Optimum Charge and Performance for R432A under Window Type Air-Conditioner Systems

Haider ali hussen
Mechanical Engineering Department, University of Al-Mustansiriyah/Baghdad.
Email:haidereng25@yahoo.com

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ABSTRACT

The study presents optimized R432A charge by the index of power consumption of compressor and COP without changing the compressor. Test results showed that the best charge is 750 g for R432A which gave the highest coefficient of performance and less energy consumption. When charging the system with 750 grams, the amount of charge rate down to 42.3% compared with the original gas. The percentage increase in coefficient of performance are 9.85%, 8.39%, and 8.93% and the proportion of low energy consuming are 7.53%, 7.31%, and 7.34% at ambient temperatures of 30 ºC,34 ºC and 38 ºC respectively for R432A compared to R22. Increasing ambient temperature lead to increase in the power consumed and decrease coefficient of performance.

Keywords: Charge, Performance, Power Consumption, Refrigerant, Air Conditioner System,

الشحنة المثالية و اداء غاز R432A تحت منظومة مكيف هواء شباكي

الخلاصة

هذه الدراسةِ قدمت الشحنة المثالية التي حَسّنتْ بالإستهلاكِ الكهربائيِ في الضاغطِ ومعامل الاداء لغاز R432A بدون تَغيير الضاغطِ. أظهرت نَتائِجَ الإختبارِ بأنّ أفضل شحنة هي 750 g لغاز R432A الذي اعطى المعاملِ الأعلى للأداءِ وأقل إستهلاكِ لطاقة. عندما يَشْحنُ النظامَ في 750 غرامِ، فان كمية الشحنة تقل بنسبة 42.3 % بالمقارنة بالغازِ الأصلي. الزيادة في معاملَ الأداءِ هي 9.85 % ,8.39% و8.93 % ونسبة إسْتِهْلاك طاقةِ اقل 7.53 %,7.31%و7.34% عند درجات حرارة بيئيةِ 30 ,34 و38 درجة مئوية على التوالي لغاز R432A بالمقارنة مَع R22. تُؤدّي درجةُ الحرارة البيئية المتزايدة إلى الزيادةِ في الطاقة المستهلكة وتَنْقصُ معاملَ الأداءِ المنظومة.

Nomenclature

Symbol	Definition	Units
COP	Coefficient of performance	------
I	Current	**Amper**
V	Voltage	**Volt**
h	Specific enthalpy	**kJ/kg**
T	Temperature	**ºC**
P	Pressure	**N/m^2 (Pa)**

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2412-0758/University of Technology-Iraq, Baghdad, Iraq

q	Heat gain or rejected per Kg	**KJ/Kg**
V	Velocity	**m/s**
W	Work input to the compressor	**Watts**

Subscripts

air	Ambient temperature
c	Actual compressor input power
rf	Refrigerating effect

INTRODUCTION

Due to the destruction of the earth's ozone layer by chlorine molecules from chlorofluorocarbon (CFC), many companies have been developed the alternative refrigerants to replace CFC refrigerant and hydro chlorofluorocarbon (HCFC) refrigerant in the refrigeration and air-conditioning applications for example, HFC-417A, HFC-404A, HFC-407C, HFC-134a, and HFC-410A.The resulting environmental concerns about ozone depletion and global warming have stopped the production of R12 and other CFC refrigerants and the production of HCFCs will stop in 2015[1]. Hydrocarbon refrigerants are natural fluids with no ozone depletion potential (ODP) for example R432A [table 1] (ASHRAE Listed, 2007) [2]. It's also offer very low global warming potential (GWP) of 3 while those of HFCs are 1,300 ~4,400. It's also offer other advantages such as 40~50% reduction in charge, 5~10% increase in energy efficiency, also 'drop-in' capability without requiring major system changes [2].Meyer (2000)[3] studied experimentally the performance of a number of R-22 alternatives in vapor compression experimental setup. The evaporating temperatures ranged between (-20 °C to 20 °C) at condensing temperature of (55 °C). R-134a revealed highest coefficient of performance, although the differences in cooling COP between R-134a, and R-22 was on average less than 2%. It is followed by R-404A which has cooling COP approximately 12% lower than that of R-134a for this high condensing temperature condition. The cooling COP of R-407C was the lowest, on average 17% below that of R-134a. INEOS FLOUR (2005)[4] company presented a comparison of data of R407A supermarket with the other similar plant R22.The experimental results showed that the cooling capacity is improved by (26%)when using R407A also C.O.P is enhanced by (18%). L.D. Yang(2012)[5] presented original R22 wall room air conditioner with a cooling capacity of 2.4 kW and energy efficiency ratio (EER) of 3.2 is retrofitted with a compressor of a 20% larger displacement to charge R290 and R1270 for performance experiments. The results show that for R1270, only adopting a same kind mineral lubricant of higher viscosity would supply 2.4% higher cooling capacity and 0.8% higher EER than those of the original R22 system under normal condition, and for R290, adopting the larger displacement compressor simultaneously would also obtain better performance. Alternative systems all have higher increase rate and greater increment in both cooling capacity and EER than the original R22 system when outdoor temperature decreases. The R1270 system has great increase in cooling capacity and negligible decrease in EER. Refrigerant charge distribution is also investigated and it indicates that the charge with in both heat exchangers and compressor ought to be reduced.

Cycle Performance

In a window type Air-Conditioner system, the refrigerant undergoes phase change from liquid to vapor and then from vapor to liquid in a closed cycle absorbing the heat in the evaporator and rejecting it at the condenser. Figure (1) shows Circuit diagram of a window type air-conditioning system. For the evaporating process between inlet (point 4) and outlet (point 1) evaporator, according to the steady flow energy equation [6],

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

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

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

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 Win, Btu / lb (kJ / kg), is given as

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

The consumed power (Wc, kJ/s) is calculated from the knowledge of the current (I), voltage (V) and the power factor (*COS Ө)* [7]

$$Wc = I\ V\ COS\ \Theta \qquad ...(4)$$

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

$$-q = h_2 - h_3 \qquad ..(5)$$

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

$$h_3 + 0 = h_4 + 0$$

$$\text{Or} \quad h_3 = h_4 \qquad ...(6)$$

Coefficient of cycle performance is defined as the ratio of the refrigerant effect to the work required to produce it [8]

$$COP = \frac{q_{rf}}{w_{in}} = \frac{h_1 - h_4}{h_2 - h_1} \qquad ...(7)$$

The refrigerant circulate rate $\dot{m}$, lb/h (kg/ s) is calculated from the know cooling capacity Q_{rf} , Btu /h (W) or condenser capacity Q_{con} , Btu /h (W) , as below

$$\dot{m} = \frac{Q_{con}}{q_{con}} = \frac{Q_{rf}}{q_{rf}} \quad ..(8)$$

Experimental Apparatus

The experimental procedures are as follows. The performance of the original R22 air conditioner with 1300 g charge is recorded under normal condition. In the second stage, after accomplishing the experiments with R22, the original air conditioner was removed out of R22, and then charged with R432a. The quantity of refrigerant is weighted by a sensitive balance. The performance of the air conditioner is first assessed at different charges to determine the optimum charge, and then assessed at the optimum charge at different ambient temperatures. A Mitsubishi trade mark of window type Air-Conditioner cooling unit is selected to be as a test rig. The specifications of the experimental system are given in Table (2). Figure (2) show photographs of the unit, and manifests the instrumentation and measurement tools. Three temperature gauges were installed on the refrigerant 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 side and one low pressure side. These where manufactured by HEIZUNGWILDMANN with a temperature range of (0 to 120 ºC) at a division of (2º). The Bourdon tube gauge namely "AIRMINDER". It 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 (0 to 500psi), the division (5 psi), and two pressure gauges installed on the low pressure side having range (-30 to 220psi), the division (2 psi). The pressure gauge is filled with oil to prevent pointer vibration.. The energy consumption of the refrigeration system was measured with DT266F clamp meter to measure both the current and voltage. All temperature and pressure gauges used for experiments were calibrated in the Central Institution for Standardization and Specify Control with error ± 0.4 % for temperature gauge and error ±0.3 % for pressure gauge.

RESULTS AND DISCUSSIONS

Figures (3) to (10) present the experimental results with various charges for R432A compared with R22 to get the best charge for R432A at different ambient temperatures. Figure (3) shows the power consumption at various charges for R432A compared with R22 at (Tair=30 ºC). This figure show the power consumption between (2322W-2511W), while the power consumption between (2421 W-2612W) at (Tair=34 ºC), and (2499 W-2697W) at (Tair=38 ºC) as shown in figure (5) and figure (7), respectively. Because the refrigerant R432A has a significantly lower discharge temperature than the refrigerant R22. Decreasing discharge temperature leads to decrease in the power consumption. The advantage of a lower discharge temperature is that there will be less strain on the compressor and hence a longer compressor life. Figure (4) shows the coefficient of performance at various charges for R432A compared with R22 at (Tair=30 ^{o}C). This figure shows the coefficient of performance between (2.93-3.25) , while the coefficient of performance between (2.73-2.98) at (Tair=34 ^{o}C), and (2.55-2.8) at (Tair=38 ^{o}C) as seen in figure (6) and figure (8), respectively. Figure (9) and figure (10) show the power consumption and coefficient of performance for R432A at different ambient temperatures. Increasing ambient temperature leads to increase in the power consumed and decrease coefficient of performance. It can be seen that the best charge for R432A is 750 g since it gives the highest coefficient of performance and less energy consumption. When charging the system with 750 grams, the amount of charge rate reduced to 42.3% compared with the original gas. The percentage increase in coefficient of performance are 9.85%, 8.39%, and 8.93% and the proportion of low energy consuming are 7.53%, 7.31%, and 7.34% at ambient temperatures of 30 ^{o}C,34 ^{o}C and 38 ^{o}C respectively for R432A compared to R22.

CONCLUSIONS

The refrigerant R432A is presented as an alternative refrigerant to R22 for window type air conditioner of 2 TR capacity systems in this paper. The following conclusions can be drawn based on the results obtained:

- The amount of charge for R432A is 42.3% lower than that of R22.
- The best charge is 750 g for R432A compared with 1300 g for R22.
- The power consumption of R432A is 7.31-7.53 % higher than that of R22
- The COP of R432A is 8.39-9.85 % higher than that of R22.

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Table 1: Physical Properties (ASHRAE Listed, 2007)

ITEM	ODP	GWP	Toxicity	Flammability	Boiling point °C	Liquid density(kg/m^3)	Molecular weight(Kg/Kmol)
R432A	0	3	N	Y	-46.6	496	42.8
R22	0.05	1700	N	N	-40.8	1129	86.5

Table 2: Specifications of the experimental system

specifications of the experimental system	
Unit	2 ton model WRC-1801K3SA
Evaporator	(42.8×42.4×8.2) cm
Condenser	(60×42.4×8.2) cm
Compressor	reciprocating (Mitsubishi co., model JAH5522E-RE68295A)

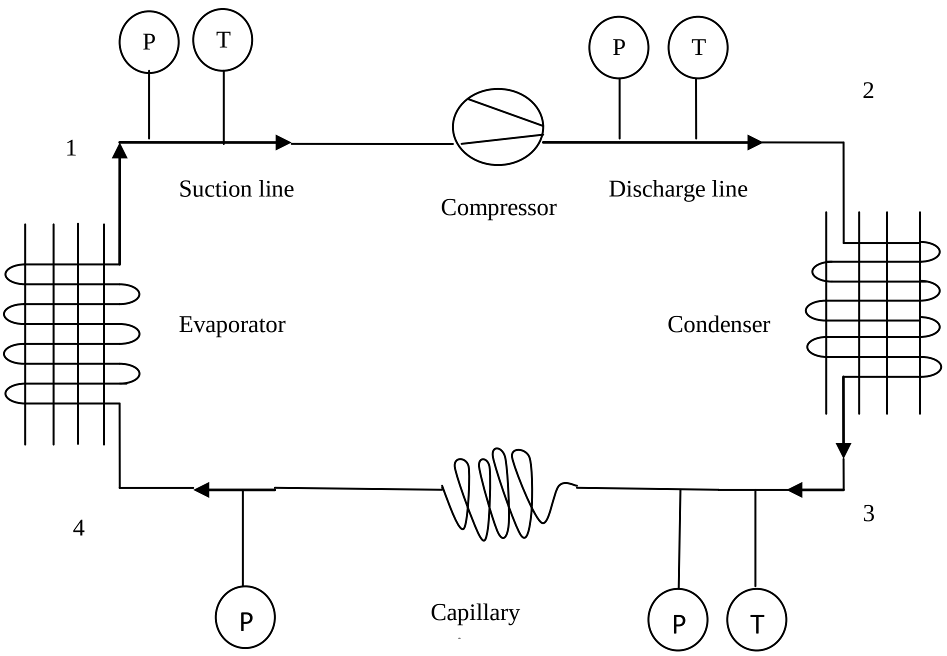
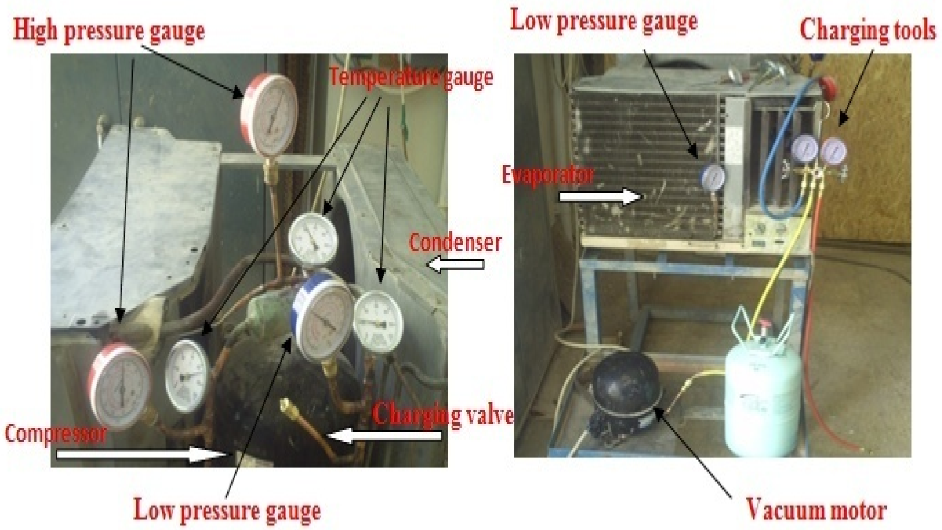


Figure (1): Schematic of a window type Air-Conditioner cycle.



(a)The measurement devices installation (b) The Test Device

Figure (2): Photograph of test apparatus.

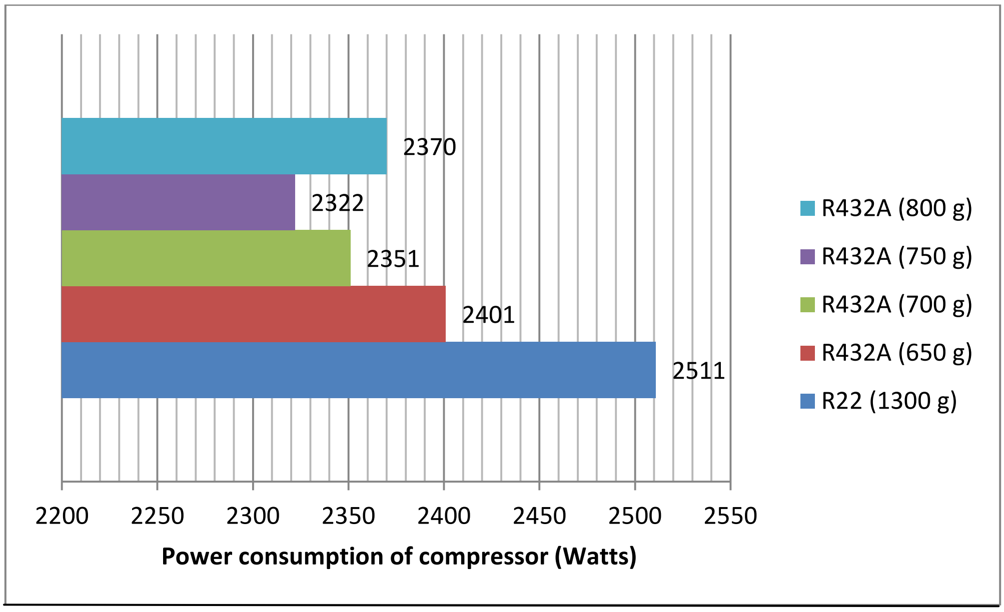


Figure (3): Power consumption at various charges for R432A compared to R22 at Tair=30 °C

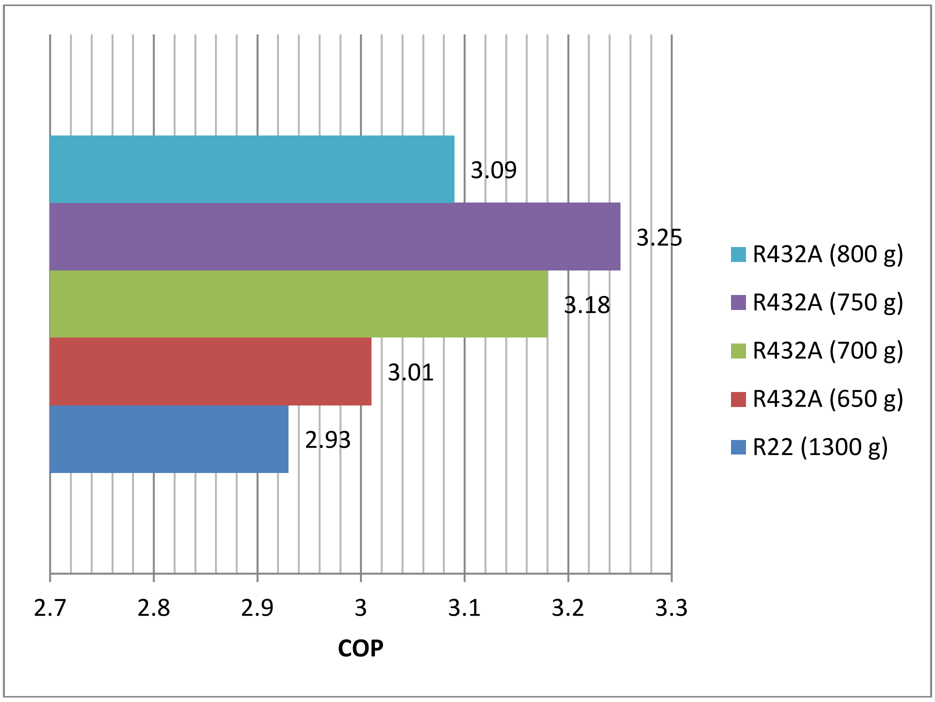


Figure (4): COP at various charges for R432A compared with R22 at Tair=30 °C

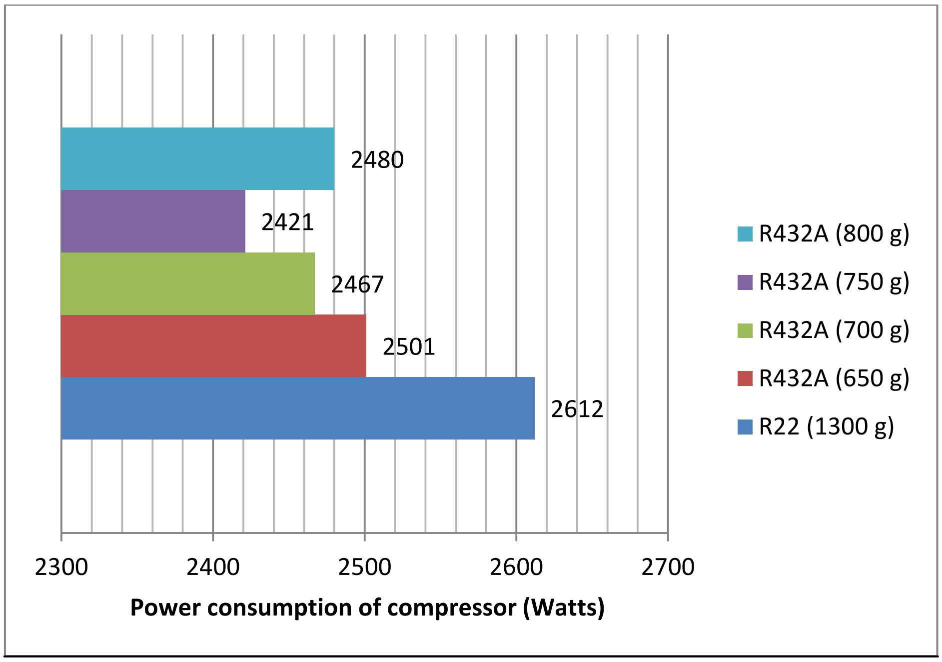


Figure (5): Power consumption at various charges for R432A compared with R22 at Tair=34 °C

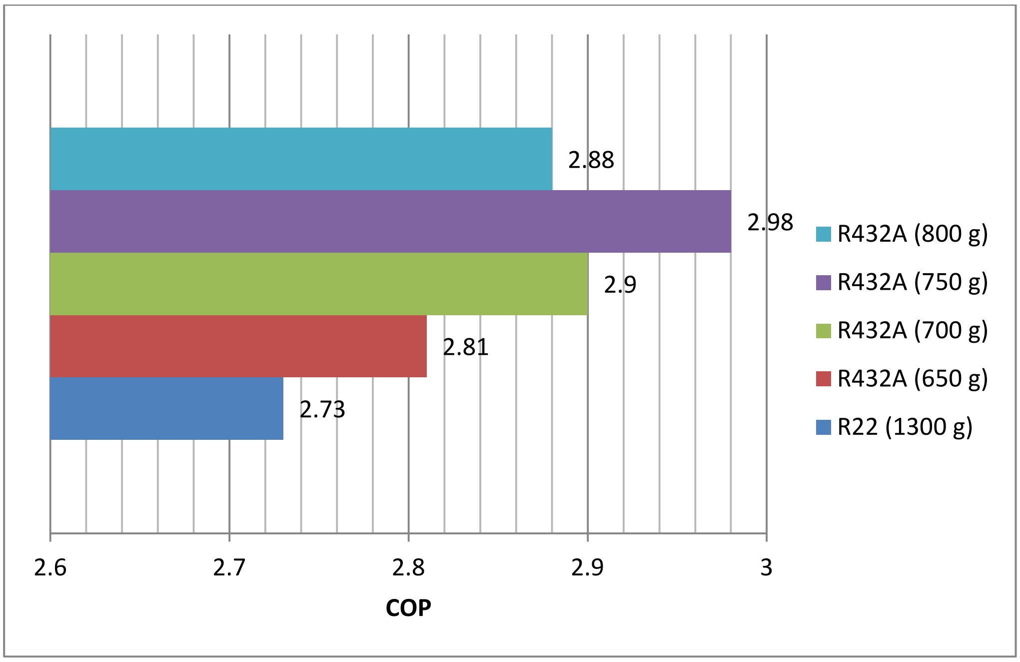


Figure (6): COP at various charges for R432A compared with R22 at Tair=34 °C

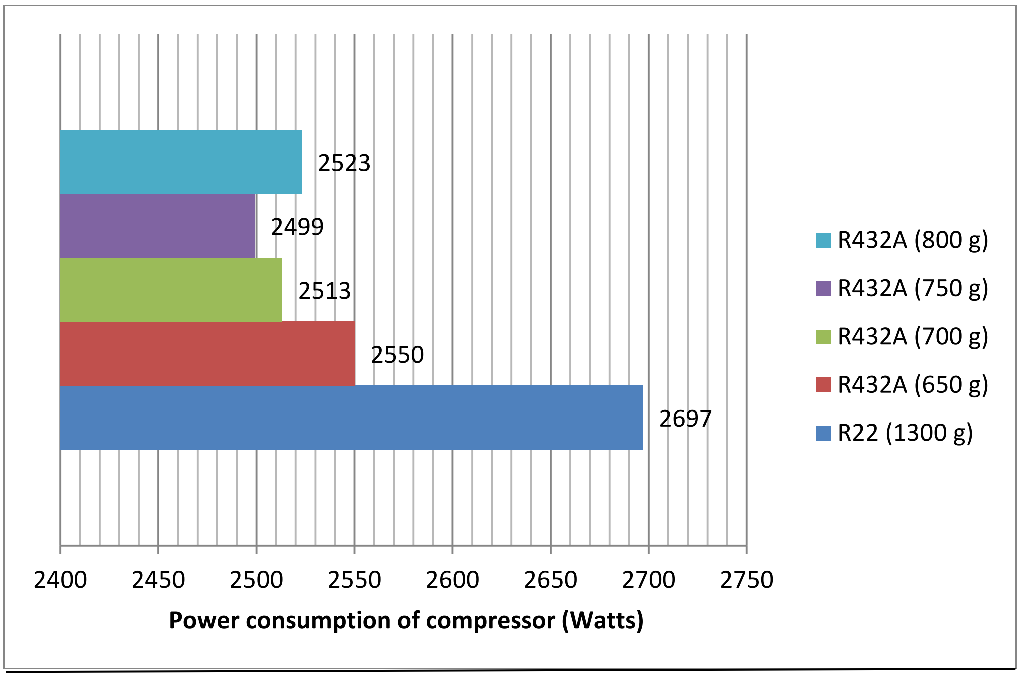


Figure (7): Power consumption at various charges for R432A compared with R22 at Tair=38 °C

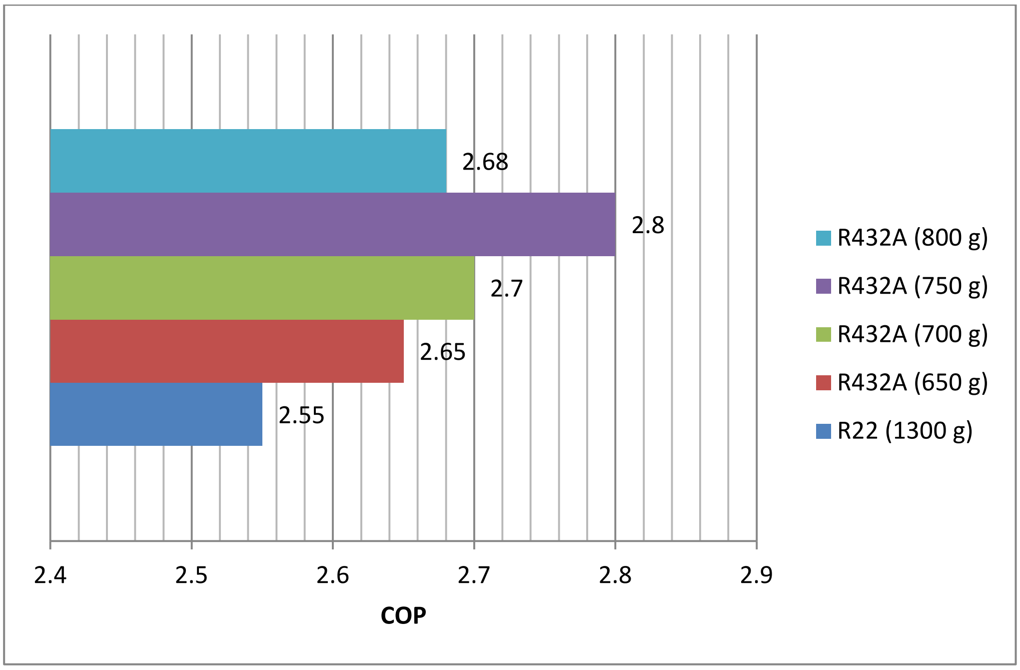


Figure (8): COP at various charges for R432A compared with R22 at Tair = 38°C

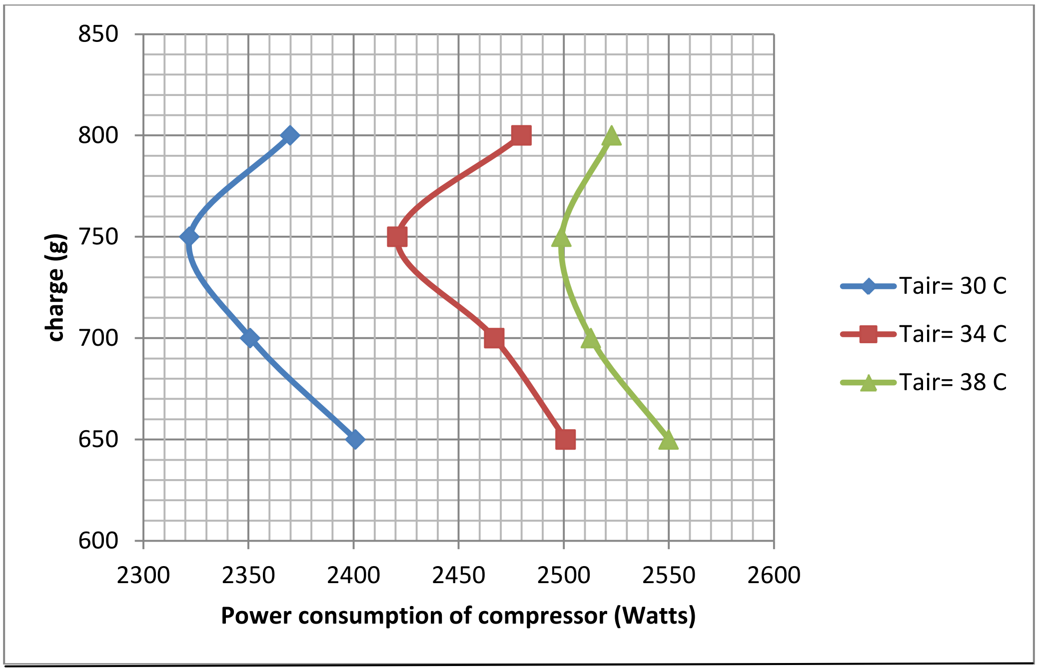


Figure (9): Power consumption at various charges for R432A with different ambient temperatures .

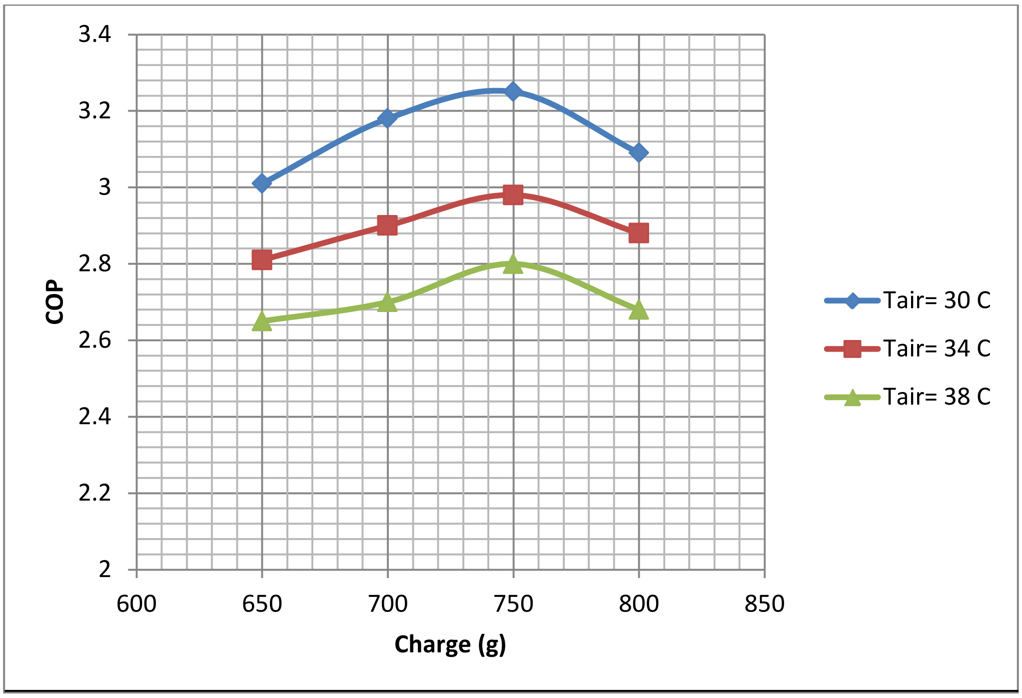


Figure (10): COP at various charges for R432A with different ambient temperatures.