

PERFORMANCE ASSESSMENT OF ALTERNATE REFRIGERANTS FOR RETROFITTING R22 BASED AIR CONDITIONING SYSTEM

by

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The performance of zero ozone depletion potential refrigerants is investigated when retrofitted in R22 based air conditioning system. The options evaluated are R407C, R417A, R422D, R427A, and R438A. In order to arrive at most suitable alternative(s) to R22, energy and exergy performance of candidate refrigerants is carried out and compared against that of R22. The COP and exergy efficiencies showed that none of selected refrigerant is as efficient as R22 however their values suggests that each may be considered as potential substitute for retrofitting. Having comparable COP to others but low cooling capacity of R417A makes it less attractive. With comparatively reduced COP, lowest exergy efficiency and highest mass-flow rate, makes R422D the least desirable option. The R407C, R427A, and R438A emerged as most attractive substitutes. The lower discharge temperatures of substitutes will enhance the compressor life. Further, for substitutes there may be a possible change out of expansion valve.

Key words: ozone depletion potential, alternative, COP, exergy efficiency

Introduction

The R22, a hydrochloroflourocarbon, widely used in refrigeration and air conditioning systems as refrigerant is a controlled substance under the Montreal Protocol due to its ozone depletion potential (ODP). As intended by this protocol, developed countries have already banned R22 to be used in new equipment, while on the other side its phase out process is in implementation. Followed by the accelerated reduction in consumption, the complete phase out of R22 for developed countries is scheduled for 2020. Similarly the developing countries are subjected to cease its production and consumption till 2030 [1].

Compliance to the phasing out of R22 is a legal obligation which has given rise to the search for environmental friendly alternatives that can effectively replace R22. In this respect studies have being conducted on substances with zero ODP that can potentially replace R22 and further investigations are in progress to explore the most promising alternates for R22. The important issue to be addressed in assessing the alternate to be used as refrigerant instead of R22 is its performance relative to R22. On the other hand the identification of most suitable candidate to replace R22 is essential. While environmental and safety compatibility are not very complex,

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the efficiency performance which entirely depends on the thermodynamic behavior of the fluid requires rigorous attention and is also important to the user of the system.

Keeping in view the previous discussion, researchers have evaluated a number of refrigerants as possible substitutes to R22. Devotta *et al.* [2] theoretically assessed the suitability of HFC-134a, HC-290, R407C, R410A, and three blends of HFC-32, HFC-134a, and HFC-125 as alternatives to HCFC-22 in vapor compression system. It was concluded that HC-290 showed closeness to HCFC-22 while HFC-134a and R410A requires new compressor and heat exchangers. The R407C was suitable for retrofitting. Spatz *et al.* [3] tested R404A, R410A, and R290A as alternatives to R22 in medium and low temperature refrigeration system. It was concluded that system modifications for R404A and R410A are needed for efficient performance and R290A has similar performance to R22. Using different heat exchangers with three air conditioners models of R410A, Chen [4] showed that the performance of R410 could reach to that of R22. Aprea and Renno [5] made experimental investigation on R417A as substitute to R22 in vapor compression system of cold store. The authors showed that *COP* of R417A was 14% lower and exergy destruction was 14% higher than R22. Aprea *et al.* [6] further investigated the system of [5] with R507C, R407C, and R417A. The evaluation method was based on regulating the refrigerating capacity by changing the speed of compressor. The authors concluded that R22 performance was best followed by R407C. It was claimed that 12% reduction in energy consumption with R407C is possible with variable speed control. Arora and Sachdev [7] analyzed R422A, R422B, R422C, and R422D for low temperature application. It was reported that on the basis of *COP* and exergy efficiency R422B suitable alternative to R22. Park *et al.* [8, 9] reported that for both air conditioners and heat pumps R431A and R432A are best drop in refrigerants for R22 due to their efficiency. An on-site energy performance study on 160kW R22 water chiller was performed by Torella *et al.* [10] when R22 is substituted with R417A and R422D. The outcomes indicated that a decrease in cooling capacity, compressor power and discharge temperature have occurred for both refrigerants. A study by Rocca and Panno [11] on R417A, R422A, and R422D when substituted for R22 in vapor compressor plant showed that no major system modifications are required however improvement in energy efficiency is needed. The system of [11] was further explored by Messineo *et al.* [12] with R404A, R407C, and R417A. Their results showed that none of the substitute was as efficient as R22. Allgood and Lawson [13] reported that when R438A is used as alternative for R22, the *COP* is 5% to 10% lower than R22 and both have similar energy efficiency ratios. The R22, R134a, R507a, R404A, and R717A were investigated under constraint compression ratio and compressor outlet temperature in vapor compression system by Stanciu *et al.* [14]. Studies revealed that under the imposed restrictions R717A emerged as most satisfactory substitute for R22. Lopis *et al.* [15] examined R422A and R417B as alternatives to R22 for low and medium evaporating temperatures. For both the candidate a reduction in cooling capacity and *COP* was reported. The R22 walk in cooler when retrofitted with R422D was experimentally investigated by Aprea *et al.* [16]. For R422D a 20% decrease in *COP* and 45% increase in mass-flow rate was noticed. It was concluded that higher fan speeds are required for R422D to reject thermal power. Chakravarthy and Deva Kumar [17] experimentally studied R407A and R407C in one ton R22 air conditioner and declared both as suitable alternatives. The experimental test on R22 split air conditioner when retrofitted with R410A and R417A by Bolaji [18] showed that R417A is better choice for retrofitting. On the other hand Oruc and Devecioglu [19] claimed that R424A has better performance than R417A when used as alternative in R22 split air conditioners. Ramu *et al.* [20] made investigation on three mixtures of R32/R125/R600a as R22 alternatives. It was reported that with modified condenser, the mixture with mass composition of 0.40/0.40/0.20 can be a suitable replacement.

Similarly, Vandaarkuzhali and Elansezhian [21] declared that a mixture of R22/R152a of 70:30 has highest discharge temperature and *COP* and these mixtures can work safely without system modifications. Venkatiah and Venkata [22] theoretically assessed R134a, R404A, R407C, R410A, and R507A, R290A, and R600a in place of R22 in air conditioners. None of the refrigerants was reported with all the characteristics as of R22. Boumaza [23] focused on natural refrigerants including R290, R600a, and R717A to replace R22. It was found that on the basis of overall performance R290 is better choice followed by R600a.

It is seen that the focused substitutes for R22 are R134a, R407C, R410A, R417A, R422D and some hydrocarbons including R290, R600a, and R717A. The R134a and R410A requires system modifications if used as replacements. The R407C, R417A, and R422D are regarded as appropriate for retrofitting but their energy efficiency needs to be addressed. From efficiency point of view, R290, R600a, and R717A are better choices for retrofitting but safety concerns make them less attractive. It is observed that the performance of refrigerant is related to working conditions and application it is used for. Therefore, it is not necessary to expect the same performance from particular refrigerant in every system. The previous work on finding the replacements for R22 is based on two main approaches. One approach is making considerable changes to system design and the other is retrofitting the existing system components with alternate refrigerants. Both the approaches have been used but in most of the present work either the common substitutes are evaluated or lesser number of substitutes are selected from available options. It is rare to see the consolidated selection set consisting of comparatively greater number of substitutes meant separately either for new systems or retrofitting the existing systems. This bounds the possibility of discovering other substitutes that might perform even better and also limits the inter-comparison of substitutes. Further, limited number of studies are found on exergy efficiency which provide even better idea in establishing most acceptable alternative. Also, the methods and procedures employed for experimental work are time consuming and costly. Analytical method which uses the thermodynamic properties of the refrigerants can estimate with good accuracy how the system will perform when refrigerant is changed [24].

Owing to the previous discussion, investigation on R407C, R417A, R422D, R427A, and R438A, a zero ODP and A1 safety classified alternatives, declared suitable for retrofitting R22 systems by US Environmental Protection Agency [25] are investigated by employing a well-established thermodynamic model. The objective is to arrive at the most suitable substitute from both energy and exergy point of view to retrofit our R22 system.

System description

The system is vapor compression system basically designed for R22 refrigerant. The compressor has six cylinders with total swept volume of 1249 cm³ and is driven by electrical motor. Its maximum rotational speed is 1070 rpm as per manufacturer. The condenser is designed for counter flow operation having shell and tube arrangement in which refrigerant pass through tubes and water through shell. Thermostatic expansion valve is installed just before the evaporator and is heat insulated. The valve adjusts automatically as well as manually by mean of knob. Evaporator coil is exposed to the space to be cooled. The specifications of the system are summarized in tab. 1. A schematic diagram of the system is presented in fig. 1. Figure 2 provides P-H diagram on which respective processes are marked.

Mathematical model

In order to have a robust analysis criteria the system performance is analyzed both from energetic and exergetic point of view.

Table 1. System specifications

Refrigerant	R22
Compressor	
Number of cylinders	6
Displacement volume	1249 cm ³
Maximum speed	1070 rpm
Operating pressure	18 bar (HP), 0.5 bar (LP)
Inlet temperature	10-20 °C
Maximum outlet temperature	90 °C
Oil type	Mineral
Evaporation temperature	4 °C
Condensation temperature	40 °C
Cooling capacity	58.11 kW
Evaporating range	-12-20 °C
Cooling water temperature	-2-30 °C

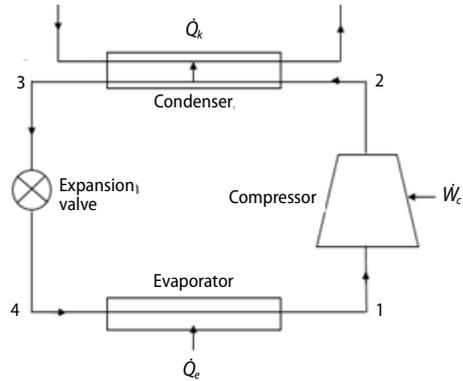


Figure 1. Schematic of vapor compression system

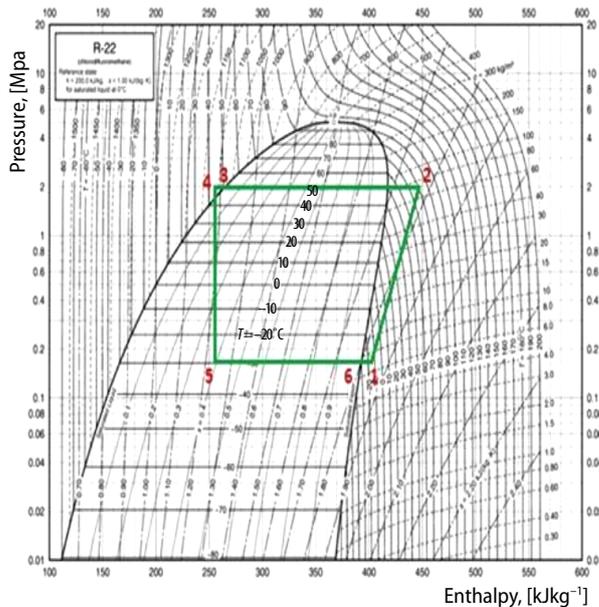


Figure 2. P-H diagram of R22 with respective processes, [ASHRAE Handbook]

Energy analysis

Under steady-state conditions with negligible kinetic and potential energies, for a control volume the energy balance is applied in accordance with [26].

Following the state points on PH diagram:

The compressor power is:

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

The heat rejected by condenser is:

$$\dot{Q}_k = \dot{m}(h_2 - h_4) \quad (2)$$

The expansion process is isenthalpic.

Heat absorbed by evaporator is:

$$\dot{Q}_e = \dot{m}(h_6 - h_1) \quad (3)$$

The overall energetic performance is expressed by COP and is given:

$$COP = \frac{\dot{Q}_e}{\dot{W}_c} \quad (4)$$

Additionally, in accordance with [27] the volumetric efficiency is:

$$\eta_{vol,ac} = 0.96 \left\{ 100 - CL \left[\left(\frac{P_{dis}}{P_{suc}} \right)^{\frac{1}{n}} - 1 \right] \right\} \quad (5)$$

As per [28] the compressor displacement rate is:

$$\dot{V}_{\text{disp}} = V_{\text{swept}} \frac{N}{60} \quad (6)$$

The mass-flow is then obtained from:

$$\dot{m} = \frac{\eta_{\text{vol,ac}} \dot{V}_{\text{disp}}}{v_{\text{suc}}} \quad (7)$$

The VCC is calculated:

$$VCC = \frac{\dot{Q}_e}{\dot{m} v_{\text{suc}}} \quad (8)$$

The pressure drop across expansion valve is given:

$$\Delta P_{\text{exp}} = P_4 - P_5 \quad (9)$$

Exergy analysis

In accordance with [26] for a control volume with steady flow the exergy balance is applied.

The exergy destroyed in compressor is:

$$\dot{X}_{d,c} = \dot{m} T_0 (s_2 - s_1) \quad (10)$$

The exergy destroyed in condenser is:

$$\dot{X}_{d,k} = \left(1 - \frac{T_0}{T_k}\right) \dot{Q}_k + \dot{m} [(h_2 - h_3) - T_0 (s_2 - s_3)] \quad (11)$$

The exergy destroyed in expansion valve is:

$$\dot{X}_{d,\text{exp}} = \dot{m} T_0 (s_5 - s_4) \quad (12)$$

The exergy destroyed in evaporator is:

$$\dot{X}_{d,e} = \left(1 - \frac{T_0}{T_s}\right) \dot{Q}_e + \dot{m} \{(h_5 - h_6) - T_0 (s_5 - s_6)\} \quad (13)$$

The exergy efficiency is then calculated:

$$\eta_{\dot{X}} = 1 - \frac{\dot{X}_{d,\text{tot}}}{\dot{W}_c} \quad (14)$$

Results and discussion

First, the energetic and exergetic parameters for R22 were calculated and treated as base data for comparison. The thermodynamic properties at the respective states are evaluated using Refprop [29]. As shown in tab. 1, for R22, from the design condition of 4 °C evaporating and 40 °C condensing temperature, the mass-flow rate is readily calculated by dividing the

cooling capacity with specific refrigerating effect. The volumetric efficiency is then calculated from eq. (7). Based on the available information, a CL value of 5% is taken. From eq. (5), for n a value of 0.533 is obtained which is assumed to be constant. Using eq. (5) again with pressure ratio of respective operating point the volumetric efficiency is calculated followed by mass-flow rate using eq. (7). The isentropic efficiency of compressor is taken as 0.8. Presently, a system has superheating of 6 °C and subcooling of 3 °C, therefore same values are used in calculations.

The effect on parameters is studied over the evaporating range of -12 °C to 16 °C with a fixed condensing temperature of 40 °C.

Figure 3 shows the mass-flow variation when system is operating under various evaporating temperatures. The highest mass-flow is observed for R422D followed by R438A being 28% to 35% and 8% to 14% higher than R22, respectively.

Figure 4 shows the variation of cooling capacity over a range of evaporating temperature at 40 °C condensing temperature. The mass-flow rate and refrigerating effect increases with increasing evaporating temperature so cooling capacity also increases with increase in evaporating temperature. It is seen that R22 and R407C has nearly same capacities. Over the considered evaporating range the cooling capacities of R422D, R427A, R438A are, respectively, lower than R22 by about 7.6% to 14.5%, 2.3% to 8.6% and 5.8% to 12.7%. The lowest cooling capacity is recorded for R417A which is about 15.8% to 22.6% lower than R22.

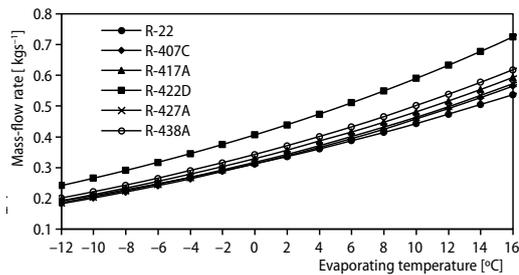


Figure 3. Effect of evaporating temperature on mass-flow rate

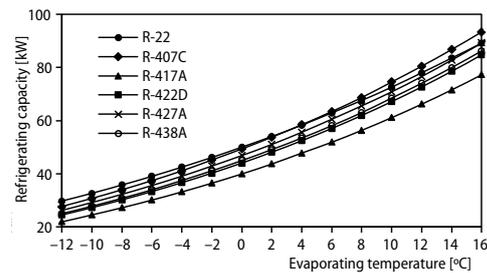


Figure 4. Effect of evaporating temperature on refrigerating capacity

The compressor power variation is presented in fig. 5. With increasing evaporating temperature the work of compression decreases but increasing mass-flow results in gradual increase in compressor power. For R407C it is observed that compressor is slightly overloaded at high evaporating temperatures. However this increase in power is only 6% relative to R22 which is acceptable.

The R422D and R427A has close compressor powers being 3.8% less than R22 while a reduction of 1.8% to 6.7% in power is noticed for R438A. The lowest power consumed is by R417A which is about 11.5% to 17% lower than R22. The results obtained indicated that in terms of power the existing compressor is capable to be used with these substitutes.

The VCC, a useful parameter in determining the size of compressor is calculated as per eq. (8) and plotted in fig. 6. The R407C and R427A have close VCC to R22. The VCC of R417A, R422D, and R438A are, respectively, 15.4% to 21.2%, 7.4% to 13.5% and 5.4% to 11.4% lower than R22. It is observed that in this case VCC is more affected by cooling capacity rather than by pressure ratio which is nearly similar for all the considered refrigerants. Hence it can be deduce that the current compressor is capable of drawing the volume needed for producing required cooling capacity.

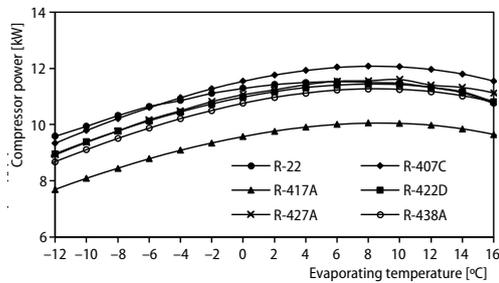


Figure 5. Effect of evaporating temperature on compressor power.

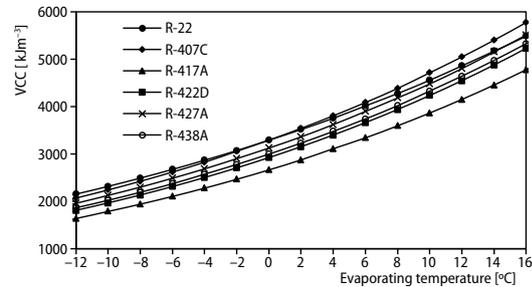


Figure 6. Effect of evaporating temperature on VCC

Investigation on the sufficiency of heat exchange area of existing condenser is made by comparing heat rejection of refrigerants. The heat rejected at various evaporating temperatures by R22 and alternates are presented in fig. 7. All the substitutes have lower heat rejection than R22 however R407C showed a slight increase of about 2.6% in rejected thermal power which can be catered by adjusting cooling water flow rate. The R417A exhibited the lowest heat rejection which is about 15.5% to 21.6% lower than R22.

As shown in fig. 8, none of the selected refrigerants have higher COP than R22 which ascertains that with alternates the plant energy efficiency will reduce. The R417A, R427A, and R438A have similar COP being 4.8% to 5.4% lower than R22. The R422D has comparatively lower COP, about 8% to 10.8% less than R22. The R407C has superior COP to others being 2.7% to 3.2% lower than R22.

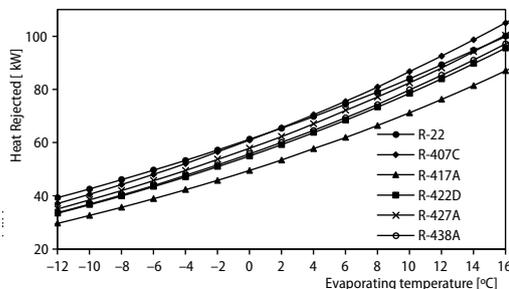


Figure 7. Effect of evaporating temperature on heat rejected

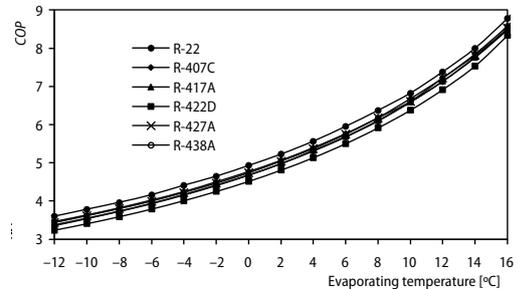


Figure 8. Effect of evaporating temperature on COP

The operational envelope of compressor is confined by its pressure ratio and discharge temperature. High pressure ratio means greater mechanical stresses thus having a direct impact on construction materials. High discharge temperatures cause thermal deterioration of lubricating oil and construction materials such as seals. As evident from fig. 9 the alternatives have slightly high pressure ratios than R22 so small adjustment in existing pressure settings will be needed. This also means that the existing pressure temperature switches/controls can be used with retrofitted system.

On the other hand all the alternatives have lower compressor outlet temperatures than R22 which may help in enhancing the overall reliability of compressor.

The suitability of expansion valve is established through the expected pressure drop that will occur across it. As can be seen in fig. 10, except for R417A, all the alternates have

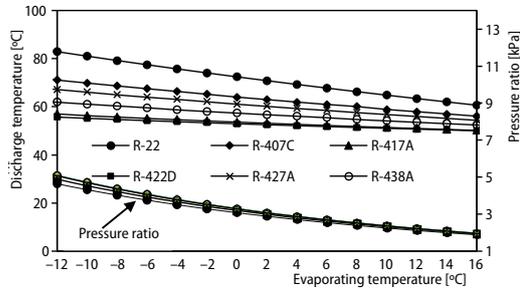


Figure 9. Effect of evaporating temperature on discharge temperature and pressure ratio

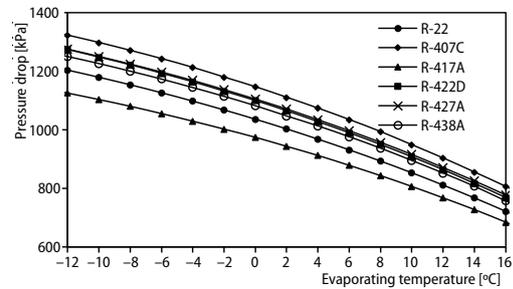


Figure 10. Effect of evaporating temperature on pressure drop across expansion valve

higher pressure drops across expansion valve than R22, the highest being for R407C for which a rise of 11% is noticed.

The difference between the pressure drop of R22 and alternate refrigerants increases with decreasing evaporating temperature. This means expansion valve becomes more critical at low evaporating temperatures. Based on pressure drop comparison there may a possible change out of expansion valve for alternate refrigerants.

The quantification of real energetic losses in components for considered refrigerants is calculated in the form of exergy destruction using eqs. (3)-(6) and then from eq. (7) respective exergy efficiency is obtained, a parameter of prime importance. As per ambient conditions the dead state temperature of 30 °C is taken. Further it is assumed that the boundary of condenser is at the same temperature as of cooling water, a valid assumption for system boundary with immediate surroundings. The space temperature is maintained 15 °C above from respective evaporating temperature ($T_s = T_e + 15$ °C).

Figure 11 shows the exergy destroyed in components for R22 and its substitutes. The highest exergy destruction is noticed for condenser and evaporator. The R22 and R407C has comparatively high destruction rates in condenser and evaporator because under the given temperature gradient both the refrigerants are subjected to high heat rejection and heat absorption, only a part of which can be transferred to surroundings or space. Except for R417A, the exergy destroyed in compressor with alternates is higher than R22, the highest being for R407C and R422D which on the average basis is about 4.8% and 2.6% higher than R22. Analysis on exergy destroyed in expansion valve revealed that all the alternates exhibits greater destruction than R22. The R422D has highest expansion valve exergy destruction, indicating a rise of 47.8% relative to R22, which is attributed to its high mass-flow.

The average percent deviation of exergy destroyed in alternate refrigerants relative to R22 is presented in fig. 12. It is observed that in terms of exergy destruction in substitutes, the compressor, condenser, and evaporator are less critical while expansion valve emerged as critical component, having high percent deviation of exergy destruction relative to R22.

Referring to fig. 13, R22 has greater exergy efficiency than its substitutes. This means R22 has superior quality of product (cooling capacity) with given fuel supplied (compressor electrical power). The R407C, R417A, R427A, and R438A have close exergy efficiencies, about 7.5% lower than R22. The worst exergy efficiency is of R422D with a reduction of 30% relative to R22.

Oil compatibility

Minor and Yokozeki [30] showed that the lubricant do not affect the performance of the system. However, the compatibility of substitute refrigerant with lubricant is important as

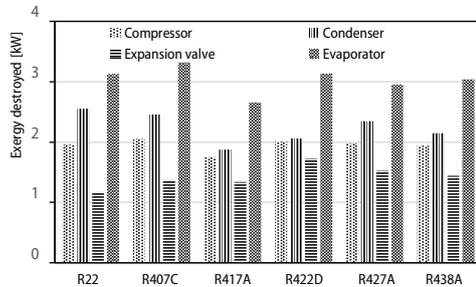


Figure 11. Exergy destroyed in components

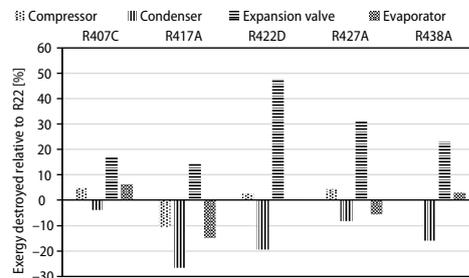


Figure 12. Percent exergy destroyed relative to R22

effective oil return and construction material adaptability are factors to be addressed during the retrofit. The R407C and R427A are not compatible with the mineral oil used with R22 as immiscibility of both refrigerants with mineral oil results in poor oil return. With R407C and R427A instead of mineral oil the polyolester oil must be used. The major disadvantages of polyolester oil is that it is expensive, highly hygroscopic and causes irritation to skin when comes in contact [7]. It is also reported that seal and gaskets for R22 are usually incompatible with polyolester oil [31]. Moreover, retrofitting becomes time consuming when the mineral oil is entirely to be flushed out. On the other hand R417A, R422D, and R438A can be used with both mineral and polyolester oil which give them an added advantage over R407C and R427A.

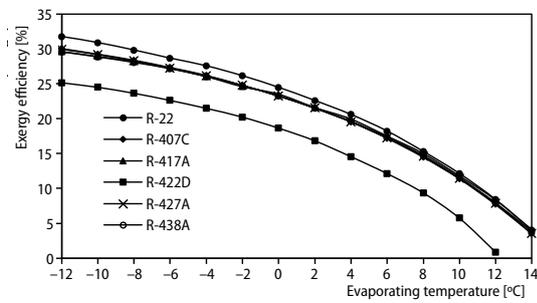


Figure 13. Effect of evaporating temperature on exergy efficiency

Conclusions

Performance assessment of R22 based system when retrofitted with ozone friendly refrigerants including R407C, R417A, R422D, R427A, and R438A is made by carrying out energy and exergy analysis. It is concluded that the system retrofitted with any of the selected refrigerants will have low energy and exergy performance than the system originally designed for R22. The *COP* of R407C, R417A, R422D, R427A, and R438A are, respectively, 2.7% to 3.8%, 4.8% to 5.9%, 8% to 10.8%, 4.8% to 5.9% and 4.8% to 5.9% lower than R22. The exergy efficiency of R407C, R417A, R427A, and R438A is 7.5% and that of R422D is 30% lower than R22. The energy and exergy analysis suggests that each of the considered candidate is potential substitute for R22 but low cooling capacity of R417A which is about 15.8% to 22.6% lower than R22 makes it less attractive. The R417A has also lowest compressor power being 11.5% to 17% less than R22 but cooling capacity can not be compromised when other refrigerants are capable of providing comparatively high capacity with power achievable by existing compressor. The R422D has comparatively lower *COP* and exergy efficiency than R22. Additionally, R422D has highest mass-flow, being about 28% to 35% higher than R22 which may not be acceptable for existing piping and auxiliaries. These facts make R422D a least desirable choice. On the basis of energy and exergy performance R407C, R427A, and R438A are most appropriate substitutes for retrofitting. Retrofitting with alternate refrigerants results in lower

compressor discharge temperatures than R22, thus enhancing the overall reliability of compressor. Compressor, condenser and evaporator do not need to be changed as required electrical power, heat rejection and cooling capacity are under the limits of these components, however, expansion valve may be changed due to higher pressure drop and exergy losses than R22. The pressure requirements of R407C, R427A, and R438A are close to that of R22 requiring minimum settings after retrofitting. This also eliminates the use of new pressure and temperature safety switches. The R438A has an advantage over R407C and R427A because it is compatible with both mineral and polyolester oil which makes the retrofitting process safe, cost effective and less time consuming.

Nomenclature

COP – coefficient of performance, [–]

CL – clearance, [%]

h – specific enthalpy, [kJkg⁻¹]

\dot{m} – mass-flow rate, [kgs⁻¹]

n – polytropic index, [–]

N – speed, [rpm]

P – pressure, [kPa]

\dot{Q} – rate of heat transfer, [kW]

s – specific entropy, [kJkg⁻¹K⁻¹]

T – temperature, [°C or K]

\dot{V} – volume flow rate, [m³s⁻¹]

VCC – volumetric cooling capacity, [kJm⁻³]

v – specific volume, [m³kg⁻¹]

\dot{W} – power, [kW]

\dot{X} – exergy, [kW]

Δ – change, [–]

Subscripts

0 – dead state temperature

ac – actual

c – compressor

dis – compressor discharge

disp – compressor displacement

d – destroyed

exp – expansion valve

e – evaporator

k – condenser

s – space

suc – suction

tot – total

vol – volumetric

Greek symbols

η – efficiency, [–]

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