Comparative Experimental Study of Low GWP Alternative for R134a in a Walk-in Cold Room

دراسة مقارنة عملية لأداء فريون ذو تأثير احتباسى حرارى منخفض بديل لفريون R134a لغرفة تبريد

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الملخص العربي:- اصبحت المشاكل البيئية الناجمة من الاحتباس الحرارى العالمى إهتماما عالميا لدى الباحثين ، ولما وُجد أن الفريونات المستخدمة حاليا ذات قيمة احتباس حرارى مرتفعة أصبح من الضرورى البحث عن بدائل لهذه الفريونات تعمل على نفس الانظمه وتكون ذات تأثير احتباسى حرارى منخفض. وعلى ذلك تم انشاء غرفة تبريد تعمل بنظام انضغاط البخار وذلك لعمل مقارنه فى الاداء بين R1234ze و R134a وذلك فى إطار البحث عن أحد الحلول لمشكلة ارتفاع قيمة الاحتباس الحرارى للفريونات المستخدمة لغرف التبريد. وبعد ان تم الاختبار تبين أن السعة التبريدية لفريون R1234ze أقل بنسبة تصل من 2 % الى 13 % من R134a . وأقل درجة تبخير يمكن التوصل إليها هى 13- مْ على عكس الفريون R134a أقل درج حرارة تبخير يمكن التوصل اليها ــ 30 مْ ومعدل استهلاك الطاقة للنظام باستخدام الفريون R1234ze أقل منه باستخدام الفريون R134a بنسبة 9% الى 15 % . وعلى ذلك فان الفريون الجديد يحبذ استخدامه فى درجات حرارة تبخير العالية والمتوسطة بعد اجراء بعد التعديلات على نظام التبريد.

Abstract: - The environmental problems induced by the ongoing increase in the global worming potential (GWP) pose a significant interest among researchers. It was found that, the currently used refrigerants are with high GWP [1], so that it becomes necessary to search for alternatives to these refrigerants that can properly operate on the same systems but with low GWP. Therefore a walk-in cold room working with vapor compression cycle is constructed to compare the performance of the high GWP refrigerant R134a to low GWP refrigerant R1234ze in a trail to provide a solution to the problem of high GWP of refrigerants currently used in cold rooms. The results obtained in this study have shown that, the cooling capacity of R1234ze was lower than that of R134a by 2% to 13%. The lowest evaporating temperature that could be reached for R1234ze is -13 ^{o}C while the lowest temperature of R134a is – 30 ^{o}C. Regarding the power consumption, R1234ze has lower power consumption than R134a by about 9% to 15% therefore it can be concluded that R 1234ze can be recommended to be used at high and medium evaporating temperature after carrying out the modification on the refrigeration cycle

1. INTRODUCTION

he continuous growth in GWP all over the world constitutes a threatening environmental phenomenon that should be studied to find scientifically based solutions. Refrigerants used in the last few decades such as CFCs and HCFCs including R11, R12, R22 and R502 have depleted the ozone layer and consequently lead to the greenhouse effect on the climate. Therefore, the use of low GWP and zero ozone depletion potential (ODP) refrigerants becomes very essential to prevent further depletion of the ozone layer and consequently guard against the greenhouse effect on climate. In the year 1996 the production of R11 and R12 were completely banned, furthermore the production of HCFCs will be completely banned after 2030. Although the ODP of R134a and R404a are zero however, they have a high value of GWP. The production of R134a may thus be completely

forbidden in the near future as it was restrained in the Tokyo Protocols signed in 1997 and Montreal Protocol [2, 3].

As a result, efforts are made in order to find out alternative refrigerants to replace the currently used high GWP refrigerants withR1234yf and R1234ze (E). R1234ze (E) will be referred simply as R1234ze. They have been proposed as alternative refrigerants for R134a, which has a GWP of 1430. Both of these refrigerants have zero ODP and a GWP value for R1234ze is 6 and for R1234yf is 4, with mild flammability and low toxicity [4-8].

In a simulation model for the steady-state analysis of domestic refrigerator-freezer validated by experimental data, it was found that, for R1234ze the evaporator capacity decreased about 21.5 % and the COP increased about 7.9% [9]. A comparative study on a system using an expander as a configuration on an internal heat exchanger on system using R1234yf, R1234ze and R134a as a working fluid, an increment of 9% to 15% for R1234yf and 11% to 20 % for R1234ze on COP with respect to basic cycle was recorded. However, the main disadvantage of these refrigerants on the system was the increase in complexity of the system and increase of cost [10]. A comparative performance analysis of low GWP refrigerants R1234yf, R1234zeand R600a inside a commercial roll-bond evaporator for household refrigerators recorded that, vaporization performances for R1234ze, R1234yf and HC600a were similar to that of R134a at evaporation temperatures, -15°C and -20 $^{\circ}$C. [11]. Internal heat exchanger (IHX) affects positively the performance of vapor compression system. By using R1234ze and R450A (R134a/R1234ze commercial mixture) as replacements for R134a at evaporation temperatures 260 K, 270 K and 280 K and condensation temperatures 300 K, 310 K and 330K, the cooling capacity increases with high compression ratios particularly for R1234ze. R134a has greater refrigeration effect and refrigerant mass flow rate. COP for R1234ze and R450A increases using IHX. [12]

As drop-in replacement in a screw chiller with 200 TR using R134a with R1234ze and D4Y a test was carried out by [13], recording low cooling capacity for R1234ze about 22.6 % with about 5.5% increase in COP. D4Y has shown nearly the same capacity as R134a and decrease in COP about 2.6%.   To evaluate energy consumption, an experiment was conducted to assess and compare the performance of R134a to that of R1234yf and R1234ze, using AHAM standard HRF-1 [14], these

refrigerants were tested as drop-in replacements, with no performance enhancing modifications to the refrigerators. It was found that, R1234yf has higher energy consumption than R-134a and R1234ze had lower energy consumption than R-134a. The results presented in [15] recommended that, in order to replace R-134a with R-1234ze in domestic refrigerators, the lower capacity would need to be addressed.

In the present study, experimental performance comparison between refrigerants R1234ze and R134a are carried out. A walk-in cold room was constructed that uses a vapor compression cycle for the two refrigerants as a working fluid, in order to study and evaluate the performance of a low GWP alternative refrigerant for R134a. The results are compared with published work of other investigators.

2. EXPERIMENTAL SETUP

2.1 construction of test rig

A walk-in cold room is designed and has been constructed to evaluate the performance and power consumption using a vapor compression cycle with R134a and R1234ze as refrigerants. Figure 1 shows a view of the main component of the vapor compression cycle which consists of the following: 1.5 hp reciprocating semi-ceiled compressor (K 150), lubricated by ester oil; air cold condenser with fan power 150 W; thermostatic expansion valve (TEN2 orifice no 02) (TXV), fan coil evaporator with fan 150 W. The evaporator is installed in a refrigerated room (2.4 m length $\times$ 1.7 m width $\times$ 2.5 height). The room is insulated by prefabricated polyurethane foam panel with thickness of 10 cm and density ρ = 40 kg/m^3. Inner and outer surface are covered with steel sheets with thickness of 0.5 mm.

2.2 Experimental test procedure

Schematic diagram of the vapor compression cycle including the measuring points of pressure and temperature is shown in Fig.2. In the vapor compression cycle there are seven points for measuring pressure and temperature (their locations can be seen in Fig. 2).

Compressor power consumption, ambient temperature and cold room air temperatures are also measured. Table 1 summarizes the instrumentation characteristics. All of these data is gathered using a data acquisition system. Then all of data is monitored and stored through a Personal Computer
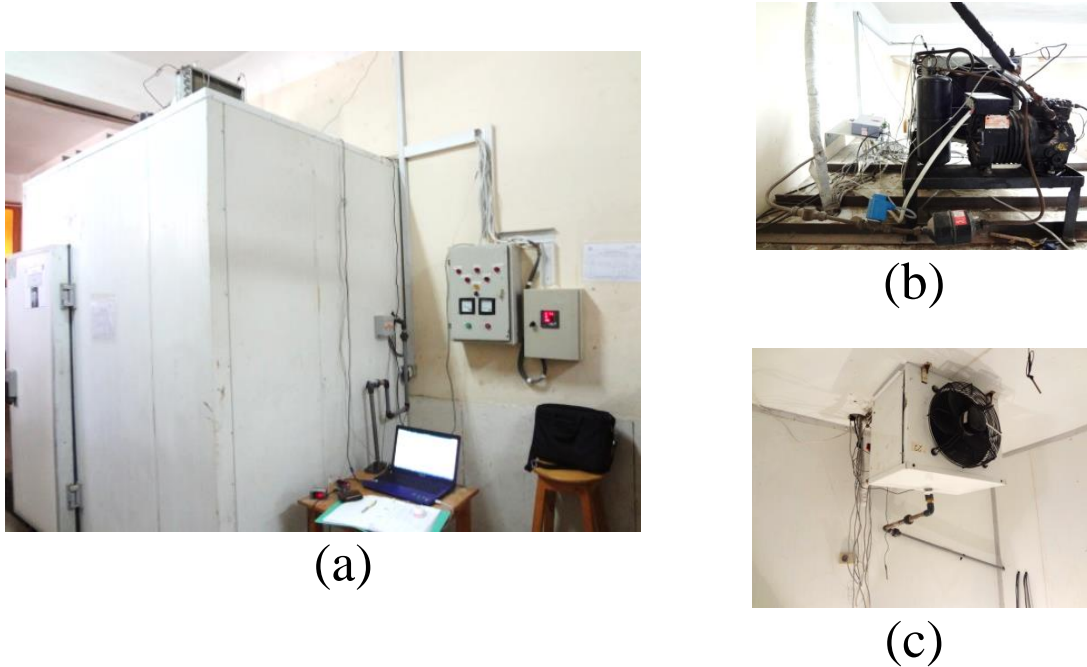
Figure 1 Photo of test rig: (a) Wall panel insulation and control panel (b) Condensing unit including all auxiliary components (c) Fan coil evaporator

Table 1: Measured parameters and equipment uncertainty

Measured parameters	Sensor	Accuracy
Air temperatures	Thermocouples type DS18B20	± 0.5 C
Refrigerant temperature	Thermocouples type DX NTC NS 6	± 0 .1 ºC
Refrigerant pressure	Pressure transducer eliwell EWPA	± 7 kPa
Power consumption	Power meter PM 1200 Shneider	± 1 %
Air Velocity	Hot wire TSI 8340 – M - GB	± 0.1 m/s

2.3 Data reduction

In order to make complete assessment of COP for vapor compression cycle, 8 experiments are carried out (4 with each refrigerant: R1234ze and R134a), at different product loads beginning with no load and using product load simulated with steady state electric heaters with capacities 200, 400 and 600 W. In order to calculate the refrigerant mass flow rate through the cycle, heat balance on condenser is done. Air mass flow rate through the condenser is calculated according to ASHRAE procedure. [16]

$$m_{air}^{\cdot} = v_{air} \times \rho_{air} \times A_{sec.} \qquad (1)$$

A hot wire anemometer is used to measure air velocity

$$m_{ref} = (m_{air} cp_{air} \Delta T_{air}) / \Delta h_{refc} \qquad (2)$$

where ΔT_{air} is the air temperature difference through the condenser.

The cooling capacity Q_c consist of the product load plus all losses from the room (transmission load, equipment load, infiltration load, cycle losses, etc.). To calculate cooling capacity of cycle, pressure and temperature are measured at inlet of TXV and outlet of evaporator.

Total cooling capacity is calculated from Eqn. 3as,

$$Q_{c.} = m_{ref.} \times \Delta h_e \qquad (3)$$

The superheating degree at evaporator outlet ranges at [7-10] ºC and subcooling degree at condenser outlet is measured as [2-6] ºC.

To calculate Carnot COP of the cycle, evaporator and ambient temperatures are measured.

$$COP_c = \frac{T_e}{T_c - T_e} \qquad (4)$$

To calculate the actual COP of cycle, cooling capacity is calculated from Eq. (3) and power consumption is measured and COP is evaluated as;

$$COP = \frac{Q_{c.c.}}{P_c} \qquad (5)$$

To calculate the actual refrigeration efficiency, η_R, [17] of the cycle, COP and COP_c are calculated and η_R is evaluated as,
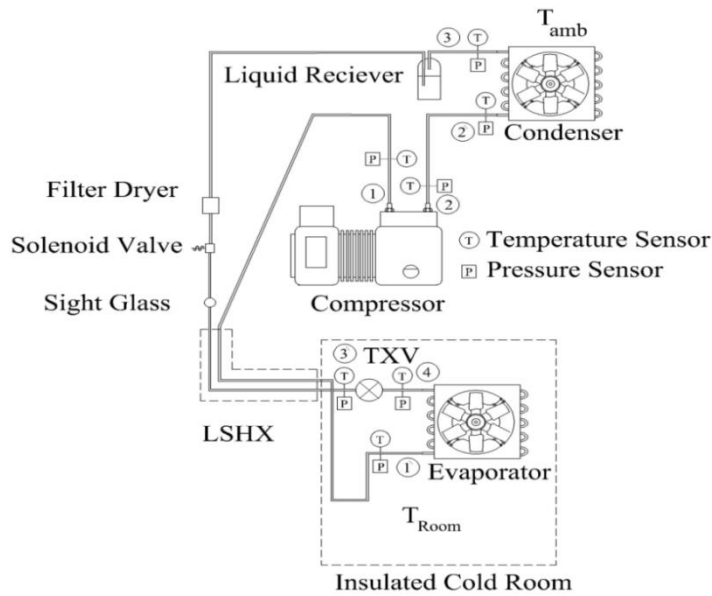
$$\eta_R = \frac{COP}{COP_c} \qquad (6)$$

Figure 2 Schematic diagram of the test rig

3. Results and discussions

This section presents and discusses the experimental results obtained using R134a and R1234ze as working fluids. The parameters measured are inlet and outlet refrigerant pressures and temperatures of compressor, cold air temperatures & outlet condenser refrigerant temperature, air velocity through condenser, power consumption and evaporator temperature. The parameters analyzed are COP, COP_C, η_R and cooling capacity to make complete assessment in performance comparison between the two refrigerants.

Discharge pressure for R134a is almost higher than R1234ze and suction pressure for R1234ze is also lower than R134a as shown in Fig. 3. For R134a the compression ratio is higher than that for R1234ze.

Discharge pressure for R1234ze are about 25-40 % lower than that of R134a at same T_e. Also suction pressure for R1234ze is ranged between 10-49 % less than that of R134a at same T_e.

The average air temperature for R134a records faster drop than the same case of R1234ze; as shown in Fig. 4. That difference in faster dropping of air temperature is due to the fact that both refrigerants have different thermophysical prosperities. At load 600 W R1234ze takes 44 min to achieve $T_{r\ avg.} = 0\ ^oC$ and 79 min $T_{r\ avg.} = -5\ ^oC$ in, while R134a achieves $T_{r\ avg.} = 0\ ^oC$ within 35 min and $T_{r\ avg.} = -5\ ^oC$ in 76 min. Temperatures of refrigerant at point 3 and 3' for R1234ze are almost lower than that for R134a as shown in Fig. 4.
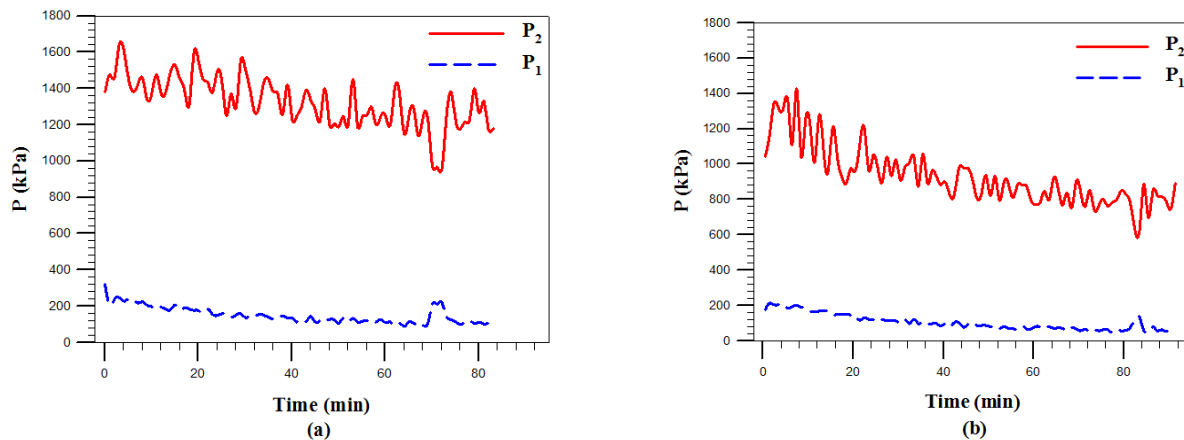

Figure 3 Compressor inlet and outlet pressures vs. time at 600 W load, (a) for R134a, (b) for R1234ze
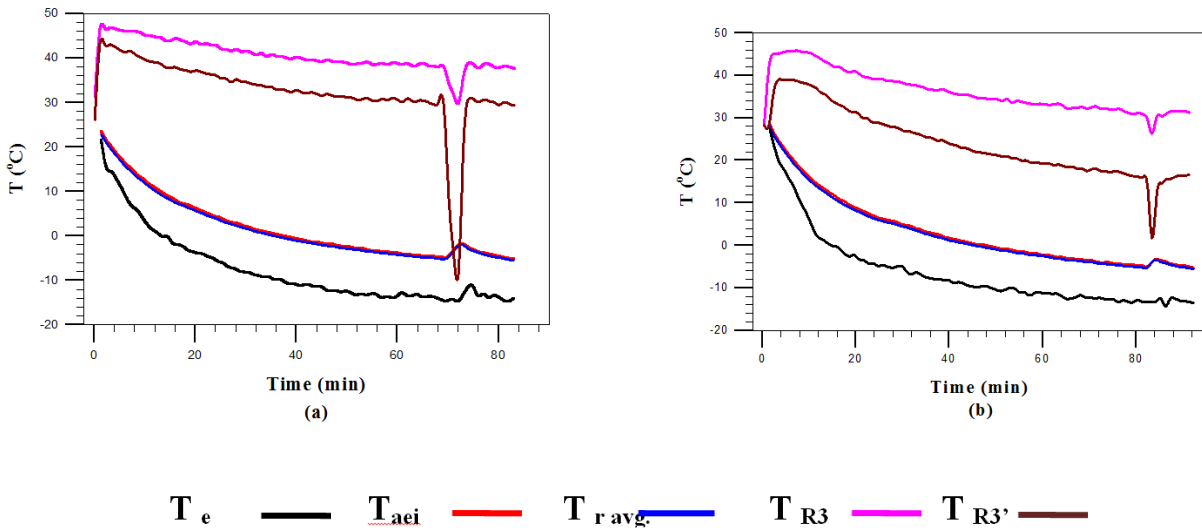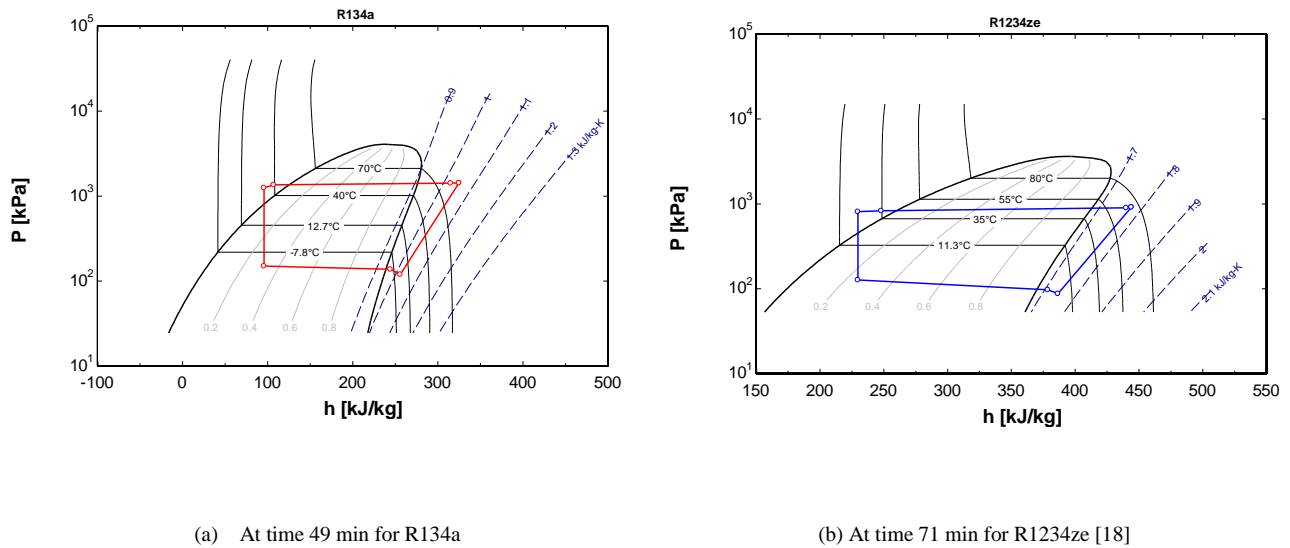
Figure 4 T_e , T_{aei} , $T_{r\,avg.}$, T_{R3} and $T_{R3'}$ vs. time at 600 W, (a) for R134a, (b) for R1234ze

(a)   At time 49 min for R134a   (b) At time 71 min for R1234ze [18]

Figure 5 P-h diagram of the two cycles at same T_e = -12.6 °C

Figure 5 (a) and (b) shows the P-h charts for both R134a and R1234ze, respectively at evaporator temperature -12.6 °C. From Fig. 5 (a), it is shown that for R134a there is pressure drop in condenser of 64 kPa and in evaporator of 12.7 kPa. Also there is increase in entropy in compression process of 0.031 kJ/kg.K. From Fig. 5 (b), it is shown that for R1234ze there is pressure drop in condenser of 72.1 kPa and in evaporator of 29.47 kPa. Also there is increase in entropy in compression process of 0.028 kJ/kg.k.

Cooling capacity of R134a is almost higher than that of R1234ze. As it is significant clear the $Q_{c.c}$ decreases by decreasing T_e. For the case of no load at the same evaporator temperature about 20 °C it is recorded that $Q_{c.c}$ is 4.54 kW for R1234ze and 4.56 kW for R134a the difference in values is about 0.4% for R1234ze less than that of R134a.
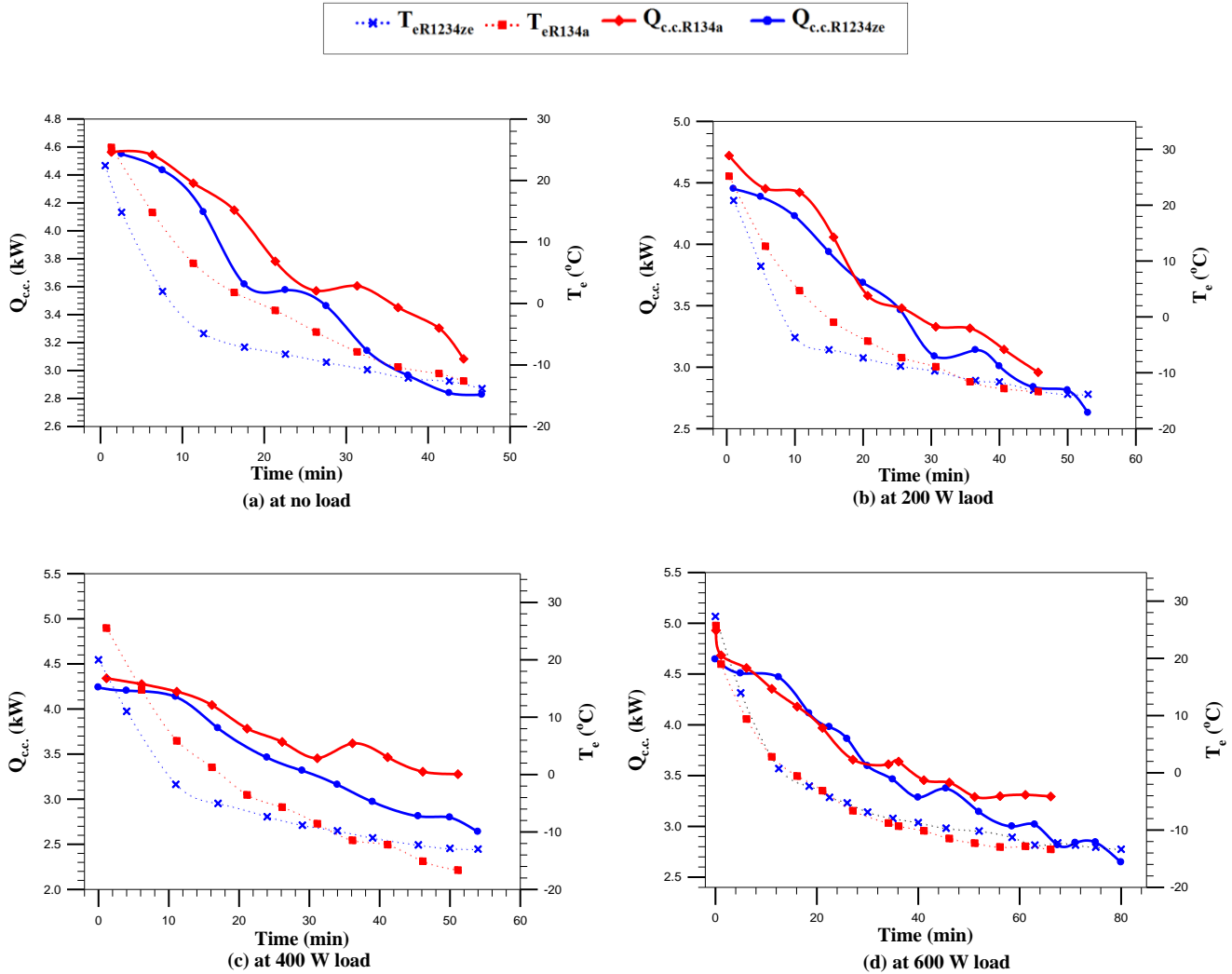
Figure 6 Q_c And T_e vs. time at different cases, (a), (b), (c) and (d)

For R134a it achieves Te = -13 °C faster than R1234ze with about 5 min as shown in Fig. 6 (a).For the case of 200 W load at the same T_e = 20 °C it is recorded that the $Q_{c.c}$ is 4.54 kW for R1234ze and 4.75 kW for R134a, it decreases by 4.4% than that for R134a. For R134a it archives T_e = -13 °C faster than R1234ze about 8 min as shown in Fig. 6 (b).

For the case of 400 W load at the same T_e = 20 °C the $Q_{c.c}$ is 4.25 kW for R1234ze and 4.35 kW for R134a with decrease of 2.3% as shown in Fig. 6 (c).

For the case of 600 W load at the same T_e =21 °C the $Q_{c.c}$ is 4.64 kW for R1234ze and 4.93 kW for R134a it decreases by 5.9 % as shown in Fig. 6 (d). $Q_{c.c}$ consists of the product load plus all loses from the room about 2.8 kW. At the end of all testes recording that the difference in $Q_{c.c}$ values for R1234ze about 2 – 13 % less than R134a.

COP of R1234ze is almost higher than that for R134a and the deference between them decreases by decreasing T_e for all cases as shown in Fig. 7. For the case of no load at the same T_e = 14.8 °C it is recorded

that the deviation about 28.5% and decreases to 12.23% at the same T_e = -7.1°C as shown in Fig. 7 (a). For the case of 200 W load, at the same T_e = -3.67 °C the deviation is significant high about 29.2% and decreases to 6.87% at T_e = -13.1°C as shown in Fig. 7 (b).For the case of 400 W load as shown in Fig. 7 (c), at the same T_e = 11.3 °C the deviation is about 31.9 % and decreases to 4.7% at T_e = -12 °C. For the case of 600 W load as shown in Fig. 7 (d), at the same T_e = 13.9 °C the deviation reaches about 33 % and decreases to 12.1% at T_e = -11.2 °C .

Total power consumption for the system using R1234ze is lower than that for R134a. For the case of no load at starting of system P_c for R1234ze is significant recorded about 12.25 % lower than that of R134a at the same T_e = 16 °C, then this difference decreased to be 9.36 % at the same T_e= -7.1 °C and then increased to be 14 % at the same T_e = -13 °C as shown in Fig. 8 (a).
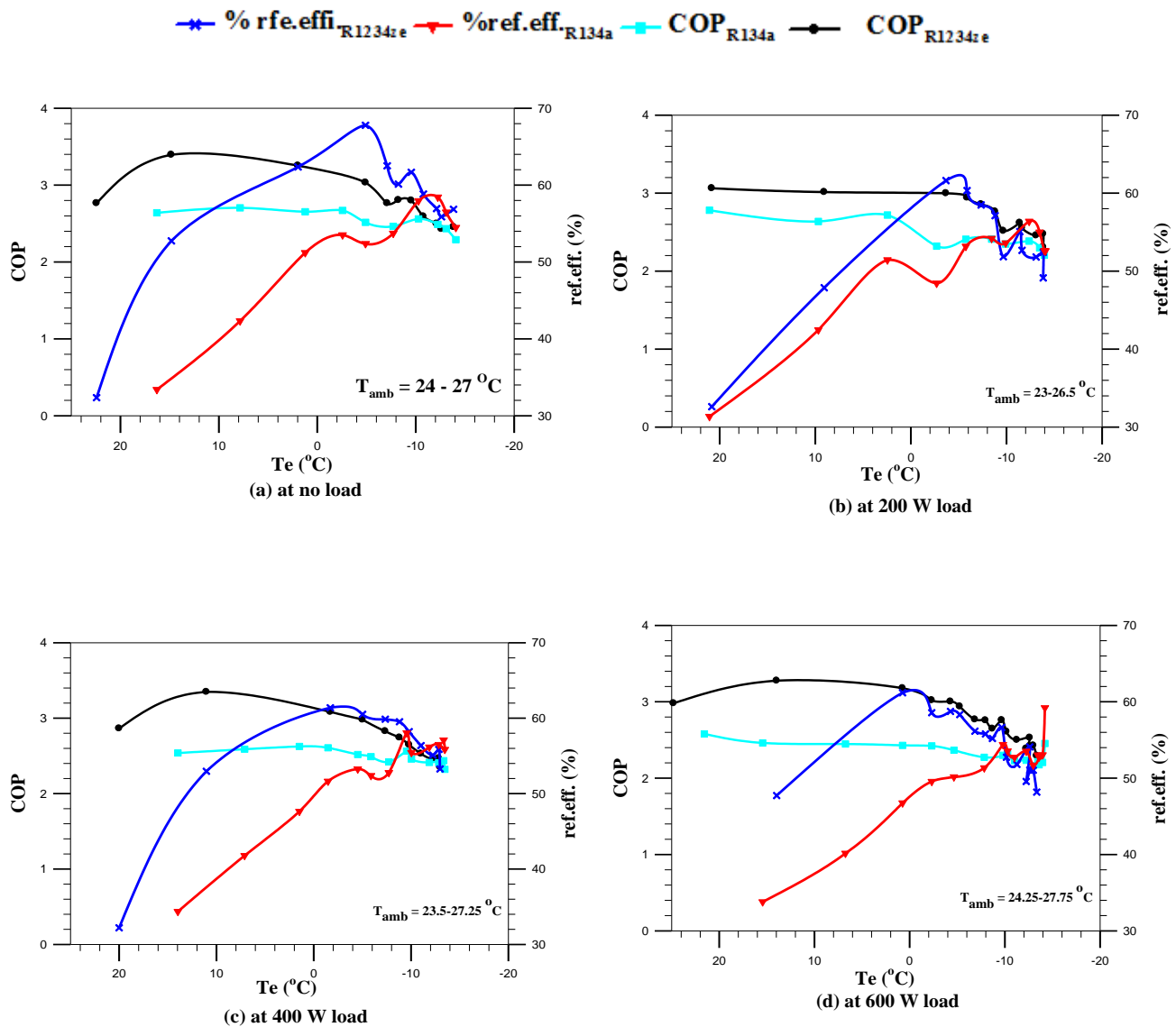
Figure 7 COP and refrigeration efficiency vs. T_e at different cases, (a), (b), (c) and (d)

For the case of 200 W load P_c for R1234ze at the same $T_e = 21$ ^{o}C is lower than R134a about 14.4 %, and at the same $T_e = -13$ ^{o}C the difference in P_c for R1234ze goes to be 15.5 % lower than that for R134a as shown in Fig. 8 (b).

For the case of 400 W load P_c for R1234ze at the same $T_e = 11$ ^{o}C is significantly lower than that of R134a

about 15 % and it goes up to about 17.4 % at the same $T_e = -13$ ^{o}C as shown in Fig. 8 (c). For the case of 600 W load P_c for R1234ze at the same $T_e = 14$ ^{o}C is recording maximum difference on values for R1234ze about 21.3 % than R134a, then it increased at the same $T_e = -13$ ^{o}C to be about 23.8 % as shown in Fig. 8 (d).
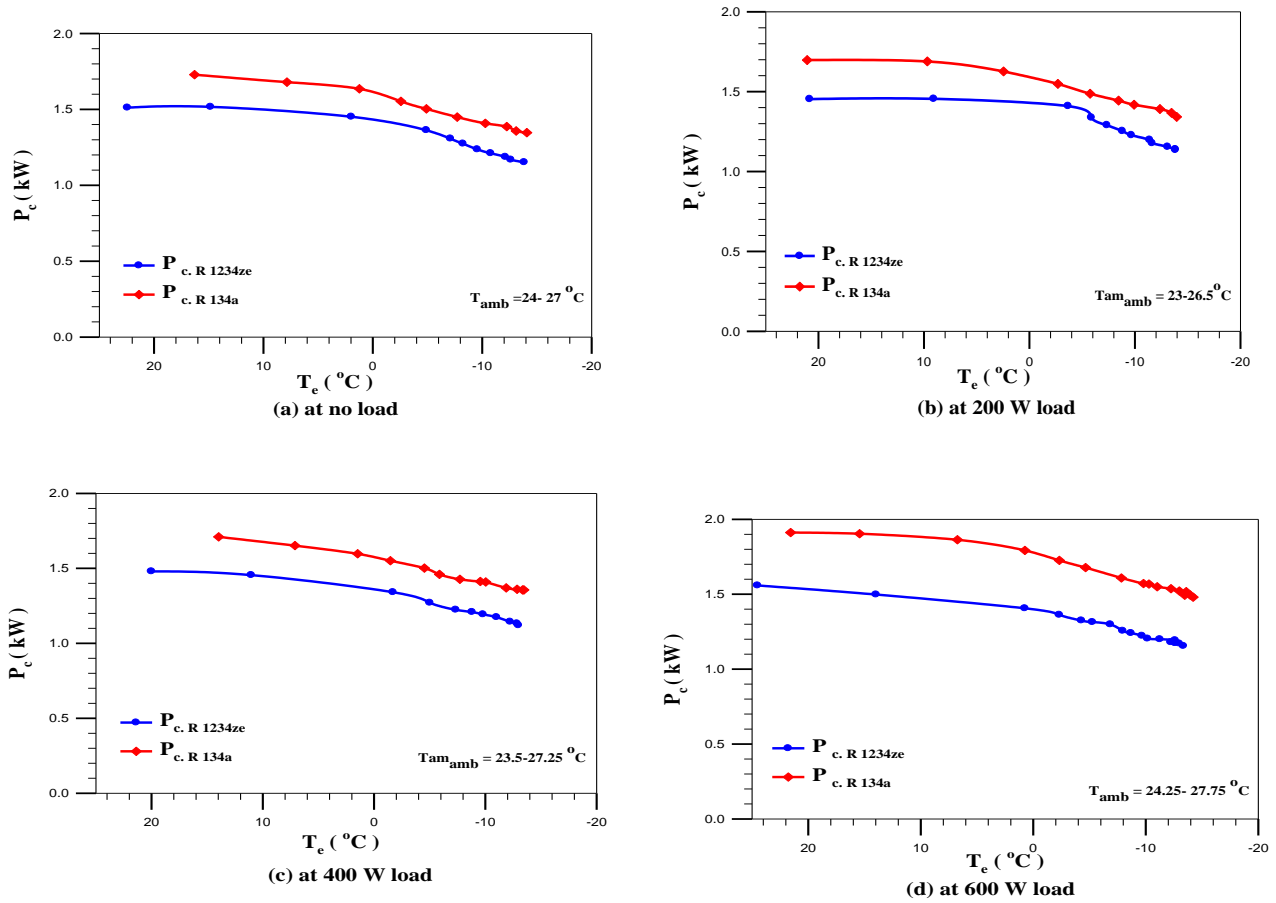
Figure 8 P vs. T_e at different cases (a), (b), (c) and (d).

4. COMPARISON WITH OTHER EXPERIMENTAL WORK

Table 2 shows a comparison between main variables of vapor compression cycle on our work with the work of Leighton D. et al. [9] as in the following tables:

Table 2 comparison between present work and Leighton et al. [8].

Variable	Leighton et al [9]	Present work R1234ze	Present work R134a
Compressor cooling capacity [kW]	0.0962	[5 : 2.6]	[5 : 2.6]
Compressor discharge pressure [kPa]	1095.4	1520 : 933	1700 : 950
Evaporator air inlet temperature [^{o}C]	-10.2	22 : -5	22 : -6
Condenser air inlet temperature [^{o}C]	32.8	30 : 24	33 : 26
Condenser volumetric air flow rate [m^3/s]	0.0283	0.67521	0.67521

Table 3 Comparison between present work at Te = -13°C and air inlet temperature to evaporator-5 ^{o}C for R134aand R1234ze, internal load 600 W

variables	R134a Present work	R1234ze Present work	R134a Leighton et al [9]	R1234ze Leighton et al [9]
COP	2.175	2.2919	1.604	1.731
Compressor power [kW]	1.5168	1.1927	0.1213	0.0882
Evaporator Capacity [kW]	3.2936	2.6464	0.1945	0.1527
Mass Flow Rate [kg/s]	0.02083	0.01855	0.001251	0.00128
Discharge Pressure [kPa]	1076.22	901.77	1196	851
Suction Pressure [kPa]	310.65	163.07	132.6	102

5. CONCLUSIONS

In this research, a commercial vapor compression system is used to refrigerate a walk-in cold room. Performance of the system using R1234ze is studied and compared with the same system using R134a. It is significantly recorded that R1234ze has a COP about 7 -

33 % higher than that of R134a in all cases. It has been shown that R1234ze requires less power than R134a with about 9 -15 %. However R1234ze has lower Q_c than R134a of about 2 - 13 % which can be considered comparable with R134a**.**

NOMENCLATURE

$\Delta h_{ref.e}$	Refrigerant enthalpy change in the evaporator kJ/kg
$\Delta h_{ref.c}$	Refrigerant enthalpy change in the condenser kJ/kg
ΔT_{air}	Air temperature difference across the condenser, K
cp_{air}	Air specific heat, kJ/kg.K
η_R	Refrigeration efficiency
Q_c	Cooling capacity, kW
P_c	Power consumption, kW
T_{aei}	Evaporator air inlet temperature, ^oC
T_e	Evaporator temperature, ^oC
T_∞, T_{amb}	Ambient temperature, ^oC
m_R	Refrigerant mass flow rate, kg/s
p_2	Compressor outlet pressure, kPa
m_{air}	Air mass flow rate, kg/s
v_{air}	Air velocity, m/s
COP	Coefficient of performance
COP_c	Carnot coefficient of performance
h	Refrigerant enthalpy, kJ/kg
ρ_{air}	Air density, kg/m^3
$A_{sec.}$	Air cross section of condenser, m^2
$T_{r\ avg.}$	Room average temperature, ^oC
T_{R3}	Refrigerant temp. at condenser outlet, ^oC
$T_{R3'}$	Refrigerant temperature at expansion inlet, ^oC
p_1	Compressor inlet pressure, kPa

ABBREVIATIONS

GWP	Global worming potential
ODP	Ozone depletion potential
TXV	Thermostatic expansion valve
AHAM	Association of home appliance manufacturers
HRF-1	Household Refrigerators, refrigerator – Freezers and freezers

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