Performance Study of a Two Ton A/C Split Unit
Using (R-407C)

By
Firas Mahmoud Rashied Makahleh



Supervisor
Professor Mahmoud Hammad

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| Examination committee | Signature |
| --- | --- |
| Prof. Mahmoud Hammad, Chairman. | ........... |
| Prof. Mohamed AL-Saad, Member. | ........... |
| Prof. Mahmoud AL-Husein, Member, External Examiner. | ........... |
| Dr. Ali Badran, Member. | ........... |

## DEDICATION

To my mother and father who gave me love and hope, to my wife who helped and encouraged me to complete this little work, to my children, brothers and sisters.

# Acknowledgements

I would like to thank my supervisor professor Mahmoud Hammad for his valuable encouragement and help to complete this work.

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# ABSTRACT

## Performance Study of a Two Ton A/C Split Unit Using (R-407C)

**By**
**Firas Mahmoud Rashied Makahleh**

**Supervisor**
**Profesor Mahmoud Hammad**

The main purpose of this research is to study the performance of 2 ton split air conditioning unit operating with R-407C as a replacement for R-22. Since Montreal protocol was signed in September 16, 1987, scientists, began a hard work to find a suitable replacement for CFC's, HCFC's and Hallons due to their high Ozone Depletion Potential (ODP). Different refrigerants and mixtures of refrigerants were tested in different applications such as refrigeration units and air conditioning units. A blend of refrigerants called R-407C (R-32/R125/R-134a) seems to be the more attractive replacement to R-22 and R-502 in air conditioning systems.

The investigated refrigerant R-407C is an alternative to R-22 with no Ozone Depletion Potential (ODP=0), low Global Warming Potential (GWP), low toxicity and low flammability. Readings for temperatures and pressures at selected points around the cycle were taken at different evaporating and condensing temperatures. The optimum charge was first decided to be 1450 grams, which gave the highest COP of about 3.75.

Tests were carried out at different condensing and evaporating temperatures, performance curves for R-407C were presented and compared with performance curves for R- 22 and R-134a taken by other works for the same unit at same evaporating, condensing and ambient temperatures. Results showed comparable performance of R-407C.

# 1-INTRODUCTION

Refrigerants are the working fluids in refrigerators, air conditioning and heat pumping systems. Refrigerant selection involves compromises between conflicting desirable thermodynamic properties.

The environmental consequences of refrigerant that leaks from system must be considered when selecting suitable refrigerants, since some refrigerants such as chlorofluoro carbons (CFC's) and hydrochlorofloro carbons (HCFC's) and hallons molecules break down and release chlorine and bromine atoms which destroys the ozone layer around the earth (ozone depletion). Montreal protocol of 1987, limited the production of CFC's, HCFC's and hallons to 0.5% by January 1, 2020.

For the above reason scientists looked forward to have alternatives with zero ozone depletion potential and low global warming potential.

As a replacement to R-22 it was found that R-407C with the above properties encourage the scientist focussing on it in air conditioning systems.

Refrigerant R-407C is a chlorine-free compound, which gives a good performance when, compared with R-22 with minimum extra cost and minimum design change.

In this work, the proposed refrigerant R-407C performance is tested in an air conditioning split unit designed originally to work with R-22 after changing the lubricant oil with polyester oil to suit the use of R-134a and R-407C.

The tests include studying the performance of R-407C at different condensing and evaporating temperatures and then a comparison with R-22 and R-134a will be presented.

# 2-LITERATURE SURVEY

Montreal protocol of 1987 aimed at complete cessation of the production of CFC's, HCFC's and hallons due to its ozone depletion potential and its contribution to the warming of the earth. Since that time researchers and specialists made tremendous efforts to find suitable replacements to CFC's and HCFC's with minimum design changes of the existing units.

An insight on the most recent researches in this field will be made here.

Ferrari, *et al.*, (1994), examined theoretically R-407C which is a mixture of (52% R-134a, 25% R-125 and 23% R-32) as a replacement for R-22. They showed that evaporator heat ratio which is defined as, the refrigeration effect for R-407C to that for R-22, at 40°C condenser temperature with no superheating or subcooling varied from 82% to 98% as the evaporating temperature varied from -40°C to 10°C, while compressor work rate varied from 85% to 98% and coefficient of performance (COP) varied from 95% to 100 % at the same variation of the evaporating temperatures.

They studied the effect of ester oil on R-22 and concluded that the use of polyol ester lubricant will reduce the over all system performance due to high solubility of the refrigerant in ester oil. They also showed that for air conditioning applications presently using R-22 (at 10°C evaporator outlet temperature) the weight mixture of R-407C (52%/25%/23%) changed to (51.7%, 25.1%, 23.2) due to the refrigerant solubility in the oil which lead to drop in COP of about 3%.

Yajima, *et al.*, (1994), studied the performance of some refrigerants and mixtures experimentally and using simulation on 0.8 ton split air conditioning unit, they showed that the COP using R-407C reached about 87% of that using R-22, while R-134a gave only 60% of the R-22 COP. Optimum refrigerant charge is approximately the same for both R-407C, R-22 and R-134a.

They also showed that the power consumed by R-407C is higher than that for R-22 and R-134a, in both heating and cooling modes.

Muir, (1994), studied the performance of R-134a, R-407 C and R-410A as alternative refrigerants for R-22. He found that R-134a has the lowest capacity and pressure of the three refrigerants, it would require to redesign all R-22 equipment in order to utilize R-134a efficiently. He conclude that R-134a is the least likely of all alternatives to be utilized in residential and commercial A/C systems and that R-407C is the simplest alternative of them to implement because it requires minimal changes to the current R-22 operating equipment. The only major change required is the use of polyol ester lubricant which is a hydroscopic material (has greater tendency to absorb moisture). He concluded that the COP of R-407C is slightly less than that for R-22 (95% of R-22) and that the cost of the system using R-407C is slightly greater than R-22. He also concluded that if the system uses R-410A then it has about 5% better COP for the same cost compared with other two alternatives.

Xiao feng, *et al.*, (1994), discussed some potential alternatives for R-22 such as R-134a, propane, R-32/ R-125, R-33/ R-152a, R-32/ R-134a, R-32/ R-125/ R-134a, R-125/ R-143a/ R-134a and R-152a/ R-134a/ R-32 by calculating their coefficients of performance and volumetric capacities. They concluded that for R-134a, and propane the compressor volume must be increased compared with R-22 with the same cooling capacity. They concluded also that ternary mixtures including R-407C are comparable to R-22 regarding both COP, volumetric capacity, and they found that (R-32/ R-152a) blend has a higher volumetric capacity but a lower COP than R-22.

temperature, 11.1°C superheating temperature from the dew point and the subcooling of 8.3°C from mean condensing temperature.

Luzzatto, *et al.*, (1994), presented the efforts carried out in Delchi Carrier, Vinasanta plant on application of R-134a and R-407C as substitute for R-22 on portable room air conditioning system. They showed that for R-134a, the compressor displacement should be 50% higher than that for R-22 and that lubricant oil, expansion device and filter should be changed while by using R-407C the COP reach 91% of R-22 system without any modification. They made a cost comparison between the systems using R-134a and R-407C. They showed that the system costs would increase 2% and 7% by using R-407C and R-134a respectively higher than the same system working with R-22.

Tarawnah, (1996), studied the performance of an A/C split unit by using mixtures of propane, butane and iso butane. Also, he used a computer algorithm based on Aspen library to calculate the enthalpy values of the mixture. He found that, increasing the propane ratio in the mixture would increase its coefficient of performance to a value greater than that of R-22.

The Association of European Refrigeration Compressor Manufacturers (ASERCOM), (1997), conducted several experiments on the use of R-407C to replace R-22. The study was carried out to evaluate the Coefficient of Performance (COP) using R-407C at different evaporating and condensing temperatures. They summarized the advantages of R-407C as non-flammable, non-toxic, available in the market, has lower discharge temperature than R-22 and a large subcooling effect.

The German compressor manufacturing company BITZER, (1998) demonstrated that R-407C has similar thermodynamic properties and performance as R-22 in the air conditioning systems within medium temperature cooling range. It was therefore preferred over the available alternatives. They showed however that the use of R-407C is not recommended for plants with flooded evaporators, since a large concentration shift/layer formation is to be expected in this type of evaporators.

The German compressor manufacturing company BITZER, (1999) showed that R-407C is preferred when compared to all other available alternatives of R-22, since it shows similar performance of R-22. They also showed that the distinctive temperature glide requires a special design for the main system components, e.g. evaporator, condenser and expansion valve.

Douglass, *et al.*, (1999) used a computer model to evaluate the performance of several leading R-22 alternatives for window air conditioners. They showed that R-407C had an optimal cost that is identical to that of R-22.

Burke, *et al.*, (1994), studied the oil behavior of the R-407C (R-32, R-125, R-134a) mixture, They found that (R-32, R-125, R-134a) mixture is miscible with ester-type oil in the temperature range from 80°C down to –70°C.

Obeidat, (2000), studied the performance of (A/C) split unit by using R-134a. He found that R-134a gives lower COP at different conditions compared to R-22, and so he concluded that R-134a is not a suitable replacement for R-22.

From previous works, all studies show that the ternary mixture R-407C is a good replacement for R-22 when the refrigeration capacity and the coefficient of performance were considered. In this work, many parameters will be studied over that in the previous works such as the electric power consumption, work of compression, isentropic efficiency, pressure ratio and discharge temperature.

# 3-COMPARATIVE STUDY

## 3.1 Introduction:

Thermodynamically, physical and chemical properties should be considered for any refrigerant under study for specific application. Ozone Depletion Potential (ODP) and Global Warming Potential (GWP) are also the most criteria to be considered. Among these are the followings:

## 3.2 Thermodynamic parameters:

### 3.2.1 Refrigeration effect:

High refrigeration effect per unit mass is preferable, since this will lead to higher refrigeration capacity. This parameter .is considered in combination with the work of compression.

### 3.2.2 Pressures:

Low condensing pressure and high evaporating pressure improve the coefficient of performance, therefore both are preferable.

### 3.2.3 Transport properties:

Thermal conductivity and viscosity affect the performance of the heat exchangers and piping. High thermal conductivity and low viscosity is desirable.

### 3.2.4 Temperature glide:

For zeotropic refrigerant blends, the difference between the bubble temperature and dew point temperature, affects the heat exchange in the condenser and the evaporator due to the decrease in temperature in the condenser and an increase in temperature in the evaporators and therefore effect the overall performance.

### 3.2.5 Freezing temperature:

The refrigerant should have low freezing temperature to prevent solidification during normal operating conditions.

### 3.2.6 Critical temperature:

The refrigerant should have high critical temperature to achieve low work of compression.

### 3.2.7 Volume flow rate per unit refrigeration capacity:

This property influences the compression rate or the type of compressor. Medium density refrigerant is desirable, since low density refrigerant requires high volume flow rate while high-density refrigerant requires high power consumption.

### 3.2.8 Coefficient of Performance:

COP is the mostly used criteria in comparing several refrigerants working at the same condensing and evaporating temperatures.

### 3.3 physical and chemical properties:

#### 3.3.1 Miscibility with oil:

Highly soluble refrigerants with oil are more desirable, since this will assure good heat transfer and good return of oil to the compressor.

#### 3.3.2 Water solubility:

Non soluble refrigerants in water are preferred since solubility leads to component damage when freezing.

### 3.4 Safety:

#### 3.4.1 Toxicity:

Low toxicity refrigerants are the least hazardous refrigerants.

#### 3.4.2 Flammability:



The good refrigerant is the non-flammable one.

### 3.5 Effect on the environment:

The effect on the environment includes the effect on the ozone layer. The refrigerant should have very low Ozone Depletion Potential (ODP) and low Global Warming Potential (GWP).

## 3.6 Comparison between R-22, R-134a and R-407C.

**Table (3.1) physical and chemical properties of 22, R-134a and R-407C (BITZER 1998,1999).**

| REFRIGERANT | R-22 | R-134a | R-407C |
|---|---|---|---|
| Composition | CHCLF2 | CH2F-CF3 | R32/R125/R134a 23/25/52% |
| Boiling temperature (° C) (1.031bar) | -41 | -26 | -44 |
| Temperature glide[1] (° K) | 0 | 0 | 7.4 |
| Density of liquid (55 ° C) kg/dm [3] | 1.06 | 1.15 | 0.99 |
| Vaporpressure[3] (7/55°C) bar | 6.2/21.6 | 3.75/14.93 | 6.2/23.4 |
| Critical temperature (° C) | 96 | 101 | 87 |
| Flammability | - | - | - |
| Toxicity (ppm) | 1000 | 1000 | 1000 |
| Ozone depletion potential ODP[2] | 0.05 | 0 | 0 |
| Global warming potential (GWP)[2] | 1700 | 1300 | 1610 |
| Specific heat (kJ/kg.K) liquid | 1.202 | 1.367 | 1.426 |
| Specific heat (kJ/kg.K) vapor | 0.792 | 0.93 | 0.997 |
| Lubricant | Mineral oil | Polyol ester | Polyol ester |

[1] Total glide from bubble to dew line

[2] ODP of R11 = 1.0

GWP of CO2 = 1.0 ( 100 a /IPCC1994)

[3] Evaporating and condensing temperatures for R-407C are mean values

### 3.6.1 physical and chemical properties:

**1. Freezing point:**

The three refrigerants have low freezing point (less than -90° C).

**2. Boiling point**

In air conditioning systems, low boiling point is desirable, and this is available for the three refrigerants under study as shown in table (3.1).

**3. Critical temperature:**

The condenser temperature must be kept below the critical temperature (87°C for R-407C) which is high enough for ordinary application.

**4. Specific heat:**

low specific heat of liquid and high specific heat of vapor will increase the subcooling state of liquid and decrease the superheating temperature, that will increase the refrigeration effect. But as shown in table (3.1) the specific heat of R-407C is higher than that of R-22 and R-134a in both liquid and vapor states.

**5. Temperature Glide:**

The high temperature glide is a disadvantage for normal applications. It can have negative influence upon the capacity of the heat exchanger.
The distinctive temperature glide requires a special design for main system components such as evaporator, condenser and expansion valve.
As shown in table (3.1) R-22 and R-134a has no temperature glide while R-407C has a high temperature glide of 7.4° K where needs some modification to offset this effect.

**6. Miscibility with oil:**

Immiscible oil can settle out in the heat exchanger and prevent heat transfer to such an extent that the plant can no longer be operated.
Table (3.1) shows the suitable lubricant for each refrigerant. It was shown that polyol ester is suitable to be used with R-134a and R-407C due to a good solubility with them, while mineral oil is suitable only for R-22.

**7. Specific volume of suction gas:**

R-134a has higher specific volume for gas at suction conditions than those for R-22 and R-407C and so larger compressor volume must be used. R-22 and R-407C have approximately the same value and therefore the same compressor can be used for both refrigerants.

**8. Toxicity:**

Table (3.1) shows that the toxicity for the three refrigerants is the same (1000 ppm).

**9. Flammability:**

Table (3.1) shows that the three refrigerants are inflammable.

**10. Ozone Depletion Potential:**

Table (3.1) shows that R-22 has low Ozone Depletion Potential (ODP = 0.05) while R-134a and R-407C have zero ozone depletion potential (ODP= 0).

**11. Global Warming Potential:**

Table (3.1) shows an approximate equal range of (GWP) for the three refrigerants, with slightly low value for R-134a.

### 3.6.2 Thermodynamic parameters:

**Table(3.2)Comparison of the thermodynamic characteristics for R-22, R-134a and R-407C (BITZER 1998,1999).**

| Refrigerant | R-22 | R-134a | R-407C |
|---|---|---|---|
| Application Range | -25° C to 13°C | -20°C to 22° C | -22°C to 13° C |
| Vapor pressure (7/55°C) bar | 6.2/21.6 | 3.75/14.93 | 6.2/23.4 |
| Refrigeration Capacity[1] (%) | 100 | 65 | 103 |
| COP[2] (%) | 100 | 90 | 97 |

1 At 5° C evaporating temperature and 50° C condensing temperature.

2 At 10° C evaporating temperature and 50° C condensing temperature.

**1. Pressure:**

Table (3.2) shows evaporating and condensing pressures for R-22, R-134a and R-407C at corresponding temperature of (7° C/55° C). As shown from the table, the compression ratio for R-134a is higher than that for the other two refrigerants, and therefore, it will consume higher power than the other refrigerants.

**2. Refrigeration capacity:**

If refrigeration capacity in a typical refrigeration cycle was 100 % using R-22, then same cycle using R-134a gives 65 %, which is very low while that of R-407C gives 103 % which is a highly comparative value.

**3. Coefficient of Performance:**

For typical refrigeration cycle if the COP using R-22 was taken as 1.0, then using R-134a will give a COP of 0.9 of that, and using R-407C will give a COP, which is 0.97.

# 4-Experimental Rig and Experimental Work Procedure

## 4.1 Introduction

The objective of this research is to study the performance of a split air condition unit by replacing R-22 by R-407C. This A/C split unit was manufactured by Petra industries. The unit is supplied with a four-way valve so it works as all year round A/C unit (heat pump). The expansion valve was replaced by manual valve to control the evaporating temperature (Obeidat, 2000).

The specifications of the unit used are:

Trade mark: Petra

Manufactured by: Petra Ind.

Compressor type: Hitachi

Condenser coil type: DX-12FPI-3 row, aluminum finned copper tube.

Evaporator coil type: DX-12FPI-2 row, aluminum finned Copper tube.

## 4.2 A/C Split Unit Components:

Figure (4.1) shows the main components of the experimental test rig used in this study.

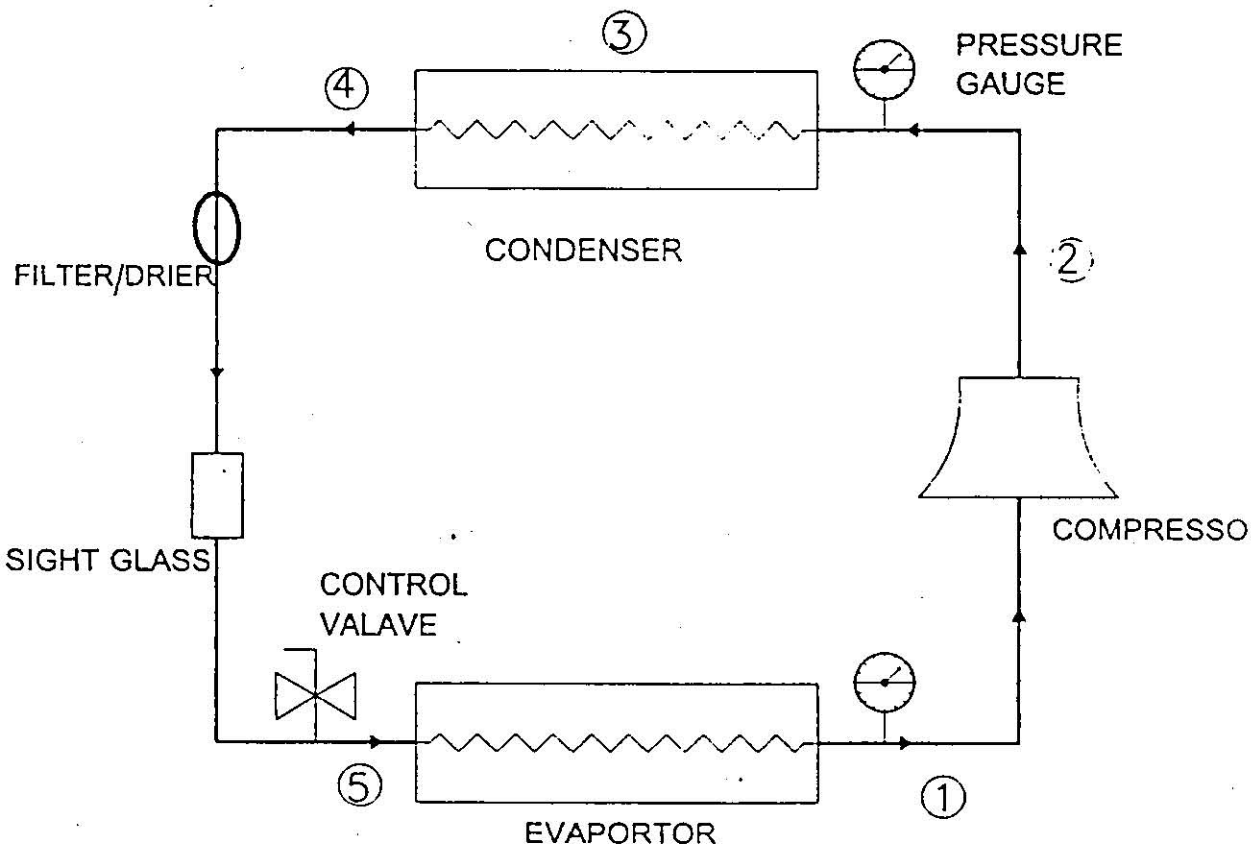



Figure (4.1) Experimental Rig

The A/C split unit consists of the following parts:

### 4.2.1 Compressor:

The compressor used here is of Hitachi type, which has the following specifications:

- Compressor cooling type: Forced air
- Copacity : 2 ton, refrigeration.
- Amps :15.1A

Mechanism: Reciprocating connection rod, 2cylinders.

Bore: 44.5mm.

Stroke: 17.5mm.

Displacement: 54.4cm$^3$/rev.

Oil Charge: 1600cm$^3$

**Specification of the drive motor:**

Permanent split capacitor type, single phase 220-240v, 50Hz, with max current of 58A.

### 4.2.2 Condenser:

The condenser coil is a DX-12 FPI-3row, aluminum finned copper tube.

### 4.2.3 Evaporator:

The evaporator is a DX-12 FPI-2row, which is manufactured from aluminum fined copper tube.

### 4.2.4 Suction Accumulator.

### 4.2.5 Filter/ drier and Strainer.

## 4.3 Experimental Work Procedure:

### 4.3.1 Introduction:

Experiments were conducted for evaporating temperature variations tests by varying the evaporating temperature at approximately constant condensing temperature and for the condensing temperature variations tests by varying the condensing temperature and maintaining the evaporating approximately constant. Both tests were conducted at approximately constant room temperature.

### 4.3.2 Variables to be measured:

During the experiments, several variables were measured.

**1. Temperature:**

The temperature was measured by thermocouples of type 'K'. These thermocouples are connected to a Microprocessor of type (Comark). The thermocouples were fixed in certain points in the system, by a tape, then insulated by (rock wool), to ensure good insulation of the thermocouples from the outside conditions.

Temperatures were measured at the following points:

1-Suction of the compressor.
2- Discharge of the compressor.
3- Inlet of the condenser.
4- Mid point of the condenser.
5- Outlet of the condenser.
6- Inlet of the evaporator.
7- Outlet of the evaporator.
8- The ambient temperature.
9- Outlet of the cooled air from the evaporator.
10- Surface of the evaporator.

**2. Pressure:**

Two pressure gauges were used, one on the suction line of the compressor, which is a low pressure gauge and the other on the discharge line which is a high pressure gauge. The low pressure gauge range is -30 to 250 psi, and the high pressure gauge range is 0 to 500 psi, with minimum reading of 0.5 psi for each.

**3. Electric power consumption:**

The electric power consumed by the compressor, the evaporator fan and the condenser fan was measured by a single-phase watt-hour meter.

The power consumed by the compressor was calculated by subtracting the power consumed by the two fans from the total power consumption. The power consumed by the two fans can be measured, during their operation alone in the beginning. The total power consumption then was measured while the compressor is working, and thereby the actual electric power consumption of the compressor in a unit time is found.

### 4.3.3 The Experimental Work Steps:

The experimental work consists of:

**A. Evacuation and charging process:**

The purpose of the evacuation of the system, is to ensure removing the air from the system, and to ensure also, removing the moisture and the R-134a refrigerant, in order to charge the system with pure R-407C.

The evacuation process was performed by connecting the hose of the gauge manifold between the vacuum pump and the suction line of the compressor, then starting the vacuum pump, continue the evacuation of the unit below the atmospheric pressure. After that, the unit was charged with R-407C beginning with a charge of 1300 grams which is increased 100 grams for each test until 1700 grams is reached.

**Methods used to indicate optimum charge of R-407C:**

**1. Using a Sight Glass:**

During the charging process by observing the flow through a sight a glass until a bubble-less stream of liquid refrigerant is obtained. But, if bubbles are found, this will indicate a shortage of refrigerant.

**2. The formation of dew on the inlet of the evaporator:**

This will indicate a good performance of the system and so, this will be the possible charge for optimum performance.

**3. Calculating the Coefficient Of Performance (COP) at different charges:**

After charging the system with several charges: 1300g, 1400g, 1500g, 1600g, and 1700g, a plot of the coefficient of performance is made for every charge at a given evaporating, condensing and ambient temperature, and from this figure the best charge is found which is corresponding to the highest coefficient of performance. In this work it was approximately 1450 grams.

**B. Charging Process:**

After determining the proper charge of optimum coefficient of performance, the system remained charged with this charge (1450 gram) and the experiments on R-407C by evaporating temperature variations test and condensing temperature variations test are performed. Evaporating temperature was varied by the manual control valve at the inlet to the evaporator while the condensing temperature was varied by using multiple speed fan used with the main unit condensing fan.

# 5-MATHEMATICAL ANALYSIS

## 5.1 Introduction

In this work, various parameters were calculated using measured data by varying evaporating temperature at constant condensing temperatures and varying condensing temperature at constant evaporating temperature for R-407C.

Results were compared with those for R-22 and R-134a taken from previous work in the same unit (Obiedat, 2000).

## 5.2 Parameters under Study:

Followings are the equations used through this work:

### 5.2.1 Refrigeration Effect:

The refrigeration effect is the quantity of heat each unit mass of refrigerant absorbs from the refrigerated space and equals:

$q_{ref} = h_1 - h_4 = h_1 - h_5$ ....................................(5.1)

Where:

$q_{ref}$ :Refrigeration effect (kJ/kg)

$h_1$: is the enthalpy of the saturated vapor at the outlet of the evaporator (kJ/kg).

$h_4$: is the enthalpy of the two phase liquid vapor mixture which enters the evaporator (kJ/kg).

$h_5$: is the enthalpy of the saturated liquid at the outlet of the condenser (kJ/kg).

$h_1$, $h_4$ and $h_5$ are the enthalpies shown in figure (5.1)

### 5.2.2 Isentropic Efficiency:

$\eta_{is} = (h_{2s}-h_1)/h_2-h_1$ ....................................(5.2)

Where

$h_{2S}$: enthalpy of the superheated vapor at the exit of compressor if the compression process is isentropic (kJ/kg).

$h_2$: enthalpy of the superheated vapor at the exit of compressor (kJ/kg).

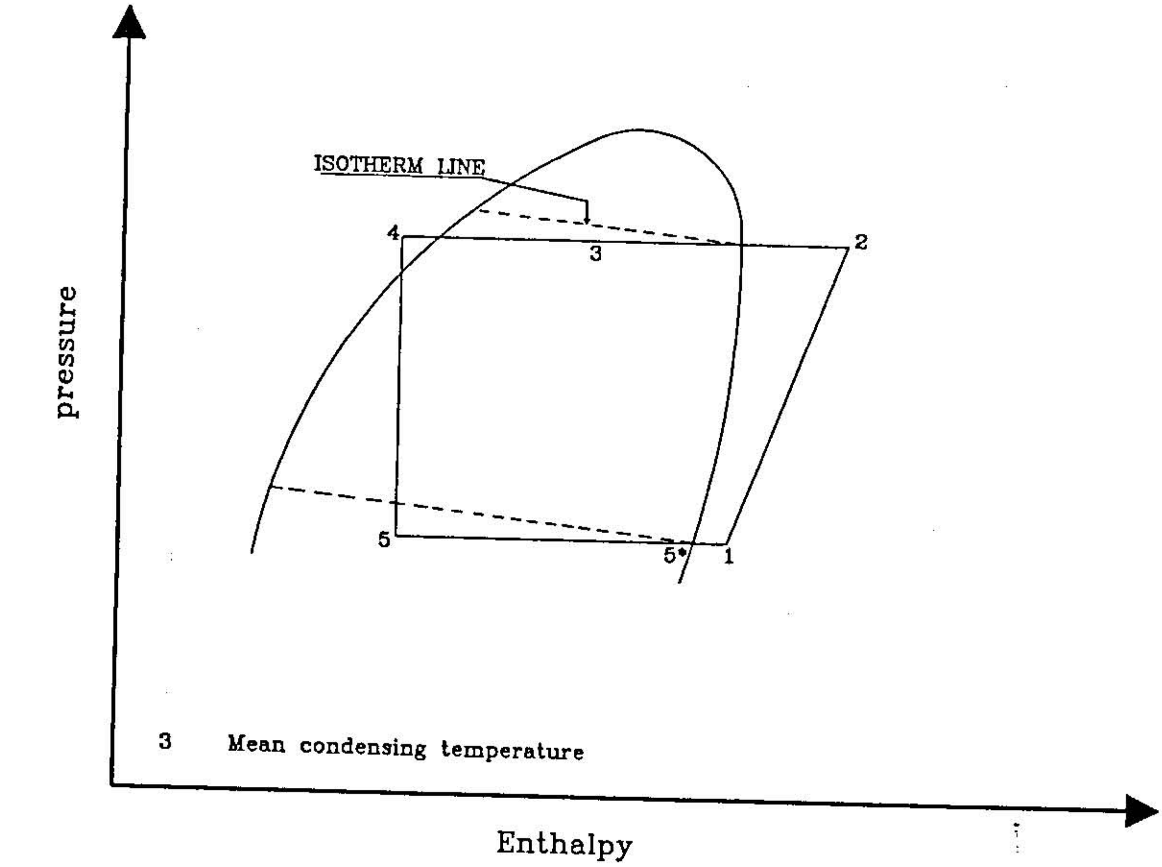



FIGURE (5.1)Pressure Enthalpy Digram for R-407C (BITZER 1998).

### 5.2.3 Mass flow Rate:

The total power consumed by two blowers and compressor was measured by a single-phase watt-hour meter, the power consumed by the two blowers when they are operated in the beginning of the unit work for 3 minutes before the compressor operation, also, was measured.

Assume that, the total power consumption is P1, and power consumed by the blowers is P2, then, the net power consumed by the compressor is (P), and so:

$P = P_1 - P_2$ ..........................................(5.3)

$P = \frac{\dot{m}\,(h_2 - h_1)}{\eta_m}$ ..........................................(5.4)

Where:

$\dot{m}$: mass flow rate (kg/s).

$\eta_m$: mechanical efficiency(0.7)

and so from equations (5.2) and (5.4) it is clearly shown that:

$\dot{m} = (\eta_m * P)/(h_2 - h_1)$............................(5.5)

# 6-RESULTS AND DISCUSSION

## 6.1 Introduction:

Tests were carried out experimentally to evaluate performance parameters by varying evaporating temperature at approximately constant condensing temperature and also by varying condensing temperature at approximately constant evaporating temperature. Both were carried out approximately at constant room temperature of 22 °C.

## 6.2 Optimum Charge:

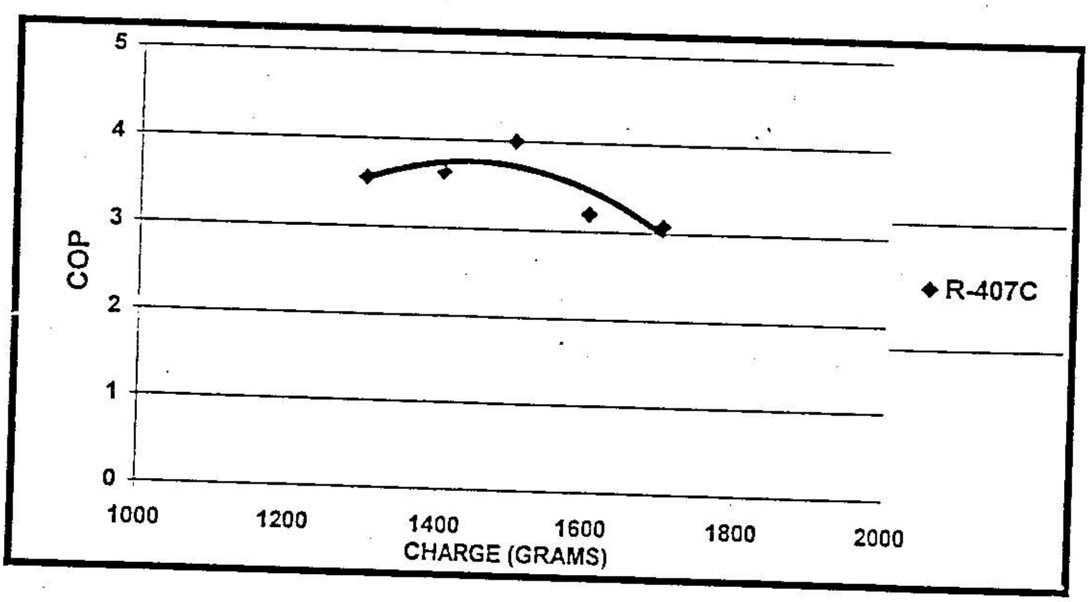



**FIGURE (6.1) COP Vs. Charge at (Tc=43° C) and (Te=5° C).**

The optimum charge can be obtained by evaluating the coefficient of performance at different charges (1700-1900) grams. Fig (6.1) shows that the optimum charge is about 1450 grams which produces a maximum COP of 3.75.

### 6.3 Evaporating Temperature Variation Tests:

Tests were carried out for R-407C at condensing temperature of (43°C) and evaporating temperatures of (5, 6, 8, 10, 11, 12)° C. Several parameters were considered which included the followings:

**1. Refrigeration Effect:**

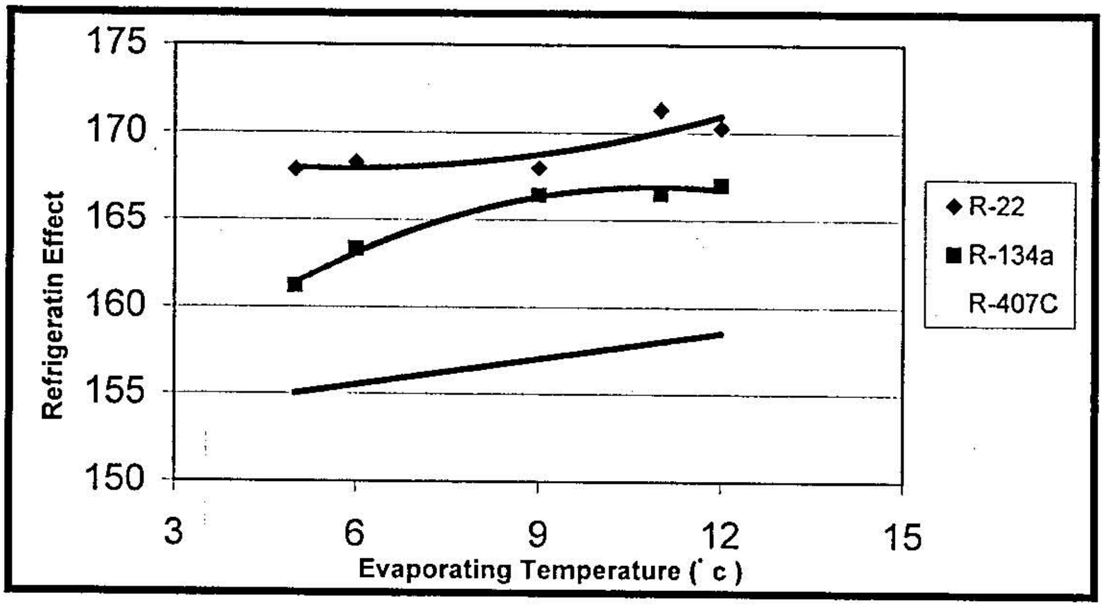



**FIGURE (6.2) Refrigeration Effect Vs. Evaporating Temperature.**

Fig (6.2) shows that for R-407C, the refrigeration effect increase as the evaporating temperature increases, which makes R-407C suitable to be used in moderate temperature systems. The increase is due to the slightly higher enthalpy of saturated vapor at higher evaporating temperatures. Fig (6.2) also shows that R-22 has the greater refrigeration effect among the three refrigerants under study. R-407C reach approximate values (92-93)% of that for R-22.

## 2. Work of Compression:

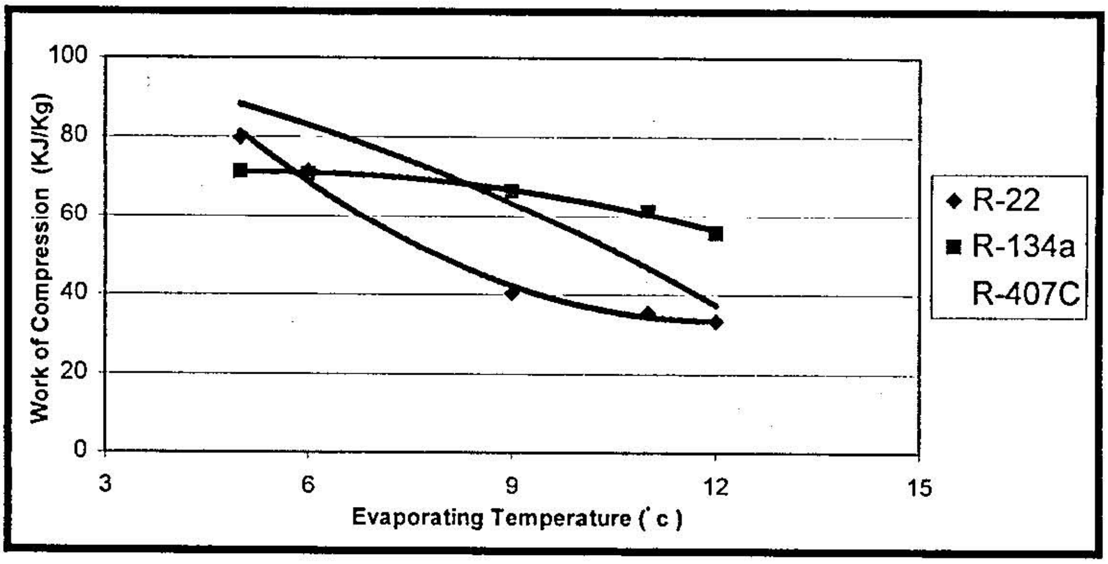



**FIGURE (6.3) Work of Compression Vs. Evaporating Temperature.**

Fig (6.3) shows that, increasing evaporating temperature decreases the work of compression since this will increase the enthalpy at the inlet of the compressor and therefore the work of compression decreases, assuming constant condensing temperature of (43°C). Fig (6.3) shows also that, below 8.5 °C the work of the compression for R-407C and R-134a is higher than for R-22. Above 8.5°C evaporating temperature the work of compression for R-134a becomes the highest among the three refrigerants under study. R-407C work of compression reaches a maximum value of 160% of that for R-22 at 8.5°C evaporating temperature and approximatelly equals that of R-22 at about 12° C evaporating temperature.

### 3. Coefficient of Performance:

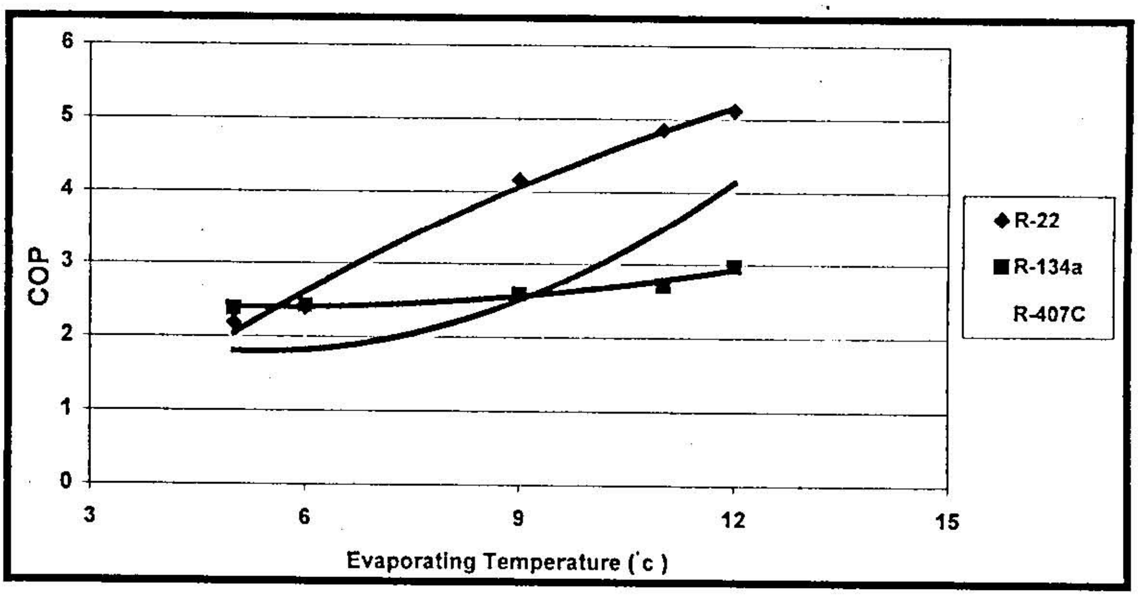



**FIGURE (6.4) COP Vs. Evaporating Temperature.**

Fig (6.4) shows that COP increases when increasing the evaporating temperature due to decrease in work of compression and the increase in refrigeration effect.

Fig (6.4) shows equal COP at low evaporating temperature and then at high evaporating temperature, R-22 has the higher COP than the other two alternatives. Fig (6.4) shows also, that above 9 °C R-407C has higher COP than that of R-134a. Maximum value of R-22 COP reaches a value of 5.1 at 12°C evaporating temperature while maximum COP value for R-407C is approximately 4.1 at the same evaporating temperature.

**4. Mass Flow Rate:**

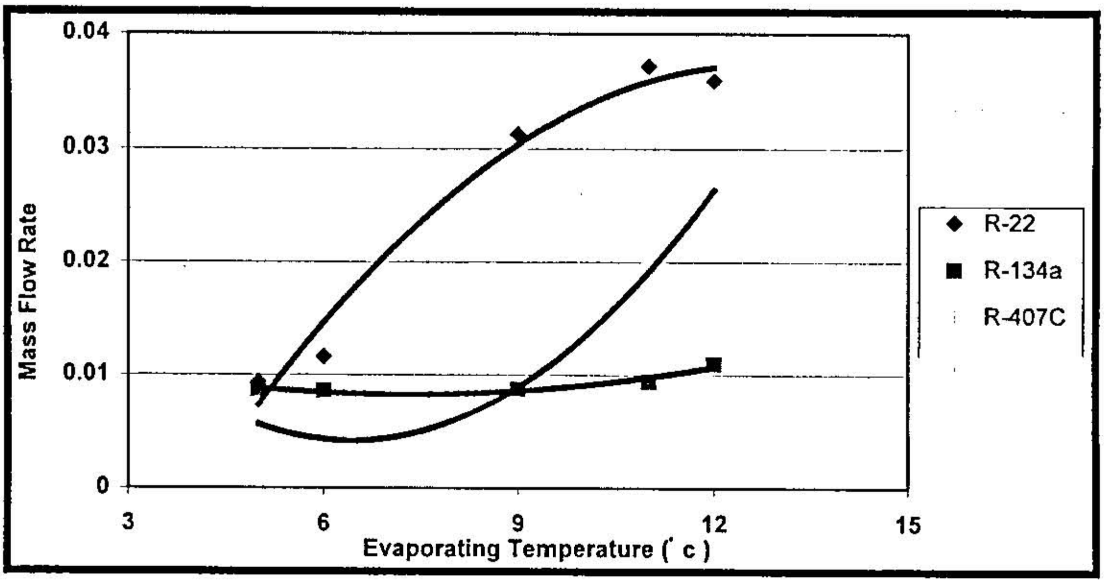



**FIG (6.5) Mass Flow Rate Vs. Evaporating Temperature.**

Fig (6.5) shows that the mass flow rate for the three refrigerants increases as the evaporating temperature increases. As the suction pressure drops, the specific volume of the gas entering the compressor increases, which together with the volumetric efficiency, reduces the mass flow rate at low evaporating temperature and vice versa. Fig (6.5) shows that the mass flow rate for R-22 is higher than those for R-134a and R-407C. the mass flow rate for R407C reaches a maximum value of 82% at 12° C of that of R-22.

## 5. Refrigeration Capacity:

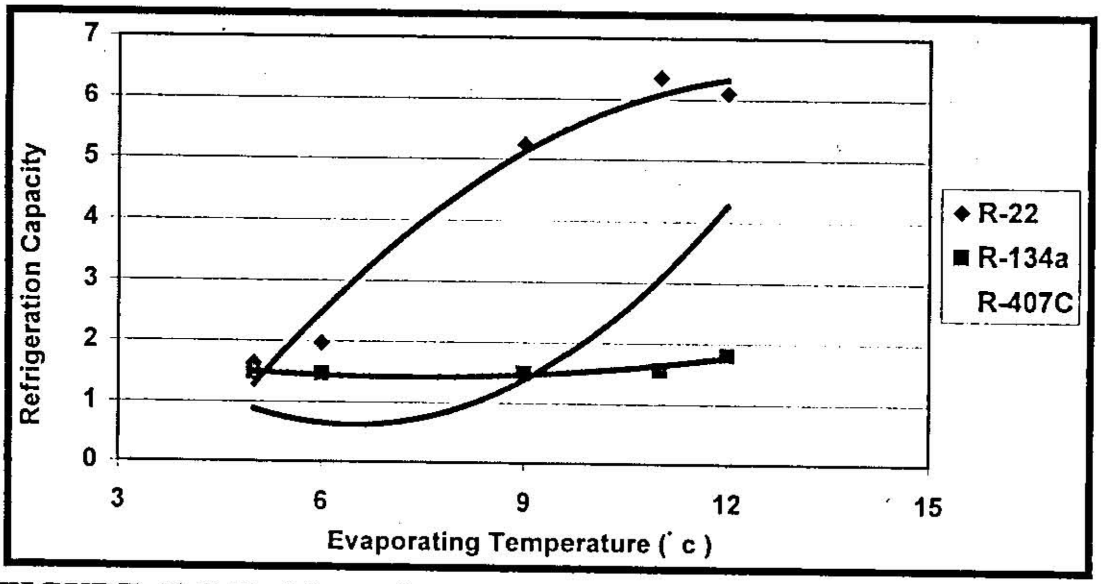



**FIGURE (6.6) Refrigeration Capacity Vs. Evaporating Temperature.**

Fig (6.6) shows that the refrigeration capacity increase with increasing evaporating temperature due to the increasing in both refrigeration effect and the mass flow rate. Fig (6.6) also shows that the refrigeration capacity of R-22 is higher than of R-407C and R-134a.

The refrigeration capacity for R-407C reaches a maximum value at 12°C evaporating temperature and equal (70 %) of that of R-22.

## 7. Heat Rejection:

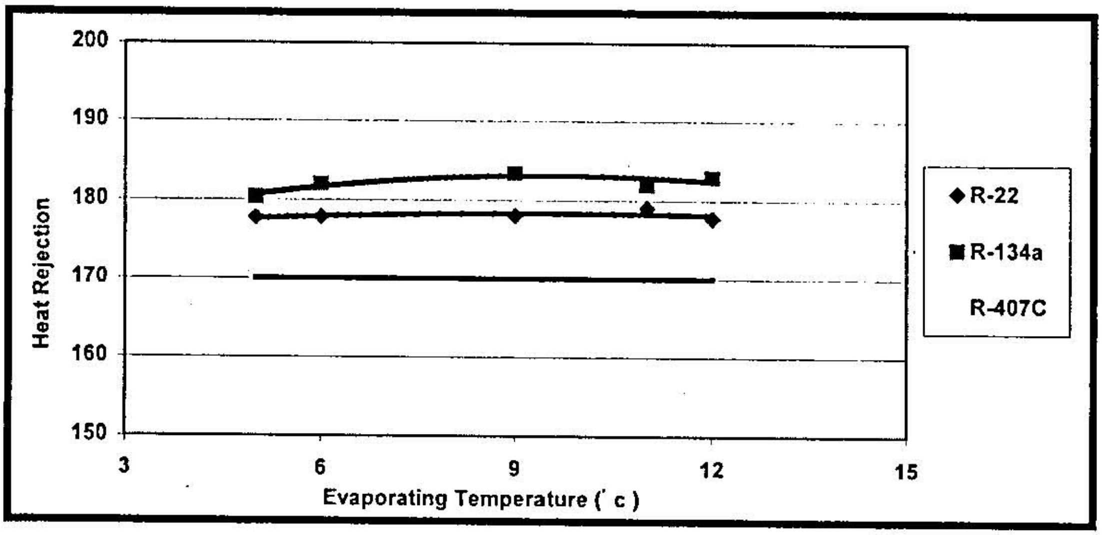
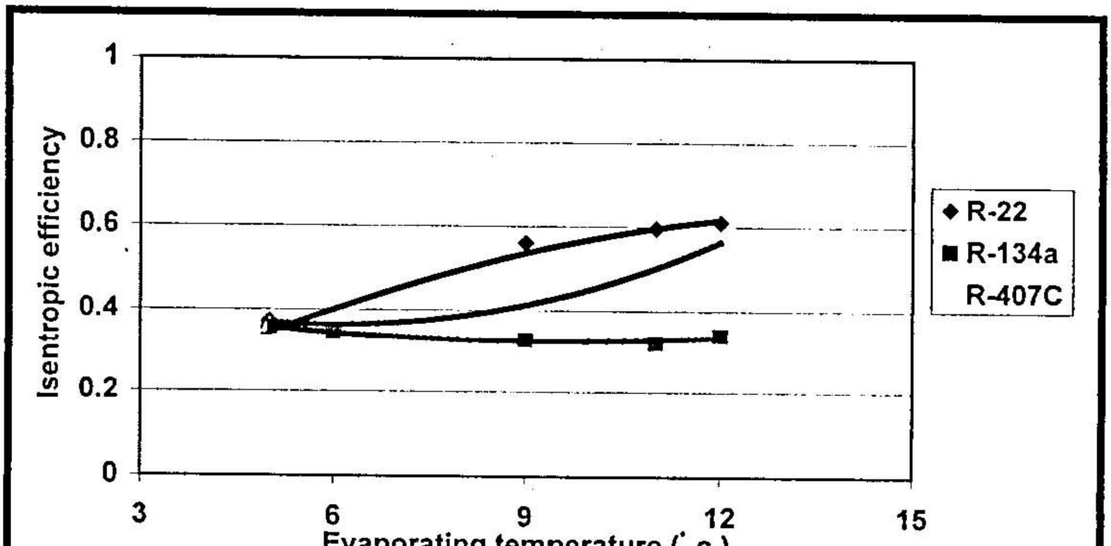



**FIGURE (6.8) Heat Rejection Vs. Evaporating Temperature.**

Fig (6.8) shows that R-407C has a lower value than both for R-22 and R-134a due to the temperature glide which decreases the heat exchange during condensation.

## 8. Isentropic Efficiency:




**FIGURE(6.9) Isentropic Efficiency Vs. Evaporating Temperature.**

Fig (6.9) shows that the isentropic efficiency increases with increasing evaporating temperature, since the enthalpy at the inlet of the compressor will increase and so the isentropic efficiency will increase.

Fig (6.9) shows also, that R-22 has higher isentropic efficiency than that for R-407C, but R-407C reaches a value of 93% Of that for R-22 and these indicates that at high evaporating temperature R-407C is more suitable to replace R-22 when compared with the ideal cycle, while it clears that R-134a is not a suitable replacement for R-22 as was found before.

**9. Pressure Ratio:**

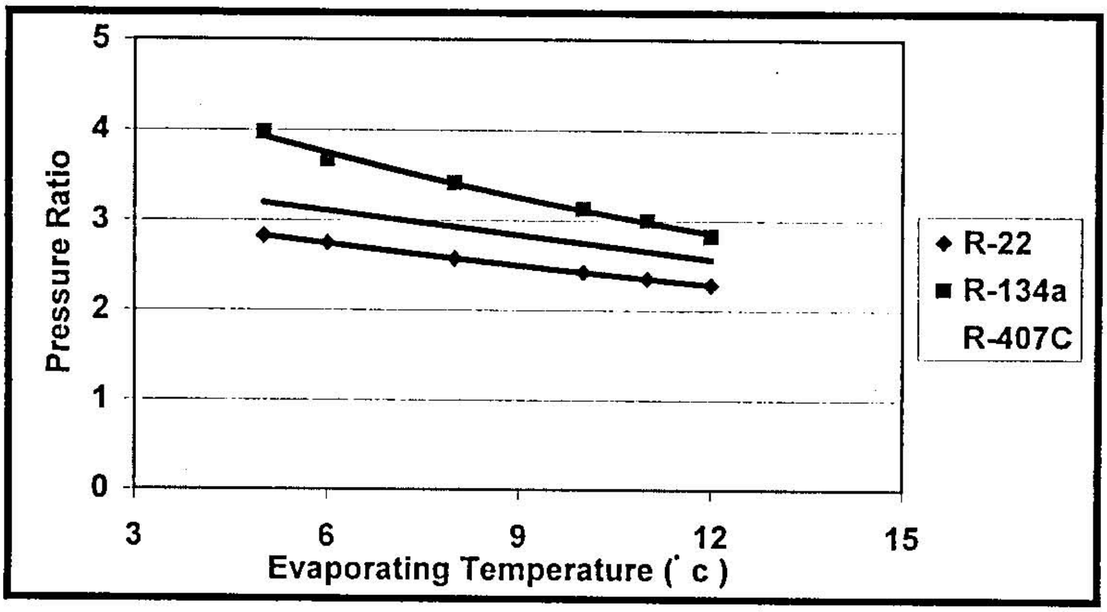



**Fig (6.10) Pressure Ratio Vs. Evaporation Temperature.**

Fig (6.10) shows that the pressure ratio decreases as the evaporation temperature increases and R-134a has a pressure ratio higher than both R-22 and R-407C. This will increase the power consumption for R-134a more than that for R-22 and R-407C, and the actuall volumetric efficiency for R-134a will increase.

**11.Discharge Temperature:**

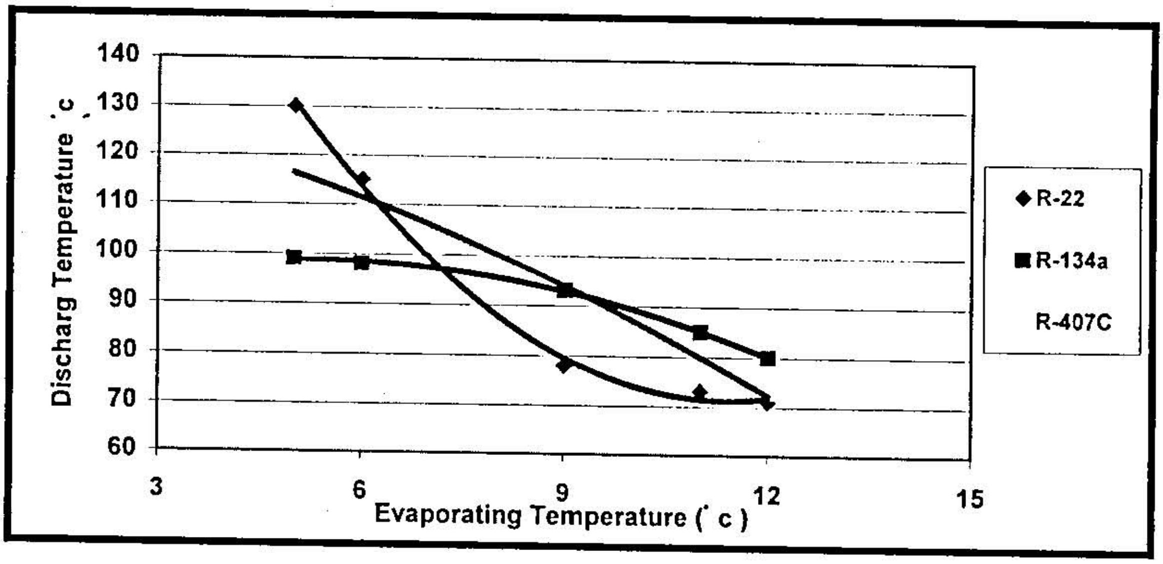



**Fig (6.11) Discharge Temperature Vs. Evaporating Temperature.**

Fig (6.11) shows that the discharge temperature decreases as the evaporating temperature increases due to decrease in the pressure ratio (compression ratio) as shown in figure (6.10). There are three regions can be shown, first at evaporating temperature less than 6° C, R-22 has a higher discharge temperature than R-134a and R-407C, secondly at an evaporating temperature between (6-9)° C, R-407C has a higher value than both R-134a and R-22 and finally when the evaporating temperature greater than 9° C, R-134a has a higher value than R-22 and R-407C.

## 6.4 Condensing Temperature Variation Tests:

These tests were carried out for R-407C at approximately constant evaporating temperature of (5°C) and condensing temperatures of (34, 36, 38, 40, 41, 43) ° C and an ambient temperature of 22°C. Several parameters were considered which include the following:

1. **Refrigeration Effect:**

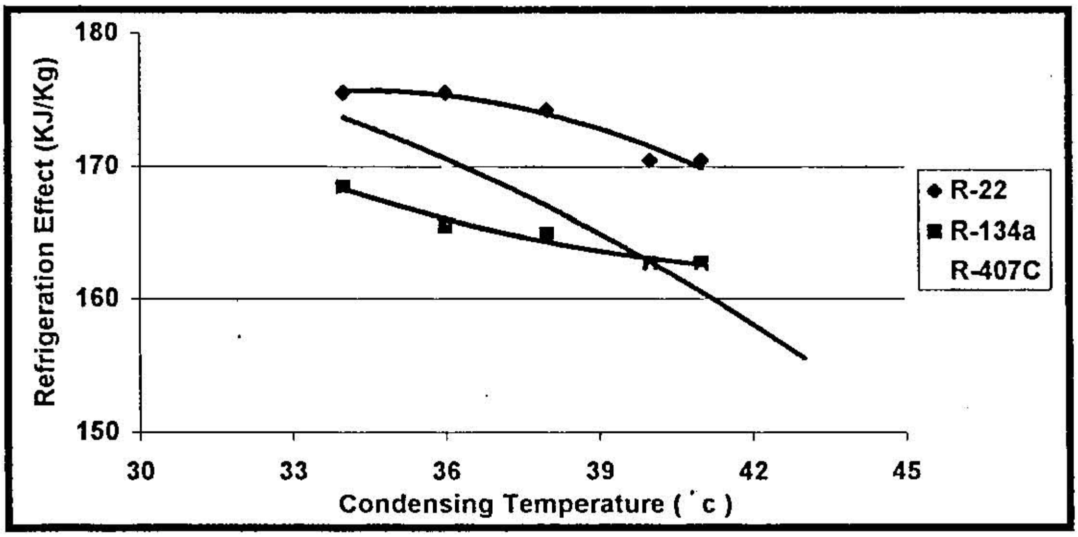



**FIGURE (6.12) Refrigeration Effect Vs. Condensing Temperature.**

Fig (6.12) shows that, the refrigeration effect for the three refrigerants decreases as the condensing temperature increases. Fig (6.12) also shows that R-22 has the higher refrigeration effect than the other two refrigerants, and as the condensing temperature decreases, the refrigeration effect for R-134a and R-407C become closer to that of the R-22.

## 2. Work of Compression:

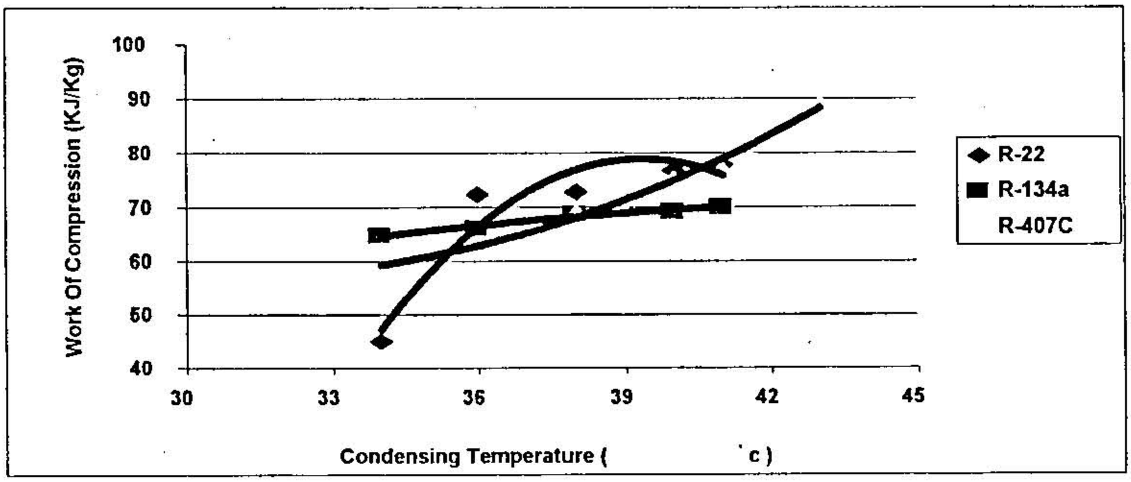



**FIGURE (6.13) Work of Compression Vs. Condensing Temperature.**

Fig (6.13) shows that, increasing condensing temperature will increase the work of compression for the three refrigerants since this will increase the enthalpy at the exit of the compressor and so the work of compression will increase assuming constant evaporating temperature of (5 ° C). Fig (6.13) shows also that, at high temperature more than 40° C, the work of compression for R-407C will be higher than the other two refrigerants while in the region between (36 °C and 40 °C) R-22 has the highest work of compression among the three refrigerants under study.

**3. Coefficient Of Performance:**

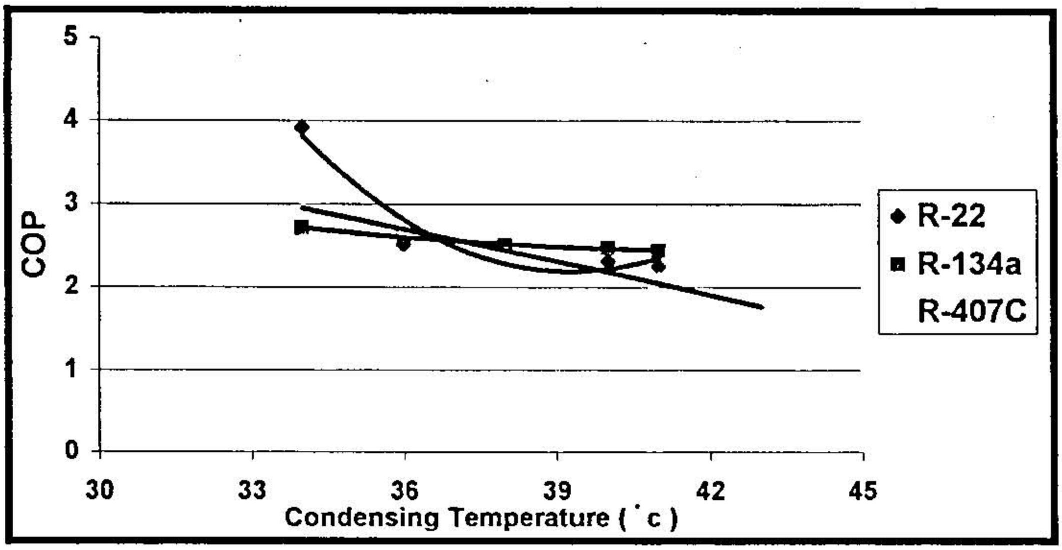



**FIGURE (6.14) COP Vs. Condensing Temperature.**

Fig (6.14) show that, the coefficient of performance for the three refrigerants decreases when increasing the condensing temperature due to the increase in the work of compression and the decrease in refrigeration effect.

Fig (6.14) shows also that at condensing temperature less than 36°C, R-22 has the highest COP, and at a condensing temperature of approximately 40 °C, R-407C reaches a value of 100% of that for R-22.

## 4. Mass Flow Rate:

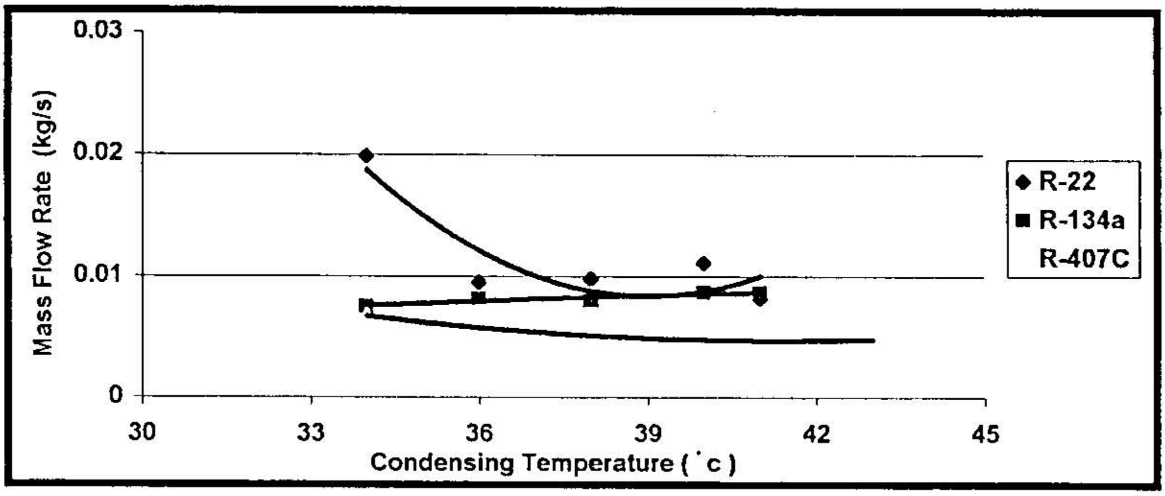
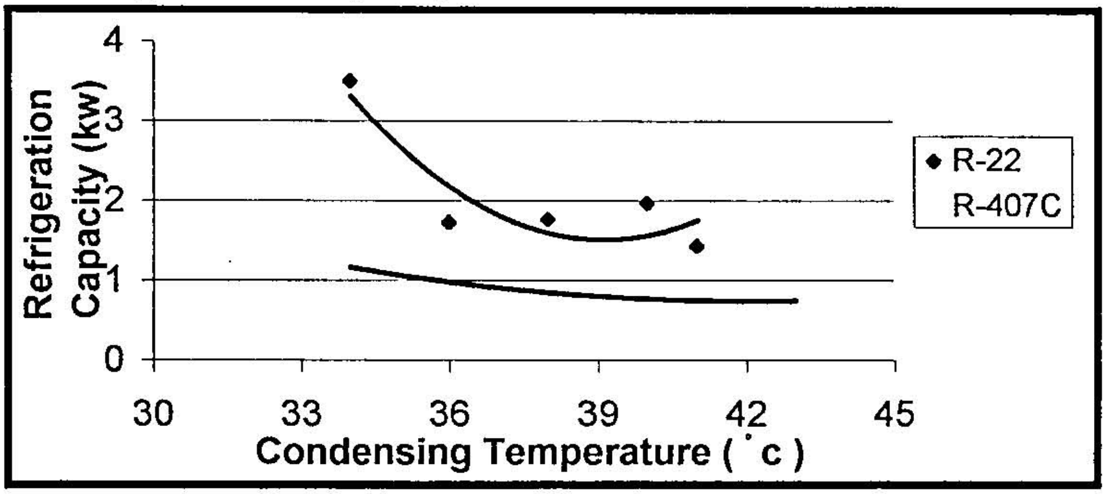



**FIGURE (6.15) Mass Flow Rate Vs. Condensing Temperature.**

Fig (6.15) shows that the mass flow rate for R-22 and R-407C decreases as the condensing temperature increases since the enthalpy at the compressor exit increases and the enthalpy at compressor inlet remain constant (constant evaporating temperature) and so, according to equation (5.5) the mass flow rate decreases.

## 5. Refrigeration Capacity:




**FIGURE(6.16) Refrigeration Capacity Vs. Condensing Temperature.**

Fig (6.16) shows that the refrigeration capacity for R-22 and R-407C rapidly drops with increasing condensing temperature due to the decreasing in both refrigeration effect and the mass flow rate.

Fig (6.16) also shows that the refrigeration capacity for R-407C reaches a value of 70% and higher of that for R-22 .

## 6. Electrical Power Consumption:

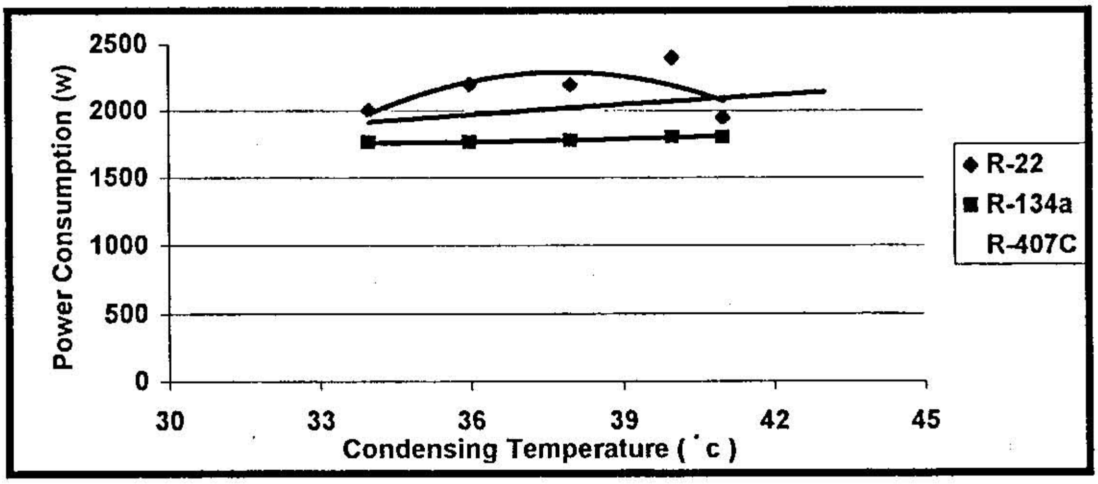



**FIGURE (6.17) Power Consumption Vs. Condensing Temperature.**

Fig (6.17) shows that the electrical power consumption for R-134a and R-407Cincrease as the condensing temperature increases while for R-22 reaches a peak value at approximately 40°C and then the electric power begins to drop. Fig (6.17) shows also, that R-22 consumes power higher than R- 134a and R-407C.

## 7. Heat Rejection:

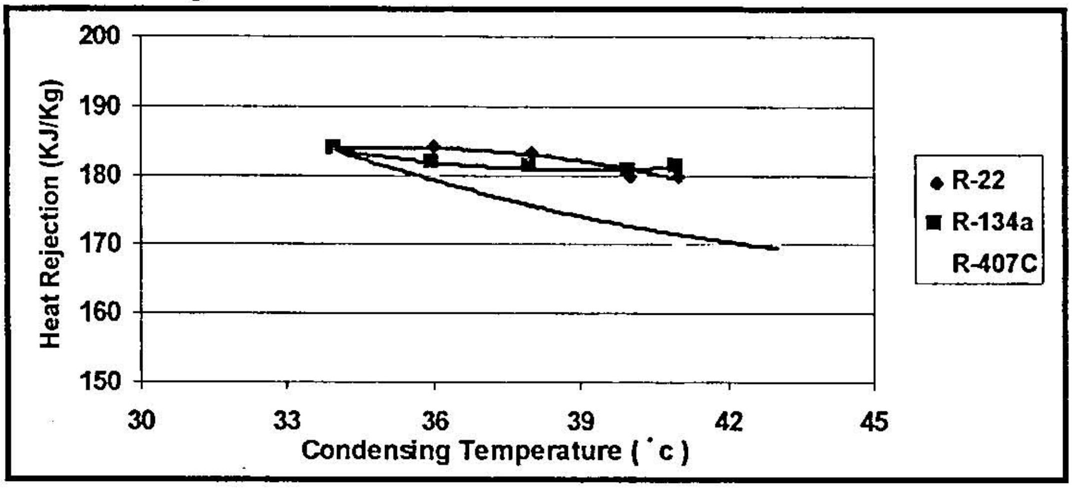
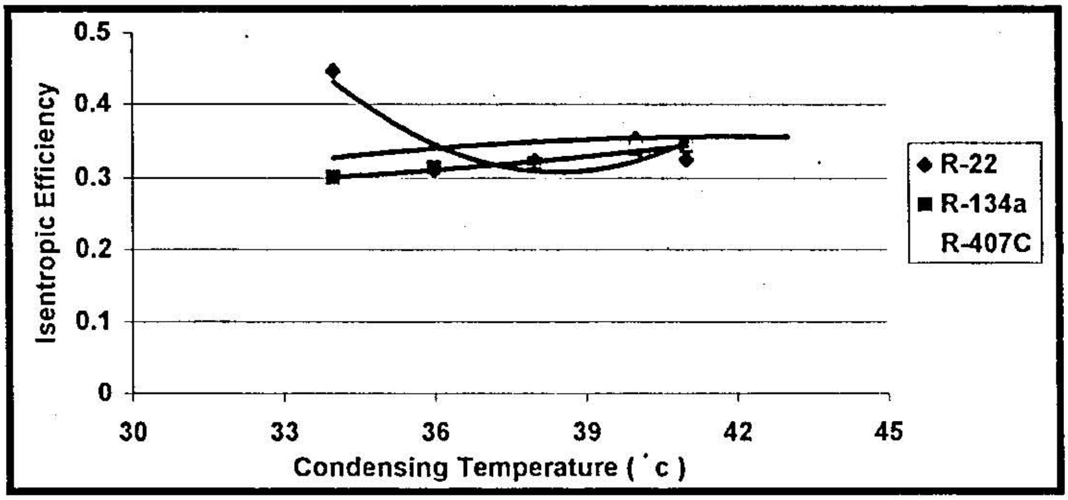



**FIGURE (6.18) Heat Rejection Vs. Condensing Temperature.**

Fig (6.18) shows that R-22 has a higher heat rejection rate than both for R-407C and R-134a and it is clearly shown that as the condensing temperature increases the heat rejection rate decreases for the three refrigerants under study.

## 8. Isentropic Efficiency:




**FIGURE(6.19) Isentropic Efficiency Vs. Condensing Temperature.**

Fig (6.19) shows that the isentropic efficiency increases for both R-134a and R-407C with increasing condensing temperature while for R-22, it decreases until a condensing temperature of 38 ° C and then begins to rise.

## 9. Pressure Ratio:

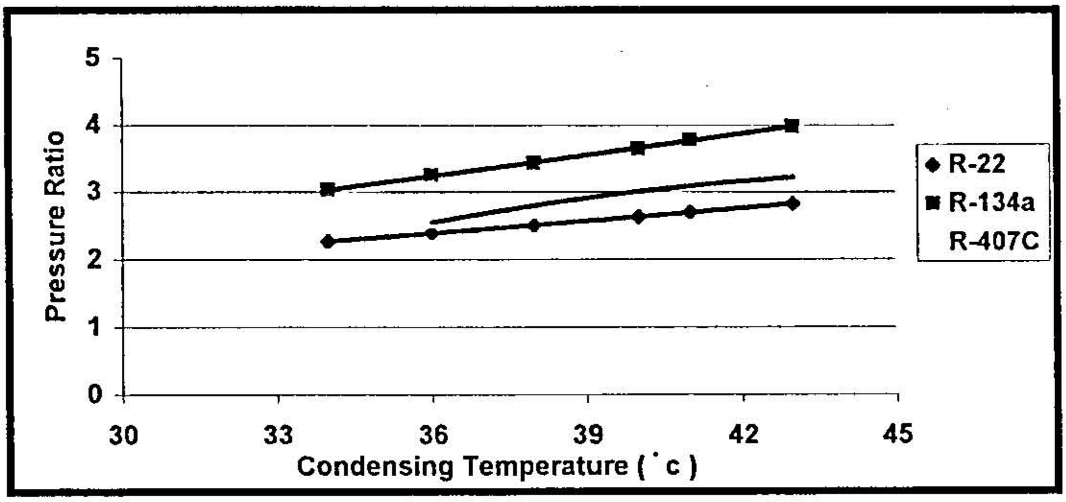



**FIGURE (6.20) Pressure Ratio Vs. Condensing Temperature.**

Fig (6.20) shows that the pressure ratio increases as the condensing temperature increases and R-134a has a pressure ratio higher than that for both R-22 and R-407C.

## 11. Discharge Temperature:

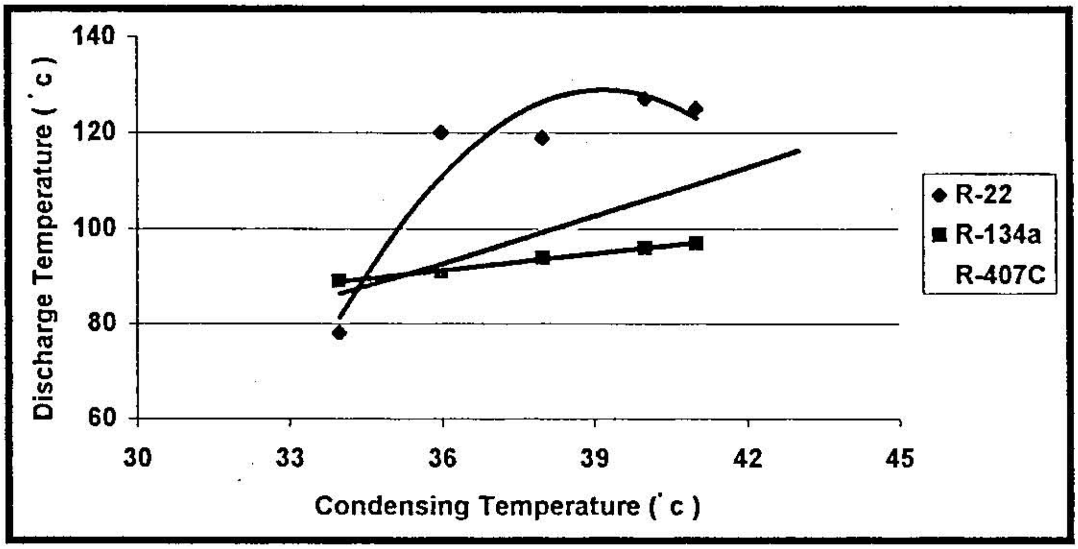



**FIGURE (6.21) Discharge Temperature Vs. Condensing Temperature.**

Fig (6.21) shows that the discharge temperature increases as the condensing temperature increases and it is clearly shown that R-22 has higher discharge temperature than the other two refrigerants.

# 7-CONCLUSION AND RECOMMENDATIONS

## 7.1 Conclusion:

This research covers an experimental study of a split A/C unit using R-407C. From this experiment the following conclusion were deduced:

1. Full set of reasonable performance curves were presented for A/C split unit working with R-407C using actual vapor compression refrigeration cycle.

2. The optimum charge for a split unit for R-407C is approximately equal to the optimum charge for the same unit working with R-22.

3. At medium and high evaporating temperature (higher than 12° C), COP, refrigeration capacity, isentropic efficiency and volumetric efficiency reaches a value between (70-83) % of that for R-22 and more.

4. The electric power consumption for R-22 is higher than that of R-407C.

5. The condenser is suitable to be used in this A/C application by using R-22 and while for R-407C the condenser needs some modification due to the temperature glide.

6. R-407C is a suitable replacement for R-22 in A/C split unit despite the small reduction in the performance when compared to R-22.

## 7.2 Recommendations:

1. Further simulation study on the same unit is recommended. Results could be compared with the results obtained from this research.

2. Due to temperature glide, further study on the design of the condenser and evaporator is recommended in order to offset this problem.

3. Further study on other A/C equipment is recommended in order to generalize the results obtained from this study.

4. Studying the effect of using mineral oil instead of using polyol ester oil.

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## ملخص الدر اسه

# دراسة أداء وحدة تبريد منفصلة قدرة ٢طن باستخدام غاز التبريد R-407C

**إعداد: فراس محمود رشيد مكاحله**

**أ لمشرف: أ.د.محمود حماد**

الهدف من الدراسة هو دراسة الأداء لوحده تكييف منفصلة قدرتها الاسمية (٢) طن تبريد عامله على غاز تبريد R-22 باستبداله بغاز R-407C. السبب في اختيار R-407C هو انه صديق للبيئة ولا يضر بطبقه الأوزون بعكس R-22.

في البداية تمت إزالة (Expansion Valve) واستبداله بآخر يمكن التحكم به يدويا حتى نتمكن من التحكم بدرجة حرارة التبخر حيث تضمن البحث تغيير درجة حرارة التبخر عند درجة حرارة تكثيف معينه , ومن ثم تغيير درجة حرارة التكثيف عند درجة تبخير ثابتة .

تم تغيير الزيت المستخدم وهو عبارة عن زيت معدني ملائم لـ R-22 , حيث تم استبداله بزيت آخر ملائم لـ R-407C والمسمى Polyol ester، حيث ان له ذا ثبيه جيده معه بالاضافه إلى انه لا يتفاعل معه كيمائيا" .

تم البحث عن الشحنه الملائمة من غـــاز R-407Cوالتـــي تعطـــي احســن كفــاءه للجهاز, حيث تم شــحن الجــهاز بعــدة شــحنات ( ١٣٠٠-١٧٠٠) غــرام وتبيــن أن الشحنه التي تعطي افضــــل أداء للجــهاز هـــي ١٤٥٠غــرام عنــد درجــة حــرارة تكثيف ( ٤٣° ) ودرجة حـــرارة تبخــير ( ٥° ) .

تنقسم التجارب إلى قســمين :

أولا : تغيير درجة حرارة التبخير عند درجة حـــرارة تكثيــف ثابتــة .

ثانيا : تغيير درجة حرارة التكثيف عند درجة حـــرارة تبخــير ثابتــه .

لقد تضمن هذا البحث منحنيـــات تبيـــن أداء الوحـــدة باســـتخدام (R-407C) وعمــل مقارنه مـــع أداء نفــس الوحـــدة باستخــــــدام (R-22) وغـــــــــــاز (R-134a)حيث تبين أن R-407C يعطـــي كفـــاءه جيـــده نســبيا"تصـــل تقريبـــا إلى COP=4.1 عند درجـــة تبخـــر١٢° م ودرجـــة حــرراه تكثيــف ٤٣° , بينمـــا كان COP =5.0 عند نفـــس الظـــروف باســـتخدام R-22 ومـــن هـــذه التجـــارب تبيـــن أن R-407C يعتـــبر بديـــلا مناســـبا لـــ R-22 فـــي وحـــدات التـــبريد المنفصلة ( A/C SPLIT UNIT) .