archives
of thermodynamics
Vol. **41**(2020), No. 2, 239–256
DOI: 10.24425/ather.2020.133631

Innovative method for heat transfer enhancement through shell and coil side fluid flow in shell and helical coil heat exchanger

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Abstract The effect of shell side and coil side volume flow rate on overall heat transfer coefficient, effectiveness, pressure drop and exergy loss of shell and helical coil heat exchanger were studied experimentally under steady state conditions. The working fluid, i.e., water was allowed to flow at three different flow rates of 1, 2, and 3 l/min on shell side (cold water) and at 1, 1.5, 2, 2.5, and 3 l/min on coil side (hot water) for each shell side flow rate at the temperatures of 298±0.4 K and 323±0.4 K, respectively. The results found that the overall heat transfer coefficient increased with increasing both shell side and coil side volume flow rates. The inner Nusselt number significantly increased with the coil side Dean number.

Keywords: Overall heat transfer coefficient; Effectiveness; Helical coil; Pressure drop

Nomenclature

A – surface area, m^2
C – constant

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C_p – specific heat, J/kgK
D – helical coil diameter, m
De – Dean number,
d – tube diameter, m
H – height of helical coil, m
h – heat transfer coefficient, W/m^{2}K
k – thermal conductivity, W/mK
L – length of tube, m
m – exponent
N – number of helical coil turn
Nu – Nusselt number
P – pitch of helical coil, m
P_d – pressure drop, kPa
Q – heat transfer rate, W
Re – Reynolds number, $= 4\rho V/\pi\mu d_i$
T – temperature, K
U – overall heat transfer coefficient, W/m^{2}K
V – volume flow rate of water, l/min
v – coil tube side fluid velocity m/s

Greek symbols

ρ – density, kg/m^3
μ – viscosity, Ns/m^2
ε – effectiveness
θ – bulk mean temperature, K

Subscripts

avg – average
c – cold
h – hot
in/i – inlet/inner
min – minimum
out/o – outlet/outside
t – tube
w – water

Abbreviations

SHCHE – shell and helical coil heat exchanger

1 Introduction

In process industries, heat exchangers are widely used for heating and cooling applications, including heat recovery system, food processing, steam power plant, medical equipment, refrigeration, and air conditioning units. The passive method, beside active method, is one that enhances the heat transfer by design modifications [1–3].

Andrzejczyk and Muszynski [4] executed an experiment to analyze the performance of helical coil tube with surface modification for laminar and turbulent fluid flow. The findings showed that the overall heat transfer coefficient increased with surface modification of helical coil tube for both parallel and counter flow arrangement. Bohdal *et al.* [5,6] studied the pressure drop and heat transfer of refrigerants during condensation in pipe minichannels. The experimental results observed that the pressure drop in a two-phase flow depends on the type of refrigerants and process parameters. It was also developed a correlation to estimate the local heat transfer coefficient. Xin and Ebadian [7] performed experiments on helical pipes with three fluids; namely air, water and ethylene glycol and correlations were developed to estimate the Nusselt number. It was also observed that when the curvature ratio was ranged from 0.0267 to 0.0884, no significant change in Nusselt number was obtained. Yildiz *et al.* [8] observed that the pressure drop and heat transfer increases due to rotation of helical pipes and also indicated that the heat transfer increases by decreasing the curvature ratio of helical pipe.

Prabhanjan *et al.* [9] conducted experiments on vertical helical coiled tube to obtain outside Nusselt number and proposed a correlation based on different characteristics length of tube. Rennie and Raghavan [10] experimented on double-pipe helical tube heat exchanger to calculate the heat transfer coefficient by Wilson plot method. The experimental data fitted well with that of the numerically obtained one for larger helical tube while it deviates for smaller helical tube. Dabas *et al.* [11] generated a computer based mathematical model to express the performance of shell and helical coil tube condenser. The predicted results deviate by 7% to 8.05% from the experimental results. Neshat *et al.* [12] studied experimentally and numerically four helical coils with different curvature ratios, under unsteady natural convection heat transfer. The results showed that the Nusselt number on the outer surface of helical coils is the function of geometrical parameters and operational parameters of the helical coil. Alimoradi numerically developed a correlation to evaluate the thermal effectiveness of shell and coil tube heat exchanger. The results found that the effectiveness increases with coil diameter, height and pitch of the coil [13].

Moawed conducted experiments on horizontal and vertical helical pipes for heating and cooling applications and studied the effect of the pitch to tube diameter ratio, length to tube diameter ratio and coil diameter ratio on the performance [14]. Similar work has been performed on helical coil tube considering different parameters. The Nusselt number increased with increase in diameter ratio and it decreased by decreasing the value of pitch to diameter ratio [15]. Ali [16] studied natural convection heat transfer from four different coil diameter to tube diameter ratios of vertical helical coil tube. It was found that the heat transfer coefficient increases with the coil length for tube diameter 0.008 m while decreases with coil length for tube diameter 0.012 m. Overall heat transfer coefficient was calculated for shell and helical coil heat exchanger (SHCHE) by Shokoumand *et al.* [17] using the Wilson plots method. It was found that the overall heat transfer coefficient for counter flow is 40% higher as compared to parallel flow. Correlations were developed for three different curvature ratios and Reynolds number ranging from 3166 to 9658 to estimate the heat transfer coefficient under steady state conditions [18]. The overall heat transfer coefficients and Nusselt number increased by increasing the curvature ratio of helical coil tube as reported by Salem *et al.* [19]. Ghias *et al.* [20] experimentally and numerically reported that the heat transfer rate increases by increasing the inlet temperature of hot water, i.e., shell and tube side flow rate of coil in the shell heat exchanger. Alimoradi and Veysi [21] numerically studied the effect of geometrical parameters of shell and coil tube heat exchanger. It was found from the results that the heat transfer rate decreases with increase in the shell diameter and the entropy generation rate increases by increasing the helical tube diameter.

According to comprehensive literature survey conducted, significant amount of studies have been performed on the shell helical coil tube heat exchanger using a larger tube diameter. The present work is focused on smaller tube diameter in shell and helical coil heat exchanger. The aim of the present work is to assess the overall heat transfer coefficient, inner Nusselt number, temperature drop between inlet and outlet of helical coil tube, pressure drop, and exergy loss of SHCHE. The effect of cold flow rate (shell side), lower than or equal to hot flow rate (coil side), on the effectiveness of SHCHE has been also discussed. In most of the studies available in literature, the effect of cold flow rate equal to or higher than hot flow rate on SHCHE can be found. In this way this work is novel and new.

2 Material and methods

Figure 1 shows the SHCHE used in the present work. The helical coil was constructed from the straight soft copper tube with the help of wooden pattern. Initially the tube was filled with fine sand and flushed with compressed air after coiling to preserve the smoothness of inner tube surface.

Figure 1: Shell and helical coil heat exchanger.

The helical coil was placed inside the shell of 0.13 m inner diameter and 0.38 m height to form SHCHE. The test section was coated with 0.003 m thick foam to minimize the heat losses to the ambient. The dimensions of heat exchangers are shown in Tab. 1. The length of helical coil tube is calculated as [22]

$$L = N\sqrt{\pi^2 D^2 + P^2}\,. \tag{1}$$

3 Experimental set-up and procedure

Figure 2 depicts the schematic diagram of experimental set-up. It consists of two galvanized iron storage tanks, insulated with foam of 0.003 m thickness. A 3 kW electrical heater was fixed at the bottom of the tank (hot water). Hot water at the temperatures 323±0.4 K was pumped to the helical coil side and returns back to the hot water storage tank.

Table 1: Geometrical parameters of heat exchangers.

Parameters	Unit	Value (present work)	Value (Jamshidi *et al.* [32])
Inner tube diameter	m	0.00579	0.009
Helical coil diameter	m	0.08	0.116
Pitch	m	0.0254	0.013
Number of helical turns	–	10.5	10
Helical coil length	m	2.65	3.64
Inner diameter of shell	m	0.13	0.14
Height of shellm	m	0.38	0.25
Shell volume	m^3	0.005	0.004

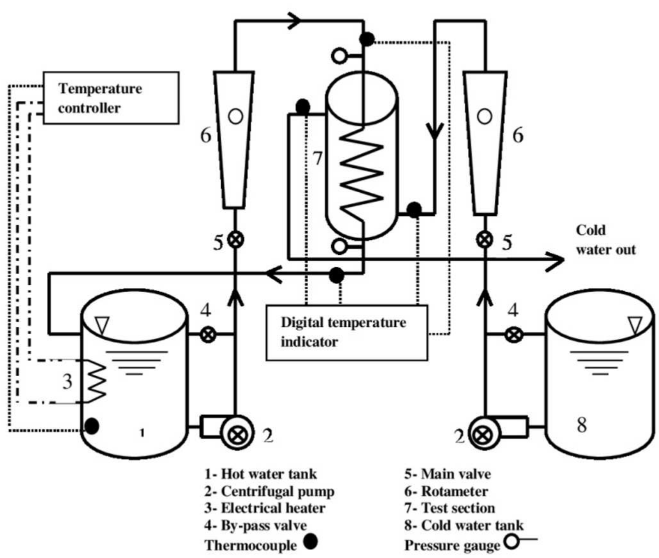

Figure 2: Schematic view of experimental set-up.

Two analog pressure gauges (Wika, range 0–207 kPa) are fixed at the inlet and outlet of helical coil to measure the pressure drop. As the hot water was cooled by the cold water (298±0.4 K) flow through shell side in counter clockwise direction, the hot water temperature got reduced but its temperature was maintained by using thermostat. The volume flow rate of hot water (coil side) V_h and cold water (shell side) V_c were measured by two rotameter (Flow Point) ranging from 0.5 l/min to 5 l/min with an accuracy of ± 2% of full scale . Dizaji *et al.* [23], Purandare *et al.* [24], and Naik [25] also used rotameters to measure flow rates up to temperature 353 K. Four K-type thermocouples were attached to each inlet and outlet of the test section. Two four channels digital temperature indicators were used to store the temperature as shown in Fig. 3. In all 15 tests were conducted for five different flow rates (1–3 l/min) of coil side for each 1 l/min, 2 l/min, and 3 l/min shell side flow rate. The set-up was allowed to come under steady state condition within ±0.4 K accuracy.

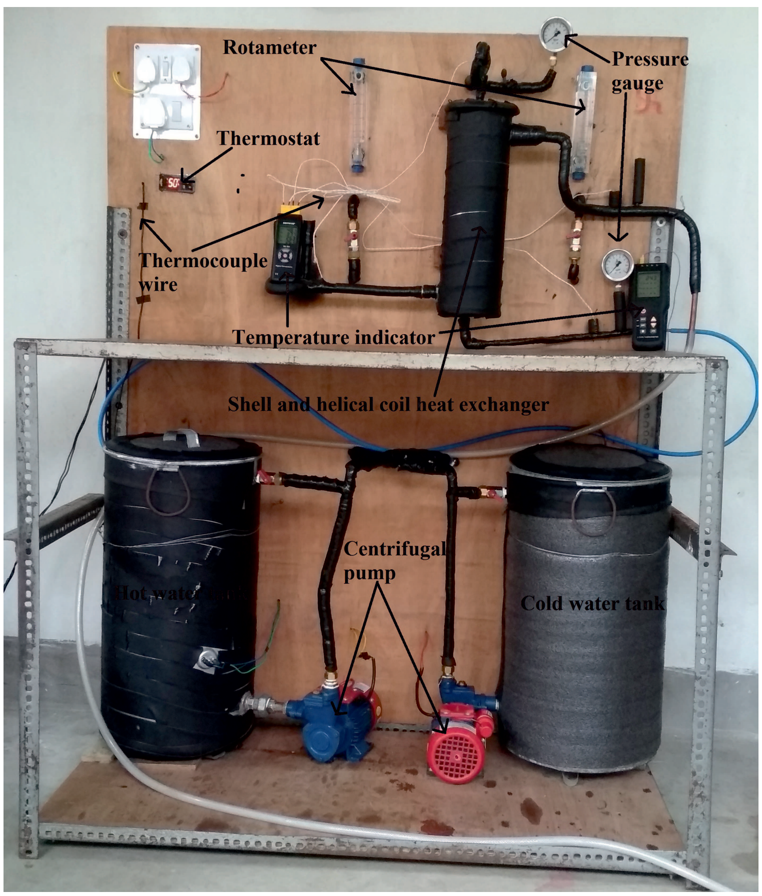

Figure 3: Pictorial view of experimental set-up.

4 Data analysis

To ensure accuracy in the result, the properties of water were calculated from the following equations at bulk mean temperature (average of inlet and outlet of fluid) θ, for both shell and coil side fluid:

- density of water is calculated from [26]

$$\rho_w = -3.570 \times 10^{-3}\theta^2 + 1.88\theta + 753.2\,, \tag{2}$$

- viscosity of water is defined by [26]

$$\mu_W = 2.591 \times 10^{-5} \times 10^{\frac{238.3}{\theta - 143.2}}, \tag{3}$$

- thermal conductivity of water is defined by [26]

$$k_w = -8.354 \times 10^{-6}\theta^2 + 6.53\theta - 0.5981, \tag{4}$$

- specific heat of water is calculated from [27]

$$C_{pw} = 1.1105 \times 10^{-5}\theta^3 - 0.0031078\theta^2 - 1.478\theta + 4631.9\,. \tag{5}$$

Heat gained by cold water in the heat exchanger was calculated as

$$Q_c = \rho_c V_c C_{pw}(T_{c,o} - T_{c,i})\,, \tag{6}$$

while heat loss by hot water in the heat exchanger was calculated as follows:

$$Q_h = \rho_h V_h C_{pw}(T_{h,o} - T_{h,i})\,, \tag{7}$$

hence the average heat transfer rate was calculated as

$$Q_{avg} = \frac{Q_c + Q_h}{2}\,. \tag{8}$$

The energy balance is performed in the data analysis section. The heat losses were found to be within 2.1% ($|Q_c - Q_h|/Q_{avg}$). This criterion is within the range in accordance with [28–30]. The overall heat transfer was then calculated as

$$U = \frac{Q_h}{A_i \mathrm{LMTD}}. \tag{9}$$

The heat exchanger operated with the counter-current flow arrangement, logarithmic mean temperature difference was calculated from

$$\mathrm{LMTD} = \frac{(T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c,i})}{\ln\left(\frac{T_{h,i} - T_{c,o}}{T_{h,o} - T_{c,i}}\right)}. \tag{10}$$

The inner heat transfer coefficient was calculated using Wilson plots [17]. In this method heat transfer coefficients are calculated without the requirement of helical coil tube wall temperature. In the present work, the shell side flow was kept constant while the coil side flow was varied for five different flow rates. The overall heat transfer coefficient can be related to the inner and outer heat transfer coefficients as follows [17]:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{A_i \ln \frac{d_o}{d_i}}{2\pi k_t L} + \frac{1}{h_o}\,. \tag{11}$$

The shell side flow rate has constant value; it can be assumed that the outer heat transfer coefficient (shell side) is constant. The inner heat transfer coefficient (coil side) is assumed to behave in the following manner with the coil tube side fluid velocity, v_i

$$h_i = C v_i^m\,. \tag{12}$$

Substituting Eq. (12) into Eq. (11), hence constant C and exponent m were obtained through the curve fitting of experimental data. The inner heat transfer coefficient can be then calculated from Eq. (12) and finally, inner Nusselt number can be determined from

$$\mathrm{Nu} = \frac{h_i d_i}{k}\,. \tag{13}$$

Effectiveness of the heat exchanger was calculated as

$$\varepsilon = \frac{Q_{\mathrm{actual}}}{Q_{\max} = (\rho V C_p)_{\min}(T_{h,i} - T_{c,i})}\,. \tag{14}$$

The total exergy loss for hot and cold fluid was calculated as [23]

$$E_{\mathrm{loss}} = T_{\mathrm{environment}}\left[\rho_h V_h C_{p,h} \ln\left(\frac{T_{h,out}}{T_{h,in}}\right) + \rho_c V_c C_{p,c} \ln\left(\frac{T_{c,out}}{T_{c,in}}\right)\right]. \tag{15}$$

5 Uncertainty analysis

The uncertainties in the results were obtained from the errors in the measurement of the volume flow rate, temperature, pressure drop and helical coil inner tube diameter. All these parameters were measured with appropriate instruments and correct method to ensure the accuracy in the results. The uncertainty in the measurement was carried out based on the Holman method. The uncertainties in the result due to some independent variables were evaluated by the following equation [31]:

$$W_R = \left[\left(\frac{\partial R}{\partial X_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial X_2} w_2 \right)^2 + \ldots\ldots + \left(\frac{\partial R}{\partial X_n} w_n \right)^2 \right]^{\frac{1}{2}}, \tag{16}$$

where R is the calculated result of experiments. The result R is a given function of the independent variables X_1, X_2,..., X_n and the uncertainties in the independent variables are w_1, w_2,..., w_n.

For example Reynolds number uncertainties can be calculated as follows:

Substituting

$$\mathrm{Re} = \frac{4\rho V}{\pi \mu d_i} \tag{17}$$

into Eq. (16) yields

$$W_{Re} = \left[\left(\frac{\partial \mathrm{Re}}{\partial V} w_V \right)^2 + \left(\frac{\partial \mathrm{Re}}{\partial d_i} w_{d_i} \right)^2 + \left(\frac{\partial \mathrm{Re}}{\partial \mu} w_\mu \right)^2 + \left(\frac{\partial \mathrm{Re}}{\partial \rho} w_\rho \right)^2 \right]^{\frac{1}{2}}, \tag{18}$$

where

$$\frac{\partial \mathrm{Re}}{\partial V} = \frac{\rho}{\pi \mu d_i}, \frac{\partial \mathrm{Re}}{\partial d_i} = -\frac{\rho V}{\pi \mu d_i^2}, \frac{\partial \mathrm{Re}}{\partial \mu} = -\frac{\rho V}{\pi \mu^2 d_i}, \frac{\partial \mathrm{Re}}{\partial \rho} = \frac{V}{\pi \mu d_i}, \tag{19}$$

thus after substituting

$$W_{Re} = \left[\left(\frac{\rho}{\pi \mu d_i} w_V \right)^2 + \left(-\frac{\rho V}{\pi \mu d_i^2} w_{d_i} \right)^2 + \left(-\frac{\rho V}{\pi \mu^2 d_i} w_\mu \right)^2 + \left(\frac{V}{\pi \mu d_i} w_\rho \right)^2 \right]^{\frac{1}{2}}. \tag{20}$$

If divide both sides of the Eq. (20) by $4\rho V/\pi \mu d_i$

$$\frac{W_{Re}}{\mathrm{Re}} = \left[\left(\frac{w_V}{V} \right)^2 + \left(\frac{w_{d_i}}{d_i} \right)^2 + \left(\frac{w_\mu}{\mu} \right)^2 + \left(\frac{w_\rho}{\rho} \right)^2 \right]^{\frac{1}{2}}. \tag{21}$$

Uncertainties in the results are estimated and summarized in Tab. 2. Uncertainty estimation presented maximum value of 3.47% for overall heat transfer coefficient, 4.93% for effectiveness, 6.22% for Reynolds number and 5.37% for pressure drop, respectively.

Table 2: Uncertainties in the measurements.

Parameter	Unit	Uncertainty
Water volume flow rates	l/min	± 0.06
Cold water inlet temperature	K	± 0.40
Hot water inlet temperature	K	± 0.40
Cold water temperature out	K	± 0.40
Hot water outlet temperature	K	± 0.40
Overall heat transfer coefficient	%	± 3.47
Effectiveness	%	± 4.93
Reynolds number	%	± 6.22
Pressure drop	%	± 5.37
Properties of water	%	± 0.10

6 Results and discussion

Figure 4 shows the effect of coil side (hot water) and shell side (cold water) volumetric flow rates on overall heat transfer coefficient, which was calculated on the basis of average heat transfer rate of shell side and coil side. Progressively the overall heat transfer coefficient increases by increasing the coil side volume flow rate as well as shell side volumetric flow rates. The present study revealed the maximum overall heat transfer coefficient of 1444.2 W/m^{2}K at V_h= 3 l/min and V_c= 3 l/min which is higher when compared with the previous works available in the literature. Jamshidi *et al.* [32] observed the value approximately equal to 1120 W/m^{2}K as an optimum overall heat transfer coefficient for the condition of $P = 0.013$ m, $D = 0.116$ m, $V_h = 3$ l/min and $V_c = 2$ l/min, but in the present work the overall heat transfer coefficient was found to be 1298.66 W/m^{2}K at the same flow rates and under similar conditions. This difference is due to the design modifications, i.e., strongly influenced by smaller tube diameter and helical coil diameter. The smaller tube diameter and helical coil diameter pertains the higher centrifugal effect, which plays a significant role to enhance heat transfer.

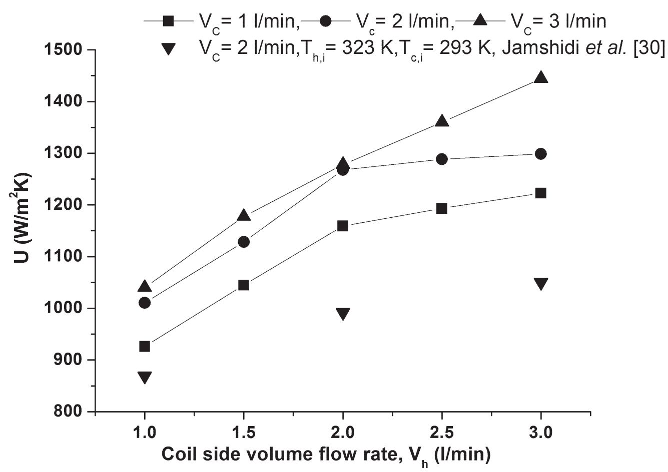

Figure 4: The variation of coil side and shell side volume flow rates on U.

Figure 5 shows the temperature difference between inlet and outlet of the helical coil tube and it decreases by increasing the coil side volumetric flow rate while it increases by increasing the shell side volumetric flow rate. Figure 6 depicts the behavior of the effectiveness which can be increasing or decreasing or a combination of both. Indeed, the effectiveness increases with increase in hot flow rate for constant cold flow rate of 1 l/min. This increasing nature is due to the minimum heat capacity $(\rho VC_p)_{\min}$ in Eq. (14) which is replaced by the heat capacity of cold flow rate. For cold flow rate of 2 l/min, the $(\rho VC_p)_{\min}$ is replaced by the heat capacity of hot flow rate up to three initial points and thereafter $(\rho VC_p)_{\min}$ is replaced by the heat capacity of cold flow rate for final two points. Hence the effectiveness fluctuates by increasing hot fluid flow rate. The effectiveness decreased by increasing hot flow rate at constant cold flow rate of 3 l/min. This decreasing behavior is due to $(\rho VC_p)_{\min}$ in Eq. (14) which is replaced by the heat capacity of hot flow rate only. This figure also shows the effectiveness of the present work at the cold flow rate 3 l/min and it follows the similar trend as available in [33].

Figure 7 represents the variation of inner Nusselt number with the coil side Dean number. The results observed that the inner Nusselt number (inner heat transfer coefficient) increases with increase in the coil side Dean number.

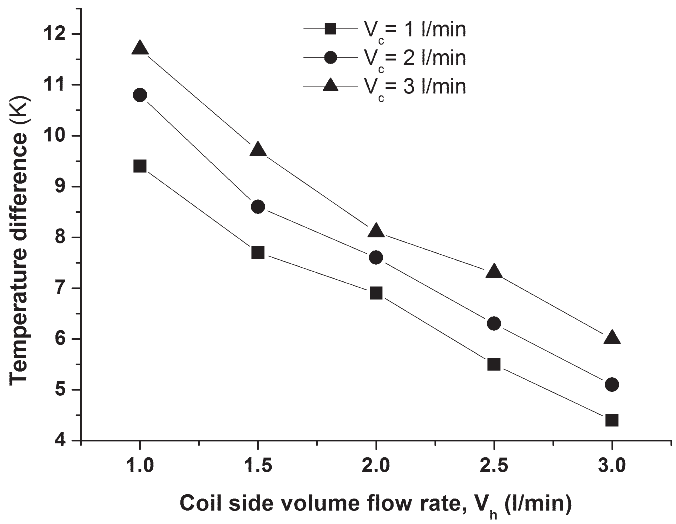

Figure 5: The variation of coil side volume flow rate on temperature difference between inlet and outlet of helical coil tube.

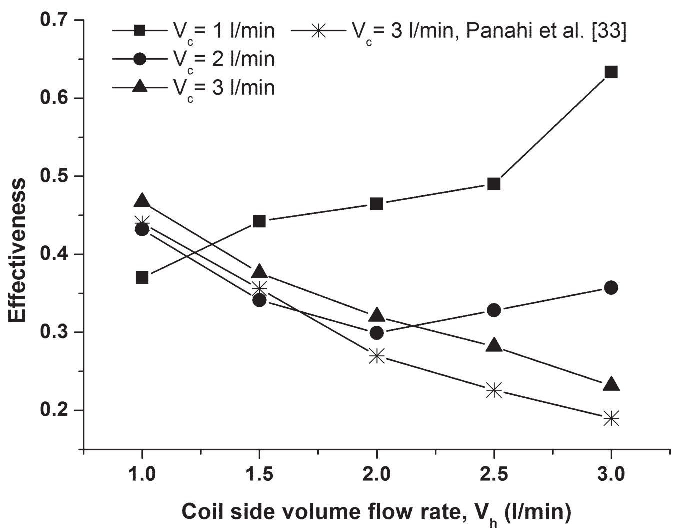

Figure 6: The effect of coil side and shell side volume flow rates on effectiveness.

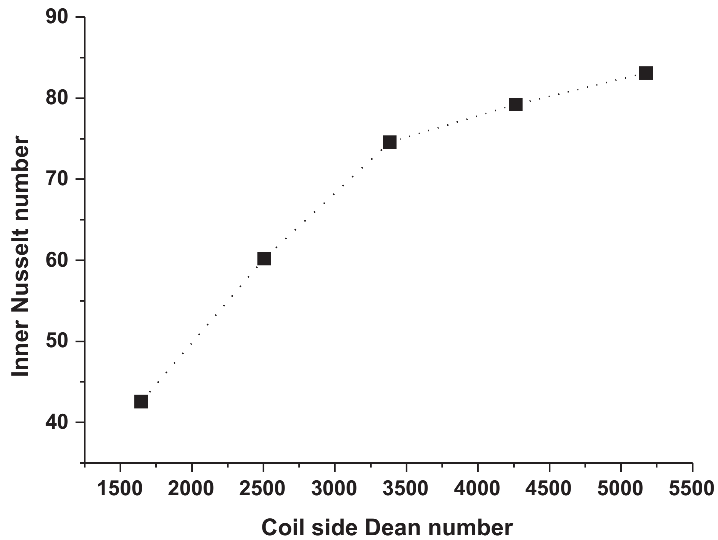

Figure 7: The effect of Dean number on coil side Nusselt number.

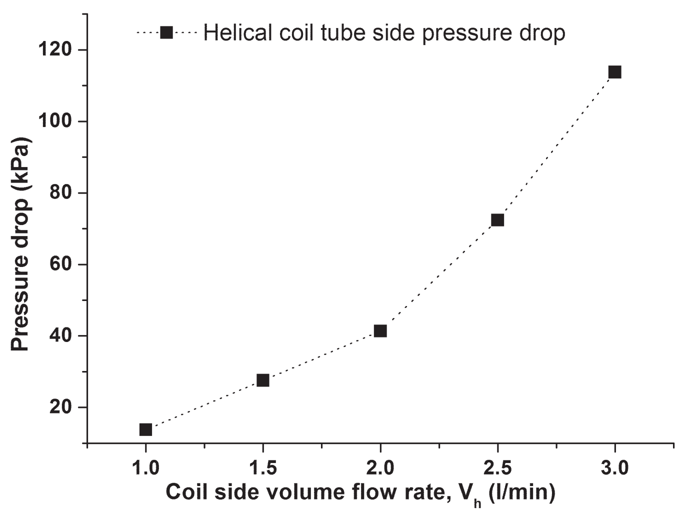

Figure 8: The effect of volume flow rate on pressure drop in coil side.

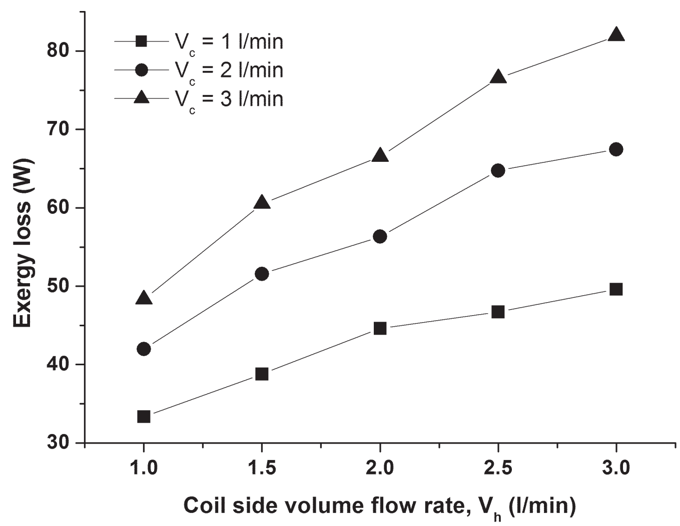

Figure 9: Variation of volume flow rate on exergy loss.

Figure 8 represents the effect of pressure drop in helical coil with the variation of coil side volume flow rate. The pressure drop in helical coil was increased with volumetric flow rate. Figure 9 shows the exergy loss in the coil side volume flow rate at volumetric flow rate of 1, 2, and 3 l/min. It can be clearly seen from the graph that with the increase in cold fluid volumetric flow rate, exergy loss is increased, which also confirms that in the present study, the 2nd law of thermodynamics is obeyed.

7 Conclusions

The experiments were conducted to analyse thermal characteristics and the performance of a shell and helical coil heat exchanger under steady state condition. The following findings of this study can be drawn.

1. The overall heat transfer coefficient increased with increasing both shell side and coil side volume flow rates. It was also noticed that heat transfer coefficient was enhanced than previously designed heat exchanger available in literature. The enhancement of overall heat transfer coefficient due to the higher centrifugal effect in helical tube, caused by heat exchanger is designed with smaller tube and helical coil diameter.

2. The inner Nusselt number (coil side) increased with the inner Dean number.
3. The effectiveness of heat exchanger and temperature difference between inlet and outlet of helical coil leads to decreases with increase in coil side volume flow rate.
4. It was found that increasing the pressure drop results with increase in the coil side volume flow rate. The shell side and coil side volume flow rates significantly affect the exergy loss of heat exchanger.

Received 4 December 2018

References

[1] Dewan A., Mahanta P., Raju K.S., Kumar P.S.: *Review of passive heat transfer augmentation techniques.* Proc. Inst. Mech. Eng. A J. Power Energy **218**(2004), 7, 509–527.

[2] Kareem Z.S., Jaafar M.N.M., Lazim T.M., Abdullah S., Abdul Wahid A.F.: *Heat transfer enhancement in two-start spirally corrugated tube.* Alexandria Eng. J. **54**(2015), 3, 415–422.

[3] Kumar R., Prabhansu, Chandra P.: *An experimental investigation of natural convection heat transfer over outer surface of vertical helical coil condenser.* J. Eng. Res. **5**(2017), 3, 162–173.

[4] Andrzejczyk R., Muszyński T.: *Performance analyses of helical coil heat exchangers. The effect of external coil surface modification on heat exchanger effectiveness.* Arch. Thermodyn. **37**(2016), 4, 137–159.

[5] Bohdal T., Charun H., Sikora M.: *Pressure drop during condensation of refrigerants in pipe minichannels.* Arch. Thermodyn. **33**(2012), 1, 87–106.

[6] Bohdal T., Charun H., Sikora M.: *Heat transfer during condensation of refrigerants in tubular minichannels.* Arch. Thermodyn. **33**(2012), 2, 3–22.

[7] Xin R.C., Ebadian M.A.: *The effects of Prandtl numbers on local and average convective heat transfer characteristics in helical pipes.* J. Heat Transfer **119** (1997), 3, 467–473.

[8] Yildiz C., Bicer Y., Pehlivan D.: *Heat transfer and pressure drops in rotating helical pipes.* Appl. Energy **50**(1995), 1, 85–94.

[9] Prabhanjan D.G., Timothy T.J., Raghavan G.S.V.: *Natural convection heat transfer from helical coiled tubes.* Int. J. Therm. Sci. **43**(2004), 4, 359–365.

[10] Rennie T.J., Raghavan J.S.V.: *Experimental studies of a double-pipe helical heat exchanger.* Exp. Therm. Fluid Sci. **29**(2005), 8, 919–924.

[11] Dabas J.K., Kumar S., Dojeja A.K., Kasana K.S.: *Modeling of a cylindrical shell and helical tube condenser of HFC-134a.* Heat Transfer-Asian Res. **45**(2016), 3, 209–227.

[12] Neshat E., Hossainpour S., Bahiraee F.: *Experimental and numerical study on unsteady natural convection heat transfer in helically coiled tube heat exchangers.* Heat Mass Transfer **50**(2014), 6, 877–885.

[13] Alimoradi A.: *Study of thermal effectiveness and its relation with NTU in shell and helically coiled tube heat exchangers.* Case Stud. Therm. Eng. **9**(2017), 100–107.

[14] Moawed M.: *Experimental investigation of natural convection from vertical and horizontal helicoidal pipes in HVAC applications.* Energy Convers. Manag. **46** (2005), 18-19, 2996–3013.

[15] Moawed M.: *Experimental study of forced convection from helical coiled tubes with different parameters.* Energy Convers. Manag. **52**(2011), 2, 1150–1156.

[16] Ali M.E.: *Experimental investigation of natural convection from vertical helical coiled tubes.* Int. J. Heat Mass Transf. **37**(1994), 4, 665–671.

[17] Shokouhmand H., Salimpour M.R., Akhavan-Behabadi M.A.: *Experimental investigation of shell and coiled tube heat exchangers using Wilson plots.* Int. Commun. Heat Mass Transf. **35**(2008), 1, 84–92.

[18] Pawar S.S., Sunnapwar V.K.: *Studies on convective heat transfer through helical coils.* Heat Mass Transfer, **49**(2013), 12, 1741–1754.

[19] Salem M.R., Elshazly K.M., Sakr R.Y., Ali R.K.: *Experimental investigation of coil curvature effect on heat transfer and pressure drop characteristics of shell and coil heat exchanger.* J. Therm. Sci. Eng. Appl. **7**(2014), 1, 11005–11009.

[20] Ghias A.S.A., Ananth S.V., Anand M.D., Devadhas G.G.: *Experimental study of thermal performance of coil in shell heat exchanger.* Indian J. Sci. Tech. **9**(2016), 13, 1–17.

[21] Alimoradi A., Veysi F.: *Optimal and critical values of geometrical parameters of shell and helically coiled tube heat exchangers.* Case Stud. Therm. Eng. **10**(2017) 73–78.

[22] Neshat E., Hossainpour S., Bahiraee F.: *Experimental and numerical study on unsteady natural convection heat transfer in helically coiled tube heat exchangers.* Heat Mass Transfer **50**(2014), 6, 877–885.

[23] Dizaji H.S., Jafarmadar S., Hashemian M.: *The effect of flow, thermodynamic and geometrical characteristics on exergy loss in shell and coiled tube heat exchangers.* Energy **91**(2015), 678–684.

[24] Purandare P.S., Lele M.M., Gupta R.K.: *Investigation on thermal analysis of conical coil heat exchanger.* Int. J. Heat Mass Transf. **90**(2015), 1188–1196.

[25] Naik B.A.K., Vinod A.V.: *Heat transfer enhancement using non-Newtonian nanofluids in a shell and helical coil heat exchanger.* Exp. Therm. Fluid Sci. **90**(2018), 132–142.

[26] Kays W., Crawford M., Weigand B.: *Convective Heat and Mass Transfer* (4th Edn.). McGraw Hill, Singapore 2005.

[27] Jayakumar J.S., Mahajani S.M., Mandal J.C., Vijayan P.K., Bhoi R.: *Experimental and CFD estimation of heat transfer in helically coiled heat exchangers.* Chem. Eng. Res. Des. **86**(2008), 3, 221–232.

[28] Naphon P.: *Thermal performance and pressure drop of the helical-coil heat exchangers with and without helically crimped fins.* Int. Commun. Heat Mass Transf. **34**(2007), 3, 321–330.

[29] Genić S.B., Jaćimović B.M., Jarić M.S., Budimir N.J., Dobrnjac M.M.: *Research on the shell-side thermal performances of heat exchangers with helical tube coils.* Int. J. Heat Mass Transf. **55**(2012), 4295–4300.

[30] Zhang Z., Zhang Y., Zhou W., Bai B.: *Critical heat balance error for heat exchanger experiment based on entropy generation method.* Appl. Therm. Eng. **94**(2016) 644–649.

[31] Holman J.P.: *Experimental Methods For Engineers* (7th Ed.). Tata Mcgraw-Hill, New York 2007.

[32] Jamshidi N., Farhadi M., Ganji D.D., Sedighi K.: *Experimental analysis of heat transfer enhancement in shell and helical tube heat exchangers.* Appl. Therm. Eng. **51**(2013), 1-2, 644–652.

[33] Panahi D., Zamzamian K.: *Heat transfer enhancement of shell and coiled tube heat exchanger utilizing helical wire turbulator.* Appl. Therm. Eng. **115**(2017), 607–615.