Performance of Alternative Refrigerant R431A on Air-Conditioning System under Real Transient Conditions

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ABSTRACT

This paper presents the performances of new ozone-friendly refrigerant (R431a) in air-conditioning system were investigated experimentally and compared with R22 under transient conditions.R431a has no ozone depletion potential and very low greenhouse warming potential of less than 43 (ASHRAE Listed,2007)[1].The variation in refrigeration rate of evaporator, work consumption of compressor and heat rejection rate of condenser with time from the start to stop of the air condition system have been examined. Test results showed that the refrigeration rate of evaporator and heat rejection rate of condenser drops almost parallel to each other, both with a decreasing rate with time. The power consumption on the other hand does not increasing much. The coefficient of performance of R431a decreased from rang 8.96 to 9.28% through (5000 sec) at different ambient temperatures. The refrigeration rate of evaporator decreased by 2.69% through 5000 sec at ambient temperature 30 C° for R431a .The heat rejection rate of condenser decreased by 2.34% through 5000 sec at ambient temperature 30 C° for R431a. Results showed that the coefficient of performance of R431a is 14.67 to 20.34% higher than that of R22. Compressor input power of R431a is 8.39 to 14.19% lower than that of R22 at different ambient temperatures.

Keywords: Transient conditions, Alternative, Ozone-friendly, Coefficient of Performance, Refrigeration System

أداء المبرّدِ البديلِ آر ٤٣١ أي على منظومة تكييف الهواء تحت ظروف استجابة عابرة حقيقية

الخلاصة

هذا البحث يقدم دراسة عملية لاداء منظومة تكييف الهواء باستخدام موائع تبريد جديدة R431a تحت ظروف استجابة عابرة حقيقية بالمقارنة مع الموائع القديمةR22 .وسيط تبريد R431a لايؤثر على

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طبقة الاوزون وكذالك GWPاقل من ٤٣.أظهرت النتائج العملية بان كمية الحرارة الممتصة من قبل المبخر وكمية الحرارة المطروحة من قبل المكثف تنخفض بشكل متوازي. لكن الطاقة المستهلكه من قبل الظاغط من جهة اخرى لا تزداد كثيراً. معامل الأداء للغاز (R431a) تقل بنسبة ٨,٩٦-٩,٢٨% خلال ٥٠٠٠ ثانية عند درجاتِ الحرارة البيئيةِ المختلفةِ . كمية الحرارة الممتصة من قبل المبخر تقل بنسبة ٢,٦٩% خلال ٥٠٠٠ ثانية عند درجة الحرارة البيئيةِ ٣٠ مئوية للغاز (R431a). كمية الحرارة المطروحة من قبل المكثف تقل بنسبة ٢,٣٤% خلال ٥٠٠٠ ثانية عند درجة الحرارة البيئيةِ ٣٠ مئوية للغاز (R431a). معامل الأداء للغاز (R431a) هو أعلى بنسبة ١٤,٦٧-٢٠,٣٤% بالمقارنه مع غاز(R22). وكذالك أقل أستهلاك للطاقه للغاز (R431a) بنسبة ٨,٣٩-١٤,١٩% بالمقارنه مع غاز(R22) عند درجاتِ الحرارة البيئيةِ المختلفةِ.

INTRODUCTION

The resulting environmental concerns about ozone depletion and global warming have stopped the production of CFC refrigerants. The production of HCFC will stop in 2015**.** Many researches were accomplished in this field. The following sections describe some of these researches which are concerned with the present study.

Kim et al.(1997)[2], presented results of performance test of R22 and four alternative fluids R134A ,R32/R134A(30/70) ,R407C and R410A.The study was performed in an experimental bread board water-to water heat pump .R407C showed the most similar performance characteristic to R22 .The cooling capacity of R407C and R32/R134A was a few percent higher than that of R22. It was only (2%).

Aprea and Renno (2003)[3] studied experimentally the performance of a vapor compression refrigeration plant using R-22 as working fluid and its substitute R-417A (R-125/R-134a/R-600). The compressor can work with the fluid R-22 and it was lubricated with polyester oil to work with the alternative refrigerants. The air temperature at the condenser has been kept at about (32°C) and (10°C) for summer and winter season respectively, while the air temperature in the cold store were maintained at (-5, 0, 5°C). The test results revealed, on average basis the coefficient of performance of R-417A was less than that of R-22 by (15 %).

Spatz and Motta (2004)[4] studied the performance of three alternative refrigerants of R-22, which were R-404A, R-410A and R-290 in medium temperature refrigeration applications, the ambient temperature was ranged between (7 to 35 °C). In this study all of the tested refrigerants exhibited reduction in the coefficient of performance as the ambient temperature increased. R-410A with the (original heat exchanger) was more efficient than R-22 over the most of operating range (less than 30 °C), the COP of R-410A increased up to (8 %) at low ambient temperature. The coefficient of performance of R-290 was slightly higher than that of R-22 by (1 to 4%) for the complete operating range. While the COP of the original (non modified) R-404A system was consistently lower than that of R-22 up to (10 %) at high ambient temperatures. On other hand, the COP of both of the modified system (by increase heat transfer area) R-404A and R-410A were significantly higher than R-22 for most outdoor temperatures.

Ki-Jung Park, Cheol-Hee Lee, Dongsoo Jung(2008)[5] presented an experimental study on the thermodynamic performance of R432A and HCFC22 is measured in a heat pump bench tester under both air-conditioning and heat pumping conditions. R432A also offers a similar vapor pressure to that of HCFC22 for 'drop-in' replacement. Test results showed that the coefficient of performance and capacity of R432A are 8.5-8.7% and 1.9-6.4% higher than those of HCFC22 for both conditions. The compressor discharge temperature of R432A is 14.1-17.3% lower than that of HCFC22.

THEORETICAL ANALYSIS

Figure (1) and (2) show the refrigeration cycle on *p-h* and Schematic diagram of air-conditioning System. The refrigerant evaporates entirely in the evaporator and produces the refrigerating effect. It is then extracted by the compressor at state point 1, compressor suction, and is compressed isentropically from state point 1 to 2. It is next condensed to liquid in the condenser, and the latent heat of condensation is rejected to the heat sink. The liquid refrigerant, at state point 3, flows through an expansion valve, which reduces it to the evaporating pressure. In the ideal vapor compression cycle, the throttling process at the expansion valve is the only irreversible process, usually indicated by a dotted line. Some of the liquid flashes into vapor and enters the evaporator at state point 4. The remaining liquid portion evaporates at the evaporating temperature, thus completing the cycle.

CYCLE PERFORMANCE

For the evaporating process between points 4 and 1, according to the steady flow energy equation [6],

$$h_4 + \frac{v_4^2}{2000} + q = h_1 + \frac{v_1^2}{2000} + w \qquad \text{... (1)}$$

Where:

$h1$, $h4$ =enthalpy of refrigerant at points 1 and 4, respectively, Btu / lb (kJ /kg)

$v1$, $v4$ = velocity of refrigerant at points 1 and 4, respectively, ft / s (m/ s)

q =heat supplied per lb (kg) of working substance during evaporation process, Btu/ lb(kJ /kg)Because no work is done during evaporation, the change of kinetic energy between points 4 and 1 is small compared with other terms in Eq. (1), and it is usually ignored. Then

$$h_4 + q = h_1 + 0$$

The refrigerating effect q_{rf} ,Btu / lb (kJ / kg), is

$$q_{rf} = h_1 - h_4 \qquad \text{... (2)}$$

For isentropic compression between points 1 and 2, applying the steady flow energy equation and ignoring the change of kinetic energy, we have

$$h_1 + 0 = h_2 + w$$
$$-w = h_2 - h_1$$

Work input to the compressor *W*in, Btu / lb (kJ / kg), is given as

$$w_{in} = h_2 - h_1 \qquad \dots (3)$$

The actual Compressor input power (*Wc*, kJ/s) is given as

$$Wc = I\ V\ COS\ \Theta$$

Where:
COS Ө = power factor ,I=amper and V= Voltage.
Similarly, for condensation between points 2 and 3,

$$h_2 + q = h_3 + 0$$

The heat released by the refrigerant in the condenser - *q*, Btu / lb (kJ / kg), is

$$-q = h_2 - h_3 \qquad \dots (4)$$

For the throttling process between points 3 and 4, assuming that the heat loss is negligible,

$$h_3 + 0 = h_4 + 0$$

Or $$h_3 = h_4 \qquad \dots (5)$$

The COP of the single-stage ideal vapor compression cycle [7] is

$$c.o.p = \frac{refrigerating\ effect}{work\ input}$$

$$= \frac{h_1 - h_4}{h_2 - h_1} \qquad \dots (6)$$

The mass flow rate of refrigerant $\dot{m}$, lb/h (kg/ s), flowing through the evaporator is

$$\dot{m} = \frac{Q_{rf}}{q_{rf}} \qquad \dots (7)$$

Where:

Qrf refrigerating capacity, Btu /h (W).

EXPERIMENTAL SETUP

The basic measurement that needed in this study is the flow rate of the refrigerant, the temperature of refrigerant and the pressure of the refrigerant too. Four temperature gauges were installed on the refrigerant side to measure the temperature of the refrigerant at different position. The technique used is by installing the thermometer immersed through the flow. This method provides a direct contact between the bulb and the refrigerant to give more accurate measured values for temperature measurements. Two temperature gauges were installed on the high pressure sides.These where manufactured by Ashcroft, with a temperature range of (2 to 150 ºC) at a division of (2º). Two temperature gauges were installed on the low pressure sides. The Bourdon tube gauge namely "AIRMINDER" was used to measure the refrigerant side pressure at different points, including pressure drop across the evaporator and condenser. Two pressure gauges were installed on the high pressure side having a range of (-30 inHg to 500psi), the division (10 psi), and two pressure gauges installed on the low pressure side having range (-30 inHg to 250psi), the division (5 psi).

The Ampere was measured by using digital glam meter, made by (266 Digital clamp meters). Voltage is measured by using Multimeter made by (PRO'SKIT 345) where the measuring is obtained continuously.The volume flow rate is measured using a Rotameter. The range of Rotameter is from 0 - 50 ml/sec. The volume flow rate was computed from Rotameter that in liquid line high-pressure side. The system was charged with the help of charging system and evacuated with help of vacuum pump to remove the moisture. After charging each refrigerant, data were collected at different ambient temperatures under transient conditions and the following performance parameters were obtained using Eqs. (1) to (6): Compressor input power (Wc), refrigerating effect (Qe) and Coefficient of Performance (COP).

RESULTS AND DISCUSSIONS

The variation in refrigeration rate of evaporator, Coefficient of Performance (COP), work consumption of compressor and heat rejection rate of condenser with time from the start to stop of the air-conditioning system are shown from in fig.(3) to (7). The comparison of variation in COP values of air-conditioning system used two refrigerants under transient conditions is shown in Figure (3).

The coefficient of performance of R431a is 14.67-20.34% higher than that of R22. Figure (4) shows the change of C.O.P versus time for gas R22 with different ambient temperatures. The C.O.P decreases from (2.646) to (2.427) with ratio (8.28%) through 5000 sec at $T_{air} = 28\ C°$ for gas R22 .But C.O.P decreases from (2.27) to (1.94) with ratio (14.54%) through 5000 sec at $T_{air} = 36\ C°$ for gas R22.This is because of the increase of discharge temperature. While figure (5) show the change of C.O.P versus time for gas R431a with different ambient temperatures. The C.O.P decreases from

(3.35) to (3.05) with ratio (8.96%) through 5000 sec at $T_{air} = 28\ C°$ for gas R431a .But C.O.P decreases from (2.921) to (2.65) with ratio (9.28%) through 5000 sec at $T_{air} = 36\ C°$ for gas R431a.

Figure (6) and Figure(7) the variation in the refrigeration rate of evaporator and heat rejection rate of condenser drops almost parallel to each other, both with a decreasing rate with time. But the power consumption does not increasing much for gas R22 and R431a respectively. In Figure (6), the heat rejection rate of condenser decreases from (8150 W) to (7690 W) with ratio (5.64%) through 5000 sec at $T_{air} = 30\ C°$ for gas R22 .But the refrigeration rate of evaporator decreases from (5840 W) to (5540 W) with ratio (5.14%) through 5000 sec at $T_{air} = 30\ C°$ for gas R22.

In Figure (7) the heat rejection rate of condenser decreases from (12800 W) to (12500 W) with ratio (2.34%) through 5000 sec at $T_{air} = 30\ C°$ for gas R431a .But the refrigeration rate of evaporator decreases from (10800 W) to (10509 W) with ratio (2.69%) through 5000 sec at $T_{air} = 30\ C°$ for gas R431a.

CONCLUSIONS

The following main conclusions of this study could be summarized as follows:

- The refrigeration system using R431a has an electrical power which less than the system using R22 by 14.19%.
- The refrigeration system using R431a has a higher C.O.P than that of R22.
- The C.O.P decreases with time.
- The refrigeration system using R431a has a discharge temperature lower than that for R22.
- Refrigeration rate of evaporator and heat rejection rate of condenser decreases with time.

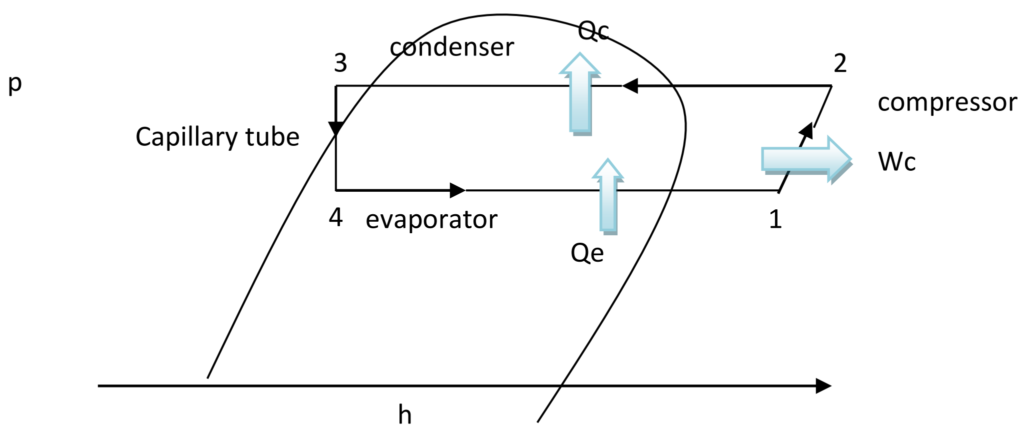


Figure (1) A Residential air-conditioning System on p-h diagram.

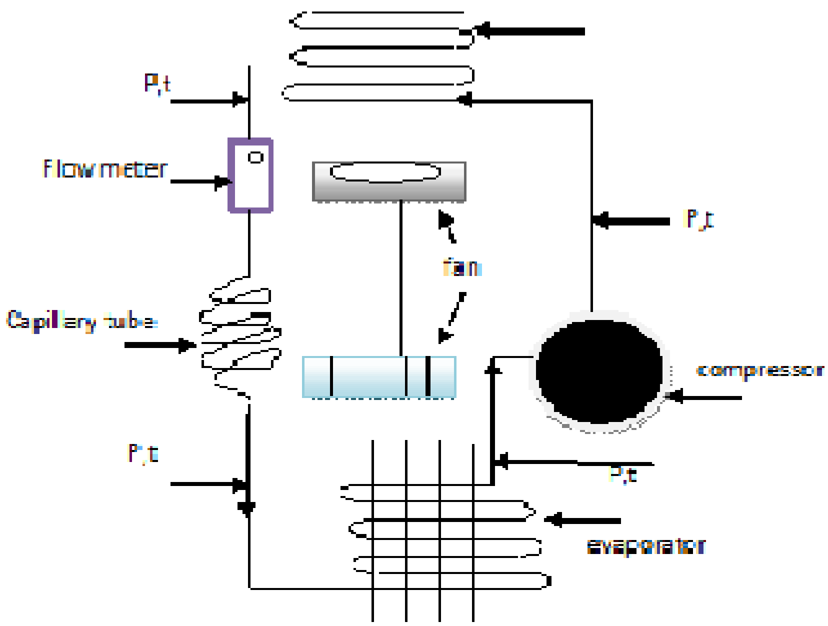
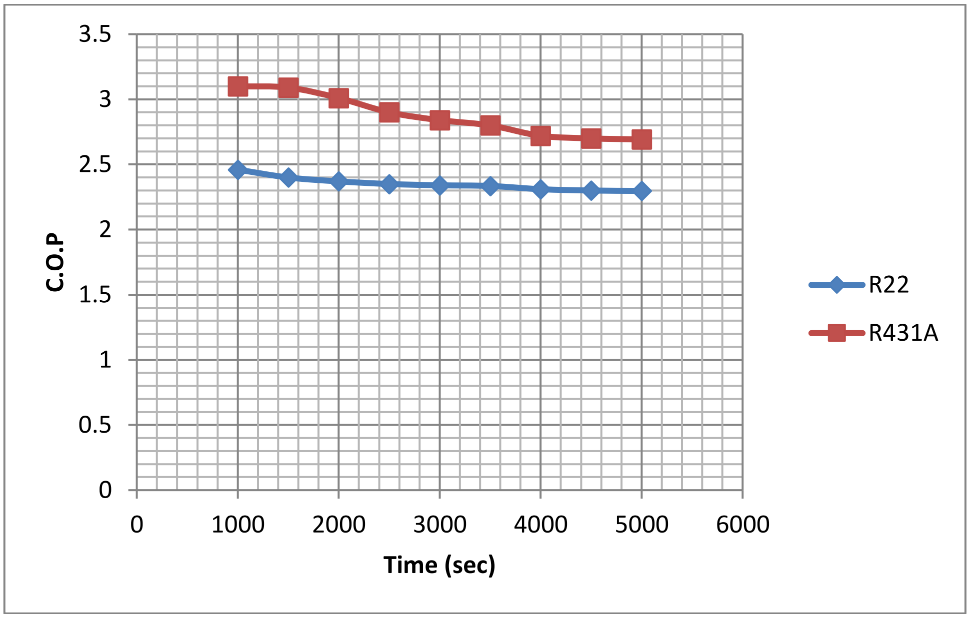


Figure (2) Schematic diagram of air-conditioning System.



Figure(3) Variation in actual COP at ambient temperatures (32 C°) under transient conditions for R22 and R431a.

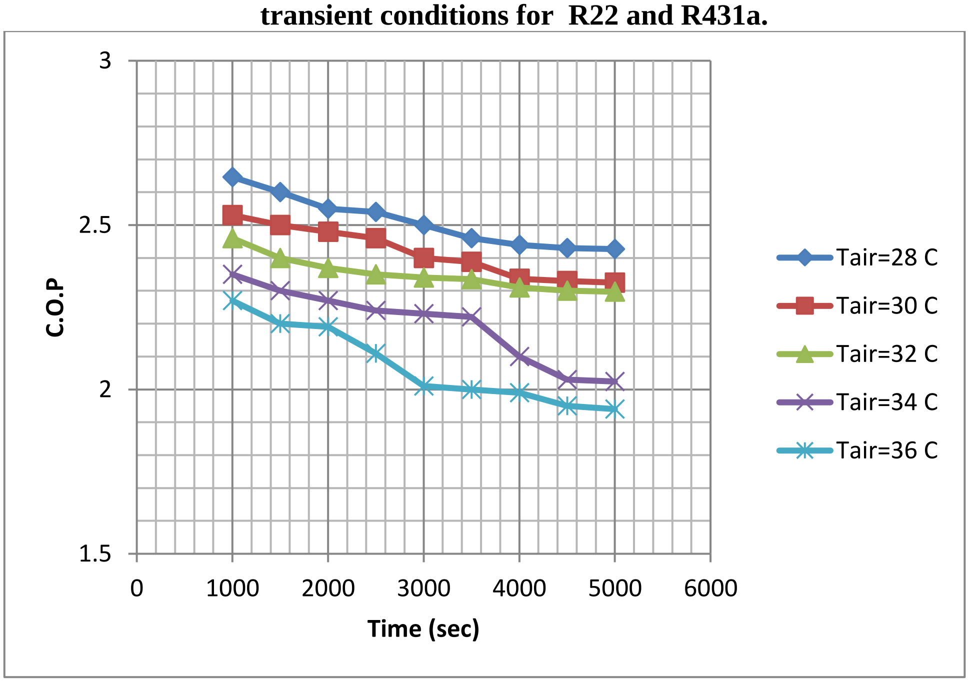


Figure (4) Variation in actual COP at different ambient temperatures under transient conditions for R22.

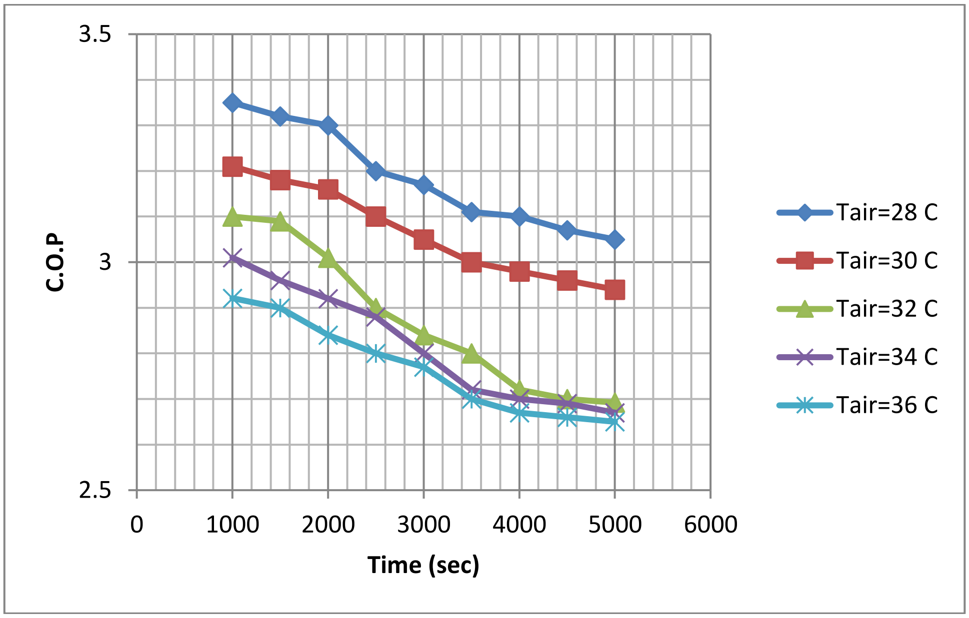


Figure (5) Variation in actual COP at different ambient temperatures under transient conditions for R431a.

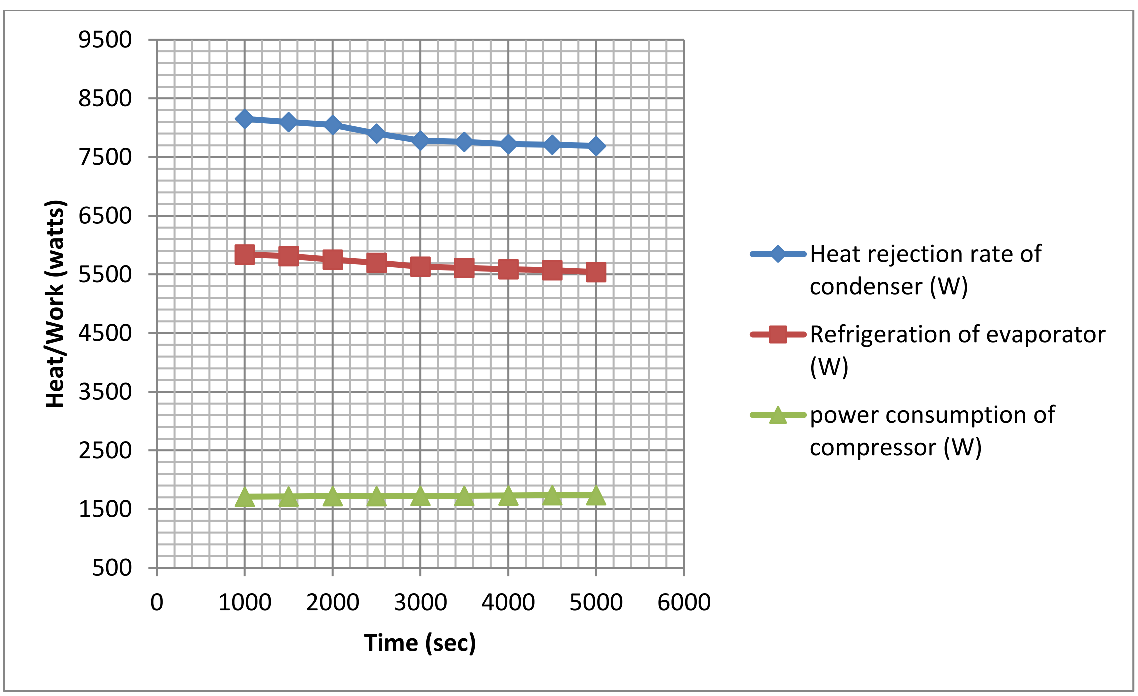


Figure (6) Performance of air-conditioning system under transient conditions at ambient temperatures (30 C°) for R22.

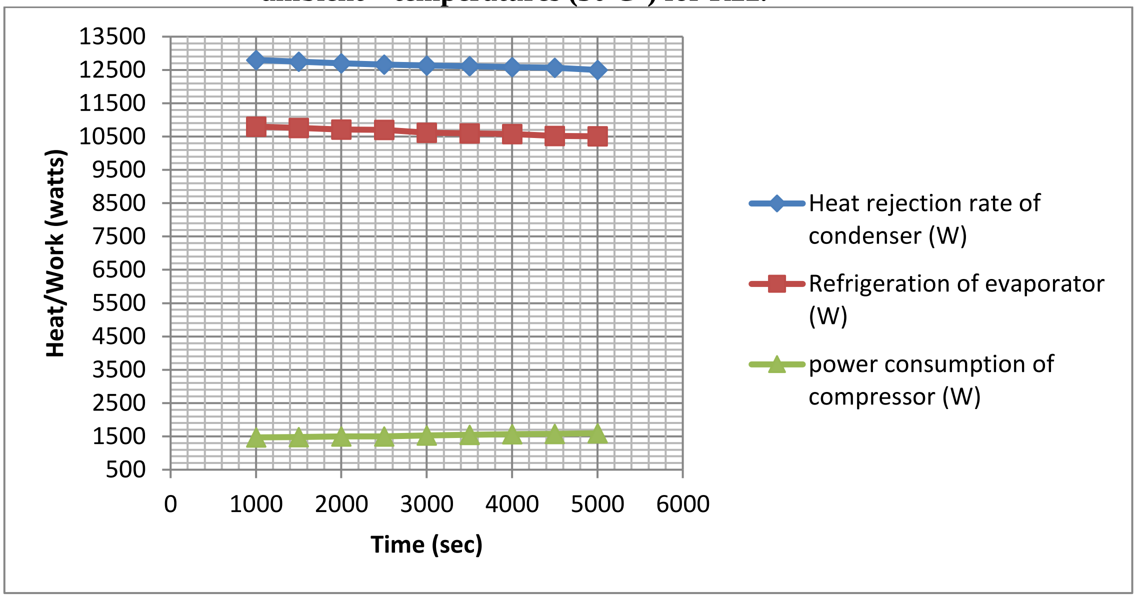


Figure (7) Performance of air-conditioning system under transient conditions at ambient temperatures (30 C°) for R431a.

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