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Research Article

Experimental and numerical investigation of heat transfer enhancement in double coil heat exchanger

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ABSTRACT

In the current work, a substantial research and cost-effective strategy has been conducted to enhance the thermal efficiency of shell and coil heat exchangers, and geometrical modification is one technique to improve the exchange of thermal energy between two or more fluids. Therefore, experimental and numerical analysis across a shell and single/double coil heat exchanger at constant temperatures of 36 °C for cold water and 65 °C for hot water are studied. Various coil pitches (baseline pitch, P-2P-P and 2P-P-2P) and mass flow rates (1 L/min for hot water and 2, 4, 6, and 8 L / min for cold water) were studied. The present experimental results for single and double coil heat exchangers were in good agreement with previous research's numerical study, with an error rate of 9% and 5%, respectively. Moreover, the numerical findings revealed that modifying the double coil pitch improves the heat transfer rate by 10% compared to a baseline case. Following the encouraging simulation findings, improving the heat exchanger's performance by utilizing more than one pitch for the same coil is a novel method that has not yet been reported. Therefore, when comparing the modified pitch of a double coil heat exchanger to a conventional coil under the same conditions ($400<Re_{sh}<2000$), the results show a 19% increase in Nusselt number.

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INTRODUCTION

Heat transfer between flowing fluids is one of the most essential physical phenomena that has attracted researchers' interest for a long time. Different types of heat exchangers are utilized in various combinations. Regardless of how these exchangers are constructed, they are all linked by a fundamental concept. Heat exchangers can be used to transfer thermal energy between two distinct fluids at different temperatures. Many applications use the heat exchanger as an important part of its operation. These applications include, for example, the energy production process, the chemical and food industries, electronics and environmental engineering [1,2].

The performance and efficiency of any type of heat exchanger is measured by the amount of heat transferred and pressure drop, and this pressure drop provides insight

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into the capital cost and energy requirements (operating cost) of the heat exchanger. The helical design of coiled tubes is one of the heat exchangers enhancing strategies that are extensively employed. The extensive usage of helical coiled tubes is due to their large heat transfer surfaces and ability to promote effective liquid mixing, which improves heat transfer coefficients [3].

Helically coiled tube heat exchangers have been widely investigated as one of the passive heat transfer improvements due to their high heat transfer coefficient and small size when compared to straight tubes [4,5]. The papers published on heat exchanger performance cannot be simply counted, but they may be categorized according to the specific problem. Several investigations have compared the helical coil against straight tubes to see how twisting affects flow. Balamurugan et al. [6] explain how the performance of the coil is dependent on the formation of a vertical vortex within the coil, and how the dean vortex affects the efficiency of the coil when compared to a straight coil [7]. The results show that the pressure drop grows linearly as the diameter of the helical coil increases, and that the pressure drop also increases linearly as the number of helical coils increases. Yamamoto et al. [8] utilized a spectral method to investigate the effects of bending the straight tube into helical tube with a circular cross-section on the flow and observed that bending had a significant impact on flow. Pablo Coronel et al. [9] compared helical heat exchangers with variable curvature ratio coils of d/D = 0.114 and 0.078 against straight tube heat exchangers with various flow rates and temperatures. The overall heat transfer coefficient (U) in helical tube heat exchangers is substantially higher than in straight tube heat exchangers.

Other studies discuss the use of nanomaterials or twisted tape to increase heat exchanger performance. Mehedi Tusara et al. [10] used the ANSYS program to simulate a 1.92 twist ratio coil in order to enhance the heat transfer in the tube. Compared to plain tube, this measurement enhanced the heat transfer rate with increasing pressure drop, the increase in the Nusselt number was 1.34-2.6 times and the friction factor was 3.5- 8 times for screw-tape tube compared to plane tube. Pourramezan et al. [11] explore the flow structure and heat transfer enhancement of laminar flow in a circular tube equipped with the innovative twisted conical strip inserts using CFD. According to the findings, lower pitches and twist angles result in greater Nusselt numbers, friction factors, and thermal performance. The heat transfer properties of the Al_2O_3 nanofluid flow were numerically studied within the shell and helical tube heat exchangers by Bahrehmand and, Abbassi [12]. The Reynolds number on the coil side ranged between 9000- 36000, the Reynolds number on the shell side ranged between 600-2600, and the Prandtl number 2.2- 8.3. The results showed that as the particle size concentration increased, the coil side, shell side and overall heat transfer coefficients improved.

Another experiment found that increasing coil diameter, coil flow rate, and mass flow rate in shell and tube heat exchangers improve heat transfer rate [13]. A comparison of several correlations reported most studies for the helically coiled heat exchanger was presented by Pramod Purandare et al. [14], as well as the overall influence of these factors on Nu and heat transfer rate. They showed that as tube diameter (d) increases with constant coil diameter (D), the curvature ratio (δ) increases, and this leads to increase the intensity of secondary flow (Dean vortices) and, hence increasing Nu. Therefore, it is desirable to have small coil diameter (D) and large tube diameter (d) in helical coil heat exchanger. Holkar et al. [15] observed that the overall heat transfer coefficient and Nusselt number increased by enhancing the secondary flow through the double helical pipes with increasing dean number. Also, during another study, it was proved that the rate of heat transfer increases with the increase in the number of turns through the double coil [16]. Rahul et al. [17] conducted another investigation in which they calculated the Nusselt number of a helical tube correlation and discovered that coil pitch has a significant influence on the outside heat transfer coefficient. Four coiled tubes with different angles of inclination (9, 15, 30, and 45 degrees) and different entrance speeds were simulated by Conte et al. [18]. The results showed that the smaller angle (9°) has a higher performance in heat transfer due to uniform fluid distribution across the tube.

Salem et al. [19] examined the convective heat transfer characteristics in shell and coil heat exchangers, as well as the friction factor for fully developed flow via helical coiled tube. Five heat exchangers with varying curvature ratios were constructed in a counter flow arrangement. By increasing the coil curvature ratio, the Nu of the two sides of the heat exchangers and the friction factor via the helical coiled tube increased. The impact of the shell side heat transfer coefficient in a shell and triple helical coil heat exchanger was investigated by Genic et al. [20] using three heat exchangers with varying geometric parameters, meaning that the Nusselt number shell side was dependent on the hydraulic diameter utilized. The influence of several geometrical shapes (Helical, Triangle, and Hexagonal) on natural convection heat transfer was also investigated using both experimental and computational methods. In comparison to the triangle and spiral coiled tubes, the hexagonal coiled tube gives higher values of the total heat transfer coefficient [21]. Experimentally, Shokouhmand et al. [22] conducted his study on three different pitches of coils to study the effect of varying degrees of curvature and curvature ratios. The results showed that the shell side heat transfer coefficients of the coils with smaller pitches are less than the ones with larger pitches.

During recent studies, there has been a trend towards a change in the design of the heat exchanger, whether it is in the change in the shell or in the coil. Tuncer et al. [23] presented a new modification that involved integrating a hollow tube into the side of the shell and placing a cold liquid heat exchanger along this tube to control fluid flow over the helically coiled tube. According to the findings, the insertion of a hollow tube into the shell side resulted in enhanced heat transfer. To improve the heat transfer efficiency of the helical coil heat exchanger, Senaa et al. [24] conducted experimental investigation to validate the results of the numerical analysis on a shell and single-coil heat exchanger, and then performed a double coil numerical analysis to demonstrate its capacity to enhance heat transfer. The results showed that, the Nusselt number in the double coil tube was 18.2% higher than in the single coil tube.

According to the previous studies and to the best of the authors knowledge, the majority of studies are focused on improving the heat transfer rate and thermal efficiency of a shell and single helical coil heat exchanger at various operating conditions. Due to the complexity, expense, and time needed in constructing experimental models, few researchers have looked into double coil heat exchangers using both numerical and experimental analysis. As a result, an experimental study is carried out to increase the heat transfer rate in a double coil heat exchanger, with the current research providing a platform for future development, as the suggested coils have not been examined in such a concept. The experimental and numerical studies in this work are divided into three parts. To validate the numerical results based on the Nusselt number shell side and Reynolds number, the first part employs single and double coils heat exchanger (a baseline case) using pure water. Once the numerical results are consistent, then a further computational analysis will be carried out using different coils within the same geometry. This model will be then fabricated once the optimal configuration in terms of heat transfer rate has been numerically established to ensure that the experimental and computational findings are compatible. Finally, a correlation was identified over a wide range of Reynolds numbers between the predicted and numerical Nusselt number of the shell side.

NUMERICAL METHODOLOGY

Problem Description and Boundary Conditions

The main objective of the current study is to increase heat transfer through shell and coil heat exchangers, as heat exchangers play a vital role in many industrial processes. Therefore, the current study examined how modifying the configuration of the double coil heat exchanger produces the greatest outcomes under the same operating conditions. Figure 1 depicts a double coil heat exchanger (baseline case) that was designed to investigate heat transfer rate and fluid flow at different pitch sizes, with D_{c1} = 114 mm for the first coil and D_{c2} = 150 mm for the second coil. Moreover, six distinct double coils were created, each with a different pitch change, however the shell of the single and double coil heat exchangers remained the same shape and dimensions. Varied boundary conditions for the shell and coil regions were simulated at various hot and cold flow rates (Reynolds number) (1L/min for hot water, (2, 4, 6, and 8 L/min for cold water) at constant temperatures of 36°C for cold water and 65°C for hot water. The parameters utilized in the numerical investigation were taken from the previous article [24] and are listed in Table 1, while Figure 2 displays the various double coil models used in the current study to determine which pitch had the greatest effects.

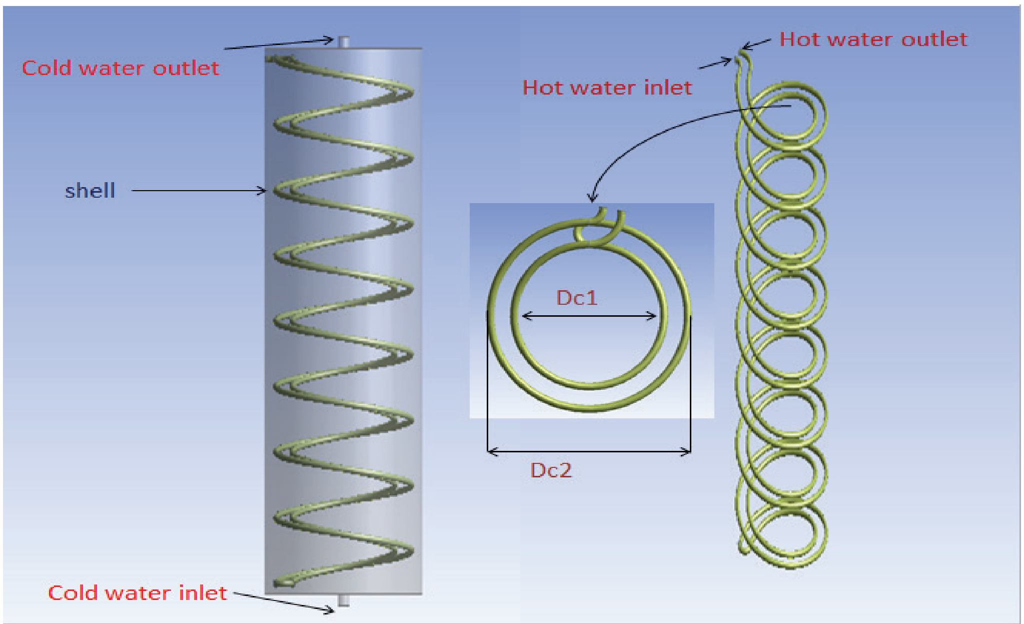


Figure 1. Schematic of the shell and double coil tube (baseline case).

Table 1. Geometric parameters of a variable pitch double coil heat exchanger

Coil No	D_{c1} mm	D_{c2} mm	di_{c1} mm	di_{c2} mm	P mm	Modified Pitch	H_c mm
1 (baseline case)	114	150	4.4	4.4	60	/	480
2	114	150	4.4	4.4	20	P-2P-P	480
3	114	150	4.4	4.4	20	2P-P-2P	480
4	114	150	4.4	4.4	30	P-2P-P	480
5	114	150	4.4	4.4	30	2P-P-2P	480
6	114	150	4.4	4.4	40	P-2P-P	480
7	114	150	4.4	4.4	40	2P-P-2P	480

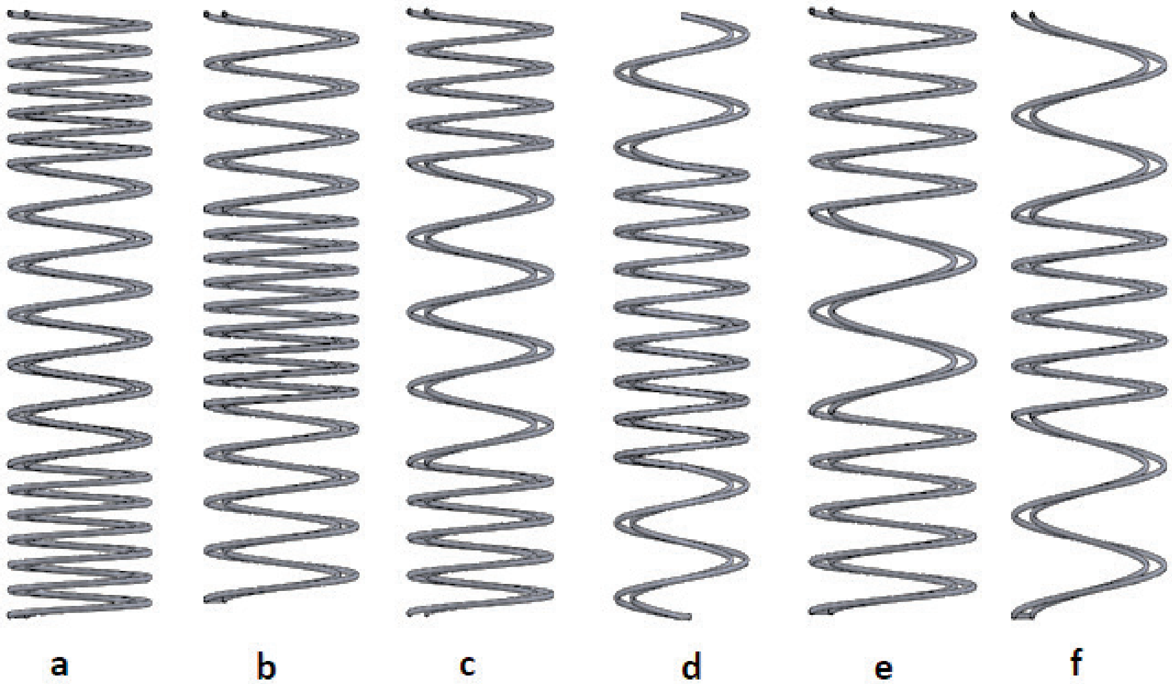


Figure 2. Various models of double heat exchanger a) P-2P-P at P=20mm, b) 2P-P-2P at P= 20mm, c) P-2P-P at P= 30mm, d) 2P-P-2P at P= 30mm, e) P-2P-P at P= 40 mm, f) 2P-P-2P at P= 40mm

Mathematical Formulation and Assumptions

The governing equations were the continuity, momentum, energy equations, and the turbulent RNG k-model was utilized to simulate the turbulent flow during the present study. These equations are mentioned in detail by Senna et al. [24].

To determine the effect of heat transfer through the shell (Q_{sh}) and coil (Q_c), the following relationships were used for numerical and experimental analysis:

$$Q_{sh} = m_{sh}.c_{p_{sh}}.(T_{co,o} - T_{co,i}) \tag{1}$$

$$Q_c = m_c.c_{p_c}.(T_{h,i} - T_{h,o}) \tag{2}$$

It is clear that the heat transfer is affected by the water temperature and the mass flow rate of water on both sides.

Because of the heat transferred on the hot side is not equal to the heat transferred on the cold side, the average binary conversion value (Q_{avg}) was utilized by Khanalari et al. [25] because for all experimental tests, the deviation between heat transfer rater of shell and coil side was not more than 2.37%:

$$Q_{avg} = \frac{Q_{sh}+Q_c}{2} \tag{3}$$

The overall heat transfer coefficient (U_o) through the heat exchanger relative to the shell side can be calculated by the following relationship:

$$U_o = \frac{Q_{avg}}{A_o.\Delta T_{LMTD}} \tag{4}$$

Where A_o represents the outer surface area of the coil while the logarithmic mean temperature difference ΔT_{LTMD} can be calculated by the following relationship:

$$\Delta T_{LMTD} = \frac{\Delta T_{in} - \Delta T_{out}}{ln\frac{\Delta T_{in}}{\Delta T_{out}}} \tag{5}$$

To calculate the lateral fluid heat transfer coefficient (h_c), the following equation can be used:

$$h_c = \frac{Q_{avg}}{A_i(T_h - T_w)} \tag{6}$$

The wall temperature of the coil (T_w) can be calculated by taking an average of four values that was numerically calculated along the length of the coil. The heat transfer coefficient for the shell-side fluid (h_{sh}) can be obtained as follows:

$$h_{sh} = \frac{1}{(\frac{1}{U_o A_o} - \frac{1}{h_c A_i}).A_o} \tag{7}$$

The Nusselt number in the coil side (Nu_c) can be calculated by using:

$$Nu_c = \frac{h_c d_{c,i}}{k_c} \tag{8}$$

K is the thermal conductivity. The Nusselt number in shell side (Nu_{sh}) can be obtained by using:

$$Nu_{sh} = \frac{h_{sh} D_{sh,h}}{k_{sh}} \tag{9}$$

The most general definition of the hydraulic diameter of the shell side ($D_{sh,h}$) based on previously published works can be stated as follows [26].

$$D_{sh,h} = \frac{4(V_{sh,i} - V_{c,o})}{A_{sh.c}} = \frac{D_{sh,i}{}^2 L_{sh} - d_{c,o}{}^2 L_c}{D_{sh,i} L_{sh} + d_{c,o} L_c} \tag{10}$$

The following relationship can be used to calculate the length of the helical tube (L_c):

$$L_c = \pi D_c N \tag{11}$$

The Reynolds number on shell side (Re_{sh}) of the heat exchanger can be expressed as follows:

$$Re_{sh} = \frac{4m_c}{\pi \mu D_{sh,h}} \tag{12}$$

The Reynolds number in the helical coil side (Re_c) of the heat exchanger can be expressed as follows:

$$Re_c = \frac{4m_h}{\pi \mu d_{c,i}} \tag{13}$$

The effectiveness of the heat exchanger (ε) is an important parameter for expressing the expected efficiency:

$$\varepsilon = \frac{actual\ heat\ transfer\ rate}{maximum\ possible\ heat\ transfer\ rate} \tag{14}$$

To evaluate both the heat transfer coefficient and the thermal size through the heat exchanger, the dimensionless parameter of the number of heat transfer units (NTU) can be utilized, which can be determined using equation [27]:

$$\mathrm{NTU} = \frac{U.A_0}{(\dot{m}.c_p)_{min}} \tag{15}$$

Grid Structure and Grid Independence Study

The accuracy of any numerical simulation is governed by the number of cells in the grid, which implies that increasing the number of cells increases the resolution of the solution. More dense grids mean higher accuracy, but at the same time, it means more cost and time. It is vital to pick a mesh size that does not alter the results as the mesh size increases in order to acquire the ideal mesh size that can capture the

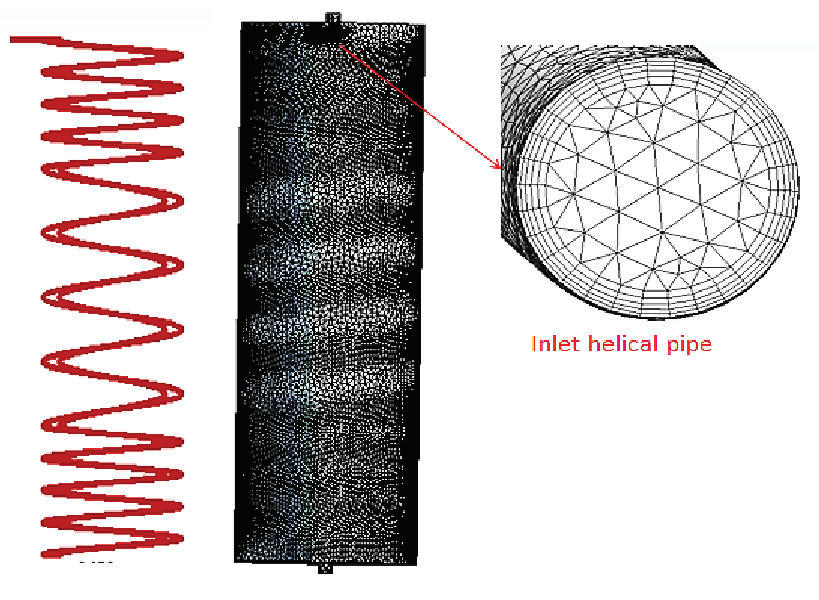
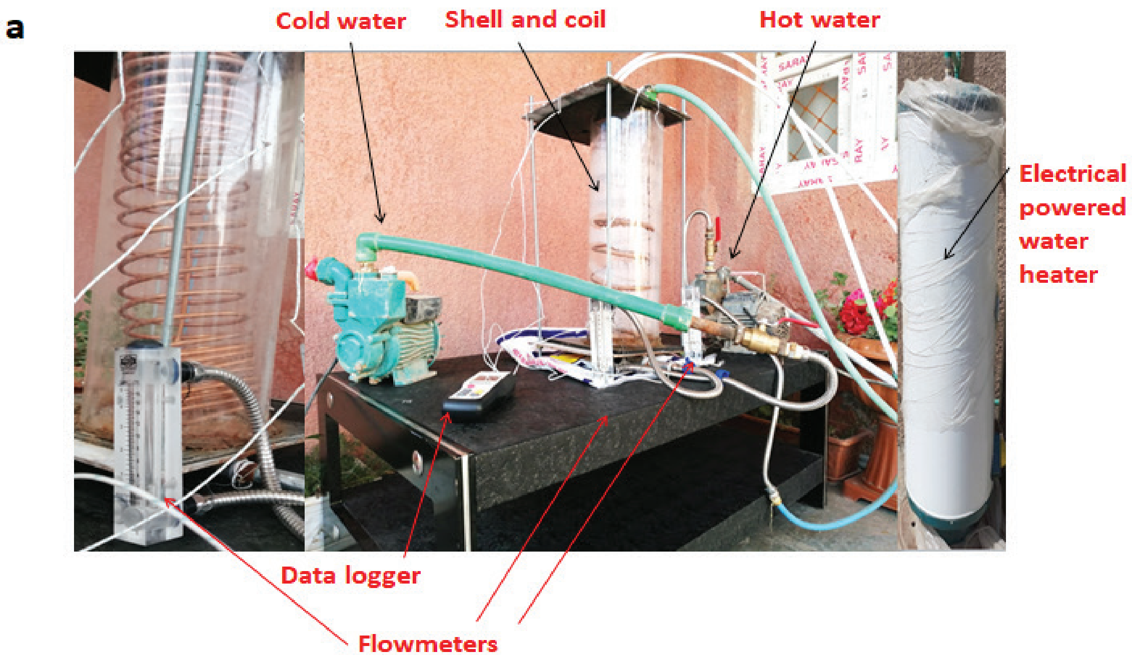


Figure 3. Generated mesh for shell and double coil heat exchanger.

Table 2. The study of mesh independency

Grid	Total mesh	$T_{co,o}$	$T_{h,o}$
G1	1934762	40	45.7
G2	2434640	40.1	45.6
G3	3357188	40.1	45.4
G4	4789892	40.2	45.3
G5	5023730	40.2	45.3

majority of flow parameters. The discretization process is a critical stage in the modeling process in any numerical simulation, and it is essentially defined by the generated mesh for the model that should be solved. In this research, Quadrilateral Dominant cells were generated (see Figure 3) using a cell size of 1 mm with refining option. Five numbers of grids are created G1 to G5 (see Table 2). Furthermore, the least change in output fluid temperature was observed in grid numbers G4 and G5. Because any further mesh refinement



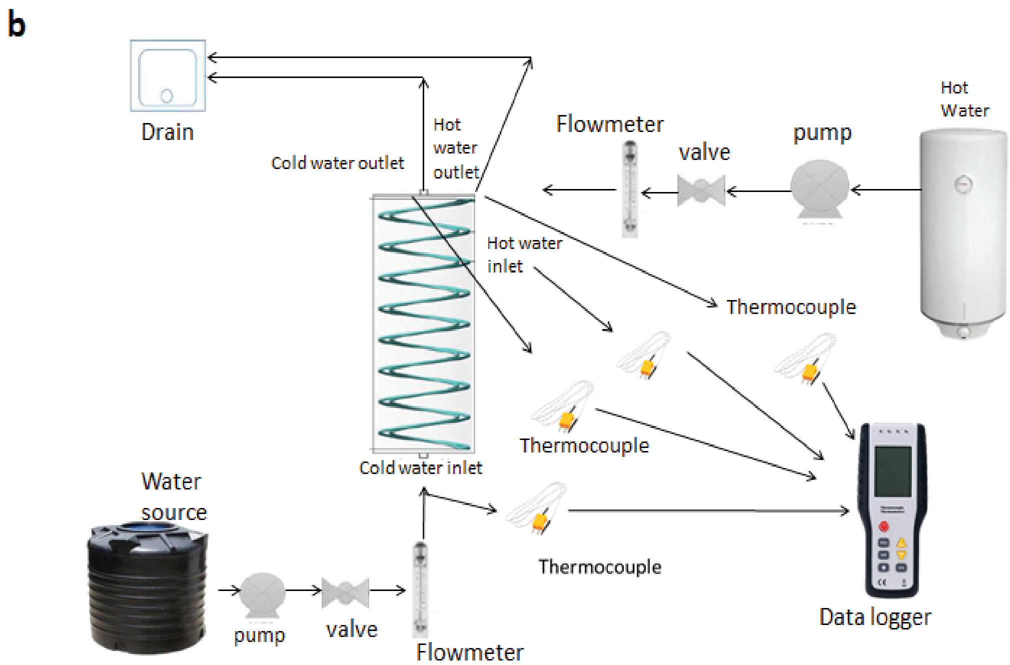


Figure 4. a) Image setup experimental work of double coil, b) Schematic diagram of the experimental set up.

had no meaningful influence on the monitoring data, the G4 grid was used in this investigation. Furthermore, when standard and realizable k-turbulence models are used, very similar results can be obtained, which is why the RNG k–turbulence model is used here [28].

Experimental Setup and Measurement Techniques

Model setup and materials

Regarding the experimental part of the investigation, Figure 4 displays an image and a schematic design of the experimental setup used in the current research. The current study employed three distinct types of heat exchangers: two double coils with variable and fixed pitch, and a single coil with variable pitch. The current helical coil was made of copper tube and was inserted in an acrylic sleeve with a diameter of 175 mm that was positioned vertically during the experiment. The experimental system mainly consisted of shell and coil, with four K thermocouples (uncertainty of ±1.5 °C) connected to a manual data logger of HT-9815 type, a pump, ball valves, and two flow meters of LZM (uncertainty of 0.25 L / min) used to measure volumetric flow rates for both hot and cold water. The four thermocouples were distributed at the inlet and outlet of hot water and inlet and outlet of cold-water to determine the hot water inlet and outlet (*Th*, *Th*) as well as the cold-water inlet and outlet (*Tco*, *Tci*). Experiments were carried out in the hot and cold mass flow rate ranges, as shown in Table 3, with constant temperatures of 36°C for cold water and 65°C for hot water.

Measurement techniques and uncertainty

The present experiment examines four different scenarios, which are mentioned in Table 4. The hot water input temperature is kept at 65°C, and the hot water flow rate is kept constant at 1 L/min. The cold water (35°C) comes from a tap on the bottom of the shell with various flow rates (from 2 to 8 L/min), while the hot water comes from an electric water heater on the coil's upper side. Both sides of the flow rate are controlled by two ball valves, and all temperatures are recorded after 30 minutes to assure the steady-state condition. The dimensions of the shell and double coil (two cases) heat exchangers, as well as the shell and single coil heat exchangers utilized in experimental and numerical simulation, are shown in Table 5. The precision of experimental data was determined by the precision of each measuring instrument and calculating technique.

Generally, the precision of the results is determined by the precision of the methodologies and the accuracy of the specific measuring equipment. In this study, to estimate the uncertainty, the Moffa [29] approach was used. Equation evaluates the uncertainties in computing a result (σ_R) that are depending on various independent variables:

$$\sigma_R = [(\frac{\partial R}{\partial x1}\sigma 1)^2 + (\frac{\partial R}{\partial x2}\sigma 2)^2 + \cdots + (\frac{\partial R}{\partial xn}\sigma n)^2]^{\frac{1}{2}} \quad (16)$$

The derived variables are represented by R, which is a function of the independent variables X1, X2,….., Xn, and indicates the uncertainty. So, to compute the uncertainty in the Nusselt number for the shell side using the following equation can be used:

$$W_{Nu_{sh}} = \sqrt{(\frac{D_h}{K_{sh}} W_{h_{sh}})^2 + (\frac{h_{sh}}{K_{sh}} D_h)^2 + (\frac{-h_{sh} D_h}{{K_{sh}}^2} W_{k_{sh}})^2} \quad (17)$$

Table 3. Test conditions for the shell and coil sides

Water mass flow rate of coil side (L/min)	Temperature for cold water (°C)	Temperature for hot water (°C)	water flow rate of shell side (L/min)
1	36	65	2
	36	36	4
	36	36	6
	36	36	8

Table 4. Geometric parameters of the shell and coil heat exchanger

Experiment No	part	t mm	D_i mm	D_{c1} mm	D_{c2} mm	di_{c1} mm	di_{c2} mm	N	P mm	Modified Pitch	H mm
1	double coil	1.6	/	114	150	4.4	4.4	8	60	/	480
	shell	5	175	/	/	/	/	/	/	/	500
2	Single coil	1.6	/	114	/	4.4	/	24	15	P-2P-P	480
	shell	5	175	/	/	/	/	/	/	/	500
3	Double coil	1.6	/	114	150	4.4	4.4	12	30	P-2P-P	480
	shell	5	175	/	/	/	/	/	/	/	500

Table 5. Average uncertainties of the performance parameters

Characteristics	Unit	Uncertainty
Hot water inlet temperature (T).	°C	±1.5
Hot water outlet temperature (T).	°C	±1.5
Cold water inlet temperature (T).	°C	±1.5
Cold water outlet temperature (T).	°C	±1.5
Coil side water flow rate	L/min	±0.25
Shell side water flow rate	L/min	±0.25

RESULTS AND DISCUSSION

The experimental and computational findings of studying shell and helically coiled heat exchangers are presented and explained in detail in this section.

Numerical Study Results

The topic of heat transfer through heat exchangers is of great interest to academics because of the volume of research and new ideas. While examining previous research, it has been started where others end rather than duplicating what had already been done. First, the experimental results of

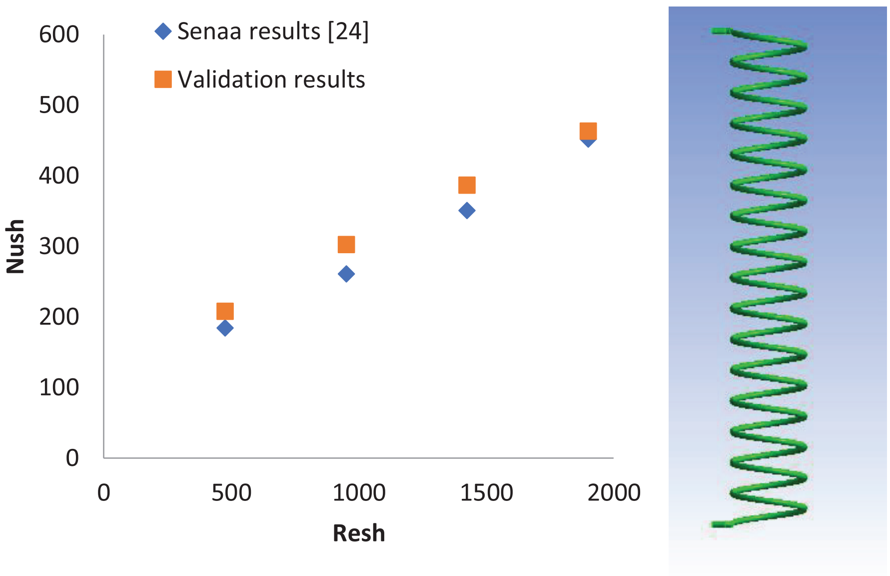


Figure 5. Validation study for a single coil inside the shell

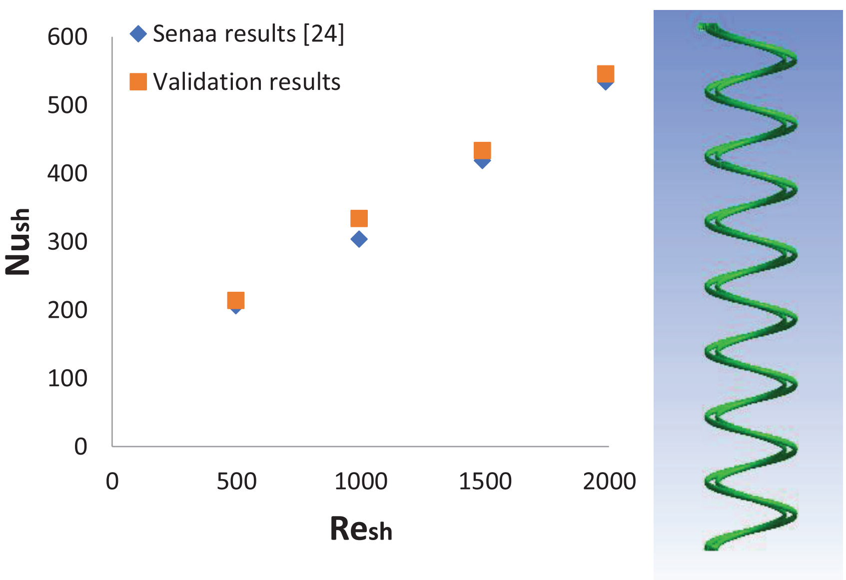


Figure 6. Validation study for a double coil inside the shell

Senaa et al. [24] are theoretically validated using single coil positioned vertically inside the shell, as shown in Figure 5, which depicts the high level of agreement attained during the validation test with a 9% error rate. The validation was also carried out on a double coil placed vertically inside the shell using the same dimensions and parameters to show how close the results are. Figure 6 shows the high level of agreement achieved during the data validation with an error rate of 5%.

Following encouraging validation findings for single and double coils within the shell, the focus turned to improving the heat transfer rate inside the double coil heat exchanger at different pitch sizes using numerical and experimental analysis. Senaa et al. [24] performed a large number of simulations with different coil diameters (dc), and second coil curvature diameters (D_{c2}) to identify the best model for the heat transfer process. Figure 7 depicts the increase in the average heat transfer rate through double coil compared to the single coil, where double coil model resulted in a higher secondary flow rate than the single coil, resulting in a greater heat transfer rate by lowering the temperature of the water leaving the coil. Therefore, following the success of the double coil inside the shell, the attention moved to enhancing heat transfer rate through the double heat exchanger at various pitch sizes.

During the current study, the optimal design of the double coil inside the shell was examined, which would produce the best possible outlet temperature and thus increase the amount of heat transferred. Furthermore, the diameter of the first coil and the diameter of the curvature of the second coil are kept constant throughout the model, and the effects of pitch modification as well as the relationship between numerous pitches are examined. Figure 2 shows schematic models with various pitch dimensions that were evaluated throughout this study, while the geometric dimensions of the shell and the first bend diameter were kept constant, and the coil side Reynolds number was 11100. The most challenging aspect of work is maintaining

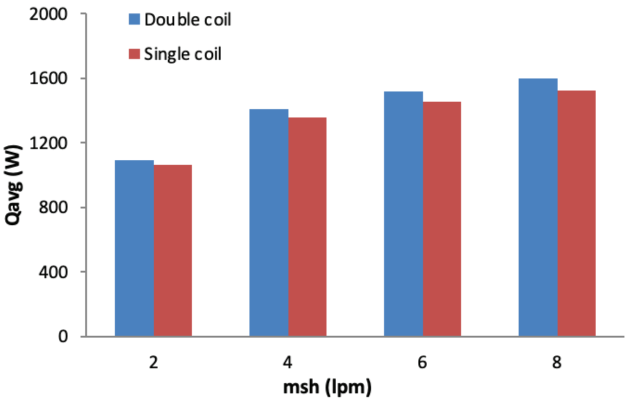


Figure 7. Comparison of average heat transfer rate variation for single and double coil at different mass flow rate.

a strong secondary flow, which enhances the rate of heat transfer in helical tubes, while changing the pitch at the same height in all cases.

In coil-tube heat exchangers, the highest tube side Nusselt number can be obtained by the lowest coil pitch and the highest tube side flow rate [13]. While a higher coil sharpness implies a faster heat transfer rate, a higher number of turns at the same coil and shell height does not always indicate a better heat transfer process. However, an increase in the number of turns does not imply a significant improvement in the Nusselt number on the shell side because this leads to a decrease in the step distance over time, which has two negative consequences: an increase in the number of turns leads to greater convergence between turns, making the helical coil behave like a straight tube, which results in lower efficiency [29]. Another hypothesis is that convection enhanced heat transfer between the tubes, and hence reducing heat transfer with the outside.

Figure 8 compares the numerical results from the simulation of six models with different pitch dimensions in terms of the average heat transfer rate, with the fourth model at P = 30 mm (P-2P-P) having the biggest improvement in the average heat transfer rate, which will be investigated experimentally. This modified pitch achieved a greater improvement in the average heat transfer because it maintained the flow of the hot water inside the coil for a longer period than the other pitches resulting in a higher convective heat transfer between cold and hot water. Therefore, it has been suggested that the changed pitch provided the coil an acceptable curvature diameter through a height of 480 mm, and then generated the secondary flow, which enhanced the average heat transfer.

Figure 9 depicts the temperature distribution in a helically coiled tube in two models: one with a variable coil pitch (P-2P-P at P=30mm) and the other with a conventional pitch (p=60mm). Because the velocity of hot water remains within the coil for a longer period of time in the first model (P-2P-P at P=30mm), a larger heat transfer occurs between the water on the shell side (cold water) and the water on the coil side (hot water). Therefore, reducing the coil pitch causes the coil's curvature diameter to increase, resulting in strength secondary flow in the coil side, which enhances the heat transfer rate in the coil tube.

Figure 10 shows how the heat transfer rate of the shell side of a double helical tube changes with the mass flow rate in the case of using more than one pitch (P-2P-P) of the same model compared to a conventional double helical tube. The heat transfer rate in the modified pitch heat exchanger increased by 10% compared to the conventional heat exchanger, suggesting that the simulation findings supported the effective design of the modified helical tube. As a consequence, the modified pitch heat exchanger (P-2P-P) was constructed during this study based on numerical simulation findings and then compared to experimental data.

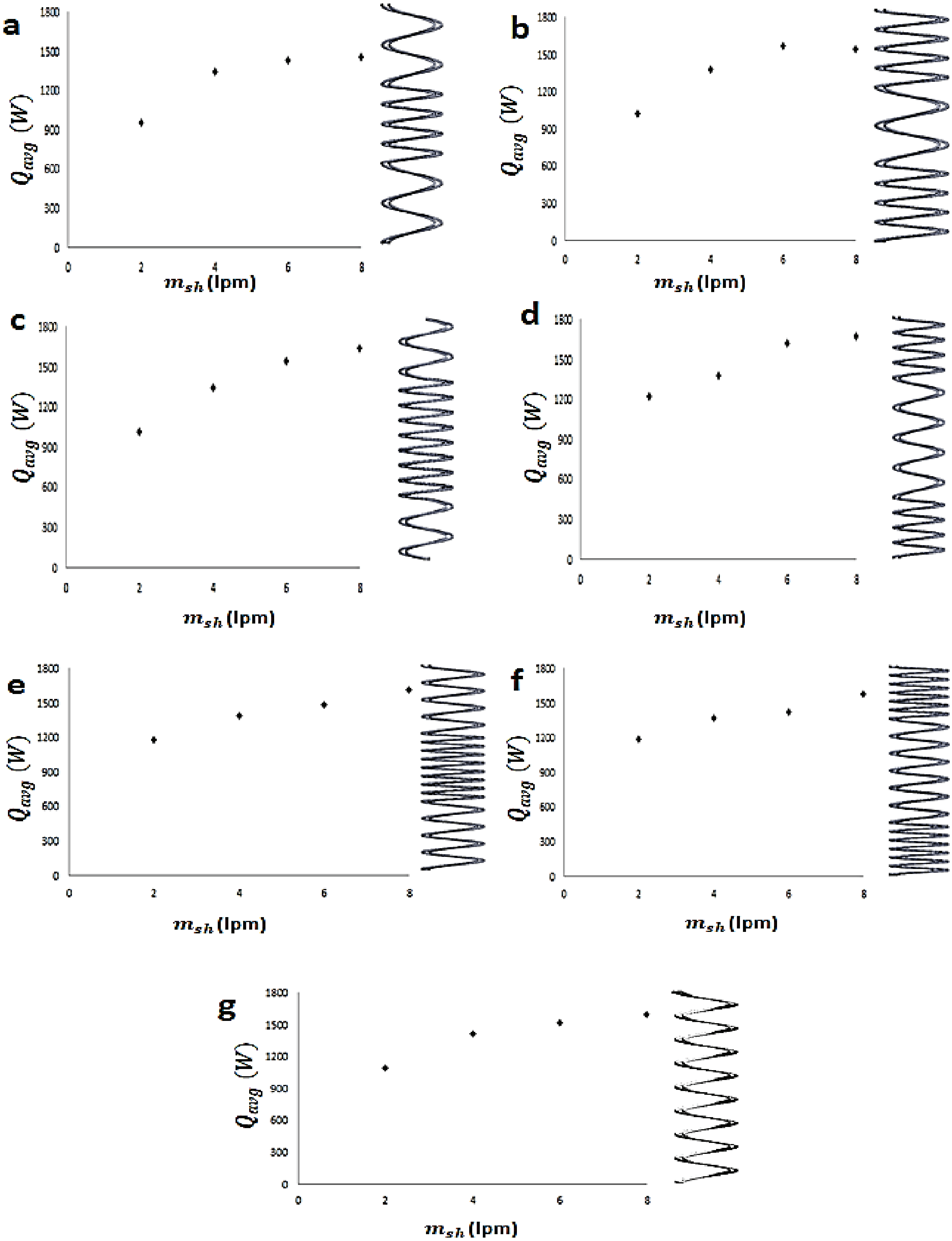


Figure 8. Variation of the average heat transfer with coil side Reynolds number 11100. a) 2P-P-2P at P=40mm, b) P-2P-P at P= 40mm, c) 2P-P-2P at P= 30mm, d) P-2P-P at P= 30mm, e) 2P-P-2P at P= 20 mm, f) P-2P-P at P= 20mm, g) conventional pitch.

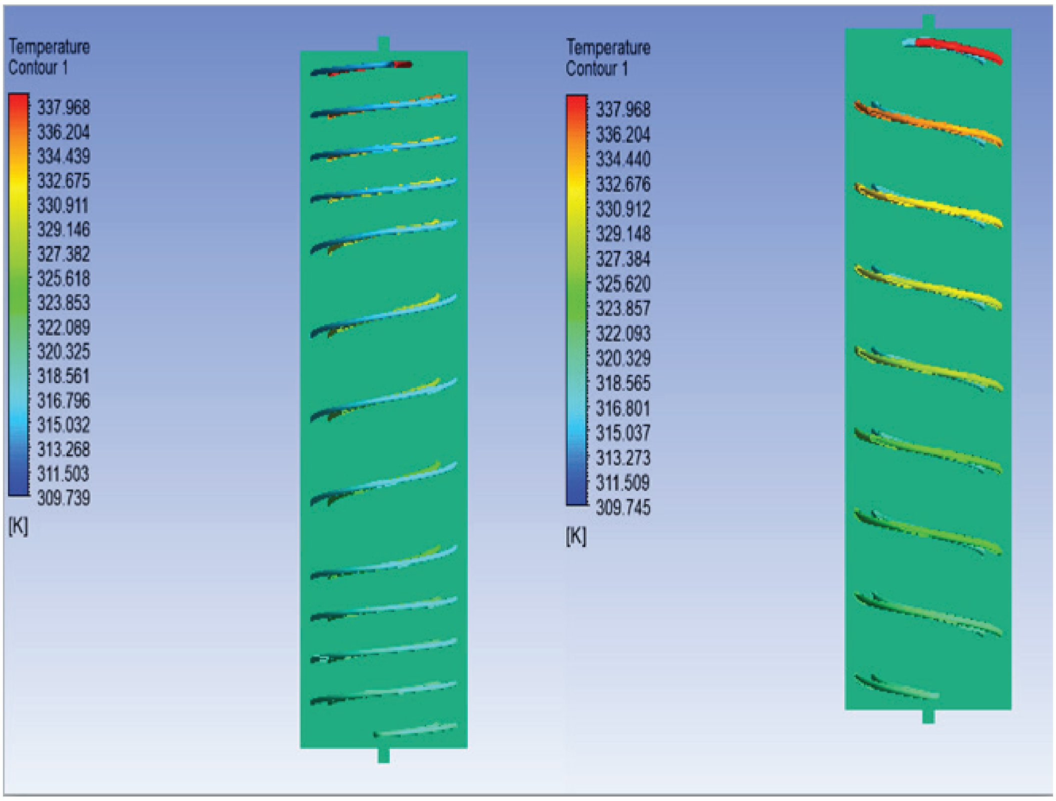


Figure 9. Temperature distribution in two models, a double coiled with modified pitch P-2P-P, and conventional pitch heat exchanger.

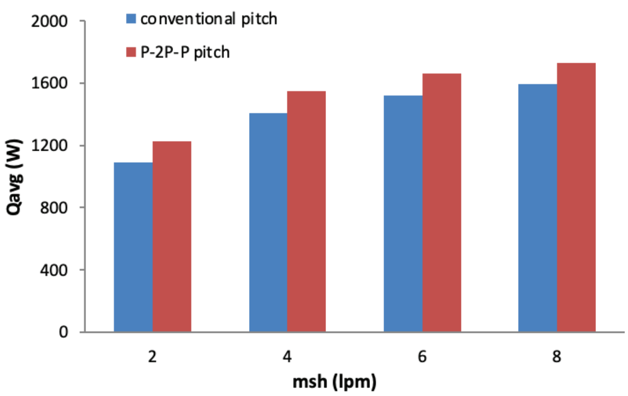


Figure 10. Average heat transfer rate variation at different mass flow rate.

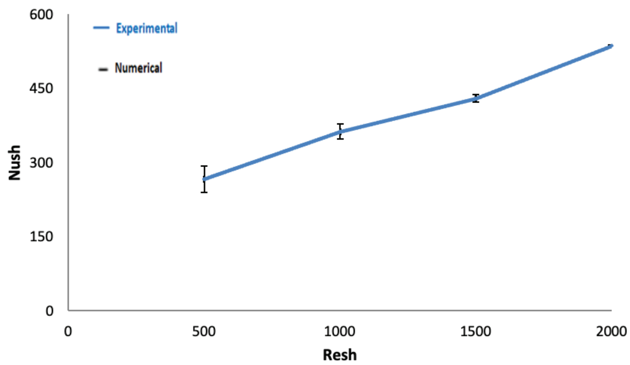


Figure 11. A comparison of the numerical simulation and experimental findings for the Nusselt number on the shell side vs. the Reynolds number.

Experimental Study Results

Experimental validation data were measured to compare numerical simulation findings of the shell side Nusselt numbers. At a constant cold flow rate of 2–8 lit/min and a constant hot flow rate of 1 lit/min, the numerical and experimental results are almost consistent. The current numerical study's conclusions are in good agreement with experimental data, as shown in Figure 11, with a deviation of less than 9%. As a result, the numerical computing approach employed in this work is assured to be appropriately applied.

Figure 12 shows the average heat transfer in the heat exchanger for a single-pitch (conventional) helical tube against a double helical coil with more than one Pitch at a varied flow rate of cold water. As indicated in the figure, the average heat transferred increases as the mass flow rate of cold water increases, it is apparent that utilizing more than

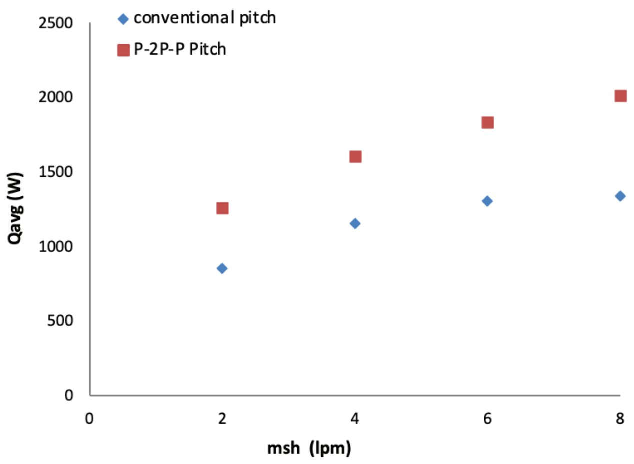


Figure 12. Comparison between average heat transfers rates.

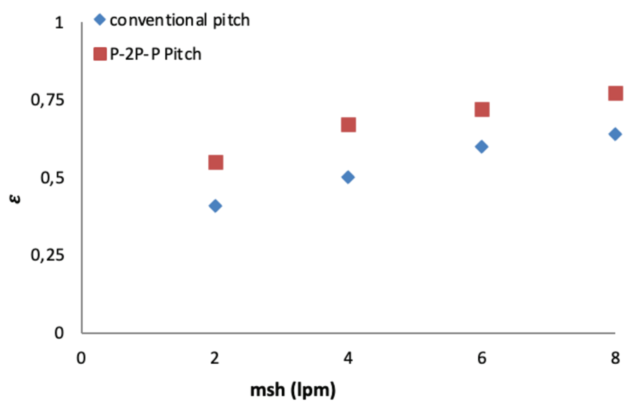


Figure 13. Effectiveness of two double-helical coil models.

one pitch through the helical tube improves the heat transfer rate by 46% compared to using a double coil with a single pitch. The amount of average heat transfers rate to the modified double coil was improved by increasing the contact area of the coil surface by using a modified pitch double coil heat exchanger instead of the conventional double coil.

The largest improvement in average heat transfer in a double helical coil heat exchanger with more than one pitch (P-2P-P) was 46%. The average heat transfer rate in for double helical coil heat exchanger with more than one pitch (P-2P-P) was estimated during this research of 1260-2010 Watts indicating that a good agreement between the current findings and previous research [12, 23-24, 30].

Figure 13 shows the discrepancy in the heat exchanger efficiency in both models, as well as the level of improvement resulting from the new double-tube design over the traditional design. As can be observed, increasing the mass flow rate of cold water flowing through the shell improves the heat exchanger's efficiency while maintaining the mass flow rate of water passing through the tube at 1 L/min. Moreover, Figure 14 illustrates how the number of heat

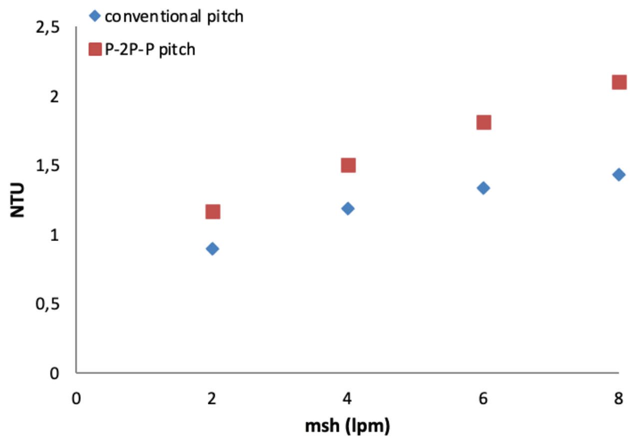


Figure 14. NTU of two double-helical coil models.

transfer units NTU changes with the shell-side flow rate Q_{sh} with a constant coil-side flow rate $Q_c = 1\ LMP$ and a constant temperature differential ΔT. Knowing that increasing the mass flow rate of the shell side leads to a considerable increase in the NTU [31], it is clear that utilizing more than one pitch of double coil increased the NTU by 34.6 %.

This improvement can be attributed to both parameters (efficiency, NTU) probably because decreasing the coil pitch then doubling it back to the same value increases the bending diameter of the coil which leads to a larger secondary flow in the side of the coil which increases the rate of heat transfer in the modified coil tube compared to conventional coils.

Under the constant Reynolds number for the coil side of 11100, the relation between the Reynolds numbers for the shell side and the Nusselt number of the double helical coil in the shell is depicted in Figure 15. The findings reveal that increasing the amount of cold-water mass flow leads to a higher velocity of the fluid in the shell, which increases the convective heat transfer coefficient and Nusselt number. When comparing the modified pitch of a double coil heat

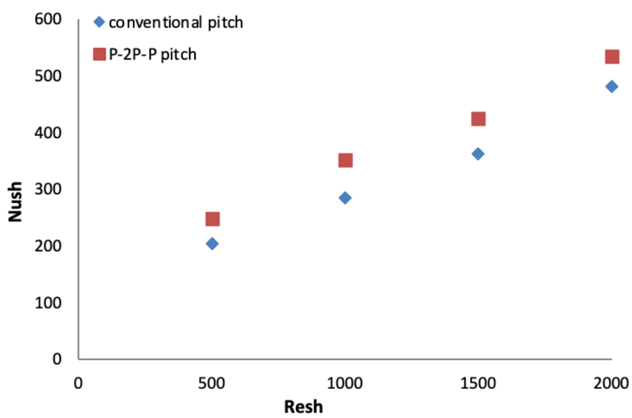


Figure 15. Nusselt numbers of two double-helical coil models.

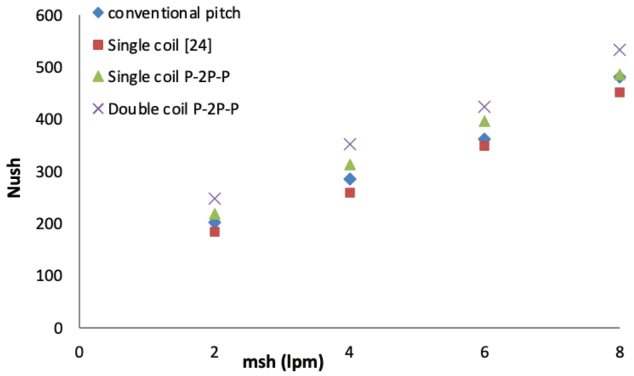


Figure 16. The Nusselt number for the shell side compared between the four models.

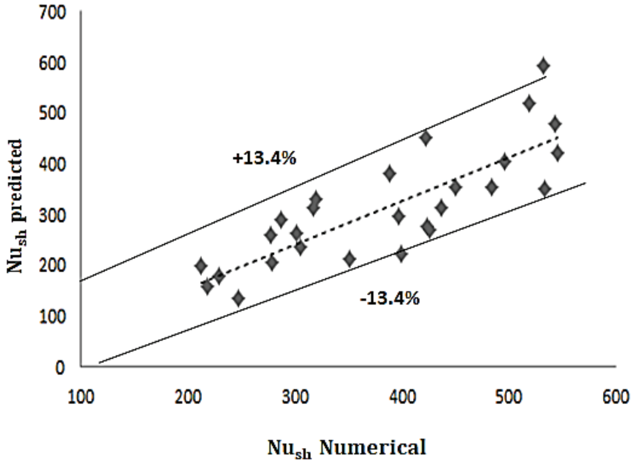


Figure 17. Numerical Nusselt numbers on the shell side compared to anticipated Nusselt numbers throughout a range of Reynolds values on the shell side.

exchanger to a conventional coil under the same conditions, the findings reveal a 19% increase in Nusselt number.

According to the Conversational Law of Mass, with no leakage, the volumetric flow that passes through the first coil is incompressible and is therefore equal to the volumetric flow that moves through the second coil. The use of more than one pitch for the coil keeps the speed of hot water inside the coil for a longer period compared to the traditional coil, which leads to higher heat transfer by convection between cold and hot water. This modification led to an improvement in Nusselt number on the side of shell.

The Nusselt number for the shell side of the three models that have been effectively addressed (one pitch double coil, single variable pitch coil, variable pitch double coil) is compared to the single pitch model in Figure 16. The results demonstrate that in both single and double models, changing the coil pitch enhanced the Nusselt number. The change of pitch leads to a change in the water path in the helical coil tube and then back to its origin path causing a force in the secondary flow and leads to greater heat transfer through the double coil model

To anticipate the Nusselt number of shell side heat exchangers, correlations were constructed using current experimental data within the helically coiled tube. The Nusselt number is linked to the Reynolds and Prandtl numbers in a helical coiled tube. Based on the data acquired from the numerical analysis [38], the Nusselt number is proportional to Pr = 0.4 in most previous studies. As a result, a relationship is created for calculating the Nusselt number of the shell:

$$Nu_{sh} = 1.913 Re_{sh}^{\ 0.647} Pr^{0.4} \tag{18}$$

As can be shown in Figure 17, the predicted correlations are in good agreement with the current experimental results, and hence, equation 18 is valid for 400 $< Re_{sh} <$ 2000, Pr_{sh} = 4.6 with a maximum variation of 13.4%.

Table 6. A general comparisons between this work and similar studies in the literature

Reference	$\dot{m}$	Q_{avg} (W)	NTU	Nu_{sh}
Ali et al. [4]	1-10 L/min		0.6-2.3	
Bahrehmand and Abbassi [12]	0.113-0.3 kg/s	3500-14000		80-160
Salem et al. [19]	1.7-11.158 l/min			50-600
Tuncer et al. [23]	1.5-3.5 l/m	2000-4600		
Senaa et al. [24]	1-8 l/m			100-600
Salimpour [32]	0.016-0.136 kg/s			10-70
Kaliannan [33]	0.3-0.8 Kg/s			30-80
Alimoradi [34]	1-6 l/m	3000-8000		50-400
Alimoradi [35]	1-7 L/min			100-1200
Kumar et al. [36]	1800-2500 l/hours			300-500
Moosavi et al. [37]	1-5 L/min		0.5-3	
The current research	1-8 L/min	800-2500	0.5-2.5	200-600

CONCLUSIONS

An attempt was made to build a new model that gives an improvement in the heat transfer process during the current study using CFD, which included a numerical analysis and experimental study to study the heat transfer properties in double coil heat exchangers. The following conclusions can be derived from the findings of this study that may be beneficial to other researchers:

- In all three models, the numerical analysis used throughout the study was in good agreement with the experimental results, with an error rate of less than 10%.
- When the shell side flow rates and the overall heat transfer coefficient both increases, the Nusselt number increases. The relevance of an increase in the total heat transfer coefficient, on the other hand, is a function of the Reynolds number on the shell side.
- When the Reynolds number was in the range ($400 < Re_{sh} < 2000$) at $Re_c = 11100$, the shell side Nusselt numbers in the double coil helical tube model with a variable pitch of P-2P-P were greater than those in the double coil helical tube model with one pitch (conventional coil).
- The findings from the three models were compared to a single coil in terms of the Nusselt number of shell side, and the results revealed that the double model with a variable pitch outperformed the other models.
- Increase the mass flow rate of cold water on the shell side would increase the effectiveness at a constant hot water mass flow rate, however the temperature of hot water at the exit decrease with an increase in cold water flow rate in the shell.
- As a function of the investigated parameters, correlations for the average Nusselt numbers of shell side heat exchangers were established.

NOMENCLATURE

A	Area, m^2
C_p	Specific heat, J/(kg.K)
D_c	Curvature diameter, m
D_e	Dean Number
D_h	Hydraulic diameter, m
f	Friction factor
H	Height, m
h	Heat transfer coefficient, W/m^2
K	Thermal conductivity, W/(m °C)
L	Length, m
$\dot{m}$	Mass flow rate, kg/s
N	Number of turns
Nu	Nusselt number
NTU	Number of heat transfer unit
P	Coil pitch, m
Pr	Prandtl number
Q	Heat transfer rate, W
Re	Reynolds number
T	Water temperature, °C
t	thickness, m
U	Overall heat transfer coefficient, W/(m2 K)
V	Velocity, (m/s)

Greek symbols

γ	Dimensionless pitch ratio
ρ	Density, kg/m^3
μ	Dynamic viscosity, kg/(m s)
ε	Effectiveness
$\Delta TLMTD$	Logarithmic mean temperature difference, K

Subscripts

Avg	Average
c	Coil side
co	Cold
h	Hot
i	Inlet
o	Outlet
sh	Shell side
w	Wall

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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GRAPHICAL ABSTRACT