Al-Qadisiyah Journal for Engineering Sciences 17 (2024) 066–076

Contents lists available at http://qu.edu.iq

Al-Qadisiyah Journal for Engineering Sciences

Journal homepage: https://qjes.qu.edu.iq

# Investigation of an automobile air-conditioner with a liquid-suction heat exchanger using R134a and R1234yf

***Ansam A. Mohammed[1]*, Wail S. Sarsam***

*Department of Mechanical Engineering, Collage of Engineering, University of Baghdad, 10071 Baghdad, Iraq*

A R T I C L E  I N F O

*Article history:*
Received 10 septemper 2023
Received in revised form 19 Novenver 2023
Accepted 9 March 2024

*Keywords*:
Refrigeration
R134a and R1234yf
Heat exchanger
ANSYS
Vapor compression
Coefficient of performance

A B S T R A C T

Air-conditioning systems (ACs) are essential in hot and humid climates to ensure acceptable ambient air quality as well as thermal comfort for buildings users. It is essential to improve refrigeration system performance without increasing the effects of global warming potential (GWP) and ozone depletion potential (ODP). The main objective of this study is to evaluate the performance of an air conditioning system that operates with a liquid suction heat exchanger (LSHX) through implementing refrigerants with zero OPD and low GWP (i.e., R134a and R1234yf). Liquid suction heat exchanger (LSHX) was added to an automobile air conditioning system (AACS).When Liquid suction heat exchanger was added to the cycle, primary results indicated that an enhancement in the cycle coefficient performance (COP) by 25.2 %and 17.3% for R134a and R1234yf respectively, and decreasing in mass flow rate of the refrigerants used ($\dot{m}_r$). Also increasing in refrigeration effect (RE) by 4.2% and 2.3% for R134a and R1234yf respectively. Presence of LSHX caused increasing in heat rejected by condenser ($Q_{cond.}$) according to increasing in sub-cooling degree and decreasing in compressor work ($W_{comp.}$), increasing in condenser exergy destruction (X. des. cond.) and thermostatic expansion device (TXV) exergy destruction, but decreasing in compressor and evaporator. Summary of previous experimental and numerical studies is presented as well.



## 1. Main text

Air-conditioning systems (ACs) are essential in hot and humid climates to ensure acceptable ambient air quality as well as thermal comfort for building users [1] Therefore, it is essential to improve refrigeration system performance without increasing the effects of GWP and ODP [2]. The next generation of refrigeration and ACs must include low-GWP refrigerants and optimized components. Several methods are used to enhance the thermal performance of a vapor compression refrigeration cycle (VCRC). One of those methods is the addition of nanoparticles to the refrigeration [3]. The use of nanoparticles as refrigerant additions has been found to be a method of improving the performance of VCRs without altering the system components [4], also using an ejector as an expansion device [5].Moreover, methods of sub-cooling can be categorized into four techniques, dedicated sub-cooling, integrated sub-cooling [6] ,condensate-assisted sub-cooling , and LSHX usage [7]. An internal heat exchanger (IHX) is a common modification of the basic cycle for enhancing its energy performance [8]) .A heat exchanger is a device that transfers heat between two fluids of different temperatures [9] IHX also known as liquid-to-suction heat exchangers (LSHX), which might be utilized to improve the energy efficiency for VCC [8].

* *Corresponding author.*
E-mail address: ansam.abdo2003m@coeng.uobaghdad.edu.iq (Ansam Mohammed)

https://doi.org/10.30772/qjes.2023.143262.1030

**Nomenclature:**

| | | | |
|---|---|---|---|
| $\dot{m}$ | Mass flow rate (kg/s) | **Abbreviation:** | |
| $h_1$ | Enthalpy at compressor inlet (kJ/kg) | AACS | Automobile air conditioning system |
| $h_2$ | Enthalpy at compressor outlet (kJ/kg) | ACs | Air conditioning system |
| $h_3$ | Enthalpy at condenser outlet (kJ/kg) | CC | Cooling capacity |
| $h_4$ | Enthalpy at evaporator inlet (kJ/kg) | COP | Coefficient of performance |
| | | DPHX | Double pipe heat exchanger |
| $h_{2s}$ | Enthalpy at the compressor outlet of isentropic compression process | AACS | Automobile air conditioning system |
| $h'_1$ | Enthalpy at LSHX vapor outlet (kJ/kg) | ACs | Air conditioning system |
| $h'_2$ | Enthalpy at compressor outlet when LSHX added (kJ/kg) | GWP | Global warming potential |
| | | HX | Heat exchanger |
| | | IHX | Internal heat exchanger |
| | | LSH | Liquid suction heat exchanger |
| | | ODP | Ozone depletion potential |
| | | P | Pressure |
| $h'_3$ | Enthalpy at LSHX liquid outlet (kJ/kg) | *Subscripts* | |
| $h'_4$ | Enthalpy at evaporator inlet when LSHX added (kJ/kg) | Comp. | compressor |
| | | Cond. | condenser |
| | | Evap. | Evaporator |

Various numerical and experimental studies on the energy and exergy analyses of VCRCs using LSHX and alternative refrigerants have been published, such as, energy and exergy analysis for low-GWP refrigerants i.e. R152a, R1234yf, and R1234ze were studied both experimentally and theoretically by [10]. The study was conducted at different evaporation and condensation temperatures, as well as compressor speeds. The compressor power, cooling capacity (CC), exergy destruction and COP were included in the investigation. Results showed that the overall exergy destruction of R1234yf was decreased by 15% as compared to R134a. An automotive air conditioning system (AACS) was created by [11] they used a thermostatic expansion valve (TXV) and a coaxial internal heat exchanger loaded with R1234yf.The system was tested with two ranges of speeds (1000 rpm and 2600 rpm). Inlet air temperatures were between (30 °C and 40 °C). The test studied two cases with and without IHX. Results for system with IHX showed a 0.8 °C decrease in evaporation temperature, 2.2% increase CC, 2% decrease in compressor power, and 3% increase in COP. A system with IHX and R1234yf showed a general improvement in performance. The thermal performance of R-1234yf as a substitute to R-134a in the AACs was studied experimentally by [12], the calculations indicated that the cooling capacity of the system with R1234yf is lower than that with R134a system, the compressor power was reduced by 11%, and COP for system with R1234yf is less than that with R134a by 14.5%. A mathematical computational model was created by [13] to determine the influence of a LSHX on the thermal performance of a VCRC using R134a, R600a, and R22. The modified system with a LSHX showed that the COP with R134a was 7% higher than that with R600a, and 12% higher than that of R22 .However, the RE of system with R600a was higher than that with R134a. R600a was considered an excellent alternative refrigerant that may be utilized in mechanical refrigeration systems.

## 2. Vapor Compression Refrigeration Cycle

The vapor compression refrigeration cycle (VCRC) consists of four major components: a variable-speed compressor, a condenser, an evaporator, and an expansion device.

Several methods were commonly used to enhance the thermal performance of a VCRC. In this study, LSHX is added to the cycle. One of the IHX types that may be used in VCRC is the double pipe heat exchanger (DPHX) [14] as shown in Fig. 1, which describes the schematic diagram of the test rig.

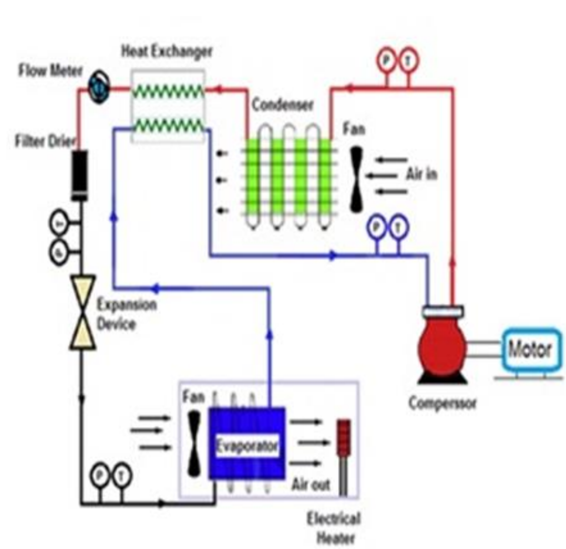



**Figure 1**. Schematic diagram of test rig

Some assumptions are made to simplify the theoretical solution, as described below:

1. Steady-state operating conditions and one-dimensional
2. The leak flows between tubes are neglected.
3. The isenthalpic process is considered for the expansion device, and the isentropic process is considered for the compressor.

4. The heat exchanger is well insulated, so the heat loss to the environment is neglected.
5. Constant refrigerant properties.
6. Pressure drops in all components are neglected.
7. The material of the heat exchanger is designed to have constant thermal properties.

## 3. Computational Model

In this work, a mathematical model was built to compute and compare the performance of AACs charged with R134a and R1234yf. The 3D heat exchanger geometry was created using the SOLID WORK 2014 design module [15]. The computational model of DPHX was solved using ANSYS (2018). The geometry is shown in Fig. 2.

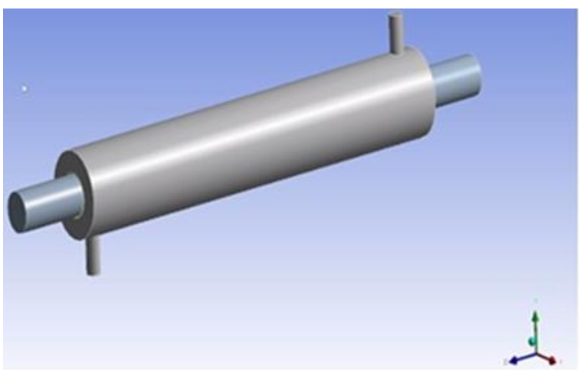


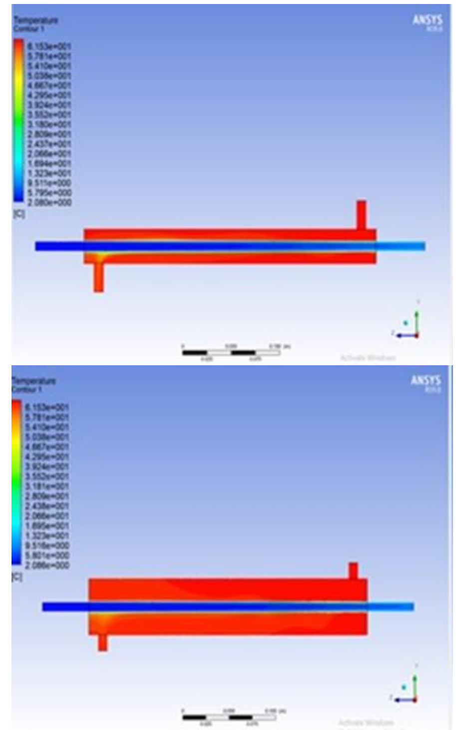
**Figure 2.** Geometry of the test suction

should have a caption. Headings should be placed above tables, center justified. Only horizontal lines should be used within a table, to distinguish the column headings from the body of the table, and immediately above and below the table. Tables must be embedded into the text and not supplied separately. Below is an example which the authors may find useful. Table 1. An example of a table Description of geometry parameters are shown in Table 1.

**Table 1.** Dimensions of simulated double tube heat exchanger

| **LSHX length L** | **Inner diameter Di** | **Outer diameter do** | **Type of flow** | **LSHX Material** |
|---|---|---|---|---|
| 30 cm | 12 mm | 2.5 cm | Counter flow | copper |

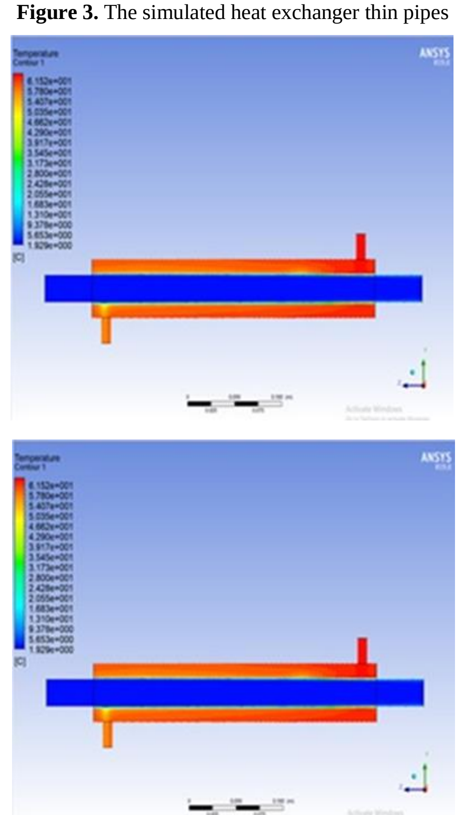
The simulated heat exchanger has two pipes one inside the other as shown in Fig. 3



(a)

(b)

**Figure 3.** The simulated heat exchanger thin pipes



(c)

(d)

**Figure 4.** The simulated heat exchanger thick pipes

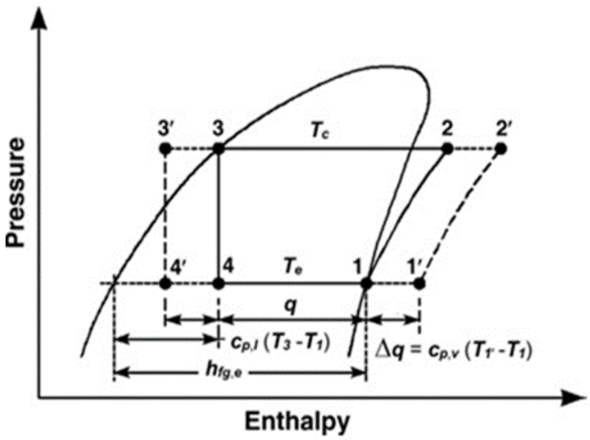



**Figure 5**. P-h Diagram of Basic VCRC (with and without heat exchanger) [16]

Fig. 5 Describes the P-h Diagram of basic VCRC (with and without HX) [16], The cycle 1,2,3,4 without heat exchanger, and the cycle 1′,2′,3′,4′ with heat exchanger.

## 4. Experimental setup

Thermal performance of VCRC was tested with R134a and compared with its alternative refrigerant R1234yf in two cases (with and without LSAHX).Three variable values of compressor speed (1000rpm,1700 rpm,2400rpm) ,and three values for thermal loads (500 watt,700 watt,1000 watt) were studied . Table 2 described the parameters studied in the experimental work.

**Table 2.** Parameters studied in the present experimental work

| Compressor speed (rpm) | Thermal load (Watt) |
|---|---|
| 1000 | 500 |
| 1700 | 700 |
| 2400 | 1000 |

The used device was an AACS which consist of multi-cylinder reciprocating compressor, fin-tube condenser, thermostatic expansion device and evaporator with (3 kW) cooling capacity, as shown in Fig. 6.
The experimental procedure was done by following the steps:

1. Charging the system with R134a.
2. Set the compressor speed on 1000 rpm &the thermal load on 500 watts.
3. After (35) minutes the device reached the steady state and data recorded from all thermocouples & pressure gages every (5) minutes.
4. Steps from (1) to (3) repeated with the second and third rotational speed each one with the three cases of thermal load and. The data was recorded every (5) minute.
5. Steps from (1) to (4) were repeated with R1234yf.
6. Heat exchanger was welded to the device.
7. Repeat steps from (1) to (5).

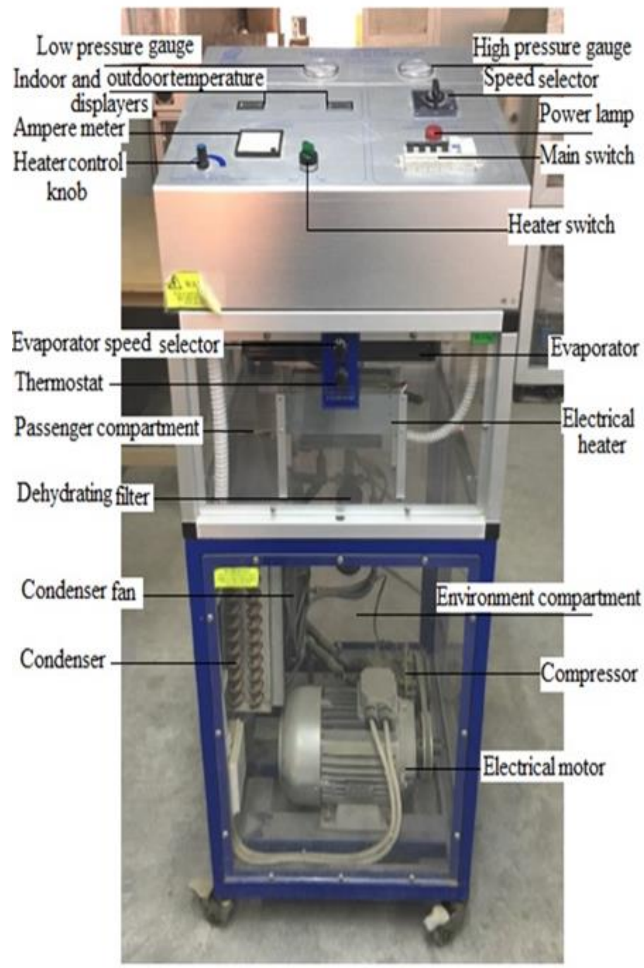



**Figure 6**. Test Rig

## 5. Thermotical analysis

To evaluate the thermal performance parameters of an ACS such as, COP, RE, heat rejected and compressor work, thermal properties of the refrigerants are required. Enthalpies at each inlet and outlet component of the cycle is extracted from Engineering Equation Solver (EES) software [17] according to the temperatures measured from the experimental work at each point of the cycle.

### 5.1. Energy Analysis of the Experimental Data

According to the first law of thermodynamic or conservation of energy for steady state [18] we need input and output states to start analytical solution.

5.1.1. Compressor

It is a device used to increase the pressure of the refrigerants. The energy equation [19] of this device is given as follows, Eq.1.

$$\dot{W}c = \dot{m}r(h_2 - h_1) \tag{1}$$

Where: $\dot{W}_c$ is the compressor work in (kW), $\dot{m}$ refrigerant to mass flow rate, $h_1$ and $h_2$ is the enthalpies of the inlet and outlet states of the compressor in (kJ/kg).

5.1.2. Condenser

One of VCRC component that doesn't need work to operate. Heat rejected in condenser [19] given as below, Eq.2.

$$Qcond. = \dot{m}r(h_2 - h_3) \tag{2}$$

Where: $Q_{cond}$. Is the heat rejected by the condenser in (kW), $h_3$ is the outlet enthalpy of the condenser in (kJ/kg).

5.1.3. Expansion Device

By assuming isenthalpic process in the expansion device, the inlet and outlet enthalpies of the capillary tube is the same [1], Eq. 3.

$$h_3 = h_4 \tag{3}$$

Where: h4 is the inlet enthalpy of the evaporator in (kJ/kg).

5.1.4. Evaporator

Evaporator's purpose is to obtain fluid with low pressure and low temperature from an expansion valve. Heat absorbed [17] by evaporator is given as below, Eq.4:

$$Qeva. = \dot{m}r\ (h_4 - h_1) \tag{4}$$

Where: $h_4$ is the inlet evaporator enthalpy in (kJ/kg).

5.1.5. Heat Exchanger

The purpose of heat exchanger is to exchanging heat between two fluids at various temperatures. It is increases the super-heating effect and sub-cooling effect [16].

Because of the variation in specific heat of the liquid and the vapor phases, the degree of sub-cooling and degree of super-heating are not equal. The actual heat transfer rate in liquid suction heat exchanger is found by [16], Eq. 5.

$$\dot{Q}_{LSHX} = Cp,v\ (T_1' - T_1) = Cp,l\ (T_3 - T_3') \tag{5}$$

Where: $C_{p,v}$ & $C_{p,l}$ is the specific heats of the vapor and liquid respectively in (kJ/kg·K). When adding LSHX compressor's work can be found by the relation below [1], Eq. 6.

$$Wcomp. = \dot{m}_r \Delta h'_{2-1} = \dot{m}_r(h'_2 - h'_1) \tag{6}$$

Heat rejected by condenser can found by the relation [1], Eq. 7.

$$Qcond. = \dot{m}_r\ (h_2' - h_3') \tag{7}$$

At throttling process, the enthalpies is equal as shown below [1], Eq. 8.

$$h'_3 = h'_4 \tag{8}$$

At evaporator the heat absorbed is found as shown below [1], Eq. 9.

$$Qevap. = \dot{m}_r\ (h^1 - h_4') \tag{9}$$

5.1.6. Coefficient of performance.

It is explaining the effect of changing external temperatures on the performance of the cycle or the ratio of useful heating and cooling supplied to the work required, Eq. 10.

$$COP\ actual = \frac{RE}{w_{actual,comp.}} = \frac{h_1 - h_4}{h_2 - h_1} \tag{10}$$

Where $w_{actual,comp}$ is the work per unit refrigerant mass in (kJ/kg) while RE is the refrigeration effect in (kJ/kg). When using LSHX the COP can be found by relation shown [16], Eq. 11.

$$COP' = \frac{h_1 - h_4\,'}{h_2\,' - h_1\,'} \tag{11}$$

### 5.2. Exergy analysis of the experimental data:

5. 2.1. Compressor.

Exergy destruction in compressor is given by the following Eq. 12.

$$\Psi_{des,comp} = To\ \dot{S}_{gen,comp}\ \dot{m}_r\ T_o(s_2 - s_1) \tag{12}$$

5. 2.2. Condenser.

Stream exergy in the condenser expressed by the following relation, Eq. 13.

$$\Psi_{des,cond} = \dot{m}_r T_o[(s_3 - s_2) + \left(\frac{q_{cond}}{T}\right)] \tag{13}$$

Where $T_o$ is the environment temperature in (K), T is the average air temperatures for inlet & outlet sides from the condenser. $T = ((T2 + T3)/2))$ in (K).

5.2.3. Expansion device.

Stream exergy for expansion device is given by, Eq. 14.

$$\Delta\Psi_{TXV} = \dot{m}_r[(h_3 - h_4) - T_o(s_3 - s_4)] \tag{14}$$

Exergy destruction for expansion device for no work neither heat transfer is given by, Eq. 15.

$$\Delta\Psi_{TXV} = T_o S_{gen,exp} = \dot{m}_r T_o(s_3 - s_4) \tag{15}$$

5.2.4. Evaporator

Evaporator stream exergy can be given by, Eq. 16.

$$\Delta\Psi_{\mathrm{des,evap}} = \dot{\mathrm{m}}_{\mathrm{r}}(\psi_4 - \psi_1) = \dot{m}_r[(h_4 - h_1) - T_o(s_4 - s_1)] \qquad (16)$$

Exergy destruction in the evaporator also can be found from the following relation, Eq. 17.

$$\Delta\Psi_{\mathrm{des,evap}} = To\dot{S}_{gen,evap} = \dot{m}_r\, T_o\,(s1 - s4 + \frac{RE}{T}) \qquad (17)$$

5.2.5. Heat Exchanger

Stream exergy for the heat exchanger can be obtained by the following expression, Eq. 18.

$$\Psi des, LSHX = \dot{m}_r[(h_3\_h_3')\ \_\ (h_1' - h_1)]\ \_\ T_o\ [(s_3\_s_3')\ \_\ (s_1' - s_1)] \quad (18)$$

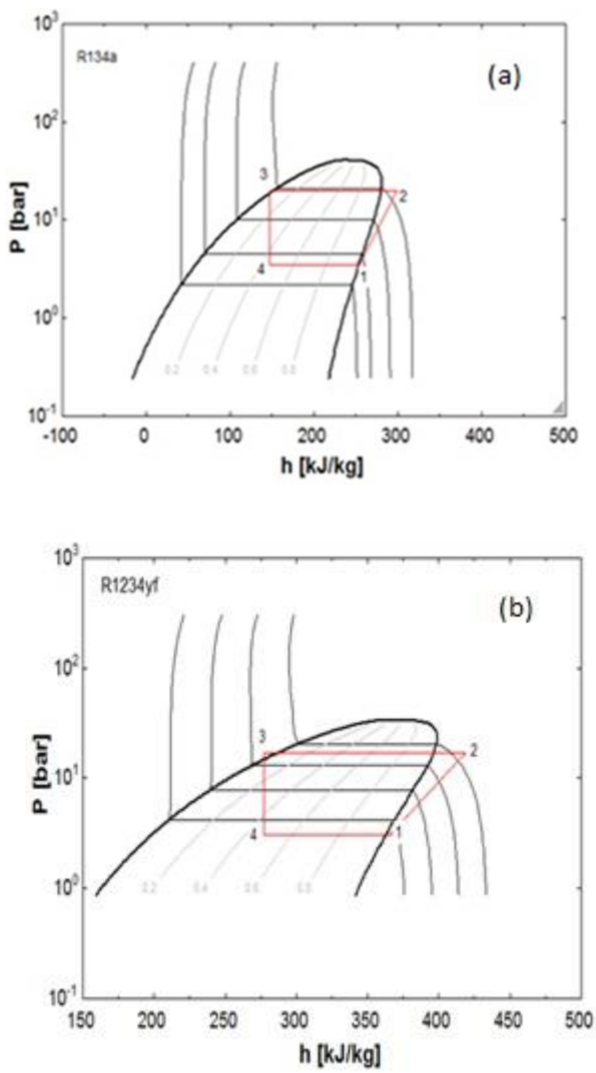



**Figure 7**. P_h Diagrams for R134 and R1234yf at 1700 rpm and 700 watt without LSHX **(a)** R134a **(b)** R1234yf

## 6. Results and discussion:

This experimental work studied the presence and absence of LSHX on an AAC work with refrigerants R134a and its alternative R1234yf .Three different values of compressors speed were used: 1000 rpm, 1700 rpm and 2400 rpm, however, three values of thermal loads 500 watt, 700 watt, and 1000 watt.

When LSHX was added, $\dot{m}_r$ was decreased, $\dot{m}_r$ of R134a with LSHX is less than that without LSHX by 21% ,and for R1234yf was decreased by 20% according to decreasing in density of refrigerants which leads to decreasing in volumetric compressor efficiency and then $\dot{m}_r$ will decrease , RE was increased because of increasing in sub cooling degree as shown in Fig.7 which represent P_h diagram without LSHX ,while Fig. 8 represent P_h diagram with LSHX.

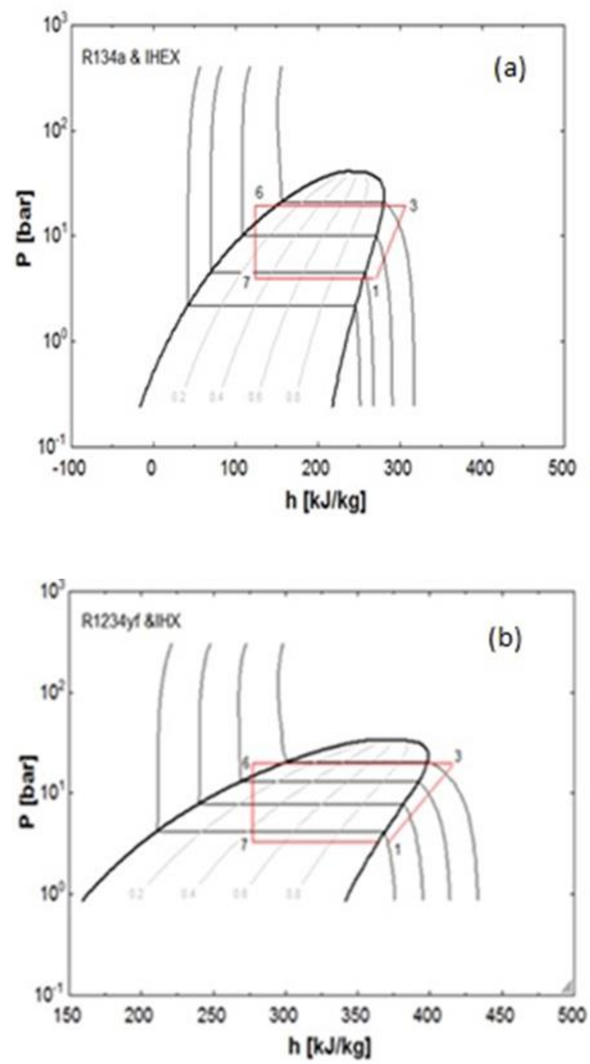



**Figure 8.** P_h Diagram for R134 and R1234yf with LSHX at 1700 rpm and 700 watt **(a)** R134a **(b)** R1234yf

Compressor work is decreasing, attributed to the decreasing mass flow rate of the refrigerants after the addition of LSHX, according to the increasing superheating degree in the evaporator outlet, which leads to increase in temperature at the compressor and condenser outlet, causing a decrease in refrigerant density, which leads to a decrease in mass flow rate and a decrease in load on the compressor, so the compressor work is decreasing., $W_{comp.}$ with R134a and LSHX was decreased by 14.6%,12.46 % and 40.12 % for the first, second, and third speed respectively, however, a decrease of 19.61 % and 30.18% for the first, second respectively was detected as R1234yf was used as described in Fig. 10.

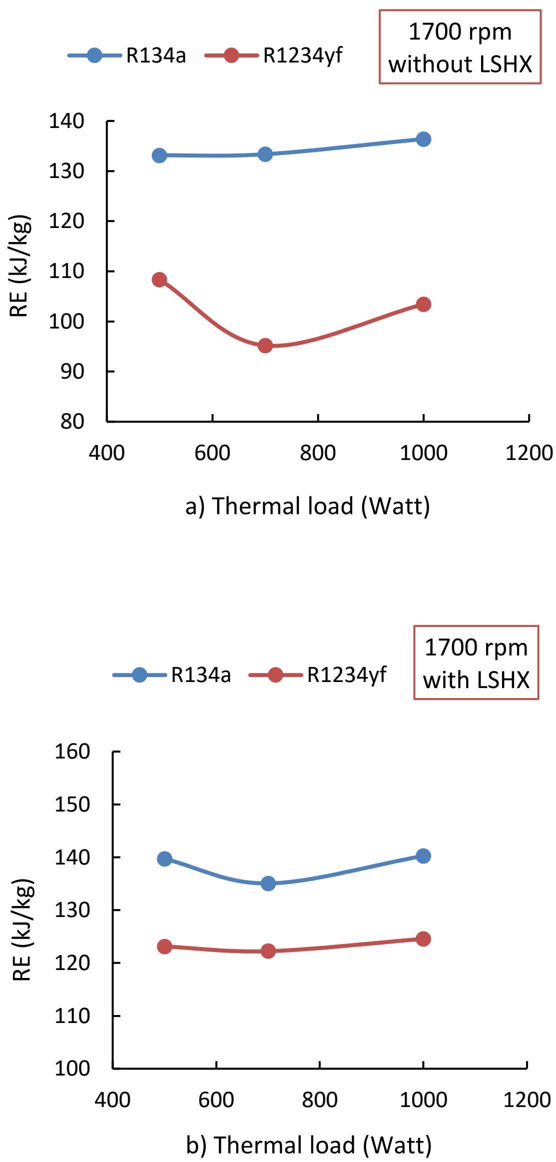



**Figure 9.** RE Vs. Thermal load at 1700 rpm for R134a and R1234yf **(a)** without LSHX **(b)** With LSHX

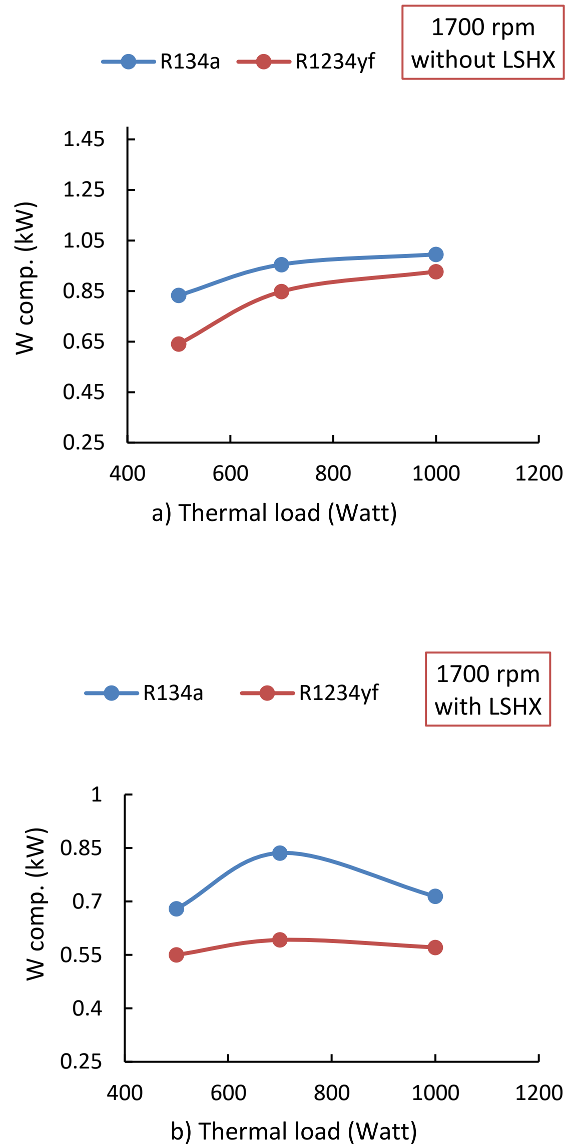



**Figure 10**. $W_{comp.}$ Vs. Thermal load at 1700rpm for R134a and R1234yf **(a)** without LSHX **(b)** With LSHX

COP was increased according to the increase in RE, and decrease in compressor work according to the relation:

$$COP_{actual} = \frac{Q_{evap.}}{W_{actual,comp.}} = \frac{RE}{w_{actual,comp.}}$$

COP with R134a and LSHX was increased by 6.93%, 25.1% and 12.6% for the first speed, second speed ,and third speed respectively, while when R1234yf was used COP increased by 11.7% , and 17.37% for the first speed and the second speed respectively as shown in Fig. 11.

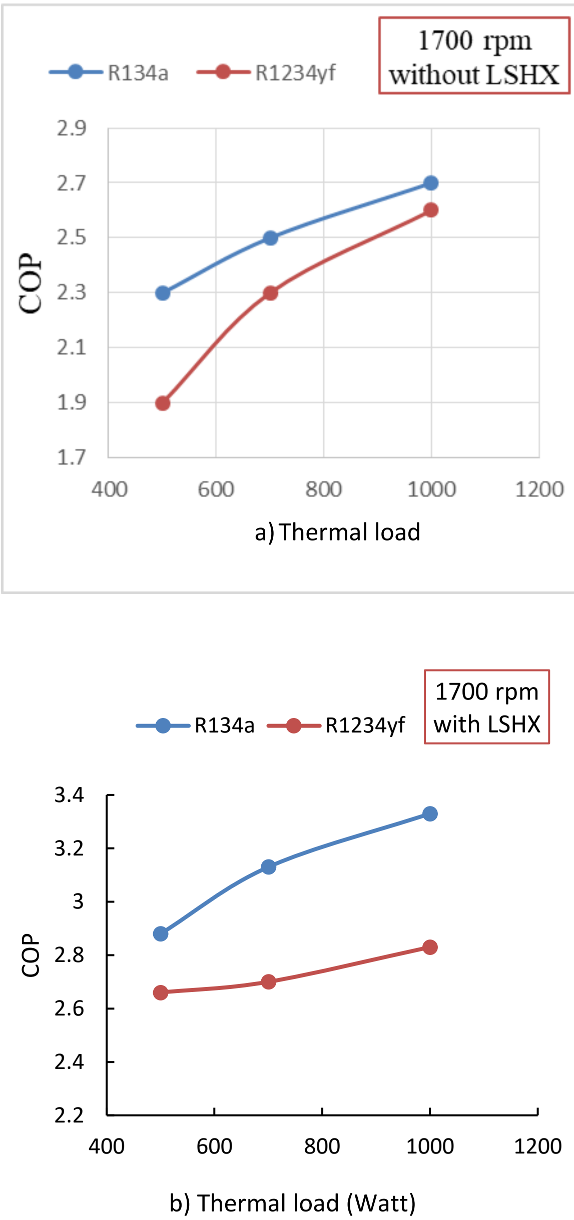



**Figure 11**. COP Vs. Thermal load at 1700 rpm for R134a and R1234yf **(a)** without LSHX **(b)** With LSHX

Heat absorbed in evaporator with R134a and LSHX was increased by 14.5% , 15.1% and 16.3% for the first, second ,and third speed respectively ,while when R1234yf was used heat absorbed increased by 11% and 13.5% for the first speed and second speed respectively.

Heat rejected by condenser increased when LSHX was added according to the increasing in sub-cooling degree and super heating degree which leads to decreasing in enthalpy that leaves the condenser and increasing in enthalpy entering the condenser, and the difference between the inlet and outlet enthalpies gives the total heat rejected by condenser, the increased values were 13.23 % ,10.58% and 10.20% for the first, second ,and third speed respectively ,while when R1234yf was used heat rejected decreased by 4.14% for the first speed and 11.1% for the second speed as shown in Fig.12.

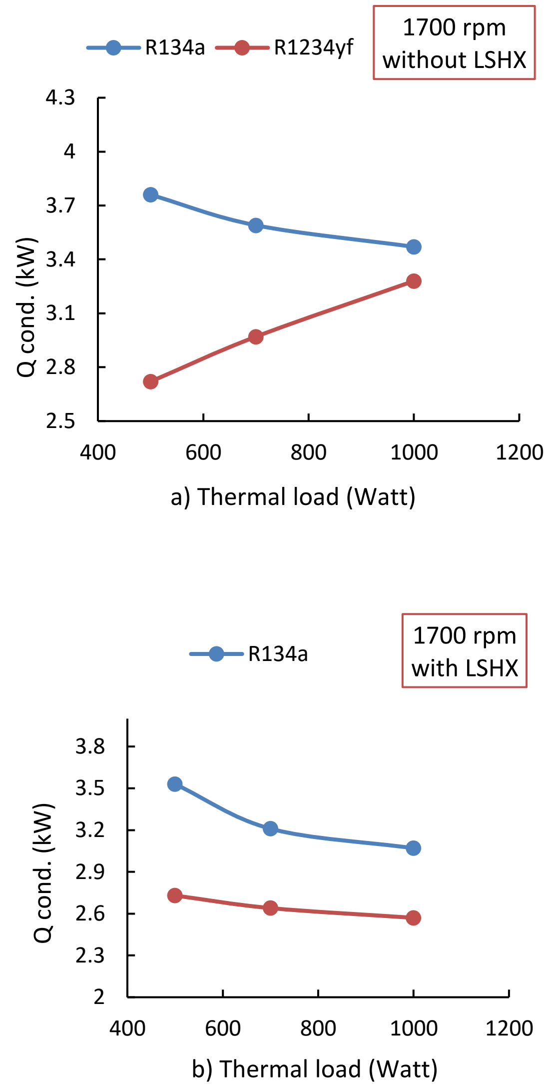



**Figure 12.** Q $_{cond}$. Vs .Thermal load at 1700 rpm for R134a and R1234yf **(a)** without LSHX **(b)** With LSHX

Exegy destruction in compressor decreased according to the decreasing in compressor work, so the compression ratio was less and the exegy efficiency was higher, which lead to lower the exergy destruction by 2.3% for the first speed, 35.6% for the second speed, and 6.29% for the third speed when R134a was used with LSHX, as R1234yf was used exergy destruction in compressor having LSHX was higher than the one without

LSHX by 15.8% and 44.48%, for the first and the second speed respectively as shown in Fig. 13.

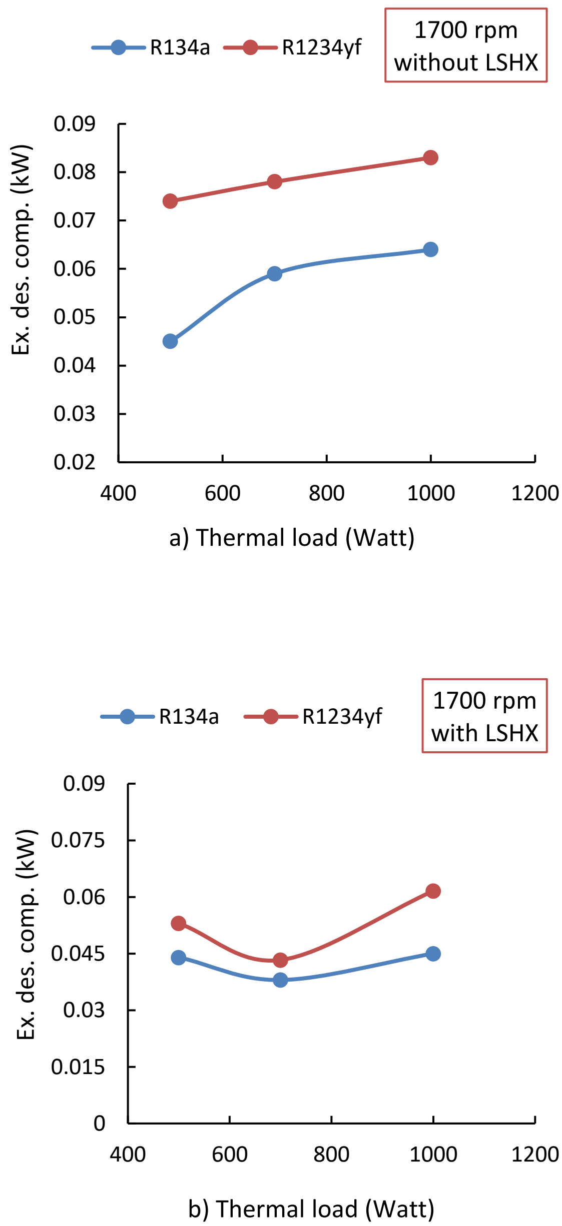



**Figure 13.** Ex. des. $_{comp.}$ Vs. Thermal load at 1700 rpm for R134a andR1234yf **(a)** without LSHX **(b)** With LSHX

Exergy destruction in condenser when LSHX was added with R134a implementation was increased by 4.49% for the first speed, 5.75% for the second speed, and 8.15% for third speed, while when R1234yf was used exergy destruction was increased by 19.84 % and 31.10 %, for the first and the second speed respectively as shown in Fig. 14. Exergy destruction in thermostatic expansion device with R134a and LSHX was increased by 11.84%, 69.46% and 49.9%, for the first, second, and third speed respectively, while when R1234yf was used and LSHX, exergy destruction was higher than that without by 49.6%, 66.6% for the first speed and the second speed respectively as shown in Fig. 15.

This increment attributed to the increasing in the entropy generation with the increasing in temperature due to higher pressure and friction

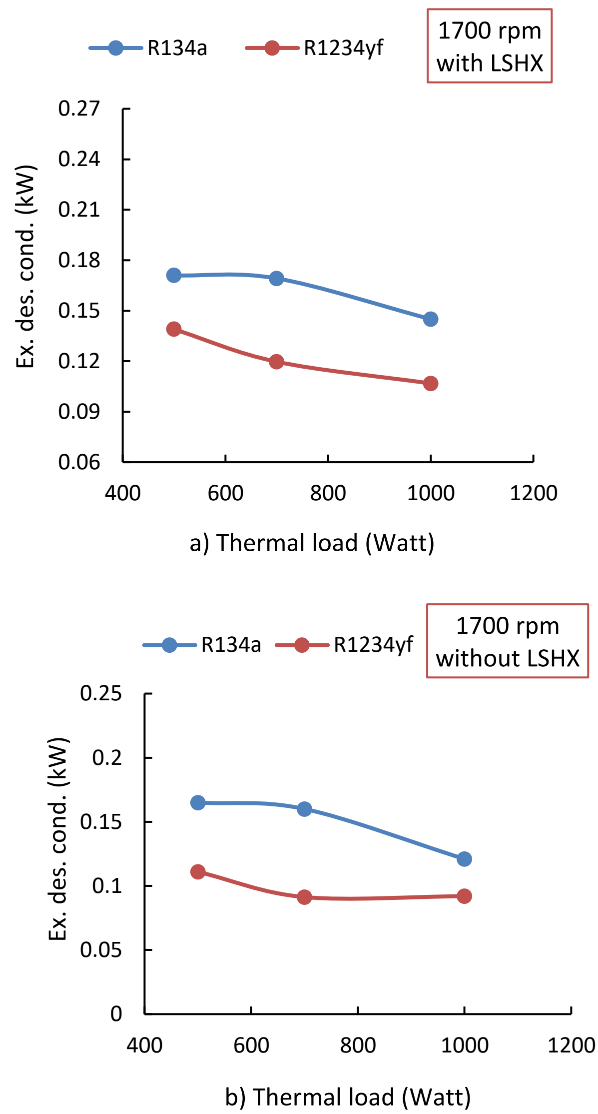



**Figure 14**. Ex. des. $_{Cond.}$ Vs. thermal load at 1700 rpm for R134a and R1234yf **(a)** without LSHX **(b)** With LSHX

Exergy destruction in evaporator with R134a and LSHX was higher than that without LSHX by 35.11 %, 45.5%, 37.03 % for the first, second, and third speed respectively, while when R1234yf exergy destruction in evaporator was less than the case of having no LSHX by 45.5% for the first sped and 23.12% for the second speed respectively as shown in Fig. 16.

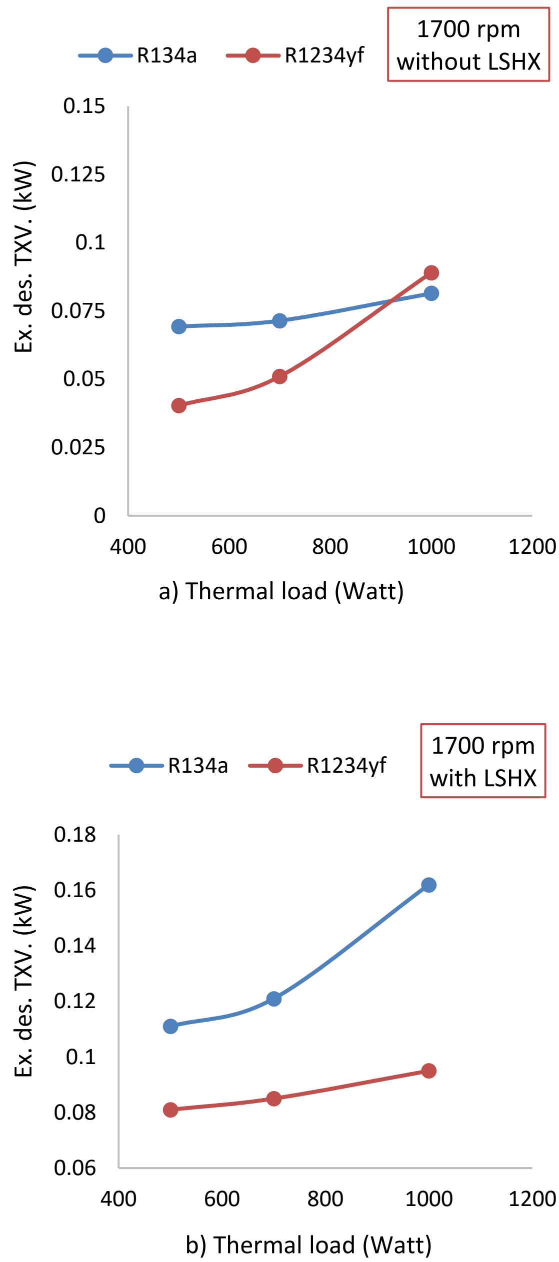



**Figure 15.** Ex. des. $_{TXV}$. Vs. thermal load at 1700 rpm for R134a and R1234yf **(a)** without LSHX **(b)** With LSHX

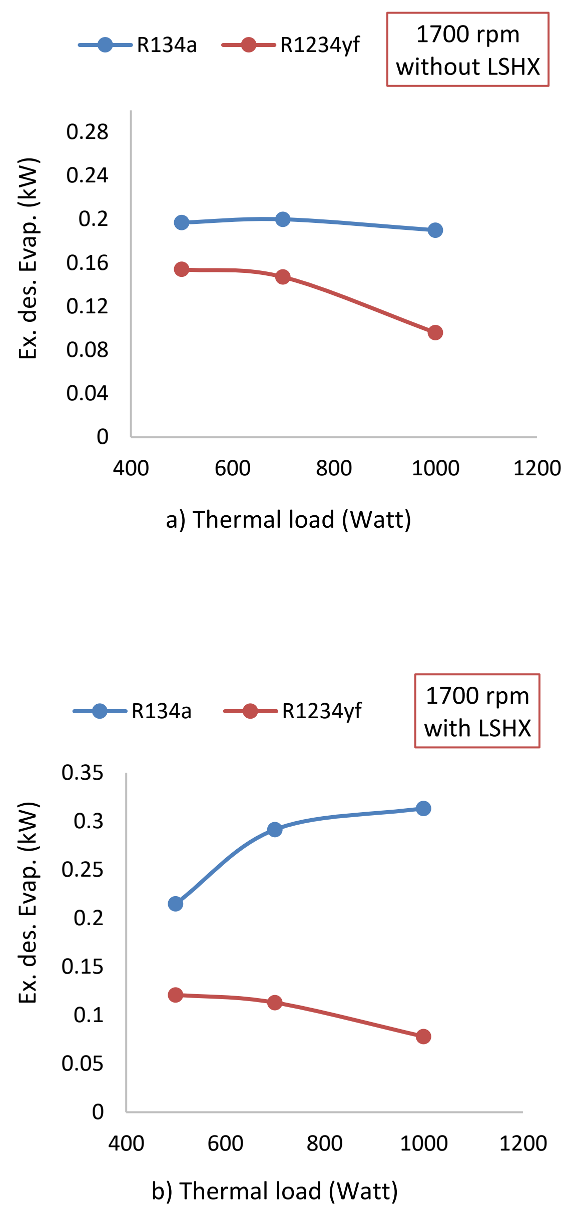



**Figure 16.** Ex. des. $_{Eva.}$ Vs. thermal load at 1700 rpm for R134a and R1234yf **(a)** without LSHX **(b)** With LSHX

Figures 10- 16 show the results of energy destruction in evaporator with R134a and LSHX. With all results the R134a is present higher than that without LSHX by different rations. While R1234yf exergy destruction in evaporator was less than the R134a. The results are agreement with the literature and for high thermal load, more tests are required in the future. In general, the experimental setup in this work provides wide range of experimental data that can be measured.

## 8. Conclusion

An AACS with LSHX and two refrigerants R134a and R1234yf were used. The experiment was carried out with three values of compressor speeds (1000, 1700, and 2400 rpm) and three thermal loads (500, 700, and 1000 watt).Energy and exergy were analyzed to all component system.$\dot{m}_r$. Was decreased when LSHX was added, according to increase in compressor temperature the density will decrease, according to the relation $\dot{m}_{ref.} = \dot{V} * \rho$ ,$\dot{m}_r$ will decrease. COP of the cycle decrease when the speed of the compressor increase, according to the decreasing in RE and increasing in $W_{comp}$., while increase when LSHX was added by 25.1% for R134a and 17.3% for R1234yf according to the increasing in RE and sub-cooling degree. $W_{comp}$. Was increased when the speed of compressor increased, because of increasing in work done by motor, while decrease when LSHX was added by 12.46% for R134a and 31.1% for R1234yf according to the decreasing in $\dot{m}_r$. RE increased when LSHX was added by 4.2% for R134a and 2.3% for R1234yf according to the increasing in sub-cooling degree. Qcond. Was increased when the speed of compressor increased, also increased when LSHX was added by 10.58% for R134a and 11.1% for R1234yf because of increasing in sub-cooling degree which leads to increasing in heat rejected by condenser. Total exergy destruction in the system was increased, according to the increasing in air streams temperature due to increasing in temperature difference between the air stream and refrigerant in condenser and evaporator which cause decreasing in exergetic performance.

## Authors' contribution

All authors contributed equally to the preparation of this article.

## Declaration of competing interest

The authors declare no conflicts of interest.


## Funding source

This study didn't receive any specific funds.


## Data availability

The data that support the findings of this study are available from the corresponding author upon reasonable request.

## REFERENCES


[1] Tritjahjono, R.I. , Sumeru, k. , Setyawan, A., and Sukri , M.F.,(2019). *Evaluation of Subcooling with Liquid-Suction Heat Exchanger on the Performance of Air Conditioning System Using R22/R410A/R290/R32 as Refrigerants*. Journal of Advanced Research in Fluid Mechanics and Thermal Sciences . Vol. 55, Issue 1, pp. 1–11.

[2] Nagasai, B.P, Kumar. K.D., Reddy, CH.CH., Rao, T.S.,(2020). *Experimental Investigation on Performance of Vapour Compression Refrigeration System with Integrated Sub-Cooling*. Test engineering & management. Vol. 83, Issue November, pp. 5104-5111.

[3] Joshi , Y.,Zanwar, D., Joshi, S., (2021) . *Performance investigation of vapor compression refrigeration system using R134a and R600a refrigerants and Al2O3 nanoparticle based suspension*. Materials Today: Proceeding. Vol. 44, pp. 1511–1519. doi:10.1016/j.matpr.2020.11.732 10.1016/j.matpr.2020.11.732

[4] Babarinde. T.O., Akinlabi, S.A., Madyira, D.M., (2018). *Enhancing the Performance of Vapour Compression Refrigeration System using Nano Refrigerants*. IOP Conf. Series: Materials Science and Engineering. ICESW 2018. Institute of Physics Publishing. https://doi.org/10.1088/1757-899X/413/1/012068

[5] Bilir, N., Ersoy, H.K.,(2009). Performance improvement of the vapour compression refrigeration cycle by a two-phase constant area ejector. *International Journal of Energy Research*. Vol. 33, Issue 5, pp. 469_480. https://doi.org/10.1002/er.1488

[6] Khan, J.R., Zubair, S.M.,(2000). *Design and rating of an integrated mechanical-subcooling vapor-compression refrigeration system.* Energy Conversion and Management . Vol. 41,Issue 11,pp. 1201-1222. https://doi.org/10.1016/S0196-8904(99)00169-7

[7] Djuanda, D., (2018). Performance study of double pipe internal heat exchanger in R-410a air conditioning system. in AIP Conference Proceedings. https://doi.org/ 10.1063/1.5046592

[8] Mota-Babiloni, A., Esbrí, J.N., Miralles, V.P., Cervera, A.B., and Maiorino, A.,(2019). *Experimental influence of an internal heat exchanger (IHX) using R513A and R134a in a vapor compression system* . Applied Thermal Engineering .Vol. 147,pp. 482-491. https://doi.org/10.1016/j.applthermaleng.2018.10.092

[9] Cengel, Y.A., Boles, M.A., (2015). *Thermodynamics* , 8th ed. 2015,United State. ISBN:987-0-07-339817-4.

[10] Alhendal, Y., ,Gomaa, Ab., ,Bedair, G., and Kalendar, A., ,(2020).*Thermal Performance Analysis of Low-GWP Refrigerants in Automotive Air-Conditioning System*. Advances in Materials Science and Engineering. Vol. 2020 https://doi.org/10.1155/2020/7967812 .

[11] Gungor, U., Hosoz. M., (2021). *Experimental performance evaluation of an R1234yf automobile air conditioning system employing an internal heat exchanger* . International Journal of Automotive Engineering and Technologies .Vol. 10,Issue 1, pp. 50_59. https://doi.org/10.18245/ijaet.842426

[12] Sharif, M.Z., Azmi1, W.H., Zawawi. N.N M., Mamat, R., and Hamisa, A.H.,(2020). R1234yf vs R134a in automotive air conditioning system: A comparison of the performance. In in IOP Conference Series: Materials Science and Engineering. Institute of Physics Publishing. https://doi.org/10.1088/1757-899X/863/1/012049

[13] Mahmood, R.A.,(2020). *Case study of liquid suction heat exchanger in a mechanical refrigeration system using alternative refrigerants.* International Journal of Engineering & Technology. Vol. 9 ,Issue 3,pp. 644_649. https://doi.org/10.14419/ijet.v9i3.30777

[14] Qian S., Yu J., Yan G.,(2017). *A review of regenerative heat exchange methods for various cooling technologies*. Renewable and Sustainable Energy Reviews. Vol. 69 , Issue November 2016 ,pp. 535-550

[15] Abdulmajeed, B.A., Jawad, H.R., (2019). *Analysis of Shell and Double Concentric Tube Heat Exchanger Using CFD Application.* Journal of Engineering. Vol. 25,Issue 11, pp. 21 _36. https://doi.org/10.31026/j.eng.2019.11.02

[16] Wantha, CH.,(2019). *Analysis of heat transfer characteristics of tube-in-tube internal heat exchangers for HFO-1234yf and HFC-134a refrigeration systems*. Applied Thermal Engineering .Vol. 157, Issue October 2018 ,pp. 113747. https://doi.org/10.1016/j.applthermaleng.2019.1137.

[17] Shah, Z.A., Zheng, Q., Mehdi, Gh., Ahmad N., Waleed, R., Rehman, A.U., and Raza, A.,(2020).*The Theoretical Framework of the Modified Organic Rankine Cycles for Improved Energy and Exergy Performances*. International Energy Journal. Vol. 20, pp. 169-180

[18] Arora, A.C., (2009) . Refrigeration and Air Conditioning . 3rd ed. 2009. New Delhi , ISBN: 9780070083905

[19] YILDIRIM1. R., KUMAŞ, K., AKYÜZ , A. Ö., (2022). *Comparative energetic analysis of a refrigeration system by using r123 and its alternative r514a refrigerants*. Techno Science . Scientific Journal of Mehmet Akif Ersoy University . Vol. 5,Issue 2 ,pp. 45_50

[20] Ghanbarpour, M., Mota-Babiloni, A., Badran, B.E., and Khodabandeh, R.,(2021) *Energy, exergy, and environmental (3E) analysis of hydrocarbons as low GWP alternatives to R134a in vapor compression refrigeration configurations*. Applied Sciences (Switzerland) .Vol.11 ,Issue 13. https://doi.org/10.3390/app11136226