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Experimental Investigation of an Alternate Refrigerant for R22 in Window Air Conditioning System

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Abstract- This paper is concerned with the future phase-out of Hydro Chloro Fluoro Carbons (HCFCs) used in the air conditioning systems. The air conditioning industry is currently evaluating alternative refrigerants for R-22. A window-type air conditioning system is selected for the tests conducted with three different types of refrigerants. These air conditioning units are spread widely in their applications and are circulating R-22 as a refrigerant. Finding an alternative refrigerant for replacing R-22 is becoming a practical problem because general use of hydro chlorofluorocarbons (HCFCs) including R-22 is promised to be banned by 2020 as per the Montreal Protocol.

It is intended to replace R-22 refrigerant by other refrigerants which are considered to be environmental friendly. In this project, two zeotrope blend refrigerants were selected to be tested as alternative refrigerants for R-22 in the window type air conditioner system viz., R-407C (mixture of R-32/125/134a), R-407A (mixture of R-32/125/134a) to their better thermal properties and acceptable pressure and temperature ranges. The alternate refrigerants to be used in the project have very less ozone depletion potential (ODP) and global warming potential (GWP). The performance of each refrigerant has been found individually and the results were used to evaluate and compare the following performance criteria: cooling capacity, Energy Efficiency Ratio and the coefficient of performance (COP).

Index Terms- Alternative Refrigerant, HCFCs, Zeotrope blend ODP, GWP.

I. Introduction

A great breakthrough occurred in the field of air-conditioning with the development of Freons by E.I.dupont.
Freons are a series of fluorinated hydrocarbons, popularly known as fluorocarbons, derived from methane, ethane etc as bases. Since their development in 1931, chloro fluoro carbons (CFCs) were thought to be ideal Refrigerants. In 1974, CFCs were tentatively identified as destructive to the ozone layer.

A. Montreal Protocol

The scientific confirmation of the depletion of the ozone layer prompted the international Community to establish a mechanism for cooperation to take action to protect the ozone layer. This was formalized by a treaty called the Vienna Convention for the Protection of the Ozone Layer, which was adopted and signed by 28 countries on 22nd march 1985 in Vienna. This led to the drafting of the Montreal Protocol on Substances that Deplete the Ozone Layer. The Protocol was signed by 24 countries and by the European Economic Community and entered into force on 1st January 1989. The treaty states that the Parties to the Montreal Protocol recognize that worldwide emissions of ozone-depleting substances (ODSs) significantly deplete and otherwise modify the ozone layer in a manner that is likely to result in adverse effects on human health and the environment. At Montreal, a 50 per cent cut by 2000 was decided on. However, this was adjusted only three years later, when full scientific evidence was available. The First Meeting of the Parties to the Protocol was held in Helsinki in May 1989, and the Parties have met every year since to review progress and discuss amendments resulting from continued research and technical developments. The provisions of the Protocol include the requirement that the Parties to the Protocol base their future decisions on the current scientific, environmental, technical and economic information assessed by panels drawn from the worldwide expert communities. The most recent scientific assessment of the current status of the ozone layer is set out in a report published by the World Meteorological Organization (WMO) entitled Scientific Assessment of Ozone Depletion 2006.

B. Refrigerant Solutions For Today's Environmental Challenges

The HVACR industry is facing two major environmental challenges today: stratospheric ozone depletion and global climate change. Stratosphere Ozone Depletion is believed to be caused by the release of certain manmade ozone depleting chemicals into the atmosphere. Arora C. P., B.K.Bhalla and Addai Gassab studied the performance of window type air conditioners using R-22 (1979). They found thirteen (13) compounds that satisfied this criterion. These are: R-115; 500, 502, 505, 506, which are already banned R-161 which is highly toxic; R-143a, 152a which are slightly flammable and R-22, 124, 125, 134 and 134a which are nonflammable. There are many other works published on the alternatives to R-22 and other ozone depletion refrigerants some of these are Kuehl S.J., Goldschmidt V.W. Steady flows of R-22 through capillary tubes (1990). Boumaza, M. M., (2007). Investigation and Comparison of Chlorine Compounds Refrigerants and their Potential Substitutes Operating at high Ambient Temperature for the Replacement of R22.Hoffman also studied the replacements for HCFCs.

C. Window Air Conditioner

A window air conditioner is a system that cools space to a temperature lower than the surroundings. To accomplish this, heat must be removed from the enclosed space and dissipated into the surroundings. However, heat tends to flow from an area of high temperature to that of a lower temperature. During the cycle, a substance called the refrigerant circulates continuously through four stages. The first stage is called Evaporation and it is here that the refrigerant cools the enclosed space by absorbing heat. Next, during the Compression stage, the pressure of the refrigerant is increased, which raises the temperature above that of the surroundings. As this hot refrigerant moves through the next stage, Condensation, the natural direction of heat flow allows the release of energy into the surrounding air. Finally, during the Expansion phase, the refrigerant temperature is lowered by refrigeration effect. This cold refrigerant then begins the Evaporation stage again, removing more heat from the enclosed space.

A typical diagram of a window air conditioner which works according to the process explained above is shown in the figure.

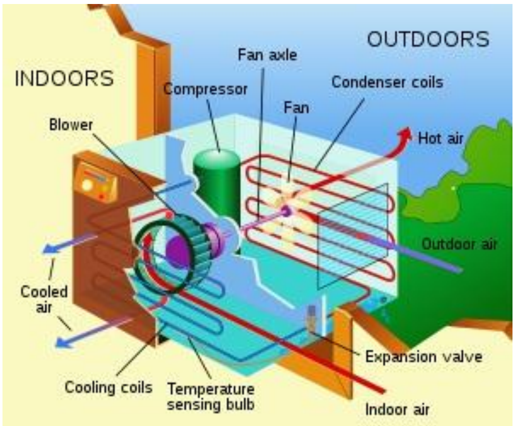

Fig.1. Construction of a window air conditioner

D. Refrigerant in Vapour Compression Refrigeration system

Window air conditioner works on the principle of vapour compression refrigeration system. The refrigerant is a heat carrying medium which during the cycle (that is compression, condensation, expansion and evaporation) in the refrigeration system absorb heat from a low temperature source and discard the heat so absorbed to a higher temperature sink.

The suitability of a refrigerant for a certain application is determined by its physical, thermodynamic, chemical properties and by various practical factors. There is no one refrigerant which can be used for all types of applications. If one refrigerant has certain good advantages, it will have some disadvantages also for a particular application. Hence, a refrigerant is chosen which has greater advantages and less disadvantages.

II. EXPERIMENTAL WORK

A. Description of the Test Apparatus

A GODREJ company window air conditioner of 1 ton refrigeration capacity was selected to be as a test rig. The overall physical dimensions of the window air conditioning system are (60 X 56 X 38) cm and 42 kg weight. Figure 2. shows the schematic diagram of the window air conditioner used in the experiment.

The unit is having single electricity phase rotary compressor. The condenser and evaporator coils are made of copper with smooth inner tube surface. The evaporator fins are hydrophilic and Condenser fins are Hydrophobic. The interrupted type of fin used in the experiment is very widely accepted method of increasing the heat transfer coefficient and creating more turbulent mixing on the air side of heat exchangers. Both compressor and condenser fins were made of alluminium.

The window air conditioner utilizes refrigerant R-22 and mineral lubricating oil. In order to provide superior lubrication with chlorine refrigerants poly ester lubricants were used. The air conditioner accommodates a three speed motor to run the condenser and evaporator fans.

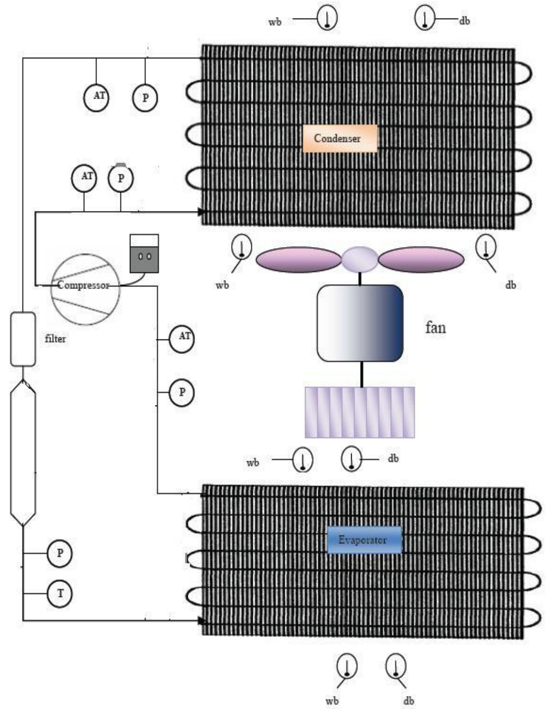

Fig.2. Schematic diagram of the window air conditioner

B. Selection of the Refrigerant

The new trend is to use zeotrope blend refrigerants in the air conditioning system. In the present experiment, two zeotrope blend refrigerants were selected to be tested as alternate refrigerants for R-22 in the window air conditioner test rig. These refrigerants were R-407C comprising of (R32/R125/R134a) in a mass fraction composition percentage as (23/25/52) and R-407A comprising of (R32/R125/R134a) in a mass fraction composition percentage as (20/40/40).

These are commercially available have been assigned an indentifying number in the 400series. Zeotrope blends shift in composition during condensing process. As the blend changes phases, more of one component will transfer to the other phase faster than the rest. This property is called fractionation. The changing composition of the liquid from one side of the heat exchanger to the other is called the temperature glide. The temperature glide will cause different values for temperature at a given pressure, depending on how much refrigerant is liquid and how much is vapour. The alternate refrigerants used in the present work have very less ODP and GWP as compared to R-22.

C. Refrigerant Charging

The refrigerant may be charged in a liquid or vapour modes. This is limited by operating factors, such as the amount of refrigerant and time of charging. Charging a refrigeration system, especially the one built-up with capillary tube control, is the most critical task. Amount of refrigerant to be charged is so selected that it maintains desired suction & discharge pressures. It is customary to charge the system with a charging cylinder on volume basis but the short-coming of this method is that since the density of refrigerant varies appreciably with temperature, one can come across erroneous quantity as the charging cylinder does not have different scales for different ambient temperatures. A better alternative method is to charge the refrigerant by weight. Charging without the aid of any equipment requires a high level of skill and human judgment. Sometimes charging is done without the aid of any equipments, this system uses suction pressure and discharge pressure as indicative of the charge quantity. However, this needs a high level of skill and human judgment.

III. TEST PROCEDURE

At the incipience of the test, the system was kept running at least 10 minutes to reach the steady state conditions. This was done by monitoring the temperature and pressure gauge for the circulated refrigerant. After that achievement, the refrigerant side measurements, temperature and pressure, and air side measurements, dry and wet bulb temperature, were recorded. These readings were taken at ambient temperature i.e.,27.3 °C DBT and 19.2 WBT to detect the performance of the window air conditioner test rig .

This procedure was repeated for the refrigerants R-407C and R407A.The tests were usually commenced at highest fan speed where the volumetric air flow rate fixed at (9.33) m3/min. as specified by the unit manufacturer company.

IV. ANALYSIS OF THE EXPERIMENTAL DATA

The data analysis involved a number of assumptions that are important to be addressed, as described below:

1. The mass flow rate of refrigerant is constant at all parts of the experimental test rig.
2. The air temperature at the entrance and exit of heat exchangers are constant and homogenous at all tubes in the front and lee sides.

The data reduction procedure includes the refrigerating effect, power consumed by compressor, heat rejected in the condenser, energy efficiency ratio calculated for both R-22 and its alternatives. In addition, (COP) was calculated from the above mentioned parameters. The properties of R-22 and other refrigerants were obtained from the published data by ASHRAE Hand Book.

V. PERFORMANCE CALCULATIONS

A. Refrigerant R-22

1. Condenser temperature, T_c = 65.4 °C
2. Condenser Pressure, P_c = 18.12 bar
3. Evaporator Temperature, T_e = 7.9 °C
4. Evaporator Pressure, P_e = 3.78 bar

Calculations for Cooling Capacity:

Readings taken from the test rig:

Mass Flow rate of Air = 0.1688 kg/sec

Input Power (Watts) = 970

Entering Air Enthalpy, h_{ae} = 55.8 kJ/kg

(Taken from psychrometric chart at Indoor air temperature i.e. 27.3 °C DBT, 19.2 °C WBT)

Leaving Air Enthalpy, h_{al} = 35.9 kJ/kg

(Taken from psychrometric chart at Leaving air temperature i.e. 18.59 °C DBT, 12.34 °C WBT)

Enthalpy Difference = Entering Air Enthalpy (h_{ae}) – Leaving Air Enthalpy (h_{al}) = 55.8 – 35.9

= 19.9 kJ/kg

Cooling Capacity = Mass flow rate of air * Enthalpy difference = 0.1688 * 19.9

= 3.359 kW

Cooling Capacity = Cooing capacity in kW * 3412.14

= 3.359 * 3412.14

= 11461.37 Btu/hr

Energy Efficiency Ratio = (Cooling capacity in Btu/hr)/ (Input Power in Watts)

= 11461.37 / 970

= 11.81

COP of system = Energy Efficiency Ratio / 3.412

= 11.81/3.412

= 3.46

From Pressure –Enthalpy Diagram (at corresponding pressure and temperatures)

Enthalpy at the beginning of compression,

h1 = 403kJ/kg

Enthalpy at the end of compression

h2 = 441kJ/kg

Enthalpy at the beginning of the expansion

h3 = 259kJ/kg

Enthalpy at the end of expansion

h4 = 259kJ/kg

Capacity of the system = 1TR = 1 * 3.5 KW

= 3.5kW

Mass Flow rate, m_r = Capacity in kW/ (h1-h4)

= 3.5 / (403-259)

= 0.0243 kg/sec

Refrigeration Effect (Re) = (h1 - h4)

= (403 – 259)

= 144 kJ/kg

Compressor work (W) = m_r x (h2 - h1)

= 0.0243 x (441 –403)

= 0.923 kW

Heat rejected in the Condenser, = m_r x (h2- h3)

= 0.0243 x (441-259)

= 4.422kW

Co- efficient of Performance,

(C.O.P.) = (h1 – h4) / h2 – h1)

= (403 – 259) / (441 – 403)

= 3.69

B. Refrigerant R-407c

1. Condenser temperature, T_c = 74. °C
2. Condenser Pressure, P_c = 19.45 bar
3. Evaporator Temperature, T_e = 10.5 °C

4. Evaporator Pressure, P_e = 3.9 bar

Calculations for Cooling Capacity:

Readings taken from the test rig:

Mass Flow rate of Air = 0.1688 kg/sec
Input Power (Watts) = 970
Entering Air Enthalpy, h_{ae} = 55.7 kJ/kg
(Taken from psychrometric chart at Indoor air temperature i.e. 27.3 °C DBT, 19.2 °C WBT)
Leaving Air Enthalpy, h_{al} = 34.9 kJ/kg
(Taken from psychrometric chart at Leaving air temperature i.e. 17.8 °C DBT, 12°C WBT)
Enthalpy Difference = Entering Air Enthalpy (h_{ae}) – Leaving Air Enthalpy (h_{al} l) = 55.8 – 35.9
= 20.8 kJ/kg
Cooling Capacity = Mass flow rate of air * Enthalpy difference = 0.1688 * 20.8
= 3.511 kW
Cooling Capacity = Cooing capacity in kW * 3412.14
= 3.511 * 3412.14
= 11980.02 Btu/hr
Energy Efficiency Ratio = (Cooling capacity in Btu/hr)/ (Input Power in Watts)
= 11980.02 / 970
= 12.26
COP of system = Energy Efficiency Ratio / 3.412
= 12.26/3.412
= 3.59
From Pressure –Enthalpy Diagram (at corresponding pressure and temperatures)
Enthalpy at the beginning of compression,
h1 = 411 kJ/kg
Enthalpy at the end of compression
h2 = 455 kJ/kg
Enthalpy at the beginning of the expansion
h3 = 260 kJ/kg
Enthalpy at the end of expansion
h4 = 260 kJ/kg
Capacity of the system = 1TR = 1 * 3.5 KW
= 3.5kW
Mass Flow rate, m_r = Capacity in kW/ (h1-h4)
= 3.5 / (411-260)
= 0.0231 kg/sec
Refrigeration Effect (Re) = (h1 - h4)
= (411 – 260)
= 151 kJ/kg
Compressor work (W) = m_r x (h2 - h1)
= 0.0231 x (455 –411)
= 1.016 kW

Heat rejected in the Condenser = m_r x (h2- h3)
= 0.0231 x (455-260)
= 4.504 kW
Co- efficient of Performance,
(C.O.P.) = (h1 – h4) / h2 – h1)
= (411 – 260) / (455 – 411)
= 3.431

C. Refrigerant R-407a

1. Condenser temperature, T_c = 76 °C
2. Condenser Pressure, P_c = 22.4 bar
3. Evaporator Temperature, T_e = 10.9 °C
4. Evaporator Pressure, P_e = 4.59 bar

Calculations for Cooling Capacity:
Readings taken from the test rig:

Mass Flow rate of Air = 0.1711 kg/sec
Power (Watts) = 970
Entering Air Enthalpy, h_{ae} = 54.8 kJ/kg
(Taken from psychrometric chart at Indoor air temperature i.e. 27.2 °C DBT, 19.1 °C WBT)
Leaving Air Enthalpy, h_{al} = 32.9 kJ/kg
(Taken from psychrometric chart at Leaving air temperature i.e. 15 °C DBT, 10.9 °C WBT)
Enthalpy Difference = Entering Air Enthalpy (h_{ae}) – Leaving Air Enthalpy (h_{al}) = 54.8 – 32.9
= 21.1 kJ/kg
Cooling Capacity = Mass flow rate of air * Enthalpy difference = 0.1711 * 21.1
= 3.610 kW
Cooling Capacity = Cooing capacity in kW * 3412.14
= 3.610 * 3412.14
= 12317.82 Btu/hr
Energy Efficiency Ratio = (Cooling capacity in Btu/hr)/ (Input Power in Watts)
= 12317.82 / 970
= 12.698
COP of system = Energy Efficiency Ratio / 3.412
= 12.698/3.412
= 3.721
From Pressure –Enthalpy Diagram (at corresponding pressure and temperatures)
Enthalpy at the beginning of compression,
h1 = 311 kJ/kg
Enthalpy at the end of compression
h2 = 346 kJ/kg
Enthalpy at the beginning of the expansion
h3 = 176 kJ/kg
Enthalpy at the end of expansion
h4 = 176 kJ/kg
Capacity of the system = 1TR = 1 * 3.5 KW
= 3.5kW
Mass Flow rate, m_r = Capacity in kW/ (h1-h4)
= 3.5 / (311-176)
= 0.0259 kg/sec
Refrigeration Effect (Re) = (h1 - h4)
= (311 – 176)
= 135 kJ/kg
Compressor work (W) = m_r x (h2 - h1)
= 0.0259 x (346 –311)
= 0.906 kW
Heat rejected in the Condenser = m_r x (h2- h3)
= 0.0259 x (346-176)
= 4.403KW
Co- efficient of Performance,
(C.O.P.) = (h1 – h4) / h2 – h1)
= (311 – 176) / (346 – 311)
= 3.857

Table.1. Comparison of performance parameters

erformance Parameters	R-22	R407C	R407A
Refrigerant mass flow rate (kg/sec)	0.0243	0.0231	0.0259
Cooling Capacity(kW)	3.359	3.511	3.611
Heat rejected in the Condenser (kW)	4.422	4.505	4.403
Compressor work (Watts)	923	1016	906
Energy Efficiency Ratio	11.81	12.26	12.698
COP	3.46	3.59	3.71

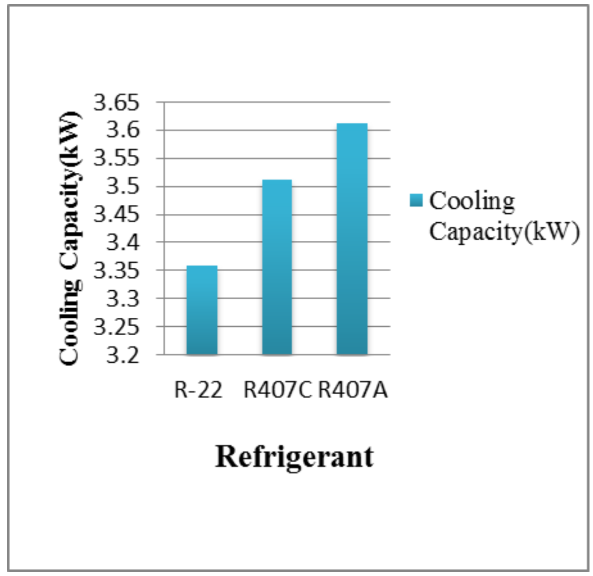

Fig.4. Cooling capacity comparison of the refrigerants

Graphs

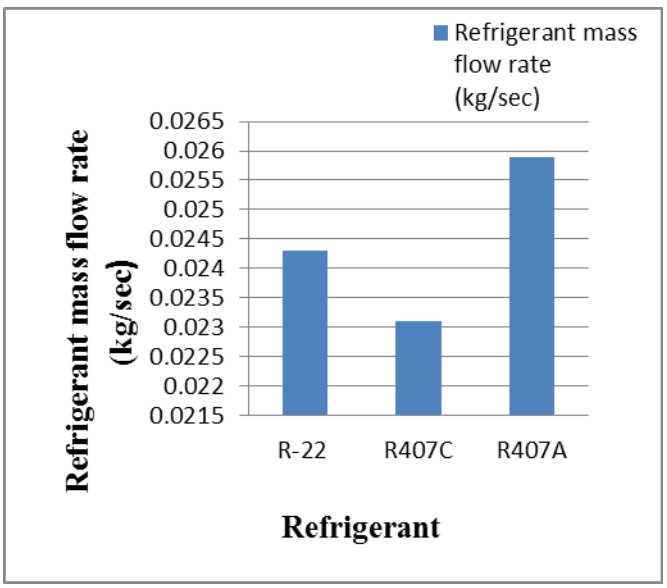

Fig.3. Refrigerant mass flow rate Comparison of the refrigerants

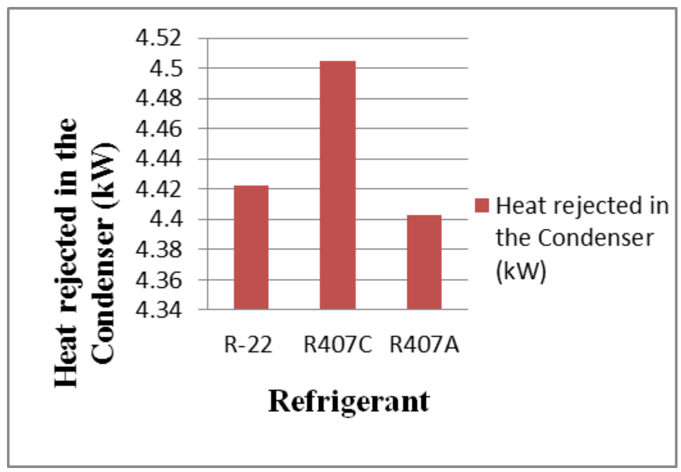

Fig.5. Comparison of heat rejected in the condenser

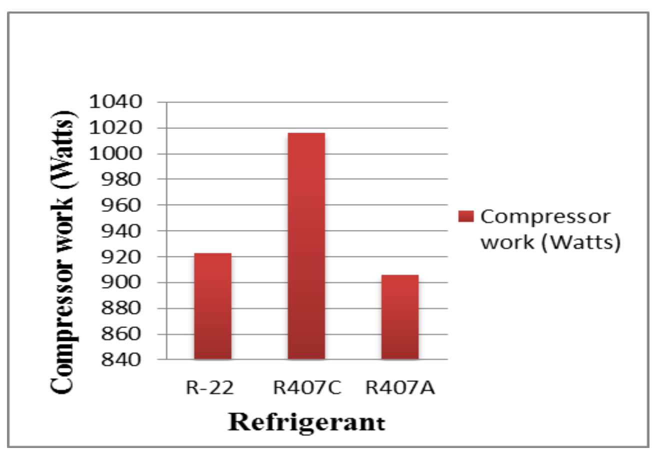

Fig.6. Compressor work Comparison of the refrigerants

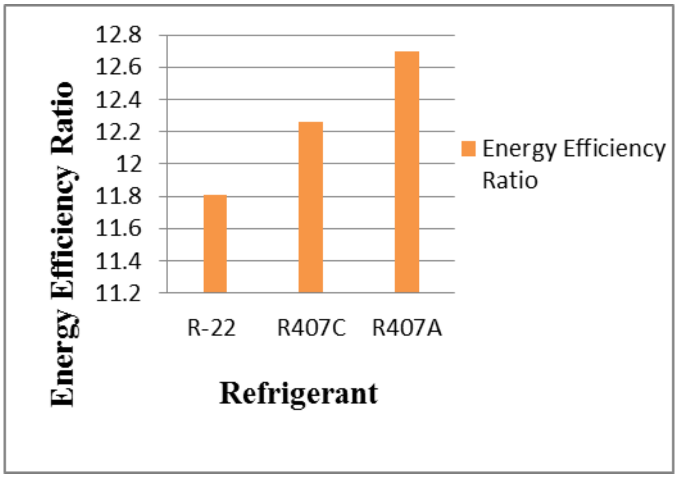

Fig.7. Energy Efficiency Ratio comparison of the refrigerants

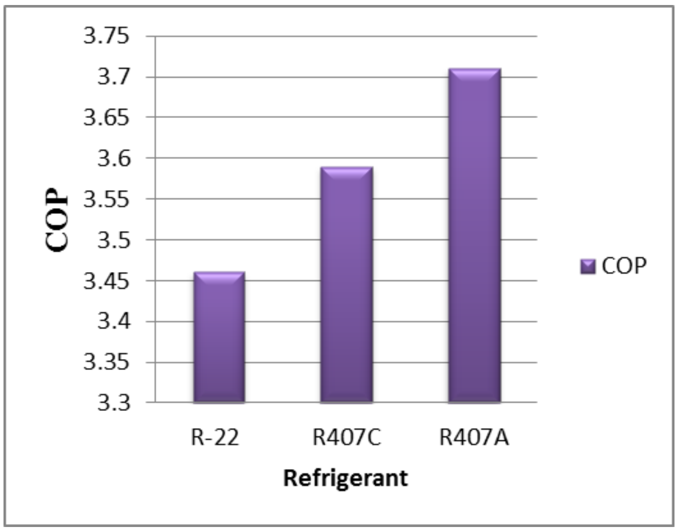

Fig.8. COP comparison of the refrigerants

VI. CONCLUSION

The present experimental work showed the following findings:

• The drop in technique of R-22 by R-407C and R-407A improved cooling capacity up to (4.5%) and (7.5%) respectively. This emphasizes a very important point that the existing evaporator circuit is very suitable for the present alternative refrigerants.

• R-407A exhibited lower power consumption than that experienced with R-22 tests by (2%). On the contrary, R-407C showed a higher consumed power than that of R-22 by (9%).

• R-407C and R-407A showed a significant increase in Energy Efficiency Rate by (4%) and (7.5%) respectively for the operating conditions presented here.

• R-407C exhibited decrease in mass flow rate than that experienced with R-22 tests by (5%). On the contrary, R-407A showed an increase in mass flow rate than that of R-22 by (6.5%).

• R-407C and R-407A showed a significant increase in COP by (3.75%) and (7.2%) respectively for the operating conditions presented here.

• The results confirmed that R-407C and R-407A are promising alternatives as a direct replacement; drop in of R-22 in RAC. Noting that the drop in technique is a feature of the refrigeration unit. Therefore, the performance of a specific alternative varies from one application to another.

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NOMENCLATURE

COP: Coefficient of Performance
h: Enthalpy, (kJ/kg)
DBT: Dry bulb temperature, (°C)
WBT: Wet bulb temperature,(°C)

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