Study the Performance of a Domestic Refrigerator with Dual-Loop Cycle Flow Condenser and Comparison with Wire-on-Tube Condenser

Dr. Dheya Ghanim Mutasher
Mechanical Engineering Department, University of Technology/Baghdad.
Email: dheya_ghanim@yahoo.com

Received on:30/6/2015 & Accepted on:17/12/2015

ABSTRACT

This paper presents an experimental investigation into the performance of domestic refrigerator system with two different types of condensers, novel design of dual-loop condenser and the conventional condenser. The geometrical configuration considered in this study is representative designing and manufacturing of novel type of dual-loop cycle condenser and compared with typical condenser. The experiments were conducted with a refrigerator designed to work with HFC134a, under no load and with different loads as (1.5, 3 and 12 litters of water). The results showed that the dual-loop condenser decreases the work done by the compressor by about (1.951% to 5.257%). Also, the effects of shape change of the condenser can be reducing the pressure drop as result of reduction the frictional loss where can rotate the flow without restriction to the flow in the dual-loop condenser

Keywords: Household Refrigerator, Dual-Loop Condenser, Wire-on-Tube Condenser, Temperature Distribution.

دراسة اداء ثلاجة منزلية مع مكثف ثنائي الحلقة ومقارنته مع مكثف سلكي

الخلاصة

تمثل هذه الدراسة بحث تجريبي في اداء نظام الثلاجة المنزلية مع نوعين مختلفة من المكثفات ، تصميم جديد من مكثف ثنائي الحلقة والمكثف التقليدي . الشكل الهندسي المطبق في هذه الدراسة يمثل تصميم وتصنيع نوع جديد من مكثف ثنائي الحلقة ومقارنته مع المكثف التقليدي. التجارب كانت تعمل على ثلاجة مصممة للعمل مع (HFC 134a). وتحت احمال متغيرة (بدون حمل ، ١.٥، ٣ ،١٢ ليتر من الماء). توضح النتائج ان المكثف ثنائي الحلقة يقلل الشغل المنجز من قبل الضاغط الى حوالي (١.٩٥١% الى ٥.٢٥٧%). كذلك، تاثير تغيير شكل المكثف يمكن ان يقلل الهبوط بالضغط كمحصلة لتقليل خسائر الاحتكاك حيث يمكن تدوير الجريان بدون اعاقة للجريان في المكثف ثنائي الحلقة.

INTRODUCTION

Condensers are basically heat exchangers in which the refrigerant undergoes a phase change. The optimal design and selection of condensers are very important for satisfactory performance of any refrigeration system. In the present paper, a dual-loop heat exchanger as elliptical shape without fins with natural air movement will be considered. There is working fluid such as refrigerant flowing inside the tubes, while the ambient air is directed across the outside surface of the tube panel.

Witzell and Fontaine [1] studied the condenser and found that any additional of wire on the external surface of condenser did not increase the rate of heat transfer from condenser. Hoke et al. [2] investigated experimentally the air side convective heat transfer for wire-on-tube heat exchangers. They Also studied the importance of angle of attack for locating the wire-on-tube heat exchangers which cooled by forced convection. Tanda et al. [3] investigated a Nusselt

https://doi.org/10.30684/etj.34.3A.12
2412-0758/University of Technology-Iraq, Baghdad, Iraq

number and operating parameters for predicting the free convection heat transfer from a wire-on-tube condenser. Quadir et al. [4] used finite element method to analyze the free convection heat transfer of wire-on- tube heat exchanger. The effects of the ambient temperature and the inside refrigerant mass flow rates on the heat transfer performance were considered. The required tube length for phase change from its initiation and completion was also determined. Islamoglu [5] showed the application of neural network model for predicting the heat transfer rate of wire-on-tube heat exchanger. Hayati et al. [6] used an adaptive neuro-fuzzy inference system model for prediction of heat transfer rate of the wire on tube type heat exchanger. Ahmed and Hayder [7] introduced the present modeling of wire and tube condensers that commonly used in a vapour compression cycle based on domestic refrigeration. The modeling results showed the effect of mass flow rate, pressure, refrigerant temperature and ambient temperature on the performance of condenser. Raiyani et al. [8] studied the effect of change in design of the condenser by changing the point contact between the tube and plate by line contact with wrapping the plate on tube. Samana et al. [9] found the enhancement of heat transfer when the oscillating heat pipe was used as the extended surface. Khlief A.K [10] studied the possibilities of replacing the refrigerant R-12 by R-134a and compared their thermal performance when working with the following variables (air cooling condenser, water cooling condenser, changing the length of the capillary tube). The results shows that in the system worked with refrigerant R-12 and air or water cooled condenser the decreasing of the length of the capillary tube from 1.92m to 1.44m will lead to increase the coefficient of performance, whereas the increasing of the length to 2.4 will lower the coefficient of performance.

The previous works used conventional heat exchanger, solid metal wire as an extended surface.

In this study, a new design is introduced by manufacturing refrigerator condenser as the dual-loop condenser. The performances of a dual-loop heat exchanger and the wire-on-tube heat exchanger have been investigated.

Analysis of the condenser

Conventional condenser consists of a steel tube bended into a single-passage serpentine shape, with wires spot welded perpendicularly on both sides, as shown in figure (1a). Condensers may be assembled with tubes in a vertical or horizontal position and the air movement can be forced or natural. From the heat transfer point of view, the exchanger is assumed to be made up of a multiplicity of horizontal tubes, with the wires acting as extended surfaces to increase the heat transfer from the wall to the external environment. The pipe bend pressure drop occurs in wire-on tube condenser causing the frictional pressure drop.

In the present study, the novel design taking the dual-loop condenser instead of a wire-on-tube has been presented. As the compressor provide the vapor of refrigerant with high pressure and temperature to the condenser, the vapor starts up and down with the first loop from the outer to the inner and continues rotating moving in the rest of loops and after that repeats the same behavior with the flow in the second loop and finally delivers the flow to the capillary tube. However, for a better understanding of the work on the dual-loop condenser, see the figure (1b).

Heat transfer rate at evaporator or refrigeration capacity, Q_E is given by:

$$Q_E = m\,(h1 - h4) \quad \dots (1)$$

Where, m is the refrigerant mass flow rate in kg/s, h1 and h4 are the specific enthalpies (kJ/kg) at the exit and inlet to the evaporator, respectively. (h1−h4) is known as specific refrigeration effect or simply refrigeration effect, which is equal to the heat transferred at the evaporator per kilogram of refrigerant.

Power input to the compressor or work of compression W_C is given by:

$$W_C = m\,(h2 - h1) \quad \dots. (2)$$

Where, h2 and h1 are the specific enthalpies (kJ/kg) at the exit and inlet to the compressor, respectively. (h2 − h1) is known as specific work of compression, which is equal to the work input to the compressor per kilogram of refrigerant.

Heat transfer rate at condenser, Q_C is given by:

Q_C = m (h2 − h3) … (3)

Where, h3 and h2 are the specific enthalpies (kJ/kg) at the exit and inlet to the condenser, respectively.

For the isenthalpic expansion process, h=Const. is given by:

h3 = h4 … (4)

The exit condition of the expansion device lies in the two-phase region, hence applying the definition of quality (or dryness fraction), we can write:

h4 = (1 − x4) hf,e + (x4*hg,e) = hf +(x4*hfg) …. (5)

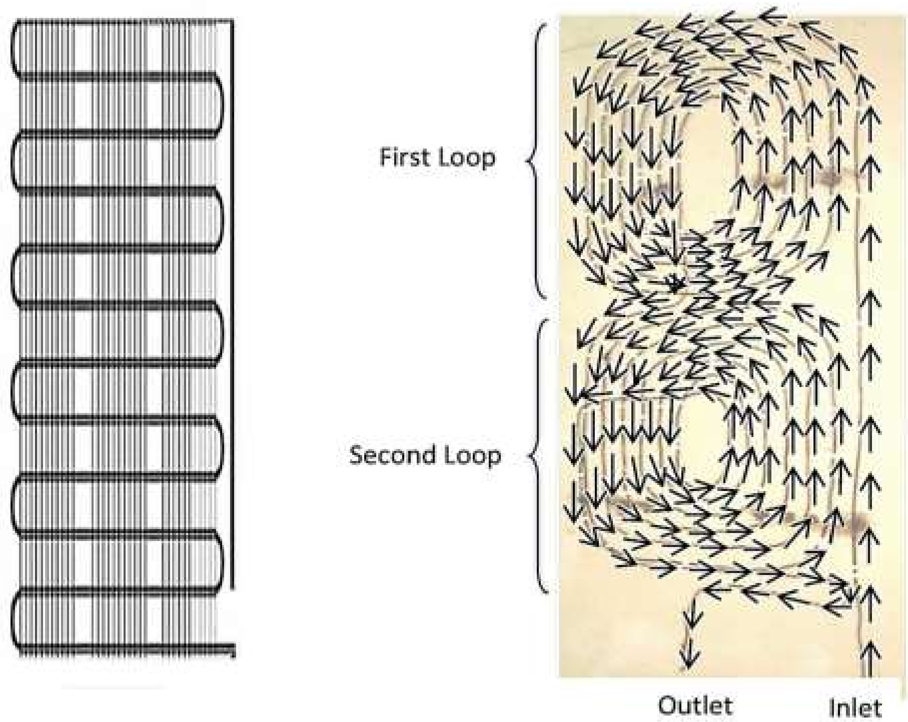
For mass flow rate, [m] is given by: Measurement of the electrical energy input, E allows the mass flow rate to be determined from the simple relationship:

$E – H = (h_{out} - h_{in}) * m$ …(6)

Where h are the enthalpy values, per unit mass of the refrigerant. These values are known from the temperature and pressure measurements through the inlet and outlet of the compressor. The heat loss H, expressed as a percentage of E, is between 5 and 7% for most compressor types.

The coefficient of performance, [C.O.P] is given by:

$C.O.P = Q_E / W_C$ …(7)



(a) (b)

Figure (1): (a) Schematic of a wire-on-tube condenser and (b) dual-loop condenser

Experimental Setup and Apparatus

The refrigerator was instrumented with pressure gauge at the inlet and outlet of the compressor, as well as one pressure gauge at the outlet of the condenser for measuring pressure drop through the condenser. (8) K-type thermocouple wires were used for measuring the surface temperatures of the condenser after calibrating these thermocouples connected to data logger type pico. When reaching the steady state condition, the measured data saved as text in the data logger. The experimental setup of the test unit and apparatus is shown in Figure 2. The system was evacuated from air and humidity using vacuum compressor before charging the system. First, the system was pressurized and then checking the leak was performed by bubble test. After that the system is ready for operation. As per the refrigerator manufacturer recommendation, quantity of the required charge for HFC134a is 140 g. In the experiment, refrigerant charge is 10% higher due to the presence of instruments and connecting lines. The refrigerator specifications are given in table 1.

During the experimentation, the temperature inside test room was 30 ± 2^oC. The experiments were conducted on the domestic refrigerator at different load conditions namely, (no load, 1.5, 3 and 12 Litter of water). The digital energy meter was used to measure energy consumption of the system. In the first, the performance of the refrigerator with the conventional condenser was measured and after that, the condenser was replaced by a dual-loop condenser, then the test results of two condensers were compared. The Schematic diagram of the experimental work is shown in Figure 3.

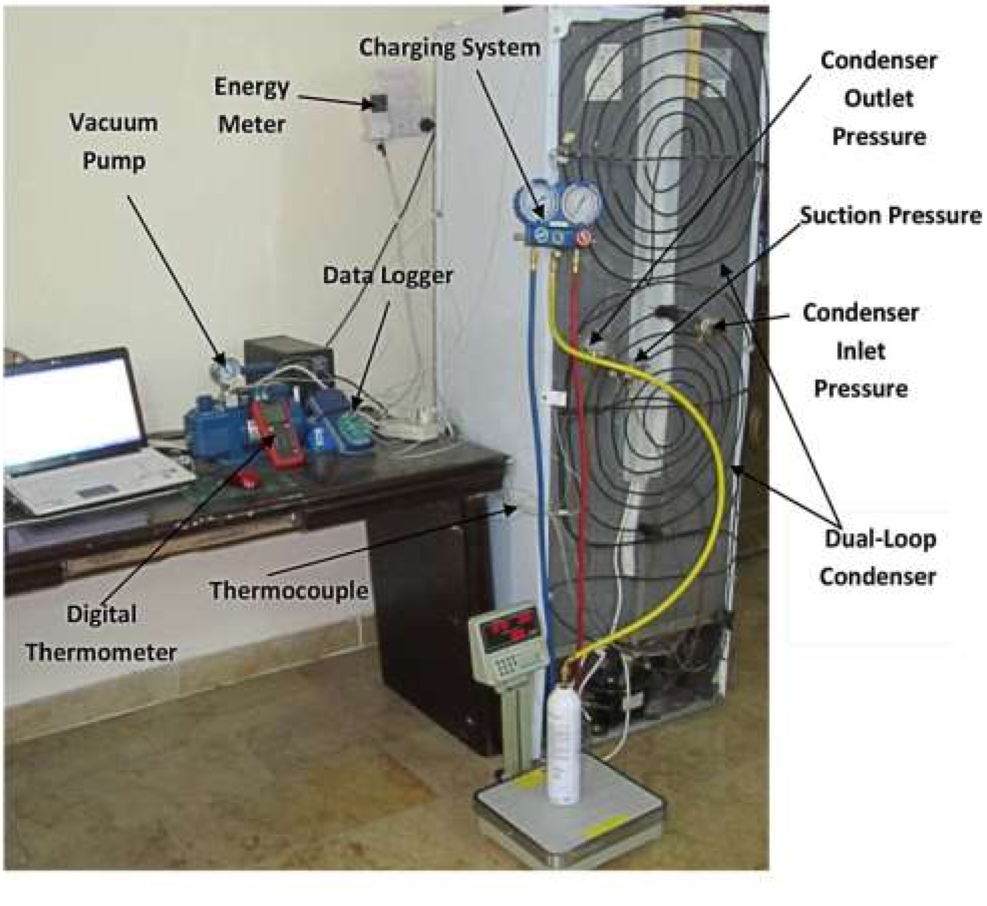


Figure (2) Experimental Setup of the Test Unit and Apparatus

Table (1): Specifications of Refrigerator Test Unit

TETN1600	Concord
Voltage (V), current (A) and frequency (Hz)	220V , 0.8A and 50Hz
Gross capacity	190L
Compressor type	Hermetic
Condenser Types	Wire-on-Tube and Dual-Loop Condenser
Refrigerant type	HFC134a
No of door	2
The charged mass	0.140 kg
Total length of the tube	19.25 m
Tube Diameter	4 mm
Tube Distance (pitch)	3.5 cm

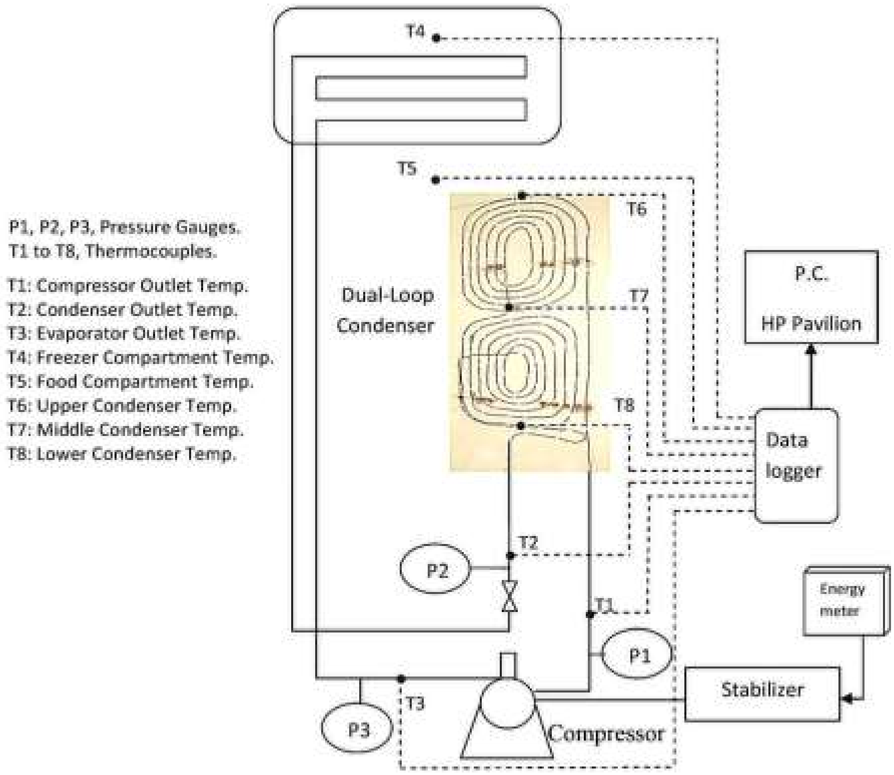
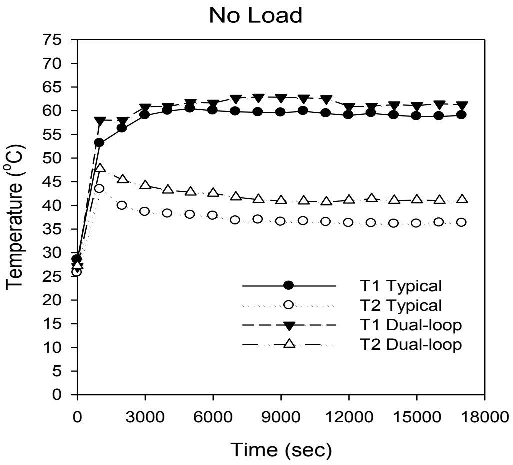


Figure (3)Schematic Diagram of the Test Unit and Apparatus

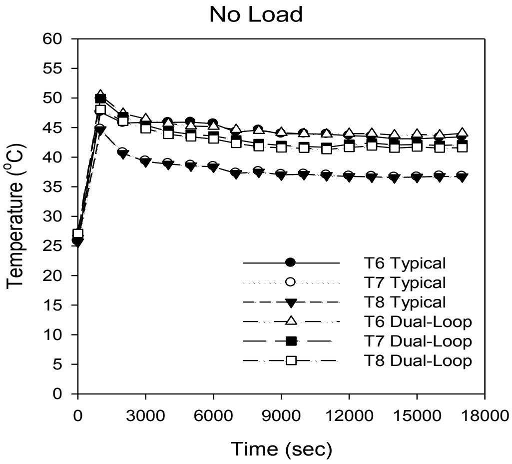
Results and Discussions

Figure 4 (a to d) shows condenser temperature at the inlet and outlet for two types, conventional and dual-loop condenser with time. The temperature difference through the dual-loop condenser is less than the conventional condenser for all load conditions, and that is because of the developed turbulent flows that increased the dissipation of heat in the tube and increasing the velocity of the flow as a result of changing the shape of the condenser to dual-loop.

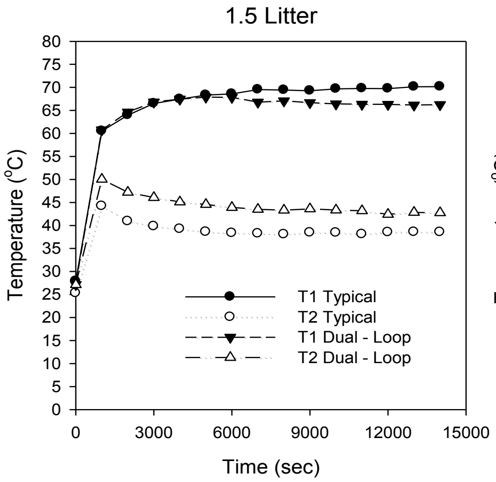
Figure 4 (e to h) manifests the condenser surface temperature for two types conventional and dual-loop condenser with time. The surface temperature of the condenser for all load conditions was non converges for the conventional condenser while in dual-loop condenser the temperature at these locations converges that because the flow characteristics are the same in all length of the tube of the dual-loop condenser.



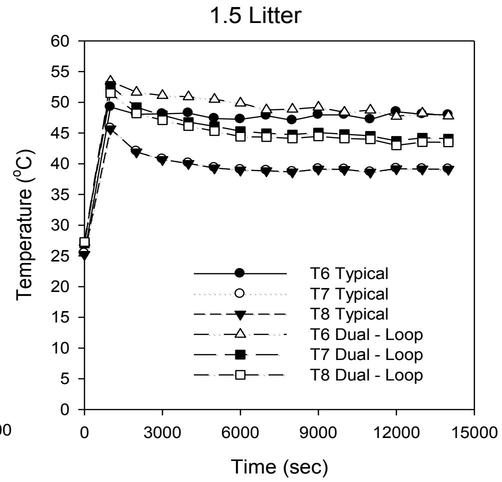
(a)



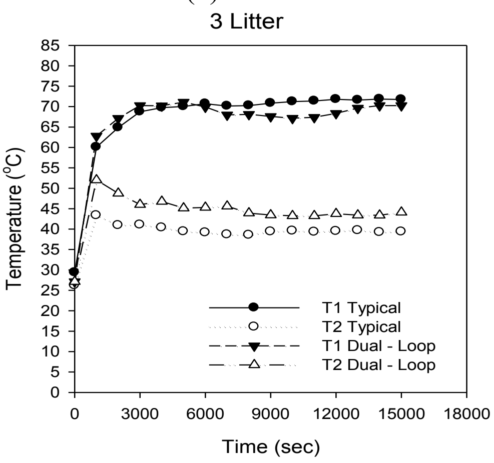
(e)



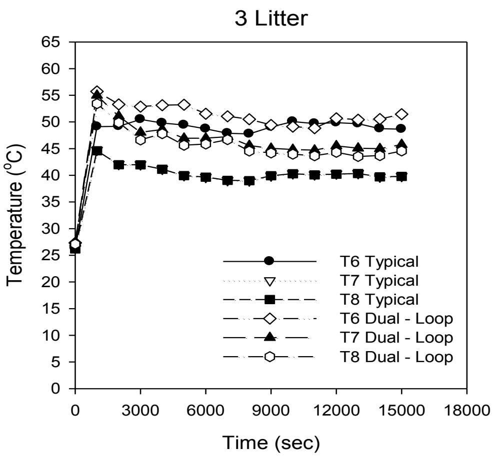
(b)



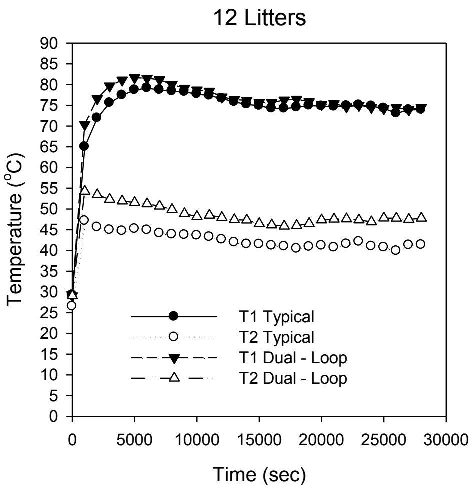
(f)



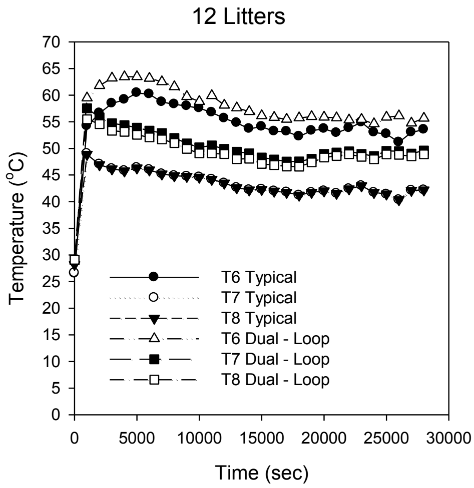
(c)



(g)



(d)



(h)

Figure (4): (a, b, c and d) the variation of temperature with time for inle and outlet of the condenser and (e, f, g and h) the Variation of the Average Temperature at the Surface of the Condenser (T6,T7 and T8) with all loads

Figure 5 illustrates the inlet and outlet temperature of the condenser for two types conventional and dual-loop condenser with steady state condition. This figure shows that the difference between the inlet and outlet temperature of the condenser is smaller for dual-loop condenser than the conventional condenser. Although the inlet temperature to the condenser at middle loaded higher for conventional but the difference between them was very low.

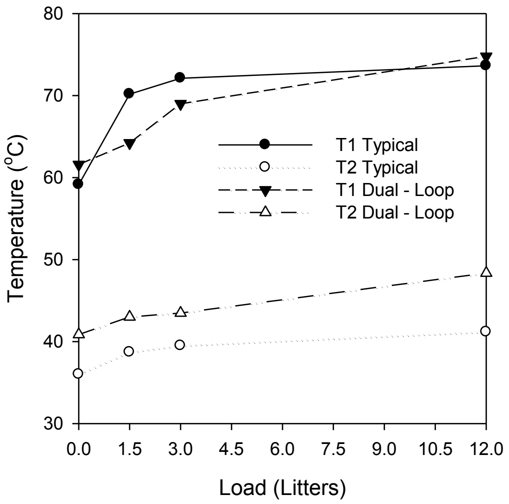


Figure (5): Variation the Inlet and Outlet Temperature of the Condenser with Load

Figure 6 reveals the average condenser temperature for two types of conventional and dual-loop condenser at a steady state condition. This figure shows the average condenser temperature was greater for the dual-loop condenser than typical condenser for all load conditions especially at high load. While the dual-loop condenser was without fins which can be reduced the heat rejected.

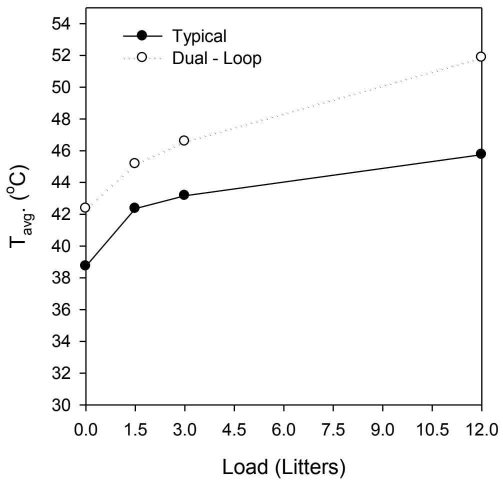
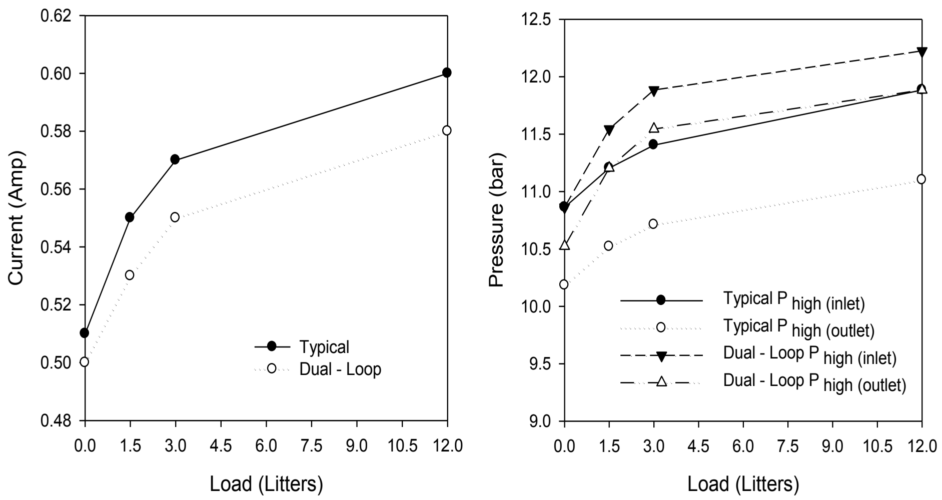


Figure (6): Variation of the Average Temperature at the Surface of the Condenser with Load

Figure 7 (a) and (b) illustrates the comparison of the current and pressure for two types typical and dual-loop condenser type with a steady state condition. On all load conditions, the current was low for the dual-loop condenser compared to the typical condenser. That is owing to the decreases of load on the compressor, the difference between the two types was very low. Also, the pressure drop for dual-loop condenser is less than the typical condenser that because decreases the frictional losses as results of improve shape of the condenser where can rotate the flow without restricted to the flow in the dual-loop condenser.



(a) (b)

Figure (7): (a) and (b) Variation of the current and the pressure with all loads

Figure 8 illustrates that the compressor work with dual-loop condenser was lower than that of typical condenser, especially at high load but the difference between them was very low. That is because in the dual-loop condenser can rotating flows easily which can reduce the load on the compressor. Although the dual-loop condenser was without fins, the coefficient of performance was close of the typical condenser. Table (2) shows the difference between the performances of two types.

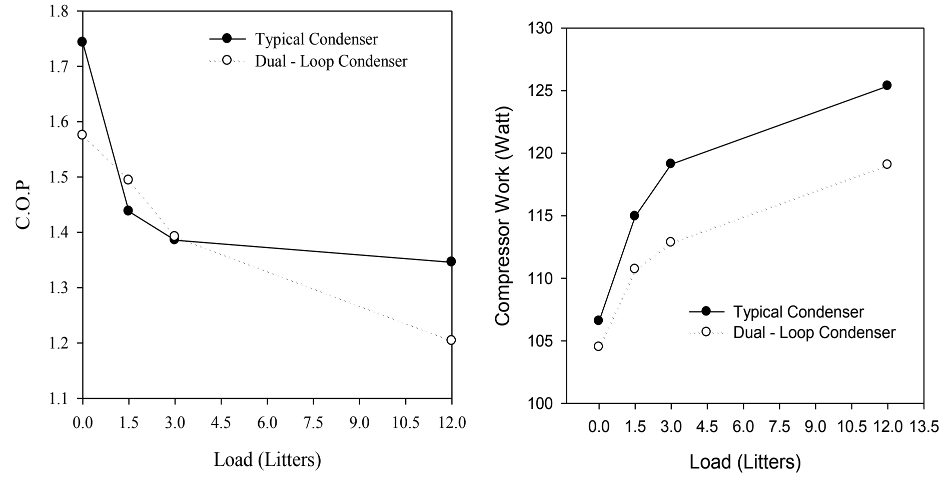


Figure (8): Variation the Compressor Work and the coefficient of performance with all Load

Table (2): The performance of Dual-Loop Condenser and Typical Condenser

	Typical Condenser (With Fins)		**Dual-Loop Condenser (Without Fins)**
No-load		**No-load**	
$Q_{(Evap.)}$	185.794 Watt	$Q_{(Evap.)}$	164.577 Watt
$Q_{(Cond.)}$	292.363 Watt	$Q_{(Cond.)}$	269.066 Watt
$W_{(Comp.)}$	106.569 Watt	$W_{(Comp.)}$	104.489 Watt
C.O.P	1.743	C.O.P	1.575
1.5 litter		**1.5 litter**	
$Q_{(Evap.)}$	165.348 Watt	$Q_{(Evap.)}$	165.431 Watt
$Q_{(Cond.)}$	280.287 Watt	$Q_{(Cond.)}$	276.137 Watt
$W_{(Comp.)}$	114.939 Watt	$W_{(Comp.)}$	110.706 Watt
C.O.P	1.438	C.O.P	1.494
3 litter		**3 litter**	
$Q_{(Evap.)}$	165.167 Watt	$Q_{(Evap.)}$	157.158 Watt
$Q_{(Cond.)}$	284.280 Watt	$Q_{(Cond.)}$	270.010 Watt
$W_{(Comp.)}$	119.113 Watt	$W_{(Comp.)}$	112.851 Watt
C.O.P	1.386	C.O.P	1.392
12 litter		**12 litter**	
$Q_{(Evap.)}$	168.807 Watt	$Q_{(Evap.)}$	143.444 Watt
$Q_{(Cond.)}$	294.166 Watt	$Q_{(Cond.)}$	262.500 Watt
$W_{(Comp.)}$	125.359 Watt	$W_{(Comp.)}$	119.055 Watt
C.O.P	1.346	C.O.P	1.204

CONCLUSIONS

The characteristic of the heat transfer from the condenser can be affected by changing geometry of the condenser. From the experiment, the following conclusions can be stated:

1- The work of compressor decreased with dual-loop condenser by about (1.951% to 5.257%).

2- The inlet and outlet temperature difference of the condenser for dual-loop condenser is smaller than the typical condenser.

3- Decreases the frictional losses as results of improved shape of the condenser where flow can rotate without restriction to the flow in the dual-loop condenser.

4- The coefficient of performance of household refrigerator with dual-loop condenser without fins was slightly higher than that of the typical condenser. The range of enhancement was by about (0.431% to 3.748%). While in low and high loads the coefficient of performance of typical condenser higher than that of the dual-loop condenser by about (9.638% to 10.549%).

REFERENCES

[1] Witzell O.W., Fontaine W.E., "What are the heat transfer characteristics of wire and tube condensers", Refrigerating Engineering 65 (1957) , PP. 33–37.

[2] Hoke J.L, Clausing, A. M, Swofford T.D., "an experimental investigation of convective heat transfer from wire-on-tube heat exchangers ", Trans. ASME J. Heat Transfer, (1997), PP.119, 348-356.

[3] Tanda, G., Tagliafico, L, "Free convection heat transfer from wire- and –tube heat exchangers", ASME J. Heat Transfer,1997, PP. 119, 370-372.

[4] Quadir G.A, Krishnan G.M, Seetharamu K.N, "Modeling of wire-on-tube heat exchangers using finite element method" , Finite Elements in Analysis and Design 38, (2002), PP. 417-434.

[5] Islamoglu Y., "A new approach for the prediction of the heat transfer rate of the wire-on-tube type heat exchanger — use of an artificial neural network model" , Applied Thermal Engineering 23, (2003), PP. 243–249.

[6] Hayati M, Rezaei A, Seifi M. , "Prediction of the heat transfer rate of a single layer wire-on-tube heat exchanger using ANFIS", Elsevier Science, International Journal of Refrigeration 32, (2009), PP. 1914–1917.

[7] Imran A.A, Jafal H.M., Numerical Modeling of Wire and Tube Condenser Used in Domestic Refrigerators, Journal of Engineering and Development,Vol. 13, No. 2, June (2009), ISSN 1813-7822, pp 1–16.

[8] Raiyani A, Sheth N. R, Mehta N. C., "thermal analysis of hot wall condenser for domestic refrigerator", International Journal of Science and Research (IJSR), Volume 3, Issue 7, July (2014), PP. 622–626.

[9] Samana T, Kiatsiriroat T, Nuntaphan A., "enhancement of fin efficiency of a solid wire fin by oscillating heat pipe under forced convection" , Elsevier Science, case studies in thermal engineering, 2, (2014), PP. 36–41.

[10] Khlief A.K., "Studying the Process of Rising the Performance of the Cooling System by Using the Two Liquids R-12 &R-134a", Journal of Engineering and Development, Vol. 32, No. 4, April (2014), pp 148–164.