ISSN: 0974-5823 Vol. 7 No. 2 February, 2022

International Journal of Mechanical Engineering

Thermal Performance Analysis of Modified Single Coil Heat Exchanger

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ABSTRACT

In the present work, attempts have been made to enhance the heat transfer rate in single-tube coiled and shell heat exchanger numerically and experimentally. Initially, the experimental results showed a good agreement with the numerical simulation results. Then, numerical analysis was carried out to find out the effect of using multiple pitches with various configurations on thermal performance and heat transfer rate. Following positive numerical findings, a single coiled tube was built with a variety of degrees while preserving the essential design parameters of tube diameter, shell diameter, shell height, and coil height. In addition, the experimentally designed heat exchanger's performance was evaluated under various settings to evaluate how efficient it was at enhancing the heat transfer process. The findings revealed that the revised design improved the secondary flow through the coil, resulting in greater heat transfer process values. On the shell side, the rate of improvement in Nusselt numbers was 15%.

Keywords: Single helical coil; Heat exchanger; Numerical analysis; Heat transfer enhancement

1. Introduction

Heat exchangers have an important role in the operation of many systems such as power plants, manufacturing industries, and heat recovery units. The centrifugal forces in the spiral coiled tubes create a pair of longitudinal vortices which increase the heat transfer coefficient. Heat transfer is the science that seeks to predict the energy transfer that may occur between physical bodies as a result of the temperature difference [1-4]. C. Sadhasivam et al [5] used a helical coiled heat exchanger to transfer heat from a single liquid in a tube in different wavy paths. An experimental analysis was performed to study the heat transfer coefficients for different cross-sections. The results proved that the mixture of ethylene glycol and water gave effective improvements in heat transfer by 10 to 15% the heat from the pipe. An experimental and numerical study on the conical ring in the annular side of the heat exchanger was conducted by Sheikholeslami et al [6]. It has been found that the pitch ratio affects the performance characteristics.

Rennie and Raghavan [7 and 8] conducted two experimental and numerical studies on double tube heat exchanger. The results showed that as the Dean number increased, the total heat transfer coefficients increased [9-11]. In order to check the flow and heat transfer characteristics of the heat exchanger, a numerical analysis was also carried out by Y. G. Jong et al [12]. It was found that the centrifugal force has an important role in the behavior of pressure drop and heat transfer due to the curvature effect. It was also found that the friction factor and the Nusselt number are proportional to the square root of the Dean number. Numerical and experimental research was performed by Q. Mahdi et al [13] to evaluate the performance of a coiled helical tube heat exchanger with wire coil inserts. The results showed that the Nusselt number increases with the increase of the Dean number and the decrease in the sharpness of the wire coils. The coiled square wire also provided higher heat transfer than the circular wire under the same conditions. Tuncer et al [14] found that the incorporation of a hollow tube into the shell side of the heat exchanger led to the regulation of fluid flow in the shell side, which improved heat transfer.

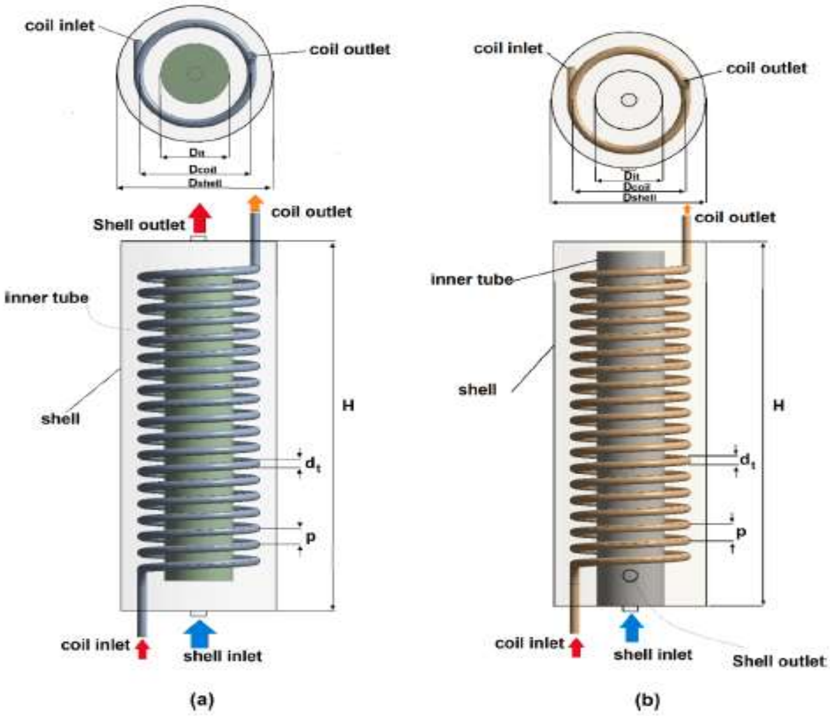

Figure 1: Presentation for simulating heat exchangers; a) traditional type. b) Modified type.

Yildiz et al [15] used a heat exchanger created by placing spring-shaped wires of varying pitches inside a helical tube. The results showed that at the same Dean number, the Nusselt number increased with decreasing the pitch/wire diameter ratio by up to five times with respect to an empty tube. During them study of the effect of curvature ratio on creating turbulent flow and heat transfer in helically coiled tubes, Piazza and, Ciofalo [16] concluded that the main parameter of flow densification in coil tubing is the curvature ratio. The coil in helical coil heat exchanger has been investigated to study fluid flow and heat transfer under turbulent condition by Mandal [17]. Hot air and cold water were used in inner tube and outer tube as working fluid. The inner tube Nusselt number of the compressed air was found to be slightly higher than the data reported in the literature for ambient conditions while the friction factor values are within the range reported in the literature for ambient conditions. The fluid flow behavior in concentric and eccentric rings with and without a rotating inner tube was numerically investigated by Bicalho et al [18]. It was found that liquid to liquid heat transfer in the ring and inner tube complicates the design of tube in tube helical coil heat exchangers, in which heating or cooling is provided by a secondary fluid, with the two liquids separated by the inner coil wall.

In an experimental study on spiral coils are placed cascading on the cylindrical shaped shell inside the heat exchanger, M.Vivekanandan [19] showed that the lateral flow rate of the shell of 10 lpm gave the highest efficiency of the different flow rates on the side of the tube from 2 to 6 lpm. Pandey et al [20] attempted to enhance the heat transfer and fluid flow properties by using a Y-shaped insert inside a circular heat exchanger tube. The results indicated that at Re = 3000, the highest heat transfer rate of 5.05 times and the thermal performance factor of 2.88 times was obtained over the smooth tube of the non-perforated input case.

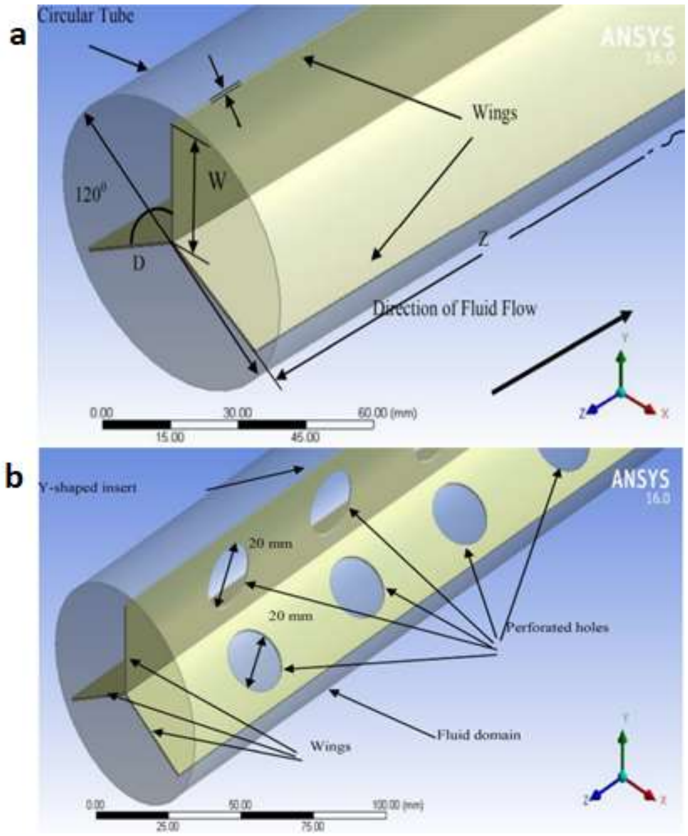

Figure 2: Physical Model of Y-shaped insert inside a circular tube a) smooth Y- shape, b) Y-Shaped insert holes [20]

Nusselt Number is one of the most important parameters to know the improvement of the new design is improved by increasing the contact area of the coil surface using single coil heat exchanger with modified pitch. As a result, Nusselt number of two single coils, one with a constant pitch and the other with a changing pitch, was compared. Furthermore, numerous papers focused on modifying the coil diameter, using nanofluids molecules, or employing fins to improve heat transfer in heat exchangers, but no more than one pitch will be employed in the same coil. By determining Nusselt number, the modified pitch technique will be compared to the conventional pitch in this study. Figure 3 shows the major structures of the present study.

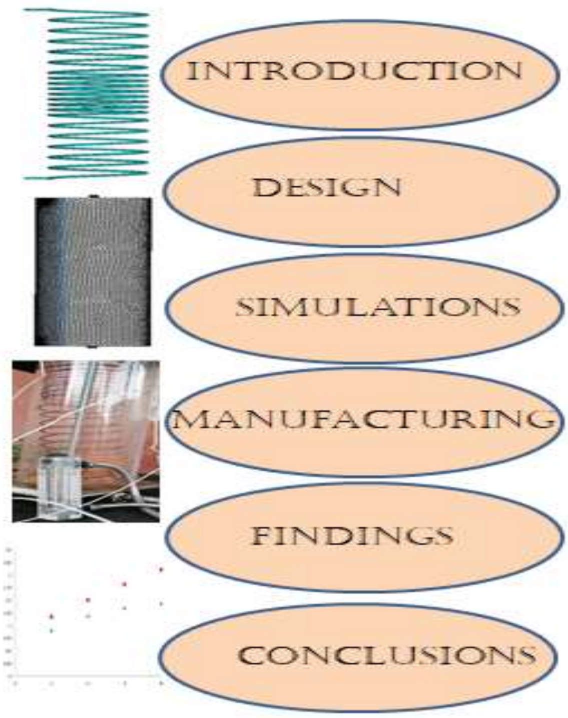

Figure 3: Diagram showing the work carried out during this research.

2. Numerical Methodology

2.1 Problem Description and Boundary Conditions

Figure 4 shows the vertical shell configuration with single coil modified pitch. Three-dimensional models are created using Solid Works as shown in Figure, these models were transferred to ANSYS R16.1 to evaluate the flow and heat transfer efficiency and outlet temperatures for hot and cold water. Different boundary conditions were set for the shell and coil regions at water mass flow rate (1L/min for hot water, (2, 4, 6, and 8 L/min for cold water) and constant temperatures for cold water (shell side) and hot water (tube side) 36°C, 65°C respectively. Figure 5 depicts the dimensions of single coil modified pitch and conventional single coil heat exchangers.

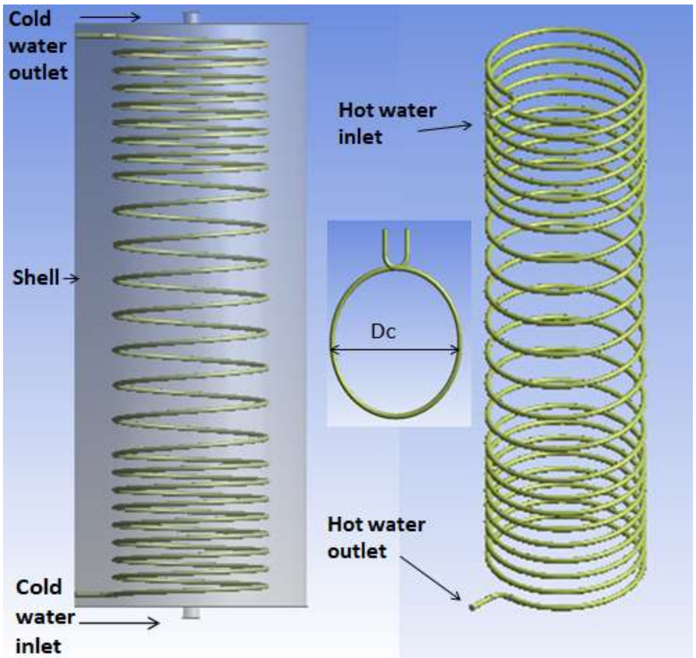

Figure 4: Schematic of shell and single coil tube (modified pitch).

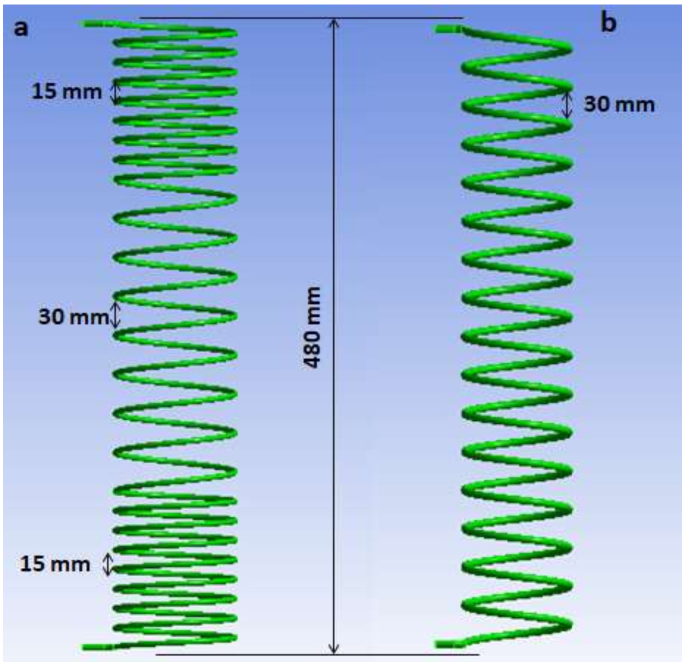

Figure 5: The design of single coil heat exchanger a) modified pitch, b) conventional pitch

2.2 Mathematical Formulation and Assumptions

The governing equations were the continuity, momentum, and energy equations, and the turbulent RNG k-model was used to simulate the turbulent flow during the current study. These equations can be showed:

Continuity equation:

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0 \tag{1}$$

Momentum equation:

$$\frac{\partial}{\partial x_i}(\rho u_j u_i) = -\frac{\partial \rho_{re}}{\partial x_i}+\frac{\partial \tau_{ij}}{\partial x_j} \tag{2}$$

Energy equation:

$$\frac{\partial}{\partial x_i}\left(\rho u_i\left(\theta+\frac{1}{2}u_j\tau_{ij}\right)\right)=\frac{\partial}{\partial x_i}(u_i\tau_{ij})+\frac{\partial}{\partial x_i}(k\frac{\partial T}{\partial x_i}) \tag{3}$$

Where $\tau_{ij}$is an expression of viscous stress tensor defined as:

$$\tau_{ij} = 2\mu S_{ij} \tag{4}$$

Where S_{ij} in turn is an expression of strain rate tensor defined as:

$$S_{ij} = \frac{1}{2}(\frac{\partial u_i}{\partial x_i}+\frac{\partial u_j}{\partial x_i}) \tag{5}$$

K is turbulent kinetic energy defined as [25]:

$$\frac{\partial}{\partial x_i}(pku_i) = G_k+\frac{\partial}{\partial x_i}\left(a_k\mu_{eff}\frac{\partial k}{\partial x_i}\right)-\rho\varepsilon \tag{6}$$

To determine the Nusselt number in shell side (outside of coil) can be obtained by using:

$$Nu_{sh} = \frac{h_{sh}D_{sh,h}}{k_{sh}} \tag{7}$$

The most general definition of the hydraulic diameter of the shell side based on previously published papers can be stated as follows [26].

$$D_{sh,h} = \frac{4(V_{sh,i}-V_{c,o})}{A_{sh.c}} = \frac{D_{sh,i}{}^2L_{sh}-d_{c,o}{}^2L_c}{D_{sh,i}L_{sh}+d_{c,o}L_c} \tag{8}$$

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

$$L_c = \pi D_c N \tag{9}$$

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 can be obtained as follows:

$$h_{sh} = \frac{1}{\left(\frac{1}{U_O A_O}-\frac{1}{h_C A_i}\right).A_O} \tag{10}$$

Table 1 shows the designation of boundary conditions. The inlet boundary condition was set as the mass flow rate for both cold and hot water, the outlet boundary condition was select as pressure outlet with zero back flow pressure. Semi Implicit Method for Pressure Related Equations was utilized to solve coupling between pressure and velocity fields. The first order upwind system was utilized for discretization pressure, momentum, energy and, RNG k-ε turbulence equations.

Table 1: Boundary conditions during numerical validation

Parameters	Shell side	Coil side
Working fluid	water	water
Material	Acrylic	Copper
Inlet temperature	36 °C	65 °C
Inlet mass flow rate	2-8 L/min	1 L/min
Outlet	Pressure	Pressure
Wall	No slip, No heat flux	Coupled

2.3 Mesh independency check

During the current research, the mesh has been improved in some areas such as the inlet and outlet of the shell and the area relatively close to the outer surface of the coil as shown in Figure 6. As for the field of the shell and coil, Quadrilateral Dominant cells were used. The mesh size of the five cells shown in Table 2 is determined by which mesh has the best capacity to capture the majority of the flow characteristics. As a result, the G4 grid was used in the study since increasing the mesh size had no obvious effect on the monitoring data.

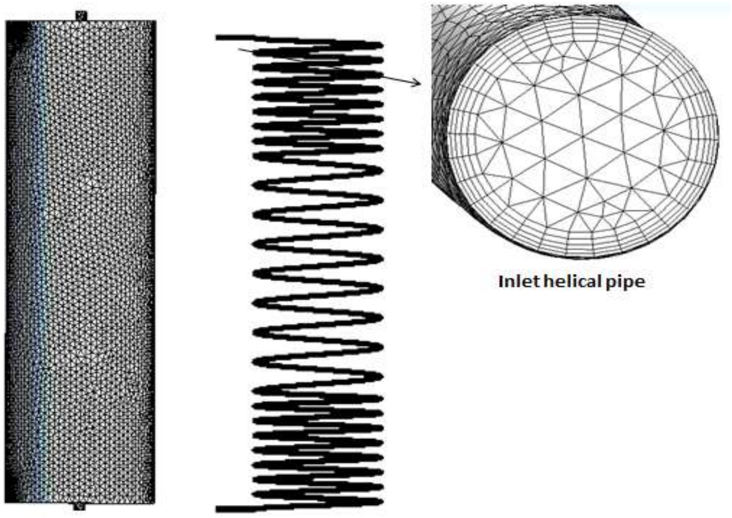

Figure 6: Generated mesh for shell and, single coil heat exchanger.

Table 2: The study of mesh independency

Grid	Total mesh	$T_{co,o}$	$T_{h,o}$
G1	1774922	39.2	46.4
G2	2328890	39.3	46.2
G3	3181784	39.5	46.2
G4	4478920	39.6	46.2
G5	4890412	39.6	46.1

3. Experimental work

3.1 Materials and setup

Figures 7a and b display an image and a schematic design of the experimental setup used in the current study. The current study employed one type of heat exchangers: single coils with variable pitch and comparing it with the previous paper [2], which was practically working on a single coil with a traditional 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 placed vertically during experiments. Experimental system consisted of shell and coil, with four K thermocouples (uncertainty of ±1.5 °C) linked to a manual data logger of HT-9815 type, pump, ball valves, and two flow meters of LZM (uncertainty of 0.25 L / min) used to calculate volumetric flow rates for both hot and cold water. The four thermocouples were distributed at the inlet, outlet of hot water and inlet and outlet of cold water to determine the hot water inlet and outlet ($T_{h,i}$, $T_{h,o}$) as well as the cold-water inlet and outlet ($T_{co,i}$, $T_{co,o}$).

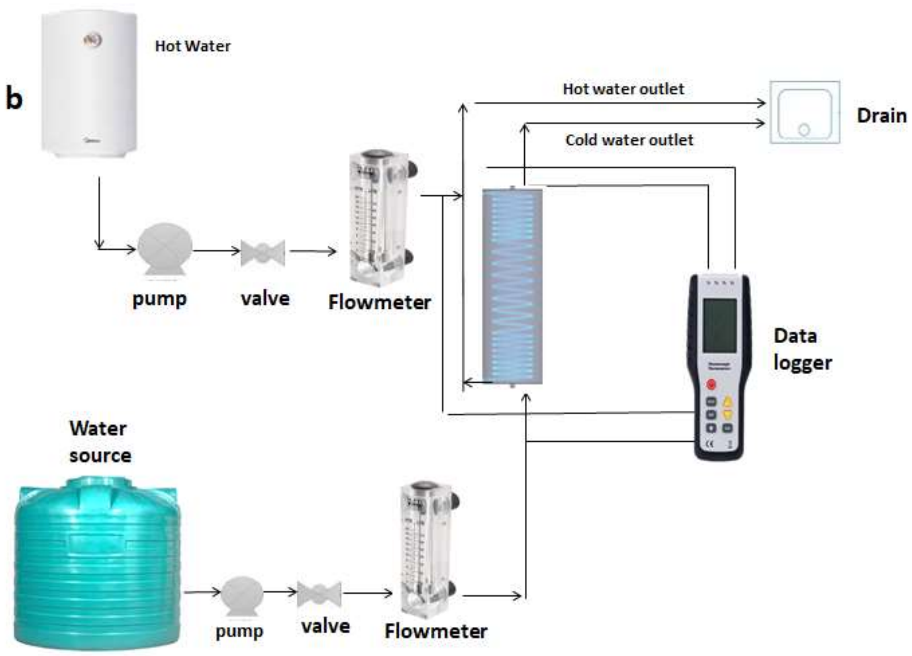

Figure 7: a) Image setup experimental work of single coil, b) Schematic for experimental setup.

3.2 Experimental procedure

The performance tests of single coil heat exchanger have been conducted to determine heat transfer and flow behavior. The performance tests have been done in 4 different flow rates of cold water (shell side) and at constant flow rate of hot water (coil side). In the experiments, the inlet temperature of hot water has been adjusted to a set temperature (65°C) the inlet temperature of cold water has been adjusted to a set temperature (35°C). After reaching steady conditions (after 30 minutes) the values have been recorded. The dimensions of the shell and single coil heat exchangers utilized in experimental and numerical simulations are shown in Table 3. The precision of experimental data was calculated by the precision of each measuring instrument and calculating technique.

Table 3: Geometric parameters of the shell and coil heat exchanger

Section	t mm	D_i mm	D_c mm	di_c mm	N	P mm	Modified Pitch	H mm
Single coil	1.6	/	114	4.4	24	15	P-2P-P	480
shell	5	175	/	/	/	/	/	500

4. Results and Discussion

In this section, the experimental and computational results of helical shell and coil heat exchangers are presented and discussed

4.1 Numerical Study Results

Figure 8 shows temperature distribution counters for the shell and single coil modified pitch heat exchangers at a mass flow rate of 8L/min for cold water. The heated water on the coil's side continually exchanges heat with the cold water on the shell's side, as can be seen in the figure. In other words, the hot water heats the water around the shell. By comparing the output temperature to the input temperature, it can be shown that the heat transfer is effective and clearly seen.

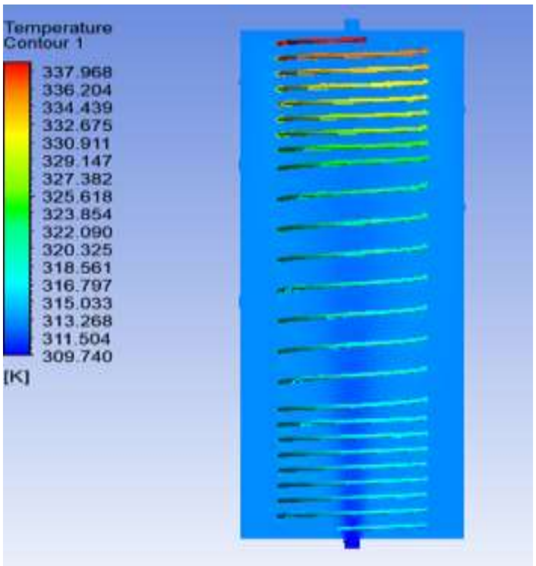

Figure 8: Contour of the temperature for shell and single coil modified pitch.

As mentioned in this paper, a study was conducted using a conventional single coil and compared with a modified pitch coil through the heat exchanger. Figure 9 shows the temperature distribution of the single tube helically coiled in both models (conventional coil and modified pitch coil) at cold water mass flow rate (shell side) 8 L/min and at hot water mass flow rate (coil side) 1 L/min. In contrast to the heat distribution in the conventional coil, the use of more than one pitch through one coil enhanced the secondary flow rate and so offered a more uniform heat distribution.

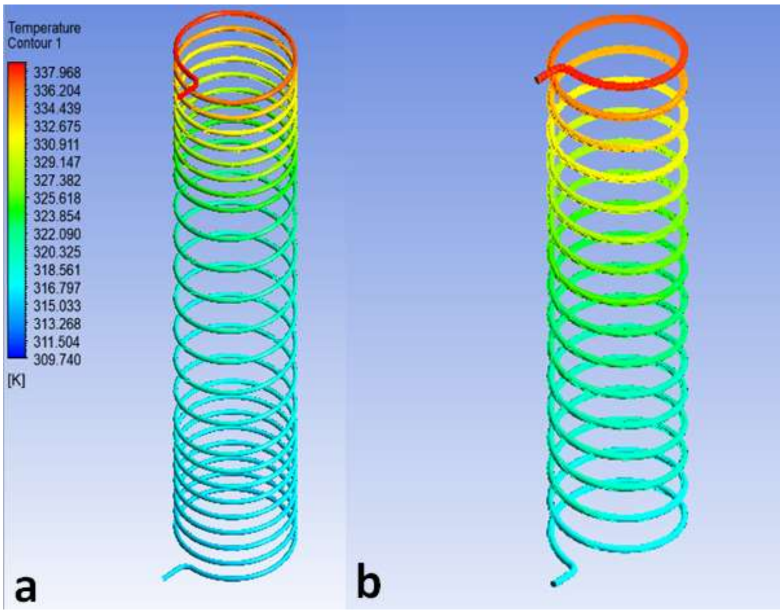

Figure 9: Temperature distribution in single helically coiled tube; a) Modified pitch, b) conventional pitch

The temperature distribution of the modified pitch single helical coiled tube is shown in Figure 10. Figure 10a illustrates the temperature distribution in a single helically coiled tube for the shell side at a mass flow rate of 2 L/min, whereas Figure 10b exhibits the temperature distribution in a single helically coiled tube for the shell side at a mass flow rate of 8 L/min. The figure clearly shows that increasing the flow rate of cold water increased the rate of heat exchange between the coil and the surroundings, implying an increase in the heat exchanger's thermal efficiency.

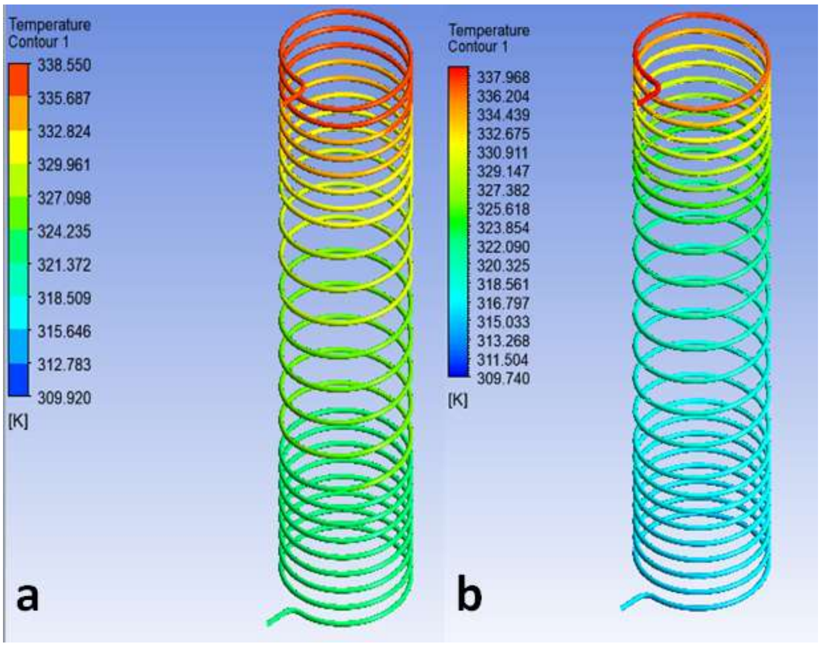

Figure 10: Temperature distribution in single helically coiled tube Modified pitch: a) at 2L/min for cold water, b) at 8L/min for cold water

4.2 Experimental Study Results

The experimental validation data were measured to compare the findings of the numerical simulation of Nusselt numbers on the shell side. At a constant cold flow rate of 2-8 L/min and a constant hot flow rate of 1 L/min, the numerical and experimental results appeared to be similar. The numerical findings of this study agree well with the experimental data, as shown in Figure 11, with a variation of less than 10%. As a consequence, the digital computing technique employed in this research is properly implemented.

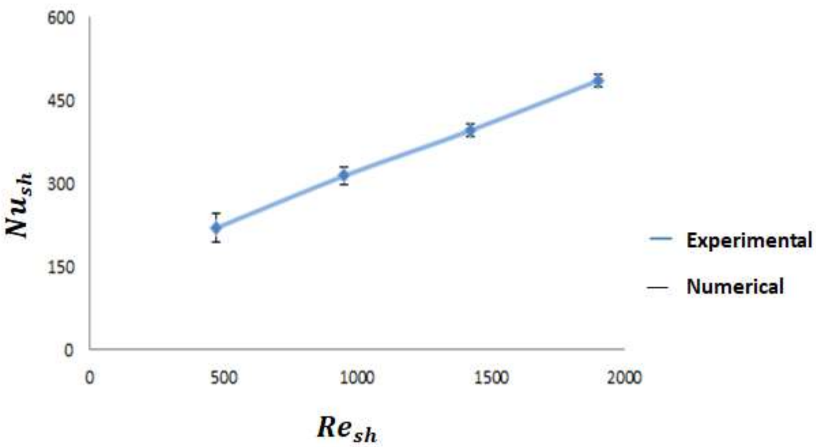

Figure 11: A comparison of the numerical simulation and experimental findings for the Nusselt number on shell side vs. Reynolds number.

At a cold-water mass flow rate, Figure 12 shows the relationship between the Nusselt number of a single coil with a conventional pitch [2] and a modified pitch single coil (shell side). Figure 12 demonstrates that increasing the quantity of cold-water mass flowing into the enclosure increases the velocity of the liquid in the enclosure, resulting in larger Nusselt numbers. When the tuned pitch of a single-coil heat exchanger was compared to that of a conventional coil under the identical conditions, the Nusselt number increased by 14%.

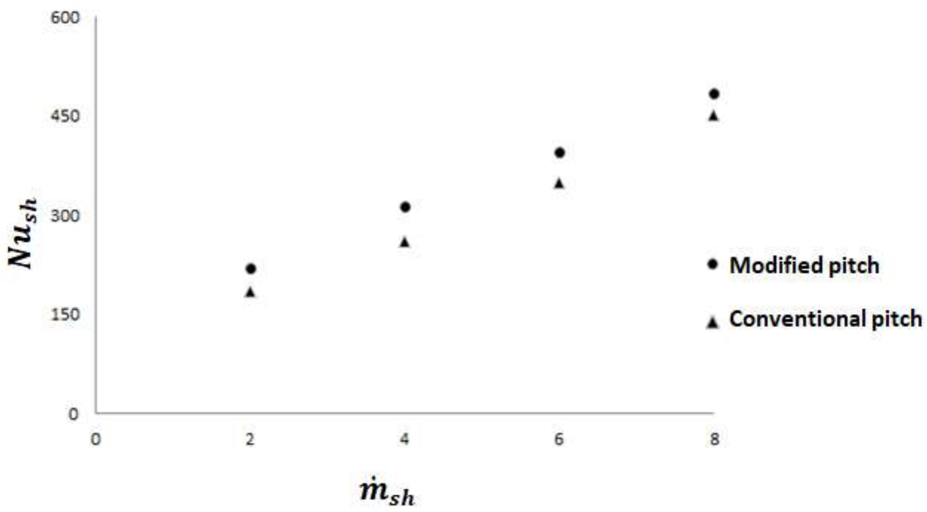

Figure 12: Nusselt numbers of two single-helical coil models.

The Nusselt number in single helical coil heat exchanger with more than one pitch (P-2P-P) was estimated during this study of 150-500 indicating that a good agreement between the current findings and previous papers which are illustrated through the Table 4.

Table 4: A general comparison between this study and similar studies in the literature

Reference	$\dot{m}$	Nu_{sh}
Salimpour [21]	0.013-0.122 kg/s	15-50
Seena [2]	1-8 L/min	100-600
Saravanan [22]	0.3-0.8 Kg/s	30-80
Alimoradi [23]	1-6 L/min	50-400
Alimoradi [24]	1-7 L/min	100-1200
Kumar [25]	1800-2500 L/hours	300-500
Salem [26]	1.7-11.158 L/min	50-600
The present study	1-8 L/min	150-500

5. Conclusions

In the present study, the thermal performance of a single tube coiled helically and the shell of a heat exchanger is improved by using a new coil configuration. The following conclusions can be drawn from the results of this study, which may be a step towards the progress of studies in this field:

- The performance of the modified helical coil was compared with that of the conventional coil using CFD approach to determine the appropriate configuration.
- The modified coil heat exchanger was manufactured taking into account the results of CFD simulation, and was experimentally tested where the numerical analysis showed good agreement with the experimental results, with an error rate of less than 10%.
- The Nusselt numbers for the shell side with variable pitch of P-2P-P were greater than those of the conventional helical tube model and at the Reynolds number range 400<Re_{sh} <2000.
- At last, the pitch correlation and investigation of improvements in the evaluation of single coil heat exchanger can be studied in the future papers.

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