# AN EXPERIMENTAL INVESTIGATION INTO ALTERNATIVE REFRIGERANTS IN AN AUTOMOTIVE AIR CONDITION SYSTEM UNDER DIFFERENT CHARGE CONDITIONS +

**Ayad Khudhair Al-Nadawi*** **Faik Lateef Saleh***

**Abstract:**

**In this paper, the performances of automotive air conditioner using R12 as a working fluid and its substitute R134a and R600a are experimentally studied. The measured data are obtained from an automotive air conditioning test facility utilizing R12 as a refrigerant. The air conditioner is tested under different charge conditions and compressor speeds. The experiments are conducted at a relatively high temperature which exists in reality at Middle East countries. The coefficient of performance, cooling capacity, compressor power, discharge temperature, and discharge pressure are measured and analyzed to quantify the influence of the refrigerant charge and compressor speed on the steady-state operation. The experimental data obtained from the test apparatus bring into view the relationship between the above mentioned parameters in the automotive air conditioning system. The tests proved that 1000, 1250, and 750 g are the best charge for R12, R134a, and R600a, respectively.**
**Key words: Refrigeration, Alternative refrigerants, R12, R134a, R600a, Performance.**

# دراسة عملية لبدائل موائع التثليج في منظومة تكييف هواء السيارة وتحت ظروف اوزان مختلفة لموائع التثليج

أياد خضير النداوي فائق لطيف صالح

المستخلص:

في هذا البحث، تمت دراسة اداء منظومة تكييف الهواء في السيارات بأستخدام R12 كمائع تثليج و بدائله من R134a و R600a عمليا. تم اختبار المنظومة تحت اوزان مائع مختلفة و بسرع للضاغط مختلفة. تم جمع القراءات بدرجة حرارة جو عالية كالتي موجودة في دول الشرق الاوسط. تم حساب و تحليل معامل الاداء، سعة التبريد، سرعة الضاغط، درجة حرارة المائع بعد الضاغط، وضغط المائع بعد الضاغط لمعرفة تأثير اختلاف كمية شحنة الغاز و سرعة الضاغط على اداء دورة مستقرة. التجارب برهنت بأن القراءات 1000، 1250، 750 g هي الافضل شحنة للموائع R12 ، R134a ، R600a ، بالتتابع.
الكلمات المفتاحية: تثليج، بدائل الغازات، R12, R134a, R600a، الاداء.

+ Received on 4/10/2015 , Accepted on 28/7/2016 .
*Asst. Lecturer/ Technical Institute in Anbar

**Nomenclatures:**

| |
|---|
| AAC: Automotive air conditioning. |
| CFC: Chlorofluorocarbon |
| COP: Coefficient of performance. |
| GWP: Global warming potential |
| h : Specific enthalpy kJ $kg^{-1}$ |
| HC: Hydrocarbon |
| HCFC: Hydrochlorofluorocarbon |
| $K_P$: Coverage factor. |
| M: Stands for "function of ". |
| $\dot{m}_r$: Refrigerant mass flow rate kg $s^{-1}$ |
| ODP: Ozone depletion potential |
| $Q_{evap}$: Cooling capacity (kW). |
| $q_e$: The best estimated quantity. |
| Q: Discrete value |
| SSSF: Steady state steady flow. |
| $u_q$: Standard uncertainty. |
| U: Expand uncertainty. |
| W: Power (kW). |
| Subscript |
| comp: compressor. |
| evap: Evaporator. |
| in: inlet. |
| out: outlet. |
| r: refrigerant. |

## Introduction:

Several environmental problems have appeared during the last decades. One of the serious threats to the environment is the stratospheric ozone layer depletionand global warming caused mainly by chlorofluorocarbons (CFCs) and hydrochloroflourocarbons (HCFCs)refrigerants. It is believed that Chlorine contained in CFC and HCFC is the main factor causing the ozone depletion. Besides, the presence of CFC and HCFC in the environment plays a significant role in the green house effect which is expected to cause many unwanted environmental effects. These effects were not considered when the refrigeration companies first used these refrigerants as working fluid in air conditioner, refrigerator, heat pump and automotive air conditioner. The Montreal protocol was held in September 1987 to phase out ozone depleting substances within a fixed time period. According to this protocol, CFC used mainly in refrigeration and air condition equipments, was expected to phase out by 1996 in the developed countries. Developed countries will phase out HCFC by 2030 while developing countries have been provided a grace period of ten years i.e. phase out by 2040 [1]. Because thermo-physical properties of R134a are very similar to those of R12; R134a was identified as a replacement of R12. Although R134a has a zero depleting potential (ODP), it has 1430 global warming potential (GWP) per 100 years, which is very high (see Table 1). The emission of R134a to the environment is increasing the concentration of greenhouse gases by direct way such as leaks or by indirect ways like performance of refrigeration plants. For all of these reasons, the use and production of R134a will be terminated in the near future. Over years, several research works have been conducted in order to find the attractive refrigerants. However, most of works found in literature focus on the study to replace the conventional refrigerant in air conditioners, water chillers, and domestic refrigerators.

Radermacher R. presented a computer study and tests of fifteen binary zeotropic refrigerant mixtures consisting of the components R23, R32, R125, R134a, R143a and R125a investigated as possible replacement fluids of R22. The two mixtures of R32/R134a and R32/R125a showed COP has improved over R22 of up to 24% at the same capacity as with R22 while using counter flow heat exchanger in evaporator and condenser [3].Kim M.S. & Didion D.A. presented thesimulation of adiabatic and isothermal process of binary and ternary refrigerant mixtures; R32/134a and R32/125/134a.

**Table 1- Properties of various refrigerants**

| Refrigerant | Chemical Formula | Molecular Mass | Boiling point at 101.325 Kpa °C | Freezing Point °C | Critical Temperature °C | Critical Pressure Kpa | Atmosphere life time, years | ODP | GWP | Safety Group |
|---|---|---|---|---|---|---|---|---|---|---|
| R11 | $CCl_3F$ | 137.4 | 23.708 | -110.47 | 197.96 | 4407.6 | 45 | 1 | 4750 | A1 |
| R12 | $CCl_2F_2$ | 120.9 | -29.752 | -157.05 | 111.97 | 4136.1 | 100 | 1 | 10900 | A1 |
| R22 | $CHClF_2$ | 86.5 | -40.81 | -157.42 | 96.145 | 4990.0 | 12 | 0.055 | 1810 | A1 |
| R134a | $CH_2FCF_3$ | 102.03 | -26.074 | -103.3 | 101.06 | 4059.3 | 14 | 0 | 1430 | A1 |
| R600 | $CH_3CH_2CH_2CH_3$ | 58.1 | -0.49 | -102.7 | 151.98 | 3796.0 | 0 | 0 | ~20 | A3 |
| R600a | $CH(CH_3)_2CH_3$ | 58.1 | -11.75 | -157.42 | 134.66 | 3629.6 | 0 | 0 | ~20 | A3 |

Source, ASHRAE Handbook [2]

The results of this study show that the refrigerant mixture left in the system remains in a nonflammable region during the isothermal vapor leak for both cases of binary and ternary mixture [4]. Steven Brown J. et al., evaluated performance merits of $CO_2$ and R134a automotive air conditioner using vapor compression and transcritical cycle simulation models. The analysis showed R134a having a better COP than $CO_2$ with the COP disparity being dependent on compressor speed (system capacity) and ambient temperature [5].

Tolouee conducted an experiment on water chiller to study R407C as a replacement of R22 refrigerant. The effect of leak on the system performance was studied too. It has been observed that the adverse effect of leak beyond 30% on the capacity becomes more significant[6]. Mohanraj et al, presented the experimental results of energy efficient hydrocarbon (HC) mixture which consisted of 45% R290 and 55% R600a as a drop in substitute for R134a at various mass charges of 50, 70, and 90 g in domestic refrigerator. The results show that (R290/R600a) mixture has better COP, lower power consumption compared to R134a [7].

Atik.K & Atkas. A investigated the performance change in automotive air conditioning system depending upon the refrigerant leakage. The experimental data reveal that the best cooling capacity was achieved at 500 g refrigerant charge level[8].

Herbert M. & Mohan D. studied the effect of refrigerant charge level of R22 and M20 (80% R407C and 20% HC blend by weight) on a window air conditioner under different outdoor conditions. It was reported that the decrease in refrigerant charge level of about 7% reduced the system refrigerant capacity by 11.3% with R22 while with M20 refrigerant it was reduced by 6.9% only [9].

Tarrad A. H. & Abbas A.K, presented experimental analysis of window type air conditioner using R12, R407C, and R407A. The results show that the drop in technique of R22 by R407C and R407A improved cooling capacity up to 12% and 25% respectively. R407C exhibited lower power consumption than that experienced with R22 tests by 4%. On the contrary, R407A showed a higher consumed power than that of R22 by 4%. R407C and R407A showed a significant increase in COP by 21% and 27% respectively [10].

Jiang L. &Hrnjak P. presented analysis of location of refrigerant inventory and theoretical results of charge reduction in small commercial refrigeration systems. The result shows most of the charge is retained in the condenser and liquid line[11].

Gaurav & Kumar R. presented a comparison of energy and exergy analysis of R134a, R152a, R290, R600, and R600a in a domestic refrigerator. The analysis showed that R152a has the highest value of COP and exergy efficiency than the other refrigerants [12]. In the present study, the main concern is to obtain experimental data to obtain replacement refrigerants in AAC systems. R134a and R600a were investigated as drop-in substitutes of the original R12 refrigerant. The experimental study was done with various refrigerants, charges, air temperatures and compressor speeds to determine COP, cooling capacity, and the compressor power. Different amounts of the same type charge refrigerant were charged into the system and then the measurements were repeated.

## Experimental apparatus and test procedure:

A schematic diagram of the experimental apparatus is shown in Fig.1. The AAC system was originally designed to work with R12. The main modification to the standard automotive air conditioning system is the addition of measuring devices. The air conditioner with capacity of 4.2 kW consists of finned tube evaporator and condenser, a swash type compressor with 132 cc/rev displacement volume, magnet clutch, receiver tank and thermal expansion valve. The evaporator and condenser are made from copper tubes and aluminum fins. The evaporator has laminated type fin geometry whereas; the condenser has corrugated type fin geometry. Sight glass is used to verify that sub-cooled liquid is exiting the condenser. A 9.5 mm diameter filter drier is installed before the expansion valve to remove the contaminants. The compressor pulley is driven by a 3.7 kW, 3-phase and 3000 rpm DC electrical motor attached to control panel. This motor drives a shaft coupled to the compressor through a pair of pulleys and a V-belt. The test unit is suitably modified to connect thermocouples, flow meter and pressure gauges. The refrigerant temperature is measured by employing seven J-type thermocouples mounted (touched) at different locations along the refrigerant loop, two at evaporator inlet, one at compressor inlet, discharge temperature, after the flow meter, condenser inlet and outlet.

A selector is used to control the temperature readings. These readings are displayed by HD-2000 digital surface equipment with the range of temperature from (-200 to 1200)℃. The system is an instrument of three Bourdon tube pressure gauges. The pressure connection ports are fixed (welded) into the tubes into which the refrigerant flows. The open type wind tunnel is used to conduct air flow through evaporator. Fresh air is passed through evaporator and condenser by means of centrifugal and axial fans respectively. The air temperature at the evaporator and condenser inlet has been kept at 30 ℃ and 45 ℃ by means of an electrical heater resistance. The heater resistance is controlled by using power supply (variable 220V and constant 110 V). The wet and dry bulb temperatures of air were measured at air inlet and outlet section of condenser and evaporator. The dry bulb temperature is measured by using a mercury glass thermometer (0-100)℃ and the wet bulb temperature is measured by using wet cotton wick on the bulb of a mercury glass thermometers. The refrigerant mass flow rate is measured by float type flow meter (0.5-5) lit $min^{-1}$, its location is shown in Fig.1. The refrigerants used in the present study are R12, R134a and R600a. The characteristics of refrigerant added to the system are listed in Table 2.

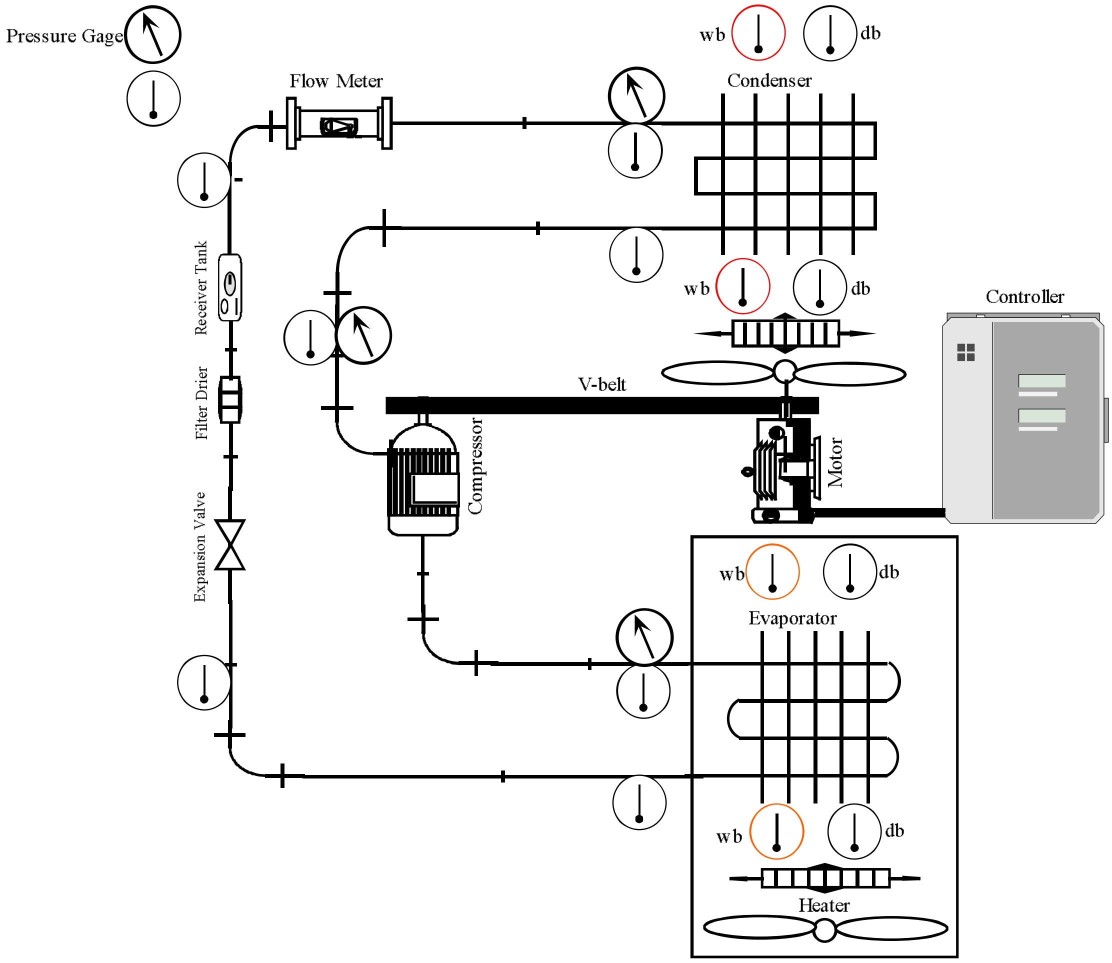


**Fig. 1- Schematic diagram of experimental apparatus.**

**Table 2 - Refrigerant added to the system and their characteristic**

| Refrigerant | Saturation temperature at 101.33 kPa[c] (°C) | Latent heat at 101.33 kPa (kJ $kg^{-1}$) | Vapour specific volume ($m^3$ $kg^{-1}$) | |
|---|---|---|---|---|
| | | | At -30 °C | At +30 °C |
| R12 | -29.75 | 166.17 | 0.16057 | 0.02377 |
| R134a | -26.07 | 216.97 | 0.22594 | 0.02664 |
| R600a | -11.75 | 365.11 | 0.72839 | 0.09542 |

[c] From ASHRE [2]

A parameter, associated with the measurement that characterizes the dispersion of the values that could reasonably be attributed to the measurand is known as uncertainty [13]. Therefore, the following equations may be used to calculate the uncertainties in the derived experimental values.

$$Q = M(Z_1 \ldots\ldots\ldots\ldots . Z_n) \qquad (1)$$

$$\left(\frac{u_q}{q_e}\right)^2 = \sum_{i=1}^{N}\left(\frac{u_z}{z}\right)^2 \qquad (2)$$

$$U = K_P u_q \qquad (3)$$

For 98% as a degree of belief ascribed to the true value of Q being inside the interval, $K_P$ will be taken as 2.327, [13]. From calibration, it was found that the maximum error in the temperature reading was 1.2 ℃ and minimum temperature error was – 1.9℃ considering the highest reading of the meter range. On the other hand, the maximum error of the pressure reading was +0.4 bar and the minimum pressure error was 0 bar. These results will be used to find the uncertainties in the derived values. See Table 3.

**Table 3- The Uncertainties in the Derived Experimental Values**

| Derived Experimental Values | Uncertainty |
|---|---|
| Refrigeration Capacity | ∓0.9 % |
| Compressor Power | ∓ 1% |
| Coefficient of Performance COP | ∓ 0.2 % |

To compare different refrigeration capacities of the AAC system using R12, R134a, and R600a, the tests were conducted by varying two variables:(1) the refrigerant charge and (2) the compressor speed. Initially, the refrigerant in the AAC was totally discharged prior to the tests and then recharged with the desirable refrigerant charge. The system was charged by 500, 750, 1000, and 1250 g for each refrigerant. At each refrigerant charge, the engine speed was set constant at 500, 1000, 1500, 2000, 2500, and 3000 rpm. The evaporator air temperature was set constant at 30 ℃ while the condenser air temperature was adjusted to 45. The measurements were taken at each compressor speed, refrigerant charge, after the system had been fully stabilized. The test began at lowest refrigerant charge, and compressor speed then stepped up to the highest values. First, the refrigerant charge was set constant then the tests were conducted at different compressor speeds. To maintain stability of the AAC, the system was run 15 minutes prior to each test condition.

## Method:

For computing the system performance, the following assumptions were made:

- The air temperature at the entrance and exit of heat exchangers is homogenous and constant at all points.
- The mass flow rate is constant at all parts of the automotive air conditioning units.
- The enthalpy change in the expansion valve is negligible.

The system analysis of the current study is based on SSSF process. The parameter concerning the system performance was calculated as follows:

$$Q_{evap} = \dot{m}_r(h_{evap\ out} - h_{evap\ in}) \quad (4)$$

Power (W) given to the refrigerant in the compressor can be expressed by:

$$W = \dot{m}_r(h_{comp\ out} - h_{comp\ in}) \quad (5)$$

The coefficient of performance (COP) can be calculated by:

$$COP = \frac{Q_{evap}}{W} \quad (6)$$

Here, the enthalpy changes according to the related pressure and temperature values were obtained by coolpack [14].A typical prediction of the refrigeration cycle is given in the p-h diagram for sample of results shown in Fig. 2.

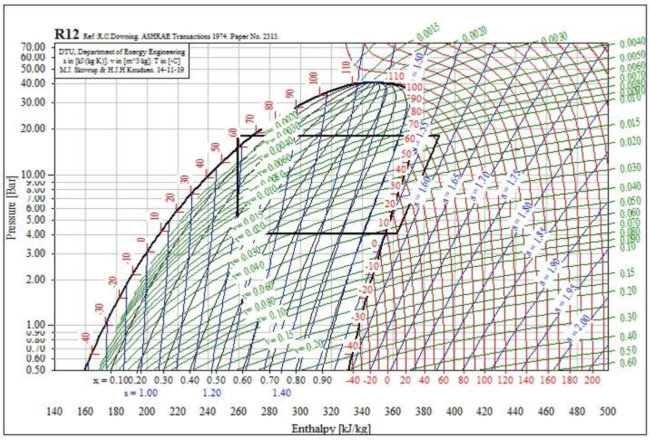


**Fig. 2- A typical p-h diagram for sample data**

## Results and discussion:

R12, R134a, and R600a refrigerants charged and analyzed to compare their performance.The parameters evaluated included the refrigeration capacity, compressor power, coefficient of performance (COP), discharge temperature, discharge pressure. Fig. 3(a) shows the variation of refrigerant capacity as a function of compressor speed under different charge conditions (750, 1000, and 1250 g). It should be stated that when the compressor speed increases, the cooling capacity shows a little influence and a slight increment at (750 g) refrigerant amount. From the same figure, it is obvious that cooling capacity shows increasing trend at (1000 g) refrigerant amount while it is almost constant at (750 g) refrigerant amount. The maximum value of cooling capacity was found to be approximately 3.27 kW at refrigerant amount of (1000 g) when the compressor speed was 2500 rpm for R12.

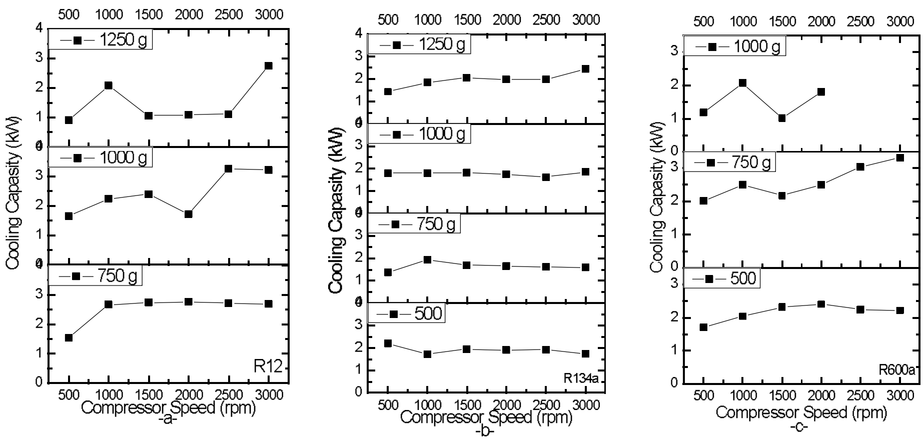


**Fig.3- Effect of different charges on refrigerants capacity of (a) R12, (b) R134a and (c) R600a.**

A comparison between (1000 g) and (1250 g) tests over (750 g) as a base value shows that the cooling capacity of the system is decreased by 60 % ( 1250 g and at 2000 rpm)and increased up to 21% (1000 g and at 2500 rpm) as a maximum value over its base charge. Therefore, the test resultsdemonstrate clearly that the cooling capacity with (750 g) is very stable compared to the results showed by (1250 g ) refrigerant amount and the increasing stable trend is obvious at

(1000 g) refrigerant amount. Further, the average value of cooling capacity for 750, 1000, and 1250 g is2.51, 2.4, and 1.5 kW respectively. The results of (1000 g) and (1250 g) exhibit around -4 % and -40 % respectively over its base charge. For R134a, the data shown in Fig. 3(b) reveal almost constant cooling capacity with compressor speed variation for all refrigerant amounts. The maximum value of cooling capacity was around 2.45 kW at refrigerant charge of (1250 g) when the compressor speed was 3000 rpm. This can be attributed to the increase in mass flow rate due to increase in refrigerant charge and speed of the compressor. However, the cooling capacity was decreased by 12.5 % (500 g and at 2000 rpm) over (750 g) refrigerant charge and at the same compressor speed. Whereas, the cooling capacity increased by 53 % (1250 g and at 3000 rpm) over the base refrigerant charge. Further, the maximum average cooling capacity was found to be 1.96 kW at (1250 g) refrigerant charge which exhibits increase by 19 % over the base refrigerant charge. It can be concluded easily that the best results are shown by (1250 g) and this refrigerant charge exhibits the best one for R134a regarding the cooling capacity. For the test with R600a shown in Fig. 3(c), it was found that the discharge pressure is very high and the whole system shows instability during the test and this instability increases as the refrigerant charge and compressor speed increase. The instability started to be very obvious at (1000 g) tests with 2500 rpm and 3000 rpm compressor speed. Because of all of these, the evaluation and comparison with (1250 g) refrigerant charge have not been reported. It is clear that the test results show high stability and linear trend at (500 g and 1000 g) refrigerant charge. The refrigerant capacity obtained from (500 g and 750 g) test results increases as the compressor speed increases. It must be noted that the maximum value of cooling capacity was found to be 3.3 kW at (750 g) refrigerant charge and 3000 rpm as a result of increasing of refrigerant velocity. Also, the cooling capacity was decreased by 53 % (1000 g and at 1500 rpm) and it increased by 9.4 % (750 g and 2000 rpm) over the base charge. The maximum average cooling capacity was around 2.59 k W at (750 g) refrigerant charge. The (500g) charge test decreased by 42 % over the base refrigerant which indicates clearly that (750 g) charge is the best choice for R600a. Fig. 4(a) shows the relationship between the compressor power and the compressor speed of R12 under different charge conditions. It is found that when the compressor speed increases, the compressor power also increases as a result of increasing in mass flow rate of the refrigerant except particularly for the 1000g (R12).In fact, there is a decrease from 1000 to 2000 rpm and also for the 500g (R600a), there is a decrease from 2500 to 3000rpm.

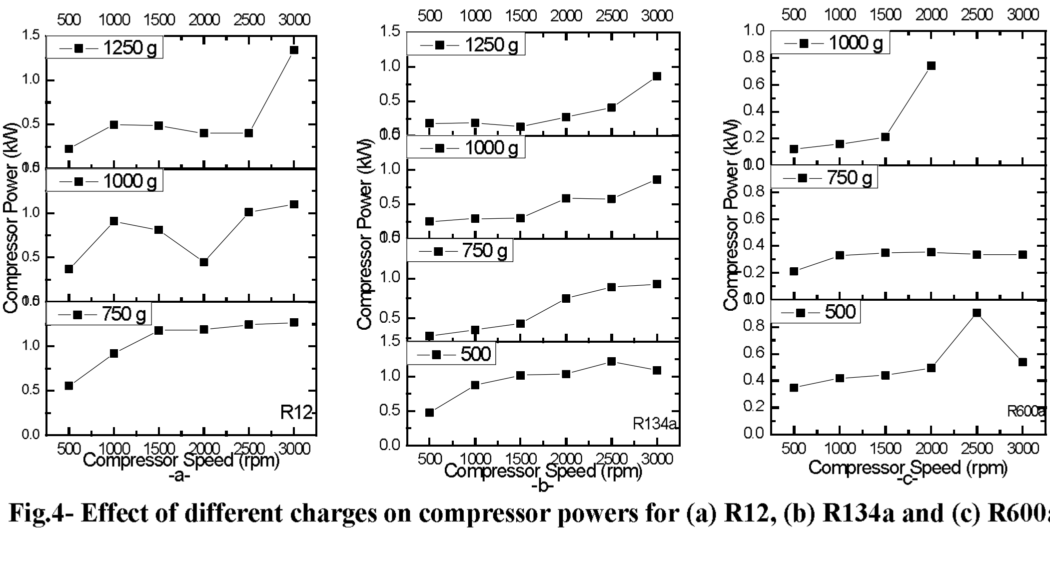


**Fig.4- Effect of different charges on compressor powers for (a) R12, (b) R134a and (c) R600a.**

The maximum compressor power was around 1.3 kW at (750 g) and 3000 rpm. However, the compressor power was decreased by 66% (1250 g and at 2000 rpm) as a maximum value over the base charge refrigerant. On the other hand, it increases by 5.5% (1250 g and 3000 rpm) as a maximum value over the base refrigerant. This can be explained by the effect of isentropic slope of p-h diagram. When the refrigerant charge is increased, the degree of superheat after the evaporator decreases. As a result of that, the isentropic slope increases $\left\{\left(\frac{\partial p}{\partial h}\right)_s = \frac{p}{RT}\right\}$ and hence less power consumption is required. Also, the average compressor power of (1000 g) results was around 0.77 k W and for (1250 g) results was around 0.56 k W which exhibits decreases by 27% and 47% respectively over its base charge. The compressor power data of R134a show the same behavior as that shown in R12 results, Fig.4 (b). The compressor powerincreases linearly when the compressor speed increases. Due to high specific heat of R134a, the dome of the p-h diagram has tilted much more to the right. That means, the compressor power increases at low charge as a result of increasing of the isentropic slope at high degree of superheat. From test results, it was found that the higher power consumption occurred at low charge and vice versa. However, this behavior was very clear at (500 g) charge. The maximum compressor power was 1.22kW (500g and at 2500 rpm) which is about 37 % higher than its base charge. It is obvious that the compressor power of (500 g) results (at low compressor speed) is higher than its base charge by twice. The average compressor power of 500, 750, 1000, and 1250 g was around 0.951, 0.604, 0.478, and 0.34 kW respectively. The (500 g) results show an increase by 57.5% over its base charge while (1000 g) and (1250 g) results show a decrease by 21% and 44% respectively over its base charge. Fig.4(c) shows the compressor power versus the compressor speed for R600a. The compressor power increases when the compressor speed increases as well. So, clearly, it shows the same trend as that of R12 and R134a refrigerants. The compressor consumed higher power at low charge for the reasons mentioned earlier. However, sometimes it may consume higher power at higher refrigerant charge. For instance, the maximum value of power consumed was 0.74 k W (1000g at 2000 rpm) and it exhibited 100% higher value than the base charge. This can be attributed to the refrigerant droplet which passed to the compressor as the degree of superheat after the evaporator disappeared. The average compressor power was 0.523, 0.447 and 0.307 k W for 500,750, and 1000 g respectively. For (500 g) and (1000g), the data shows about 17% and -31% respectively over its base charge.

Fig.5(a) shows the relationship between the coefficient of performance (COP) and compressor speed under different charge conditions. It can be clearly seen that COP decreases with increasing the compressor speed except particularly for the 1000g (R12) and 1250g (R12). This is because the increase in compressor power is more than that of cooling capacity. However, at low refrigerant charge, the decrease in COP shows a linear behavior compared to the higher refrigerant charge results. The COP increases by 66.4% (1000 g and at 2000 rpm) as a maximum value over the base refrigerant. This can be attributed to the increase in mass flow rate as a result of increase in the charge of the refrigerant.

The average COPs of 750, 1000, 1250 g were around 2.4, 3.32, and 2.8 respectively. For (1000 g) and (1250 g), the data shows increases in COP by 38.3% and 17% respectively over its base charge. In Fig.5 (b), the COP of R134a versus the compressor speed was reported. The test results show the same trend as R12 but it is more stable and linear. The COP increases when the refrigerant change increases. On the other hand, it decreases when the compressor speed increases. This can be attributed to the effect of high mass flow rate and compressor power. Due to the effect of mass flow rate and compressor power, the COP of (1250 g) is much higher than that of (750 g) results. However, the average cooling COPs of (500 g) and (1000 g) were about -33.3% and 26.6% respectively over its base charge.

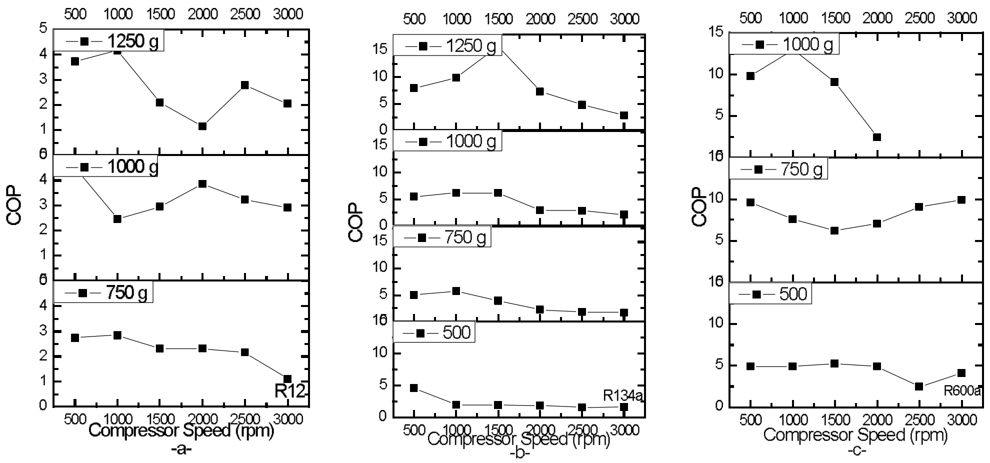


**Fig.5- Effect of different charges on COPfor(a) R12, (b) R134a and (c) R600a.**

For R600a, Fig.5(c) shows that COP of (1000 g) decreases severely with the increase in the compressor speed. On the other hand, the COP of (500 g) was almost constant. This is attributed to the effect of the compressor power which shows the same trend as that of COP behavior.However, the COP was decreased by 73% (500 g and at 2500 rpm) as maximum value over the base charge and increased by 71% (1000 g and at 1000 rpm) as a maximum value over its base charge. The average COP was about -47% and 4.3% for (500g) and (1000g) respectively over their base charges. The test results prove that (750 g) results was the best charge for R600a because of its high COP and stability compared with the other refrigerant charge.

## Conclusions:

The following Conclusions have been drawn from the research:

1. The effect of compressor speed is more pronounced than the cooling capacity.
2. The test results of R134a are more stable compared to those of R12 and R600a.
3. At low compressor velocities, the power consumption is low and then the COP of the system increases.
4. For R134a and R600a, the compressor used more power at low charge due to the effect of its high specific heat.
5. The test results of R600a at high charge and compressor speed could not be analyzed and compared due to its instability.
6. R134a and R600a are suitable replacements for R12. They yield higher COP than R12. Additionally, R134a is more suitable with regard to its safety while R600a is more suitable with regard to its environmental characteristics.
7. For R134a, the test device was not modified and there was no problem with compressor and refrigerant lines.
8. For R600a, the test device needs to be redesign to adapt to the characteristics of R600a, especially at high compressor speeds.
9. 1000, 1250, and 750 g proved to be the best charge for R12, R134a and R600a, respectively.

**Acknowledgement:**

The authors would like to acknowledge the help from the Middle Technical University /Technical Institute in Anbar/ Machinery and Equipment Dept.

**References:**

1- Verma J.K., Satsangi A., Chaturani V., “A Review of Alternative to R134a ($CH_3CH_2F$) Refrigerant”, International Journal of Emerging Technology and Advanced Engineering, Vol.3, Issue 1, pp 300-304, 2013.
2- ASHRAE Handbook-Fundamental (SI), “American Society of Heating, Refrigerating, and Air Conditioning Engineers”, Chapter 29, 2009.
3- Radermacher R., "Performance Evolution of Chlorine Free Zeotropic Refrigerant Mixtures in Heat Pump", International Institute of Refrigeration Conference, 1992.
4- Kim M.S. & Didion D.A., “Simulation of Isothermal and Adiabatic Leak Processes of Zeotropic Refrigerant Mixture”, HVAC &Research, Vol.1, No.1, pp. 3-20, 1995.
5- Steven Brown J., Samuel F., Yana-Motta and Domanski P., “Comparative Analysis of an Automotive Air Conditioning Systems Operating with CO2 and R134a”, International Journal of Refrigeration, ELESVIER, USA, 25, pp. 19-32, 2002.
6- Tolouee C.M., “Experimental Study of Zeotropic Refrigerant Mixture HFC-407C as A Replacement for HCFC-22 in Refrigeration and Air Conditioning Systems”, PhD Thesis, Swinburne University of Technology, School of Engineering and Science, 2006.
7- Mohanraj M., Jayaraj S., and Muraleedharan C., “Improved energy Efficiency for HFC134a Domestic Refrigerator Rotfitted with Hydrocarbon Mixture (HC290/HC600a) As Drop in Substitute”, Department of Mechanical Engineering, National Institute of Technology, Vol.11, Issue 4, pp.29-33, 2007.
8- Atik K. & Aktas A., “An Experimental Investigation of The Effect of Refrigerant Level on An Automotive Air Conditioning System”, Thermal Science and Technology, Turkey, pp. 11-17, 2011.
9- Herbert M. & Mohan D., “An Experimental Analysis of the Effect of Refrigerant Charge Level and Outdoor Condition on a Window Air Conditioner”, Thermal Science, Vol.14, No.4, India, pp. 1121-1138, 2010.
10- Tarrad A.H & Abbas A.K, “Evolution Of A Proper Alternative Refrigerant For R-22 In Air Conditioning Systems”, Emirates Journal for Engineering Research(EJER), Vol.15, Issue 2, UAE, pp.37-47, 2010.
11- Jiang L. & Hrnjak P., “Refrigerant Charge Reduction in Small Commercial Refrigerant Systems”, International Refrigeration and Air Conditioning Conference, Perdue, USA, pp. 14-17, 2014.
12- Gaurav and Kumar R., “Performance Analysis of Household Refrigerator with Alternate Refrigerants”, International Journal of Innovative Research in Science, Engineering and Technology, Vol. 3, Issue 4, India, 2014.
13- Lira I., “Evaluating the Measurement Uncertainty-Fundamental and Practical Guidance”, Institute of Physics, IOP Publishing Ltd, USA, 2002.
14- Jacobson A., Rasmussen B.D, Skrovrup M.J. and Anderson E., “Coolpack package”, Department of Energy Engineering, Technical University of Denmark, 2001.