International Journal of Current Engineering and Technology
E-ISSN 2277 – 4106, P-ISSN 2347 - 5161

Available at http://inpressco.com/category/ijcet

Research Article

Experimental Analysis & Performance Evaluation of a Split Air Conditioning System using R22, R407C and R410A

K. Ravi Kumar[A*], B. Uma Maheshwar Goud[B], B. Sudheer Prem Kumar[B] and P. Bhramara[B]

[A]Department of Mechanical Engineering, Al-Habeeb College of Engineering & Technology, Hyderabad, India
[B]Department of Mechanical Engineering, Jawaharlal Nehru Technological University Anantapur, Anantapur, India

Accepted 10 Dec 2014, Available online 13 Dec2014, **Vol.4, No.6 (Dec 2014)**

Abstract

The main objective of the Paper is to present the performance analysis of a split air conditioning system of capacity 1.5tonnes using three refrigerants viz., R22, R407C(azeotriopic mixture of R32/R125/R134a, 23%/25%/52% by weight) and R410A(azeotropic mixture of R32/R125, 50%/50% by weight). The experimental test was conducted at standard ISO 5151 by maintain indoor & outdoor air temperatures at 27°C DBT, 19°C WBT and 35°C DBT, 24°C WBT respectively in an Air conditioning test room. 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 Co-efficient of performance (COP).

Keywords: *Air Conditioner; Refrigerant; Cooling Capacity; EER; COP; Performance; R22; R407C; R40A*

1. Introduction

As CFC (Chlorofluorocarbon) and HCFC (hydro chlorofluoro carbons) refrigerants which have been used as refrigerants in a vapor compression refrigeration system were known to provide a principal cause to ozone depletion and global warming, production and use of these refrigerants have been restricted. Therefore, new alternative refrigerants should be searched for, which fit to the requirements in an air-conditioner or a heat pump, and refrigerant mixtures which are composed of HFC(hydroflurocarbon) refrigerants having zero ODP(ozone depletion potential are now being suggested as drop-in order mid-term replacement. An air-conditioning system using new alternative refrigerants must be modified or newly designed because thermo-physical properties of these alternative refrigerants differ from those of conventional refrigerants in order to maintain or improve the performance of the cycle, the operating characteristics of individual components of the cycle should be clarified with new alternative refrigerants.

In 1974, CFCs were tentatively identified as destructive to the ozone layer (Domanski, 1997). For the next decade, this relationship was investigated by the World Meteorological Organization and the United Nations Environment Programme (WMO/UNEP) in 1985. The Montreal Protocol (1987), which was agreed to by nearly one hundred and fifty countries, froze CFC consumption in 1989 and pledged to cut it in half by 1998. In 1992, the Copenhagen Amendments went even further, and halted the production of CFCs in developed countries by 1996.

Chlorodifluoromethane (R22 or HCFC22)has been used as a refrigerant withinrefrigeration, industrial cooling, airconditioning and heating applicationsfor many years. The low ozone depletionpotential of R22 compared to CFC11 andCFC12 combined with its excellent refrigerantproperties have helped facilitate thetransition away from CFCs. However, evenif HCFCs, such as R22, are less damaging tothe ozone layer than CFCs, they still containozone-destroying chlorine. In addition, R22is a greenhouse gas and the manufactureof R22Results in a byproduct (HFC23) thatcontributes significantly to global warming.Therefore R22 is being phased out andreplaced with ozone-friendly refrigerants.

World Wide Ozone Depletion Legislation

Worldwide legislation has been enacted through the United NationsEnvironmental Programme (UNEP) to reduce stratospheric ozone depletion. TheMontreal Protocol was approved in 1987 to control the production of suspected ozone –depleting substances, among them chlorofluorocarbons (CFCs) and hydro chlorofluorocarbons(HCFCs), commonly used as refrigerants in the HVACR industry. The MontrealProtocol has a provision to conduct and review future scientific, technical and economicassessments, and adjust the legislation accordingly. Further evidence in the early 1990sdid suggest a phase out of ozone – depleting substances was in order. Amendments weremade to the protocol in London (1990), in Copenhagen (1992) and in Vienna (1995). Nochanges have been made since 1995.

*Corresponding author **K. Ravi Kumar** is working as Assistant Professor; **Dr. B. Uma Maheshwar Goud, Dr.B. Sudheer Prem Kumar** and **Dr. P. Bhramara** are working as Professors

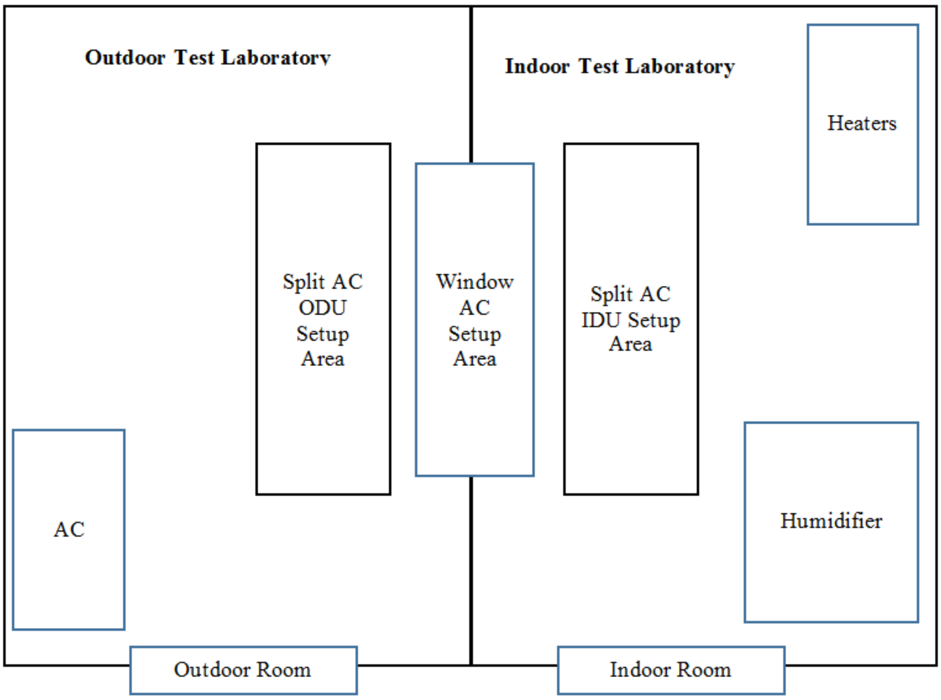


Fig.1. Psychometric Laboratory view

Refrigerant Solutions for Today's Environmental Challenges

The HVACR industry faces two major environmental challengestoday stratospheric ozone depletion and global climate change. Stratosphere Ozone Depletion isbelieved to be caused by the release of certain manmade ozone - depleting chemicals into the atmosphere. A compromised ozone layer results in increased ultraviolet (UV) radiation reaching the earth's surface, which can have wide ranging health effects. Legislation has been enacted worldwidethrough the Montreal Protocol to phase out the production of these chemicals. This phase out is in progress, and the scientific evidence indicates repair to the ozone layer is underway. Global Climate Change is believed to be caused by the buildup of greenhouse gases in the atmosphere. The primary greenhouse gas is carbon dioxide (CO2), created by fossil - burning power plants. These gases trap the earth‟s heat, causing global warming,Legislation is not yet in place, but any policy changes of legislation will be enacted through the Kyoto protocol. CFC and HCFC refrigerants used by the HVACR industry are suspected ozone - depleting substances. CFC, HCFC and HFC refrigerants are considered greenhouse gases. In addition, HVACR equipment is a major power consumer. Therefore, the industry is part of these environmental challenges.

2. Experimental Setup

Air conditioner test room is constructed according to ASHREA standards. It consists of two rooms made of adiabatic material of rigid polyurethane, one is indoor side room and another is outdoor side room. The outside room ambient is controlled from 25°C to 55°C. While indoor side has either temperature control or load control. For testing or air conditioner, maintaining of temperatures is important so these two rooms are made of adiabatic material. The method use in air conditioner test room for finding capacity is air enthalpy method. These two rooms are separated by an opening in to which the non-ducted equipment (test air conditioner) is mountedwhich the non-ducted equipment (test air conditioner) is mounted.

The equipment shall be installed in such amanner similar to a normal installation. We can test up to 4 TON refrigeration capacity in our test room. The air conditioner evaporator coil is in indoor side and condenser coil is in outdoor side. There should not be any leakage between indoor side and outdoor side.The equipments associated with air conditioner test room as follows:

(a) Panel Boards
(b) Split AC Indoor Unit Installation
(c) Split AC Outdoor Unit Installation
(d) Packaged Air conditioners
(e) Humidifiers
(f) Code Tester
(g) Temperature measuring instrument
(h) Heaters
(i) Sampling device

3. Cooling Capacity Test and Calculations

Test Conditions

The test conditions that should be maintained are as follows:

ISO 5151 standard

Indoor side		Outdoor side	
DBT °C	WBT °C	DBT °C	WBT °C
27	19	35	24

Table 1: Physical Properties of R22, R407c and R410a

Property	R22	R407C`	R410A
Replaces	N/A	R-22	R-22
ChemicalFormulae/composition	$CHClF_2$	R32/R125/R134a $CH_2F_2/CHF_2CF_3/CH_2FCF_3$ 12/25/52 % weight	R32/R125 CH_2F_2/CHF_2CF_3 50/50 % weight
Molecular Weight	86.47	86.2	72.58
Boiling Point at1atm^{o}C	-40.8	-51.53	-43.56
Critical TemperatureTe in ^{o}C	96.24	86.74	72.13
Critical Pressure inkpa	4981	4619	4926.1
ODP	0.05	0	0
GWP	2000	1420	1350

Procedure

Out let of air conditioner (which is going to be tested) is attached to the receiving chamber of the code tester through the proper ducting.

- Before starting the main switch, ensure that all the switches of panel Board are in disable or not.
- Before starting the test, clean all the sensors and fill water in all sampling devices.
- Set the temperatures values, which are going, to be maintained in PID'S of the panel boards
- Switch on the test unit by adjusting dimmer to 230 volts.
- Switch on the humidifier ad packaged air conditioner of both the rooms.
- Switch on the heaters that are controlled by panel boards.
- Switch on the temperature measuring instruments and computer to record the data. Sets scan control of temperature measuring instrument to 30 seconds in Data logger software, so that it scans all the temperatures for every 30 seconds.
- Wait for some time for stabilizing the room.
- If room is not stabilized and temperatures obtain is less then set value, then switch ON the Extra heater and adjust the duct dampers to get the required temperature.
- If we are getting more temperature than the set value by using extra heater then switch off one heater among three of each 3 KW in the duct that are controlled by the panel boards
- If temperatures are stabilized i.e. the temperature should have ±1°C difference with, set value or arithmetic mean should have ±0.3°C for DBT and ±0.2°C for WBT.**For example:** we are maintaining DBT–35°C, WBT-24°c in outdoor side and DBT-27°C, WBT-19°C indoor side (according to ISO standard).
- When temperature is stabilized there arithmetic mean value should have
- After stabilization, to maintain the test conditions for 4 hours, and record the data after 30 minutes so that there are 7 set of readings for every 5 min.
- During this 30 min duration, record the code tester nozzle pressure drop. When receiving chamber pressure shows 0.0 and record code tester DBT and WBT readings also.

The sensor temperature that should be recorded are:

102 Indoors side DBT
103 Indoors side WBT
104 Outdoors side DBT
105 Outdoors side WBT
106 Indoor side Code tester DBT
107 Indoor side Code tester WBT

Average the recorded data. So that we can get more appropriate one value.

For convenience not indoor side room temperature from 102 and 103 sensors and leaving air condition from 106 and 107 sensors.

For calculation of CFM:

$Cfm = C_i * A_i * Y_i * \sqrt{2D_p/p} * 3600 * 0.5885$

Where
ρ=density of air at the air sampling condition kg/m^3.
C_i=Discharge co-efficient of its nozzle dimensionless.
Ai=Area of the nozzle in m^2.
Yi=expansion factor, dimensionless.
Dp=pressure drop across the nozzle in Pa.

C_i, Y_i can be calculated from ASHRAE 41

Formulae for cooling capacity calculations:

Volume Flow rate of air Q_{va}(m^3/sec)= Cfm/2118.88

Enthalpy difference D_h= Enthalpy of moist air entering - Enthalpy of moist air leaving
Mass flow rate of air M_a(kg/s) =Volume flow rate of air/Specific volume of air
Cooling capacity in KW =Mass flow rate of air*Enthalpy difference
Cooling capacity in Btu/hr=Cooling capacity in KW*3412.14
Cooling capacity in Ton of refrigeration=Cooling capacity in KW/3.5167

Sample Calculations

Refrigerant R-22

Outdoor side conditions ^oC		Outdoor side conditions ^oC	
DBT	35±0.3	DBT	27±0.3
WBT	24±0.2	WBT	19±0.2

1. Condenser temperature (Tc - in^{o}C) : Tc = 54.5
2. Condenser Pressure (Pc – in bar): Pc=18.89
3. Evaporator Temperature (Te- in^{o}C) : Te=7.2
4. Evaporator Pressure(Pe – in bar) : Pe=4.89

Pressure ratio (Pc/Pe) = 18.89/ 4.95 = 3.816

Readings Note down from Laboratory:

Indoor Air Temperature
DBT in ^{o}C = 27.08
WBT in ^{o}C = 18.94
Outdoor Air Temperatures
DBT ^{o}C = 35.01
WBT in ^{o}C = 24.98
Leaving air Temperatures
DBT in ^{o}C = 16.21
WBT in ^{o}C = 11.53

Specific volume of Air =0.831m^3/kg(From psychrometric chart at leaving condition)

$$\text{Density of Air} = \frac{1}{Specific\ volume\ of\ air}$$

$$= \frac{1}{0.831} = 1.2033\ kg/m^3$$

Area of Nozzle

$$Ai = \frac{\pi}{4}D^2 = \frac{\pi}{4}(100/1000)^2$$
$$= 0.00785m^2$$

Cubic feet per minute

CFM=Ci*Ai*Yi* SQRT(2*Dp)/ρ)*3600*0.5885
=0.985 * 0.998 * 0.00785 * SQRT ((2*456)/ 1.2033)) * 3600 * 0.5885
= 450.10

Volume flow rate (Qv) =Cfm/2118.88
= 450.10 /2118.88
= 0.2127m^3/sec

Mass Flow rate of Air= volume flow rate/ Specificvolume
= 0.2156/0.925
=0.2556kg/sec
Power (Watts) = 1951
Entering Air Enthalpy = 54.56 kJ/kg(taken from psychrometric chart at Indoor air temperature
i.e. 27.02^{o}C DBT,18.94^{o}CWBT)
Leaving Air Enthalpy=33.5 kJ/kg (taken from psychrometric char

At leaving air temperature

i.e. 16.21^{o}C DBT, 11.53^{o}CWBT)

Enthalpy Difference = Entering Air Enthalpy (h_e)
– Leaving Air Enthalpy (h_{le})
= 54.56 – 33.5
= 21.06 kJ/kg

Cooling Capacity in KW =
Mass flow rate of air * Enthalpy difference
= 0.2556 * 21.06
= 5.38
Cooling Capacity in Btu/hr =
Cooing capacity in KW * 3412.14
= 5.38 * 3412.14
= 18357.31
Energy Efficiency Ratio =
(Cooling capacity in KW)/ (Input Power in KW)
= 5.38/ 1.951
= 9.409
COP of system =Energy Efficiency Ratio / 3.412
= 9.409/3.412
= 2.75

4. Results and Conclusions

From the above calculations we can observed that the performance of Air conditioning system of capacity 1.5 Tones has been evaluated experimentally with R22, R407C and R410A. It can be stated that the Performance parameters Refrigerant mass flow rate, compressor work, Cooling Capacity, Energy Efficiency Ratio (EER) and COP are given below in Table 7.1

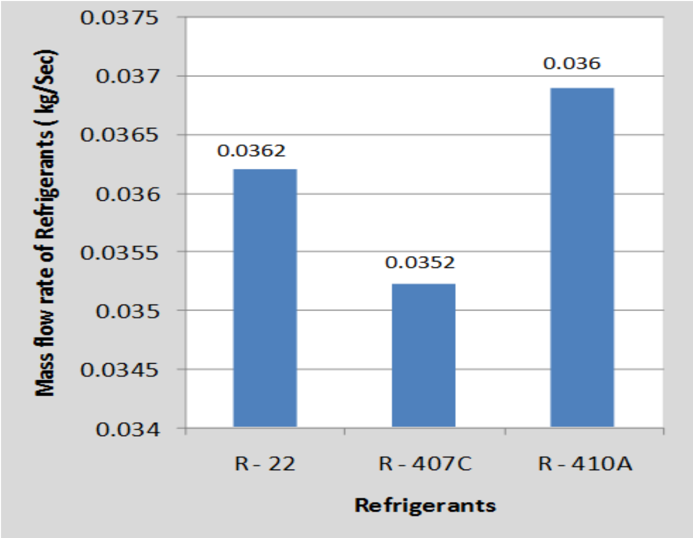
This Experimental test has been conducted on split air conditioning system by maintaining indoor and outdoor air conditions at ISO5151 standard i.e. indoor at 27^{o}CDBT, 19^{o}CWBT and 34^{o}CDBT, 24^{o}C WBT at outdoor respectively. And the suction and discharge pressures are taken at global test condition i.e. evaporator temperature at 7.2^{o}C and condenser temperature at 54.5^{o}C

Table 2:comparison of Performance parameters

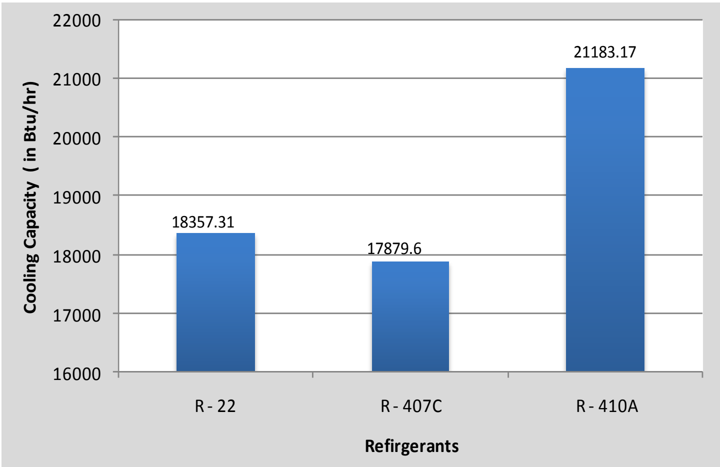
Performance parameters	R-22	R-407C	R-410A
Refrigerant mass flow rate	0.0362	0.0352	0.0369
Compressor Power (KW)	1.951	2.028	1.998
Cooling Capacity(KW)	5.38	5.24	6.208
Heat Rejection in Condenser (KW)	6.66	6.799	6.47
EER	9.41	8.81	10.47
COP	2.75	2.58	3.07

The graph 1 is the plot of Mass flow rate of refrigerant with respect to the Refrigerants R22, R407C and R410A at standard global condition i.e. evaporator temperature at 7.2oC and condenser temperature at 54.5oC. The figure indicates that increase of mass flow rate for R410A refrigerant when comparing with R22 and R407C

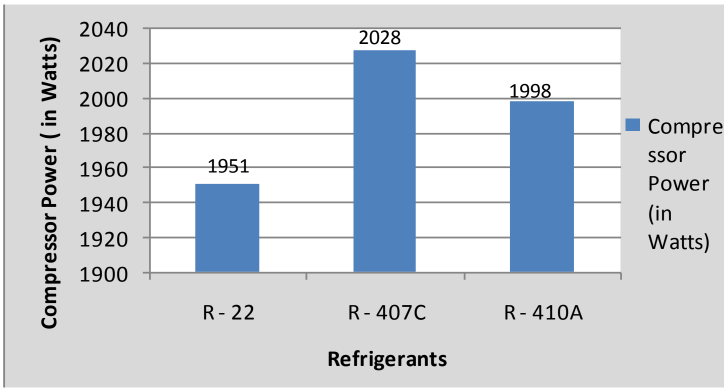
The graph 2 is the plot of cooling capacity with respect to the Refrigerants R22, R407C and R410A at standard global condition i.e. evaporator temperature at 7.2oC and



Graph 1: Refrigerants Vs Mass flow rate



Graph 2: Refrigerants Vs Cooling Capacity

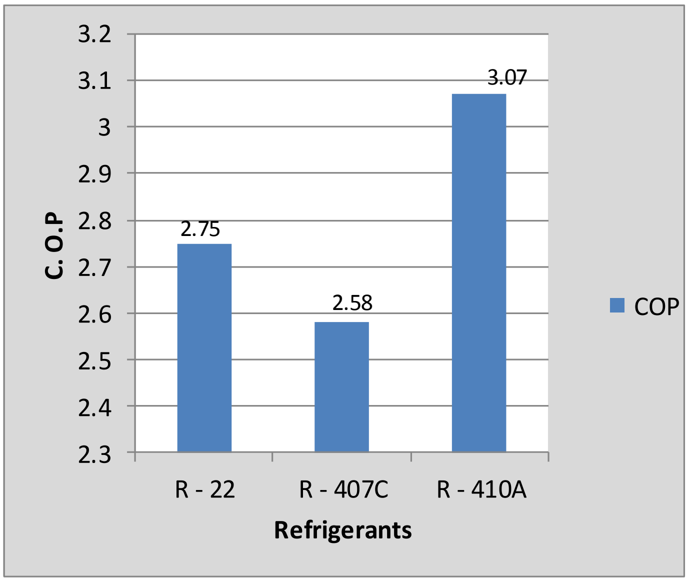


Graph 3: Refrigerants Vs Compressor Power

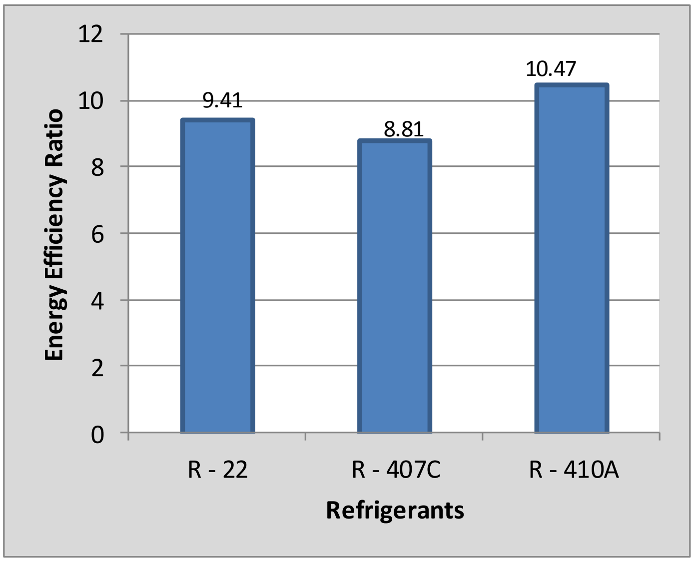
condenser temperature at 54.5oC. The figure indicates that the compressor work is more for the Refrigerant R407C, when comparing with R22 & R 410A.

The graph 3 is the plot of compressor work with respect to the Refrigerants R22, R407C and R410A at standard global condition i.e. evaporator temperature at 7.2oC and condenser temperature at 54.5oC. The figure indicates that the compressor work is more for the Refrigerant R407C, when comparing with R22 & R 410A.

The Graph 4 is the plot of Co-efficient of Performance with respect to the Refrigerants R22, R407C and R410A at standard global condition i.e. evaporator temperature at 7.2oC and condenser temperature at 54.5oC. The figure indicates that the COP is more for the Refrigerant R410A



Graph 4: Refrigerants Vs COP



Graph 5: Refrigerants Vs EER

is more for when comparing with R22 & R 410A.

The Graph 5 is the plot of Energy Efficiency Ratio with respect to the Refrigerants R22, R407C and R410A at standard global condition i.e. evaporator temperature at 7.2oC and condenser temperature at 54.5oC. The figure indicates that the Energy Efficiency Ratio is more for the Refrigerant R410A is more for when comparing with R22 & R 410A.

Conclusions

From the experimental analysis it can be concluded that the Refrigerants R22, R407C and R410A were tested on a split air conditioning system of capacity 1.5 Tones at ISO 5151 standard. Where the air temperatures are maintained at 35oC DBT, 24oC WBT at outdoor side and 27oC DBT, 19oC WBT at the indoor side. At this standard condition the important parameters suction pressure, discharge pressure and Evaporator temperature; Condenser pressures are similar for R22 and R407C, where as for R410A it is slightly more comparing with R22.From the above results the performance parameters are concluded below.

- Cooling Capacity: The cooling capacity of the system is 2.7% less for R407C when compare with R22 and 15% more for R410A.
- Energy Efficiency Ratio: The Energy Efficiency Ratio of the system is 6.4% less for R407C comparing with R22, whereas for R410A it is 11.2% more.
- Compressor Work: Compressor work for R22 and R407C are almost similar, it is almost 6% more for R407C, for R410A it is 3% more when compare with R22.
- Refrigerant Mass flow Rate: the refrigerant mass flow rate is 3% less for R407C when compare with

R22, but whereas for R410A it is increasing slightly about 1.9%.
- COP: The Co-efficient of performance of the system with for R407C is 6.2% less when compare with R22, but for R410A it is 11.6% more comparing with R22.

But the above performance parameters variations are quite small for three refrigerants, hence their use is recommended as alternatives for R22 refrigerant. Finally it can be concluded that R410A is a better substitute to R22 than R407C in an air-conditioning system.

References

American Society Heating Refrigeration and Air Conditioning.*ASHRAE Hand Book*. 2001

Yumrutas R, Kunduz M, Kanoglu M. (2002) Exergy analysis of vapor compression refrigeration systems.Exergy, An international Journal;2(4);266-72.

Wernick, B.J (2004) Effectiveness method, RACA journal.

C.P Arora, Refrigeration and Air conditioning. Tata McGraw-Hill Book Company

Akintunde, M.A. (2004b). Experimental Investigation of The performance of Vapor Compression Refrigeration Systems.Federal University of Technology, Akure, Nigeria.

Granryd, E. (Nov 2010) Analytical expressions for optimum flow rates in evaporators and condensers of heat pumping systems International Journal of Refrigeration, Volume 33, Issue 7, Pages 1211-1220

Eckert, E.R.G.; Goldstein, R.J.; Ibele, W.E.; Patankar, S.V.; Simon, T.W.; Strykowski, P.J.; Tamma, K.K.; Kuehn, T.H.; Bar-Cohen, A.; Heberlein, J.V.R.; (Sep 1997), Heat transfer-a review of 1994 literature, International Journal of Heat and Mass Transfer40-16, 3729-3804

Stoecker, W.F. and Jones J.W. (1982), Refrigeration & Air Conditioning**.**McGraw-Hill Book Company, Singapore

Technical Manual Air-conditioning application, Tecumseh. Tecumseh Products India Pvt. Ltd., Hyderabad.

Calorimeter Test Facility Lab manual **(**Ctflm) Tecumseh Products India Pvt. Ltd.

S.S. Banwaitand S.C. Laroiya, Properties of refrigerant and psychometric tables and charts