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Effect of Different Compressor Speeds on the Energy and Exergy of an Automobile Air-Conditioner Using R134a in the Absence and Presence of a Liquid-Suction Heat Exchanger

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Energy analysis; Liquid-suction heat exchanger; Refrigeration system; R134a; Thermal performance; Variable speed compressor.

Highlights:

- *Low global warming potential (GWP) and ozone depletion potential (ODP).*
- *Liquid suction Heat exchanger (LSHX).*
- *ANSYS Fluent.*
- *Different compressor speeds.*

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Abstract: Refrigerant R134a has been widely utilized in automotive air conditioning systems (AACSs); R134a has a high global warming potential (GWP) of 1430 despite having zero ozone depletion potential (ODP). Coming refrigeration systems must include refrigerants with low GWP and zero ODP. The aim of this experimental study is to evaluate the thermal performance of an (AAC) with different values of compressor speeds, i.e., (1000, 1700, and 2400 rpm) and two thermal loads, i.e., (500 and 1000 Watt) with the absence and presence of liquid suction heat exchanger (LSHX) using R134a. The results showed that adding LSHX enhanced the COP cycle by 7.18%, 10.7%, and 3.09% for the first, second, and third speed, respectively, at 500 Watt, while the enhancements were 10.27 %, 23.3 %, and 11.5 % for the first, second, and third speed, respectively, at 1000 Watt. Increasing the compressor speed decreased COP due to a reduction in RE and increased the compression effect, increasing the work done by the motor on the compressor that caused a reduction in COP. The compressor exergy destruction ($X_{des.\ Comp.}$) decreased when LSHX was added by 6.13%, 2.22%, and 18.8% for the first, second, and third speed, respectively. However, $X_{des.\ comp.}$ increased with compressor speed due to the system's pressure difference rise because of decreasing evaporation and increasing condensation pressures. As a result, the entropy generation increased. The increase in discharge temperature and pressure of the compressor led to a high friction force between the moving part of the compressor and the refrigerant, so the energy losses increased. Increasing the compressor speed decreased the total exergy performance of the cycle by 5.8 %, 7.5 %, and 16.7 % for the first, second, and third speed, respectively, due to increasing the compressor discharge temperature, increasing the $X_{des.\ comp.}$ and thermostatic expansion device and decreasing condenser and evaporator. Increasing $X_{des.\ comp}$ was higher than the destruction in the condenser and evaporator, which canceled the effect of others, so the total exergy performance of the cycle decreased.

تأثير السرع المختلفة للضاغط على الأداء الحراري لجهاز تكييف سيارة باستخدام R134a في غياب ووجود مبادل حراري بين خطي السائل والسحب

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الخلاصة

يعتبر مائع التثليج المستخدم (R134a) من الموائع الشائع استخدامها في الوقت الحاضر في منظومات تكييف السيارات (AACS) ومنظومات التثليج ويعتبر مائع التثليج مائع مناسب لتقليل من استنفاذ طبقة الأوزون(ODP) والحد من امكانات الاحترار العالمي (GWP) تعتبر منظومات تكييف الهواء ضرورية في الاجواء الحارة والرطبة وذلك للحصول على درجة حرارة مناسبة للمستخدمين داخل المبنى .يجب ان تشمل انظمة التثليج الحالية موائع تثليج ذات قدرة منخفضة التأثير على الاحترار العالمي (GWP) وامكانية استنفاذ طبقة الاوزون (ODP) .الهدف الرئيسي من هذه الدراسة العملية هو تقييم الاداء الحراري لمكيف هواء سيارة (AACS) باستخدام سرع مختلفة للضاغط بوجود وعدم وجود مبادل حراري بين خطي السائل والسحب (LSHX) باستخدام مائع التثليج (R134a). تم تنفيذ هذا العمل التجريبي على مكيف هواء سيارة باستخدام ثلاث سرع للضاغط (١٠٠٠ دورة في الدقيقة ,١٧٠٠ دورة في الدقيقة ,٢٤٠٠ دورة في الدقيقة) مع قيمتين للأحمال الحرارية (٥٠٠ واط ,١٠٠٠ واط) بوجود وعدم وجود مبادل حراري أظهرت النتائج تحسن بإداء منظومة التكييف بصورة عامة للسيارة عند اضافة المبادل الحراري .حيث ازداد معامل الاداء الحراري بنسبة (٧,١٨ %) في السرعة الاولى وكانت الزيادة في السرعة الثانية بنسبة (١٠,٧٪) اما في السرعة الثالثة فكانت الزيادة بنسبة (٣,٠٩٪) عند قيمة الحمل الحراري (٥٠٠ واط) بينما كانت نسبة الزيادة (١٠,٢٧٪) و (٢٣,٣٪) و (١٥,٥٪) عند السرعه الاولى والثانية والثالثه على التوالي عندما كانت قيمة الحمل الحراري (١٠٠٠ واط). عند زيادة سرعة الضاغط فان معامل الاداء الحراري يقل وذلك لزيادة الشغل المسلط على المحرك. عند اضافة المبادل الحراري قلت جودة الطاقة (الاكسيرجي) في الضاغط بنسبة (٦,١٣٪) و(٢,٢٢٥) و(١٨,٨٪) للسرعه الاولى والثانية والثالثة على التوالي ولكن عند ازدياد سرعة الضاغط فان قيمة الطاقة المستنزفة في الضاغط ازدادت وذلك لانخفاض ضغط التبخير وازدياد ضغط التكثيف والذي بدوره يؤدي الى زيادة ضغط وحرارة الضاغط مما يولد قوة احتكاك عالية بين مائع التليج واجزاء المحرك مما يؤدي الى زيادة فقدان الطاقة.

الكلمات الدالة: تحليل الطاقة، سرعة متغيرة للضاغط، منظومات التكييف، مبادل حراري، معامل الاداء الحراري، R134a.

1.INTRODUCTION

The present study experimentally investigated the performance of an automobile air conditioner system (AACS) with a liquid suction heat exchanger (LSHX) and variable speed compressor. The heat exchanger is a device where heat is exchanged between two fluids without mixing [1]. Many studies on adding LSHX in the Vapor compression refrigeration cycle (VCRC) have been established to study the influence of adding LSHX on the thermal performance of automobile air conditioners [2]. Adding a heat exchanger to an automobile air conditioning system might enhance the performance of the cycle [3]. Using nanoparticles as refrigerant additions is a method of improving the vapor compression refrigeration system (VCRS) performance without altering the system components [4], besides using an ejector as an expansion device [5]. One of the techniques found to enhance the efficiency of the vapor compression refrigeration system is sub-cooling using a liquid suction heat exchanger [6]. Ref. [7] conducted an experimental comparative study between the thermal performance of ACs of VCRS with and without LSHX charged with R134a with two compressor speeds. The temperatures at all points of VCRC, i.e., with low and high-pressure points and indoor and outdoor temperatures, were recorded. The results showed improved COP and refrigeration effects when LSHX was installed. An experimental investigation of adding LSHX to an automobile air conditioning system conducted by Ref. [8] with different compressor speeds. The results showed that for compressor speeds of 800–1800 rpm, the percentage of COP with a system working with R1234yf and IHX was decreased by 0.3-2.9%, respectively, i.e., lower than the R134a system. At a compressor speed of 2500 rpm, the COP enhanced by 0.9%. An internal heat exchanger (IHX) is a common modification of the basic cycle to enhance its energy performance [9]. A heat exchanger can be a separate device, such as a heater or condenser, or a component of a device, such as the heat exchangers found in old refrigerators [10]. Using a thermostatic expansion valve (TXV), a coaxial internal heat exchanger (IHX), and refrigerant R1234yf, an automotive air conditioning system (AACS) was created by Ref. [11] with two compressor speeds (1000 rpm and 2600 rpm). The inlet air temperatures were between (30 °C and 40 °C). The test was studied with IHX and without IHX cases. The results showed that when IHX was added, the COP enhanced by 3%, the evaporating temperature decreased by 0.8°C, the cooling capacity increased by 2.2 %, and the compressor power decreased by 2%. The results indicated that the system with IHEX and R1234yf improved system performance. Some additions have been used to increase the performance of LSHX, such as nanofluid that increases the convective heat transfer [12] and metal foam to enhance the heat transfer of double pipe heat exchanger fins [13]. Ref. [12] theoretically studied the effect of adding LSHX to a refrigeration system. The studied heat exchanger comprised 17 tubes with 6mm and 4mm diameters and 300mm and 225mm lengths. The results showed that the 300mm length and 4mm diameter heat exchanger enhanced COP more than other cases.

2.EXPERIMENTAL PROGRAM

2.1.Apparatus and Procedures

The test apparatus used in the present study was an automobile air conditioner consisting of four basic components: compressor, condenser, expansion device, and evaporator. The test apparatus is shown in Fig. 1. The test rig worked with refrigerant R134a, and a liquid suction heat exchanger was added to the system, as shown in Fig. 2, designed by ANSYS. Figure 3 describes the schematic diagram of the test apparatus with the LSHX. Some parameters were analyzed, such as the coefficient of performance, refrigeration effect, heat rejected in the condenser, compressor work ($W_{comp.}$), and exergy destruction for each system component. The results were compared for two cases, i.e., with and without LSHX, three compressor speeds, i.e., 1000 rpm, 1700 rpm, and 2400 rpm, and two thermal loads, i.e., 500 Watt and 1000 Watt). The thermal properties of R134a are illustrated in Table 1.

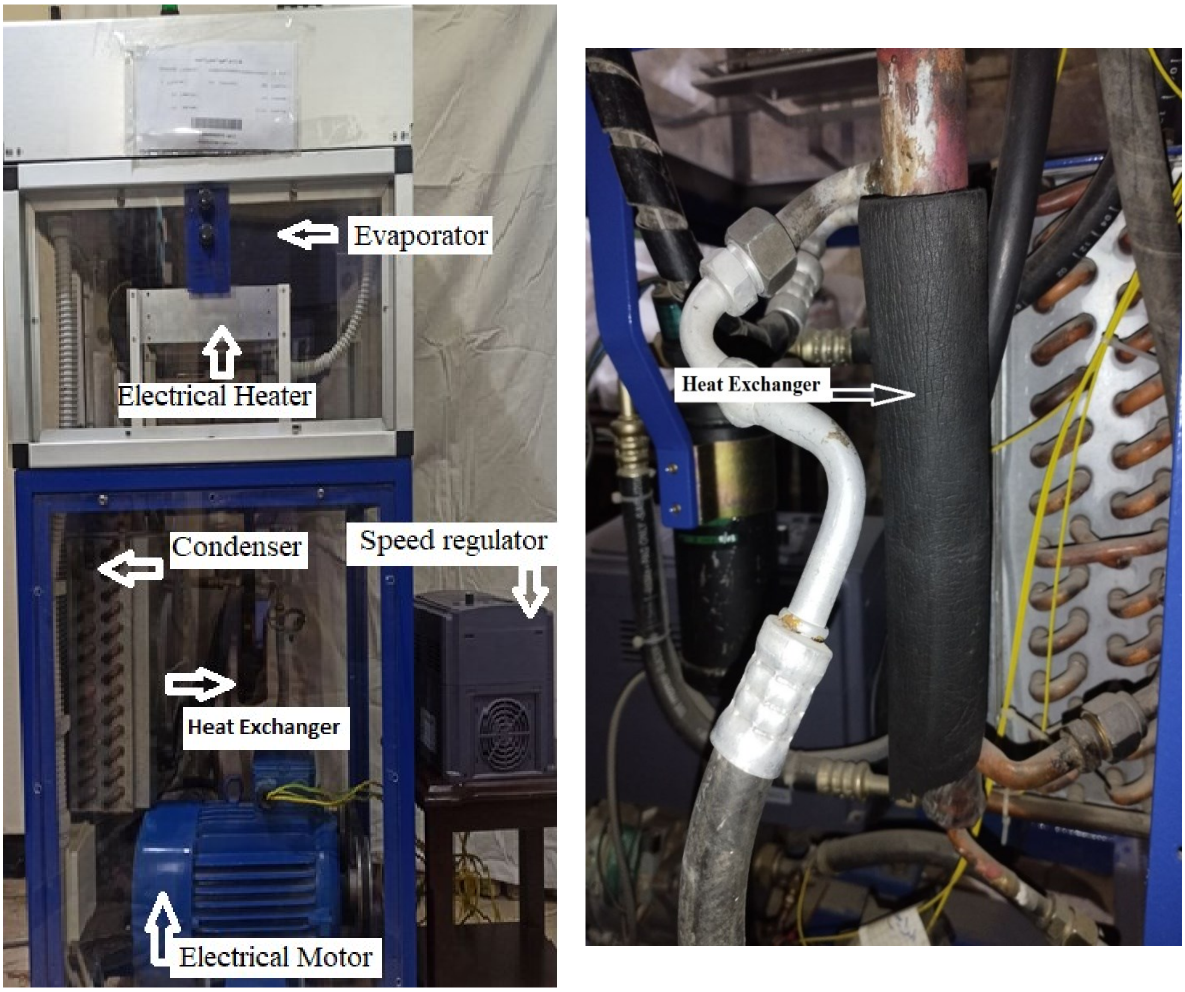

Fig. 1 Test Apparatus.

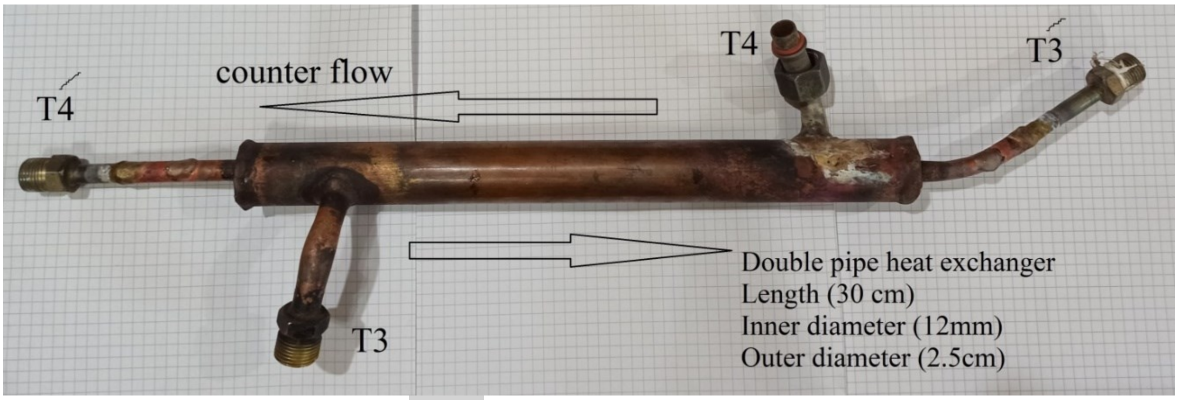

Fig. 2 Heat Exchanger Added.

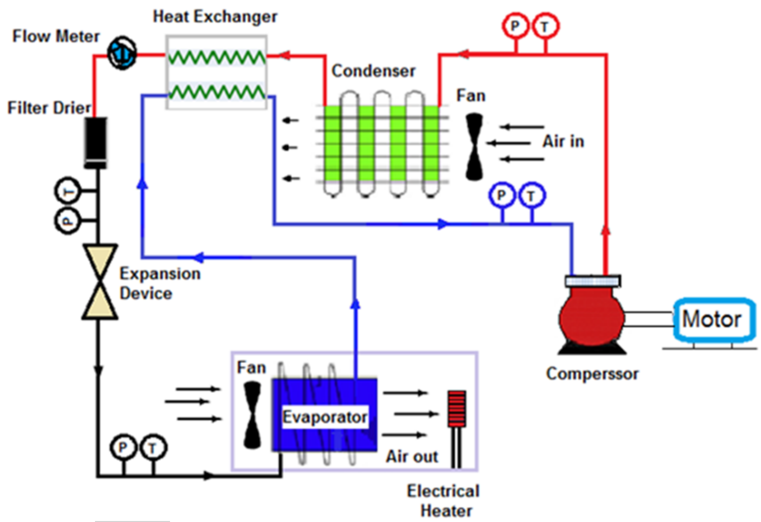

Fig. 3 Schematic Diagram of the Test Apparatus.

Table 1 Thermal Properties for R134a [14].

Thermodynamic Property	R134a
Chemical Formula	CF_3CH_2F
GWP	1430
ODP	0
Molar Mass (kg/kmol)	102.03
Boiling Point at 1 atm (K)	247.08
Freezing Point (K)	169.85
Critical Pressure (MPa)	4.90
Critical Temperature (K)	374.21
Critical Density (kg/m3)	511.90
Cost (USD)	25

2.2. Experimental Procedure

The experimental procedure is listed below:

1- Charging the system with R134a.
2- Setting the compressor speed at 1000 rpm and the thermal load at 500 watts.
3- The device was operated and waited to reach a steady state.
4- Steady state achieved after 33 minutes of operating.
5- The data was recorded every five minutes for all thermocouples and pressure gauges.
6- Set the compressor speed at 1700 rpm and then 2400 rpm with a thermal load of 500 Watt then with 1000 Watt and recorded all data obtained.
7- A Heat exchanger was added to the system.
8- All steps above were repeated with the added heat exchanger.

2.2.1. Mathematical Procedure

Energy and Exergy were analyzed for each component of the cycle with and without LSHX, as follows:

2.2.1.1. Energy Analysis

2.2.1.1.1. Coefficient of Performance (COP)

Without LSHX [14]:

$$\mathrm{COP} = \frac{\mathrm{RE}}{\mathrm{W_{actual,comp.}}} = \frac{h_1 - h_4}{h_2 - h_1} \quad (1)$$

With LSHX [15]:

$$\mathrm{COP} = \frac{\mathrm{RE}}{\mathrm{W_{actual,comp.}}} = \frac{h_1 - h'_4}{h'_2 - h'_1} \quad (2)$$

where h_1 *and* h'_1 are the inlet enthalpies to the compressor without and with LSHX, respectively, in (kJ/kg), h_2 *and* h'_2 are the outlet enthalpies from the compressor without and with LSHX, respectively, in (kJ/kg), and h_4 and h'_4 are the inlet enthalpies to the evaporator without and with LSHX, respectively, in (kJ/kg).

3.1.2. Compressor Work (W comp.)

Without LSHX [14]:

$$\mathrm{Wcomp.} = m_r \Delta h_{2-1} = \dot{m}_r (h_2 - h_1) \quad (3)$$

With LSHX [15]:

$$\mathrm{Wcomp.} = \dot{m}_r \Delta h'_{2-1} = m_r (h'_2 - h'_1) \quad (4)$$

where W_{comp} is the compressor work in (kW), and $\dot{m}_r$ is the refrigerant mass flow rate in (kg/s).

3.1.3. Heat Rejected by Condenser ($Q_{cond.}$)

Without LSHX [14]:

$$\mathrm{Qcond.} = \dot{m}_r (h_2 - h_3) \quad (5)$$

With LSHX [15]:

$$\mathrm{Qcond.} = \dot{m}_r (h'_2 - h_3) \quad (6)$$

where $Q_{cond.}$ is the heat rejected in (kW), and h_3 is the enthalpy of the condenser outlet in (kJ/kg).

3.1.4. Heat Absorbed by Evaporator ($Q_{evap.}$)

Without LSHX [14]:

$$\mathrm{Qevap.} = \dot{m} (h_1 - h4) \quad (7)$$

With LSHX [16]:

$$\mathrm{Qevap.} = \dot{m} (h_1 - h4') \quad (8)$$

3.2. Exergy Analysis

3.2.1. Exergy Destruction in Compressor (Ex. des. comp.) [17]

Without LSHX:

$$\Psi_{des,comp} = To\, \dot{S}_{gen,comp} = \dot{m}_r T_o (s2 - s1) \qquad (9)$$

With LSHX:

$$\Psi_{des,comp} = To\dot{S}_{gen,comp} = \dot{m}_{ref} T_o (s'2 - s'1) \qquad (10)$$

where $\Psi_{des,comp}$ is the rate of exergy destruction in the compressor, s_1 and s_2 are the entropies in the inlet and outlet of the compressor, respectively, without LSHX, and s'_1 and s'_2 are the entropies in the inlet and outlet of the compressor, respectively, with LSHX.

3.2.2. Exergy Destruction in Condenser (Ex. des. cond.) [18]

Without LSHX:

$$\Psi_{des,cond} = To\dot{S}_{gen,cond} = \dot{m}_r T_o ((s3 - s2) + \frac{q_{cond}}{T}) \qquad (11)$$

With LSHX:

$$\Psi_{des,cond} = To\dot{S}_{gen,cond} = \dot{m}_{ref} T_o [(s3 - s'2) + (\frac{q_{cond}}{T})] \qquad (12)$$

where T_o is the environment temperature, and T is the average temperature for the inlet and outlet from the condenser.

3.2.3. Exergy Destruction in Thermostatic Expansion Device (Ex. des. TXV) [19]

Without LSHX:

$$\Delta\Psi_{TXV} = T_o S_{gen,exp} = \dot{m}_r T_o (s_3 - s_4) \qquad (13)$$

Without LSHX:

$$\Delta\Psi_{TXV} = \dot{m}_{ref} [T_o (s'_3 - s'_4)] \qquad (14)$$

3.2.4. Exergy Destruction in the Evaporator (Ex. des. evap.) [20]

With LSHX:

$$\Delta\Psi_{des,evap} = \dot{m}_r (\psi_4 - \psi_1) = \dot{m}_{ref} [(h_4 - h_1) - T_o (s_4 - s_1)] \qquad (15)$$

With LSHX:

$$\Psi_{des,evap} = To\, \dot{S}_{gen,evap} = \dot{m}_{ref} [(h_4 - h_1) - T_o (s_4 - s_1)] \qquad (16)$$

where T is the average temperature for the air side inlet and outlet of the evaporator temperature. The enthalpies of all input and output points of the cycle components were extracted from Engineering Equation Solver programming (EES) based on the temperature and pressure values taken from the experimental work.

3.3. Experimental Sets

Two sets of experiments were examined. In the first set, all experiment parameters were examined without using a liquid suction heat exchanger. The test included three compressor speeds: 1000rpm, 1700rpm, and 2400 rpm, and two thermal loads: 500 Watt and 1000 Watt. The second set included a liquid suction heat exchanger with the parameters mentioned above. Table 1 describes the different parameters for these sets.

Table 2 Operating Parameters Values of the Present work.

Parameters	Values
Refrigerants used	R134a
Compressor speed	1000 rpm, 1700 rpm, 2400 rpm
Thermal load	500 watt, 1000watt
Case 1	Without LSHX
Case 2	With LSHX

3. RESULTS AND DISCUSSION

3.1. Results of the Basic Cycle Without LSHX

The studied parameters were the refrigeration effect, compressor work, coefficient of performance, heat rejected in the condenser, and exergy destruction in the compressor, condenser, thermostatic expansion device, and evaporator. Table 2 represents the experimental results for the first set without a liquid suction heat exchanger. The results showed that the compressor work ($W_{comp.}$) increased with the compressor speed based on the increasing compression ratio and discharge pressure in the compressor, which required more power to operate. The cycle COP decreased due to the high work done by the motor at high speeds. The refrigeration effect of the cycle also decreased due to decreasing the COP and increasing $W_{comp.}$, according to the relation below:

$$COP = \frac{RE}{W_{actual,comp.}} \qquad (17)$$

The Heat rejected in the condenser increased due to increasing the compressor discharge temperature and speed.

Table 3 Experimental Results of the First Set for a Basic Cycle Without LSHX.

Parameters	Without LSHX							
	COP	RE (kJ/kg)	Qcond. (kW)	Wcomp. (kW)	Ex.des. comp. (kW)	Ex.des. cond. (kW)	Ex. Des. TXV (kW)	Ex. Des. Evap. (kW)
1000rpm,500 watt	3.2	140.3	2.26	0.619	0.038	0.23	0.089	0.225
1000rpm,1000watt	3.76	130.66	2	0.896	0.041	0.223	0.064	0.127
1700 rpm,500 watt	2.6	133.13	3.76	0.833	0.045	0.265	0.096	0.267
1700 rpm,1000 wat	2.7	128.4	3.47	0.995	0.064	0.275	0.069	0.19
2400 rpm,500 watt	2.26	126.3	4.5	1.073	0.069	0.294	0.112	0.284
2400rpm,1000watt	2.69	123.06	4.3	1.21	0.065	0.292	0.072	0.225

3.2.Second Set Results of the Cycle with adding LSHX

The parameters examined in the first set above were examined in the second set, i.e., by adding the LSHX. When LSHX was added to the cycle, the results showed an enhancement in the cycle COP with the three examined compressor speeds; however, it remained decreasing with the increase in the compressor speed. When the LSHX was added, the compressor work decreased, while the refrigeration effect increased, and the heat rejected by the condenser increased. Table 4 represents the experimental results of the cycle with the liquid suction heat exchanger added. The experimental work pressure limits are illustrated in Table 5. Figure 4 shows the P-h diagram of the VCRC for R134a with and without LSHX. Figures 5 to 12 show the effect of compressor speed on the energy analysis parameters at different thermal loads with and without LSHX. When a liquid suction heat exchanger was added to the cycle, the coefficient of performance increased by 7.18%, 10.7%, and 3.09% for the first, second, and third speed, respectively, a 500-Watt thermal load. When the thermal load was 1000 Watt, the coefficient of performance increased by 10.6%, 23.3%, and 7.8% for the first, second, and third speed, respectively. Figures 5 and 6 show that increasing the compressor speed decreased the coefficient of performance due to increasing the compressor work.

Table 4 Experimental Results of the Second Set for the Cycle with LSHX.

Parameters	With LSHX							
	COP	RE (kJ/kg)	Qcond. (kW)	Wcomp. (kW)	Ex.des. comp. (kW)	Ex.des. cond. (kW)	Ex. Des. TXV (kW)	Ex. Des. Evap. (kW)
1000rpm,500 watt	3.43	147.8	2.39	0.441	0.035	0.265	0.093	0.21
1000rpm,1000watt	3.83	144.23	2.30	0.585	0.039	0.259	0.075	0.12
1700 rpm,500 watt	2.88	139.7	3.93	0.679	0.044	0.265	0.11	0.225
1700 rpm,1000 wat	3.3	140.26	3.06	0.714	0.045	0.281	0.091	0.133
2400 rpm,500 watt	2.33	132.06	4.74	0.854	0.056	0.294	0.118	0.249
2400rpm,1000watt	2.9	128.8	4.49	0.882	0.049	0.31	0.099	0.197

Table 5 Evaporator and Condenser Pressure Values of the Experimental Work.

	Condenser pressure (bar)	Evaporator pressure (bar)
without LSHX	20	3.5
with LSHX	19.2	4

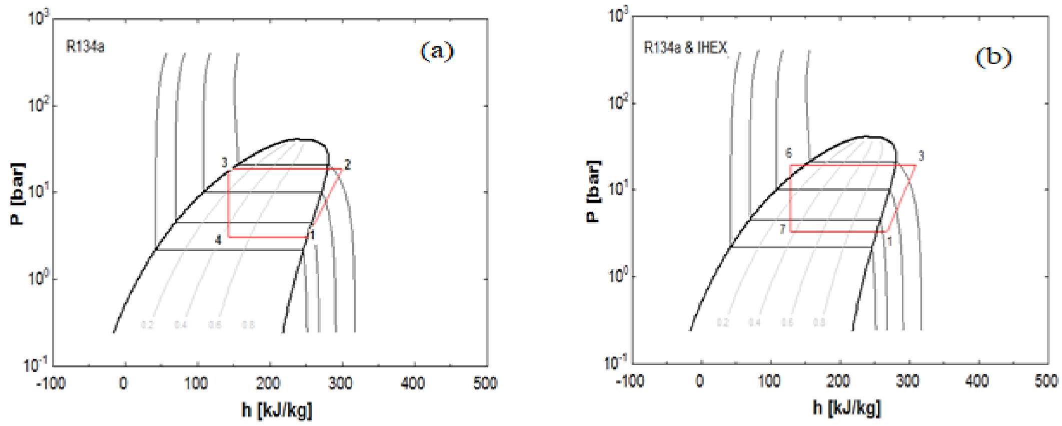

Fig. 4 P-h Diagrams of VCRC Working with R134a: (a) without LSHX (b) with LSHX.

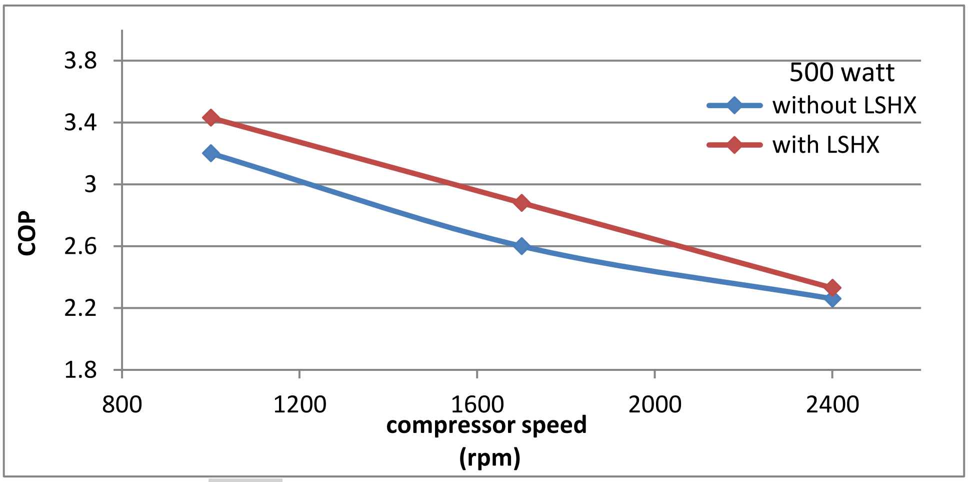

Fig. 5 Cop Vs. Compressor Speed at 500 Watt.

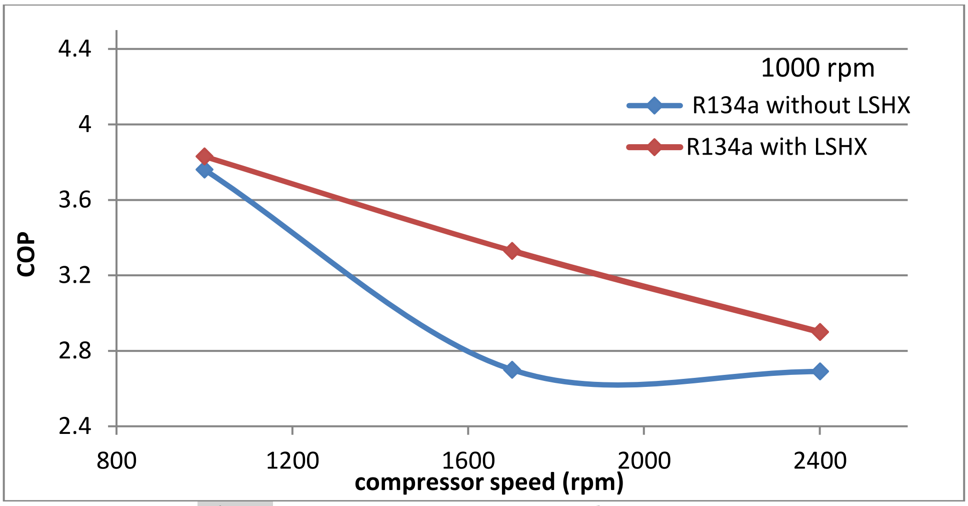

Fig. 6 COP Vs. Compressor Speed at 1000 Watt.

The refrigeration effect increased when the liquid suction heat exchanger was added due to increasing the sub-cooling degree. At a thermal load of 500 Watt, the refrigeration effect increased by 5.36%, 4.9%, and 4.5% for the first, second, and third speed, respectively, while at a thermal load of 1000 Watt, the refrigeration effect increased by 10.38%, 9.23%, and 4.6% for the first, second, and third speed, respectively; the same trend is in [16]. Figures 7 and 8 show that increasing the compressor speed decreased the cycle's refrigeration effect. When the liquid suction heat exchanger was added to the cycle, and the thermal load was 500 Watt, the compressor work decreased by 28.9%, 18.4%, and 20.41% for the first, second, and third speed, respectively. When thermal load was 1000 Watt, the compressor work decreased by 34.7%, 28.2%, and 27.3% for the first, second, and third speed, respectively. Figures 9 and 10 represent the relation between the compressor work and speed. The compressor work increased with compressor speed due to the increasing in compression ratio, which requires more power to operate. The heat rejected by the condenser increased with adding a liquid suction heat exchanger due to increasing the sub-cooling degree and superheating degree, decreasing the enthalpy leaving the condenser and increasing the enthalpy entering the condenser. The difference between the inlet and outlet enthalpies gives the total heat rejected by the condenser; the increment values were 5.7%, 4.5%, and 5.3% for the first, second, and third speed, respectively, at a thermal load of 500 watt. While at the thermal load of 1000 Watt, the increment values were 15.3%, 5.7%, and 4.4% for the first, second, and third speed, respectively. However, increasing the compressor speed increased the heat rejecter due to increasing the compressor work and condensing temperature, decreasing the condensation latent heat; therefore, the heat rejected by the condenser increased. Figures 11 and 12 show the relation between the compressor speed and the heat rejected by the condenser with and without liquid suction heat exchanger.

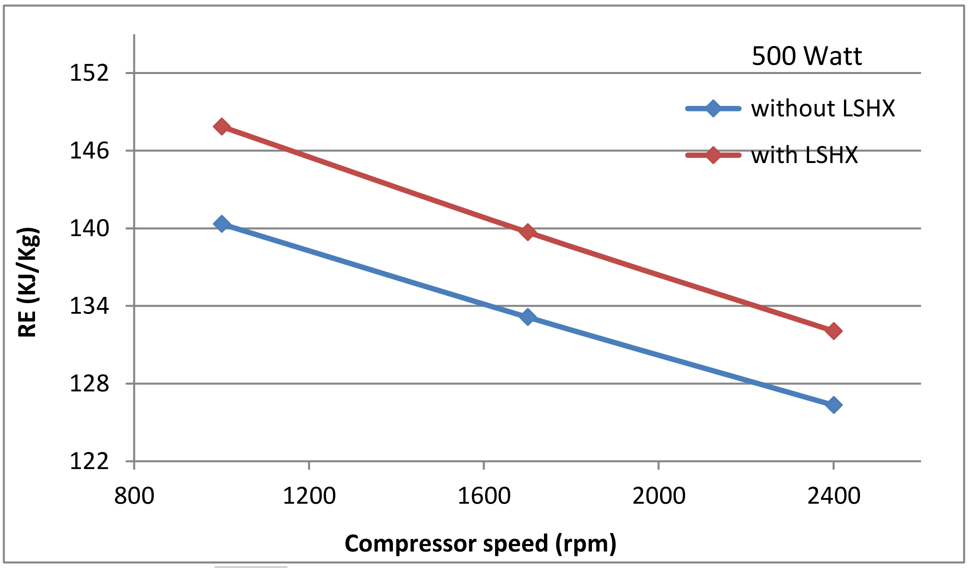

Fig. 7 RE Vs. Compressor Speed at 500 Watt.

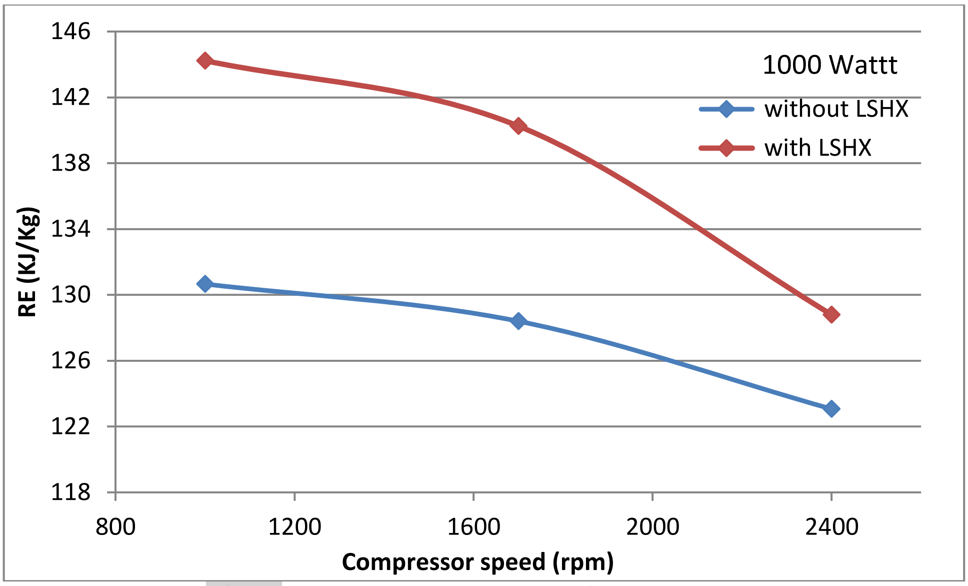

Fig. 8 RE Vs. Compressor Speed at 1000 Watt.

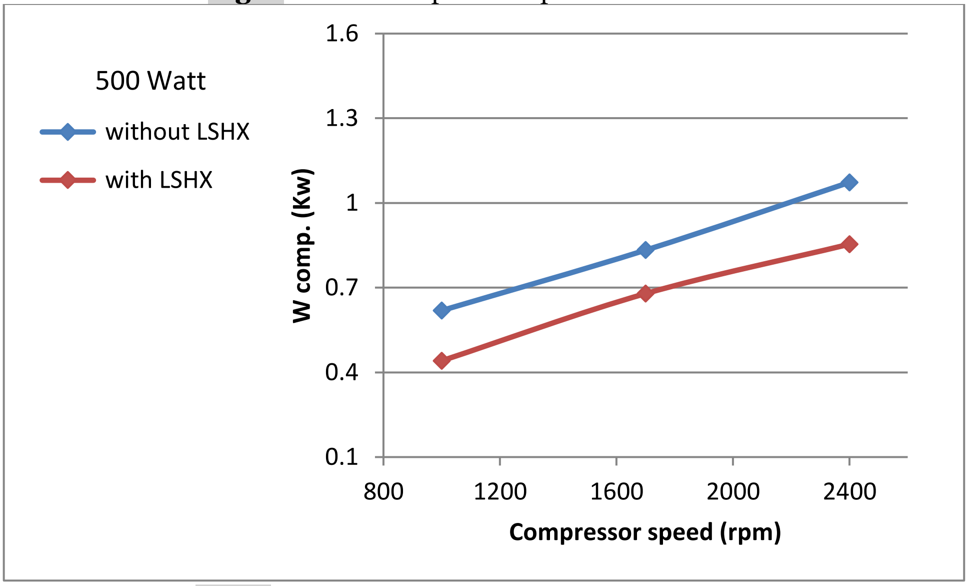

Fig. 9 W_{comp}. Vs. Compressor Speed at 500 Watt.

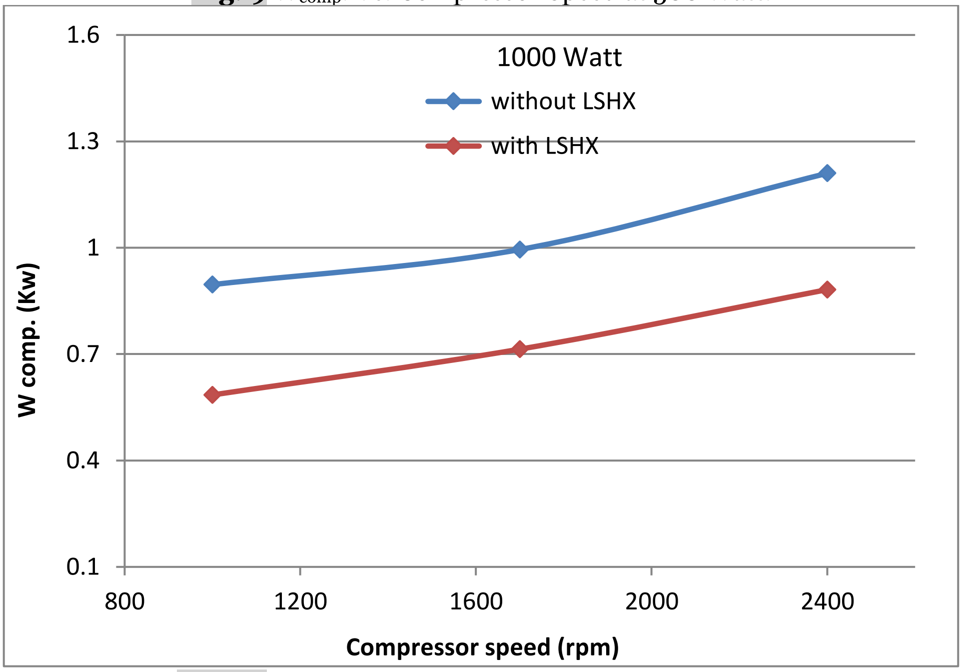

Fig. 10 W_{comp}. Vs. Compressor Speed at 1000 Watt.

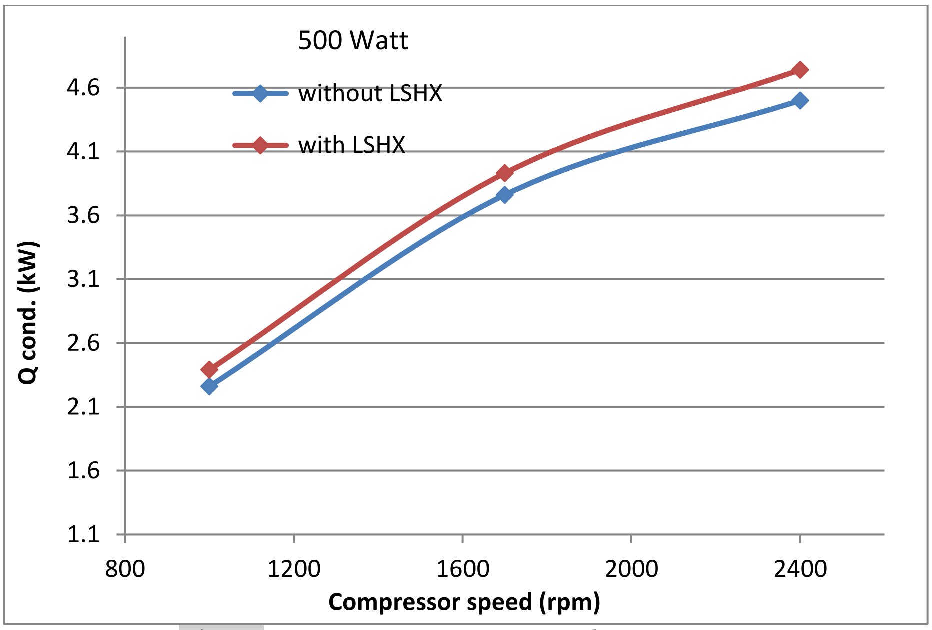

Fig. 11 $Q_{cond.}$ Vs. Compressor Speed at 500 Watt.

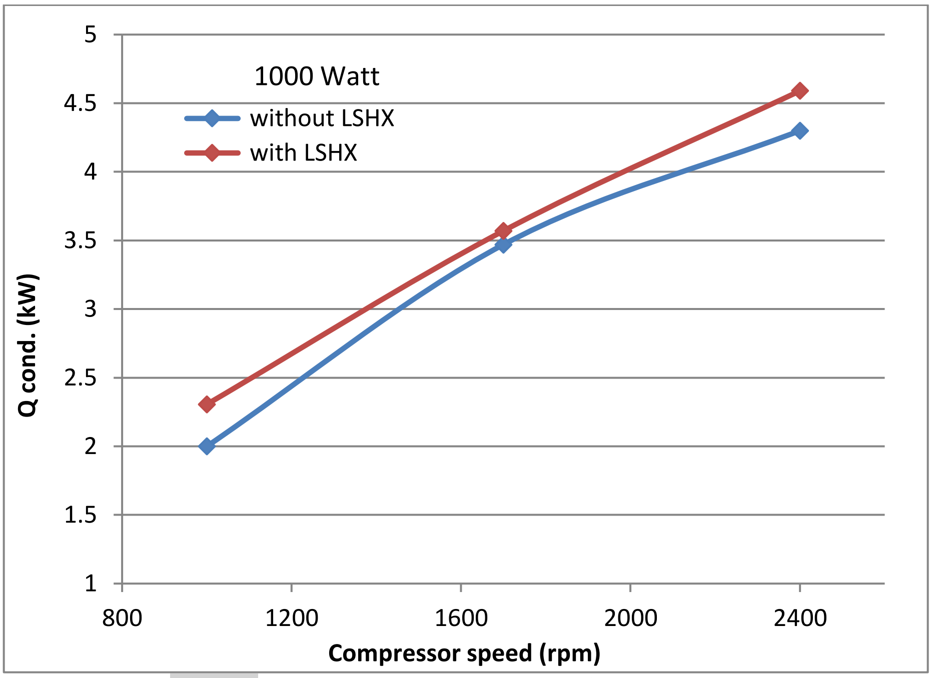

Fig. 12 $Q_{cond.}$ Vs. Compressor Speed at 1000 Watt.

Figures 12 to 20 represent the effect of compressor speed on the parameters of exergy analysis at different values of thermal loads with and without LSHX to an automobile air conditioner system. When the LSHX was added to the cycle, the compressor's exergy destruction decreased by 7.8%, 2.2%, and 18.8% for the first, second, and third speed, respectively, at a thermal load of 500 Watt. When the thermal load was 1000 Watt, the decreasing values were 4.8%, 29.6%, and 24.6% for the first, second, and third speed, respectively. The compressor exergy destruction decreased due to the decrease in compressor work, so the compression ratio was low, and the exergy efficiency was high, leading to low exergy destruction. When the compressor speed increased, exergy destruction in the compressor also increased due to the increased pressure difference of the system because of the decrease in evaporation and the increase in condensation pressures. As a result, the entropy generation increased. The increase in discharge temperature and pressure of the compressor caused a high friction force between the moving part of the compressor and the refrigerant; therefore, the energy losses increased. As a result, these reasons increased exergy destruction in the compressor. Figures 13 and 14 show the relation between the compressor speed and EX. Des. Comp.

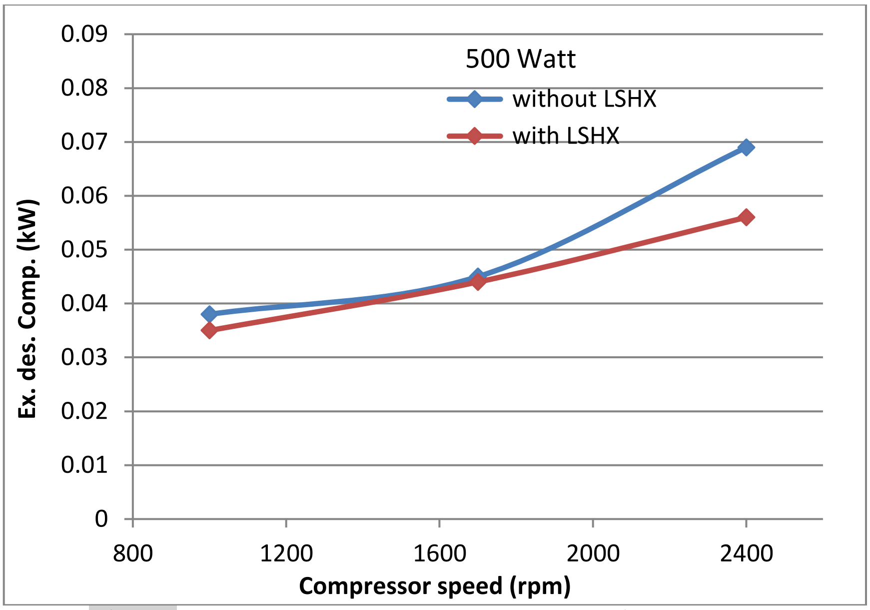

Fig. 13 Ex. Des. Comp. Vs. Compressor speed at 500 Watt.

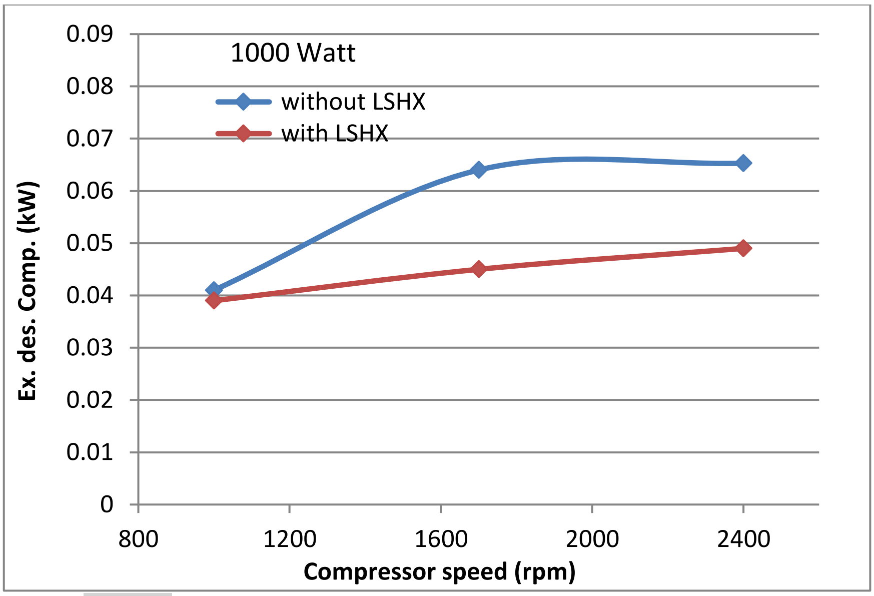

Fig. 14 Ex. Des. Comp. Vs. Compressor speed at 1000 Watt.

Exergy destruction in the condenser increased with the compressor speed and adding LSHX by 15.2%, 6.03%, and 6.8% for the first, second, and third speed, respectively, at the thermal load of 500 Watt. While at the thermal load of 1000 Watt, the increment values were 16.1%, 7.25, and 16% for the first, second, and third speed, respectively, due to increasing the system pressure difference because the condensation pressure increased and the evaporation pressure decreased, resulting in more irreversibility and entropy generation, i.e., more exergy destruction. Figures 15 and 16 show the relation between the compressor speed and Ex. Des. Condenser. The exergy destruction in TXV increased with the compressor speed due to increasing the refrigerant mass flow rate; therefore, the pressure difference of the system increased due to increasing the condensation pressure and decreasing the evaporation pressure, resulting in more irreversibility and entropy generation, i.e., more exergy destruction. Also, the exergy destruction in TXV increased when LSHX was added by 4.49%, 5.2%, and 5.88% for the first, second, and third speed, respectively, at 500 Watt, while at 1000 Watt, the increment in destruction was 17.1%, 31.1%, and 37.5% for the first, second, and third speed, respectively. Figures 17 and 18 show the relation between the compressor speed and Ex. Des. TXV with and without LSHX.

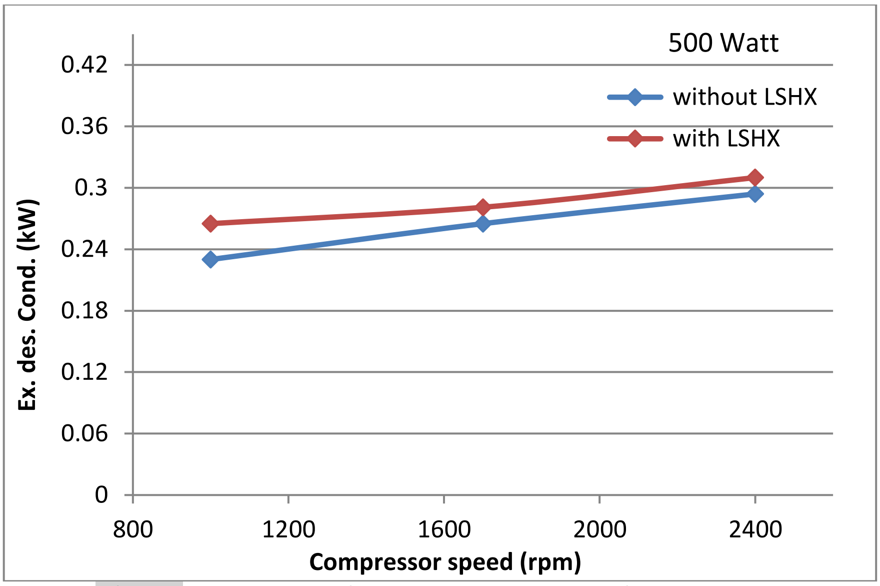

Fig. 15 Ex. Des. Cond. Vs. Compressor Speed at 500 Watt.

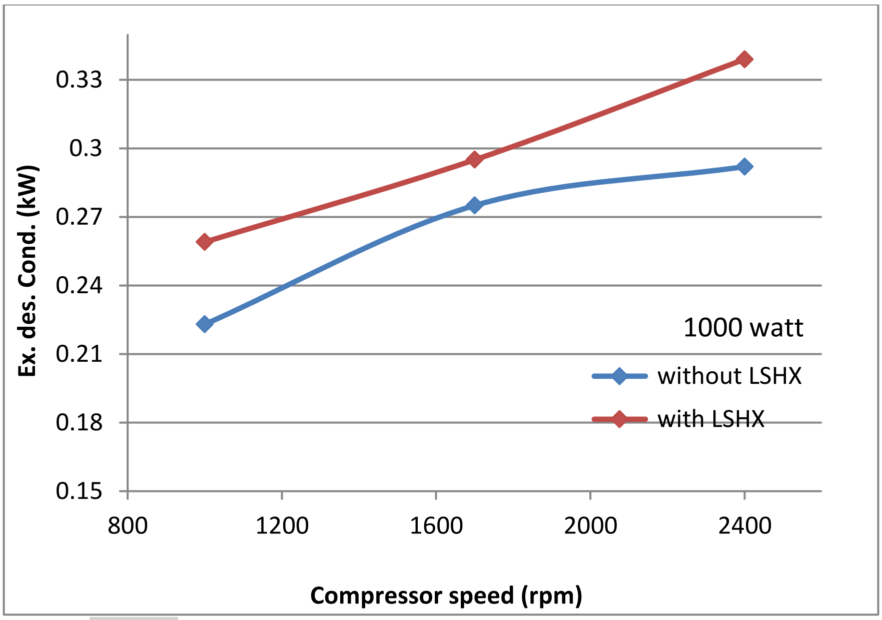

Fig. 16 Ex. Des. Cond. Vs. Compressor Speed at 1000 Watt.

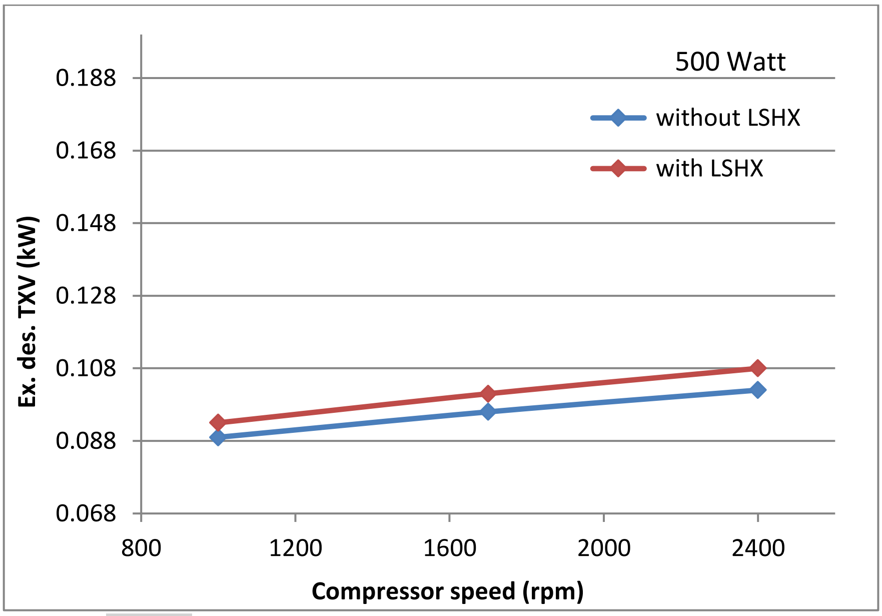

Fig. 17 Ex. Des. TXV Vs. Compressor Speed at 500 Watt.

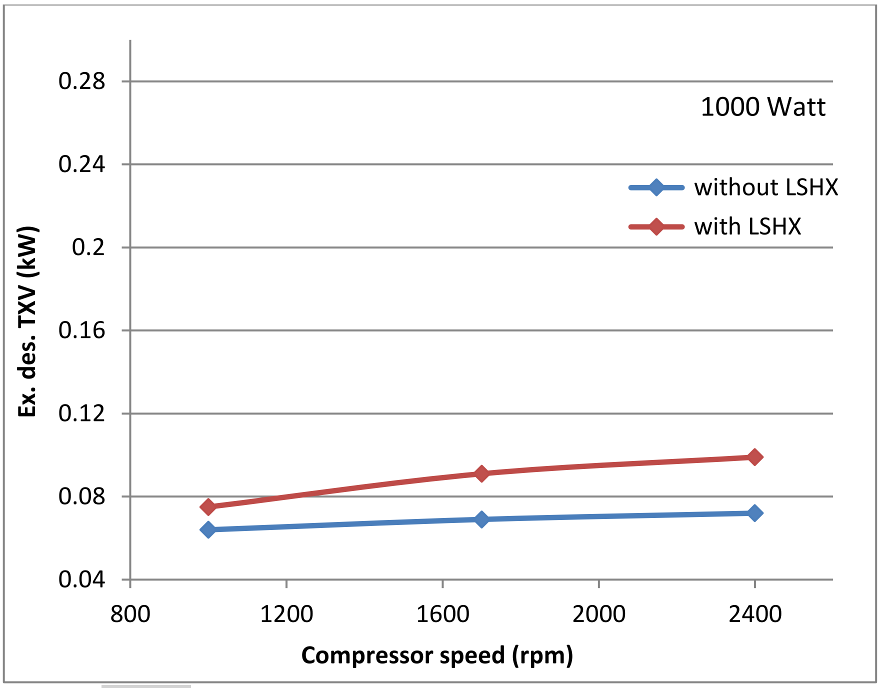

Fig. 18 Ex. Des. TXV Vs. Compressor Speed at 1000 wW.

The exergy destruction in the evaporator increased with the compressor speed due to increasing the refrigerant mass flow rate; therefore, the pressure difference of the system increased due to increasing the condensation pressure and decreasing the evaporation pressure, resulting in more irreversibility and entropy generation, i.e., more exergy destruction. However, the exergy destruction in the evaporator decreased when LSHX was added by 6.6%, 15.7%, and 12.3% for the first, second, and third speed, respectively at 500 Watt, while the decreasing values at 1000 Watt were 5.5%, 8%, and 13.6% for the first, second, and third speed, respectively, as described in Figures 19 and 20. Table 6 compares the results of the present study with the other studies.

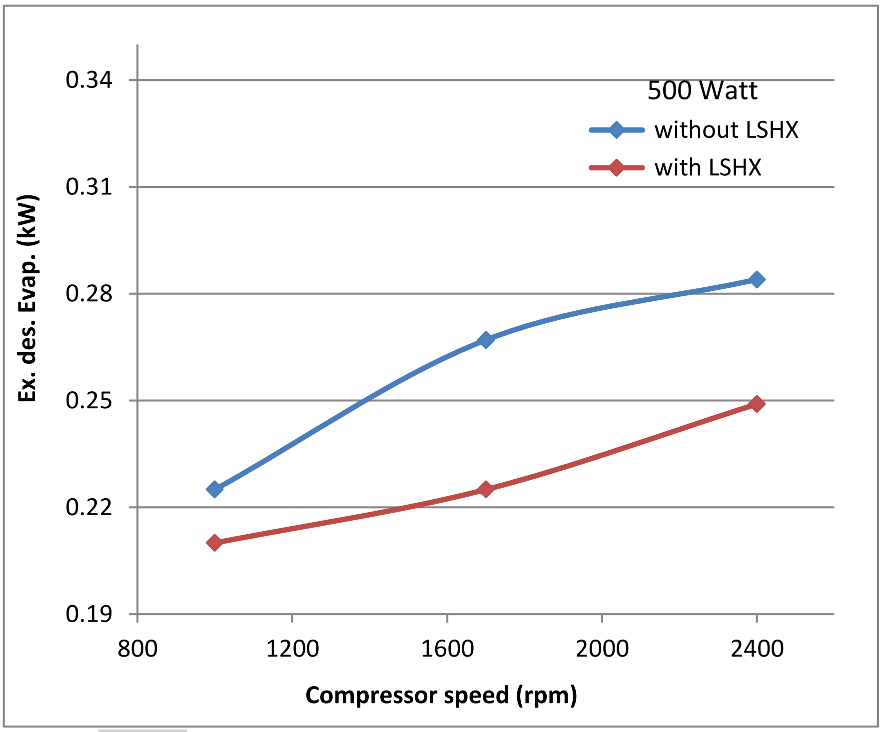

Fig. 19 Ex. Des. Evap. Vs. Compressor Speed at 500 Watt.

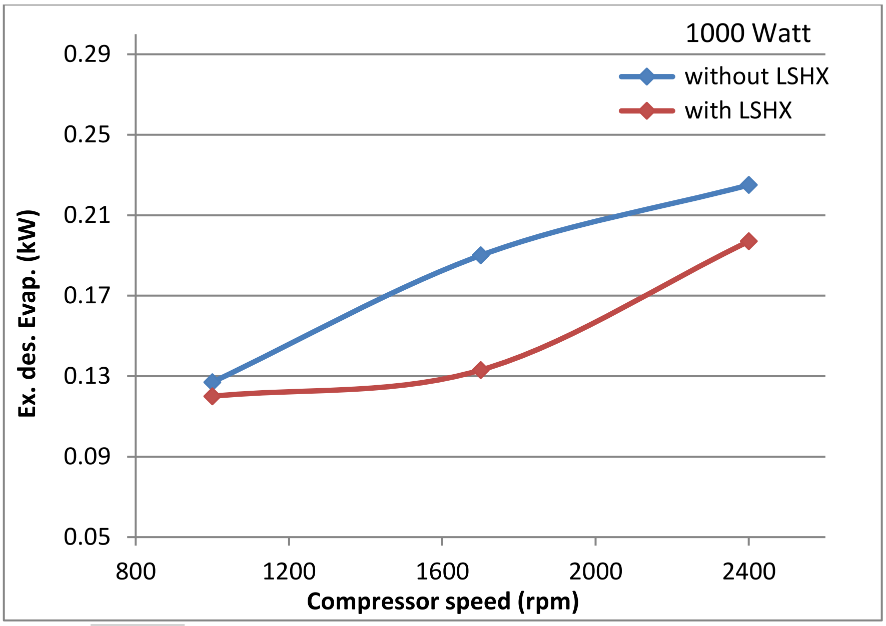

Fig. 20 Ex. Des. Evap. Vs. Compressor Speed at 1000 Watt.

Table 6 Comparison between the Present and Previous Results.

Parameters	COP	RE (kJ/kg) Increased	Qcond. (kW) Increased	Wcomp. (kW) Decreased	Ex.des. comp. (kW) Decreased	Ex.des. cond. (kW) increased	Ex. Des. TXV (kW) increased	Ex. Des. Evap. (kW) Decreased
Present study 1000 rpm (with LSHX)	10.6%	10.38%	15.3%	34.7%	4.8%	16.1%	17.1%,	5.5%
Present study 1700 rpm (with LSHX)	23.3%	9.23%	5.7%	28.2%	29.6%	7.26%	31.1%	8%
Present study 2400 rpm (with LSHX)	7.8%	4.6%	4.4%	27.3%	24.6%	16%	37.5%	13.6%
Elewi [5] (with LSHX) at 1400rpm	12.2%	_____	_____	12.1%	_____	______	______	_____
Elewi [5] (with LSHX) at 2900 rpm	48%	____	______	17.2%	_____	_______	______	_____

4.CONCLUSIONS

The following points are the summary of the present study conclusions:

- When a liquid suction heat exchanger was added to the cycle, the thermal performance of the cycle was enhanced by 10.9%, 23.3%, and 7.8% for 1000rpm, 1700rpm, and 2400rpm, respectively, at 1000 Watt; however, it decreased with increasing the compressor speed, varying the refrigerant mass flow rate value. As the compressor's speed increased, the compressor's discharge temperature increased, requiring more power to operate. Therefore, the compressor work increased, and the refrigeration effect decreased with the increasing the compressor speed. As a result, the thermal performance decreased, based on the relation below:

$$COP = \frac{\mathbf{RE}}{w_{actual,comp.}}$$

- The heat rejected by the condenser increased with adding LSHX and increasing the compressor speed due to increasing the compressor work and the condensing temperature, decreasing the condensation latent heat.
- The total exergy destruction of the cycle decreased when LSHX was added due to decreasing the destruction values of all cycle components since the total destruction is the summation of destruction values in all cycle components. While the total exergy destruction of the cycle increased with the compressor speed, the total exergy efficiency decreased because it is a function of exergy destruction.Increasing the total exergy destruction values is attributed to the increase in the refrigerant mass flow rate and increasing the compressor discharge pressure, which requires more power to work; therefore, the work and destruction in the compressor increased.

- Since the total exergy destruction increased with the compressor speed, the efficiency of the cycle decreased.
- The absence of LSHX in a system working with R134a is a good addition to enhancing the system's thermal performance and increasing efficiency.

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NOMENCLATURE

AACSs	Automobile air conditioning system
COP	Coefficient of performance
Cp	Specific heat capacity, J/(kg °C)
EES	Engineering equation solver
IHX	Internal heat exchanger
LSHX	Liquid suction heat exchanger
P	Pressure
$Q_{evap.}$	Heat absorbed by evaporator (kW)
$Q_{cond.}$	Heat rejected by condenser (kW)
RE	Refrigeration effect
S	Entropy
T	Temperature, °C
VCRC	Vapor compression refrigeration cycle
W_{actual}	Actual work kW
$W_{comp.}$	Compressor work (kW)
X	Exergy
$\dot{m}$	Mass flow rate (kg/s)
h_1	Enthalpy at compressor inlet (kJ/kg)
h_2	Enthalpy at compressor outlet (kJ/kg)
h_3	Enthalpy at condenser outlet (kJ/kg)
h_4	Enthalpy at evaporator inlet (kJ/kg)
h_1'	Enthalpy at LSHX vapor outlet (kJ/kg)
h_2'	Enthalpy at compressor outlet when LSHX added (kJ/kg)
h_3'	Enthalpy at LSHX liquid outlet (kJ/kg)
h_4'	Enthalpy at evaporator inlet when LSHX added (kJ/kg)
Greek symbols	
ρ	Density, kg/m^3
Subscripts	
Comp.	Compressor
Cond.	Condenser
Evap.	Evaporator
TXV.	Thermostatic expansion valve

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