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Experimental Assessment of R134a and Its Lower GWP Alternative R1234yf in an Automobile Air Conditioning System

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ABSTRACT

Reducing global warming potential (GWP) of refrigerants is needed to the decrease of ozone-depleting of refrigeration systems leakages. Refrigerant R1234yf is now used to substitute R134a inside mobile air conditioning systems. Thermodynamic properties of R1234yf are similar to R134a. Also, it has a very low GWP of 4, compared to 1430 for R134a, making it a proper choice for future automobile refrigerants. The purpose of this research is to represent the main operating and performance differences between R1234yf and R134a. Experimental analysis was carried out on the automotive air conditioning system (AACS) with 3 kW nominal capacity, to test and compare the performance of R134a with R1234yf. Experiments were accomplished for both refrigerants in almost the same working conditions and procedure with a range of ambient temperature varied from 26^{o}C to 50^{o}C. Parameters studied were ambient temperature, type of refrigerant in the system at compressor speed 1450 rpm, and internal thermal loads of passenger room. The performance characteristics of the system, including COP and cooling capacity, were studied by changing different parameters. The results show that COP of R134a is higher than R1234yf by 12.6%, while the refrigeration effect of R134a is higher than R1234yf by 25%. This shows that R1234yf is a suitable and good candidate for drop-in replacement of R134a in AACS.

Keywords: automotive air conditioning, Global Warming Potential, R134a, R1234yf, energy analysis.

التقييم التجريبي لـ R134a مع البديل منخفض الاحترار العالمي R1234yf في نظام تكييف هواء سيارة

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الخلاصة

الحد من إمكانات الاحترار العالمي (GWP) لموائع التثليج ضروري لتقليل استنفاد طبقة الأوزون في أنظمة التثليج. يستخدم مائع التثليج R1234yf حالياً كبديل عن R134a داخل أنظمة تكييف الهواء والتثليج المتنقلة. ان الخصائص الحرارية الديناميكية لمائع التثايج R1234yf تشبه R134a. كذلك، تملك هذه الموائع الجديدة معامل احتباس حراري قليل يقارب 4 ،

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مقارنة مع قيمته لـ R134a والذي يعادل 1430، مما يؤدي الى اعتباره اختيار مناسب وبديل ملائم لمائع التثليج للسيارات في المستقبل. أن الغرض من هذا البحث هو تمثيل الاختلافات التشغيلية الرئيسية وكذلك معامل الأداء بين R1234yf و R134a. تم إجراء التحليل التجريبي لنظام تكييف هواء السيارات بسعة تبريد 3 كيلو واط ، لتقييم ومقارنة أداء R134a مع R1234yf. وقد اجريت التجارب لكلا الممائعين في نفس ظروف العمل تقريباً مع تغيير درجات الحرارة المحيطة من 26 درجة مئوية إلى 50 درجة مئوية. تم تغيير درجة حرارة المحيط عند سرعة الضاغط 1450 دورة في الدقيقة ، والحمل الحراري الداخلي لغرفة الركاب تم تثبيته عند 1300 واط. تمت دراسة خصائص الأداء للنظام ، بما في ذلك COP وقدرة التبريد ، من خلال تغيير المتغيرات المختلفة اعلاه. أظهرت النتائج أن معامل الاداء لـ R134a أعلى من R1234yf بنسبة 12.6 ٪ ، في حين أن التأثير التبريدي لـ R134a أعلى من R1234yf بنسبة 25 ٪. هذا يدل على أن R1234yf هو مرشح مناسب وجيد لاستبدال R134a في نظام تكييف السيارات.

الكلمات الرئيسية: تكييف هواء السيارات ، الاحترار العالميGWP ، مائع التثليج R134a ، مائع التثليج R1234yf، تحليل الطاقة.

1. INTRODUCTION

Refrigeration and the components that work on this principle have indeed made the human life easier, the cooling system in the vehicle is designed to keep cold, comfortable, and safe when the temperature outside is high. Most automobile air conditioning systems in Iraq operate with R134a refrigerant. Research has revealed that this refrigerant is harmful to the environment. Today, there are many refrigerants alternatives of R134a, but the question which is of utmost importance is and which refrigerant works the best. So, the current vehicle air conditioning system refrigerant (R134a), which is used in all motor vehicles since the early nineties, will be phased out in the coming years. **Fig. 1** shows the refrigerant improvement.

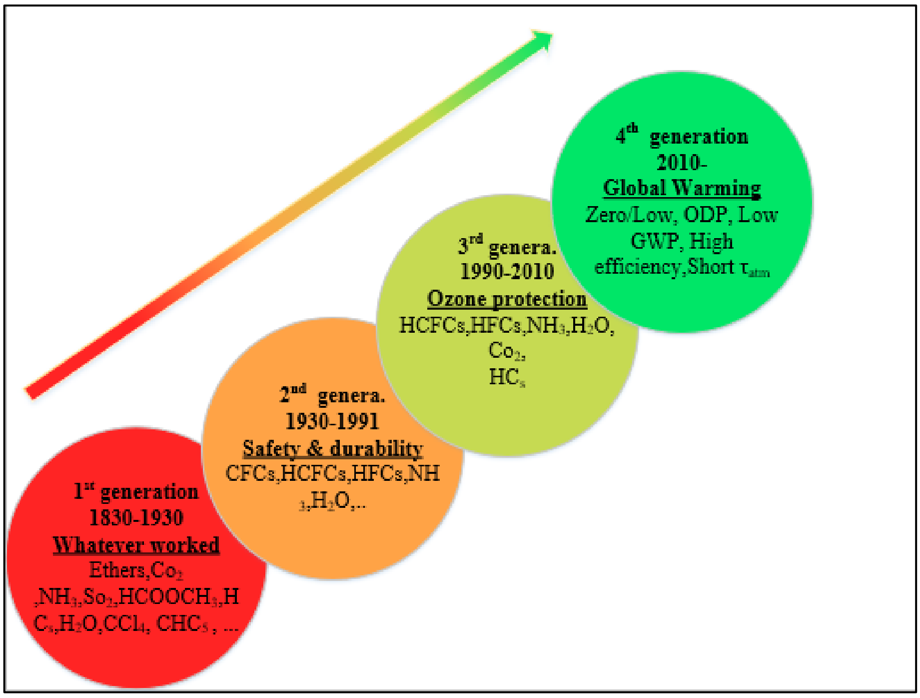


Figure 1. Refrigerant Improvement.

Many researchers investigated experimentally and numerically the performance of R1234yf as a replacement for R134a in the automotive air conditioning system. **Zilio et al., 2011** analysed experimentally and numerically the performance of R1234yf as a replacement for R134a in the automobile air conditioning system. The system consists of four main components compressor, condenser, thermal expansion valve (TXV), and the evaporator. It has a cooling capacity of 5.8 kW and volumetric flow rate of compressor 7.8 m3/h and applied some modification on the system. The experimental results showed that the coefficient of performance (COP) and cooling capacity for the R134a system was higher than R1234yf. While the numerical simulation showed that they enhance the front area for the condenser and the evaporator moreover the COP for R1234yf was more than of R134a at same cooling capacities. **Lee and Jung, 2012** used experimentally R1234yf as an alternative for R134a in the heat pump bench (HPB) in winter and summer weather environments. The tested calculations showed that it is possible to use R1234yf as the alternative of R134a due to very low global warming potential (GWP) and zero ozone depletion (ODP), as well the coefficient of performance (COP) for R134a was 0.8 - 2.7% higher than R1234yf. While the capacity of R134a was about 4% higher than of R1234yf. As for charge of R134a, it was 10-11% greater than that of R1234yf. The temperature of compressor discharge for R134a was 6.4 °C - 6.7 °C higher than that of R134a. **Cho et al. 2013 studied** experimentally the performance of automotive air conditioning (AAC) evaluated by charging the R1234yf and R134a. Then they installed an internal heat exchanger (IHX) to improve system performance. R1234yf showed lower cooling capacity by 7% and the power consumption up to 4%. When comparing the performance between two refrigerants, it revealed that the cooling capacity decreased about 7% while COP lowered by 4.5% at the R1234yf system without the IHX, but when IHX was used, those reduced by up to 2.9% and 1.8%, sequentially. **Ansari et al. 2013** used a mathematical computational model to calculate a coefficient of performance (COP), exergy efficiency, exergy destruction and efficiency in the vapor compression refrigerant system operated on the working fluid R1234yf and R1234ze as a replacement of R134a was developed. They found that the exergy efficiency and COP for R1234ze and R134a were the same; it decreased by 5% when increasing the evaporator temperature, while with R1234yf it was increased by 5%- 14.5% respectively. The exergy destruction occurred in the condenser, compressor, expansion valve and evaporator, respectively. **Molés et al., 2014** studied a single stage vapour compression system evaluated theoretically by using R1234ze (E) and R1234yf. They observed that the COP increased from 11% to 20% of R1234ze and incremented of 9% to 15% for R1234yf compared with the R134a. Using an IHX in a cycle led to COP of HFO1234yf of 4% to 8% smaller than R134a and thermal power evaporator (Q_e) about 4% to 7% lower. With R1234ze the Q_e was about 25% to 27% lower but COP equal the R134a. Finally, the value of IHX effectiveness was above 45%. **Sethi et al. 2016** studied experimentally the performance of the GWP as R1234ze(E) and HFO1234yf as alternative for HFC134a evaluated in the vending machine. The theoretical comparison showed that the pressure for R1234yf was approximately similar to R134a while the pressure of R1234ze (E) was lower than R134a; that affected on the decreased capacity about 25%. While the experiments indicated that the performance of R1234yf and R134a was equal. R1234ze (E) had a lower cooling capacity about 25%. **Shi et al., 2016 studied** the performance of R1234yf system experimentally, and the improvement of the system through the thermal expansion valve (TEV). The results indicated that the performance of R1234yf system improved significantly by adjusting the setting of the TEV. The TEV charged with R134a the system showed the optimal system. Therefore, when comparing with the original TEV, for the R1234yf system the COP increased by 8% and cooling capacity increased by 11.3%. **Direk et al. 2017a** studied experimentally the performance of R1234yf as a replacement to R134a in the AACS studied. The results indicated that the COP and cooling capacity for R1234yf was 7.5 % to 16.5 % and 13.9 % to 20.4 lower than that of R134a, sequentially. As well as determine the effect of internal heat exchanger (IHX) on the experimental system with HFO1234yf. Also, they found that the cooling capacity of R1234yf was increased by 6.3 % to 8.6 % and 6.4 % to 9.9 % when raising the temperature of airstream about (27 °C and 35 °C) sequentially. Moreover, the COP of the system was raised by 2.8 % to 7.4 % and 2.4 % to 4.8 % when raising the temperature of airstream about (27^{o}C and 35^{o}C) sequentially. **Vaghela, 2017** studied theoretically the alternative refrigerants substitute for R134a evaluated in the automotive air conditioning system (AACS), as R1234yf, R404A, R152a, R407C, R600a, R290, R410A and compared it with the R134a. Using the software REFPROP and engineering equation solver (EES) acquired the thermodynamic properties. The results indicated that R600a and R290 couldn't be the replacement of R134a in the AACS because of high flammability. The saturation pressure for R410A, R407C and R404A was very large subsequently it couldn't be used in AAC system. Finally, R1234yf had the COP 6.3% smaller than R134a, so it was a suitable alternative refrigerant as a drop-in replacement of R134a in AAC system due to very small GWP. Wherefore, could be replaced in the AACS with the smallest modification. **Sethi and Hrnjak, 2014** studied and compared the oil retention and pressure drop characteristics of refrigerants R1234yf and R134a with POE32 oil. They used high-speed videos to identify the flow regimes as the mass flux varied. Results showed that R1234yf have the same cooling capacity with R134a, as well as very similar oil retention. In the present research, the energy analyses for an AACS originally working with R134a was experimentally carried out at the ambient temperature from 26 to 50^{o}C by using R1234yf, and Table 1 showed the characteristics of the refrigerants.

Table 1. Characteristics of the refrigerants.

Thermodynamic Property	R1234yf	R134a
Chemical Formula	$C_3F_4H_2$	CF_3CH_2F
GWP	4	1430
ODP	0	0
Molar Mass (kg/kmol)	114.04	102.03
Boiling Point at 1 atm (K)	243.70	247.08
Freezing Point (K)	-	169.85
Critical Pressure (MPa)	3.38	4.90
Critical Temperature (K)	367.85	374.21
Critical Density (kg/m^3)	478.01	511.90

2. TEST APPARATUS

2.1 Test Rig

To evaluate the performance of alternative refrigerant R1234yf, it was tested in AACS and compared to a baseline established with R134a.The AACS used for test is composed of multi-cylinder reciprocating compressor including an electromagnetic clutch, conventional condenser type fin-tube and the tube made from aluminum, internal equalizer types for thermostatic expansion valve and evaporator is designed the same as used in the automobile. The main parameters are shown in Table 2.

Table 2. Parameters of the main components.

Component	Description
Compressor	Displacement 87 cc, Max. Speed 2900 rpm
Condenser	330mm x 330mm x 90 mm, No. of rows 4, 26 tubes at each row, type air-cooled, fin tube type
TXV	Internal equalizer type
Evaporator	Cooling capacity 3kW, fin tube type

The experiment tests were carried out to specify the performance of the AACS for different conditions. Therefore, the test procedure is described by the following steps and the condition of operating as shown in **Table 3** :

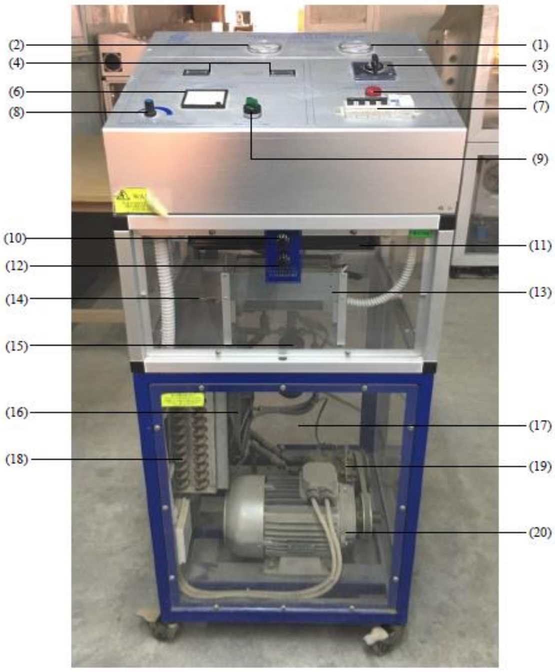
1. Charging the system in the refrigerant.
2. The ambient temperature varied from 26 to 50 ^{o}C by 4 increments.
3. The compressor rotational speed was 1450 rpm.
4. The thermal load inside the passengers' compartment was set on 1300 W.
5. Data from all thermocouples were recorded every 15 minutes.
6. The steps from (1) to (5) were repeated for a new refrigerant.

Table 3. Condition of operating

Description	Range
Ambient Temperature	26 to 50^{o}C
Compressor Speed	1450 rpm
Thermal Load	1300 Watt
Refrigerant	R134a – R1234yf

2.2 Instrumentation and Measurements

A test room with 2×1.5×2 m dimensions was used to simulate the environment, and it was made from sandwich panel covered by galvanized sheets of iron. The test room is so highly insulated that the heat was enclosed inside. Heater battery with a capacity of (2200 watts) was used to simulate the environment temperature. A thermocouple type K was used to measure temperature in the different locations and the number of thermocouples was 12 distributed on the inlet and outlet of each component. A turbine flow meter was used to measure the volumetric flow rate for the refrigerant. Two pressure gauges of bourdon type used to evaluate the low and the high pressure for the system. **Fig. 2** showed the test rig. **Fig. 3** showed a schematic diagram for the AACS.



(1) High-pressure gauge (2) Low-pressure gauge (3) Compressor speed selector (4) Indoor and outdoor temperature displayers (5) Power lamp (6) Ampere meter (7) Main switch (8) Heater control knob (9) Heater switch (10) Evaporator speed selector (11) Evaporator (12) Thermostat (13) Electrical heater (14) Passenger compartment (15) Dehydrating filter (16) Condenser fan (17) Environment compartment (18) Condenser (19) Compressor (20) Electrical motor

Figure 2. Test Rig.

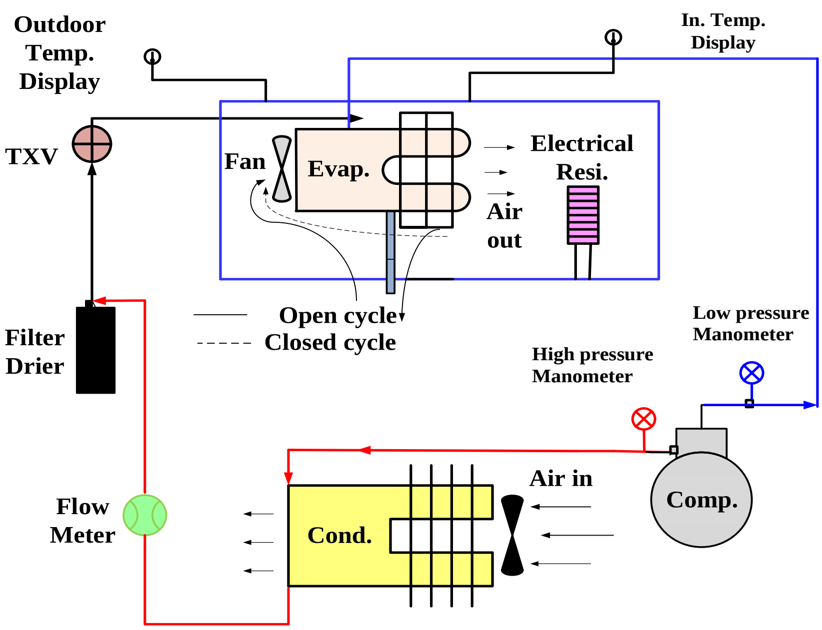


Figure 3. Schematic diagram to show the sequence of the cycle.

2.3 Experimental Data
To determine the performance parameters such as COP, cooling capacity, heat rejection, pressure ratio, and compressor work, it is required to find the thermodynamic properties of the refrigerant at selected points on the cycle. The experimental results are calculated based on the refrigerant enthalpy at the inlet and outlet of each component. Enthalpy values are calculated using the average measured temperature and pressure of the refrigerant at each state point. Properties of refrigerants are extracted from the engineering equation solver (EES) software. The volume flow rate of refrigerant is measured directly by a turbine flowmeter.

3. ASSUMPTIONS

1. Steady-state
2. The fan work is so small that can be neglected.
3. Changes in potential and kinetic energies are neglected.
4. Adiabatic process in expansion valve and compressor.
5. No heat loss over the rubber tubes.
6. The pressure drop across the discharge valve is isenthalpic process.
7. The oil effects on the refrigerant properties are neglected.

4. ENERGY ANALYSIS
A mathematical model was developed in this study to predict the performance for AACS charged with R134a and its alternative refrigerants R1234yf and compare between them. Depending on the conservation of energy equation or the first law of thermodynamics for steady-state flow given by **Çengel and Boles, 2005** needs only input and output states to begin analytic calculations.

4.1 Compressor

A compressor is used to get high pressure for a refrigerant. The work for this device was supplied by an external source as a rotating shaft. Then the work of compressor can be found theoretically and actually as in Eq. 1 and 2 below respectively **Khalifa, et al.2016**:

$$W_{\text{ideal, comp.}} = \dot{m}_r \Delta h_{2s-1} = \dot{m}_r (h_{2s} - h_1) \quad (1)$$
$$W_{\text{actual, comp.}} = \dot{m}_r \Delta h_{2-1} = \dot{m}_r (h_2 - h_1) \quad (2)$$

where $W_{\text{ideal, comp.}}$ is the required work in the process of isentropic compression in (kW), (h_{2s})is the outlet enthalpy from the compressor which is related to the processes of isentropic compression in (kJ/kg), (h_1) is the inlet enthalpy at the compressor in (kJ/kg), $W_{\text{actual, comp.}}$ is the actual work required of the non-isentropic compression process in (kW) and (h_2) is the outlet enthalpy related to the actual cycle in (kJ/kg) as indicated by the curve (1-2s) & (1-2) in **Fig. 4** respectively **Mahdi,et al. 2018**.

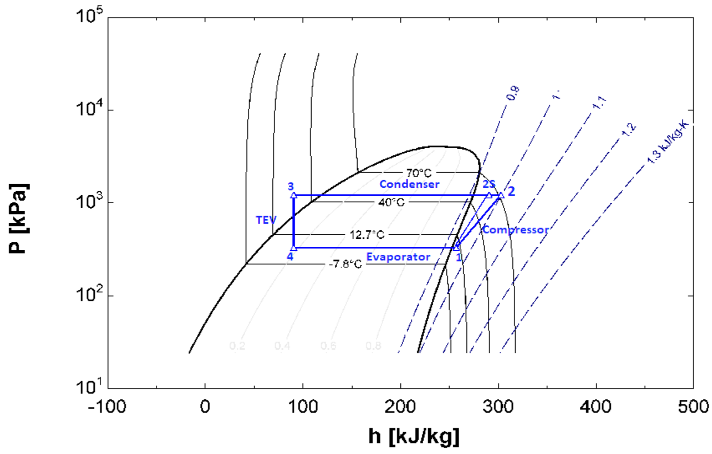


Figure 4. Vapour Compression Refrigerant Cycle.

To find the compression ratio for a compressor which is defined as the ratio of discharge pressure to suction pressure at saturated conditions, expressed in Pa or kPa as in Eq. 4, **Dincer and Kanoglu, 2010.**

$$\mathrm{CR} = \frac{P_d}{P_s} \tag{3}$$

where CR is a compression ratio, P_d and P_s the pressure at discharge and suction line respectively.

4.2 Condenser

The condenser, as any heat exchanger does not require any work to operate. Therefore, Eq. 4 will be:

$$Q = \dot{m}_r \Delta h \tag{4}$$

Thus, to find the heat rejected by the condenser, Eq. 5 is applied.

$$Q_{\mathrm{cond.}} = \dot{m}_r \,(h_2 - h_3) \tag{5}$$

where $Q_{\mathrm{cond.}}$ represents the heat rejected in (kW) and (h_3) is the outlet enthalpy at the condenser in (kJ/kg). It is indicated by the curve (2-3) in **Fig. 4**.

4.3 Thermostatic Expansion Valve

The purpose of the expansion valve is to control the flow of refrigerant from the high pressure at condenser of the cycle to the low pressure at the evaporator by throttling process as shown the vertical line (3-4) in **Fig.4**. So, the energy equation shows that the enthalpy is constant across the expansion valve as Eq. 6.

$$h_3 = h_4 \tag{6}$$

where h_4 is the enthalpy at the evaporator inlet in (kJ/kg).

4.4 Evaporator

The purpose of the evaporator is to get low-pressure and low-temperature refrigerant from the expansion valve. Then analysis of the evaporator is almost the same as that of the condenser. The refrigeration effect or the heat absorption process inducted via the straight line (4-1) in **Fig. 4**. So, Eq. 7 will be:

$$Q_{\text{evap.}} = \dot{m}_r\ (h_1 - h_4) \tag{7}$$

where $Q_{\text{evap.}}$ represents the heat absorbed in (kW) and (h_4) is the outlet enthalpy at the evaporator in (kJ/kg). To find RE, which is the refrigerant effect in (kJ/kg), Eq. 8n was used.

$$\text{RE.} = h_1 - h_4 \tag{8}$$

4.5 Coefficient of Performance

The seasonal coefficient of performance COP considers the influence of changing outside temperatures on the performance of the cycle. Therefore, for a complete cycle COP which signifies the ratio of the absorbed heat to the work of the compressor is given by Eq. 9:

$$\text{COP}_{\text{actual}} = \frac{Q_{evap.}}{W_{actual,comp.}} = \frac{\text{RE}}{w_{actual,comp.}} = \frac{h_1 - h_4}{h_2 - h_1} \tag{9}$$

where $W_{\text{actual,comp}}$. is representing the work per unit mass of refrigerant in (kJ/kg) and RE represents the refrigerant effect in (kJ/kg).

To compare the performance for the actual cycle with the ideal cycle, the Carnot cycle was used and the COP of it was calculated as shown by Eq. 10, which is considered a first way for comparing:

$$\text{COP}_{\text{Carnot}} = \frac{T_{evap.}}{T_{cond} - T_{evap.}} \tag{10}$$

where $T_{\text{cond.}}$, and $T_{\text{evap.}}$ are the saturated temperatures at condensing pressure and evaporating pressure respectively in K.

5. RESULTS AND DISCUSSION

Fig. (6) indicates the effect of ambient temperature on the COP. This phenomenon is clarified by analysing the effect of ambient temperature on the condensing temperature and pressure. As the ambient temperature increased, the saturation pressure in the condenser also increased. Consequently, the discharge pressure in the compressor increased. As a result, the compressor needed more power, resulting in a decrease in COP. Results showed that the COP of R134a becomes 24% higher than of R1234yf.

Fig. (7) shows the variation of compression ratio with different values of ambient temperature. It’s clear that when the ambient temperature increased, the compressor work increased, it means that the discharge pressure in the compressor increased which leads to an increase in the compression ratio. So, it’s found that compression ratio of R1234yf becomes 3.3% lower than of R134a.

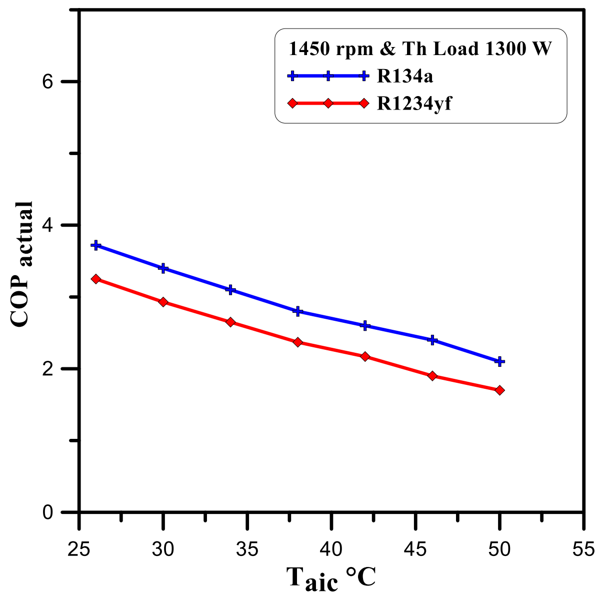


Figure 6. The effect of ambient temperature on the COP.

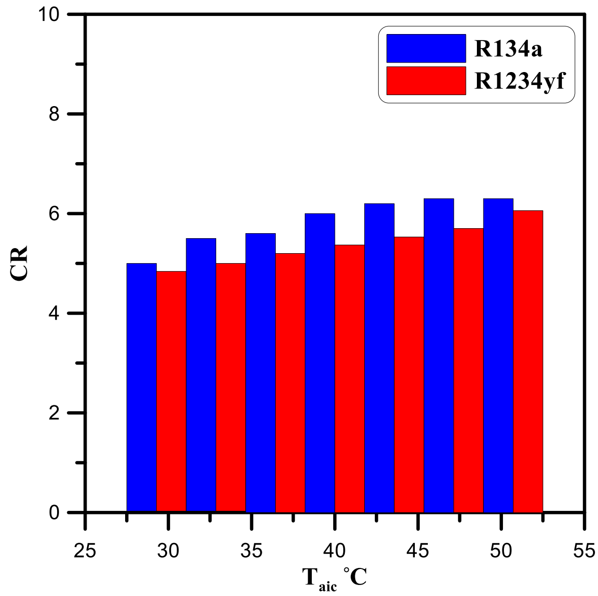


Figure 7. The effect of ambient temperature on the compression ratio.

Fig. (8) represents the effect of condenser inlet air temperature on compressor work at constant compressor speed and thermal load inside passenger room (1450 rpm and Th load 1300W). Results showed that when T_{aic} increased, the discharge pressure increased resulting in an increase in the compressor work. Therefore, When T_{aic} is 26 °C, the work of R1234yf becomes 23.4% lower than of R134a.

While **Fig. (9)** shows the effect of ambient temperature on heat rejection for both refrigerants. The rate of heat rejected from condenser decreased when increasing the ambient temperature because the compressor discharge temperature is increased. So, when T_{aic} is 26 °C, it's found that the heat rejection of R134a becomes 32% higher than of R1234yf, while when T_{aic} is 50 °C the heat rejection for R134a becomes 26% higher than R1234yf.

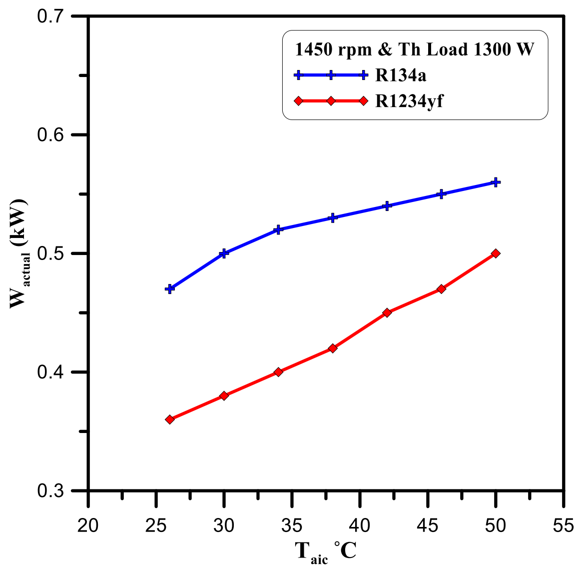


Figure 8. The effect of ambient temperature on the Work.

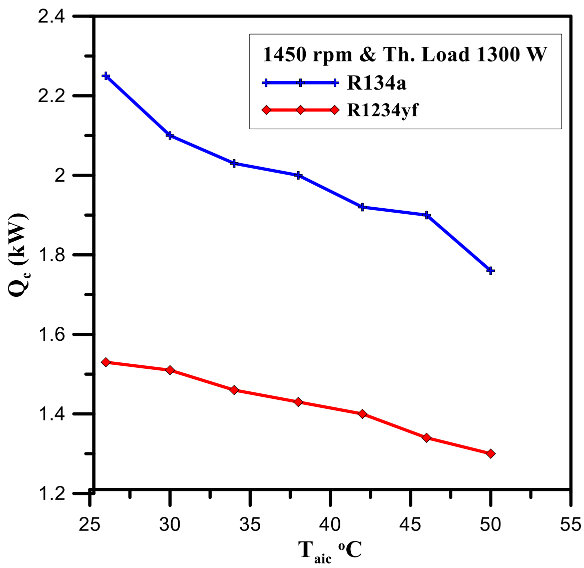


Figure 9. The effect of ambient temperature on heat rejection.

Fig. (10) shows refrigeration effect (RE) resolved by Eq. 9 which is an important parameter in refrigeration and AACS. It is obvious that RE reduced for both refrigerants with increasing T_{aic}. Moreover, R-134a is larger by about 25% than R1234yf.

Fig. (11) shows the variation of discharge pressure with different ambient temperature for both refrigerants. It is obvious that higher ambient temperature leads to higher discharge pressure. Therefore, the discharge pressure increased by 12% and 4% with increase T_{aic} =26 °C and T_{aic}=50 °C respectively.

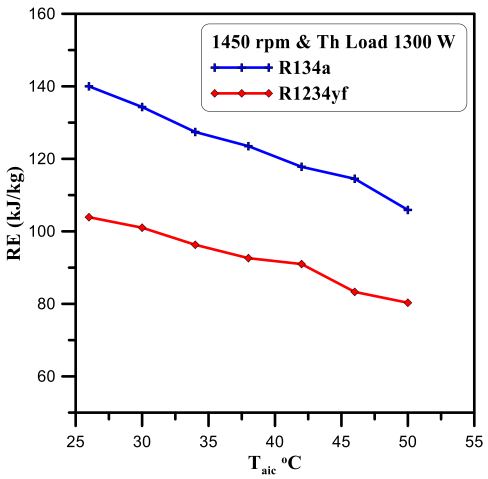


Figure 10. The effect of ambient temperature on the RE.

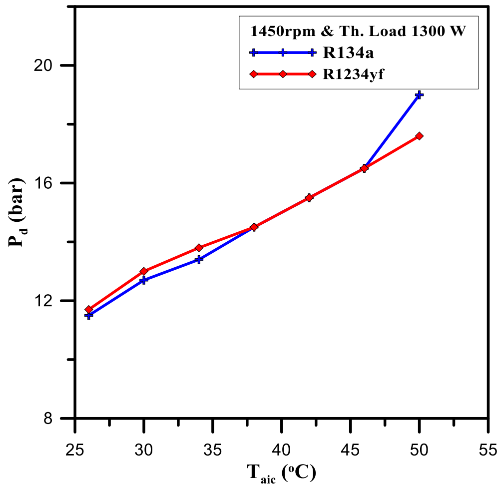


Figure 11. The effect of ambient temperature on the discharge pressure.

Fig. 12 a and b shows a P-h diagram of the R134a and R1234yf. It seems that at 50°C ambient temperature and 1450 rpm compressor speed, the discharge pressures are 19 bar and 17.6 bar for the R134a and the R1234yf cycle respectively. The R134a cycle has about 7.3 % higher discharge pressure than the R1234yf cycle. So, it will have better performance than the R134a cycle.

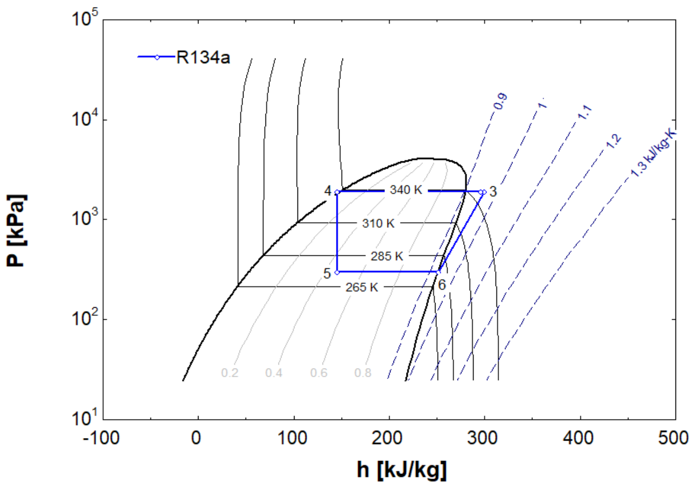


Figure 12-a. P-h diagram at 1450 rpm, 1300 W Th. Load, 50 °C for R134a.

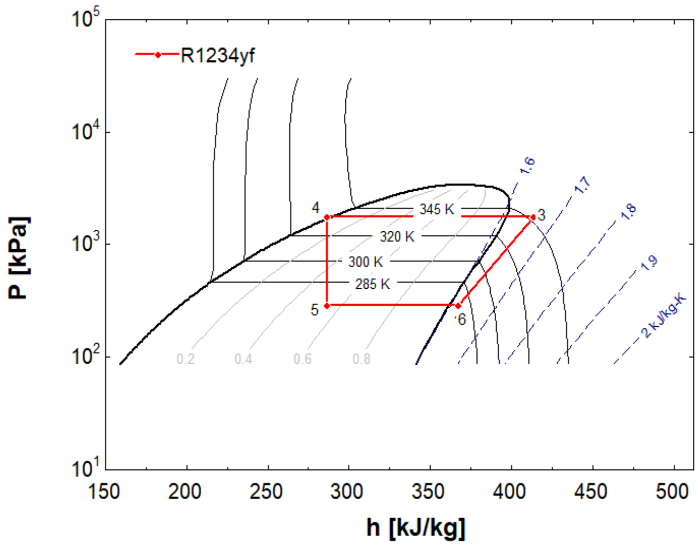


Figure 12-b. P-h diagram at 1450 rpm, 1300 W Th. Load, 50 °C for R1234yf.

Figure 12. P-h diagram of the R134a and R1234yf.

6. CONCLUSIONS

The results of this study showed that the COP values are close to each other for R1234yf and R134a refrigerants. Consequently, R1234yf, which yields a reduced COP, can be used for refrigerating systems operating with R134a without any modification. The condenser heat rejects with R134a greater than R1234yf by about 28.1%. Regarding the refrigerant effect, it was found that it decreased with increasing the ambient temperature. It was found that R134a was greater than R1234yf by 24.3 %. But it can be recommended that choosing a compressor with greater capacity should compensate the decrease of COP. It is important also to note that pressure discharge values for both refrigerants were nearly the same. This points out that R1234yf is a suitable refrigerant as an alternative for R134a and it can be securely used in the systems operating with R134a.

7. REFRENCE

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NOMENCLATURE

AACS = automoative air conditioning system.
COP = coefficent of performance, dimensionless.
CR = compression ratio.
EES = engineering equation solver.
HFO1234yf= R1234yf.
GWP = global warming potential.
IHX = internal heat exchanger.
ODP = ozone depletion potential.
TXV = thermal expansion valve.

VCRC = vapor compression refrigerant cycle.
RE = refriherant effect kJ/kg.
$\dot{m}_r$= mass flow rate of refrigerant, kg/s.
h = enthalpy, kJ/kg.
P = pressure, bar.
Qc = Heat reject from condenser kW.
Q_{evap}. = heat absorbed in the evaporator, kW.
rpm = revolution per min.
Th. Load= thermal load, kW.
W_{actual} = actual work, kW.
W_{ideal} = ideal work, kW.
aic = air inlet condenser.
comp.= compressor.
cond.=condenser.
d = discharge line.
evap.=evaporator.
s = suction line.
T_{evap}.= Evaporatore temperature, (^oC).
T_{cond}.= Condenser temperature, (^oC).