



# Exergy analysis of Shell and helical coil heat exchanger and design optimization

Audai Hussein Al-Abbas<sup>1</sup> · Akeel Abbas Mohammed<sup>1</sup> · Atheer Saleh Hassoon<sup>1</sup>

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## Abstract

In this study, experimental investigations and exergy analysis on shell and helically coiled tube heat exchanger are carried out for free convection heat transfer. The measured data are totally optimised utilizing thermodynamics rules in which exergy study is performed to investigate the thermal performance of the helical system under different operating conditions. The experimental set-up of apparatus are designed and made for cold water and hot water as a working fluid of both the shell side and helical coil side, respectively. The effects of several parameters such as geometry and operational conditions on the exergy destruction and dimensionless exergy destruction are investigated. The counter flow direction is considered under the steady state flow condition, and the critical Reynolds number was more than 4000 in this study. The main objective of this work was to clarify the effect of the volume flow rates and inlet temperatures of hot water and cold water in the shell and helical coil on exergy efficiency and pressure drop. Results showed that the exergy destruction and dimensionless exergy destruction decrease with the increase of coil pitch and Dean number. In contrast, the exergy destruction and dimensionless exergy destruction are obviously increased with the hot water flow rates or cold water flow rates. These exergy characteristics are also augmented with the values of hot water inlet temperatures and cold water inlet temperatures. The pressure drop is considerably increased with the increase of Dean number and reduced with the increase of coil diameter. While, the exergy efficiency steadily increases with the decrease of the cold water flow rates and with the increase of Dean number.

**Keywords** Exergy destruction · Shell and helical coil heat exchanger · Dean number · Optimum design · Exergy efficiency

## Nomenclature

$T$	Temperature (control volume) (K)
$T_0$	Dead state temperature (K)
$P_0$	Dead state pressure (K)
$\rho$	Density of water (kg/m <sup>3</sup> )
$u$	Velocity of water (m/s)

$\mu$	Viscosity (kg/m.s)
$z$	Elevation (m)
$g$	Specific gravity (–)
$h$	Enthalpy of water (kJ/kg)
$s$	Entropy of water (kJ/kg.K)
$\dot{W}_{CV}$	Control volume work (W)
$\dot{Q}$	Flow rate (LPM)
$\dot{m}_h \& \dot{m}_c$	Mass flow rate of hot and cold water (kg/s)
$\dot{\psi}$	Exergy flow (J/kg)
$\dot{S}_{gen.HCHE}$	Entropy generation of helical coil heat exchanger (W/K)
$\dot{\Psi}_{dest.HCHE}$	Exergy destruction of helical coil heat exchanger (W)
$e$	Dimensionless of exergy destruction (–)
$\eta_{exHCHE}$	Exergy efficiency of helical coil heat exchanger (–)
$C_{min}$	Smaller heat capacity rate (W/K)
$C_p$	Specific heat at constant pressure (J/kg.K)

## High Lights

- Experimental work and exergetic analysis of Shell and helical coil tube heat exchanger are investigated
- Geometrical effect and operational parameters on exergy characteristics and pressure drop are studied.
- Exergy efficiency for different amounts of Dean number and inlet temperatures is evaluated.

✉ Audai Hussein Al-Abbas  
aalabbas@atu.edu.iq

<sup>1</sup> Pumps Engineering Department, Al-Musaib Technical College, Al-Furat Al-Awsat Technical University (ATU), Kufa, Iraq

$Re_{in}$	Reynolds number inside helical coil heat exchanger (–)
$D_e$	Dean number (–)
W	Total uncertainty in the measurement
X	Independent variable
1 and 2	Inlet and outlet sections in coil tube
3 and 4	Inlet and outlet sections in shell
h and c	Hot and cold water
$R^+$	Function of the independent variables
$X_1, X_2, \dots, X_n$	Independent variables
$w_1, w_2, \dots, w_n$	Independent variables
$d_{in}$	Inlet diameter (m)
$D_c$	Coil diameter (m)
$\Delta T$	Temperature difference (K)
L	Length of helical coil (m)
N	Number of turns of helical coil
P	Distance between two consecutive coils [ $1.5 * D_c$ ](m)

## 1 Introduction

In industrial applications such as power generation, nuclear industries, food industries, air-conditioning systems, etc., helical coil tube heat exchangers are extensively used in these industries for a reason of higher heat transfer coefficient and condensed structure [1–3]. The curved shape of the helical coil tube significantly leads to flowing the fluid particles under the influence of the centrifugal force. And this flow pattern considerably leads to the generation of opposite rotational flow inside the tube called secondary flow. As a result of this flow configuration, it will lead to the transfer of fluid particles in the form of rotating vortices, which increases the process of transferring heat and a drop in pressure if compared with the straight tubes. However, the effect of the curved tube shape reduces turbulent flow fluctuations generated by fluid flow which leads to an increase in the Reynolds number value and leads to achieve complete turbulence flow [4–6].

The principles of availability and irreversibility in thermodynamic systems have a great role in analyzing and optimizing such complex systems as a helically coiled tube heat exchanger. Therefore, exergy analysis can be implemented for optimization the geometrical parameters and exergetic characteristics in order to increase exergy efficiency of the helical coil heat exchanger [7, 8]. Ebru Kavak Akpınar [9] reported an experimental investigation on the effect of different helical wires configurations for the heat transfer performance and exergy loss. The experimental results showed that the helical number and the pitch number of helically coiled heat exchanger have a great enhancement on the heat transfer rates and pressure drop. Both of the dimensionless numbers of heat transfer units (NTU) and exergy loss increased with the number of

helical wire and decreased with the pitch number. The experimental and theoretical study on the exergy loss (E) and entropy generation of the straight copper micro-fin tube heat exchanger was presented by Paisarn Naphon [10]. The author obtained a reasonable agreement between the measured data and predicted results for all cases of investigation. The temperatures of water, as working fluid, were in ranges of 15–20 °C and 40–50 °C for inlet cold water and inlet hot water temperatures, respectively. The trends of both values for the entropy generation and exergy loss with the hot water mass flow rates were increased and approximately similar. This happened due to the heat capacity ( $m C_p$ ) rates of hot water used. H. Sadighi Dizaji et al. [11] reported an experimental investigation on the effect of geometrical parameters, flow configuration, and thermodynamics characteristics on the total exergy loss and dimensionless exergy loss in shell and coiled tube. The results revealed that the values of the dimensionless exergy loss are dependent on the minimum specific heat capacity of hot and cold water ( $C_{min}$ ). The decrease of the inlet temperature in the shell side led to increase both of the dimensionless exergy loss and exergy loss. On the same scenario of investigation in shell and helical coil in the system, Ashkan Alimoradi [12] studied the effect of flow rates, inlet temperature, coil diameter and pitch number of the helically coiled heat exchanger on the exergy efficiency. The author found that the exergy efficiency decreased with increasing the inlet temperature difference and mass flow rates of fluid.

Recently, numerous researchers have been worked experimentally and numerically toward developing the design of heat exchangers by proposing several techniques for heat transfer enhancement [13–16]. These interesting researches have been done to achieve a higher energy efficiency in this thermal system. M.S. Emani et al. [17] studied experimentally the heat transfer performance under the condition of the laminar flow in rectangular and square channels by inserting various compound techniques. Another beneficial technique by injecting air bubbles in the heat exchanger for improving the thermal efficiency was studied by A. S. Baqir, et al. [18]. M. H. Sharqawy et al. [19] experimentally studied the effect of flow configurations (called axial, radial, and mixed flows) on the spiral-wound heat exchanger. Their investigation showed that the mixed flow configuration provides the higher heat transfer rates and pressure drop in comparison with the other flow patterns used. P. Vocale et al. [20] investigated numerically the effect of thermal boundary conditions on the heat transfer characteristics in the helically coiled tube heat exchanger. They found that the thermal boundary conditions might extensively amend the values of the convective heat transfer coefficient based on the local distribution of the Prandtl number. A. Sheeba, R. Akhil and M. J.

Prakash [21] numerically and experimentally investigated the heat transfer coefficient and fluid flow characteristics under fully developed laminar flow for the conically coiled double tube heat exchanger. They found that the Dean number has considerable effect on the Nusselt number in the annulus tube, and the use of conical heat exchanger has a significant improvement on the heat transfer coefficient in compared with the helical one.

The literature review has a small number of research regarding the experimental study and exergy analysis for helically coiled tube heat exchangers (HCHE). The present study aims to provide a good understanding in the exergy characteristics and exergy efficiency under different parameters. The main objective of this study was to perform the effect of several important parameters and exergy characteristics on exergy destruction and dimensionless exergy destruction. Different parameters are taken into consideration of experiments such as the thermo-physics, fluid flow, and geometrical parameters. The experiments is carried out at different values of Reynolds number. The hot and cold water volume flow rates, hot and cold water inlet temperatures, and geometrical parameters (coil diameter and coil pitch) are experimentally investigated for the shell and helical coil tube heat exchanger. The exergy efficiency and pressure drop are also investigated and presented in this study.

## 2 Exergy analysis

Exergy is a thermodynamics concept which can be defined as the maximum useful work that can be achieved from the reversible system in a specified environment. The obtained data from a shell and helical coil heat exchanger used for exergy analysis are considered based on developed energy equations. Therefore, no change in energy, entropy and exergy of the system with the time is taken into consideration of calculations in mathematical models. The steady state condition is assumed, and the mass flow rate enter equals that leaving component of the system, as shown in Fig. 1.

The exergy balance equation can be written as [22]:

$$\frac{d\psi_{sys}}{dt} = \sum \dot{Q}_{CV} \cdot \left(1 - \frac{T_0}{T}\right) - \sum \dot{W}_{CV} + P_0 \frac{dV}{dt} + \sum \dot{m}_i \cdot \psi_i - \sum \dot{m}_e \cdot \psi_e - T_0 \cdot \dot{S}_{gen}. \quad (1)$$

In this study, the heat transfer process of helical coil heat exchanger is an adiabatic process, and it can be assumed that the amount of hot fluid is equal with the amount of cold fluid. Exergy destruction in a steady control volume with two different types of working fluids can be written as follows:

$$\dot{\psi}_{in} - \dot{\psi}_{out} - \dot{\psi}_{dest} = \frac{d\psi_{sys}}{dt} \quad (2)$$

Where the symbol  $\psi$  representing a flow exergy and can be expressed as:

$$\psi = (h-h_0) - T_0(s-s_0) + g(z-z_0) + \frac{u^2}{2} \quad (3)$$

Exergy flow for hot water in inlet and outlet are given, as follows:

$$\dot{\psi}_{in} = (h_1-h_0) - T_0(s_1-s_0) \quad (4)$$

$$\dot{\psi}_{out} = (h_2-h_0) - T_0(s_2-s_0) \quad (5)$$

Exergy flow for cold water in inlet and outlet are given, as follows:

$$\dot{\psi}_{in} = (h_3-h_0) - T_0(s_3-s_0) \quad (6)$$

$$\dot{\psi}_{out} = (h_4-h_0) - T_0(s_4-s_0) \quad (7)$$

Now, exergy destruction for HCHE is:

$$\dot{\psi}_{dest,HCHE} = \dot{m}_h(\psi_1-\psi_2) + \dot{m}_c(\psi_3-\psi_4) \quad (8)$$

Or

$$\dot{\Psi}_{dest,HCHE} = T_0 \dot{S}_{gen,HCHE}. \quad (9)$$

Applying entropy balance to get on Entropy generation in HCHE is:

$$\dot{S}_{gen,HCHE} = \dot{m}_h(s_2-s_1) + \dot{m}_c(s_4-s_3) \quad (10)$$

Where  $s_1$  and  $s_2$  represent the entropy of hot water (kJ/kg. K) in the coil tube sections 1 and 2, respectively; and  $s_3$  and  $s_4$  are the entropy of cold water (kJ/kg.K) in the shell sections 3 and 4, respectively.  $\dot{m}_h$  and  $\dot{m}_c$  are the mass flow rates for the hot and cold water, respectively. The entropy change ( $s_{out} - s_{in}$ ) of liquids (water) can be evaluated from Eq. (11), and the specific heat capacity is kept constant.

$$(s_{out} - s_{in}) = C_p \ln \left( \frac{T_{out}}{T_{in}} \right) \quad (11)$$

The exergy efficiency of the HCHE can be defined as an increase in the exergy of cold stream divided by the decrease in the exergy of the hot stream [23], as presented in the following equation:

$$\eta_{ex,HCHE} = \frac{\dot{m}_c(\psi_4-\psi_3)}{\dot{m}_h(\psi_1-\psi_2)} \quad (12)$$

Or

$$\eta_{ex,HCHE} = 1 - \frac{\dot{\Psi}_{dest,HCHE}}{\dot{m}_h} (\psi_1-\psi_2) \quad (13)$$

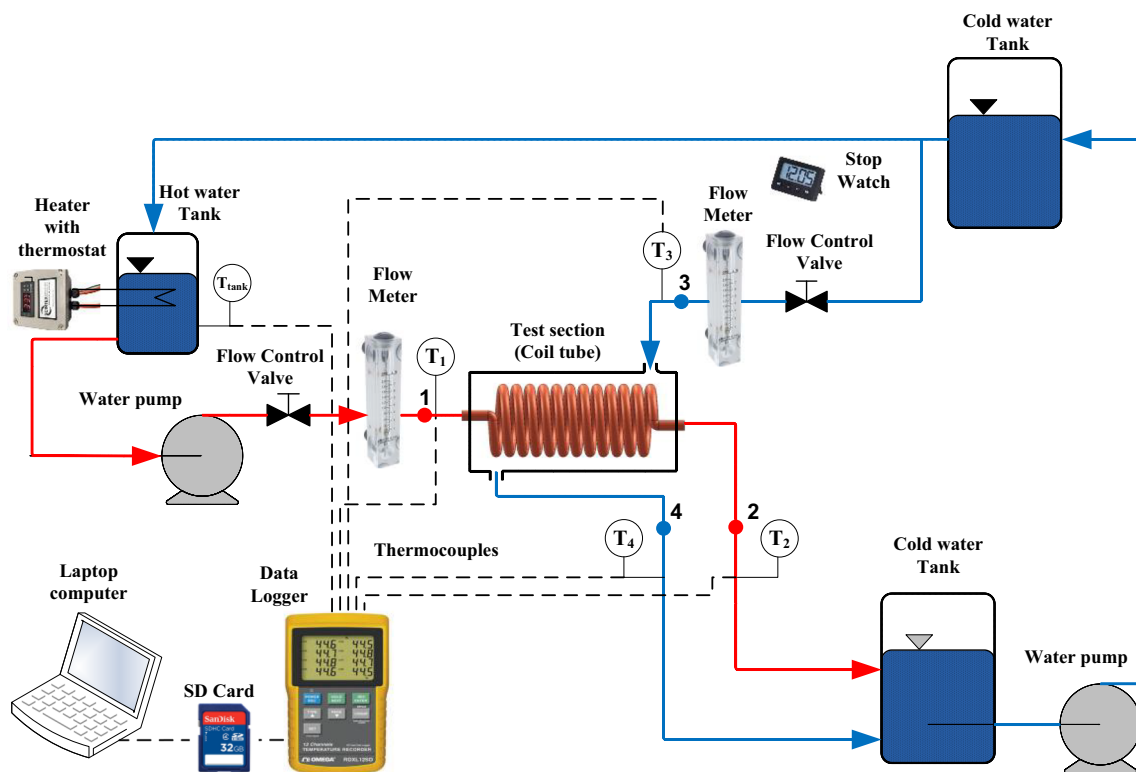


Fig. 1 Schematic diagram of the experimental setup

Finally, dimensionless of exergy destruction can be calculated as follows [11]:

$$e = \frac{\Psi_{dest.HCHE}}{\dot{m}_h} (\psi_1 - \psi_2) \tag{14}$$

The change in exergy flow for hot water (assuming kinetic and potential energy are negligible) is evaluated by:

$$(\psi_1 - \psi_2) = (h_{out} - h_{in}) - T_0 (s_{out} - s_{in}) \tag{15}$$

Where the change in enthalpy for hot fluid is:

$$(h_{out} - h_{in}) = C_p \Delta T \tag{16}$$

Substituting Eq. (11) and Eq. (16) in Eq. (15) and then Eq. (15) in Eq. (14) we get:

$$e = \frac{\Psi_{dest.HCHE}}{\dot{m}_h} C_p \left[ \Delta T - T_o \ln \left( \frac{T_{h,out}}{T_{h,in}} \right) \right] \tag{17}$$

Another definition for dimensionless of exergy destruction can be written as [24]:

$$e = \frac{\Psi_{dest.HCHE}}{T_o C_{min}} \tag{18}$$

Where :  $C_{min} = \text{Min} \left\{ Ch = \dot{m}_h C_p \text{ and } Cc = \dot{m}_c C_p \right\}$   $\tag{19}$

### 3 Experimental investigation and procedure

#### 3.1 Experimental set-up

The experimental set-up for the HCHE is shown in Fig. 1. In this experimental work, the cold water is firstly taken from the upper water tank to pass through the shell side of HCHE at point 3 and departs the helical coil tube at point 4 based on the siphon effect. While, the hot water is pumped from the electric water heater tank and passed through the HCHE at point 1 and departs the coil tube at point 2. Flow meters (model number: LZM-Z) and flow control valves are set up in flow lines to control the water flow rates. Additional water tank with electric water heater (800 W) is used for the hot water source. The power of the electrical heater is adjusted by thermostat. Five thermocouples (type-K) with data logger (model BTM-4208SD) are used in the present study for measuring the water temperatures.

Under the steady state condition, the inlet and outlet water temperatures are precisely recorded through the data logger at each measuring point. After measuring the required temperatures, the cold water is pumped from the bottom cold tank to the upper cold water tank. This is continually making a water loop in order to be ready for the next test, as illustrated in Fig.1. The tests are generally carried out based on three different parameters: flow characteristics, thermodynamic properties, and geometry. For that reason, in each test the selected operating conditions are practically adjusted according to the

design of helical coil tube examined. The calibration of the flow meters is implemented using the measuring container and digital stopwatch. For each test, the initial thermal conditions were attained. The experimental work was conducted under the counter flow direction at different values of Reynolds numbers. The specifications and geometrical details for the test section (HCHE) are mentioned in Table 1.

### 3.2 Tests procedure

The thermal and hydraulic performances of shell and helical coil heat exchanger were examined under different conditions in this study. Many key variables and parameters are taken into consideration of experiments such as the fluid flow, thermodynamics, and helix geometry parameters, as shown in Table 2. As seen in Table 2, the effect of flow characteristics, thermodynamics properties, and geometry are studied under the counter flow direction in order to improve the heat transfer processes between the hot water in the HCHE and cold water in the shell. Also, it can be noted that the volume flow rates were limited to be less than 6 (L/min.) and the critical Reynolds number is more than 4000 for all examined cases due to the high pressure drop produced in the helical coil tube. However, for each key parameter was taken into consideration of this experimental study the other variables were set as constant. For illustration, the variables that affect the performance of the HCHE are:

- The effect of hot water inlet temperature through the HCHE on heat exchanger performance was studied in the range of 57 to 60 °C.
- The effect of cold water inlet temperature through the HCHE on heat exchanger performance was studied in the range of 27 to 31 °C.
- The effect of volume flow rate of water through the HCHE on heat exchanger performance was studied in the range of 2.5 to 5 L/min.
- The effect of coil diameter and coil pitch of HCHE on the heat exchanger performance was studied in the range of 7 to 24 cm and 3 to 11 cm, respectively.

**Table 1** Specifications and detailed geometrical information of the HCHE

Geometrical description	Values
Number of turns (N)	12
spacing between consecutive coil turns (P)	3 cm
Outer diameter of tube ( $d_{out}$ )	1.5 cm
Inner diameter of tube ( $d_{in}$ )	1.2 mm
Diameter of helix (coil) ( $D_c$ )	12.2 cm
Heigh of coil ( $H_c$ )	40 cm
diameter of shell ( $D_{sh}$ )	220 cm
Height of shell ( $H_{sh}$ )	52 cm

For all key variables mentioned above, the test procedures are carried out with ambient temperature of 25 °C. Furthermore, the inlet and outlet water temperatures and volume flow rates were accurately measured every 10 min by using data logger and flow meter, respectively. Based on the experience, the suction and discharge pressures in this investigated thermal system are no longer changed to ensure more accurate experimental results achieved under steady state conditions.

### 3.3 Measurement uncertainty

In order to estimate the measurement uncertainties of the experimental data results, several dependent and independent variables such as exergy destruction, dimensionless exergy destruction, and Reynold number ( $Re_{in} = \left(\frac{\rho u d_m}{\mu}\right)$ ), which are combined with Eqs. (8) and (14) can be taken into consideration of calculations the uncertainties for every independent variables. The uncertainties of these variables ( $W_{R^+}$ ) can be evaluated, as follows [25]:

$$W_{R^+} = \sqrt{\left(\frac{\delta R^+}{\delta X_1} w_1\right)^2 + \left(\frac{\delta R^+}{\delta X_2} w_2\right)^2 + \dots + \left(\frac{\delta R^+}{\delta X_n} w_n\right)^2} \quad (20)$$

Where  $R^+$  is a function of the independent variables  $X_1, X_2, \dots, X_n$ , and  $w_1, w_2, \dots, w_n$  are the independent variable uncertainties. The detailed information of uncertainty results of this study are listed in Table 3.

## 4 Results and discussion

### 4.1 Exergy destruction

#### 4.1.1 The effect of hot and cold water flow rates on exergy destruction

Prior to carrying out the measurement procedures in this experimental investigation, the hot and cold water inlet temperatures are kept constant at about 57 °C and 27 °C, respectively. The effects of hot and cold water volume flow rates on exergy destruction are carried out with a range of the Dean number ( $De = Re_{in} \sqrt{d_{in}/D_c}$ ). The experimental results of the effects of hot and cold water flow rates on both exergy destruction and dimensionless exergy destruction are shown in Figs. 2 and 3. As presented in Fig. 2, by raising the amount of hot water flow rates or cold water flow rates, the amount of exergy destruction increases. Certainly, the amount of exergy destruction enhanced due to the augmentation of cold water flow rates. The smallest and highest rates of exergy destruction are achieved for 2.5LPM (liter per minute) and 5LPM,



**Table 2** Information for tested conditions

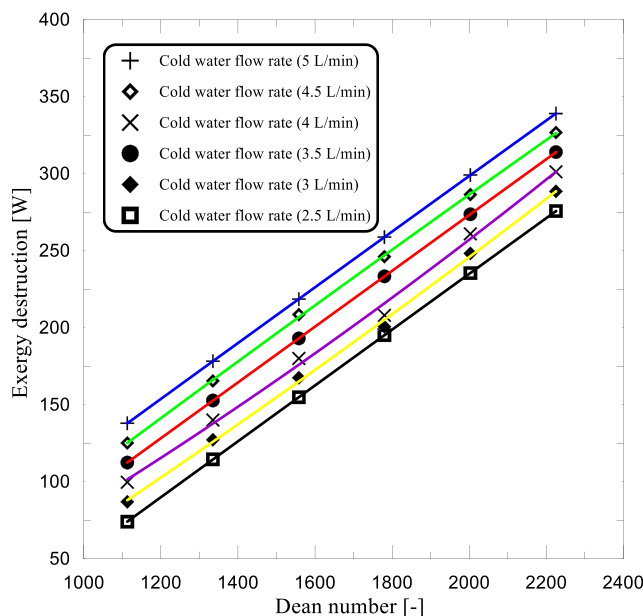
Tested conditions	Flow parameters		Thermodynamics parameters		Helix geometry parameters	
	Q <sub>out</sub> (L/min)	Q <sub>in</sub> (L/min)	T <sub>c,in</sub> °C	T <sub>h,in</sub> °C	diameter (cm)	Pitch (cm)
Effect of flow characteristics	2.5	(2.5–5)	27	58	12.2	3
	3	(2.5–5)	27	58	12.2	3
	3.5	(2.5–5)	27	58	12.2	3
	4	(2.5–5)	27	58	12.2	3
	4.5	(2.5–5)	27	58	12.2	3
Effect of Thermodynamic properties	3	3	27	57–60	12.2	3
	3	3	29	57–61	12.2	3
	3	3	31	57–62	12.2	3
Effect of geometry	3	3	27	58	7	(3–9)
	3	3	27	58	14	(3–9)
	3	3	27	58	24	(3–9)

respectively. Additionally, for lower value of hot water flow rate which is represented in the Dean number, for instance  $De = 1113$ , cold water flow rate has less significant effect on exergy destruction in compared with higher value of Dean number ( $De = 2225$ ). Exergy destruction enhancement might be attained due to the augmentation of volume flow rate. However, This can be explained because additional values of hot or cold water flow rates can be resulted in more values of Reynolds number (Dean number) during the HCHE. And evidently, higher value of Reynolds number led considerably higher rate of the heat transfer process. By using other words, more temperature difference ( $\Delta T$ ), which is the most reasonable cause of exergy destruction, occurred. This aspect for helical coiled tubes is more substantial due to the secondary flows generated which are because of the centrifugal forces in compared with straight tubes, as described clearly in Ref. [11]. As a result, this flow configuration enforces the water toward the outer wall of tube and then causes additional value of exergy destruction in the HCHE.

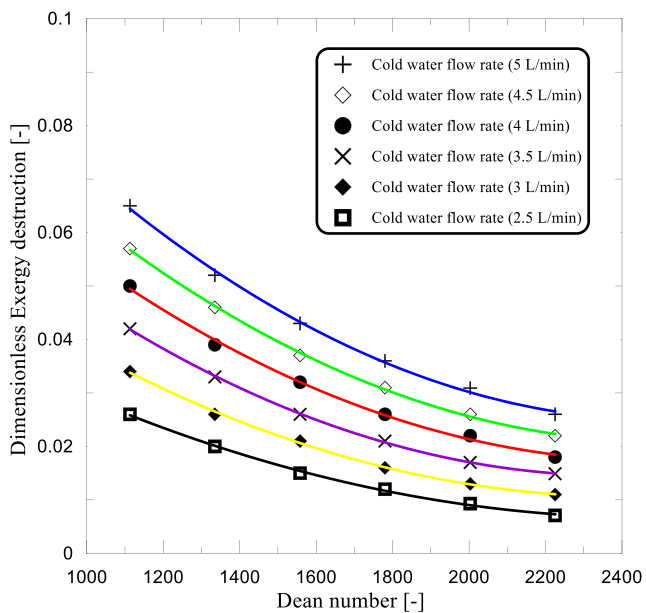
**Table 3.** Uncertainty of parameters for experimental work

Independent variables	Variable errors
Data Logger(Temperature readers)	$\mp 0.7$
Hot Side Temperature (°C)	$\mp 0.5$
Cold Side Temperature (°C)	$\mp 0.5$
Hot Side Mass flow rate (LPM)	$\mp 0.02$
Cold Side Mass flow rate (LPM)	$\mp 0.02$
Dependent variables	Variable errors
Exergy Destruction (W)	$\mp 1.44$
ExergyEfficiency (%)	$\mp 0.039$

As seen in Fig. 3, the amount of dimensionless exergy destruction ( $\epsilon$ ) reduces with the increase of hot water flow rate, and on the other hand it enhances with the increase of cold water flow rate. The explanation of this phenomenon is associated with Eq. (14). In fact, augmentation of hot water flow rate results in enhancement of both numerator and denominator of Eq. (14). Nevertheless, the amount of  $(T_o C_{min})$  in Eq. (18) is not associated with the cold water flow rate. For this reason, augmentation of cold water flow rate enhances the dimensionless exergy destruction. Furthermore the augmentation of hot water flow rate reduces the dimension less exergy



**Fig. 2** Effect of hot and cold water flow rates on Exergy destruction in the HCHE

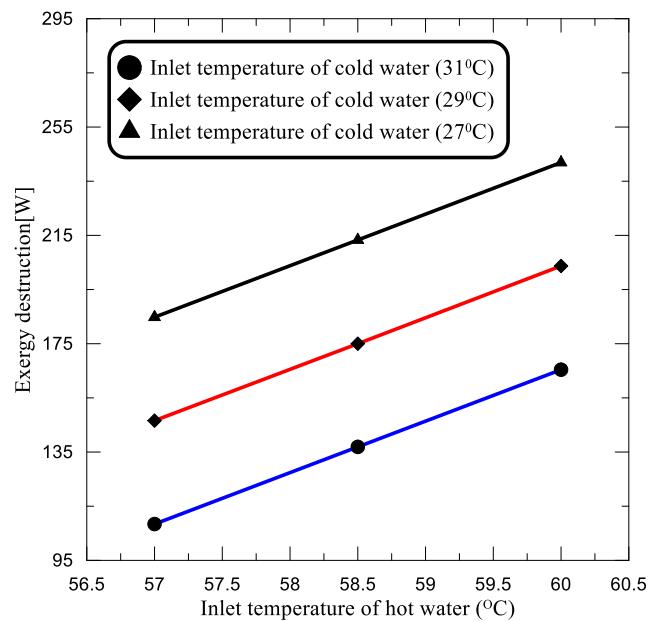


**Fig. 3** Effect of hot and cold water flow rates on dimensionless exergy destruction in the HCHE

destruction. However, for all tested cases of the cold water flow rates, the tangent lines (slopes) of dimensionless exergy destruction curves are approximately comparable. However, these results were highly confirmed with the Ref. [24].

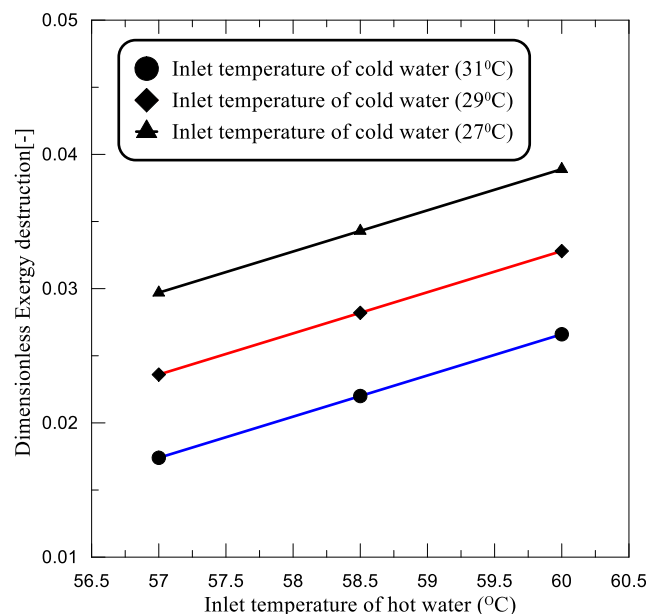
#### 4.1.2 The effect of hot and cold water inlet temperatures on exergy destruction

In the present investigation, the flow rates and geometrical parameters are kept constant, and the cold water and hot water inlet temperatures were varied in ranges of 27–31°C and 57–62°C, respectively. In general, Prandtl number is a prospective pointer of specific heat capacity, viscosity, and thermal conductivity as the most important thermodynamic properties of the working fluid. For this reason, hot and cold water inlet temperatures are changed so as to obtain different values of water Prandtl number; and the effects of water inlet temperatures on exergy destruction are estimated. In Figs. 4 and 5, the effect of inlet temperatures on exergy destruction and dimensionless exergy destruction are presented, respectively. It can be clearly seen in two figures that both of exergy destruction and dimensionless exergy destruction are obviously increased with the increase of hot water inlet temperatures and cold water inlet temperatures. As generally observed, augmentation of inlet temperature difference ( $\Delta T$ ) between hot water and cold water increases both exergy destruction and dimensionless exergy destruction. In fact, the explanation of this result is basically associated with exergy destruction due to inlet temperature difference which is the



**Fig. 4** Effect of hot and cold water inlet temperatures on exergy destruction in the HCHE

most important cause of exergy destruction in heat exchangers. For the dimensionless exergy destruction, it is supposed to observe that the value of  $(T_o C_{min})$  in Eq. (18) is constant in this fraction. Consequently, dimensionless exergy destruction ( $\epsilon$ ) enhances with the increase of  $\Psi_{dest. HCHE}$  in Eq. (18). While augmentation of hot water inlet temperature improves dimensionless exergy destruction because the specific heat capacity of hot water is higher than that of the cold water.



**Fig. 5** Effect of hot and cold water inlet temperatures on dimensionless exergy destruction in the HCHE

### 4.1.3 The effect of geometrical parameters on exergy destruction

In Figs. 6 and 7, the effects of coil pitch and coil diameter on the exergy destruction and dimensionless exergy destruction in the HCHE are presented. The fluid flow and thermodynamic properties are maintained constant, as seen in Table 2. It can be seen that the number of turns of coil is alike for the helically coiled tube heat exchanger with different diameters of coil. For this reason, the HCHE with higher value of coil diameter possesses noticeable added length in the tube based on the following mathematical relation:  $(L = N\sqrt{(\pi D_c)^2 + P^2})$ . Where: L is the length of helical coil, N equals the number of turns of helical coil, D<sub>c</sub> is the coil diameter, and P represents the distance between two consecutive coils [1.5\* D<sub>c</sub>]. Therefore, additional length in tube leads to further time for hot water, and that is caused higher heat transfer rate and further exergy destruction, as clearly showed in Fig. 6. Also, the effect of coil pitch on thermal properties and exergetic characteristics is actually insignificant. On the other hand, smaller values of coil pitch produces a powerful secondary flow which enhances exergy destruction during the HCHE. The dimensionless exergy destruction reduces as the coil diameter increases. The reason of that it can be easily found in Eq. (17). From the interesting point of investigation, the value of temperature difference (ΔT) in Eq. (17) augments as the coil diameter increases, and that is due to the additional length in tube and leads as a result to optimal heat transfer rate. Consequently, the value  $(\dot{m}_h C_p [\Delta T - T_o \ln(\frac{T_{h,out}}{T_{h,in}})])$  in Eq. (17) is increased and dimensionless exergy destruction is

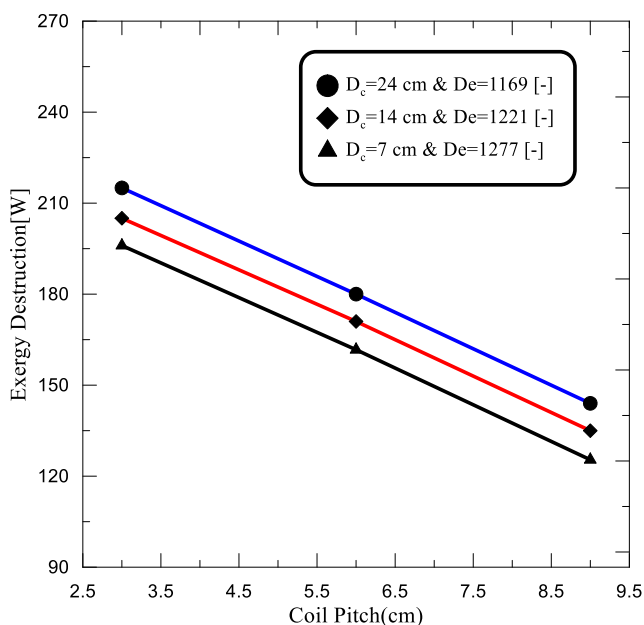


Fig. 6 Effect of coil pitch and coil diameter on exergy destruction in the HCHE

definitely reduced. Also, it can be seen in Figs. 6 and 8 the values of flow rates are similar for all studied cases, although each investigated case produces various values of Dean number. For verification, these investigated results regarding the exergy analysis for the coil diameter and coil pitch are also confirmed with the similar work of A. Alimoradi [12].

As far as the pressure drop is concerned in the HCHE, the effect of geometrical and operational parameters on the pressure drop is also presented in this study for optimum design. The pressure drop is generally dependent on several independent variables such as fluid properties, fluid flow rates, and helical coil geometry. For that reason, the effect of coil diameter on the pressure drop inside the helical tube under a range of water flow rates is presented in Fig. 8. As seen in Fig. 8, the pressure drop is increased with the augmentation of the water flow rates. But, it is clearly decreased with the increase of helical coil diameters. However, this behavior is occurred due to the change of the coil geometry, which led to increase velocity of fluid; and consequently affected to increase the Reynolds number (Re) and Dean number (De), as mentioned early in subsection 4.1.1. This means that with higher values of De the loss of pressure increases. Both of the pressure drop and Dean number values are increased from 1.858 to 14.07 Pa and from 984.8 to 1970, respectively. These findings are confirmed with similar experiment [25] and exergy works [26].

### 4.2 Exergy efficiency

From the interesting point of investigation, good understanding of the thermal and hydraulic characteristics of this type of heat exchanger is very important to improve the thermal performance and exergy efficiency. In this study, exergy

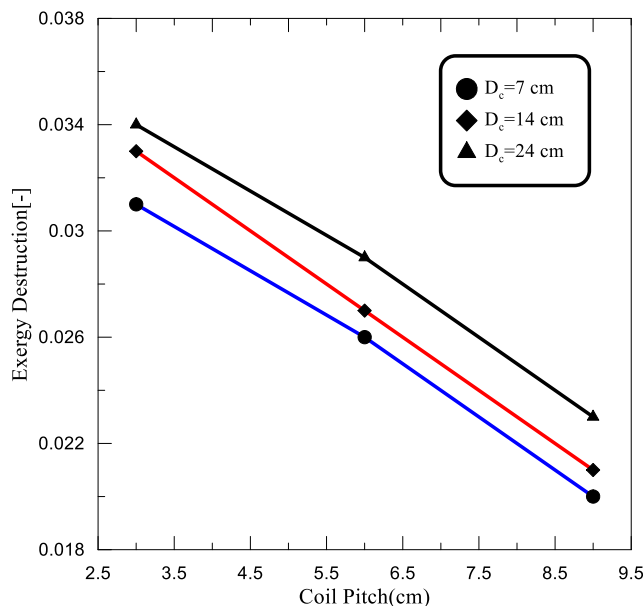
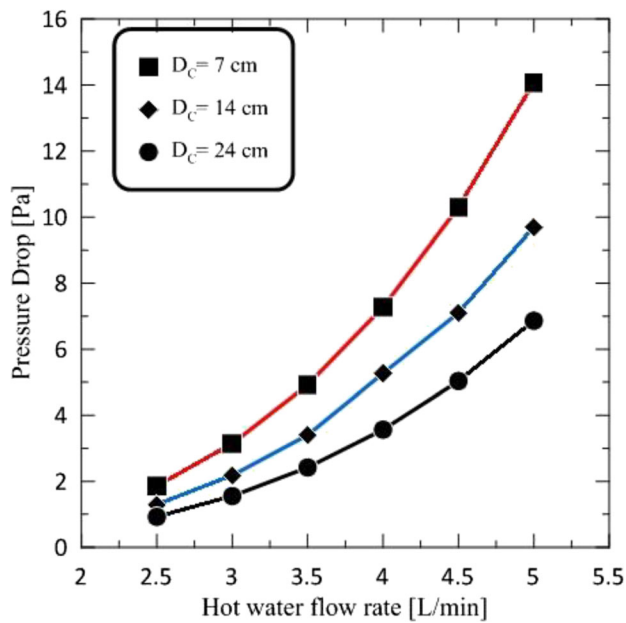
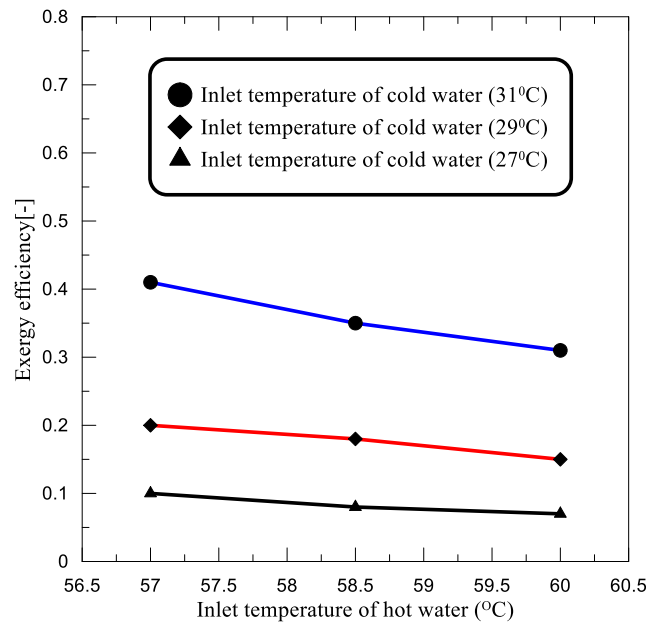


Fig. 7 Effect of coil pitch and coil diameter on dimensionless exergy destruction in the HCHE



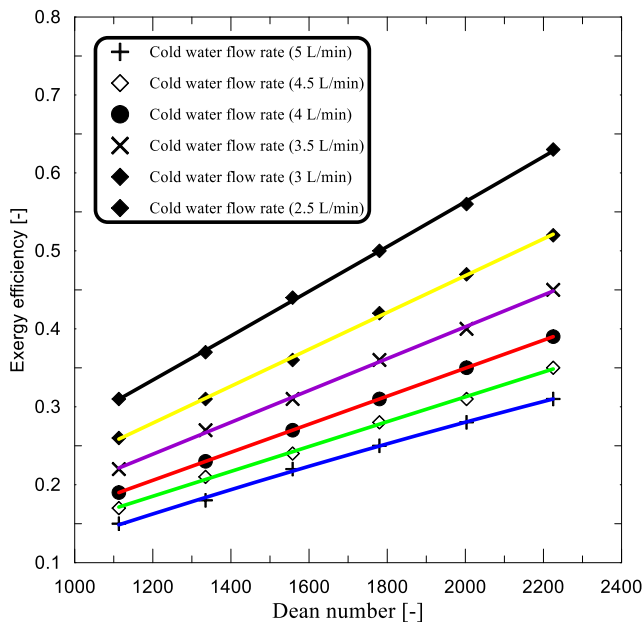


**Fig. 8** Effect of coil diameter and volume flow rate on pressure drop in the HCHE



**Fig. 10** Effect of hot and cold water inlet temperatures on exergy efficiency in the HCHE

efficiency was evaluated for different volume flow rates and different inlet temperatures, as presented in Figs. 9 and 10. As seen in Fig. 9, the value of the exergy efficiency raises when both of the hot water flow rate (represented implicitly in Dean number) and the cold water inlet temperatures increase. Moreover, it is somewhat reduced with the augmentation of cold water flow rates (as presented in Fig. 9) and hot water inlet temperature (as presented in Fig. 10). Obviously, it can be seen that, even though the augmentation of cold water flow



**Fig. 9** Effect of hot and cold water flow rates on exergy efficiency in the HCHE

rate increases the value of the numerator of Eq. (12); but, the exergy efficiency is reduced because of the reduction of the cold water outlet temperature and hot water outlet temperature. Also, it can be concluded from Eq. (12) that the exergy efficiency is increased with the decrease of the inlet temperature difference between the cold water and hot water. However, these results of the present work are confirmed with the similar results of Dizaji et al. [11]. For the same assumptions of the experiment procedures of the fluid flow and thermodynamics properties, they concluded that the exergy efficiency is decreased with the increase of hot water inlet temperatures in the coil side and with the decrease of cold water inlet temperatures in the shell side. That is particularly happened due to the increase in the amounts of both the exergy destruction and dimensionless exergy destruction in the helical tube heat exchanger. Finally, this exergy analysis study might be provided a good information for the exergy characteristics in the helically coiled tube heat exchanger and its optimal design, particularly in the enhancement of exergy destruction ( $\psi_{dest, HCHE}$ ) and dimensionless exergy destruction under different operating BCs. In short, the results considerably showed that the geometrical and operating parameters have a great importance on the exergy efficiency of the helical system.

## 5 Conclusion

The experimental investigations and exergy analysis for the shell and helical coil tube heat exchange under free convection heat transfer are carried out in the present study. For steady

state condition, the tests are conducted according to three different parameters: fluid flow, thermodynamic properties, and geometry. The present paper describes the effects of aforementioned parameters on the exergy characteristics, exergy efficiency, and pressure drop. For the geometrical investigation, the exergy destruction and dimensionless exergy destruction ( $\epsilon$ ) are increased with the increase of pitch diameter and reduced with the increase of pitch coil. For the effect of inlet water temperatures, the amounts of exergy destruction and dimensionless exergy destruction ( $\epsilon$ ) are clearly increased with the increase of the inlet temperatures of the hot and cold water; and that is happened due to  $C_{\min}$ . For water flow effect, the exergy destruction ( $\dot{W}_{dest. HCHE}$ ) is obviously increased by augmenting the amount of hot water flow rates or cold water flow rates. That is achieved due to the augmentation of Dean number through the HCHE. The dimensionless exergy destruction ( $\epsilon$ ) is also enhanced by augmenting cold water flow rates (shell side). Whilst, the augmentation of hot water flow rates (coil side) reduces the dimensionless exergy destruction. That is dependent on specific heat capacities of the hot and cold water. However, it is noted that the exergy efficiency ( $\eta_{exHCHE}$ ) is linearly increased with the increase of Dean number and with the reduction of volume flow rates of cold water. A clear reduction in pressure drop is noted with the increase of coil diameter. Finally, this study is significantly affected the thermal and hydraulic characteristics of the HCHE. However, an air injection technique with using nano fluids is a promising technique to improve the thermal performance of the shell and helical coiled tube heat exchanger, as a continuous research in this issue of heat exchanger.

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