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Design and Optimizing Thermal and Hydraulic Performance of a Helical Coil in Tube Heat Exchanger: Numerical Analysis and Experimental Validation

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Abstract

To enhance thermal performance of a heat exchanger in compact volume, the research aims for optimization of helical coil in tube heat exchanger. The helical coil increases the heat transfer area and turbulent flow of fluids enhances heat transfer rate. The thermal and hydraulic performance can be improvised by altering the mean diameter, pitch of the helical tube, mass flow rate of cooling fluid without changing the length of the heat exchanger. Mean diameter, however, adversely affects the overall heat transfer coefficient but increases the hydraulic performance. Mass flow rate of cooling fluid, on the other hand slightly increases thermal efficiency. To examine the optimum structure of heat exchangers with constant volume, the current numerical studies are conducted for different mean diameter, pitch length, mass flow rate. Experiment is carried out for 35 mm pitched coil that can withstand 600 psi of fluid pressure, and it can cool down 335K hot oil to 330K. Maximum overall heat transfer co-efficient of the heat exchanger 305.651 W/m^{2}k which is obtained for pitch of 45 mm and mean diameter of coil 95 mm and 0.08 kg/sec of oil flow rate.

Keywords: Helical coil, Heat transfer efficiency, Numerical Analysis, Ansys simulation.

1. Introduction

The helical tube in shell heat exchangers are being employed in the HVAC systems, piping systems, chemical reactors for its compact design and comparatively ease in manufacturing process [1], [2]. Because the heat exchangers are small, they require less volume and weight and have a better heat transfer rate. The shape of the coil leads to generating a centrifugal force that generates rotational flow. Consequently, heat transfer process increases, and pressure drop decreases in contrast to straight pipes [3], [4]. Impact of pitch and helix number of a helically coiled heat exchanger was assessed through experiments[5]. Experiments also shows that, increasing pitch size increases heat transfer coefficient of shell [6]. He established two correlations that forecast heat transfer coefficient on external and internal surface. The following are the correlations,

$$Nu_c = 0.152\, De^{0.431}\, Pr^{1.06}\gamma^{-0.277} \quad (1)$$

$$Nu_c = 19.64\, Re^{0.513}\, Pr^{0.129}\gamma^{0.938} \quad (2)$$

Another experiment is conducted by M. Moawed [7], where he studied forced convection of helical coil from outer surface in an airflow wins tunnel system with open circuits for constant heat flux. The effect of Prandtl numbers and geometric configurations on the heat transfer characteristics in helical tubes are analyzed by Xin and Ebadian et al. [8] Jayakumar studied thermal performance on coil side of the heat exchanger numerically and experimentally. He observed using working fluids' temperature dependent characteristics to forecast the heat transfer coefficient more accurately. In any heat exchanger where heat transfer takes place between fluids, different boundary conditions, for example, constant heat flux and wall temperature cannot be used to predict heat transfer. Not only Iso-thermal steady state, also non- isothermal unsteady conditions in helically coil are studied for Newtonian, non-Newtonian fluid is studied by Pawar et al. [9].

Then effects of helical coils' curvature, Prandtl number, Reynold's number on Nusselt number and friction factor are studied by experiments conducted by Hardick et al. [10]. Pimenta observed experimentally [11] the heat transfer coefficient of Newtonian and non-Newtonian fluids for constant wall temperature conditions where flow condition is fully developed laminar flow. Wilsons' plot experiment apparatus is used to determine the heat transfer coefficient of shell and coil side of the heat exchanger [12]. Moreover, the relationship between geometric characteristics and fluid flow are investigated on heat transfer rate. The heat transfer rate of the heat exchangers can be increased by incrementing the mean diameter of helical coil, coil pitch and mass flow rate in shell and tube. The correlations suggested coil side Nusselt number, demonstrate how the effects of fluid mass flow rate, mean

diameter and shell diameter vary considerably between equations. All the researchers showed only different empirical formulas and predicted thermal and hydraulic behavior of this type of heat exchangers, but the present work attempts to elaborate the thermo-fluidic behavior of the compact heat exchanger made in university.

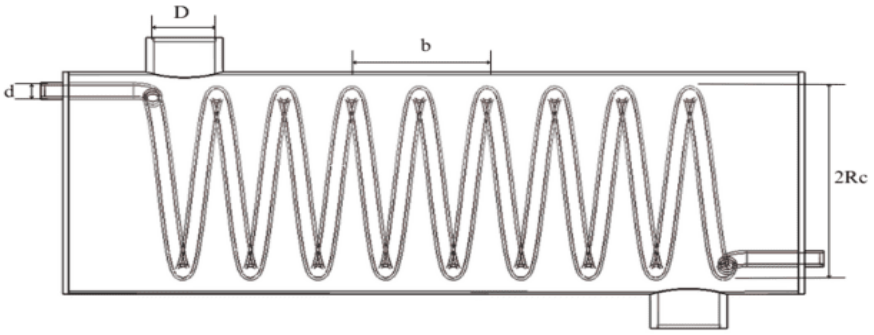


Fig. 1. A typical helical coil in a tube heat exchanger

2. Numerical Modeling

2.1 Description of computational domain

Fig .1 shows the diagram of heat exchanger. The helical coil has a diameter of (d) 6.58mm (3/16 inch.) and the pitch of the helical coil (b) is 35mm which varies from 25mm to 45mm in the numerical model. The mean diameter of the helical coil (2Rc) is 104 mm which is not constant as well. The value varies from 95 mm to 125 mm. The shell diameter (D) is 127mm. For different cases, the flow rate of fluid is varied and so is the geometric parameter of the heat exchanger. Detail geometrical dimensions are listed in Table. 1. Water is chosen as cooling fluid that flows through shell side and engine oil is used as hot fluid that flows through the helical coil.

Table 1. Specific geometrical dimensions

Sl. no	Pitch (mm)	Mean Dia (mm)	Length (mm)
1	25	95	380
2	30	100	380
3	35	105	380
4	40	110	380
5	45	120	380

2.2 Governing equations

A model must be developed to investigate heat exchangers' hydraulic and thermal characteristics. Ansys fluent is used to determine different parameters using the finite element method. For this, some assumptions are taken, (a) thermos-physical properties are constant, (b) radiation is neglected, (c) incompressible flow, (d) steady-state condition. At the same time, some correlations are used to determine different parameters using python and hand calculations.

Dean number, $De = Re(\frac{d}{2Rc})^{0.5}$ (3)

Xin and Ebadian [8] suggested the Nusselt number of coil side as,

$$Nu_c = (2.153 + 0.318De^{0.643})Pr^{0.177} \quad (4)$$

Conditions to apply the correlation:

$20 \leq De \leq 2000$

$0.0267 \leq d/2Rc \leq 0.0884$

The heat exchanger satisfies both conditions. So, equation. 3 is applicable for determining Nusselt number for coil side. For Reynolds' number of coil and shell side, hydraulic diameter of shell side is needed to determine. Dimensionless pitch is the ratio of pitch and mean diameter of coil.

Hydraulic diameter of shell, $D_h = \frac{D^2 - 2\pi Rc^2 {d_0}^2 \gamma^{-1}}{D + 2\pi R_c d_0 \gamma^{-1}}$ (5)

Reynolds' number of shell, $Re = \frac{v\rho D_h}{\mu}$ (6)

Velocity of flowing fluid can be determined by, $v = \frac{\dot{m}}{\pi {R_c}^2}$ (7)

Eqn. 2 is used for determining the Nusselt number for coil side. Nusselt number will be used to calculate the heat transfer coefficient (h) of coil and shell side of the heat exchanger using the equation. 8

$$Nu = \frac{h*Characteristic\ Diameter}{k} \quad (9)$$

Where, characteristic diameter is D_h and d for shell and coil side of heat exchanger respectively.

From eqn. 9, heat transfer coefficient (h) can be determined for both shell and coil side. Then the coefficients will be used to calculate the overall heat transfer coefficient (U_o). This parameter is the most important for heat exchanger. To determine U_o, eqn. 10 is used. Here, fouling factors will be used in both hot and cold side.

$$\frac{1}{U_o} = \frac{1}{h_{shell}} + \frac{1}{h_{coil}} + f.f_{shell} + f.f_{coil} \quad (10)$$

For a heat exchanger, pumping power is a very important parameter that is affected by the friction factor of heat exchanger. For this situation, friction factor of coil side is more important than shell side. Friction factor can be used by eqn. 11 studied by Churchill et al. [13]

$$f_c = [\left(1 + \frac{0.18}{\left(\left(1+\frac{35}{De}\right)^2\right)^{0.5}}\right)^2 + \left(1 + \frac{d}{3R_c}\right)^2 (\frac{De}{88.33})]^{0.5} \quad (11)$$

2.3 Boundary and operating conditions

Table 2. Boundary and operating conditions

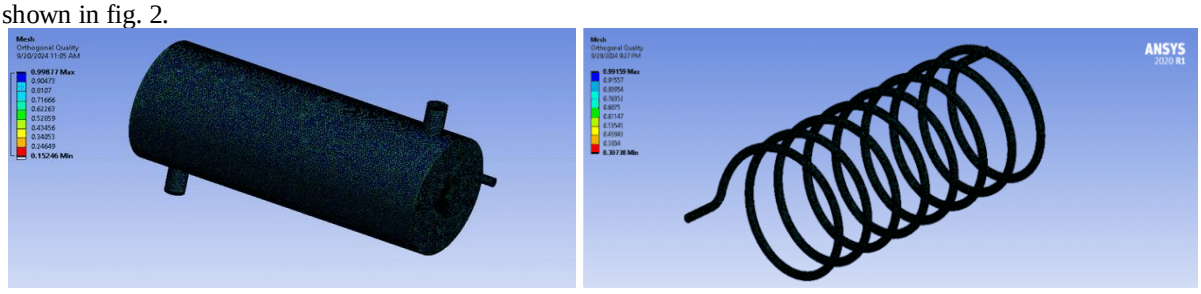
Water inlet (K)	Oil inlet (K)	Water mass flow (Kg/s)	Oil mass flow (kg/s)
293	335	0.10	0.02
293	335	0.15	0.04
293	335	0.20	0.06
293	335	0.25	0.08
293	335	0.30	0.10

2.4 Numerical methods and experiments

The model is made and colling water through outer shell and hot oil is flown through helical tube from oil reservoir to conduct the experiment and collect outlet temperature, pressure drop. Then a mathematical model is built on PyCharm IDE for further calculation to generate graphs. Then, the Finite elements based CFD method is used for supporting previous simulations. Ansys Fluent 2020 r1 is used for the operation. For the simulation, Simple algorithm with energy method and viscous model is used. As converging criteria, residual was set as 10^{-6} and momentum and energy were used as 0.3 and 1.0 respectively.

2.5 Meshing of heat exchanger

Meshing was done using Ansys and for accurate results some steps were taken, for example refinement, inflation layers and finer meshing. As the geometry is complicated, there are some drops in quality while meshing the shell side of the heat exchanger. Orthogonal quality is quite acceptable for the mesh of overall heat exchanger that is shown in fig. 2.



(a) (b)

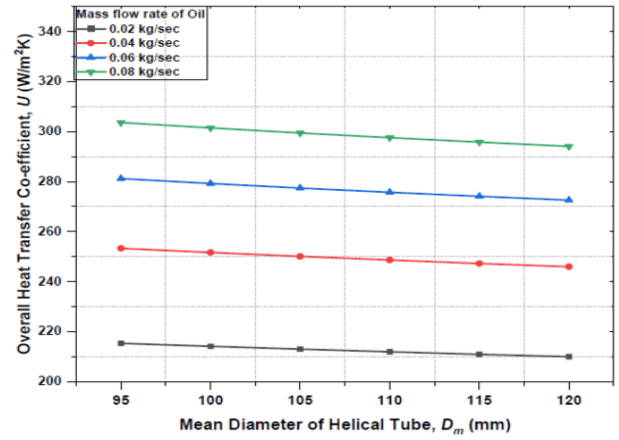
Fig. 2. Orthogonal quality of mesh of (a) shell side, (b)helical coil of the heat exchanger

3. Result and discussions

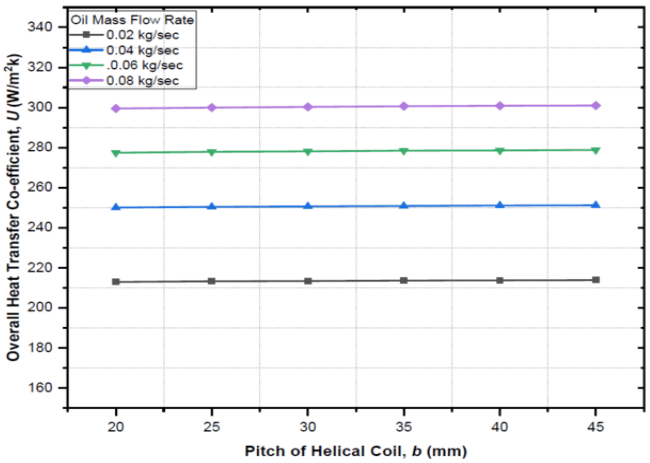
3.1 Heat transfer characteristics

Fig. 3(a) depicts the variation of U_o with mean diameter of helical coil. When diameter increases the overall heat transfer coefficient drops. And U_o is maximum when flow rate of oil through the coil is maximum for a given

diameter. As higher flow rate ensures more turbulence, heat transfer rate increases, so the overall heat transfer coefficient. Unlike mean diameter, with increasing pitch of helical tube, there is very slight increasement can be shown in overall heat transfer coefficient, shown in fig. 3(b). For example, for a constant mean diameter (95mm) and same mass flow rate of water and hot oil, when pitch is 40 mm, U is 305.488 W/m^{2}k whereas, when pitch is 35 mm, U is 305.209 W/m^{2}k (0.09% increment). Similar results can be seen for different water flow rates. A very slight increment in overall heat transfer coefficient can be seen when the water flow rate is increased. Around 0.3% of maximum increment can be observed for every situation studied.



(a)

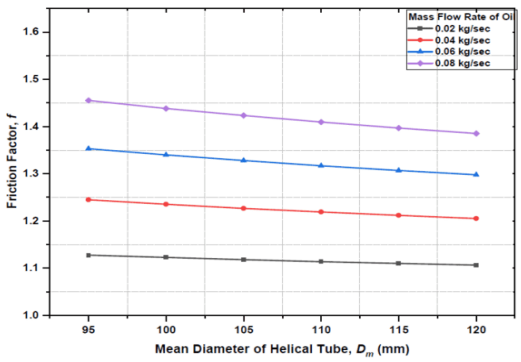


(b)

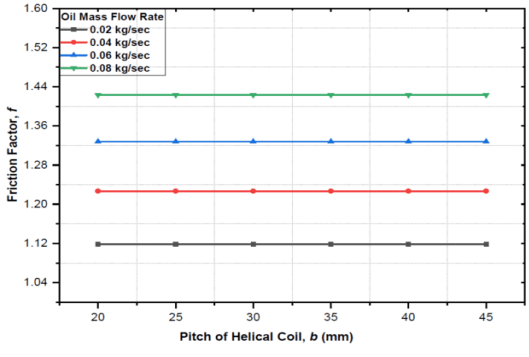
Fig. 3. Variation of overall heat transfer coefficient (U_o) with (a) mean diameter, (b) pitch of helical coil

3.2 Friction factor characteristics

Friction factor is significant for operating a heat exchanger because it directly affects the pumping power for it. For this heat exchanger, pressure inside helical tube is more important than pressure inside the shell side. So, the friction factor for helical tube is observer. Fig. 4(a) shows the relation of friction factors with mean diameter of coil. With increasing mean diameter, turbulence drops slightly and thus, friction factors drop. And the friction factor is not affected by pitch of helical tube that can be seen in fig. 4(b).



(a)

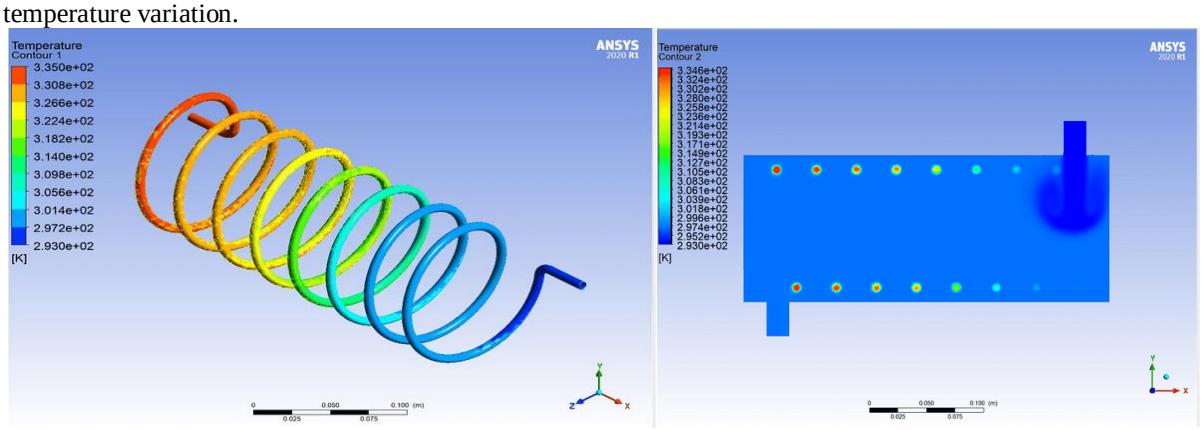


(b)

Fig. 4. Variation of friction factor (f) with (a) mean diameter, (b) pitch of helical coil

3.3 Temperature characteristics

Temperature contour of shell and coil side can be seen in fig. 5(a). For hot oil flow of 0.02 kg/sec and water mass flow rate of 0.1 kg/sec, a simulation is performed and temperature of variation of oil side is observed. And a plane is taken at XY plane and middle of the heat exchanger. At the plane, temperature variation can be seen in fig. 5(b). Temperature distribution can be triggered by (i) water flow rate through shell side, (ii) turbulence in shell side. As turbulence cannot be increased in coil side where hot oil is flowing, shell side flow disruption has impact on temperature variation.



(a) (b)

Fig. 5. Temperature distribution at (a) helical coil side, (b) transverse section of heat exchanger for Oil flow rate of 0.02 kg/sec and water flow rate of 0.1 kg/sec

3.4 Overall performance of the heat exchanger

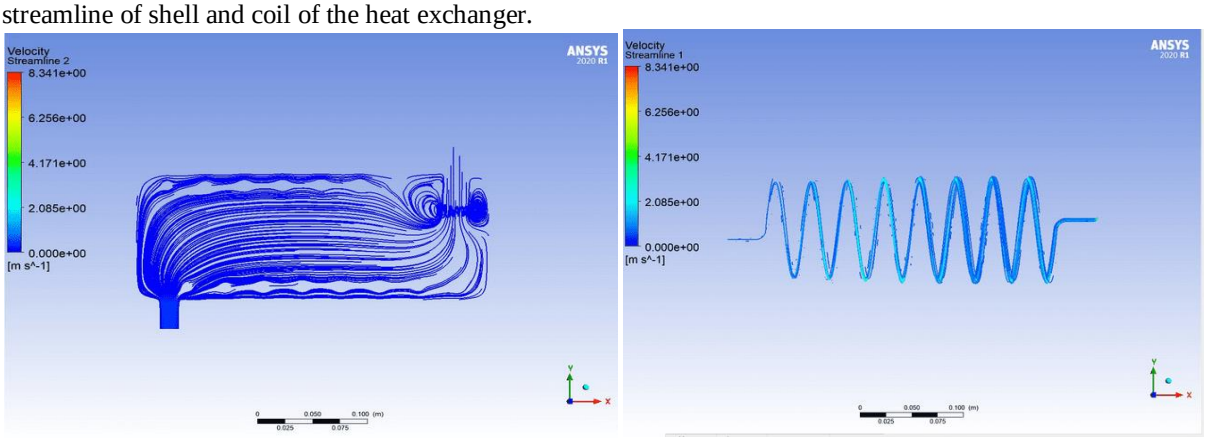
Most of the calculation is done using python modeling. And similar performance can be seen in Ansys simulation. For different oil flow rates and constant mass flow rate water of 0.1 kg/sec, a simulation is done, and results can be seen in table. 3.

Table 3. Results from simulation

Oil mass flow rate (Kg/sec)	Interface temperature (K)	Water outlet temp. (K)	Oil outlet temp (K)
0.02	306.1	297.24	310.36
0.05	307.8	297.18	315.68
0.06	308.6	297.22	318.46
0.08	309.0	297.22	320.22

3.5 Streamline of flow inside heat exchangers

There is significant turbulence that can be observed in inlet of shell side and density of streamline is significantly increased in outlet of shell side. As a result of turbulence, hot oil temperature falls in outlet. Figure. 6 shows streamline of shell and coil of the heat exchanger.



(a) (b)

Fig. 6. Streamline of velocity distribution at (a) shell side, (b) coil side.

3.6 Optimization of the system

For the best output from the heat exchanger, the overall heat transfer coefficient (U_o) should be maximized, and friction factor should be as low as possible. Overall heat transfer coefficient (U_o) is maximum (305.651 W/m^2k) when pitch of helical coil is 45 mm and mean diameter is 95 mm, oil mass flow rate is 0.08 kg/sec and water flow rate is 0.3 kg/sec.

4. Conclusion

The effect pitch and mean diameter of helical tube is examined numerically on overall hydraulic and thermal performance of the heat exchanger. Different geometrical and flow parameters are examined. The heat exchangers' numerical modeling leads to the following conclusion,

1. Mean diameter ($2R_c$) of the helical coil decreases the overall heat transfer coefficient (U_o) and decreases the friction factor in coil side.
2. Pitch of helical coil has very little impact on overall heat transfer coefficient (U_o) and no impact on friction factors.
3. The overall heat transfer coefficient (U_o) increases relatively little if the mass flow rate of cooling water increases. Approximately 0.3% increment of U_o.
4. Mass flow rate of oil has significant impact on U_o. With the increasing mass flow rate, both overall heat transfer coefficient and friction factor (f) increase significantly.

5. References

[1] W. H. Fleming, J. A. Khan, and C. A. Rhodes, "Effective heat transfer in a metal-hydride-based hydrogen separation process," *International Journal of Hydrogen Energy*, vol. 26, Jul. 2001, doi: 10.1016/S0360-3199(00)00129-4.

[2] P. K. Sahoo, Md. I. A. Ansari, and A. K. Datta, "A computer based iterative solution for accurate estimation of heat transfer coefficients in a helical tube heat exchanger," *Journal of Food Engineering*, vol. 58, no. 3, pp. 211–214, Jul. 2003, doi: 10.1016/S0260-8774(02)00370-9.

[3] "Numerical investigation of heat transfer intensification in shell and helically coiled finned tube heat exchangers and design optimization - ScienceDirect." Accessed: Sep. 28, 2024. [Online]. Available: https://www.sciencedirect.com/science/article/abs/pii/S0255270117304105

[4] "Prediction of heat transfer coefficients of shell and coiled tube heat exchangers using numerical method and experimental validation - ScienceDirect." Accessed: Sep. 28, 2024. [Online]. Available: https://www.sciencedirect.com/science/article/pii/S1290072915301319

[5] E. K. Akpinar, "Evaluation of heat transfer and exergy loss in a concentric double pipe exchanger equipped with helical wires," *Energy Conversion and Management*, vol. 47, no. 18, pp. 3473–3486, Nov. 2006, doi: 10.1016/j.enconman.2005.12.014.

[6] M. R. Salimpour, "Heat transfer coefficients of shell and coiled tube heat exchangers," *Experimental Thermal and Fluid Science*, vol. 33, no. 2, pp. 203–207, Jan. 2009, doi: 10.1016/j.expthermflusci.2008.07.015.

[7] M. Moawed, "Experimental study of forced convection from helical coiled tubes with different parameters," *Energy Conversion and Management*, vol. 52, no. 2, pp. 1150–1156, Feb. 2011, doi: 10.1016/j.enconman.2010.09.009.

[8] R. C. Xin and M. A. Ebadian, "The Effects of Prandtl Numbers on Local and Average Convective Heat Transfer Characteristics in Helical Pipes," *Journal of Heat Transfer*, vol. 119, no. 3, pp. 467–473, Aug. 1997, doi: 10.1115/1.2824120.

[9] S. S. Pawar and V. K. Sunnapwar, "Studies on convective heat transfer through helical coils," *Heat and Mass Transfer*, vol. 49, pp. 1741–1754, Dec. 2013, doi: 10.1007/s00231-013-1210-3.

[10] B. K. Hardik, P. K. Baburajan, and S. V. Prabhu, "Local heat transfer coefficient in helical coils with single phase flow," *International Journal of Heat and Mass Transfer*, vol. 89, pp. 522–538, Oct. 2015, doi: 10.1016/j.ijheatmasstransfer.2015.05.069.

[11] T. A. Pimenta and J. B. L. M. Campos, "Heat transfer coefficients from Newtonian and non-Newtonian fluids flowing in laminar regime in a helical coil," *International Journal of Heat and Mass Transfer*, vol. 58, no. 1–2, pp. 676–690, Mar. 2013, doi: 10.1016/j.ijheatmasstransfer.2012.10.078.

[12] N. Jamshidi, M. Farhadi, D. D. Ganji, and K. Sedighi, "Experimental analysis of heat transfer enhancement in shell and helical tube heat exchangers," *Applied Thermal Engineering*, vol. 51, no. 1–2, pp. 644–652, Mar. 2013, doi: 10.1016/j.applthermaleng.2012.10.008.

[13] R. L. Manlapaz and S. W. Churchill, "FULLY DEVELOPED LAMINAR FLOW IN A HELICALLY COILED TUBE OF FINITE PITCH," *Chemical Engineering Communications*, vol. 7, no. 1–3, pp. 57–78, Jan. 1980, doi: 10.1080/00986448008912549.