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التحقيق التجريبي للانتقال الحراري بالحمل الحراري الطبيعي على السطح الخارجي لمكثف لفائف حلزونية عمودي

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الخلاصة

تم دراسة الانتقال الحراري الطبيعي على السطح الخارجي لمكثف لفائف حلزونية (HCC) مع ذلك المتواجد بسلك على الأنبوب (WOTC) في الهواء تجريبيا تحت تأثير ظروف مستقرة. تم اختبار نوعين من المكثفات وهي: مكثف لفائف حلزونية من قطر لفائف مختلفة لنسبة لقطر الأنبوب ومكثف يتواجد السلك على الأنبوب. أجريت التجارب للتدفق المضطرب خارج سطح اللفائف الحلزونية وكذلك لمكثف السلك على الأنبوب وتم تحليل المقارنة بينهما. أظهرت النتائج أن انخفاض نسبة القطر وكذلك المسافة بين اللفائف الحلزونية يؤدي إلى زيادة في معامل الانتقال الحراري الخارجي. كان معامل الانتقال الحراري في الخارج لمكثف اللفائف الحلزوني أعلى بمقدار 12.42% بالمقارنة مع المكثف التقليدي ذو السلك على الأنبوب. ولوحظ أيضا أن (COP) في الثلاجة المحلية المصممة بطريقة (HCC) أعلى بنسبة 12.77% من (WOTC) الحالية.

An experimental investigation of natural convection heat transfer over outer surface of vertical helical coil condenser

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ABSTRACT

Free convection heat transfer over the outer surface of helical coil condenser (HCC) and existing wire-on-tube condenser (WOTC) in air was experimentally studied under steady-state conditions. Two types of condensers are tested: namely, helical coil condenser of different coil diameter to tube diameter ratio and an existing wire-on-tube condenser. The experiments were conducted for turbulent flow outside, the helical coil surface as well as, conventional wire-on-tube condenser surface and comparative analysis was done. The experimental results show that decrease in diameter ratio as well as pitch leads to increase in outside heat transfer coefficient. The heat transfer coefficient was higher 12.42% outside for the helical coil condenser in comparison with the conventional wire-on-tube condenser. It was also observed that the COP of the domestic refrigerator with designed HCC is 12.77 % higher than existing WOTC.

NOMENCLATURE

D	Coil diameter, m
d	Tube diameter, m
g	Acceleration due to gravity, m/s^2
h	Heat transfer coefficient, W/m^{2}K
k	Thermal conductivity, W/mK
L	Length of tube, m
$\dot{m}$	Mass flow rate of water, l/min
Nu	Nusselt number
P	Coil pitch, m
P_C	Pressure drop in condenser, Pa
Pr	Prandtl number,
Ra	Rayleigh number, $[g\beta L^3 \nu^{-1} \alpha^{-1}(t_{as}-t_a)]$
t	Temperature, °C
T	Mean film temperature, °C

Greek symbols

α	Thermal diffusivity, m^2/s
β	Coefficient for thermal expansion, K^{-1}
μ	Dynamic viscosity, Pa.s
ν	Kinematic viscosity, m^2/s

Subscripts

a	Ambient
i	Inner/inside
o	Outer/outside
as	Average condenser surface
s	Condenser surface

INTRODUCTION

Natural convective heat transfer from helical coil condenser having vertical orientation, is pertinent to some heating and cooling manufacturing applications, namely, refrigeration and air conditioning units, process plants, nuclear power plant, heat recovery system, power plant, and so forth. Helical coil heat exchanger is a device used to transfer heat between two or more fluids for numerous industrial applications. Liu and Sakr (2006), Yildiz *et al*. (1996) and Liebenberg and Meyer (2007) classified enhancements of heat transfer techniques into two categories. The active technique is one which involves some external power inputs for the enhancement of heat transfer, whereas the passive techniques do not require such external power for enhancement, but this method commonly uses geometrical modification to the flow channel. One of the most imperative passive techniques is the use of helical coil tubes. According to the investigators Futagami *et al*. (1998), Patankar *et al*. (1974), Prabhanjan *et al*. (2002), Mao *et al*. (1010), Korane *et al*. (2012), and Kumbhare *et al*. (2012), helical coil heat exchangers are widely used in industrial applications because of the compact structure and high heat transfer coefficient. The flow alteration in helical coil is due to centrifugal forces. Ali (1994, 1998, 2004, and 2006) conducted a series of experiments, and the results confirm that the average heat transfer coefficient increases as the number of turns in helical coil decreases for a fixed diameter ratio. The other comparable work studied how to examine the average Nusselt number for oil that was found higher than that of water with the same Grashof number.

Thermal performance and pressure drop of the helical coil heat exchangers, with and without helically crimped fins, have been reported by Naphon (2007). It was found that an average heat transfer rate increases as the mass flow rate of hot and cold water increases. It was also observed that the friction factor decreases with increasing the hot water mass flow rate. Janssen and Hoogendoorn (1978) concluded that the effect of the overall heat transfer coefficient, in case of a constant wall temperature and a constant average heat flux boundary condition, is little. Zachar (2012) studied the case of fluid to fluid boundary condition experimentally and found that the outside heat transfer rate somewhat depends on the inner flow rate of any helical tube in case of increasing temperature differences between the bath fluid temperature and the coil inlet temperature. Ghorbani *et al*. (2010) experimentally pointed out the equivalent diameter of shell as the best characteristic length. Rabin and Korin (1996) developed a mathematical model for thermal analysis of helical heat exchanger for long-term thermal energy storage in soil for use in arid zones. Amori (2014) examined two types of helical coiled tube, a vertical single helical coiled tube and vertical meshed helical coils. The results show that meshed coils have higher heat transfer as compared to a single coil, and also the pressure drop for the meshed coils was found lower than that for the single coil, which lowers the pumping power requirement.

Prabhanjan *et al*. (2004) conducted natural convection heat transfer from helical coiled tubes and investigated the correlation between the outside Nusselt number and the Rayleigh number using characteristic lengths at a constant water bath temperature. Rennie *et al*. (2005) tested a double-pipe helical heat exchanger and calculated the overall heat transfer coefficients. The inner tube and the annulus heat transfer coefficient were found using Wilson plots technique, and the results show that the experimental data fit with the numerical data for larger coils. Rennie *et al*. (2006) numerically studied how to obtain the effect of fluid thermal properties on the heat transfer for double-pipe helical coil. Mohammed (2011, 2009) reported that the overall heat transfer coefficient and Nusselt number increase when the flow rate of the coolant and curvature ratio increases. Neshat *et al*. (2014) conducted both experimental and numerical studies for unsteady natural convection heat transfer from the outer surface of helical coils. The results illustrated that the shell side fluid temperature is a function of the mass flow rate in the tube, fluid specific heat, and geometrical parameters. Al-Hajeri *et al*. (2007) and Lin *et al*. (2007) found that the heat transfer coefficient and overall heat transfer coefficient of refrigerant side decrease as the saturation temperature increases, and the pressure drop in helical tubes increases as the mass flux of refrigerant increases. Jayakumar *et al*. (2008) conducted experimental and CFD estimation of heat transfer in helically coiled heat exchangers, and the correlation was developed to compute the inner heat transfer coefficient of the helical coil.

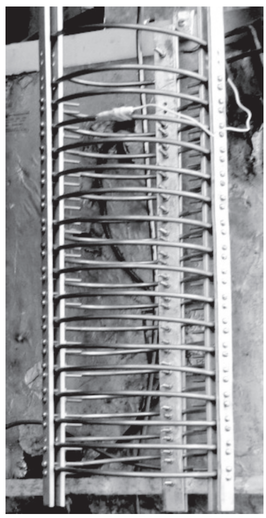
Although there are a number of works available on the outside natural convection heat transfer, there is still room to discuss whether it gives a reliable prediction on the outside heat transfer coefficient of helical coil condenser involving different configurations. The objective of this experimental study is to evaluate the outside convective heat transfer coefficient over vertical helical coiled tube and compare the result with the value obtained for the wire on-tube condenser with the same length and tube diameter.

Table 1. Experimental studies on heat transfer through helical coils

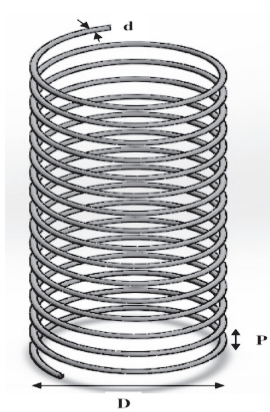
Authors	Working fluid	Configuration	Conditions	Observations
Prabhanjan *et al*. (2002)	Water	Helical coil and straight tube	8300≤ Re≤ 41400, Pr ≤ 0.05	Helical coil heat exchanger increases heat transfer as compared with straight tube.
Mao *et al*. (2010)	Water	Helical coil	40000≤ Re ≤500000	Heat transfer coefficients of the helical tube were close to those of the straight tube under the same flow condition.
Ali (2006)	Oil	Vertical helical coil tube	250 ≤ Pr ≤ 400	The average Nusselt number for oil is higher than that for water at the same Grashof number.
Ali (1994)	Water	Vertical helical coil tube	10^{11}≤Ra≤ 10^{15}	Heat transfer coefficient varies with tube diameter.
Ali (2004)	Glycerol-water	Vertical helical coil tube	Steady-state	Heat transfer coefficient is enhanced either by reducing either the diameter ratio (D/d_o) or the number of coil turns.
Ghorbani *et al*. (2010)	Water	Vertical helical coil in shell	Mixed convection	The reduction of outside heat transfer by increasing the coil surface area.
Amori (2014)	Water	Vertical single coil tube and meshed triple coiled tubes	2.67 ≤ṁ≤ 7.08 l/min	A considerable amount of heat transfer increase in meshed coil tube compared with single coil tube.
Prabhanjan *et al*. (2004)	Water	Vertical coil in bath	12000 ≤ Ra ≤ 27000	The predicted outlet temperatures were close to the experimental values.
Rennie *et al*. (2005)	Water	Double-pipe helical heat exchanger	0.1≤ṁ≤0.9 l/min Wilson plots technique	There was a deviation obtained for the small coil but the experimental data fit well with the numerical data for the larger coil.
Mohammed (2009)	Water	Vertical helical coil	Free convection 4800 ≤De≤ 5700	Enhancement of the overall heat transfer coefficient as the flow rate of coolant and curvature ratio increases.

MATERIAL AND METHODS

The test section was constructed from a copper tube of an inner diameter 6 mm and length 9 m, and aluminium plated fins of thickness 3 mm. The physical dimensions of four helical coil condensers used in the present study are shown in Table 2. A helical coil condenser consists of soft copper tubing, screw and nut, and aluminium plated fins. The helical coil condenser was constructed initially from straight copper tubing with the help of a pattern. The tube was filled with fine sand to preserve the smoothness of the inner surface and was flushed with compressed air. Three pairs of aluminium plated fins were tightened on the helical coil by screw and nut for maintaining helical pitch.



(a)



(b)

Figure 1.(a) HCC used in the present experiment; (b) 3D model of helical coil condenser.

Table 2. Physical dimensions of helical coil condensers used in the experiment.

Condensers used in the experiment	L (m)	d_i (m)	d_o (m)	D (m)	P (m)	D/d_o (-)
WOTC	9	0.006	0.00635	-	-	-
HCC-1	9	0.006	0.00635	0.225	0.055	35.433
HCC-2	9	0.006	0.00635	0.225	0.030	35.433
HCC-3	9	0.006	0.00635	0.175	0.055	27.559
HCC-4	9	0.006	0.00635	0.175	0.030	27.559

Experimental setup and procedure

Figure 2 shows a schematic diagram of the experimental setup. A domestic refrigerator of gross capacity 165 litres was selected for the experimental apparatus working on vapour compression refrigeration system. The existing wire-on-tube was replaced by the helical coil condenser in between the compressor and dryer (filter). The sufficient amount of refrigerant 134a was charged in the experimental setup. In this experiment, five digital thermometers were used. Out of these, one was for measuring ambient room temperature, and the remaining four were fixed on the surface of each condenser as shown in Figure 3. All digital thermometers were calibrated prior to installation with an accuracy of $\pm 1^{O}C$. The pressure gauge (make-Think Auto) was used in this experiment, measures up to 200 psi, and also reads from 0 to 30 psi for vacuum.

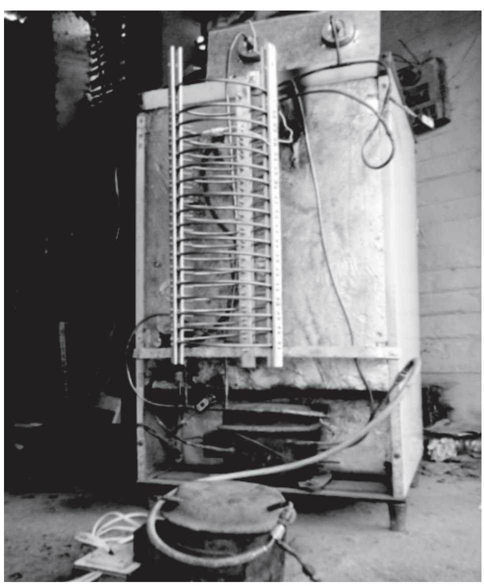


Figure2. Schematic diagram of experimental setup

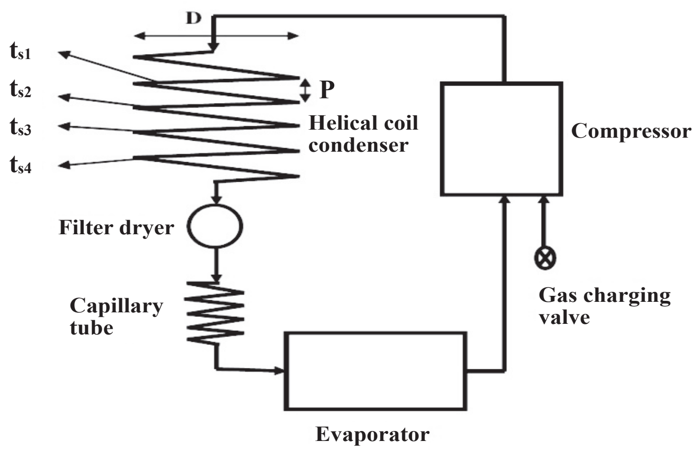


Figure3. Flow diagram of the experimental set-up.

A series of experiments (one existing wire-on-tube and four helical coil condensers) were conducted in a closed room maintained at constant temperature (31.9 ± 0.1^{O}C) with the help of an external heating and cooling system by varying the configuration of the helical coil. For each experiment, the helical coil condenser was installed in between the compressor and the dryer (filter), and experimental setup was allowed to run for 2 hours at a constant flow rate of refrigerant inside the condenser initially set. The surface temperatures and ambient temperature of the existing condenser as well as each helical coil condenser were recorded after steady-state conditions were attained.

ANALYSIS OF EXPERIMENT

In the present study, the physical properties of air flowing outside the condenser tube test section are assumed to be constant along the length of coil. The thermo-physical properties of air are obtained as of polynomial function of temperature Kays *et al*. (2005).

The density of air is calculated from

$$\sigma_{air} = 1.076{\times}10^{-5}T^2 - 1.039{\times}10^{-2}T + 3.326 \quad (1)$$

The viscosity of air is defined by

$$\mu_{air} = 5.21{\times}10^{-15}\,T^3 - 4.077{\times}10^{-11}T^2 + 7.039{\times}10^{-8}T + 9.19{\times}10^{-7} \quad (2)$$

The thermal conductivity of air is defined by

$$k_{air} = 4.084{\times}10^{-10}T^3 - 4.519{\times}10^{-7}T^2 + 2.35{\times}10^{-4}T - 0.0147 \quad (3)$$

The specific heat of air is given by

$$c_{p,air} = -4.67{\times}10^{-6}T^3 + 4.837{\times}10^{-3}T^2 - 1.599T + 1175 \quad (4)$$

Once the physical properties of air were calculated from Equations (1-4) at mean film temperature, $\theta = \frac{t_{as}+t_a}{2}$, where $t_{as} = \frac{t_{s1}+t_{s2}+t_{s3}+t_{s4}}{4}$ is the average surface temperature outside condenser tube and t_a is the ambient temperature. The outside Nusselt number , was then calculated using the empirical correlation for free convection turbulent flow from Equation (5), Kothandaraman and Subramanyan (2010).

$$\mathrm{Nu_o} = 0.10(\mathrm{Ra})^{1/3}, \qquad 10^9 > Ra < 10^{13} \tag{5}$$

where Ra is the Rayleigh number. The Nusselt number was then used to calculate the outside heat transfer coefficient based on the tube length as characteristic length.

The uncertainties are given in Table 3, which were calculated from Equation (6) given below:

$$w_R = [\left(\frac{\partial R}{\partial x_1} w_2\right)^2 + \left(\frac{\partial R}{\partial x_2} w_2\right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n} w_2\right)^2]^{1/2} \tag{6}$$

where R is the calculated result of the experiments, w_R is the uncertainty in the result. The result R is a given function of the independent variables x_1, x_2, … , x_n. The uncertainties in the independent variables are w_1, w_2, … , wn.

Table 3. The uncertainties of calculated values.

Calculated value	Uncertainty (%)
t_a	3.279
t_{as}	3.22
Pc	0.25

RESULTS AND DISCUSSION

Experiments have been carried out on the existing wire–on-tube condenser and helical coil condensers with varying geometry. The results obtained from experiments were analysed. The results cover the Rayleigh number over a range of 3.02428×10^{11} to 4.10744×10^{11}, Prandtl number of around 0.7002, and diameter ratio range of 35.433 to 27.559. The ambient temperature is conditioned to approximately 39.9^{0}C for all the experiments.

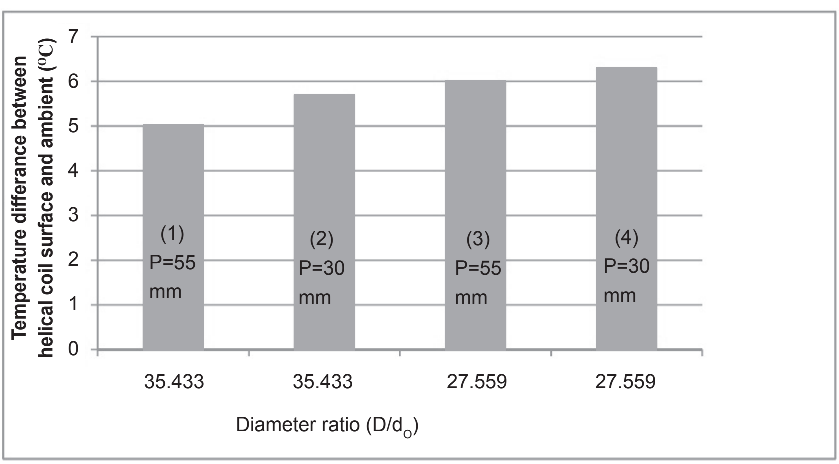


Figure 4. Difference between surface and ambient temperature versus diameter ratio (D/do).

The performance of the designed helical coil condenser in air was examined for constant refrigerant flow rate inside the helical coil condenser. Figure 4 shows that if the diameter ratio decreases, the difference between helical coil exposed surface and ambient temperature increases for the same pitch. The diameter ratio of HCC-1 and HCC-2 is equal to 35.433, whereas, for HCC-3 and HCC-4, it is equal to 27.559. It was observed that as the pitch of the coil decreases for the same diameter ratio (D/do), the difference between exposed surfaces and ambient temperature increases. The difference between helical coil surfaces and the ambient temperature is greatly influenced by the diameter ratio (D/do) and pitch of the helical coil.

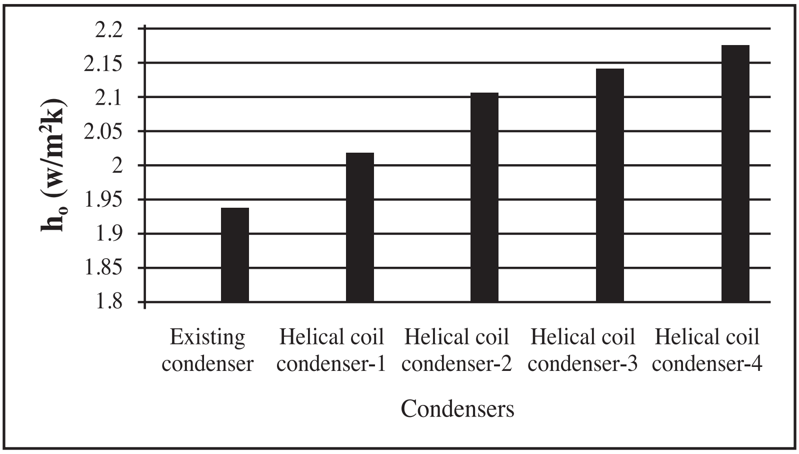


Figure 5. Outside heat transfer coefficient for all five condensers.

A bar diagram representation of the outside heat transfer coefficient for the existing condensers WOTC, HCC-1, HCC-2, HCC-3, and HCC-4 on the basis of experimental results is shown in Figure 5. This shows the comparative analysis of all five condensers. The diameter ratio (D/d_o = 27.559) in case of helical coil condenser-3 and helical coil condenser-4 gives a higher value of the outside heat transfer coefficient than the diameter ratio (D/d_o = 35.433) used in case of helical coil condenser-1 and helical coil condenser-2. When compared with the existing condenser (i.e., WOTC), the increase in the outside heat transfer coefficients was found to be 4.27 %, 8.83 %, 10.67 %, and 12.42 % for HCC-1, HCC-2, HCC-3, and HCC-4, respectively. The experimental pressure drop in HCC-4 was found to be 5.07%, which is higher than that of the existing WOTC.

CONCLUSIONS

The results show that a decrease in diameter ratio increases the outside heat transfer coefficient. With a decrease in pitch while keeping the diameter ratio constant, it was found that the outside heat transfer coefficient increases. The outside heat transfer coefficient of the used helical coil condenser was found to be 12.42 % more as compared to that of the existing wire-on-tube condenser. It was observed that the amount of the outside heat transfer coefficient of HCC-4 was the highest among all. Its value was 1.58 %, 3.29%, and 7.81%, respectively for HCC-3, HCC-2, and HCC-1. It was found that the COP of the domestic refrigerator with designed the HCC is 12.77 % higher as compared to that of the existing WOTC.

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