A New Finned Tube Helical Coil Heat Exchanger an Experimental Study

By

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HIGHLIGHTS

1. A new vertical helical heat exchanger was designed.
2. Fins with a triangular shape were added on the outside surface of the tube.
3. A comparison study (heat transfer rate, effectiveness, and thermal performance factor) between the plain and finned vertical helical tube heat exchanger was done.
4. All obtained results for the finned tube helical heat exchanger were promising and superior to the unfinned one.
5. The study of finned helical coil tube heat exchanger shows better heat transfer compared to an enhancement with nanofluid.

ABSTRACT

The performance of the triangular finned vertical helical coil heat exchanger experimentally and mathematically was investigated. Hot water flows inside the helical tube at 60 °C while cold water enters the shell at 30 °C counter currently. A triangle copper fin with 1 cm height, 2 cm base length, and 0.3 cm thickness at 4 cm pitch distance was welded at the external surface of the helical copper coil tube. Along the coiled tube123 fins were welded on the outside surface of the helical coil tube heat exchanger.

The results showed a good enhancement ratio compared to the unfinned one. The heat enhancement was 12 % with effectiveness 11 %. The improvement in the Nusselt number was 16.5 %. These results confirm that the added fins to the helical coil heat exchanger modified and improve the helical heat exchanger which is suitable for the industrial design with better performance.

Keywords: vertical helical heat exchanger, finned tube, triangular fin, performance, heat transfer coefficient

1- Introduction

The helical tube heat exchanger has many applications such as cooling process, chemical reactors, marine cooling systems, air-condition, and heating systems, etc. **Vinous and Ahmed [1]** investigated the temperature distribution through a horizontal helical coil heat exchanger experimentally and numerically. The copper tube formed helically on a 90 mm diameter immersed in a Perspex shell of 1 m length and 150 mm internal diameter. The water used in the shell and the tube at 65°C and 35°C respectively. Four different mass flow rate selected for the shell and tube sides 6, 8, 10, and 12 L/min. The experimental results showed good agreement with the numerical one. **Mhaske and Palande [2]** studied the experimentally and numerically helical tube heat exchanger which was wounded with the wire at the inner tube and compare it with a conventional heat exchanger. One phase system at counter flow type was used. The hot water at 60 ^{o}C comes through the inner tube while the cold water at 30 ^{o}C enters the outer tube. The water flow rate ranged from 480 to 1200 LPH. The results showed that the tube which wounded with wire enhances the heat more than the conventional helical coil heat exchanger. **Ammar and Ali [3]** investigated experimentally and numerically the heat transfer coefficient of a double-helical coiled tube heat exchanger. The water was the working fluid in the system for both parallel and counter flow type. Hot water temperature ranged from 40 ^{o}C to 70 ^{o}C which enters the tube at a constant mass flow rate of 0.0724 kg/s. While the cold water enters through the shell at temperature (20 to 40 ^{o}C) with mass flow rate varies from 0.032 to 0.0721 kg/s. The mass flow rate effect on axial temperature distribution was investigated. Also proved counter flow type is more effective than parallel type. **Nada et al. [4]** investigated the characteristics and performance of the helical coil tube with and without fins. The test is applied on four different shell diameters of (10, 15, 20, and 25 cm). Two-phase is used in the test, the water enters the shell and the refrigerant enters the coiled tube. The tube is from Aluminum material of 8 mm in internal diameter and 0.5 mm in thickness. The obtained results showed an increase in Nusselt number, Reynolds and Grashof number over the unfinned tube case. **Vinous and Bashar [5]** studied the performance of a triangular finned tube heat exchanger experimentally and numerically. Two meters copper tube was used with a 20 mm inner diameter. Copper fins in a triangular shape were added on the tube outside the surface. The fin dimensions were 1 cm in length, 1 cm in height, and 1 mm thickness. Perspex shell was used with a 54 mm internal diameter. The water and air are the working fluid in the system. Cold air flows through the shell at a mass flow rate ranged from 0.001875 to 0.003133 kg/s while hot water enters through the tube side. The performance of the finned and unfinned heat exchanger was compared. The experimental results showed that the finned tube heat exchanger enhances the heat transfer rate (3.252 to 4.502) times than the unfinned tube. The numerical results have a good agreement with experimental results. **Viraj et al**. **[6]** studied the experimental analysis of helical coil heat exchanger with and without fins. A copper helical tube of 7.96 mm outer diameter and 0.81 mm thickness was used in the work. Circular fins from copper material brazed on the external surface of 24 mm external diameter, 7.96mm internal diameter, and 2 mm width. The helical coil immersed in the insulated shell with 500 mm in length. The coil diameter is 0.21 m and the pitch is 0.03 m. the hot and cold water is entered at both the coil and the shell sides by centrifugal pumps, respectively. The obtained results declared the increase in the hot flow rate will increase the heat exchanger effectiveness. The helical coil with fins will enhance the heat more than the non-finned coil due to increasing the effective tube heat transfer surface area. **Karima Amori [7]** presented an experimental study of a helical coil heat exchanger immersed in cold water. Single and triple-helical coiled heat exchangers were used. Hot water enters the tube at a flow rate ranged from (2.67 to 7.08 l/min). The inlet temperatures are (50 60 70 and 80 "C). The results show that the triple helical coil heat exchanger is better than a single helical coil heat exchanger which was a result of enhancement in heat transfer and effectiveness. While lower pressure drop values were obtained in the first one. **Ali et al., [8]** made a comparison between pure water with Nanofluid TIO2 at a volume concentration of 0.1 and 0.2 % in a vertical helical heat exchanger. Copper tube with (0.5 mm external diameter, 9 mm internal diameter, and 93mm coil diameter was used. The total turns of the coil were 13 turns with a 17 mm pitch distance. The helical coil was immersed in a steel shell. The mass flow rate in the shell kept constant at 15 kg/s while varied at the helical tube. The results show that using Nanofluid presents good results as compared to pure water. Nanofluid increasing the thermal conductivity of the used fluid causing an increase of system heat transfer rate.

The main object of any operating heat exchanger was to increase the amount of heat exchange between the two fluids in the heat exchanger. This object was investigated by designing a new triangular finned coiled tube heat exchanger. The performance and the amount of heat transfer enhancement of the new finned coiled heat exchanger were studied and investigated. A comparison with the traditional helical tube heat exchanger without an enhancement was also done.

2. Experimental work

A new type of vertical helical coil heat exchanger enhancement was designed. The test rig is divided into two parts, main parts, and the auxiliary parts. The main parts are:-

1- Perspex shell: 100 cm shell with two ends flanges and 13.5 cm internal diameter as shown in the following figure.

Figure 1: Perspex shell

2- Copper tube: copper material with purity 99.9 % and dimensions of (1.27 cm) external diameter and (0.71 mm) in thickness. The total length of the tube is 789 cm. The tube turned diameter was 7 cm with 3.2 cm pitch distance. The total number of turns was 25. So, the test section length was 80 cm with 15 cm in the region.

Two tube shape cases were done. The first case was done for a normal vertical helical coiled tube as shown in Fig. 2.

Figure 2. Helical tube without enhancement

While in the second case a passive technique enhancement on the external surface of the coil tube was done as shown in Fig. 3.

Figure 3 Finned helical coil tube

A triangular copper fins with a curved base in order to match the coil tube outside surface without gapping were welded. The dimensions of each fin were 2 cm in the base, 1 cm in the height, 0.3 mm in the thickness as shown in Fig. 4. The pitch distance between every two successive fins was 4cm. The total fins on the outside coiled tube surface would be 123 fins as seen in Fig. 4.

Figure 4. Triangular fin picture

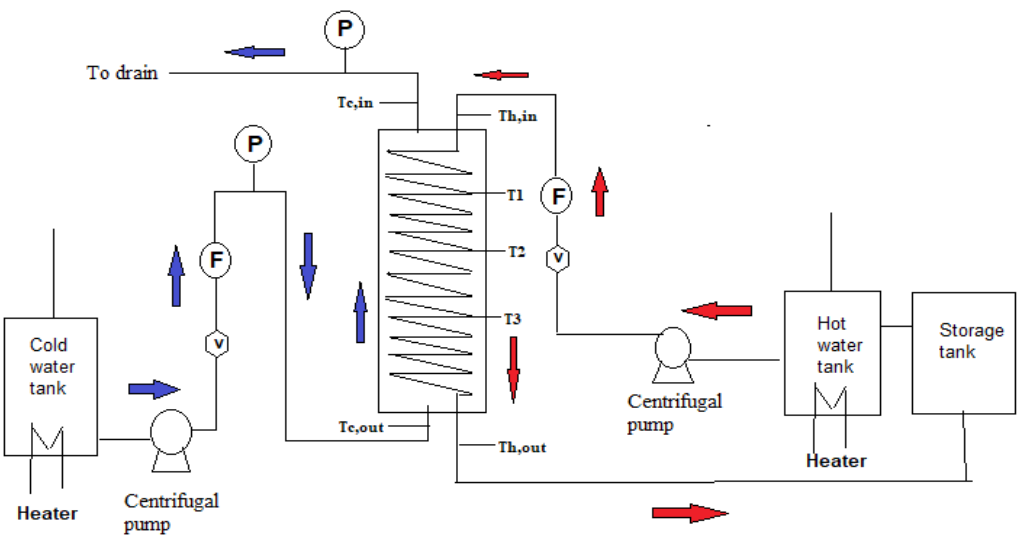
3- two water tanks: water was used as a working fluid. Hot water at 60 ºC enters the coil tube while the cold water enters the shell at 30 ºC.

Auxiliary parts: - these parts consist of measurement tools used.

Seven thermocouples were used in the test section. Four thermocouples were fitted in the entrance and exit regions of each shell and tube sides. While three thermocouples were fitted at an equal distance (25 cm) on the outside surface of the tube test section.

Two flow meters measure the flow amount of cold and hot water enters the shell and tube sides with four different volumetric flow rates (1.8, 3, 4, 5 L/min).

The pressure drop through the helical coil heat exchanger was measured by using two pressure gauges fitted at the shell ends.



P: - pressure gages F: - flow meter

T: - thermocouples v: - valve

Figure 5. Schematic diagram of a vertical helical coil heat exchanger

Fig. 5 shows the schematic diagram of a complete rig with all main and auxiliary parts.

2.1. The Experimental Work Boundary Condition and Assumptions.

The boundary conditions and assumptions which the experiment work depended on are:-

1- Steady-state operation conditions.

2- No heat generation.

3- One phase flow.

4- Constant specific heat and physical properties.

5- Insulated shell surface.

6- Neglect the potential and kinetic energy changes.

2.2. Experimental work Procedure

Sixteen runs for each helical coil heat exchanger with and without fins were done. Each run could be summaries as follows:

1- Coldwater at 30 ^{o}C was pumped to the shell side while hot water was supplied to the tube side at 60 ^{o}C counter currently. An electrical controller was added to each waterside to ensure constant temperature sources.

2-cold water fed to the shell side at a selected flow rate (1.8, 3, 4, or 5 L/min).

3- Hot water was fed to the shell side at (1.8, 3, 4, 5 L/min) respectively.

4-After 20 minutes to ensure steady-state the temperature and pressure were measured and recorded through the rig overall system.

The same procedure was repeated for all shell volumetric flow rates.

To check the accuracy of the readings, the experiment repeated three times

3. Theoretical Work

Several variables and coefficients were calculated to ensure the effectiveness and best performance of the proposed finned shape helical coiled heat exchanger. These factors are

I-Heat transfer rate calculations

The helical coil heat exchanger enhanced heat more than the straight tube heat exchanger as a result of the turns which made a centrifugal force. This force would generate a radial velocity component in a normal direction to the primary flow that would produce a twin of symmetrical vortices in the tube [9].

The heat transfer coefficient could be calculated from the following relations:-

$$q = \dot{m}\, Cp\, (Ti - To) \qquad (1)$$

The hot water enters the tube while the cold water enters the shell, so the relations become [10]:-

For the shell side:-

$$qc = \dot{m}c\, Cpc\, (Tc,o - Tc,i) \qquad (2)$$

For the tube side:-

$$qh = \dot{m}h\, Cph\, (Th,i - Th,o) \qquad (3)$$

The shell was insulated by the glass wool to prevent heat losses to the surrounding. The test must reach to the equilibrium state at (qh = qc).

II-Reynolds number

The increase in the Reynolds number will cause an increase in the heat transfer coefficient that leads to a rise in heat transferring [11]. The laminar flow at the shell side. Reynolds number ranged from 430.61 to 1196.08. The equation that could be applied at the shell side was [12]:-

$$\mathrm{Re}_c = \frac{\rho_c v_c D_h}{\mu_c} \tag{4}$$

Hydraulic diameter when the tube was helically coiled [9]:-

$$D_h = \frac{Ds^2 - \pi Dc\left(d_o\right)^2 \gamma^{-1}}{Ds + \pi Dc\left(d_o\right)\gamma^{-1}} \tag{5}$$

Where $\gamma = \dfrac{p}{\pi\, Dc}$ (6)

At finned helical coil tube De will be found from:-

$$D_e = \frac{4\, Ae}{Pe} \tag{7}$$

$$Ae = A_c - A_f N_f \tag{8}$$

$$A_c = \frac{\pi}{4}\left(Ds^2 - d_o^2\right) \tag{9}$$

The fin base which is manufactured at the curved shape to fit on the external surface of the coil as shown in Fig.6 will obtain from below relation

$$A_f = A_{total} - A_{segment} \tag{10}$$

$$A_{total} = \frac{1}{2} H_f b_f \tag{11}$$

$$A_{segment} = \frac{1}{2} Rc^2 \left(m - \sin 2\theta\right) \tag{12}$$

$$\mathrm{Sin}\ \theta = \frac{1}{Rc} \tag{13}$$

$$m = \frac{2\theta\pi}{180} \tag{14}$$

$$Pe = \pi\,(Ds + do) + (2\,H_f N_f) \tag{15}$$

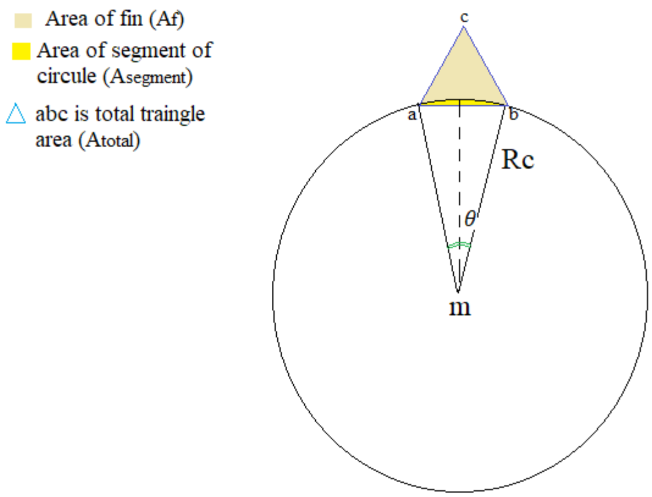


Figure 6. Fin area calculation

III-Nusselt Number

Nusselt number equation of the inside the helically coiled tube is:-

$$Nu_i = \frac{h_i di}{k_h} \tag{16}$$

Where (hi) is achieved by cooling Newton's law [16]:-

$$h_i = \frac{q_h}{A_{si}(T_m - T_s)} \tag{17}$$

The coiled tube surface area can be calculated from [17, 18]:-

$$A_{si} = \pi \, di \, L_c \tag{18}$$

And

$$L_c = N\sqrt{(2\pi R_c)^2 + b^2} \tag{19}$$

$$T_m = \frac{T_{h,i} + T_{h,o}}{2} \tag{20}$$

$$T_s = \frac{T1 + T2 + T3}{3} \tag{21}$$

Nusselt number of the outer side of the helical coil tube will be:-

$$Nu_o = \frac{h_o D_h}{k_c} \tag{22}$$

Where (ho) can be obtained from:-

$$ho = \frac{qavg}{A_{so} \Delta T_{\log}} \tag{23}$$

$$A_{so} = \pi \, do \, Lc \tag{24}$$

For finned helical coil tube (ho) is calculated from:-

$$ho = \frac{qavg}{A_{eff} \Delta T_{\log}} \tag{25}$$

$$A_{eff} = A_{so} + A_f \tag{26}$$

The log mean temperature difference for the counter flow is [20, 21]:-

$$\Delta T_{log} = \frac{\Delta T1 - \Delta T2}{\ln\left(\frac{\Delta T1}{\Delta T2}\right)} \quad (27)$$

$$\Delta T1 = T_{h,i} - T_{c,o} \quad (28)$$

$$\Delta T2 = T_{h,o} - T_{c,I} \quad (29)$$

IV-Effectiveness

The effectiveness is the ratio of the actual to maximum heat transfer amount as shown in equations below:-

$$\varepsilon = \frac{qact}{qmax} \quad (30)$$

$$qact = q_h = q_c \quad (31)$$

$$q_{max} = \dot{m}_{min}\, Cp\, (T_{h,i} - T_{c,i}) \quad (32)$$

V-Thermal performance factor

The enhancement ratio of heat transfer to the ratio of friction factor is known as the thermal performance ratio (TPF). TPF was used in the heat exchanger to compare different used passive techniques [22]. The overall performance of the finned helical tube to the unfinned helical tube could be obtained from [23]:-

$$TPF = \frac{\frac{Nu_f}{Nu_o}}{\text{¿¿}} \quad (33)$$

(TPF $\geq$ 1)

Where f (friction factor) obtained from pressure drop relation

$$\Delta P = f ¿) (\frac{L}{D_h} ¿ \qquad (34)$$

The properties of water and copper tube have been considered from the heat transfer book JP Holman [25].

4. Results and discussion

The helical coil heat exchanger enhances the heat transfer more than a straight tube as a result of the turns that creates a centrifugal force which generates radial velocity component perpendicular to the main flow direction. Vortices would appear in the tube which increases the heat transfer from the tube side to the shell side.

The addition of fins would increase the enhancement of the helical tube due to the increment in the tube surface area that would increase the convection heat transfer to the fluid. The increment in mass flow rate caused an increase in the heat transfer rate. The increase in water velocity raised the convection heat transfer coefficient value. Nusselt number which is directly proportional to Reynolds number. When Reynold's number increased Nusselt number would be increased also. At laminar water flow at the shell side, the enhancement in finned helical tube heat exchanger is 16.5 % more than the unfinned helical tube heat exchanger. This enhancement was a result of the presence of the triangular extended surfaces which increase the convection heat transfer coefficient which leads to an increase in Nusselt number. A comparison was done between finned and unfinned helical tube heat exchanger as shown in Fig. 7.

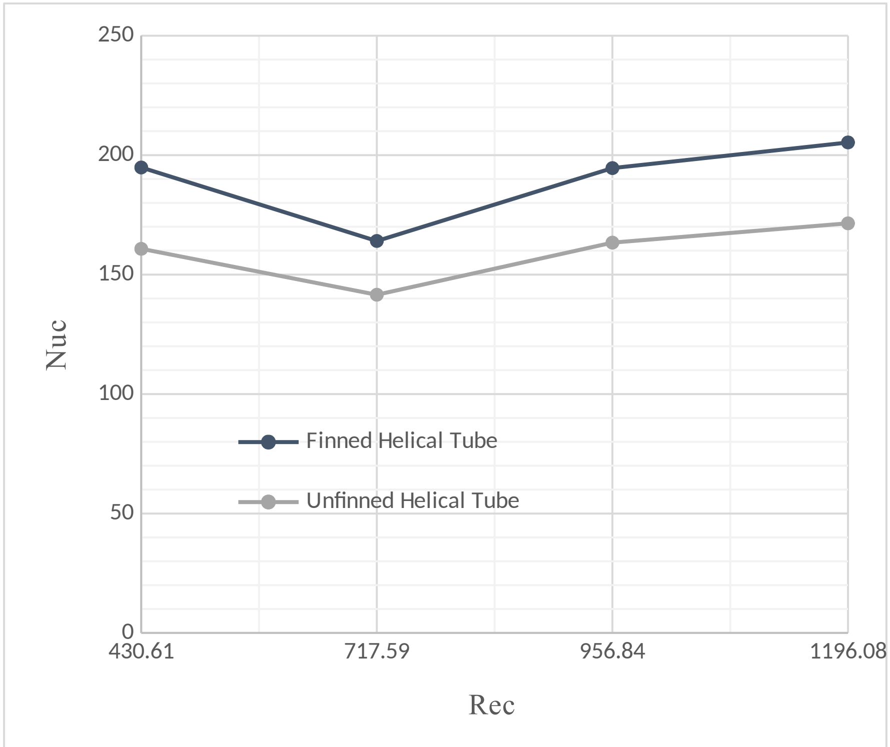


Figure 7: Comparison between the finned and unfinned helical coil heat exchanger heat transfer rate and Reynolds number at cold shell side where Reh=19613

Reynolds number is inversely proportional to the helical coiled tube heat exchanger effectiveness as shown in the following figure.

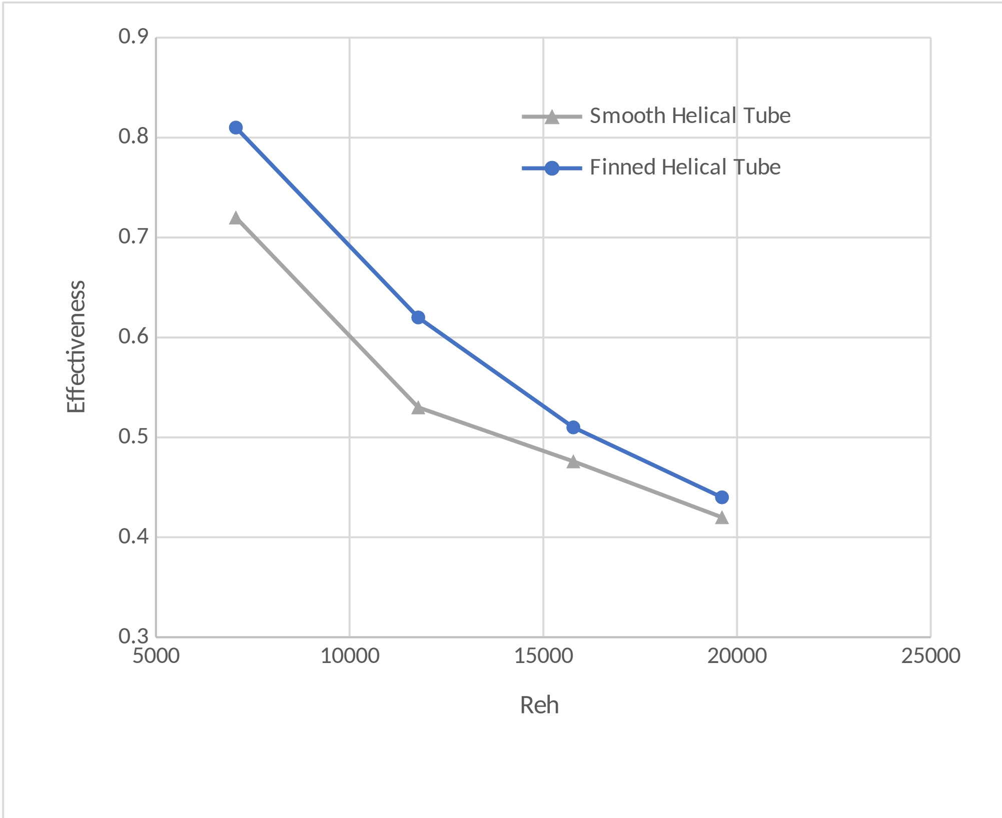


Figure 8: The relation between the effectiveness and tube side Reynolds number where Rec=1196.08.

The increase in the maximum heat transfer of the finned helical coil heat exchanger leads to a decrease in the heat exchanger effectiveness. The finned helical tube heat exchanger effectiveness is higher than the smooth helical tube as a result of the increase in the actual heat transfer rate of the finned case. The improvement in the helical tube with fins is 11% when compared with the helical tube without fins.

Figure 9 shows the average ratio of thermal performance factor (TPF) which increases with increasing Reynolds number at the shell side Nusselt number increase for each water flow rate value. Generally, the results show a good contract with the condition because of the obtained results are more than 1. The ratio obtained from the finned and unfinned helical tube is reached to 1.213, so, the experiment rig considers a high activity.

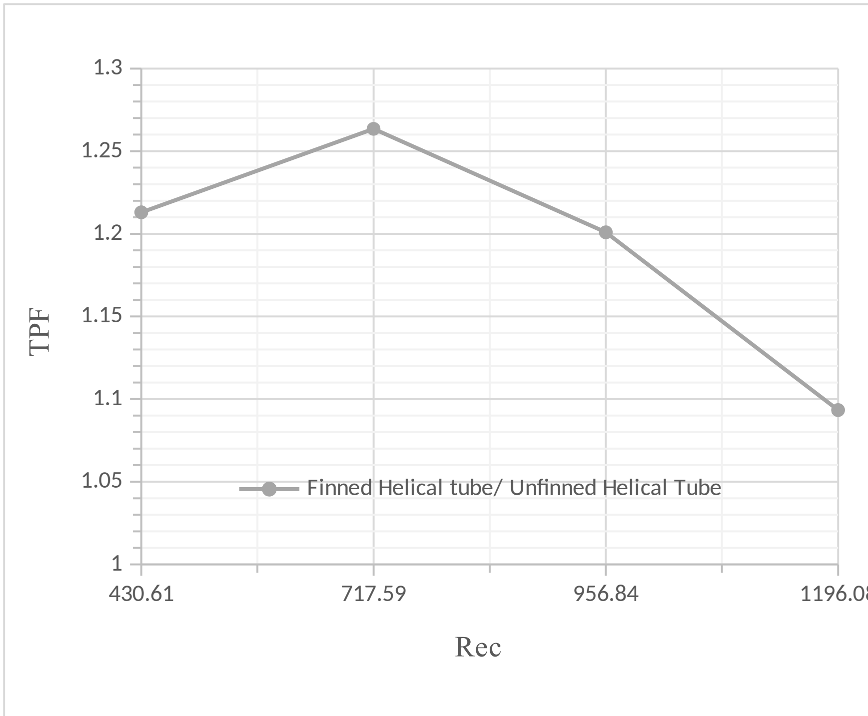


Figure 9 The relation between average TPF and Re. the number at the cold water side (shell)

5. Validation

Nanofluid, as well as fins, considered as one method of heat enhancement in the helical tube heat exchanger. So, to investigate which way is more active than the other, Ali et al. [8] study was chosen for comparison with the present work. The study of Ali included the enhancement by using nanofluid of TiO2 - water in the vertical helical coil heat exchanger. Ali et al., [8] used copper tube (10.5mm) outer diameter, (93 mm) coil diameter, and (17 mm) pitch with 13 turns. The nanofluid volume concentration are 0.1% and 0.2 %. The second one is chosen for the comparison. Figure 10 presents the relation between Dean's number and Nusselt number for the present work and Ali work at the turbulent side. At the shell side, the cold flow rate chosen for current work to comparison is 5 L/min but the flow of Ali work is 15 kg/s. The values of Dean and Nusselt number at existing work are (2779.81, 4633.02, 6208.25, and 7718.61) and (132, 141.28, 166.93, 193.27), respectively, while Ali work Dean Number is ranged from 1000 to 4000 and the Nusselt number ranged from 105 to 170. The results show that the finned helical tube is more active than using Nanofluids at different Dean number due to increasing convection heat transfer to surrounding and increment of surface area.

This comparison proved that the fins technique was more efficient than the nanofluid enhancement techniques. This was a result of more enhancing heat transfer in finned helical coiled tube heat exchanger rather than using nanofluid. Besides that, the nanofluid enhancing techniques had a disposal water problem and dangerous to handling without safety care which was not needed to handle a finned helical coiled tube. The enhancement with nanofluid decreases with time due to the aggregate and increase in nanofluid particle size through the operation process and change to a micro size which needs regeneration and ultrasonic redistributor which an additional cost during the operation. The triangular finned helical tube has a low cost and needs a small time to manufacture. Also, it considers clean technique as compared with nanofluid that effects on human health.

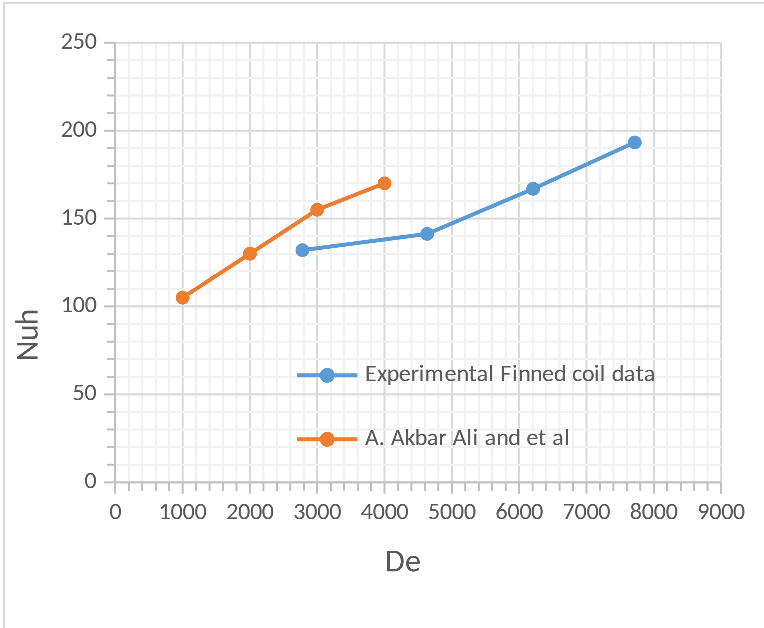


Figure 10 Comparison between experimental data of finned coil heat exchanger and pervious study using Nanofluid**.**

6. Conclusions

The outcomes of the present study could be summarized as follows:

1. The addition of fins on the tube outside the surface of the helically coiled tube heat exchanger would improve the heat transfer rate by increasing the turbulence in the shell side.
2. The addition of fins will increase the surface area and raise the transferring of heat in the heat exchanger
3. Heat transfer rate increases with increasing flow rate and Reynolds number.
4. Nusselt number increases with the Reynolds increment or fluid flow rate velocity increments.
5. The effectiveness reduces with Reynolds number increases.

6. The addition of fins gave better enhancement in the coil heat exchanger compared to enhancement with nanofluids besides the elimination of the disposal problem.

Symbols

Symbol	**Notation**	**Unit**
q	Heat transfer rate	KW
$\dot{m}$	Mass flow rate	Kg/s
$\tilde{V}$	Volume flow rate	L/min
T	Temperature	^oC
Cp	Specific heat with constant pressure	KJ/kg. k
Re	Reynolds number	Dimensionless
v	velocity	m/s
d	Tube diameter	m
A	Area	m^2
D_h	Hydraulic diameter	m
Dc	Coil diameter	m
P	pitch	m
Ds	Shell diameter	m
De	Equivalent diameter	m
A_f	Area of Fin	m^2
N_f	Number of fins	dimensionless

Ac	Cross section area	m^2
H_f	Height of fin	m
b_f	Base of Fin	m
Pe	Wetted perimeter	m
h	Convection heat transfer coefficient	$W/m^2.^oC$
k	Conduction heat transfer coefficient	$W/m.^oC$
Nu	Nusselt number	dimensionless
As	Surface area	m^2
Tm	Mean temperature	^oC
Ts	Average surface Temperature	^oC
L	Tube length	m
L_c	Coil length	m
ΔT_{log}	Log mean temperature difference	^oC
A_{eff}	Effective area	m^2
ε	Effectiveness	dimensionless
f	Friction Factor	dimensionless
ΔP	Pressure drop	Kpa
TPF	Thermal	dimensionless

	performance factor	

Subscripts

Symbol	**Meaning**
i	Inlet
o	Outlet
c	Cold
h	Hot

Greek Symbols

Symbol	**Notation**	**Unit**
ρ	Density	Kg/m^3
μ	Dynamic viscosity	N.s/m^2
γ	Dimensionless pitch	dimensionless
θ	Angular Displacement	Degree

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