inner coil, an increase in the annulus flow rate eventually leads to an asymptotic overall heat transfer rate. It is apparent that this asymptotic value is reached at a lower Dean number for the larger coil than for the smaller coil.

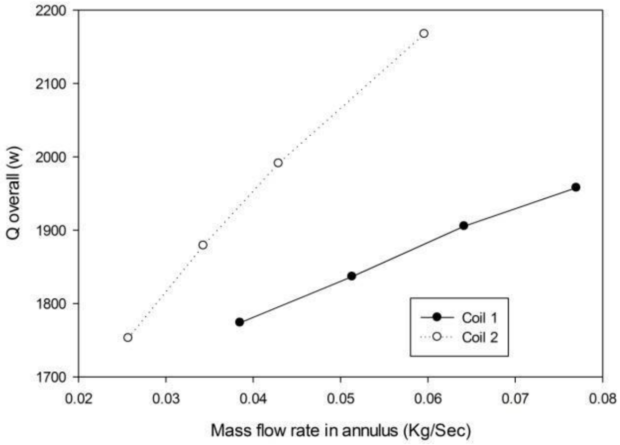

Fig. 4. Overall heat transfer rate versus Mass flow rate in annulus

B. Experimental Nusselt number

The validity of the experimental apparatus and procedures was first confirmed by comparing the obtained results of the Nusselt number (Nu) using water as the working fluid in the helical coil heat exchanger with the results reported in the literature [17, 18]. As shown in Fig. 5, the experimental results for the heat transfer rate are in good agreement with those of previous studies. The investigation covered laminar and turbulent flow regimes in both the inner coil and annulus, with Reynolds numbers ranging between 850 and 6000.

The Nusselt number was then calculated using the following formula:

$$Nu_i = \frac{h_i d_i}{k} \quad (12)$$

$$Nu_o = \frac{h_o d_h}{k} \quad (13)$$

Where d_h is the hydraulic diameter of the outer coil.
Reynolds number of the annular space between the two tubes can be calculated with:

$$Nu_o = \frac{h_o d_h}{k} \quad (14)$$

The variation of the Nusselt number with the Reynolds number, calculated based on the tube diameter, is shown in Figure 5. The figure indicates that Nusselt numbers increase with increasing Reynolds numbers and furthermore, increasing the dimensions and pitch of the coiled tube results in higher Nusselt numbers. It is evident that the differences among the various coil geometries are more prominent at high Reynolds numbers, as an increase in Nusselt number is observed when the annulus-side Reynolds number increases. The graph's trend clearly shows that the Nusselt number is a function of the Reynolds number, with a higher Reynolds number leading to a greater Nusselt number and improved heat transfer coefficient for any fluid.

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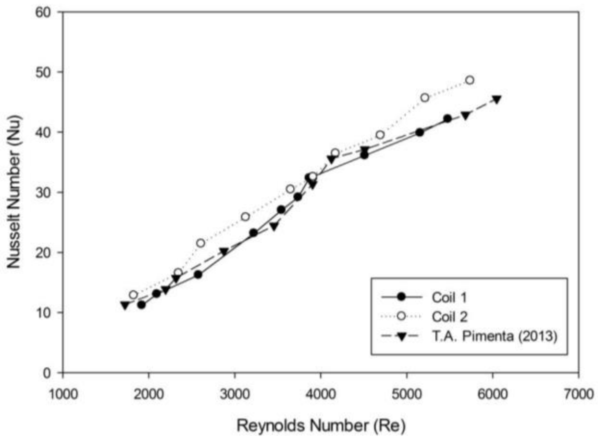

Fig. 5. Variation of Nusselt number with Reynolds number

C. *Frictional characteristics*

Figures 6 and 7 show the relationship between the friction factor and Reynolds number for coil 1 and coil 2, respectively. As observed from the figures, the friction factor decreases consistently as the Reynolds number increases. The reduction rate is rapid in the laminar flow region and gradually slows down as the flow approaches turbulent flow. Additionally, the pressure drop in the annular space between the two tubes and the pressure drop inside the inner tube were both measured experimentally using a digital pressure gauge at various Reynolds numbers. The friction factor (f) can be determined using the following formula [20]:

$$f = \frac{\Delta P}{0.5\left(\frac{1}{D_h}\right)\rho V^2} \tag{15}$$

The physical properties of the fluid are evaluated at the average fluid temperature (Tm). When the Reynolds number is less than 150, the friction factors for the helical coil are quite similar to those in a straight tube [20]. This indicates that the flow pattern is mainly influenced by the viscous forces, and the effect of the centrifugal force is not significant. However, when the Reynolds number exceeds 150, the friction factor values for the helical coil begin to deviate towards higher values, and this deviation becomes more prominent for Reynolds numbers greater than 500. This can be attributed to the growing influence of the centrifugal force on the flow pattern.

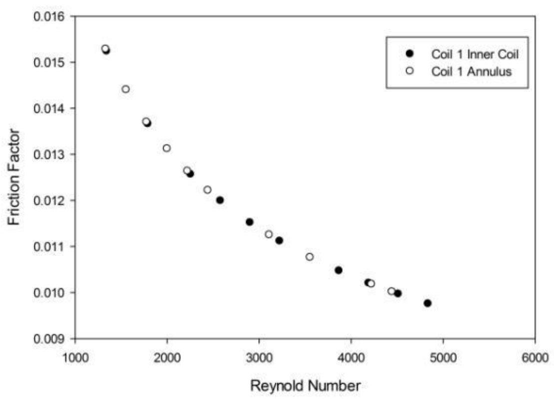

Fig. 6. Variation of friction factor with Reynolds number

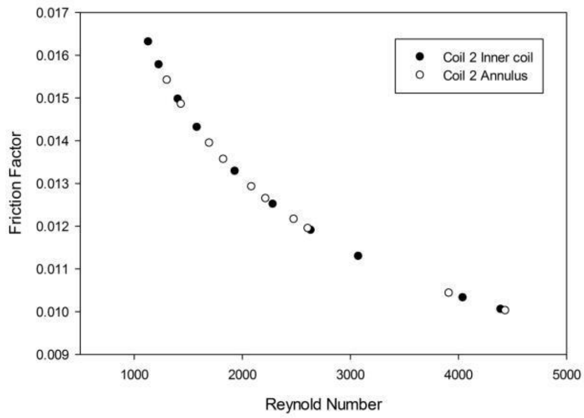

Fig. 7. Variation of friction factor with Reynolds number

D. *Dean Number*

Dean number is a parameter that characterizes the flow in a helical pipe. The relationship between the Nusselt number inside the tube and the Dean number for two different coil pitches is shown in Fig. 8. The figure indicates that increasing the coil pitch results in a higher inner Nusselt number. This can be explained by the fact that higher coil pitches correspond to an increase in secondary flow up to a certain extent. Based on the comparison of experimental data with available correlations, it can be concluded that when coil pitches are significant, and the Dean numbers are low (De < 3000), the correlation of a constant temperature boundary condition predicts the present data quite accurately [10].

$$Nu = 0.152 De^{0.431} Pr^{1.06} \gamma^{-0.227} \tag{16}$$

Dean Number for flow in inner coil was calculated from equation 17.

$$De = Re\left(\frac{r}{Rc}\right)^{0.5} \tag{17}$$

The modified dean number for the annulus is calculated as if it would be for a normal Dean number, except that the curvature ratio used is based on the ratio of the radius of the outer tube to the radius of curvature of the outer tube, and the Reynolds number based on the hydraulic radius of the annulus

$$De = \frac{\rho \vartheta_o}{\mu}\left(\frac{Do^2 - Di^2}{Do^2 + Di^2}\right)\left(\frac{Do - Di}{R}\right)^{\frac{1}{2}} \tag{18}$$

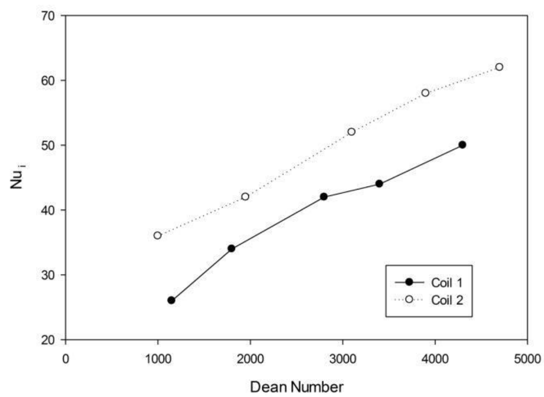

Fig. 8. Variation in Nusselt Number with Dean Number

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V. Conclusion

The present study uses experimental methods to investigate the heat transfer characteristics of double-pipe helical coil heat exchangers with water as a working fluid. The investigation covers a range of Reynolds numbers from 1000 to 6000, with two different-sized coils examined to improve the inside heat transfer rate. Various inflow rates and temperatures have been studied to assess the impact of flow parameters on the overall heat transfer coefficient of the heat exchanger. The primary objective is to experimentally evaluate the performance of the double-pipe helical coil heat exchanger in a counter-flow arrangement, considering the overall heat transfer rate, convective heat transfer coefficient, mass flow rate, and Reynolds number. The experimental results are compared with available data from previous studies, and the following findings were obtained:

- The convective heat transfer coefficient increases with the increase in mass flow rate in the annulus; flow conditions in the annulus have a significant effect on the overall heat transfer coefficient.
- Overall heat transfer coefficient increases with an increase in Reynolds number, indicating a shift to turbulent flow regimes.
- An increase in the Reynolds number leads to an increase in the Nusselt Number.
- The friction factor decreases faster in the laminar flow region and decreases slowly as the flow approaches turbulent flow in the inner tube.
- Nusselt Number increases with an increase in Dean Number. It is also observed that annulus Nusselt numbers have a strong linear relationship with Dean Numbers for the range of flow rates studied in this research.

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