International Journal of Advance Research and Innovation
Vol. 6(3), Jul-Sept 2018, pp. 89-99
Doi: 10.51976/ijari.631814
www.gla.ac.in/journals/ijari

Article Info
Received: 18 Aug 2018 | Revised Submission: 28 Aug 2018 | Accepted: 02 Sept 2018 | Available Online: 15 Sept 2018

Comparative Performance Evaluation of Vapor Compression Refrigeration System for Drop-in Replacement of R22

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ABSTRACT

Present work reports a detailed theoretical and experimental performance analysis of vapor compression refrigeration system (VCRS) using alternative refrigerants to find out the substitute of R22. First, a theoretical investigation is done using R134a, R410A R407C, R438A as the potential drop-in refrigerants for the same setup and compressor displacement volume using a thermodynamic model developed in Engineering Equation Solver (EES) software. The results of this analysis indicate that R438A is a better substitute to R22. Then a similar analysis is done to compare the performance of R22 and R438A for the same cooling capacity and input conditions but different setup and compressor displacement volume. The results indicate that R22 results in a better efficiency. In the end, experimental comparison of R22 and R438A is done on the same experimental set up designed for R22 and the results of first theoretical analysis are verified.

Keywords: *Vapor Compression; Drop-In; Refrigerants; Experimental Work.*

1.0 Introduction

Recently, the ozone depleting potential (ODP) and global warming potential (GWP) have become the most important criteria in the development of new refrigerants apart from the refrigerant CFCs and HCFCs, both of which have high ODP and GWP, due to their contribution to ozone layer depletion and global warming [1]. In spite of their high GWP, alternatives to refrigerant CFCs and HCFCs such as hydro fluorocarbon (HFC) refrigerants with their zero ODP have been preferred for use in many industrial and domestic applications intensively for a decade. HFC refrigerants also have suitable specifications such as non-flammability, stability, and similar vapor pressure to the refrigerant CFCs and HCFCs [2-3]. The problems of the depletion of ozone layer and increase in global warming caused scientists to investigate more environmentally friendly refrigerants than HFC refrigerants for the protection of the environment such as hydrocarbon (HC) refrigerants of propane, isobutene, n-butane, or hydrocarbon mixtures as working fluids in Vapor-compression refrigeration systems (VCRS).

R22 has been predominantly used in residential air conditioners and heat pumps for the past few decades and its sales volume has been the largest among various refrigerants. Even though the ozone depleting potential of R22 is not as high as CFCs, it still contains ozone depleting chlorine and hence the parties to the Montreal protocol decided to phase out R22 eventually and the regulation for the HCFC production has begun from 1996 in the developed countries.

For the past years, various alternative refrigerants for R22 have been proposed and tested in an effort to comply with the Montreal protocol [4].

Recently, ASHRAE listed R438A as a possible candidate to replace R22. R438A is a non-ozone depleting zeotropic blend of HFC 32, HFC 125, HFC134a, HC 600 and HC 601a (8.5/45.0/44.2/1.7/0.6 wt% respectively) [5]. Retrofitting the existing R22 equipment to an alternative refrigerant is a viable, cost effective option for the equipment owner. R438 is a non-ozone

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depleting hydrofluorocarbon (HFC) refrigerant blend which can be used to retrofit existing HCFC refrigerant R22 refrigeration and air conditioning systems with direct expansion evaporators and positive displacement compressors. In most systems, the existing mineral oil (MO) or alkyl benzene (AB) lubricant can be used, reducing the costs and time required for the retrofit by eliminating the need to change to a polyester (POE) lubricant.

2.0 Literature Survey

Allgood et al. [5] reviewed the performance characteristics and oil return properties of R-438A from both laboratory tests and actual R-22 system retrofits including the topics like refrigerant physical and environmental properties, material compatibility information, oil return properties, and capacity, energy efficiency, and operating data from both compressor calorimeter tests and actual R-22 refrigeration and air conditioning system retrofits.

Campagna et al. [6] compared the physical properties and system performance characteristics of the two new fluids R410A & R407C with R22 and attempts to answer the question: "Will R-410A Replace R-407C in the future? Kalla et al. [7] evaluate the performance of refrigerants R22, R407C, R432A, R438A and NM1 (R32/R125/R600a) in order to find a suitable alternative refrigerant for HCFC 22.

Bhargav et al. [8] investigated the performance of the refrigerator using azeotropic mixture of propane and isobutane and compared with the performance of refrigerator when R134a, R12, R22, R290, R600a is used as refrigerant. The effect of condenser temperature and evaporation temperature on COP, refrigerating effect, condenser duty, work of compression and Heat Rejection Ratio where investigated. “No Oil Change” R-22 Replacement Comparison Guide by Chemours [9] suggests that R-22 retrofits. Freon MO99 is U.S. EPA SNAP-approved, compatible with mineral oil and POE, and has the closest performance match to R-22 compared to other “no oil change” replacements.

Lampugnani et al. [10] studied the performances of R290 in comparison with R22, from the theoretical as well as from experimental point of view. The influence of R290 on compressor reliability has been also evaluated analysing the bearing load and considering both the materials compatibility and the oil solubility; the lower operating temperature has a positive impact on compressor reliability. In the present work, the above-mentioned aspects have been considered and a computational model is developed for evaluating COP, mass flow rate of refrigerant, Cooling Effect and compressor work for various alternate refrigerants on system designed for R22. As the quest for finding a suitable alternative refrigerant for R22 in air conditioners goes on, this study carries out performance assessment of some chosen refrigerants. Refrigerants studied are not having any ozone depletion potential. Table 1 depicts the physical characteristics of the potential candidates selected for the replacement of R22.

Engineering Equations Solver [EES] is the software that has been used for the simulation purpose. The refrigerants studied are R-22, R134A, R410A, R438A, R290 and R407C. To the authors’ knowledge, there is no specific work reported on the energy and exergy performance comparison of the above mentioned refrigerants as an alternative to R22 in residential air conditioners.

The main objective of this paper is to investigate which of these five refrigerant blends can be a potential drop-in alternative to R22. Literature survey emphasizes that most promising alternate refrigerants for R22 are R438A and R290 as they can be used as the drop-in replacements in the systems designed for R22. Both of them provide comparable performance to R22 under similar operating conditions and both of them are having zero ODP.

Table 1: Physical And Environmental Characteristics of Selected Refrigerants [3, 11]

Properties	Refrigerants					
	R22	**R438A**	**R290**	**R407C**	**R134a**	**R410A**
ODP	0.055	0	0	0	0	0
GWP	1810	2265	3	1774	1430	2088
Molecular weight	86.5	94.1	44.1	86.2	102.03	72.6
Normal boiling point	-40.8	-42	-42.1	-43.6	-26	-48.5
Critical	96	85	96.8	86.7	101.1	72.8

Temp. (OC)						
Critical Pressure (bar)	40.7	43	42.5	46.7	40.67	48.6
Latent Heat (At 4OC)	201.78	175.46	369.17	213.16	195.52	221.25
Flammability, category	Non-Flammable, A1	Non-Flammable, A1	Flammable, A3	Non-Flammable, A1	Non-Flammable	Non-Flammable
Lubricant	Mineral oil/alkyl benzene	Mineral oil/alkyl benzene/POE	Mineral oil/alkyl benzene	Polyol ester	Polyalkaline glycol	Polyvinyl ether

Therefore, the main objectives of present work are as follows:

- To develop thermodynamic model of VCRS in EES using mass and energy equations.
- To do simulation for wide range of operating conditions.
- To find out best substitutes of R22 theoretically and also by experimentally substituting the refrigerants in system designed for R22.

3.0 Thermodynamic Analysis

In vapor compression refrigeration cycle from the energy analysis point of view, the measure of performance of the refrigeration cycle is the coefficient of performance (COP), which is defined as the net refrigeration effect produced per unit of work required.

3.1 Assumptions

The energy analysis presented in this work is based on the following relevant assumptions:

- System is in steady state.
- Clearance ratio (R) of the compressor used is 2.5%.
- The efficiency of the condenser and evaporator is 100%.
- There is no enthalpy change across the expansion device.
- The entropy remains constant across the compressor.

3.2 Vapor compression refrigeration cycle

Fig 1: Vapor Compression Refrigeration System

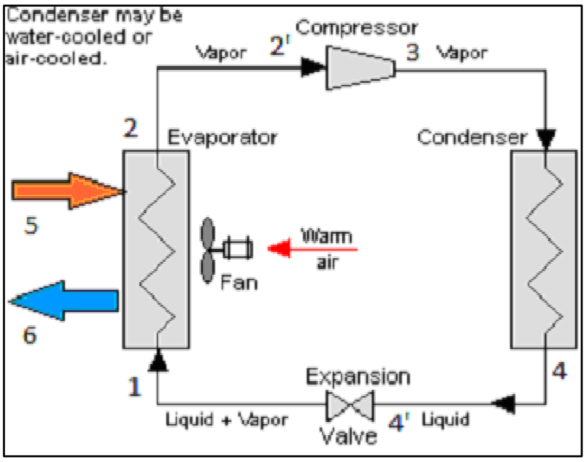
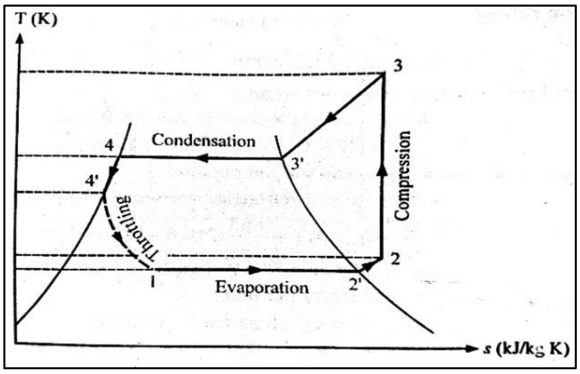


Fig 2: T-S Diagram of VCRS



The vapor-compression uses a circulating liquid refrigerant as the medium which absorbs and removes heat from the space to be cooled and subsequently rejects that heat elsewhere. Fig. 1 depicts a typical, single-stage vapor-compression system; whereas, Fig. 2 shows corresponding T-s diagram. This system has four components: a compressor, a condenser, a thermal expansion valve (also called a throttle valve or metering device), and an evaporator. Circulating refrigerant enters the compressor in the thermodynamic state known as a saturated vapor and is compressed to a higher pressure, resulting in a higher temperature as well.

The hot, compressed vapor is then in the thermodynamic state known as a superheated vapor and it is at a temperature and pressure at which it can be condensed with either cooling water or cooling air flowing across the coil or tubes.

This is where the circulating refrigerant rejects heat from the system and the rejected heat is carried away by either the water or the air (whichever may be the case).

3.3 Energy analysis

Evaporator: Evaporator abstracts the heat (Q_e) from the space maintained at temperature T_r and it is given by Equation (1):

$$Q_e = \dot{m}(h_2 - h_1) \quad (1)$$

m: mass flow rate of refrigerant (kg/s)

h_2: enthalpy at outlet to evaporator (kJ/kg)

h_1: enthalpy at inlet to evaporator (kJ/kg)

Compressor: The isentropic efficiency of the compressor is given by

$$N_V = 1 - R[\frac{V_3}{V_2'} - 1] \quad (2)$$

$$N_V = \frac{\dot{m}V_2'}{P_d} \quad (3)$$

N_v: Isentropic compressor efficiency

V_2': Suction volume (m^3)

V_3 : Discharge volume (m^3)

The isentropic work (W_{cs}) input to the compressor is given by Equation (4):

$$W_{cs} = \dot{m}(h_3 - h_2') \quad (4)$$

h_3 : enthalpy at outlet of compressor (kJ/kg)

h_2': enthalpy at inlet to compressor (kJ/kg)

Condenser: The heat rejected by the condenser (Q_{cond}) to the atmosphere is given as:

$$Q_{cond} = \dot{m}(h_3 - h_4) \quad (5)$$

h_4: enthalpy at outlet to condenser (kJ/kg)

h_3: enthalpy at inlet to condenser (kJ/kg)

Throttle valve: In throttle valve, the enthalpy remains constant.

According to the first law, the measure of performance of the refrigeration cycle is the COP and is equal to the net refrigeration effect produced per unit of work required. It is expressed as Equation (6):

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

Q_e: Evaporator heat W : Compressor work

Also the tonnage of the system in kilowatts is calculated as:

$$Ton = \frac{Q_e}{3.5} \quad (7)$$

4.0 Results and Discussion

4.1 Theoretical comparison of refrigerants on the same system

A computational model is developed for carrying out the energy analysis of different alternate refrigerants on the same system using Engineering Equation Solver software.

The input data assumed for the computation of is furnished below:

1. Displacement volume of the compressor P_d is 0.001632
2. Evaporator temperature $T_e$is 4 ^oC.
3. Condenser temperature T_c 50 ^oC
4. Clearance Ratio R of the compressor is 2.5%.
5. There is no enthalpy change across the expansion device.
6. The entropy remains constant across the compressor.
7. The state of the refrigerant is assumed to be pure vapor in the suction line and pure liquid after the condensation.
8. Heat transfer across the condenser and evaporator is 100% efficient.

Theoretical performance comparison of various alternate refrigerants for the same system, i.e. for the same compressor displacement and other design parameters as designed for R22 provides us the following conclusions as given in Table 2.

Table 2: Theoretical Performance of Various Refrigerants

Properties	Refrigerant					
	R22	R290	R438	R134a	R410a	R407c
Comp Work (kJ)	1.14	0.9718	0.66104	0.7095	1.775	1.172
Compressor Pressure(kPa)	1943	1713	2077.759	1319	3061	2199
Evap Pressure (kPa)	566.2	535.2	510.79	337.9	902.2	521.8
Pressure ratio	3.432	3.201	4.0677	3.902	3.393	4.215
Mass flow rate (kg/s)	0.03693	0.01792	0.037801	0.02509	0.05372	0.0327

Cooling effect (kJ)	5.2507	4.33 09	4.10 367	3.24 39	7.35 588	4.41 18
Isen. Comp eff (m3/Kg)	0.9474	0.94 5	0.92 43	0.92 86	0.94 58	0.92 45
COP	4.606	4.45 7	6.16 81	4.57 2	4.14 4	3.76 6
Tonnage (Tonne)	1.5	1.23 7	1.17 24	0.92 69	2.10 2	1.26
Discharg e Temp	69.86	54.6 1	52.8 4	54.7 7	68.3 6	69.1 4

Fig 3: Comparison of Compressor Work

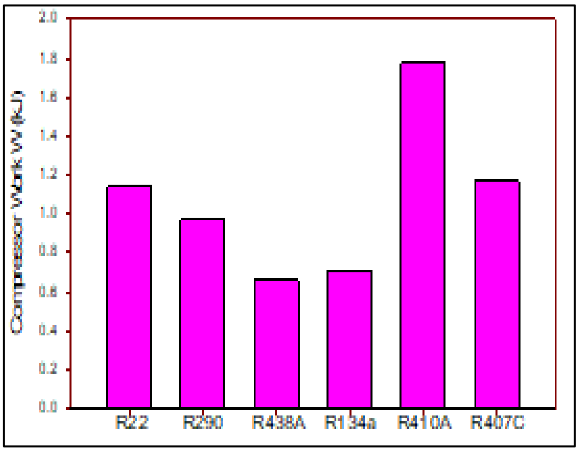
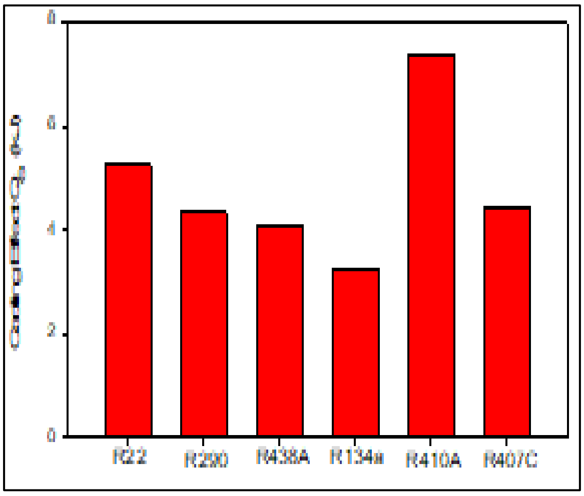


Fig 4: Comparison of Cooling Effect



4.1.1 Compressor work

Compressor work is a major component in measuring the energy efficiency of a system. After comparing various refrigerants on the basis of their compressor work, we imply from Table 2 and Fig. 3 that compressor work for R438A is minimum i.e., 42% less than that for R22 and maximum for R410A i.e., 55.70% more than that for R22. Hence, R438A is a more power efficient alternative to R22.

4.1.2 Cooling effect

The cooling effect is as important as COP in refrigeration. Fig. 4 and Table 2 show the evaporator cooling effect, Qe for air-conditioning of various alternate refrigerants as compared to R22 for a given compressor. R410A showed 40% higher capacity while R134a showed 38% lesser capacity than R22. Also, R438A showed 21% lesser cooling effect than R22.

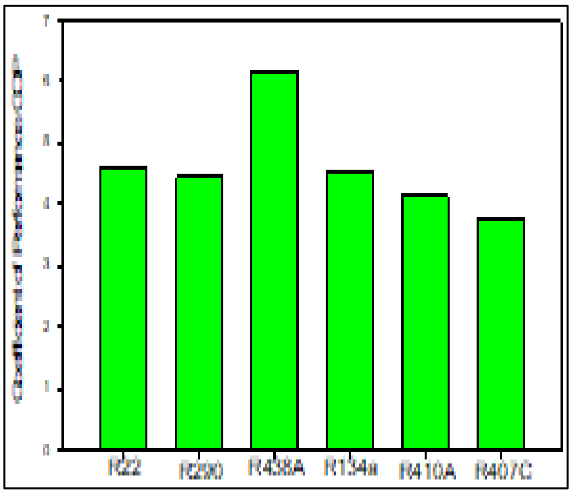
4.1.3 Coefficient of performance

In refrigeration and air-conditioning, COP is a measure of energy efficiency for a given device charged with a specificrefrigerant. Fig. 5 shows the COPs of all the alternate refrigerants for air-refrigerants for air-conditioning.

As seen in this figure, the COP of R438A is 34% higher while the same for R407C is 18% lower than that of R22. One of the reasons for the improved efficiency of R438A is the decrease in compressor work. As listed in Table 2 and Fig. 3 the compressor work of R438A decreased 42% as compared to that of R22.

Test results demonstrate that R438A is a good alternative to replace R22 in air-conditioners from the standpoint of energy efficiency.

Fig 5: Comparison of Coefficient of Performance

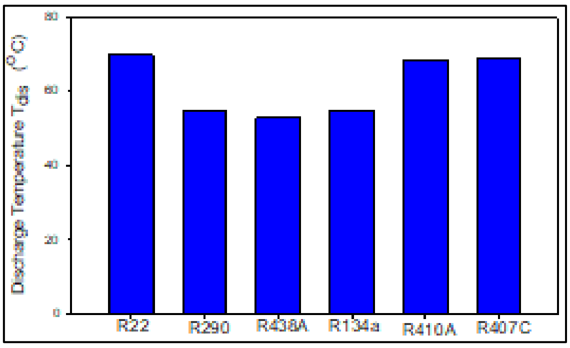


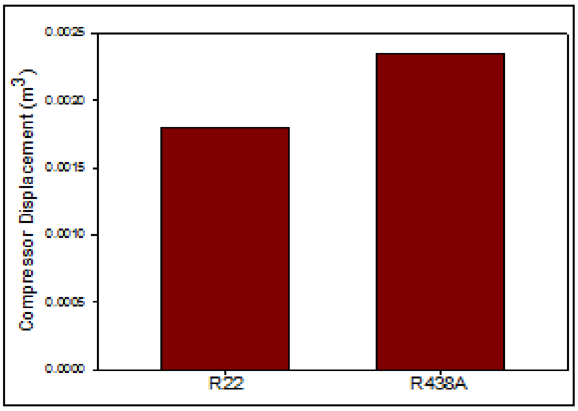
4.1.4 Compressor discharge temperatures

The lifetime and reliability of the system as well as the stability of the refrigerant and lubricant should be considered when alternative refrigerants are considered. These characteristics can be examined indirectly by measuring the compressor discharge temperature (T_{dis}).

Fig. 6 shows the compressor discharge temperatures of R22 and other alternate refrigerants for air-conditioning conditions. As depicted in Table 2 and Fig. 6, R438A showed 24 % decrease in discharge temperature when compared to R22. From this observation, it can be safely concluded that R438A would be appropriate from the viewpoint of system reliability and refrigerant.

Fig 6: Comparison of Discharge Temperature Stability.

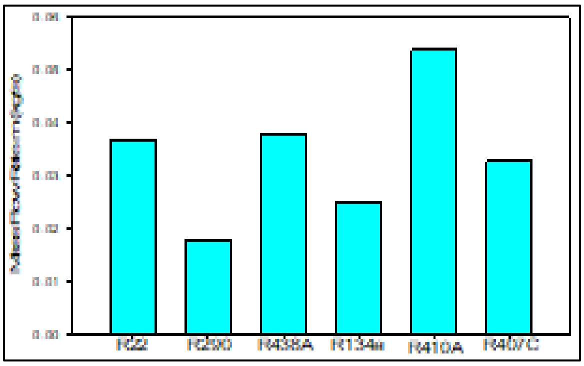




4.1.4 Mass flow rate

Mass flow rate of the refrigerant governs both the Cooling Effect and the compressor work. Table 2 and Fig. 7 shows that the mass flow rate for R410A is 45% higher while that of R290 is 51% lower than R22. While Mass flow rate for R438A is 2.35% higher. Hence Mass flow rate of R438A and R22 is almost comparable.

Fig 7: Comparison of Mass Flow Rate



While all the alternate refrigerants other than R290 and R438A require compressor lubricant change, these two refrigerants are compatible with the existing mineral oil lubricant in the R22 system. Also these two refrigerants do not require any major system design changes unlike the other refrigerants. The only negative characteristic, typical of the hydrocarbons, of R290 is, obviously, the flammability. The risk associated to the explosion possibility, related to hermetic compressors cannot be ignored for domestic appliances. An additional precaution is the elimination of the starting relay and use for propane applications only PSC motor compressor. In case of real need of a High Starting Torque Compressor, the opportunity of an explosion proof enclosure for electrical components should be considered.

Test results indicate that R438A is a good ‘drop-in’ replacement without requiring major changes in major components including the compressor.

In fact, resizing and redesigning of compressors is very costly and the ‘drop-in’ feature of R438A is very advantageous from the viewpoint of manufacturing cost and efficiency of the system. But the energy efficiency comes at the cost of the Cooling Effect while using R438A as it decreases by 21%. Therefore, further analysis is done for the same Cooling Effect for different systems and comparison is done taking R438A as the only potential candidate for substitution of R22.

A computational model is developed for carrying out the energy analysis of R438A and R22 for some cooling capacity on different systems, i.e. different compressor displacements and other design parameters, using Engineering Equation Solver software.

4.2 Theoretical comparison of refrigerants on different systems for same cooling capacity

Table 3: Theoretical Performance of R22 and R438A

Properties	Refrigerant	
	R22	R438A
Compressor Work (kJ)	1.279	1.5334
Compressor Pressure (kPa)	2048.8	2163.65
Discharge Temperature (^{o}C)	77.15	66.09
Evaporator Pressure (kPa)	516.74	463.38
Pressure Ratio	3.96	4.67
Mass Flow Rate (kg/s)	0.03635	0.04763
Specific Cooling Effect (kJ)	144.4	110.2033
Compressor Displacement (m^3)	0.001798	0.0023418
COP	4.118	3.4229

The input data assumed for the computation of results are furnished as-

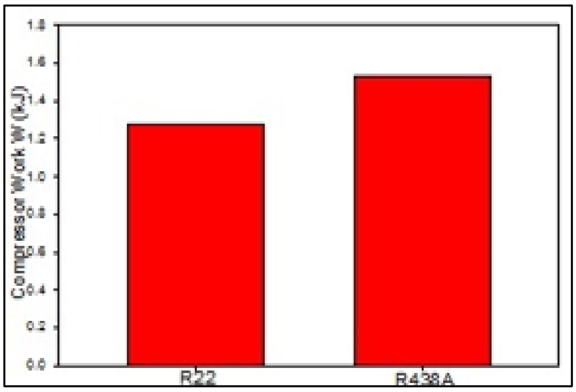
1. Cooling capacity of the system 5.25 KW or 1.5 Tonne.
2. Evaporator temperature $T_e$is 4 ^{o}C.
3. Condenser temperature Tc 50 ^{o}C
4. Clearance Ratio R of the compressor is 2.5%.
5. There is no enthalpy change across the expansion device.
6. The entropy remains constant across the compressor.
7. The state of the refrigerant is assumed to be pure vapor in the suction line and pure liquid after the condensation.
8. Heat transfer across the condenser and evaporator is 100% efficient.

Theoretical performance comparison of R438A and R22 for the same cooling capacity but different compressor displacements provides us the following conclusions:

4.2.1 Compressor work

Compressor work is a major component in measuring the energy efficiency of a system. After comparing the compressor work required by R22 and R438A to produce the same cooling effect, we imply from Table 3 and Fig. 8 that compressor work for R438A is approximately 20% more than that for R22. The reason for the increase in the compressor work can be the increase in the compressor displacement volume and the mass flow rate required and the same is explained below.

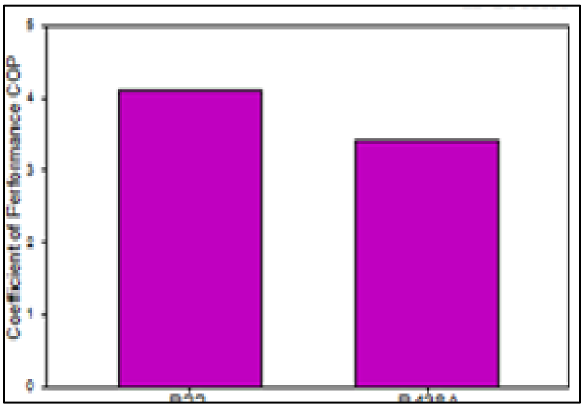
Fig 8: Comparison of Compressor Work



4.2.2 Coefficient of Performance

Fig. 9 shows the COPs of all the alternate refrigerants for air-conditioning. As seen in this figure, the COP of R438A has decreased than that of R22 and it is approximately 17% lesser. One of the reasons for the decreased efficiency of R438A is lesser specific cooling effect because of which more mass flow rate of refrigerant in required which leads to an increase in compressor work.

Fig 9: Comparison of Coefficient of Performance



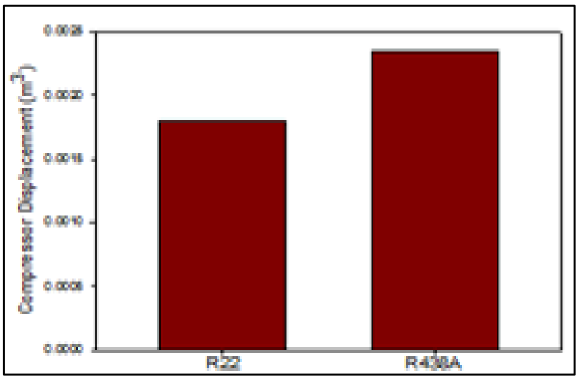
4.2.3 Compressor displacement volume

Compressor displacement volume decides the compressor work which is further required for measuring the energy efficiency of a system.

After comparing the compressor displacement volumes for R22 and R438A required to produce same cooling effect, we imply from Table 3 and Fig.

10 that displacement volume required for R438A is 30% more than that for R22 thus requiring more work to be done by the compressor.

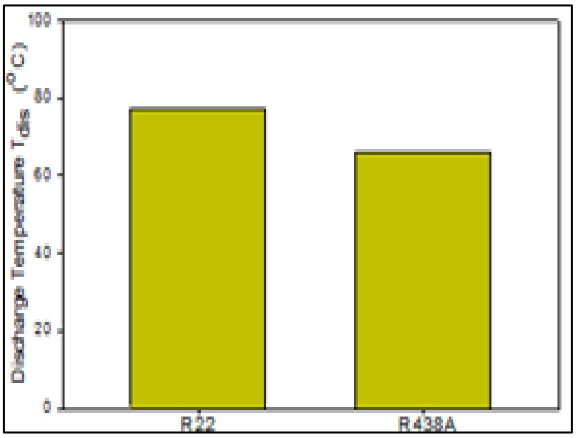
Fig 10: Comparison of Compressor Displacement



4.2.4 Compressor discharge temperatures

Fig. 11 shows the compressor discharge temperatures of R22 and R438A for air-conditioning conditions. As depicted in Table 3 and Fig. 11, R438A showed approximately 15% decrease in discharge temperature when compared to R22. This decrease on temperature contributes to system reliability and refrigerant stability and a similar trend was observed in the previous analysis.

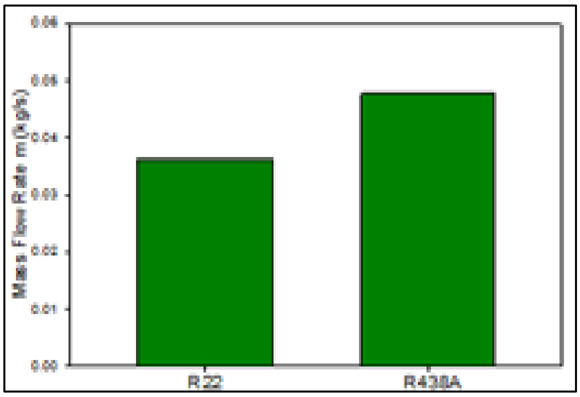
Fig 11: Comparison of Discharge Temperature



4.2.5 Mass flow rate

Mass flow rate of the refrigerant has a direct effect on both the Cooling Effect and the compressor work of the system. Table 3 and Fig. 12 shows that the mass flow rate for R438A is 31% than that of R22. Higher mass flow rate is required to produce a similar cooling effect because of 13% lesser latent heat of R438A than that of R22. This increase in mass flow rate of the refrigerant results in an increase in the compressor work required to produce similar cooling effect.

Fig 12: Comparison of Mass Flow Rate



This analysis shows us that it may not be much beneficial from the point of view of energy efficiency to use R438A in new systems of similar cooling capacity as the COP of the system has reduced by 17% than that of R22. The reason for the same is lower latent heat of R438A because of which approximately 31% more mass flow rate of refrigerant is required to produce a similar cooling effect which further results into 30% more compressor displacement and 20% more compressor work. But using R438A in the existing systems designed for R22 has proved to be beneficial from both the aspects of environmental protection and energy efficiency. Both these benefits come at a cost of 21% lesser Cooling Effect than R22. Further experimental analysis is done to compare the performance of R22 and R438A on the same system designed for R22 to verify the theoretical results from previous analysis.

4.3 Experimental study

4.3.1 Setup description

The window air-conditioner is composed of the basic components of a vapor compression refrigeration system: a hermetically sealed reciprocating compressor, a condenser, a capillary tube and an evaporator, and such attachments as accumulator and fans. A 1.5 tonne window air conditioner was chosen as the experimental setup and to measure the refrigerant pressures, the unit was fitted with four pressure gauges out of which one was compound pressure gauge fitted on the suction line (Fig. 13). Also for temperature measurement, eight thermocouples were fitted at various positions across the system. Pressure gauges have a least count of 10 psi while the compound gauge has the same of 2 psi. Thermocouples have an accuracy of ±0.1 ◦C.

Fig 13: Different Views of Experimental Set Up.

4.3.2 Experimental comparison

The input data assumed for the computation of experimental results is furnished below:

1. Displacement volume of the compressor P_d is 0.001632
2. Clearance Ratio R of the compressor is 2.5%.
3. The temperature and pressure readings displayed by the thermocouples and pressure Gauges are true and accurate.

Experimental performance comparison of R438A was done in comparison to R22 on the system designed for R22 under similar operating conditions for different cooling loads, i.e. Low cooling load (Low Cool) and High cooling load (High Cool) by changing the blower speed. The following results were obtained:

Table 4: Experimental Performance of R22 and R438

Properties	Low cool	High cool	Low Cool	High Cool
Compressor Work (kJ)	0.6808	0.573	0.41202	0.34554
Discharge pressure (kPa)	2068.42	1999.47	2137.374	2068.427
Discharge temperature (0C)	96	92	64	62
Evaporator Pressure (kPa)	372.317	372.317	358.527	358.527
Pressure Ratio	5.555	5.37	5.961	5.769
Mass Flow Rate (kg/s)	0.01985	0.01984	0.02244	0.02232
Cooling Effect (kJ)	3.684	3.601	2.785	2.902
Tonage (Tonne)	1.053	1.029	0.7956	0.8291
Isentropic Compression efficiency	0.8984	0.8994	0.8596	0.8551
COP	5.414	6.284	6.759	8.402

4.4.1 Compressor work

After comparing various refrigerants on the basis of their compressor work, we imply from Table 4 and Fig. 14 that compressor work for R438A is 39% less than that for R22 as expected from the theoretical analysis.

4.4.2 Cooling effect

Fig. 15 and Table 4 show the evaporator Cooling Effect of both the refrigerants. R438A showed 24% lower capacity in comparison R22 as can be seen from the previous results.

Fig 14: Comparison of Compressor Work

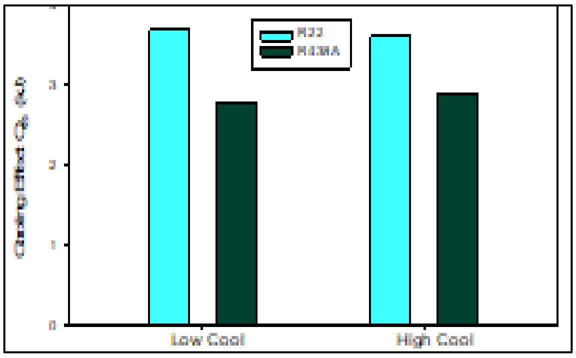
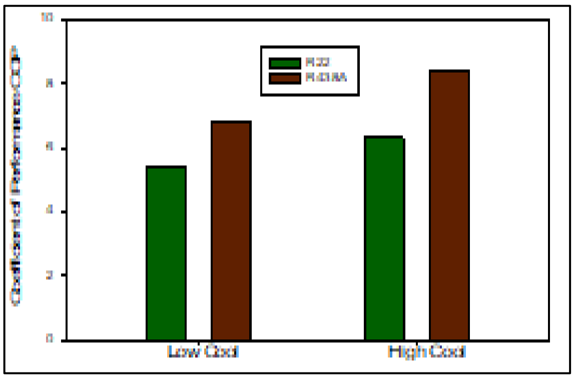


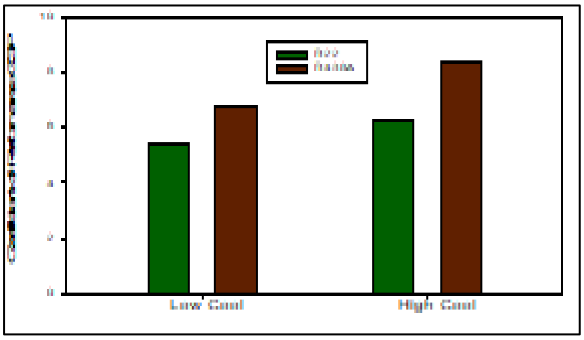
Fig 15: Comparison of Cooling Effect



4.4.3 Coefficient of performance

Fig. 16 shows the experimental COPs of R438A and R22. As seen in this figure, the COP of R438A is 16-24% higher (The variation results due to varying cooling load conditions) than R22. The same can be verified from the theoretical analysis.

Fig 16: Comparison of Coefficient of Performance



4.4.4 Discharge temperature

Fig. 17 shows the compressor discharge temperatures of R22 and R438A for similar operating pressure conditions. As depicted in Table 4 and Fig. 17, R438A showed 33% decrease in discharge temperature when compared to R22. A similar trend was depicted inthe theoretical analysis.

Fig 17: Comparison of Discharge Temperature

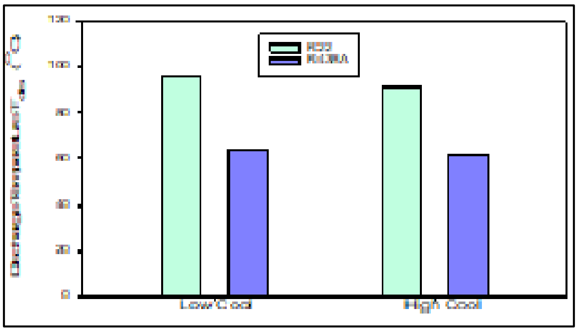
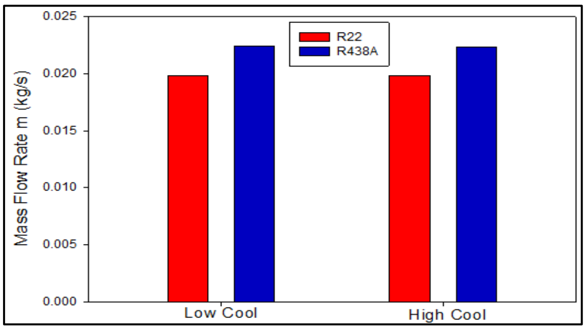


Fig 18: Comparison of Mass Flow Rate



4.4.5 Mass flow rate

Fig. 18 and Table 4 show the mass flow rate comparison of R22 and R438A for varying cooling load conditions. As depicted in Fig. 18, R438A showed 13% increase in mass flow rate when compared to R22. A similar trend was depicted in the theoretical analysis.

Suction and discharge pressures and in field retrofits have been similar to those measured in laboratory tests. In most systems, no system pressure or temperature set point changes are needed.

After performing energy analysis of all the studied refrigerants, R438A was found to be the promising alternative to R22. The energy performance of R438A has been compared to R22 theoretically as well as experimentally. The results point out that the energy efficiency of R438A is higher than that of R22. However, higher efficiency for R438A comes at a cost of reduced Cooling Effect. At high cooling load conditions, COP of R438A is higher by 24% while at low cooling load conditions it is higher by 16%.

5.0 Conclusions

In the present work, an extensive energy analysis of R22, R438A, R134a, R410A, R407C, R290 in a vapor compression refrigeration system has been presented. The conclusions of the present analysis are summarized below.

1. COP for R438A is better than others for the same design conditions and compressor displacement volume. COP of R438A is found to be 34% higher theoretically while the same is found to be 16-24% higher experimentally depending on cooling load conditions. The increase in COP comes at a cost of decrease in the Cooling Effect by approximately 20%
2. Analysis for the same Cooling Effect but different compressor displacement volume and design conditions shows us that COP of R438A system decreases by 17%. It is because of 17% lesser latent heat of R438A which results in 31% more mass flow rate further causing a 20% increase in the compressor work.

Hence, R438A proves to be a better alternative to R22 if energy efficiency and environmental friendliness are our main focus parameters. Further theoretical and experimental analysis of R410A can be done as an alternative to R22 as the Cooling Effect shows an increase of 40% when compared to R22 and the same comes at a cost of only 10% lesser COP than R22.

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