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

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Abstract

The performance of zero Ozone Depletion Potential (ODP) refrigerants is investigated when retrofitted in R22 based air conditioning system. The options evaluated are R407C, R417A, R422D, R427A and R438A. In order to establish the objective of arriving at most suitable candidate/s to substitute R22, energy and exergy performance assessment of candidate refrigerants is carried out and compared against that of R22. Studies revealed that none of selected refrigerant is as efficient as R22 however COPs and exergy efficiency values suggests that each may be considered as potential substitute for retrofitting. The COPs of R407C, R417A, R427A and R438A are respectively 2.7% to 3.8%, 4.8% to 5.9%, 4.8% to 5.9% and 4.8% to 5.9% lower while their exergy efficiencies are about 7.8% lower than R22 but low cooling capacity of R417A, about 15.8% to 22.6% lower than R22 makes it less attractive. With comparatively reduced COP of about 8% to 10.8%, lowest exergy efficiency, about 26.2% lower than R22 and highest mass flow, about 28% to 35% higher than R22, makes R422D the least desirable option. R407C, R427A and R438A emerged as most attractive substitutes but compatibility of R438A with mineral oil gives it an added advantage, making the retrofit process less time consuming and cost effective. The substitutes exhibited lower discharge temperatures than R22 thus enhancing the compressor life. Further, for substitutes there may be a possible change out of expansion valve.

Keywords: ODP, retrofitting, R22, R407C, R417A, R422D, R427A, R438A, COP, exergy efficiency.

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Nomenclature

$\boldsymbol{COP}$	coefficient of performance
$\boldsymbol{CL}$	clearance, %
$\boldsymbol{h}$	specific enthalpy, kJ/kg
$\boldsymbol{\dot{m}}$	mass flow rate, kg/sec
$\boldsymbol{P}$	pressure, kPa
$\boldsymbol{\dot{Q}}$	rate of heat transfer, kW
$\boldsymbol{s}$	specific entropy, kJ/kgK
$\boldsymbol{T}$	temperature, ^{0}C or K
$\boldsymbol{\dot{V}}$	volume flow rate, m^3/sec
$\boldsymbol{v}$	specific volume, m^3/kg
$\boldsymbol{\dot{W}}$	power input, kW
$\boldsymbol{\dot{X}}$	exergy, kW
$\boldsymbol{VCC}$	volumetric cooling capacity, kJ/m^3

Greek letters

$\boldsymbol{\eta}$	efficiency
$\boldsymbol{\Psi}$	exergy rate of fluid, kJ/kg
$\boldsymbol{\Delta}$	change

Subscripts

$\boldsymbol{0}$	dead State temperature
$\boldsymbol{ac}$	actual
$\boldsymbol{c}$	compressor
$\boldsymbol{d}$	destroyed
$\boldsymbol{dis}$	compressor discharge
$\boldsymbol{disp}$	compressor displacement
$\boldsymbol{exp}$	expansion valve
$\boldsymbol{e}$	evaporator
$\boldsymbol{k}$	condenser
$\boldsymbol{suc}$	suction
$\boldsymbol{tot}$	total
$\boldsymbol{vol}$	volumetric

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Chapter 1

Introduction

1.1 Background:

Refrigeration and Air Conditioning have an array of applications from food preservation to the thermal comfort they provide to humans. Initially the working substances used in these systems were chloroflourocarbons (CFCs). “In 1970s extensive research was carried out on the substances that deplete the Ozone Layer of which the important discovery was the identification of manmade CFCs as Ozone destroying substances by Molina and Rowland in 1974 [1]”. “The Ozone layer acts as an atmospheric shield which protect the life on earth against harmful ultraviolet radiation [2]”. “There is compelling evidence that exposure to ultraviolet radiation affects human health and is the cause of skin cancer and cataracts [3]”.

This harm to Ozone layer by CFCs and other chemicals became increasingly concerned in the world`s nations and efforts were made to address this critical issue. In 1987 a treaty ‘Montreal Protocol on the substance that deplete the Ozone layer’ was signed for taking corrective actions and to eliminate the Ozone Depleting Substances (ODS). The Montreal Protocol establishes legally binding controls on the national production and consumption of ODS. Via this protocol the CFCs were banned and there production was stopped by 1995. “Hydrochloroflourocarbons (HCFCs), were developed as transitional substitutes for CFCs and are also subject to be phase out because of their ODP and Global Warming Potential (GWP) [4]”.

1.2 HCFC22/R22:

Among all the HCFCs the most commonly used is chlorodiflouromethane or HCFC22 or R22 as refrigerant. It has been widely used in air conditioning and in medium and low temperature applications within the commercial and industrial refrigeration [20]. These included unitary air conditioners, cold storage, food refrigeration equipment, chillers, and industrial process refrigeration [4]. Having ODP of 0.055 and GWP of 1810 R22 is no longer considered as acceptable for use and is scheduled to be phased out under the Montreal Protocol.

1.3 Global Scenario of R22:

"European Union and Japan have banned the import of R22 systems since 2004 [20]. The US which has already reduced to 75% the consumption and production of R22 by 2010 and from onwards it is completely banned to be used in new equipment. By 2020 its production and importing will be completely stopped [4]". This also applies to all other developed countries. Similarly the developing countries including China and others will reduce its usage to 35% by 2020 and completely phased out it until 2040.

Pakistan became the member of Montreal protocol in 1992. In 2013 Pakistan imported 2731.09 Metric Tons of R-22 contributing to OPD Ton of 150.20 [31]. The National Ozone Unit (NOU) is actively involved to meet the targets for HCFCs set for developing countries. The NOU had already started working to phase out the HCFCs including HCFC22 from the industry. Moreover HCFC phase out plan was launched in 2011 by which Pakistan will be able to eliminate the use of HCFCs from Refrigeration and Air Conditioning and other sectors.

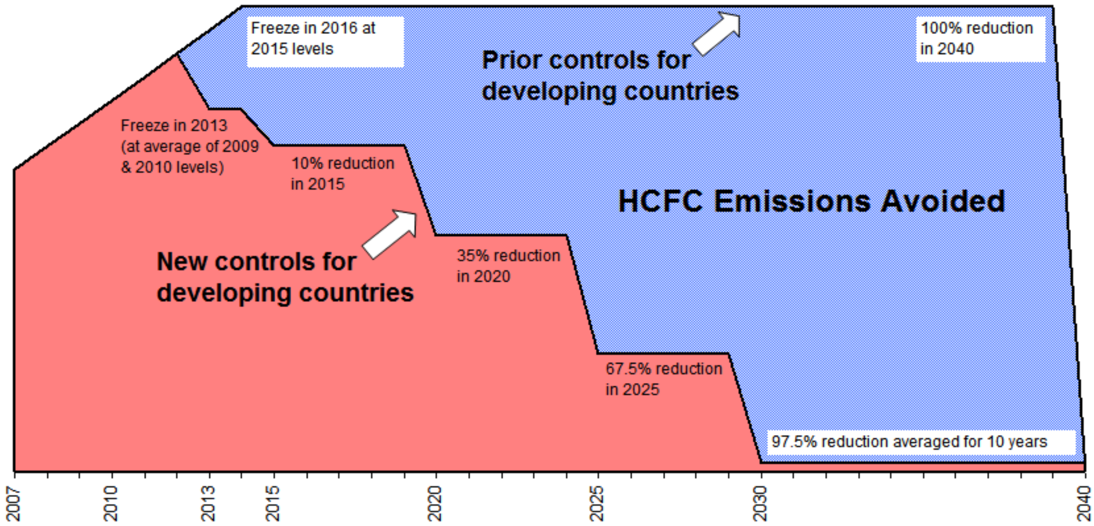

Figure 1.1 New stronger Montreal protocol Controls for R22 phase-out [source: US EPA]

It is a fact that large number of plants are still running on R-22 but due to restrictions there has been an increased interest in the alternatives to R22. In the last few years zero ODP refrigerants have been developed and many companies have extended much efforts to further develop and identify the refrigerants able to give performance to R22 in the particular application.

1.4 Retrofitting R22 Systems:

Instead of replacing the existing R22 equipment with new equipment, the other viable option available is retrofitting it with suitable substitute. Retrofitting is also a cost effective option when the establishment of a complete new plant is not feasible. Studies on retrofitting R22 systems has been carried out and many other new refrigerants are going to be investigated.

The important issue to be addressed when system is retrofitted with an alternate refrigerant is its performance as of R22. Similarly identification of the most competitive candidate to replace R22 is also very important. In current developing situation, there has been no such kind of refrigerant which can perfectly replace R22 in most of the existing equipment [32]. However criteria exist on the basis of which potential candidates can be assessed. This assessment of replacing R22 is done by considering its environmental protection, efficiency level and safety level. As stated, environmental protection is measured by OPD and GWP. Efficiency assessment is done by carrying out the energetic/exergetic analysis by calculating parameters including Coefficient of Performance (COP), compressor power, discharge temperature, mass flow rate, refrigerating capacity, exergy destroyed and exergy efficiency. These are then compared with base parameters of replaced refrigerants, providing useful information to check the suitability of new refrigerant. Safety level is also very important because flammability and toxicity are key issues. While environmental and safety compatibility are not very complex, performance analysis which entirely depends on the thermodynamic behaviour of the fluid requires rigorous attention. The experimental procedures used for the verification that an existing system will work satisfactorily with alternate refrigerant is a lengthy and expensive procedure. Analytical method which uses the thermodynamic properties of the refrigerants can estimate with good accuracy how the system will perform when refrigerant is changed [22].

"The first assigned R22 substitute to reach the market was R407C but since then a number of alternatives has appeared [33]". In recent years refrigerant blends, which are the mixtures of two or more pure refrigerants have become popular in retrofitting R22 systems. These have predictable properties and potential exists to further explore newly developed blends in order to arrive at the best suitable substitute.

This thesis is concerned with the assessment of various refrigerant blends for retrofitting R22 air conditioning system. Performance analysis is carried out for each of the selected refrigerant and compared with that of R22. Performance evaluation can be carried out experimentally or analytically. Conducting experimental studies is useful for obtaining near accurate results but time and cost associated with experiments make it expensive. Also it is not feasible to conduct a number of experiments when variety of options are available. A well-established analytical method can be employed for carrying out the analysis of the system when working fluid is changed and results can be obtained with good approximation. We have used analytical approach by employing a mathematical model based on first and second law of thermodynamics.

1.5 Goals and Objectives:

1) To investigate the performance of alternate refrigerants when retrofitted for R22.
2) To check the feasibility of retrofitting for the considered alternatives.
3) To identify the most competent alternative to substitute R22.

Chapter 2

Literature Review

To meet the legal obligations imposed by Montreal Protocol on R22 many researchers and scientists have studied other refrigerants that may replace R22. For a refrigerant to replace R22, its evaluation in terms of performance relative to R22 is of prime importance. A number of refrigerants are evaluated as possible substitutes to R22. The desirable candidate is the one which have similar performance like that of R22 and lower environmental impact. The work carried out in assessing the alternate refrigerants for R22 is presented as follows.

Devotta et al. [5] 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. The study was performed on VCS with IHX for various evaporating and condensing temperature of 55^0C. Among the candidates HC-290 showed closeness to HCFC-22 but safety risks are the major concern. HCFC-134a was most efficient but it required larger compressor due to its low cooling capacity. Due to higher pressure and compressor power R410A also required new compressor and heat exchangers. It was reported that R407C is suitable for retrofit due to same compressor displacement as HCFC-22.

Spatz et al. [6] evaluated R404A, R410A and R290 as alternatives to R22 in medium temperature refrigeration systems. A thermodynamic model was developed for a prediction of system performance of each refrigerant compared to R22. It was shown that R404A and R410A when modified for optimal performance shows better efficiency than R22. R410A and R290A showed similar performance but the latter has safety issues which makes R410 a better choice. The limitation of this study is the modification of the system for improving the efficiency. In many cases modification is not a good option as it adds costs.

Aprea and Renno [7] investigated R417A as a substitute to R22 in cold store application. The vapour compression plant which is employed in cold stores was experimentally setup and design parameters were selected accordingly. Energetic and exergetic study was performed for summer and winter conditions. To simulate these conditions condenser air temperature of 32^0C and 10^0C was maintained and cooling capacity was held constant. Tests showed that on average COP of R417A was 14% lower than R22. The exergy destroyed in components was 14% higher than R22. For both the refrigerants the highest efficiency defects were in compressor and heat exchangers.

Aprea et al. [8] investigated the energy performance of the same vapour compression plant used in cold stores. The selected refrigerants against R22 were R507C, R407C and R417A. The evaluation method used was regulating the refrigerating capacity by changing speed of the compressor with 30Hz to 50Hz current supply. The measured parameters were then compared with on/off cycles of the compressor working with 50Hz. It was shown that R22 performance is best followed by R407C. It was also shown that the variable speed control is advantageous than on/off control. The authors reported that using variable speed control, for R407C a 12% reduction in energy consumption is possible.

Chen [9] made a comparative study by simulating one R22 baseline and three R410A air conditioners models having same cooling capacities but different heat exchanger constructions. It was found that R410A nearly as R22 and at low ambient temperatures R410A could be more efficient. Also the use of R410A will enable to develop air conditioners with high operation efficiency and LCCP index for R410A could be lowered by reducing its indirect global warming impact.

An energetic and exergetic analysis of R422 series refrigerants was performed by Arora and Sachdev [10]. These refrigerants included R422A, R422B, R422C and R422D. The computational model was developed carrying out the analysis. Parameters computed were volumetric cooling capacity, compressor discharge temperature, COP, exergetic efficiency and exergy destruction in system components. The effect of varying evaporating and condensing temperature was studied on theses parameters. They reported that discharge temperatures od R422 series were lower than R22 by about 20^0C to 70^0C in the evaporating range of -40^0C to 10^0C. Among the selected refrigerants the volumetric cooling capacity of R422A was higher compared to others. COP and exergetic efficiency values show that R22 exhibits higher values followed by R422B. When condensing temperature was increased, decrease in VCC, COP and exergetic efficiency was noted for all the refrigerants. Also compressor and expansion valve needs improvement as these components presents the largest efficiency defects. The authors concluded that on the basis of COP and exergetic efficiency R422B is suitable alternative to R22.

Park et al. [11,12] used R431A and R432A as drop in refrigerant for R22 in air conditioners and heat pumps. In both applications COPs and cooling capacities of these substitutes were higher and the discharge temperatures were lower compared to R22. Although results showed that these are good options but due to hydrocarbons their safety hazards make them less attractive.

An on-site study on R22 water chiller of the building was performed by Torrella et al. [13]. Effect on the energy performance of the chiller was studied when R22 is replaced with R417A and R422D. The chiller capacity was 160kW having two independent cycles. Experiments were carried out over a wide range of ambient temperatures and performance was based on mass flow, COP, cooling capacity and compressor power. The authors observed that there was a decrease in cooling capacity and compressor power of R417A and R422D. The compression ratio of both the substitutes was higher than R22 but the discharge temperature were lower.

Allgood and Lawson [14] discussed the performance of R438A when used in R22 systems in low, medium and high temperature applications. Laboratory tests on R438A for its performance relative to R22 were conducted and compared with the results obtained from actual field retrofits. The results from both the sources were in well agreement with laboratory tests. It was reported that cooling capacity of R438A was within 5% to 10% in broad range of evaporating temperatures and energy efficiency ratio of R438A is similar to R22. Moreover there is no change of lubricating oil required for R438A during retrofit.

R22 vapour compression refrigerating plant was experimentally evaluated by Rocca and Panno [15] for drop in tests with R417A, R422A and R422D. In this experiment the cooling load was water ethylene glycol mixture. COP against mass flow rate of water ethylene glycol mixture and as a function of evaporation temperature was calculated. In both the cases COPs of the candidates were less than R22.The compression ratios and discharge temperature were higher and lower than R22 as also reported by [25]. Although these refrigerants do not require any major changes but incrementing energy efficiency is still a challenge.

The system studied by [15] was further investigated by Messineo et al. [23] when R22 is replaced by R417A, R407C and R404A. Using the same procedure and methodology they also concluded that performance of none of the refrigerants exceeded R22 and further refrigerants need to be evaluated for energy efficiency and environmental friendliness.

Stanciu et al. [16] studied the effect of the refrigerant in VCS on its operation and performance limits. For a compression ratio of between 2 and 6 with constrained outlet temperature of 140^0C, R22, R134a, R507a, R404a and R717 were investigated. Under the imposed conditions, R717 has lowest compression ratio and vaporization temperature and highest COP, volume flow rate and specific refrigerating power and also shown substantial increase in COP with superheating. R134a has higher operation limit and higher vaporisation temperature than R22. This method can be very useful in system optimization and retrofitting under working constraints.

R422A and R417B were analysed as substitutes for R22 in low and medium evaporating temperatures by Llopis et al. [17]. Both theoretical and experimental study was carried out. The energy parameters were evaluated in the evaporating rang of -31^{0}C to -17^{0}C and in condensing range of 30^{0}C to 48^{0}C. It was noted that cooling capacity for the substitute refrigerants decreases. The authors reported that at -30^{0}C evaporating and 40^{0}C temperature there is 12.1% and 11.6% reduction in cooling capacity of R417B and R424A. Also at these temperatures a reduction of 20.3% and 26% in COP was recorded for R417B and R422A. Differences in values were obtained from theoretical and experimental studies. The authors associated this difference due to variation in volumetric efficiency in real conditions.

Wang et al. [18] used R404A as a replacement to R22 in a cooling system of a manufacturing process. The outlet coolant temperature required for the process was -15^{0}C. Over a range of coolant outlet temperature it was shown that COP of R404A is better than R22. The effect on the performance of the condenser cooling water temperature was also evaluated at coolant temperature of -15^{0}C. Also in this case the cooling capacity and COP of R404A was higher than R22. Electronic expansion valve controlled by a controller was also tested for temperature and power consumption. It was found that this scheme was more accurate and cost effective option for retrofitted R404A system.

Aprea et al. [19] experimentally investigated the R22 walk in cooler when retrofitted with R422D by varying refrigeration capacity with external air temperature of 24^{0}C and superheat of 7-10^{0}C. Based upon the results on average COP and mass flow rate was 20% lower and 45% higher than R22. Furthermore R422D needed higher fan speed because the condenser area was not enough to reject thermal power.

Ashok and Kumar [20] experimentally investigated two zoetrope blend refrigerants R407A and R407C as an alternates to R22 in window air conditioning system. 1 ton window air conditioner utilizing R22 and mineral oil as lubricant was selected for a test. Refrigerant side temperature and pressure and air side dry bulb and wet bulb temperature at an ambient temperature of 27.3^{0}C dry bulb and 19.2^{0}C wet bulb were recorded and refrigerating capacity, compressor power, heat rejection in condenser, energy efficiency ratio (EER) and COP was calculated for R22, R407A and R407C. Results showed that cooling capacity for R407A and R407C was improved by

7.5% and 4.5% and the compressor power for R407A and R407C was lowered and higher by 2% and 9% .EER increased by 7.5% and 4% and the mass flow increased by 6.5% for R407A and decreased by 5% for R407C compared to R22. Both were declared as suitable alternatives.

Bolaji [21] experimentally tested R22 split air conditioner when retrofitted with R410A and R417A. Performance parameters were measured by changing evaporating temperature. It was observed that average refrigeration capacity of R417A and R410A was higher and lower by 1.9% and 14.2% while COP was 2.9% higher and 8.4% lower than R22. Similarly other characteristics also proved that overall performance of R417A was better and suitable choice for retrofitting. As previously reported by [17, 20] here also the R410A performance deviated from R22.

Three mixtures of R32/R125/R600a having mass compositions of 0.45/0.45/0.10, 0.40/0.40/0.20 and 0.35/0.35/0.30 by Ramu et al. [24] as alternatives to R22. Tests were carried out at evaporating temperature of between -10^0C and 10^0C and condensing temperature of 35^0C, 45^0C and 55^0C. Results showed that the mixture of 0.40/0.40/0.20 has better thermal and performance characteristics and can be a suitable replacement. However modification will be needed to counteract the high heat rejection of this mixture.

Kuzhali and Elansezhian [25] made experimental tests on R22/R152a mixture in an air conditioning system. R22/R152a mixture in the ratio of 30:70, 50:50, 70:30 by mass at ambient temperature of 32^0C were tested. It was recorded that a mixture of 70:30 has highest discharge temperature and COP and R22/152a mixture can work safely without system modifications.

Venkatiah and Rao [26] theoretically assessed R13aa, R404A, R407C, R410A, R507A, R290 and R600a in place of R22 in 5.276kW air conditioner. The analysis was carried out between -5^0C to 15^0C evaporating and 55^0C condensing temperature. Results showed that R600a has highest COP but it needed largest compressor displacement. R404A and R507A consumed largest compressor power and COP of R404A was the lowest. Among all R22 has the highest discharge temperature. It was reported that no refrigerant has all the characteristics as of R22.

The performance of R22 split type air conditioner having cooling capacity of 2.05kW was investigated by Oruc and Devecioglu [27] when R22 is replaced by R417A and R424A. Evaluation was based on energy and exergy criteria. The room temperature was kept at 22^0C and measurements of various parameters were obtained at ambient conditions of 25^0C, 30^0C and 35^0C. It was reported that COP of R22 was higher than R417A and R424A, however R424A has 5%-16% higher COP than R417A. The discharge temperatures of these two substitutes were lower than R22. As of [34] they also showed that exergy efficiency of R22 is highest and exergy defects in expansion device and compressor are higher in alternate refrigerants. They concluded that R417A and R424A has almost similar performances but R424 is more suitable due to its overall better performance. High GWP of R424A is still an issue.

Boumaza [28] focussed on the use of natural refrigerants including R290, R600a and R717 to replace R22. The author analytically carried out the comparative performances in terms of COP, volumetric refrigerating capacity and compressor load by varying evaporating and condensing temperatures. Results showed that R290 was most suitable followed by R600a but flammability and toxicity are serious issues which need to be mitigated.

It is seen that the focussed substitutes for R22 are R134a, R407C, R410A, R417A, R422D and some hydrocarbons including R290, R600a and R717A. R134a and R410A requires system modifications if used as replacements. 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.

Owing to the above discussion, 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.

Chapter 3

Mathematical Modelling

Refrigeration cycle consists of a sequence of thermodynamic processes in which heat is drawn in from cold body and rejected to hot body by consuming energy. These processes are accomplished through various system components constituting the whole cycle. Before going to discuss the governing equations involved in cycle analysis, it will be useful to understand the terms, definitions and processes involved in the cycle.

3.1 Vapour Compression Cycle:

Vapour compression cycle is widely implemented for air conditioning. The system under investigation is also based on this cycle. In this cycle vapour undergoes phase changes through a series of processes. The cycle consists of following major components.

- Compressor
- Condenser
- Expansion Valve
- Evaporator

Figure 3.1 shows the systematic arrangement of these components and a sequence of processes which are as follows:

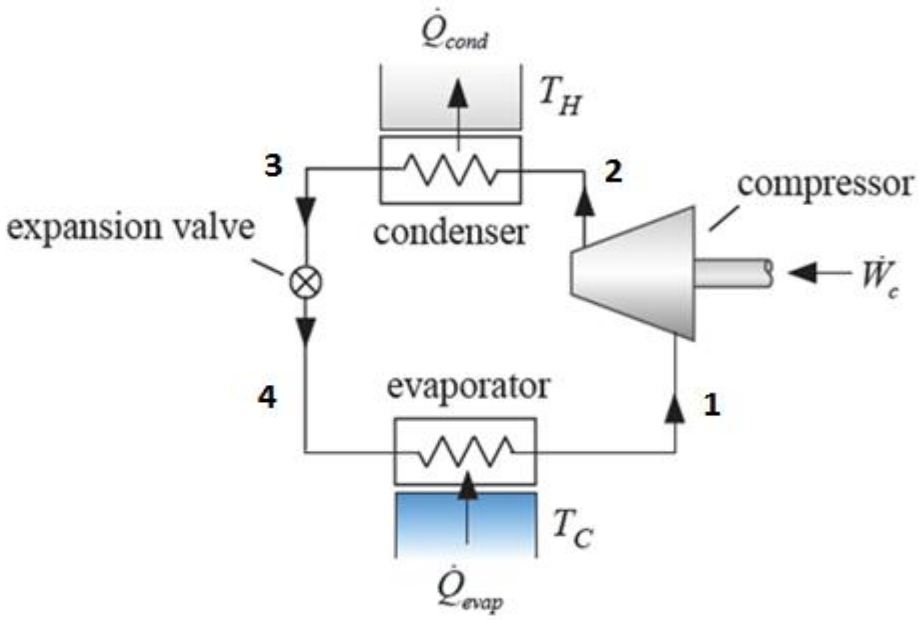

Figure 3.1 Schematic of conventional vapour compression system [36].

Process 1-2: Compression of vapour refrigerant. The refrigerant comes as a low pressure gas and enters the compressor where it is compressed and is discharged with vapour with high pressure and temperature.

Process 2-3: The vapour is fed into the condenser giving off its heat to surroundings. At the end of this process the refrigerant is liquid having high pressure.

Process 3-4: The high pressure liquid enters the expansion valve where it restricts the flow of liquid and lowers its pressure when it leaves the valve.

Process 4-1: This low pressure liquid then moves to evaporator where it picks up from inside air and changes from liquid to vapour. This sequence of processes is again repeated and the cycle continues to work.

3.2 Pressure Enthalpy Diagram:

Process 2-3 and 4-1 are the condensation and evaporation process occurring at constant pressure. Process 1-4, the compression may be assumed as isentropic while from 3-4 the expansion process is isenthalpic. Figure 3.2 represents these processes on P-H diagram with refrigerant being superheated and subcooled after evaporation and condensation. The enthalpies at the start and end of the process are used to calculate various parameters.

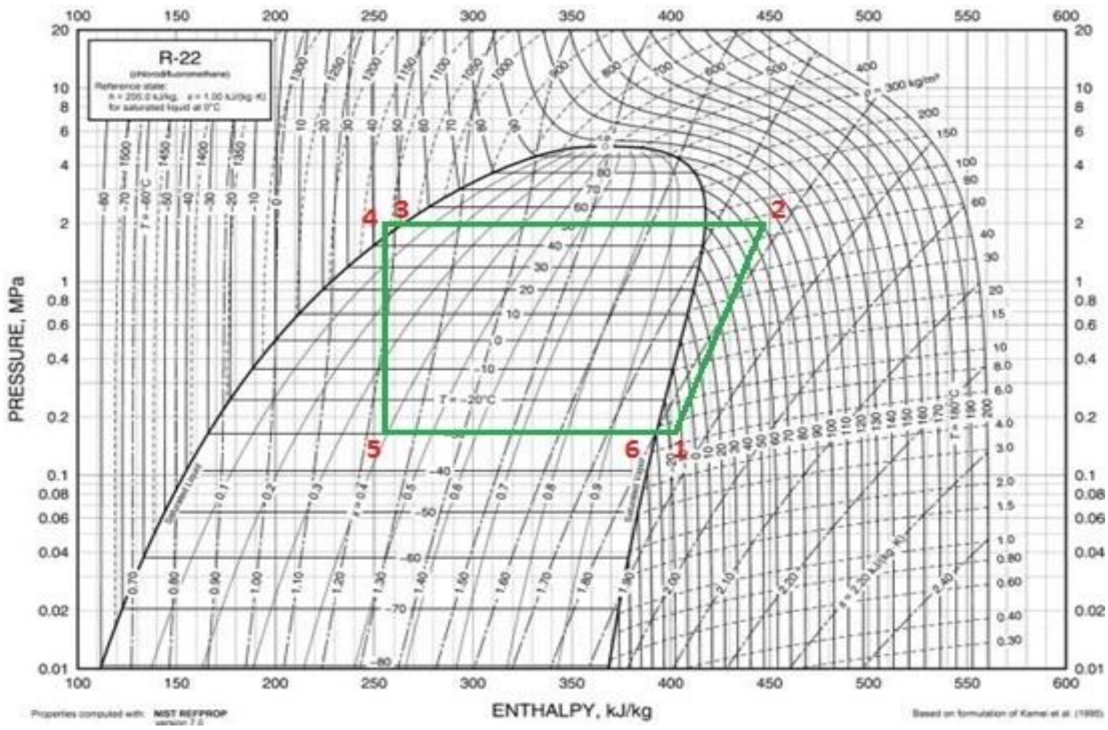

Figure 3.2 Pressure enthalpy diagram of R22 with respective processes [ASHRAE Handbook].

3.3 VCS Performance Parameters:

The performance of Vapour Compression System (VCS) is measured in terms of parameters defined below.

3.3.1 Coefficient of Performance (COP)

The overall energetic performance of the plant is evaluated by its COP defined as the ratio of refrigeration capacity to the electrical power supplied to the plant.

3.3.2 Refrigerating Capacity

The total amount of heat absorbed from the space or body to be cooled in unit time. It is measured in watts.

3.3.3 Circulation Rate

It is the mass flow rate of the refrigerant within the system.

3.3.4 Compressor Power

The power consumed by compressor, calculated as the product of mass flow rate and compressor specific work.

3.3.5 Discharge Temperature

The temperature at outlet of the compressor. It affects compressor`s life and provides an indication of the operational limit. The lubricating oil is affected by the temperature as high temperatures thermally deteriorates it, making it unfit for use.

3.3.6 Compression Ratio

The ratio of outlet to inlet pressure of the compressor.

3.3.7 Volumetric Cooling Capacity (VCC)

The amount of refrigeration produced per unit volume of the refrigerant. It is given as the ratio of refrigerating capacity to volume flow rate of refrigerant and useful in determining the size of compressor.

3.3.8 Exergy Efficiency

The degree to which the given amount of energy is effectively utilized. For vapour compression cycle it is defined as the ratio of minimum (reversible) work input to useful work input.

3.4 Governing Equations:

In order to quantify the parameters the equations of energy balance and exergy balance can be conveniently used.

3.4.1 Energy Balance

Assuming the flow as steady flow in the control volume, the energy balance equation can be written as [35]

$$\dot{Q} + \dot{W} = \dot{m}(h_{out} - h_{in}) \tag{3.2}$$

3.4.2 Exergy Balance

The quantification of real energetic losses is important to measure the criticality of systems components when they operate with different refrigerants. Exergy analysis provides useful information on the energy degradation that occurred in components during energy transfers. This degradation of energy is calculated in the form of exergy destruction. In accordance with [35], for a control volume with steady flow the exergy balance is written as

$$\left(1 - \frac{T_0}{T_s}\right)\dot{Q} + \dot{W} + \dot{m}(\psi_{in} - \psi_{out}) - \dot{X}_d = 0 \tag{3.3}$$

Where $\psi_{in} - \psi_{out} = (h_{in} - h_{out}) - T_0(s_{in} - s_{out})$ and T_s is the temperature of the source at which heat is transferred.

The first and second term in equation (3.3) is the exergy rate transfer by heat and work. The third term is the difference of exergy rate transfer by mass at inlet and outlet of the control volume. The last term quantifies the exergy destroyed within the boundary of the control volume.

3.4.3 Assumptions in Calculations:

The calculations performed are based on the following assumptions

1) The flow through the control volume is steady flow.
2) Compressor isentropic efficiency is 0.8.
3) Compressor and expansion valve are adiabatic.
4) Pressure drops in components are negligible.
5) Kinetic and potential energy of fluid is negligible.
6) A superheat of 6^0C and subcooling of 3^0C is used as per available information.

3.5 Calculation of Parameters

Applying equation (3.2) and (3.3) to evaporator, following the nomenclature of PH diagram, we have

$$\dot{Q}_e = \dot{m}(h_1 - h_5) \tag{3.4}$$

$$\dot{X}_{d,e} = \left(1 - \frac{T_0}{T_e}\right)\dot{Q}_e + \dot{m}\{(h_5 - h_1) - T_0(s_5 - s_1)\} \tag{3.5}$$

Applying equation (3.2) and (3.3) at inlet and outlet of compressor the compressor power and exergy destroyed in compressor is

$$\dot{W}_c = \dot{m}(h_2 - h_1) \tag{3.6}$$

$$\dot{X}_{d,c} = \dot{W}_c + \dot{m}\{(h_1 - h_2) - T_0(s_1 - s_2)\} = \dot{m}T_0(s_2 - s_1) \tag{3.7}$$

The function of the condenser is to reject heat. Heat rejection rate is calculated by applying energy balance. To calculate the exergy destruction, exergy balance is applied at condenser inlet and outlet sections. Here $\dot{W} = 0$.

$$-\dot{Q}_k = \dot{m}(h_3 - h_2)$$

Or

$$\dot{Q}_k = \dot{m}(h_2 - h_4) \tag{3.8}$$

$$\dot{X}_{d,k} = \left(1 - \frac{T_0}{T_k}\right)\dot{Q}_k + \dot{m}\{(h_2 - h_4) - T_0(s_2 - s_4)\} \tag{3.9}$$

The expansion process is assumed isenthalpic therefore $h_4 = h_5$ and

$$\dot{X}_{d,exp} = \dot{m}T_0(s_5 - s_4) \tag{3.10}$$

The COP of the compressor is calculated as

$$COP = \frac{\dot{Q}_e}{\dot{W}_c} \tag{3.11}$$

3.6 Additional Relations:

As per definition the volumetric cooling capacity is given by

$$VCC = \frac{\dot{Q}_e}{\dot{m} * v_{suc}} \tag{3.12}$$

The compressor displacement rate is the volume swept per unit time, mathematically it is calculated as

$$\dot{V}_{disp} = V_{swept} * \frac{N}{60} \tag{3.13}$$

The clearance in the compressor cylinder causes the re-expansion of the gas and an amount of refrigerant is lost due to leakage past the rings of piston and pressure drop across suction and discharge valves [34]. The combined effect of both phenomena is the lower suction volume than actually swept by compressor. Therefore according to [30] volumetric efficiency is introduced and is written as

$$\eta_{vol,ac} = 0.96\{100 - CL\left[\left(\frac{P_{dis}}{P_{suc}}\right)^{\frac{1}{n}} - 1\right]\} \tag{3.14}$$

CL is the percent clearance volume and is fixed for a given compressor. P_2/P_1 is the discharge to suction pressure ratio. The typical values for *CL* are between 4-7%. As noted volumetric efficiency is the function of pressure ratio and decreases with increasing pressure ratio. In this case the mass flow is calculated as

$$\dot{m} = \frac{\eta_{vol,ac} * \dot{V}_{disp}}{v_{suc}} \tag{3.15}$$

The total exergy destroyed in the system is

$$\dot{X}_{d,tot} = \dot{X}_{d,c} + \dot{X}_{d,k} + \dot{X}_{d,exp} + \dot{X}_{d,e} \tag{3.16}$$

The exergetic efficiency of the system which is a measure of how nearly the actual performance approaches the ideal is calculated as

$$\eta_{ex} = 1 - \frac{\dot{X}_{d,tot}}{\dot{W}_c} \tag{3.17}$$

The pressure across expansion valve is calculated as

$$\Delta P = P_4 - P_5 \tag{3.18}$$

3.7 Dew Point and Midpoint Protocol:

Unlike pure refrigerant the evaporation and condensation process in refrigerant mixtures do not occur at constant temperature. During constant pressure phase change, in refrigerant mixtures there is a difference in temperature at evaporator inlet and outlet. Similarly a variation in temperature exists and condenser inlet and outlet. This difference is described by term glide. For evaporator it is the difference between evaporator dew point and evaporator inlet temperature. For condenser it is the difference between condenser dew point and condenser bubble point temperature. Refrigerant R407C, R417A, R422D, R427A and R438A are mixtures of 3-5 pure components with respected mass compositions. Their P-H diagrams have sloping temperature lines during evaporation and condensation. For comparing their performance with R22 two methods can be employed that are dew point and midpoint definitions. In dew point approach the evaporating and condensing temperatures are defined on dew points, the saturation vapour temperatures. In midpoint approach a mean evaporating and mean condensing temperatures are defined which are considered as the temperatures that would prevail along the entire length of evaporator and condenser. In both cases the degree of superheat is calculated as the difference in compressor inlet and evaporator outlet temperature. Many researchers have recommended midpoint approach for comparing the refrigerant mixtures with pure refrigerants however dew point method can also be used [29]. The standard AHRI 540-2004 also treats the evaporating and condensing temperatures of refrigerant mixtures as midpoint values for compressor performance. In our analysis a midpoint protocol approach is used for alternate refrigerants for comparing their performance with R22.

Chapter 4

Results and Discussion

4.1 Description of the Working Unit:

The working unit to be analysed consists of the following main components

- Six cylinder reciprocating compressor
- Cross flow water cooled condenser
- Thermostatic expansion valve
- Evaporator

The system is basically designed for R22 refrigerant. The compressor has six cylinders with total swept volume of 1249cm^3.It is driven by electrical motor which is connected to it through belt and pulleys. 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 is 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 summarised in table 4.1.

Table 4.1 Specifications of the system.

Compressor	
No. of cylinders	6
Swept Volume	1249 cm^3
Operating Pressure	18bar (HP), 0.5bar(LP)
Maximum Speed	1070rpm
Compressor inlet temperature	10^{0}C-20^{0}C
Maximum outlet temperature	90^{0}C
Refrigerant	R22
Oil type	Mineral
Evaporation temperature	4^{0}C
Condensation temperature	40^{0}C
Cooling capacity	58.11kW
Evaporating temperature range	-12^{0}C to 20^{0}C
Cooling water entering temperature	-2^{0}C to 30^{0}C

4.2 Performance Analysis:

The air conditioning system which is meant to be retrofitted with R22 alternative is theoretically assessed for the energy and exergy performance when refrigerant is changed. The theoretical investigation has an advantage allowing the user to make good approximations for each potential candidate in short period of time rather than individually conducting time consuming and costly experiments to arrive at suitable replacement. The options evaluated to replace R22 include R407C, R417A, R422D, R427A and R438A. First, the results for R22 are obtained. These results formed the basis for system constraints and comparison. Three different methods are used for comparison and discussed.

4.2.1 Energy Analysis

Based on the information of Table 4.1, performance parameters are evaluated for R22. As per nomenclature of figure 3.2, the thermodynamic properties are tabulated as follows

Table 4.2 Thermodynamic properties of R22 ($T_e = 4$°C , $T_k = 40$°C).

State	Temperature ^{0}C	Enthalpy kJ/kg	Entropy kJ/kg.K	Specific volume m^3/kg
1	10	411	1.7611	0.0430
2	62	436.5	1.7799	0.0174
3	40	249.6	1.1665	0.000886
4	36.6	245.1	1.1520	0.000874
5	4	245.1	1.1628	0.00896
6	4	406.5	1.7450	0.00416

From equation (3.4) the mass flow rate is calculated as

$$\dot{m} = \frac{58.11}{406.5 - 245.1} = 0.36kg/sec$$

Using equation (3.6) the power consumption of compressor is

$$\dot{W}_c = 0.36(442.9 - 411) = 11.5kW$$

COP of compressor from equation (3.11) at these conditions is

$$COP = \frac{58.11}{11.5} = 5.1$$

From equation (3.8) the heat rejected by the condenser is

$$Q_k = 0.36(442.9 - 249.6) = 69.5kW$$

The volume flow rate at compressor inlet is the product of mass flow and suction specific volume.

$$\dot{V}_{suc} = 0.36 * 0.0430 = 0.0154m^3/sec$$

Using equation (3.12) the volumetric cooling capacity is

$$q_{vol} = \frac{58.11}{0.0154} = 3773kJ/m^3$$

With equation (3.13) the displacement rate is calculated as

$$\dot{V}_{dis} = \frac{1249}{10^6} * \frac{1070}{60} = 0.0222m^3/\sec\ or\ 22.2dm^3\ /sec$$

Table 4.3 shows the results for R22 at 4^0C evaporating and 40^0C condensing temperature. Compressor inlet temperature is 10^0C.

Table 4.3 Results for R22 ($T_e = 4°C$, $T_k = 40°C$).

Parameter	Value
Discharge Temperature, ^{0}C	69.2
Mass Flow, kg/sec	0.36
Volume Flow, m^3/sec	0.0154
Ref. Capacity, kW	58.1
Compressor Power, kW	11.5
COP	5.1
Heat Rejected, kW	69.5
VCC, kJ/m^3	3773

In carrying out the retrofit assessment for alternate refrigerants the results of R22 are treated as base data for comparison. As these results are based on the actual design condition of the plant they define the limiting conditions specific to this operating point. For an alternate refrigerant to produce the desired cooling capacity of 58.1kW at this condition, the compressor power should be either lower or must not exceed 11.5kW. At this condition the COP of the compressor is 5.1 and volume flow is 15.4dm^3/sec. The condenser is limited to reject 69.5kW. This is important as insufficient heat dissipation will adversely affect the performance of the plant. However cooling water flow rate may be adjusted to certain level to meet heat dissipation requirement with new refrigerant. Using the procedure as for R22, calculations are made for R407C, R417A, R422D, R427A and R438A which are explained in section 4.2.2.

4.2.1.1 Calculations for Alternate Refrigerants:

4.2.1.1.1 Refrigerating Capacity as Constraint

The refrigerating capacity of 58.1kW is kept constant for calculating performance measures at 4^0C evaporating and 40^0C condensing temperature. The compressor inlet temperature of 10^0C is maintained. For alternate refrigerants the midpoint protocol is employed for calculations. The results are presented in table 4.4.

Table 4.4 Results for R22 and alternate refrigerants, constant Ref.cap ($T_e = 4°\text{C}$, $T_k = 40°\text{C}$).

Parameter	R22	R407C	R417A	R422D	R427A	R438A
Dis. Temp ^{0}C	69.2	62	52.8	52.1	59.3	56.1
Mass Flow, kg/sec	0.36	0.36	0.47	0.53	0.39	0.44
Vol. Flow, dm^3/sec	15.4	15.3	18.7	17.1	16.1	16.7
Ref. Capacity, kW	58.1	58.1	58.1	58.1	58.1	58.1
Comp. Power, kW	11.5	12	12	12.5	12	12
COP	5.1	4.9	4.8	4.6	4.9	4.8
Heat Rejected, kW	69.5	70	70	70.6	70	70.2
VCC, kJ/m^3	3773	3803	3110	3395	3618	3474

The results in table 4.4 are presented in terms of percentage change from R22 in table 4.5.

Table 4.5 Percent change of parameters of alternate refrigerants from R22, constant Ref.cap.

Parameter	R22	R407C	R417A	R422D	R427A	R438A
Dis.Temp, ^{0}C	69.2	-10.4%	-23.6%	-24.7%	-14%	-19%
Mass Flow, kg/sec	0.36	0	+30.5%	+47.2%	+8.3%	+22.2%
Vol. Flow, dm^3/sec	15.4	+0.6%	+21.4%	+11%	+4.5%	+8.4%
Ref. Capacity, kW	58.1	0	0	0	0	0
Comp. Power, kW	11.5	+4.4%	+4.4%	+8.6%	+4.4	+4.4%
COP	5.1	-4%	-5.8%	-9.8%	-4%	-5.7%
Heat Rejected, kW	69.5	+0.7%	+0.7%	+1.6%	+0.7%	+1%
VCC, kJ/m^3	3773	+0.8%	-17.5%	-10%	-4%	-7.8%

The above results indicate that the discharge temperatures of substitute refrigerants are lower than R22. Mass flow of R22 and R407C are similar while that of other candidates are higher than R22. This is due to the lower refrigerating effect of other alternate refrigerants. The highest mass flow is of R422D which is about 47.2% higher than R22. Despite of less compressor work, the high mass flow of R422D results in its highest compressor power being 8.6 % percent higher than R22. The important result to be noted here is the compressor power limit which is exceeded by each refrigerant. The suction volume flow of each refrigerant is also greater than R22. The highest volume flow is of R417A which is about 21.4% higher than R22. Increase in volume flow may not be handled by compressor thus reducing effective mass flow rate. Less mass flow means less cooling capacity. Among the candidates R407C and R427A have highest COP. For both the refrigerants the COP is about 4% less than R22. The heat rejection values show that there is no significant difference between the heat rejected by R22 and others.

Comparison on the basis of single operating point do not provide sufficient knowledge for evaluating the refrigerants. Thermodynamic properties of refrigerants changes when operating conditions vary. Therefore the performance at each operating point is also different. As air conditioning system may operate at different conditions, knowledge about the behaviour of individual refrigerant specific to that operating point is necessary to justify the comparison. In order to have a clear understanding of the performance of each refrigerant in our system, performance curves are generated by varying evaporating and condensing temperatures. The evaporator temperature is varied between -12^{0}C to +16^{0}C against a fixed condensing temperature of 40^{0}C.

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. The pressure ratios and discharge temperatures of R22 and its alternatives are presented in figure 4.1 and figure 4.2. As evident the alternatives have slightly high pressure ratios than R22 so small adjustment in existing pressure settings will be required after retrofitting the system. 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 will help in enhancing the overall reliability of compressor.

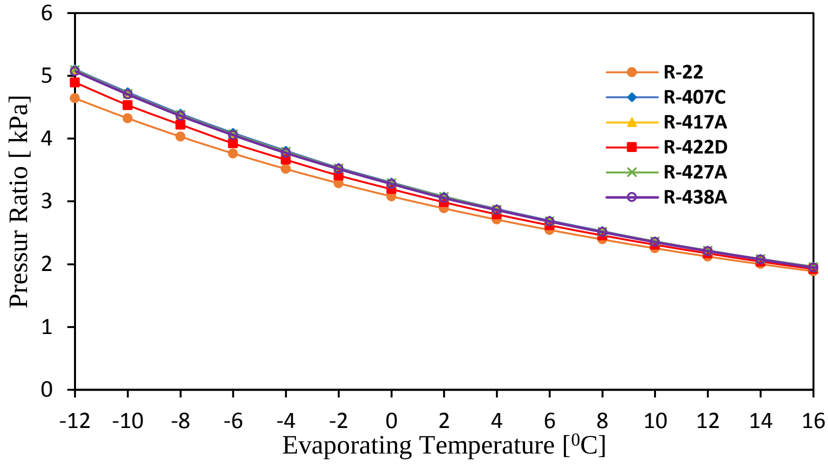

Figure 4.1 Effect of evaporating temperature on pressure ratio ($T_k = 40°C$).

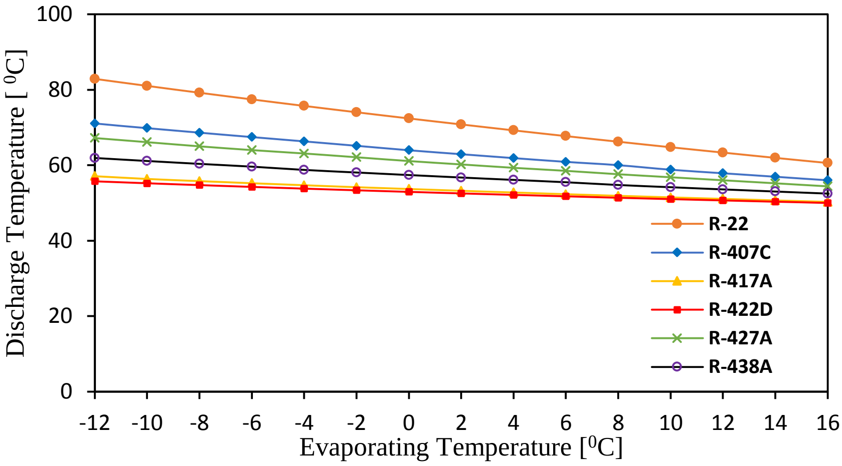

Figure 4.2 Effect of evaporating temperature on discharge temperature ($T_k = 40°C$).

Figure 4.3 shows the effect of evaporating temperature on mass flow rate. Since the refrigerating effect do not vary considerably for small temperature differentials, the mass flow rate between two close operating conditions is approximately similar. At all the evaporating temperatures the mass flow of R422D is higher than others. R22 and R407C have almost similar masses over the entire range.

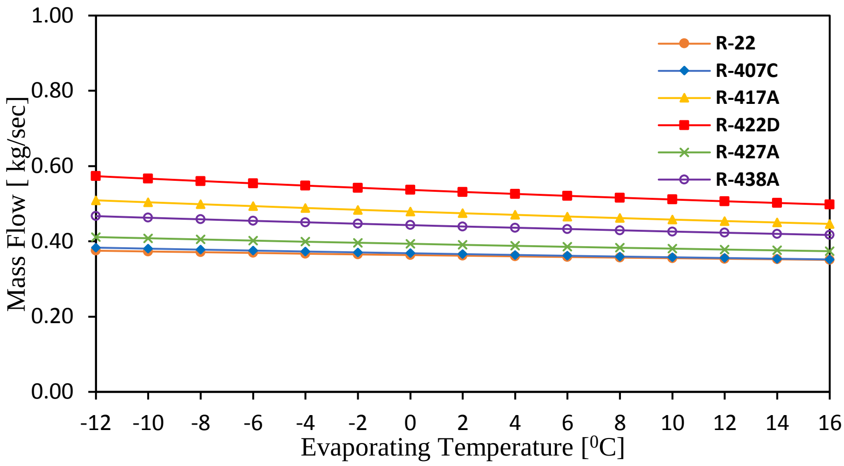

Figure 4.3 Effect of evaporating temperature on mass flow rate, constant ref.cap ($T_k = 40$°C).

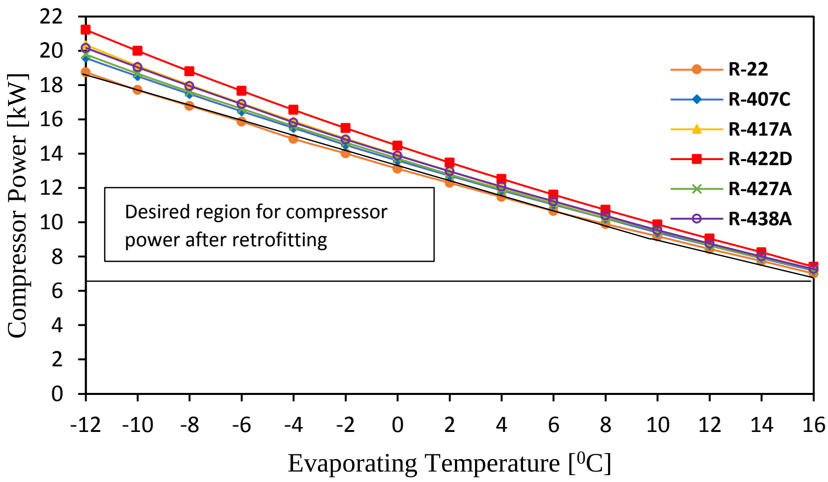

Figure 4.4 Effect of evaporating temperature on compressor power, constant ref. cap ($T_k = 40$°C).

The effect of evaporating temperature on compressor power when cooling capacity is constant is presented in figure 4.4. Clearly at other operating conditions also, the compressor power needed for alternate refrigerants is higher than R22. R422D needs largest compressor power due to its high mass flow rate.

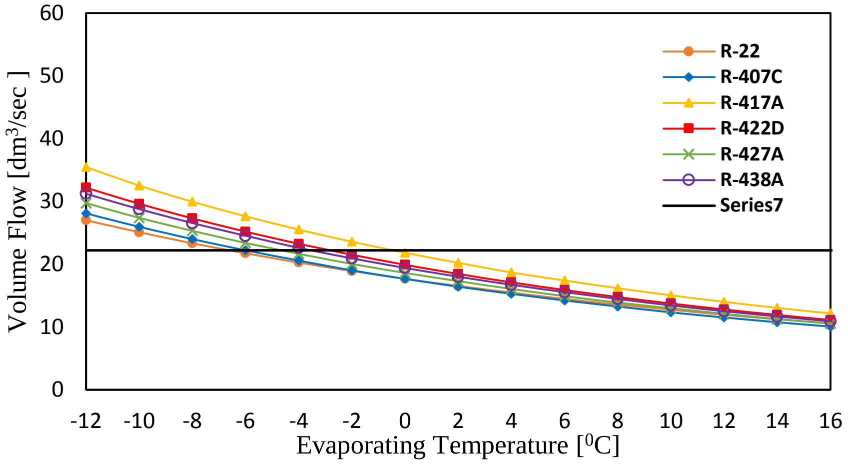

Figure 4.5 Effect of evaporating temperature on volume flow rate, constant ref.cap ($T_k = 40°C$).

The effect of evaporating temperature on volume flow is shown in figure 4.5. The volume flow of all the refrigerants is higher than R22. At the evaporating temperature of -8^{0}C the volume flow of R22 is 23.3dm^3/sec which is not possible because the compressor ideal swept volume is 22.2dm^3/sec. Similarly the volume flow of other refrigerants exceeds this value when evaporating temperature starts decreasing from 0^{0}C respectively. This scenario means that the volume flow needed for producing the cooling capacity of 58.1kW cannot be handled by the compressor after evaporating temperature of 0^{0}C

Figure 4.6 depicts the condenser heat rejected with varying evaporating temperature. R22 and R407C having lower masses but the heat rejected is similar because their specific heat rejection is higher. This slight variation in heat rejection can be catered with adjustment in cooling water flow rate.

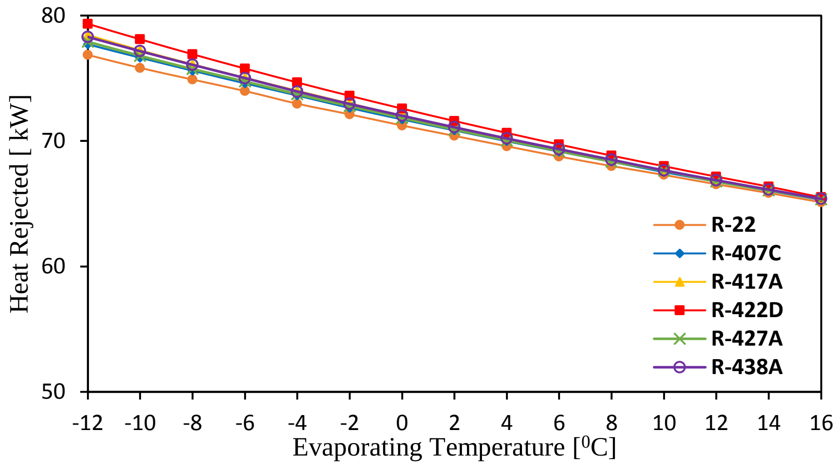

Figure 4.6 Effect of evaporating temperature on heat rejected, constant ref.cap ($T_k = 40°C$).

The compressor power and volume flow scenarios shows that the analysis based on constant refrigerating capacity violates system constraints and this criteria is not suitable for comparing the performance of refrigerants in a system in which modifications are to be avoided. In subsection 4.2.1.1.2 a method based on constant volume flow is used and results obtained are presented.

4.2.1.1.2 Volume Flow as Constraint

In this approach the volume flow of 15.4dm3/sec at the 4^0C evaporating and 40^0C condensing temperature is kept constant. The mass flow was then calculated by dividing volume flow by suction specific volume at compressor inlet. The results are given in table 4.6. The percent change of parameters relative to R22 are provided table 4.7.

Table 4.6 Results for R22 and alternate refrigerants, constant vol. flow ($T_e = 4°C$, $T_k = 40°C$).

Parameter	R22	R407C	R417A	R422D	R427A	R438A
Dis. Temp ^{0}C	69.2	62	52.8	52.1	59.3	56.1
Mass Flow, kg/sec	0.36	0.37	0.39	0.47	0.37	0.40
Vol. Flow, dm^3/sec	15.4	15.4	15.4	15.4	15.4	15.4
Ref. Capacity, kW	58.1	58	48	52.3	55.7	53.5
Comp. Power, kW	11.4	12	10	11.3	11.4	11.1
COP	5.1	4.9	4.8	4.6	4.9	4.8
Heat Rejected, kW	69.2	70.5	57.8	63.2	67.2	64.6
VCC, kJ/m^3	3773	3803	3110	3395	3618	3474

Table 4.7 Percent change of parameters of alternate refrigerants from R22, constant volume flow.

Parameter	R22	R407C	R417A	R422D	R427A	R438A
Dis.Temp, ^0C	69.2	-10.4%	-23.6%	-24.7%	-14.3%	-19%
Mass Flow, kg/sec	0.36	+2.7%	+8.3%	+30.5%	+2.7%	+11%
Vol. Flow, dm^3/sec	15.4	0	0	0	0	0
Ref. Capacity, kW	58.1	0	-17.4%	-10%	-4.1%	-7.8%
Comp. Power, kW	11.5	+4.3%	-14%	-1.7%	-0.9%	-3.5%
COP	5.1	-4%	-5.8%	-9.8%	-4%	-5.7%
Heat Rejected, kW	69.5	+1.5%	-16.8%	-9%	-3.3%	-7%
VCC, kJ/m^3	3773	+0.8	-17.5	-10	-4	-8

The results obtained with constant volume flow show that the mass flow of alternate refrigerants is greater than R22. In contrast to the mass flow rate calculated with constant refrigerating capacity, the mass flow calculated with constant volume flow increases with increase in evaporating temperature. This is due to the fact that here mass flow is inversely proportional to suction specific volume which decreases with increase in evaporating temperature. Understanding of the mass flow of refrigerant at various temperatures is important because it directly controls the refrigerating capacity and compressor power. The highest mass flow is for R422D which is 30.5% higher, followed by R438A being 11% higher than R22 at the evaporation temperature of 4^0C. R407C and R427A have close mass flows with R22. In figure 4.7, the effect of evaporating temperature on mass flow is presented. It is observed that when evaporating temperature increases the difference in mass flow of alternate refrigerants and R22 increases. From -12^0C to 2^0C the mass flow of R22, R407C and R427A are almost same but from 4^0C mass flow of R407C start increasing. This is because of at relatively high temperatures the specific suction volume of R407C and R427A decreases more rapidly than R22. Between 4 to 16^0C on the average basis the mass flow of R407C and R427A is 5% is higher than R22.

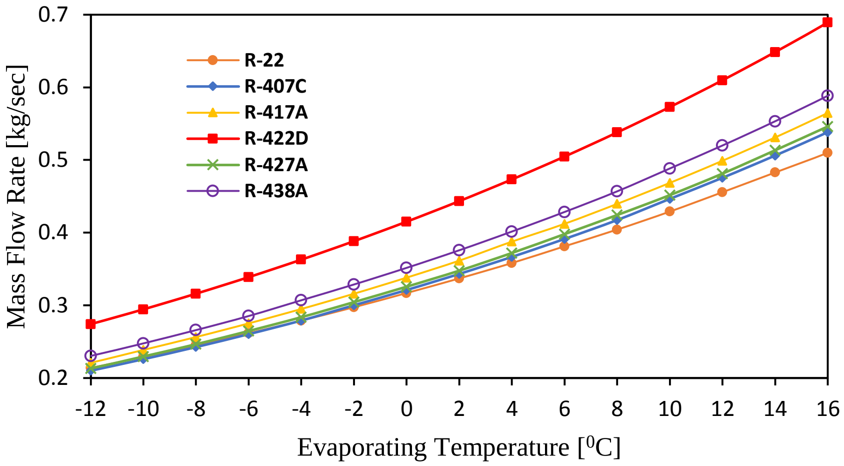

Figure 4.7 Effect of evaporating temperature on mass flow rate, constant vol.flow ($T_k = 40°C$).

The refrigerating capacity and compressor power trends are presented in figure 4.8 and 4.9. At the design point the capacities of R417A, R422D and R438A are respectively lower by 17.4%, 10% and 7.8%. Also at the entire evaporating temperature range their capacities are consistently lower than R22. After R422D the highest mass flow is of R417A yet it has the lowest refrigerating capacity which is attributed to its low refrigerating effect. Also R417A has lowest compressor power of 10kW. The refrigerating capacity of R438A is only slightly higher than R422D but both have similar compressor power. In this case R438A has an advantage over R422D because of its low mass flow rate. The capacities and compressor power of R407C are directly influenced by its mass flow variation. From 4^{0}C the refrigerating capacity of R407C start getting higher than R22 however the compressor power start increasing earlier at about -4^{0}C. On the average basis, between -4^{0}C and 16^{0}C the compressor power of R407C is 5.4% higher than R22. As stated earlier, at any operating point the compressor power of R22 is treated as constraint, this increase in refrigerating capacity cannot be effectively utilized. For R427A the difference in capacity relative to R22 decreases with increasing temperature. Moreover the compressor power of R427A is very similar to R22.

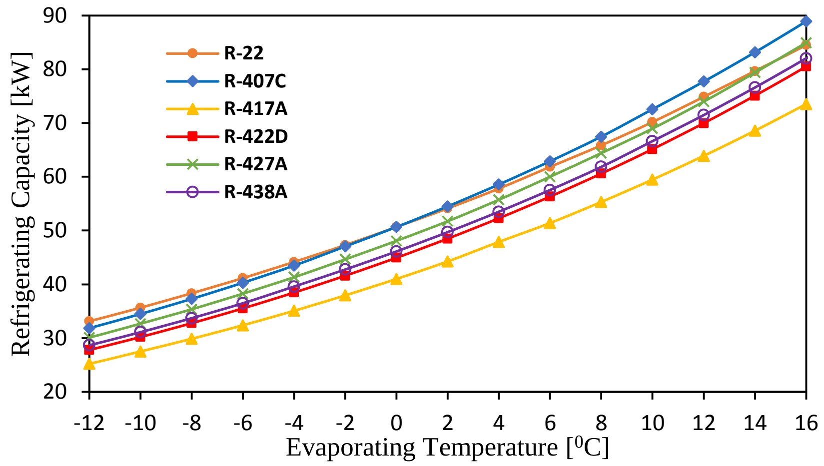

Figure 4.8 Effect of evaporating temperature on refrigerating capacity, constant vol.flow ($T_k = 40°C$).

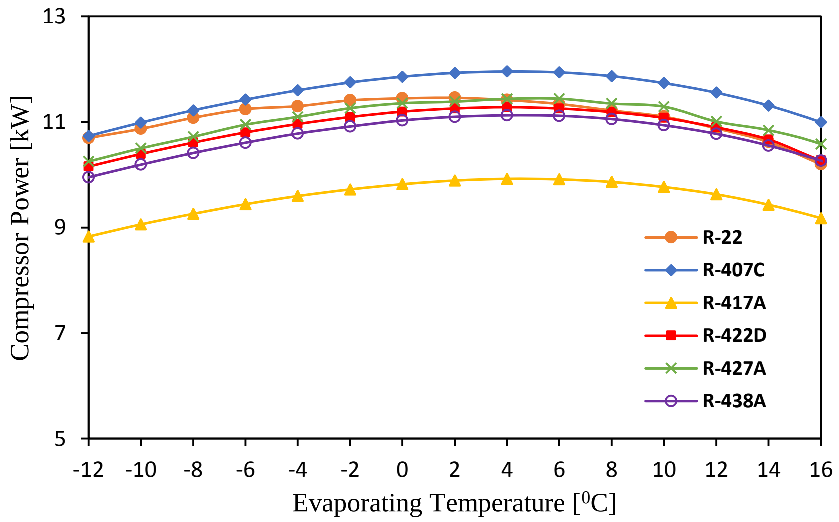

Figure 4.9 Effect of evaporating temperature on compressor power, constant vol.flow ($T_k = 40°C$)

VCC, an important parameter in determining the size of compressor is plotted in figure 4.10. R407C and R427A have close VCCs to R22. The VCCs 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 will be capable of drawing the volume needed to produce required cooling capacity.

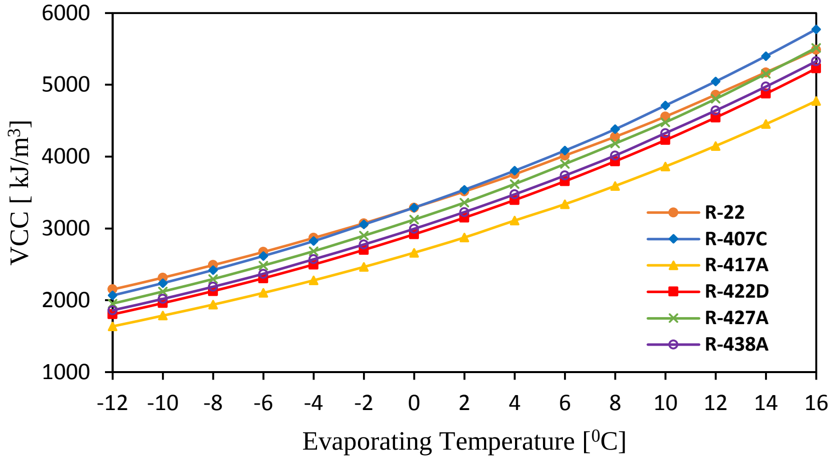

Figure 4.10 Effect of evaporating temperature on VCC ($T_k = 40°C$).

The increase in mass flow of the refrigerants also increases the condenser heat rejected. The heat rejected by each refrigerant is shown in figure 4.11. As evident from the figure except for R407C all the refrigerants has lower heat rejection than R22. The lowest heat rejected at the design point as from table 4.7 is of R417A which is 16.8% less than R22. R422D and R438A has similar heat rejections. For R407C at low temperatures the heat rejected is slightly lower than R22 however with increasing evaporating temperature heat rejection of R407C gets higher than R22. At the evaporating temperature of 4^0C the heat rejected by R407C is 1.5% higher than R22. On the average basis between 0^0C to 16^0C the heat rejected by R407C is 3.2% higher than R22.

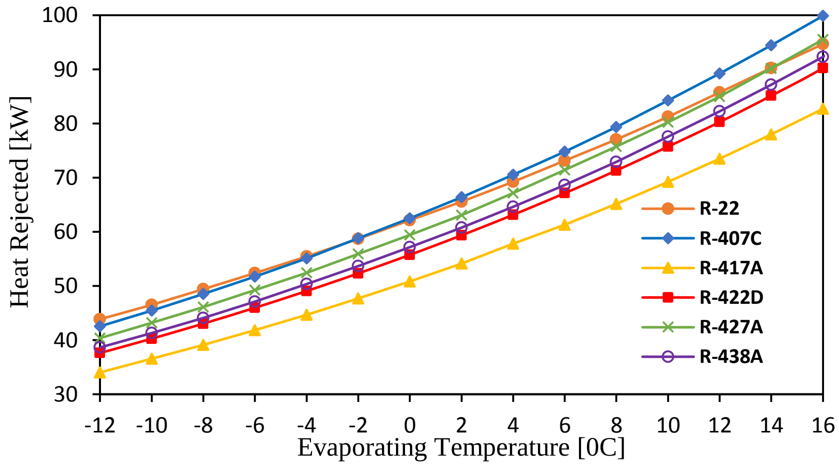

Figure 4.11 Effect of evaporating temperature on heat rejected, constant vol.flow ($T_k = 40°C$)

Calculations based on constant volume flow well addressed the inherent constraints of the system. The limitation of this procedure is not taking into account the volumetric efficiency which effects the suction volume flow in the compressor. As discussed in section 3.5 the clearance volume in cylinder and leakage losses reduces the suction volume flow. This reduction is a function of pressure ratio and can be expressed in terms of volumetric efficiency. The volumetric efficiency has direct impact on mass flow which in turn controls the refrigerating capacity and power consumption. It is therefore important to investigate the performance of refrigerants considering the volumetric efficiency. In the section below, a method based on volumetric efficiency is used for calculating the mass flow. Using this value of mass flow remaining parameters are obtained.

4.2.1.1.3 Volumetric Efficiency Approach

On the basis of the available information 5% clearance volume is taken, the clearance volumetric efficiency at the design point from equation (3.15)

$$\eta_{vol,ac} = \frac{0.36 * 0.043}{0.0222} = 69.7\%$$

The curve of actual volumetric efficiency of compressor is obtained from testing the compressor in real conditions [34]. Actual volumetric efficiency curve of compressor for various pressure ratios is obtained from experimental testing as in [7, 34]. As seen it is moderately a linearly function of pressure ratio and its curve can be approximated. For close approximation of the performance of refrigerants over a range of temperatures, information of volumetric efficiency is important as it is a function of pressure ratio. Varying evaporating or condensing temperatures varies pressure ratio thus affecting volumetric efficiency. With increasing pressure ratio volumetric efficiency decreases. Equation (3.14) is employed to find the corresponding actual volumetric efficiency at respective point and then mass flow is obtained using equation (3.15) Based on the available information, a *CL* value of 5% is taken. From (3.14), with 69.7%, a value of 0.533 for *n* is obtained which is assumed to be constant. Using (3.14) again with pressure ratio of respective operating point, volumetric efficiency is calculated followed by mass flow rate using (3.15).

Table 4.8 Results for R22 and alternate refrigerants, using vol.efficiency. ($T_e = 4$℃ , $T_k = 40$℃).

Parameter	R22	R407C	R417A	R422D	R427A	R438A
Dis. Temp ^{0}C	69.2	62	52.8	52.1	59.3	56.1
Mass Flow, kg/sec	0.36	0.37	0.39	0.47	0.37	0.40
Vol. Flow, dm^3/sec	15.4	15.4	15.4	15.4	15.4	15.4
Ref. Capacity, kW	58.1	58	48	52.3	55.7	53.5
Comp. Power, kW	11.5	12	10	11.3	11.4	11.1
COP	5.1	4.9	4.8	4.6	4.9	4.8
Heat Rejected, kW	69.2	70.5	57.8	63.2	67.2	64.6
VCC, kJ/m^3	3773	3803	3110	3395	3618	3474

Table 4.9 Percent change of parameters of alternate refrigerants from R22, using vol. efficiency.

Parameter	R22	R407C	R417A	R422D	R427A	R438A
Dis.Temp, ^{0}C	69.2	-10.4%	-23.6%	-24.7%	-14.3%	-19%
Mass Flow, kg/sec	0.36	+2.7%	+8.3%	+30.5%	+2.7%	+11%
Vol. Flow, dm^3/sec	15.4	0	0	0	0	0
Ref. Capacity, kW	58.1	0	-17.4%	-10%	-4.1%	-7.8%
Comp. Power, kW	11.5	+4.3%	-14%	-1.7%	-0.9%	-3.5%
COP	5.1	-4%	-5.8%	-9.8%	-4%	-5.7%
Heat Rejected, kW	69.5	+1.5%	-16.8%	-9%	-3.3%	-7%
VCC, kJ/m^3	3773	+0.8	-17.5%	-10%	-4%	-8%

It is interesting to note that the parameters evaluated at the design condition of 4^{0}C evaporating and 40^{0}C condensing temperature using constant volume flow and volumetric efficiency approach are equal. This is due to similar pressure ratios of R22 and alternate refrigerants at this condition. In order to study the effect of volumetric efficiency when the system is operating under different conditions the parameters are calculated and plotted. For each evaporating temperature the mass flow is calculated from corresponding volumetric efficiency. This mass flow is then used to calculate other parameters. Figure 4.12 shows the effect of evaporating temperature on mass flow rate.

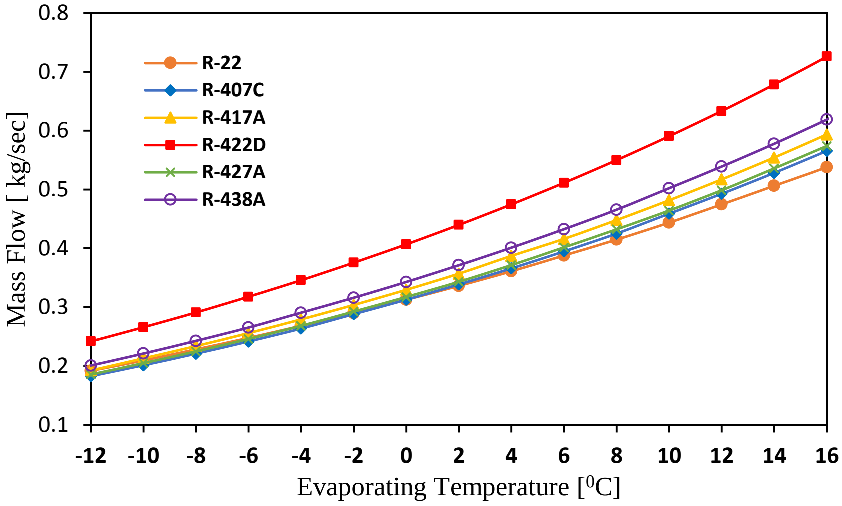

Figure 4.12 Effect of evaporating temperature on mass flow rate using vol. eff ($T_k = 40°C$)

As seen by using volumetric efficiency approach, mass flow variation characteristics are identical to constant volume flow approach. However there are differences in mass flow values calculated with efficiency approach. At low evaporating temperatures the pressure ratio is high which decreases volumetric efficiency thus decreasing mass flow. The increase and decrease in mass flow can be better visualized through volume flow which is calculated as the product of compressor displacement rate and volumetric efficiency at the respective point. In figure 4.13 the variation of volume flow is presented. It is seen that unlike constant volume flow approach, here the volume flow do not remain constant at the constrained value of 15.4dm^3/sec. At lower temperatures the volume flow of refrigerants is less than 15.4dm^3/sec. From about 8^{0}C the volume flow of each refrigerant start increasing with increasing evaporating temperature. Due to its better efficiency, mass and volume flow of R22 are less affected compare to others. Among the alternate refrigerants R422D has better volumetric efficiency. At low temperatures the volumetric efficiency effects are more visible.

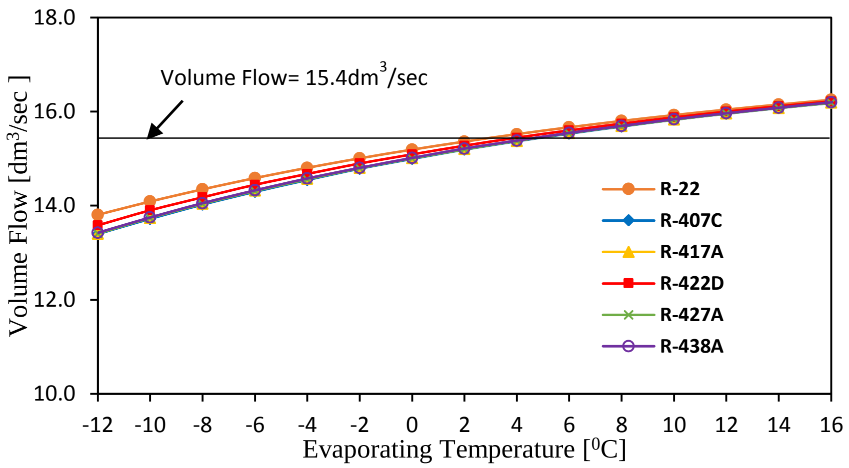

Figure 4.13 Effect of evaporating temperature on volume flow, using vol.eff ($T_k = 40$°C).

Obviously mass flow influences the capacity, compressor power and heat rejection. These parameters are calculated with mass flow obtained using volumetric efficiency and are plotted as shown in figure 4.14, figure 4.15 and figure 4.16.

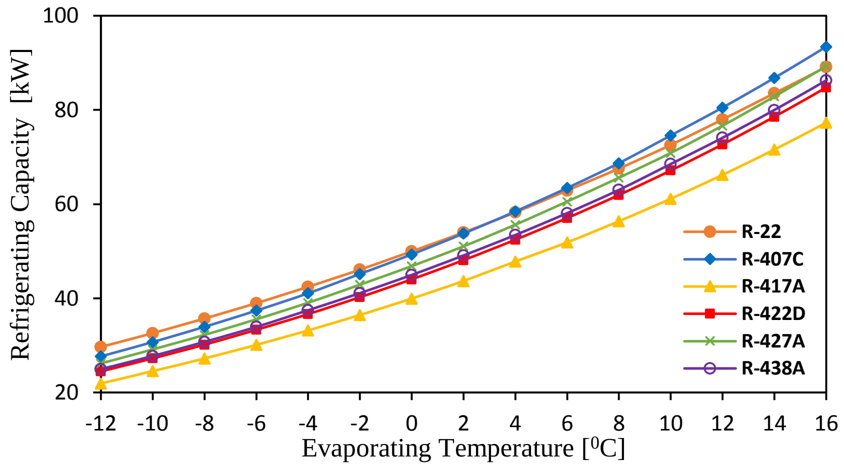

Figure 4.14 Effect of evaporating temperature on refrigerating capacity using vol.eff ($T_k = 40$°C).

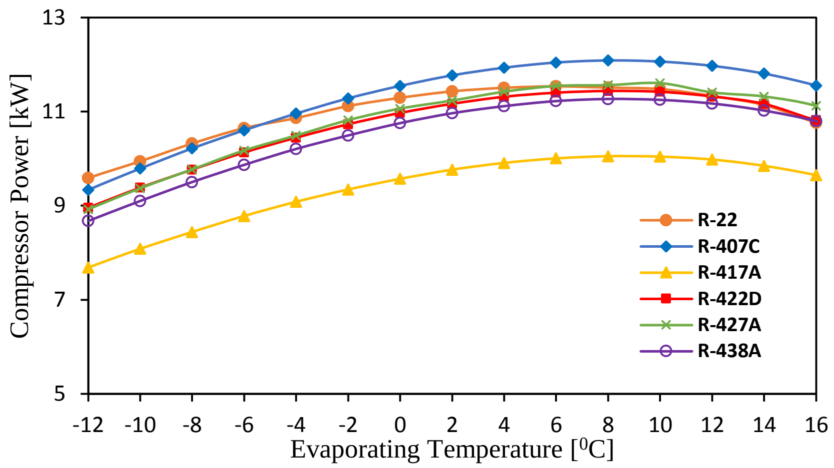

Figure 4.15 Effect of evaporating temperature on compressor power, using vol. eff ($T_k = 40$°C).

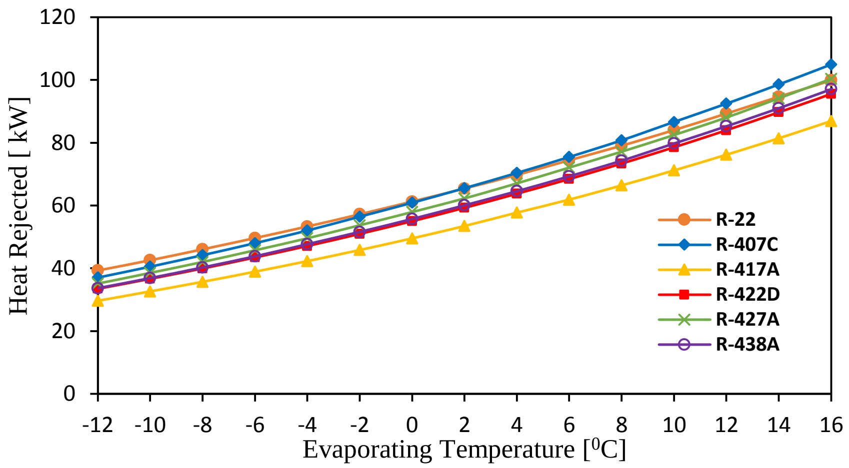

Figure 4.16 Effect of evaporating temperature on heat rejected, using vol. eff ($T_k = 40$°C).

The expansion valve in our system can be regulated to achieve the desired pressure drop. However if the pressure drop differences between R22 and alternate refrigerants are greater it may not be possible to have the required level of adjustment. To investigate this scenario a pressure drop of R22 and alternate refrigerant across the expansion valve are plotted as shown in figure 4.17.

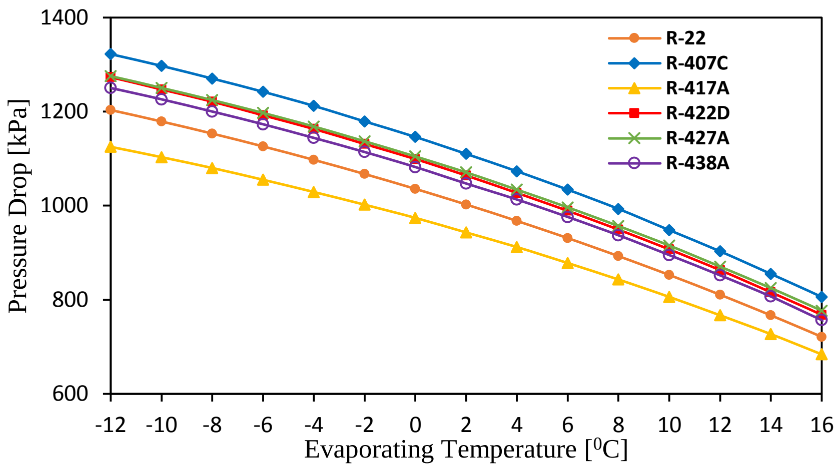

Figure 4.17 Effect of evaporating temperature on pressure drop across expansion valve ($T_k = 40°C$).

Clearly the pressure drops across the expansion valve of R407C, R422D, R427A and R438A are higher than R22. Only R417A has lower pressure drops than R22. It is that the highest pressure drops are of R407C. R422D and R427A has similar pressure drops. The difference between the pressure drop of R22 and alternate refrigerants increases with decreasing evaporating temperature. This means expansion valve become more critical at low evaporating temperatures. Based on pressure drop comparison we can conclude that there may a possible change out of expansion valve for alternate refrigerants.

From the results it is noted that the performance parameters based on volumetric efficiency have the same behaviour as constant volume flow. Use of volumetric efficiency is important to sort out any unusual behaviour of the refrigerant when pressure ratio changes. It is observed that in terms of volumetric efficiency none of the refrigerants showed significant deviation relative to others. This is due to nearly similar pressure ratios at the particular operating point. Over the entire evaporating range, on an average basis values of parameters of each of the refrigerants are tabulated for constant volume flow and volumetric efficiency method. Keeping in view the observed trends, two intervals of evaporating temperatures are selected. One from -12^{0}C to 0^{0}C and second from 2^{0}C to 16^{0}C. Table 4.10 and table 4.11 provide values for the interval of -12^{0}C to 0^{0}C and 2^{0}C to 16^{0}C.

Table 4.10 Average value of parameters in the evaporating range of -12^{0}C to 0^{0}C ($T_k = 40$°C).

Parameter	R22	R407C	R417A	R422D	R427A	R438A
Dis. Temp ^{0}C	77.5 77.5	67.5 67.5	55.3 55.3	54.5 54.5	64 64	60 60
Mass Flow, kg/sec	0.26 0.25	0.26 0.24	0.28 0.26	0.34 0.32	0.27 0.25	0.29 0.27
Vol. Flow, dm^3/sec	15.4 14.5	15.4 14.3	15.4 14.3	15.4 14.4	15.4 14.3	15.4 14.3
Ref. Capacity, kW	41.5 39.4	40.7 38	32.7 30.5	36 33.7	38.6 36	36.9 34.4
Comp. Power, kW	11.1 10.5	11.4 10.5	9.4 8.7	10.7 10.1	11 10.1	10.6 9.8
COP	3.7 3.7	3.6 3.6	3.5 3.5	3.3 3.3	3.5 3.5	3.5 3.5
Heat Rejected, kW	52.6 50	52.1 48.4	42.1 39.2	46.3 43.8	49.5 46.1	47.5 44.2
VCC, kJ/m^3	2694 2694	2645 2645	2124 2124	2331 2331	2509 2509	2397 2397

NOTE: The first value in the cell is based on constant volume flow. The second value is based on using volumetric efficiency.

Table 4.11 Average value of parameters in the evaporating range of 2^{0}C to 16^{0}C ($T_k = 40$°C).

Parameter	R22	R407C	R417A	R422D	R427A	R438A
Dis. Temp ^{0}C	65.6 65.6	59.4 59.4	51.7 51.7	51.2 51.2	57.2 57.2	54.5 54.5
Mass Flow, kg/sec	0.42 0.43	0.44 0.45	0.46 0.47	0.56 0.58	0.44 0.45	0.48 0.49
Vol. Flow, dm^3/sec	15.4 15.8	15.4 15.7	15.4 15.7	15.4 15.8	15.4 15.7	15.4 15.7
Ref. Capacity, kW	68.6 70.7	70.7 72.4	58 59.5	63.5 65.3	67.4 69.1	65 66.6
Comp. Power, kW	11 11.3	11.7 12	9.7 10	11 11.3	11.2 11.4	11 11.1
COP	6.3 6.3	6.1 6.1	6 6	5.8 5.8	6.1 6.1	6 6
Heat Rejected, kW	79.6 82.1	82.4 84.4	67.7 69.4	74.1 76.6	78.6 80.5	75.8 77.7
VCC, kJ/m^3	4454 4454	4592 4592	3769 3769	4126 4126	4375 4375	4215 4215

NOTE: The first value in the cell is based on constant volume flow. The second value is based on using volumetric efficiency.

The above results indicate that for the interval of -12^{0}C to 0^{0}C, the values of parameters obtained by taking into account the volumetric efficiency are less than constant volume flow method. It is seen that for the evaporating interval of -12^{0}C to 0^{0}C, volumetric efficiency method consistently gives lower values. This is this due to dominant effects of volumetric efficiency at lower temperatures. From 2^{0}C to 16^{0}C, volumetric efficiency method gives higher values than constant volume flow method. However this increase at higher temperatures is less compared to decrease at lower temperatures. It is to be noted that the discharge temperature, COP and volumetric capacity remain unaffected by using any of the method. In table 4.12 percent change in values based on volumetric efficiency relative to constant volume flow are presented.

Table 4.12 Percent error in parameters using constant volume flow method.

Parameter	-12^{0}C to 0^{0}C	2^{0}C to 16^{0}C
Dis. Temp ^{0}C	0	0
Mass Flow, kg/sec	-3.8% to -7.7%	2.1% to 3.6%
Volume Flow, dm^3/sec	-5.8% to -7.1%	2% to 2.6%
Ref. Capacity, kW	-5.1% to -6.8%	2.4% to 3.1%
Compressor Power, kW	-5.4% to -8.2%	1% to 3.1%
COP	0	0
Heat Rejected, kW	-5% to -7%	2.4% to 3.4%
VCC, kJ/m^3	0	0

From the above results it can be concluded that both the methods are appropriate for carrying out analysis but with constant volume flow, results at low temperatures tend to have greater error. Considering volumetric efficiency, more realistic values of mass flow rate and other parameters are obtained. Therefore this method is more reliable than constant volume flow method. Our further comparison of refrigerants will be based on this method. In table 4.13 the average values of parameters are summarised for R22 and alternate refrigerants.

Table 4.13 Average values of parameters of over the evaporating range. ($T_k = 40°C$)

Parameter	R22	R407C	R417A	R422D	R427A	R438A
Dis. Temp ^0C	77.5 to 65.6	67.5 to 59.4	55.3 to 51.7	54.5 to 51.2	64 to57.2	60 to54.5
Mass Flow kg/sec	0.25 to 0.43	0.24 to 0.45	0.26 to 0.47	0.32 to 0.58	0.25 to 0.45	0.27 to 0.49
Vol. Flow dm^3/sec	14.5 to 15.8	14.3 to 15.7	14.3 to 15.7	14.4 to 15.8	14.3 to 15.7	14.3 to 15.7
Ref. Capacity, kW	39.4 to 70.7	38 to 72.4	30.5 to 59.5	33.7 to 65.3	36 to 69.1	34.4 to 66.6
Comp. Power, kW	10.5 to 11.3	10.5 to 12	8.7 to 10	10.1 to 11.3	10.1 to 11.4	9.8 to 11.1
COP	3.7 to 6.3	3.6 to 6.1	3.5 to 6	3.3 to 5.8	3.5 to 6	3.5 to 6
Heat Rejected kW	50 to 82.1	48.4 to 84.4	39.2 to 69.4	43.8 to76.6	46.1 to 80.5	44.2 to 77.7
VCC, kJ/m^3	2694 to 4454	2645 to 4592	2124 to 3769	2331 to 4126	2509 to 4375	2397 to 4215

Table 4.14 Average percent change in parameters of alternate refrigerants over evaporating range.

Parameter	R22	R407C	R417A	R422D	R427A	R438A
Dis. Temp ^0C	77.5 to 65.6	-12.9% to -9.5%	-28.6% to -21.2%	-29.7% to -22%	-17.4% to -12.8%	-22.6% to -17%
Mass Flow,kg/sec	0.25 to 0.43	-4% to +4.7%	+4% to +9.3%	+28% to +35%	0 to +4.7%	+8% to +14%
Vol.Flow,dm^3/sec	14.5 to 15.8	-1.4% to -0.6%	-1.4% to -0.6%	-0.7% to 0	-1.4% to -0.6%	-1.4% to -0.6%
Ref. Capacity, kW	39.4 to 70.7	-3.6% to +2.4%	-22.6% to -15.8%	-14.5% to -7.6%	-8.6% to -2.3%	-12.7% to -5.8%
Comp. Power, kW	10.5 to 11.3	0 to +6.2%	-17.1% to -11.5%	-3.8% to 0	-3.8% to +1%	-6.7% to -1.8%
COP	3.7 to 6.3	-2.7% to -3.2%	-5.4% to -4.8%	-10.8% to -8%	-5.4% to -4.8%	-5.4% to -4.8%
Heat Rejected,kW	50 to 82.1	-3.2% to +2.8%	-21.6% to -15.5%	-12.4% to -6.7%	-7.8% to -1.9%	-11.6% to -5.4%
VCC, kJ/m^3	2694 to 4454	-1.8% to +3.1%	-21.2% to -15.4%	-13.5% to -7.4%	-7% to -1.8%	-11% to -5.4%

The above tabulated results show that the discharge temperatures of alternate refrigerants are lower than R22. Among the alternate refrigerants the highest discharge temperatures are of R407C. R422D has lowest discharge temperatures. The lower discharge temperatures are beneficial for compressor as they prevent thermal deterioration of lubricating oil, offering compressor a longer life. As previously noticed here also the highest mass flow is of R422D which is about 28% to 35% higher than R22. R407C and R427A has similar mass flows. The volume flow differences in R22 and alternate refrigerants at lower temperatures are more compared to high temperatures due to dominant effects of volumetric efficiency. In terms of refrigerating capacity R407C is most efficient followed by R427A. It is interesting to note that in the evaporating range of 2^0C to 16^0C the capacity of R407C is about 2.6% higher than R22. R422D and R438A has nearly same performance with respect to R22 but R438A has slightly higher capacity than R422D. The lowest refrigerating capacity is recorded for R417A which is about 15.8% to 22.6% lower than R22 over the entire evaporating range. Compressor power consumption is as important as refrigerating capacity because it serves as the key parameter in deciding the overall efficiency of the plant. Viewing compressor power values it is noted that none of the refrigerants consumes higher power than R22, however R407C indicates an increase of about 6.2% when the system is operating in the evaporating range 2^0C to 16^0C but this is acceptable. After R407C, among the selected candidates, R422D and R427A have high compressor powers which are about 0 to 3.8% lower than R22 over the entire operating envelope. Unlike similar refrigerating capacities, R438A consumes less power than R422D. The power consumed by R438A is about 1.8% to 6.7% while that of R422D is about 3.8% lower than R22. The lowest compressor power is of R417A which is about 11.5% to 17.1% lower than R22. Looking at the COP values we notice that none of the refrigerant has higher COP than R22. R417A, R427A and R438A has similar COP being 4.8% to 5.4% lower than R22. R407C has superior COP to others being 2.7% to 3.2% lower than R22. The heat rejected and volumetric cooling capacity of alternate refrigerants are lower than R22 however R407C shows a slight increase in both parameters at elevated evaporating temperatures.

Now the energy performance assessment when the system operates at various evaporating temperatures at fixed condensing temperature has been established, investigation on performance parameters need to be made by varying condensing temperature because it also directly influences the system performance. To study this effect the condensing temperature is varied between 30^0C to 50^0C with the interval of 5^0C. Two evaporating temperatures, -10^0C and 4^0C are fixed against which the condensing temperatures are varied. The results obtained are plotted and presented in the following figures.

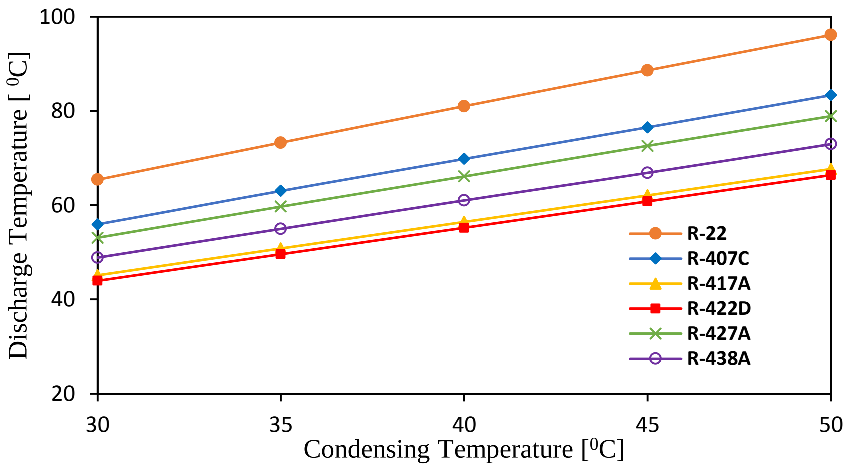

Figure 4.18 Effect of condensing temperature on discharge temperature ($T_e = -10°\text{C}$).

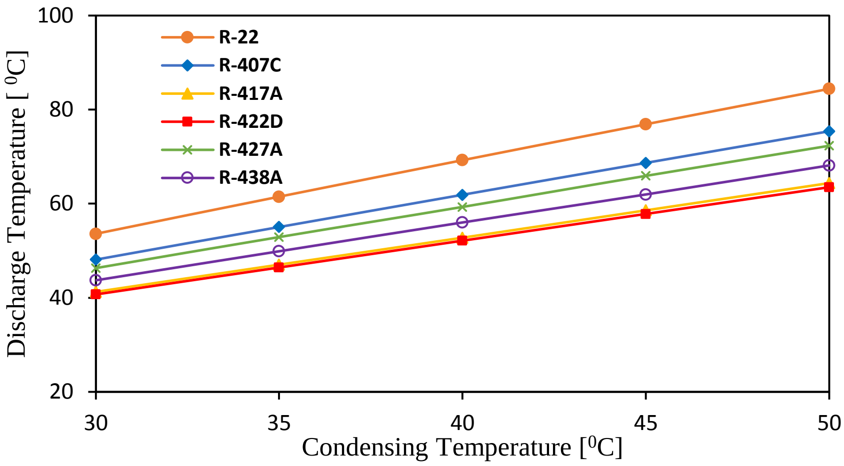

Figure 4.19 Effect of condensing temperature on discharge temperature ($T_e = 4°C$).

It is seen that the discharge temperature increases with increase in condensing temperature. The higher the condensing temperature, higher the discharge temperature. From the previous study at fixed condensing temperature of 40^0C, here also the discharge temperature of R22 is highest followed by R407C and R427A. It means that at any condensing temperature this trend would be observed. However at low evaporating temperature the discharge temperature of individual refrigerant is higher by some degrees due to increased pressure difference. For instance, the discharge temperature at -10^0C and 4^0C evaporating and 35^0C condensing temperature of R407C is 63^0C and 55^0C and of R427A is 53^0C and 60^0C.

The effect of condensing temperature on mass flow is presented in figure 4.20 and figure 4.21. R422D has maintained its highest mass flow at any condensing temperature followed by R438A. R22, R407C and R427A has close mass flows. It is observed that for any refrigerant, varying the condensing temperature at a fixed evaporating temperature, there is very less change in mass flow. This is because mass flow is dependent on suction specific volume and volumetric efficiency. The suction specific volume is same for fixed evaporating temperature and volumetric efficiency do not change rapidly unless the condensing temperature is sufficiently increased or evaporating temperature is sufficiently decreased. Also we notice that at -10^0C temperature mass flow has lower values than 4^0C due to decrease in volumetric efficiency which subsequently lowers the mass flow.

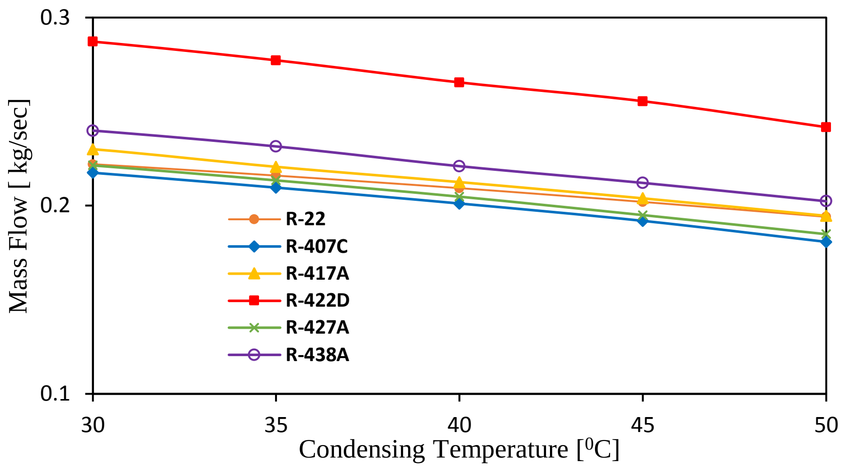

Figure 4.20 Effect of condensing temperature on mass flow rate ($T_e = -10°C$).

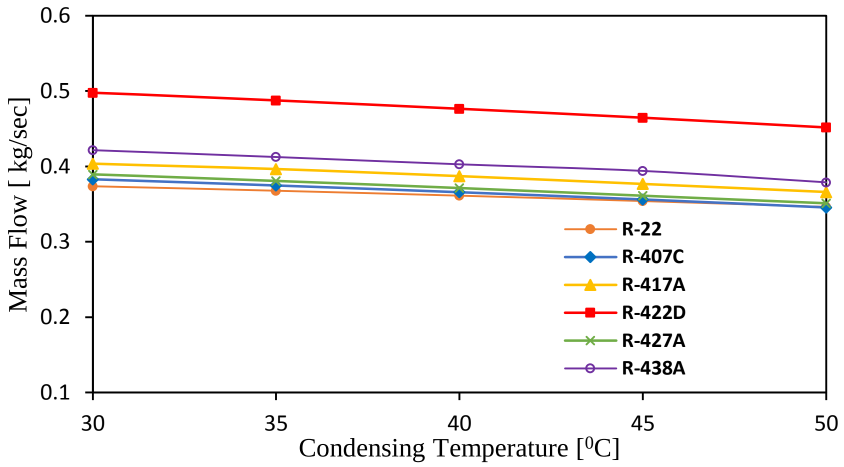

Figure 4.21 Effect of condensing temperature on mass flow rate ($T_e = 4°C$).

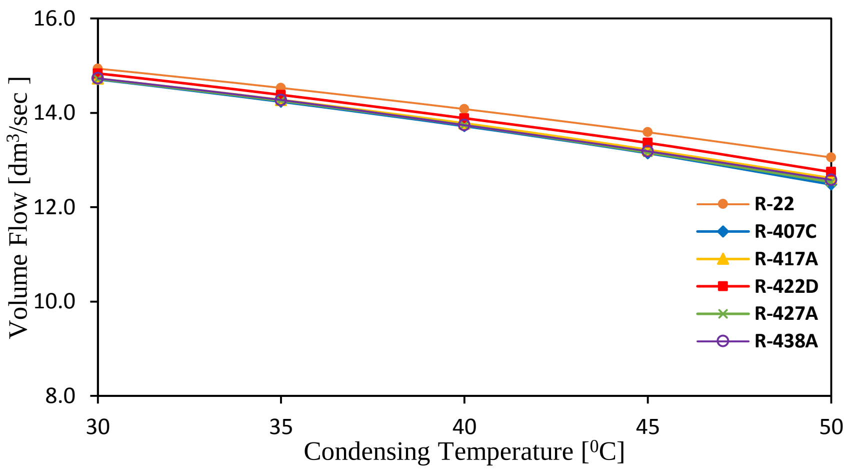

Figure 4.22 Effect of condensing temperature on volume flow rate ($T_e = -10°C$).

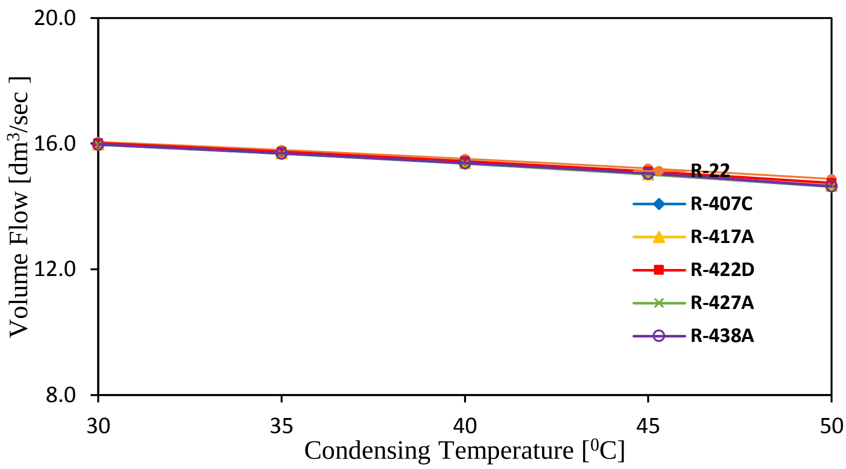

Figure 4.23 Effect of condensing temperature on discharge temperature ($T_e = 4$°C).

The impact on volume flow as a result of changing condensing temperature is shown in figure 4.22 and figure 4.23. The volume flow of R22 and alternate refrigerants are identical and decreases with increase in condensing temperature. At evaporating temperature of -10^0C, the drop in volume flow is more prominent due to lower volumetric efficiencies.

The refrigerating capacity decreases when condenser temperature increases. This is due to with increasing condenser temperature both the mass flow and refrigerating effect decrease. The highest capacity at any condensing temperature at -10^0C and 4^0C evaporating temperature is of R22 followed by R407C and R427A. As previously noted at the condensing temperature of 40^0C with varying evaporating temperature, here also the refrigerating capacity of R422D and R438A at various condensing temperatures is close to each other. R417A has lowest refrigerating capacity. We see that at low condensing temperatures the deviation in capacities of alternate refrigerants relative to R22 decreases. For example at the evaporating temperature of 4^0C, R417A has a capacity of 47.8kW which is 18% less than capacity than R22 at condensing temperature of 40^0C. Decreasing the condenser temperature to 30^0C, R417A capacity turns out to be 56kW being 14% less than R22. Other refrigerants also exhibits the same behaviour however R407C shows slight increase in capacity against R22 at high evaporating temperatures but this difference diminishes with the rise in condensing temperatures. This means that at low condensing temperatures, alternate refrigerants have better refrigerating capacity.

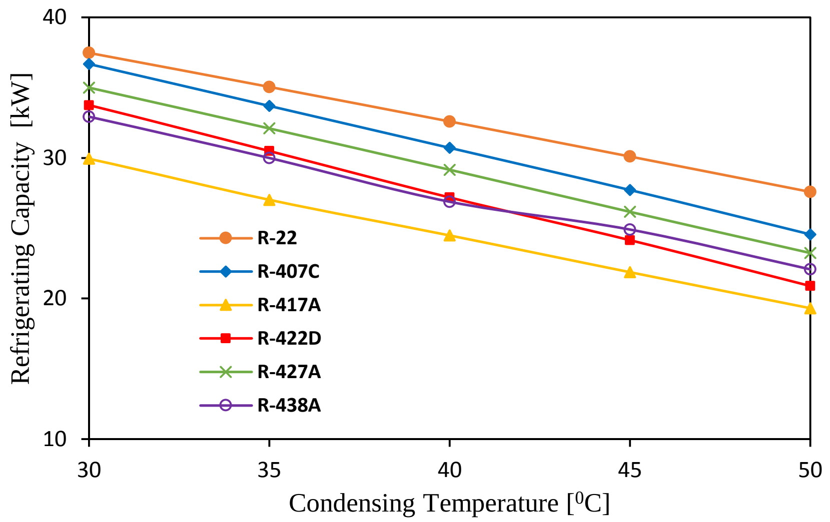

Figure 4.24 Effect of condensing temperature on refrigerating capacity ($T_e = -10$°C).

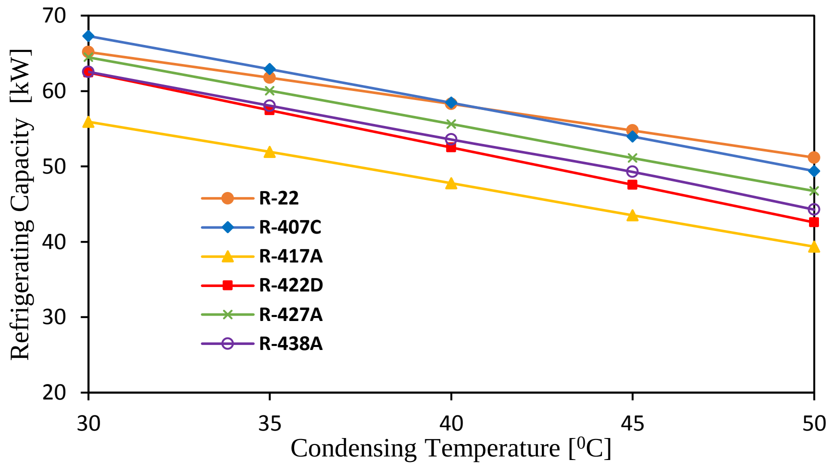

Figure 4.25 Effect of condensing temperature on refrigerating capacity ($T_e = 4$°C).

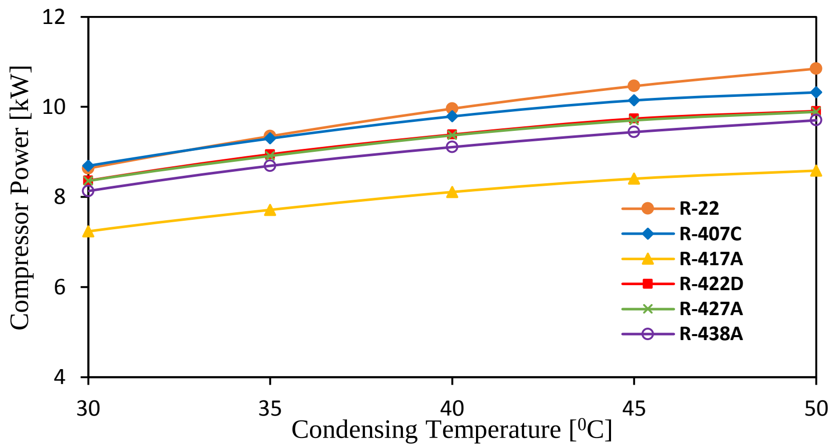

Figure 4.26 Effect of condensing temperature on compressor power ($T_e = -10$°C).

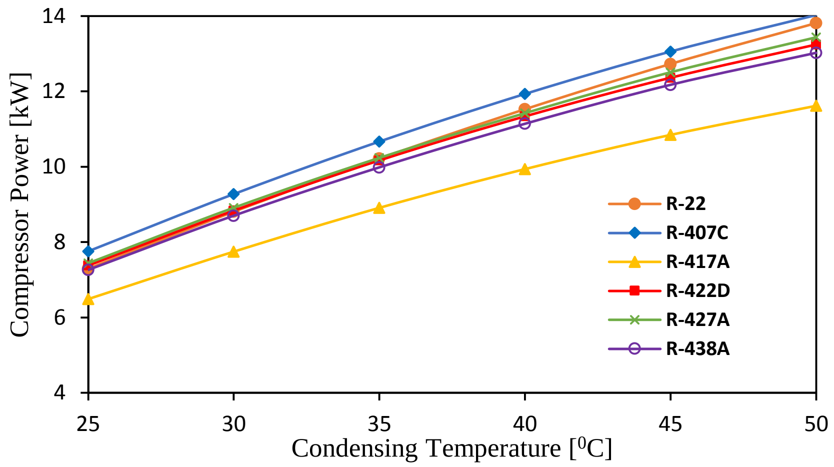

Figure 4.27 Effect of condensing temperature on compressor power (T_e = 4°C).

Compressor power variation with condensing temperature is plotted in figure 4.26 and figure 4.27. Compressor power of each refrigerant increases with increase in condensing temperature. At -10^{0}C evaporating temperature, R22 has higher compressor power than others. As noted earlier for refrigerating capacity of R407C, the compressor power of R407C also increases against R22 when the evaporating temperature rises (in our case 4^{0}C). These power differences between R22 and R407C continue to reduce when condensing temperature gets high as evident from figure 4.25. After R407C, R422D and R427A consume high compressor power followed by R438A. The lowest compressor power at any condensing temperature is that of R417A. An interesting observation is the manner in which compressor power changes for low and high evaporating temperatures over a range of condensing temperatures. The work of compression increases with increasing condensing temperature but we see at -10^{0}C evaporating temperature the compressor power do not change rapidly as that of when evaporating temperature is 4^{0}C. The reason is reduced effective mass flow due to higher suction specific volume and lower volumetric efficiency at low evaporating temperatures.

Figure 4.28 and Figure 4.29 describe the effect of condensing temperature on COP. COP of each refrigerant drops when condensing temperature rises because refrigerating effect decreases and compressor work increases. R22 has highest COP_C followed by R407C and R427A. R417A and R438A have close COP. The lowest COP is that of R422D. At the corresponding condensing temperature, COP at -10^{0}C is lower than 4^{0}C evaporating temperature due to the fact that refrigerating effect drops and compressor work rises at low evaporating temperatures.

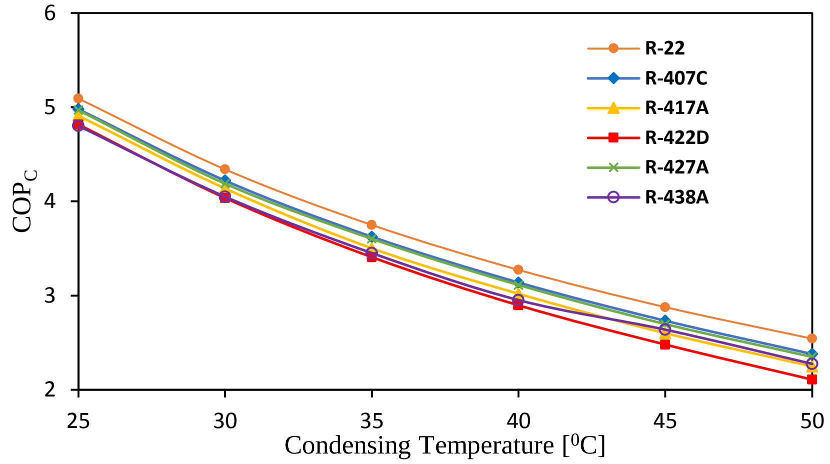

Figure 4.28 Effect of condensing temperature on COP ($T_e = -10°C$).

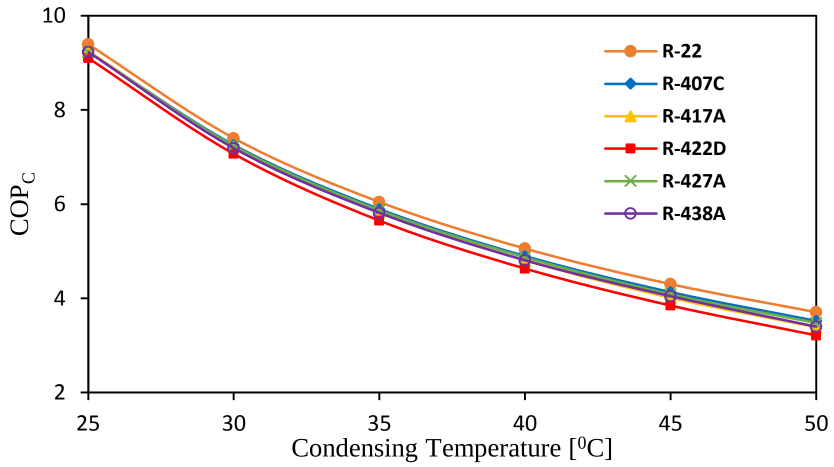

Figure 4.29 Effect of condensing temperature on COP ($T_e = 4°C$).

Investigation on the sufficiency of heat exchange area of existing condenser is made by comparing heat rejection of refrigerants. The heat rejected at various condensing temperatures by R22 and alternates are presented in figure 4.30 and figure 4.31. It is noted that at low evaporating temperature (-10^{0}C), the heat rejection of R407C is less than R22. At comparatively high evaporating (4^{0}C) and low condensing temperatures, R407C rejects more heat than R22 but gets close to R22 at high condensing temperatures. At any evaporating or condensing temperature heat rejected by R417A, R422D, R427A and R438A is less than R22.

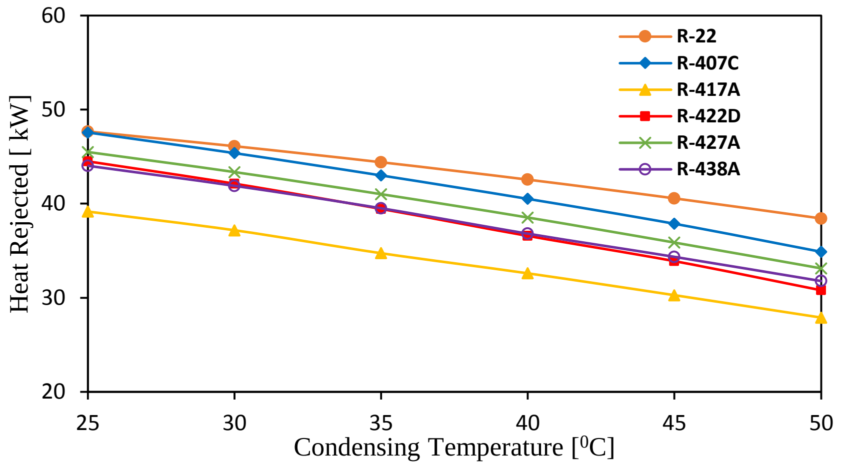

Figure 4.30 Effect of condensing temperature on heat rejected ($T_e = -10$℃).

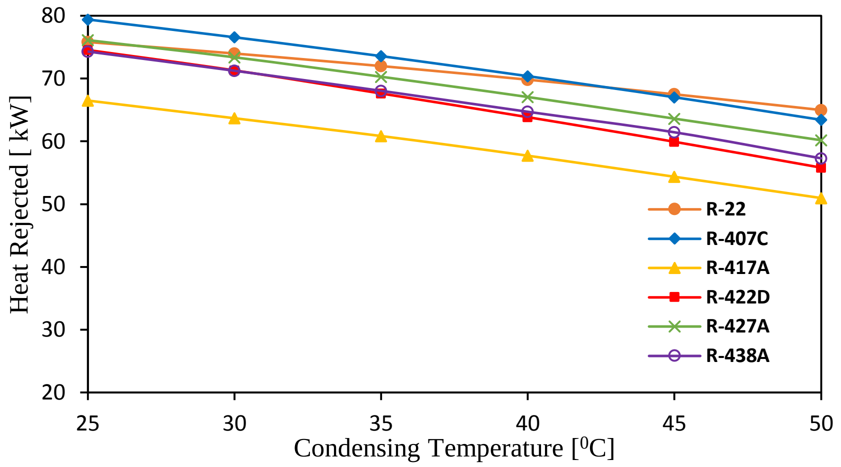

Figure 4.31 Effect of condensing temperature on heat rejected ($T_e = 4$℃).

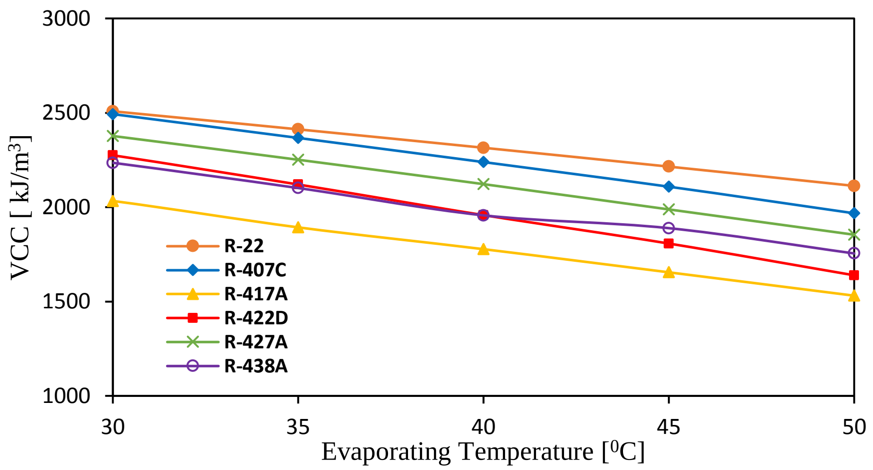

Figure 4.32 Effect of condensing temperature on VCC ($T_e = -10$℃).

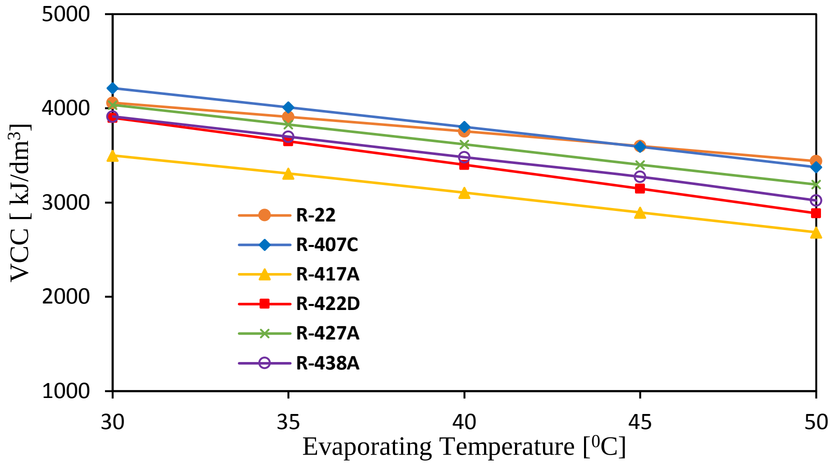

Figure 4.33 Effect of condensing temperature on VCC (T_e = 4°C).

The volumetric cooling capacity which is an important factor in estimating the size of the compressor is plotted for the considered condensing temperatures in figure 4.32 and figure 4.33. For all refrigerants it decreases with increasing condensing temperature because both refrigerating capacity and volume flow drops with rise in condensing temperature. R22 and R407C has higher VCC followed by R427A. R422D and R438A have close VCC. As a direct consequence of its lowest refrigerating capacity, lowest VCC is of R417A.

Comparing the energy performance of R22, R407C, R417A, R422D, R427A and R438A have equipped us with the information that how our system will perform if the respective alternate refrigerant instead of R22 is charged into the system. This is accomplished by simulating the conditions under which system may operate and then evaluating performance parameters related to compressor, condenser, expansion valve and evaporator. From the above analysis we see that none of the refrigerants perform better than R22 as indicated by its highest COP at any operating point. Hence there is no candidate to perfectly replace R22 however each of the considered alternate is suitable to retrofit our R22 system. The cooling capacity, compressor power, heat rejected and VCC of alternate refrigerants suggests that these components do not require any modification however there may be a possible change out of expansion valve as indicated by pressure drops. COP of the alternate refrigerants are quite comparable with R22 and any of the considered zero ODP refrigerant candidates can be picked up to substitute R22. However selection purely based on COP can be misleading in choosing the most suitable refrigerant to substitute R22 because it is the ratio of refrigerating capacity and compressor power. We calculated that at the condensing temperature of 40^0C the average COP between low and high evaporating temperatures of R407C, R417A, R422D, R27A and R438A are 2.7% to 3.2%, 4.8% to 5.4%, 8% to 10.8%, 4.8% to 5.4% and 4.8% to 5.4% less than R22. Similarly at the condensing temperature of 30^0C the COP of these refrigerants between evaporating temperature of -10^0C and 4^0C are 1.3% to 2.3%, 2.7% to 4.6%, 4% to 7%, 2.3% to 2.7% and 4% to 7% less than R22. This analysis suggest that each can be a potential candidate to substitute R22 but this is not true. The refrigerating capacity and compressor power which are the parameters of prime importance should be analysed separately in order to have a better idea of most desirable candidate to substitute R22. Having close COP to others, still R417A is least desirable because of its lowest refrigerating capacity being 15.8% to 22.8% less than R22. It also has the lowest compressor power which is 11.5% to 17% less than R22 but refrigerating capacity cannot be compromised when other refrigerants are capable of providing comparatively high capacity with power achievable by existing compressor. R422D and R438A has similar performance characteristics but the better COP makes R438A favourable than R422D. In the alternates, considering all the parameters, R407C and R427A and R438A showed close performance relative to R22. The behaviour of R407C is different in a fact that its capacity and compressor power fluctuates between low and high evaporating temperatures which is attributed to its mass flow variation, refrigerating effect and work of compression. The mass flow is dependent on suction specific volume and volumetric efficiency and the last two are direct consequence of thermodynamic properties. On the other hand R427A showed more consistent energetic performance relative to R22. From the energetic point of view we can conclude that R407C, R427A and R438A are most suitable alternates to retrofit our R22 system.

4.2.2 Exergy Analysis

First, the base case of 4^0C evaporating and 40^0C condensing temperature for R22 is considered. As per ambient conditions the dead state temperature of 25^0C (298.15K) is taken. From equation (3.5) the exergy destruction in evaporator is

$$\dot{X}_{d,e} = \left(1 - \frac{298.15}{292.15}\right) 58.1 + 0.36\{(245.1 - 406.5) - 298.15(1.1628 - 1.7450)\} = 3.2kW$$

From equation (3.7) the exergy destruction in compressor is

$$\dot{X}_{d,c} = 0.36 * 298.15(1.7799 - 1.7611) = 2kW$$

The exergy destruction in condenser from equation (3.9) is

$$\dot{X}_{d,k} = 0.36\{(442.9 - 249.6) - 298.15(1.7799 - 1.1665)\} = 3.7kW$$

The thermal exergy loss in the condenser is zero because we have assumed that the boundary temperature of the condenser at which heat is rejected is same as dead state temperature.

From equation (3.10) the exergy destruction in expansion valve is

$$\dot{X}_{d,exp} = 0.36 * 298.15(1.1628 - 1.1520) = 1.2kW$$

From equation (3.16) the total exergy destroyed is

$$\dot{X}_{d,tot} = 3.2 + 2 + 3.7 + 1.2 = 10.1kW$$

From equation (3.17) the exergy efficiency is

$$1 - \frac{10.1}{11.5} = 0.12$$

The same equations are applied to alternate refrigerants and results are presented in table in 4.15.

Table 4.15 Exergy destroyed (kW) and exergy efficiency, ($T_e = 4°\text{C}$, $T_k = 40°\text{C}$, $T_0 = 25°\text{C}$)

$\dot{X}_d$	R22	R407C	R417A	R422D	R427A	R438A
$\dot{X}_{d,c}$	2	2.1	1.8	2.1	2.1	2
$\dot{X}_{d,k}$	3.7	3.7	2.8	3.2	3.5	3.2
$\dot{X}_{d,ex}$	1.2	1.4	1.4	1.7	1.5	1.4
$\dot{X}_{d.e}$	3.2	3.4	2.7	3	3	3.2
$\dot{X}_{d,tot}$	10.1	10.6	8.8	10	10.1	9.8
η_{ex}	12	11	11.7	8	11.2	11.4

From the results we see that the total exergy destruction of R22 is higher than R417A, R422D, R427A and R438A by 0.7 to 2.3kW. However the exergy destroyed by R407C is higher by 0.5kW than R22. Among the alternates the highest exergy destroyed is of R407C and lowest exergy destroyed is of R417A. None of the refrigerants have better exergy efficiency than R22. R422D has worst exergy efficiency.

In order to better understand the exergy destruction of each of the considered refrigerant relative to R22, the exergy destroyed in individual component is calculated and plotted by varying evaporating temperature with a fixed condensing temperature of 40^0C. Figure 4.34 shows the exergy destroyed in compressor. Clearly when the compressor operates with R407C, R422D and R427A, it has higher exergy destruction rate than R22. However this difference diminishes at lower evaporating temperatures. R438 has similar compressor exergy destruction as R22. The exergy destruction in compressor is governed by two factors, the entropy generation and mass flow. Although R22 has greater entropy generation but the higher mass flow of alternate refrigerants results in their elevated exergy destruction rates. The lowest exergy destruction in compressor at any operating condition is of R417A due to its comparatively low mass flow and entropy generation.

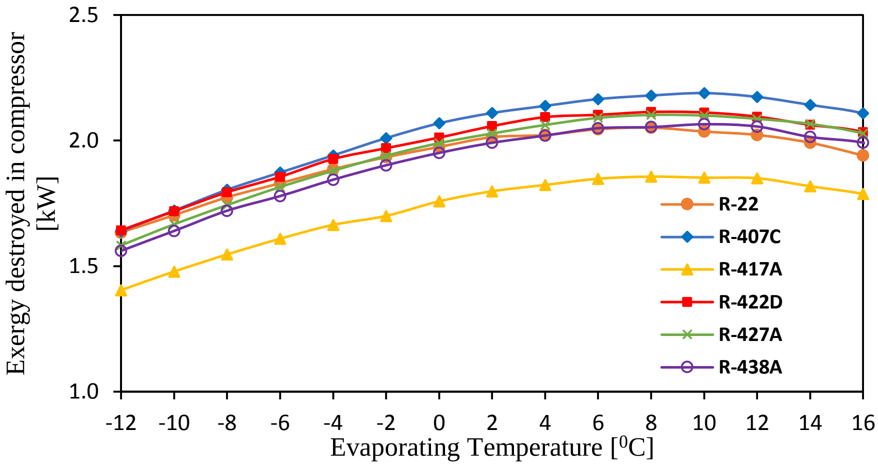

Figure 4.34 Effect of evaporating temperature on exergy destroyed in compressor ($T_k = 40°C$).

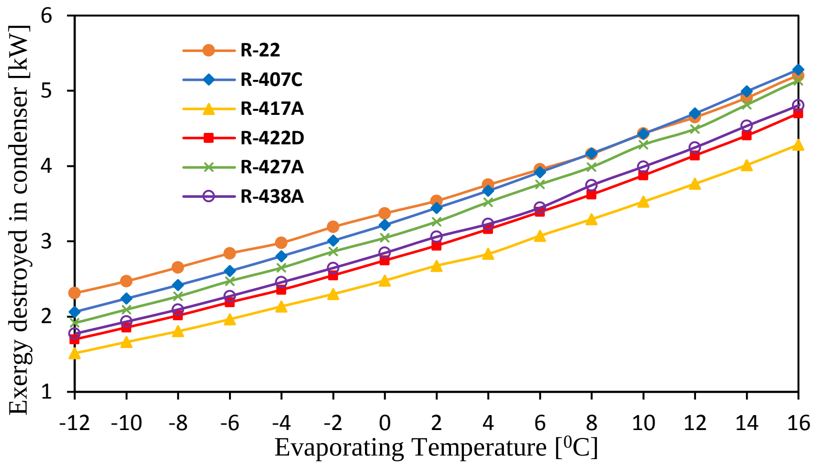

Figure 4.35 Effect of evaporating temperature on exergy destroyed in condenser ($T_k = 40$℃).

The exergy destroyed in condenser is shown in figure 4.35. It is noted that R22 has greater exergy destruction than alternate refrigerants. The exergy destruction of any of the considered refrigerant increases with increasing evaporating temperature. In the alternates the highest and lowest exergy destruction in condenser is of R407C and R417A. The factors which cause the lower exergy destruction in alternate refrigerants is their low heat rejection and low entropy produced due to low discharge temperatures.

The exergy destruction in the expansion valve is presented in figure 4.36. It is revealed that all alternate refrigerants have higher exergy destruction in the expansion valve than R22. As the expansion process is isenthalpic, the exergy destruction is mainly due to pressure drop across the valve. The exergy destruction in expansion valve decreases with increasing evaporating temperature. In the selected substitutes, R422D and R417A gives the highest and lowest exergy destruction in expansion valve.

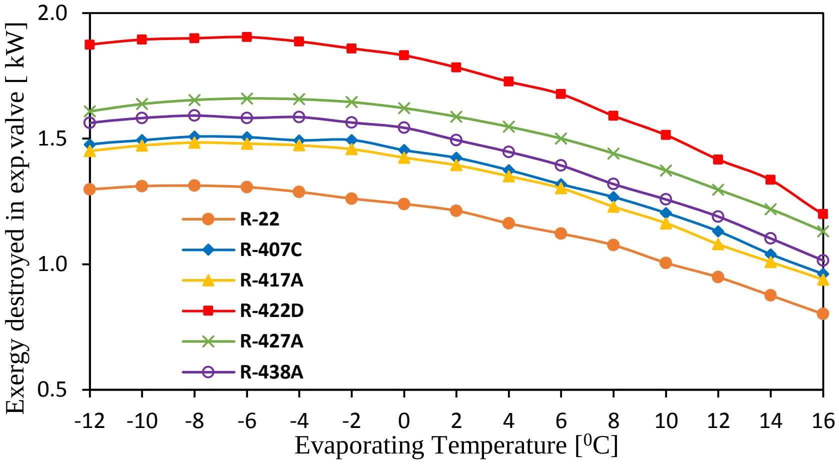

Figure 4.36 Effect of evaporating temperature on exergy destroyed in expansion valve (T_k = 40°C).

Referring figure 4.37 the exergy destroyed in evaporator increases with increase in evaporating temperature. This is because at high evaporating temperatures the cooling capacity increases while the temperature gradient between evaporating, space and dead state temperature decreases, thus reducing the heat transfer. It is considered that the space temperature is 15^{0}C above than corresponding evaporating temperature. R407C has highest exergy destroyed in evaporator followed by R22. R417A has lowest exergy destroyed in the evaporator.

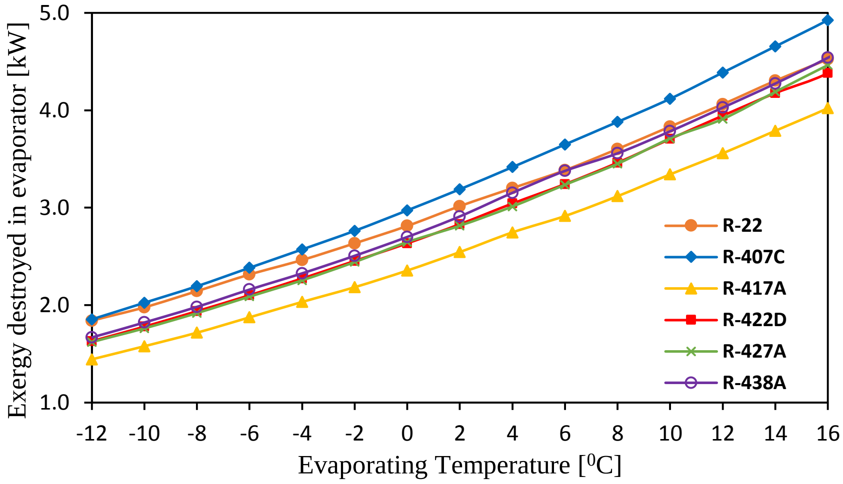

Figure 4.37 Effect of evaporating temperature on exergy destroyed in evaporator (T_k = 40°C).

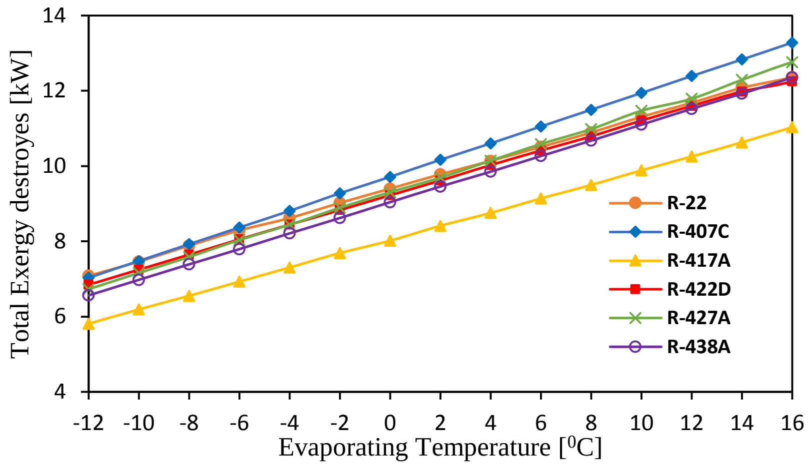

Figure 4.38 Effect of evaporating temperature on total exergy destroyed ($T_k = 40°C$).

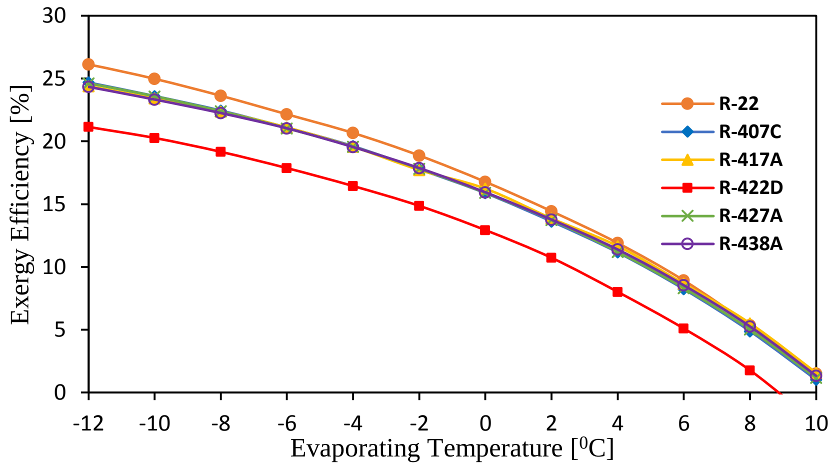

Figure 4.39 Effect of evaporating temperature on exergy efficiency ($T_k = 40°C$).

It should be noted that for a constant cooling capacity the exergy destroyed in compressor, condenser and evaporator decreases with increase in evaporating temperature however this is equivalent to our case of varying capacity as compressor power (exergy supplied) changes accordingly, giving same exergy efficiency for both scenarios.

The total exergy destroyed and the exergy efficiency are presented in figure 4.38. R407C has highest system exergy destruction while the exergy destruction of R22, R422D and R427A are close to each other. As a direct consequence of low destruction in components the lowest system exergy destruction is of R417A.

The total exergy destruction may be misleading from comparison point of view because information of the exergy supplied (compressor electrical power) is necessary whose fraction has been destroyed. The parameter, exergy efficiency provides this information and is plotted for each refrigerant in figure 4.39. As seen R22 has better exergy efficiency than others. R407C, R417A, R427A and R438A has similar exergy efficiencies. The worst exergy efficiency is of R422. The exergy destruction occurred in components in the evaporating temperature range of -12^{0}C to 16^{0}C at 40^{0}C condensing temperature is summarised in table 4.16.

Table 4.16 Exergy destroyed (kW) and exergy efficiency over the evaporating range ($T_k = 40°\text{C}, T_0 = 25°\text{C}$).

$\dot{X}_d$	R22	R407C	R417A	R422D	R427A	R438A
$\dot{X}_{d,c}$	1.6 – 1.9	1.6 – 2.1	1.4 – 1.8	1.6 – 2.1	1.6 – 2.1	1.6 – 2.1
$\dot{X}_{d,k}$	2.3 – 5.1	2.1 – 5.3	1.5 – 4.3	1.7 – 4.6	2 – 5.1	1.8 – 4.8
$\dot{X}_{d,ex}$	0.8 – 1.3	1 – 1.5	1 – 1.5	1.3 – 2	1.1 – 1.6	1 – 1.6
$\dot{X}_{d.e}$	1.4 – 4.5	1.8 – 4.1	1.4 – 4	1.6 – 4.4	1.6 – 4.5	1.7 – 4.5
$\dot{X}_{d,tot}$	7 – 12.5	7 – 13.3	5.8 – 11	6.8 – 12.2	6.7 – 12.7	6.5 – 12.4
η_{ex}	0 – 26	0 – 24	0 – 24	0 – 20	0 – 24	0 – 24

Similar to evaporating temperature the effect of condensing temperature on the exergy performance of components is investigated. The condensing temperature is varied from 30^{0}C to 50^{0}C against a fixed evaporating of 4^{0}C. The results obtained are shown in figures below.

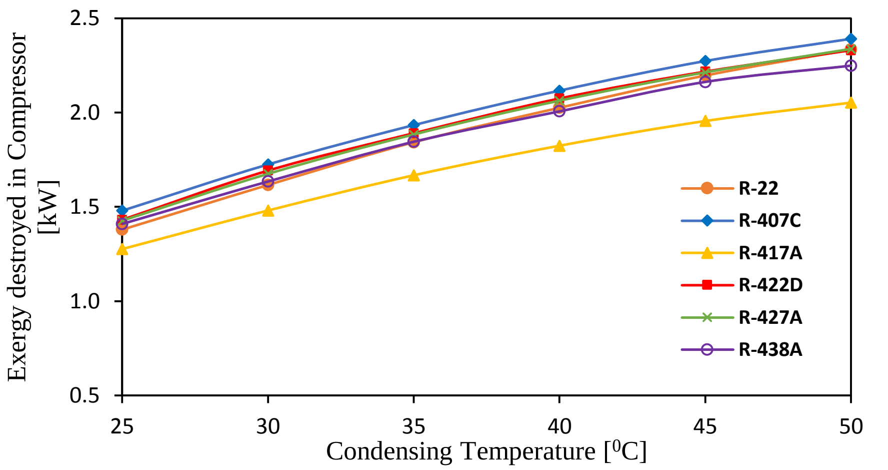

Figure 4.40 Effect of condensing temperature on exergy destroyed in compressor ($T_e = 4°\text{C}, T_0 = 25°\text{C}$).

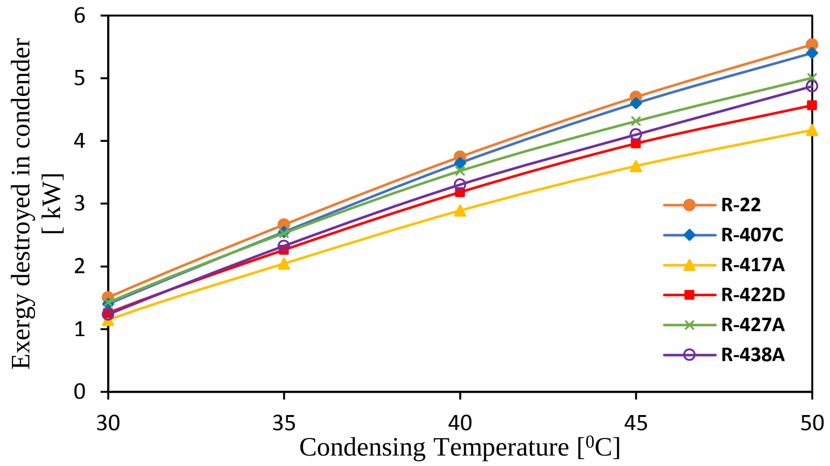

Figure 4.41 Effect of condensing temperature on exergy destroyed in condenser (T_e = 4℃ , T_0 = 25℃).

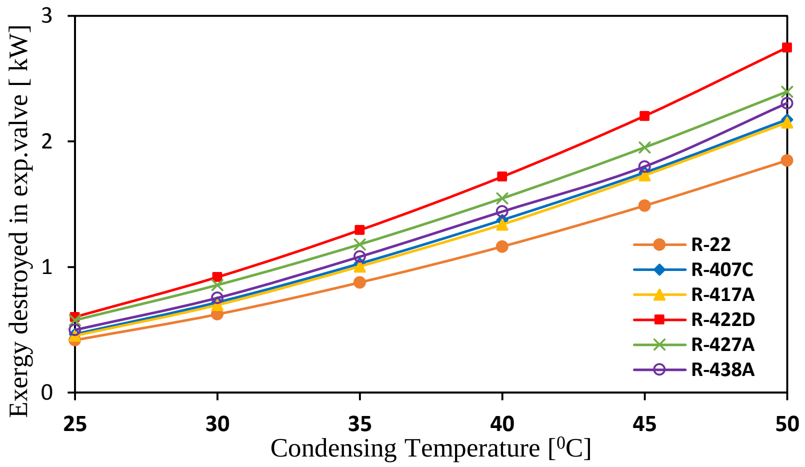

Figure 4.42 Effect condensing temperature on exergy destroyed in expansion valve (T_k = 40℃ , T_0 = 25℃).

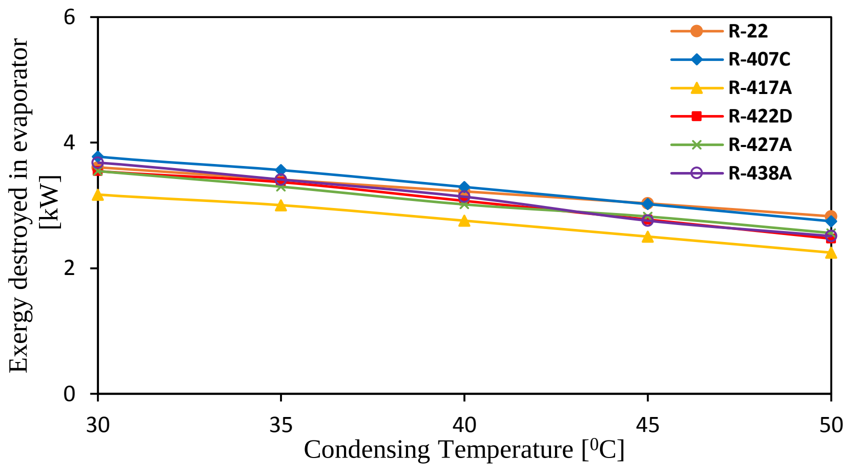

Figure 4.43 Effect of condensing temperature on exergy destroyed in evaporator (T_e = 4℃ , T_0 = 25℃).

Referring to figure 4.40 the exergy destroyed in compressor increases with increase in condenser temperature because high pressure ratio and temperature increase the entropy generation, resulting in greater exergy destruction. At any condensing temperature R407C and R417A has the highest and lowest compressor exergy destruction. R422D, R427A and R438A has similar compressor exergy destruction as R22. As evident from figure 4.41 the exergy destruction in condenser increasers with increasing condensing temperature due to increasing temperature differences and phase change process which rises entropy. Alternate refrigerants have recorded lower exergy destruction in condenser relative to R22.

As can be seen in figure 4.42 the exergy loss in expansion valve increases with increasing condensing temperature. At any condensing temperature the valve exergy destruction in alternate refrigerants is higher than R22. Investigation on the effect of condensing temperature on evaporator exergy destruction revealed that exergy losses in evaporator are reduced when condensing temperature rises as shown in figure 4.43.

It should be noted that the compressor, condenser and evaporator exergy losses decreases when cooling capacity is kept constant. In our case the cooling capacity varies due to mass flow and the respective refrigerating effect produced at any evaporating temperature. However both the methods are equivalent as compressor power (exergy supplied) changes accordingly giving same efficiency.

The total exergy destroyed and exergy efficiency are shown in figure 4.44 and figure 4.45.The total exergy destroyed increases while exergy efficiency decreases with increasing condensing temperature.R417A, R422D, R427A and R438A have lower system exergy destroyed than R22 but none of the refrigerants has better exergy efficiency than R22. This is because R22 has superior quality of product (cooling capacity) with given fuel (compressor electrical power).

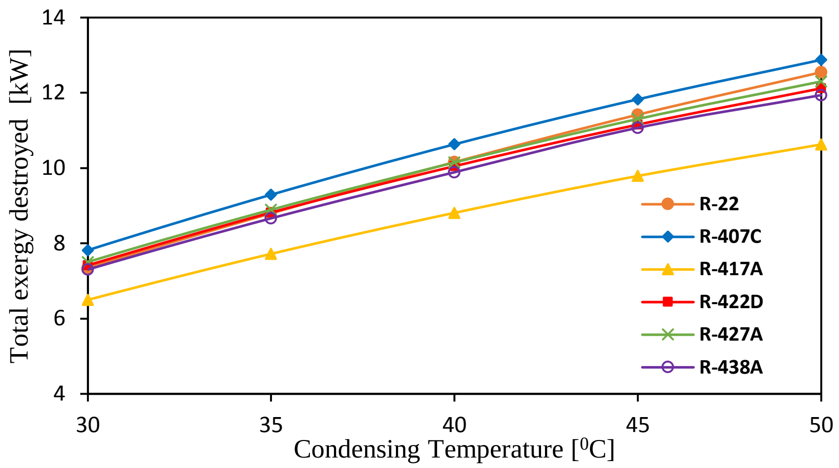

Figure 4.44 Effect of condensing temperature on total exergy destroyed ($T_e = 4$°C , $T_0 = 25$°C).

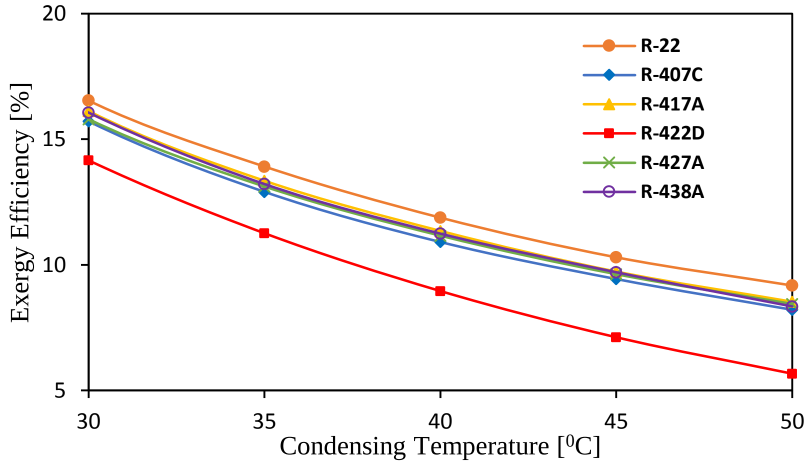

Figure 4.45 Effect of condensing temperature on exergy efficiency ($T_e = 4$°C, $T_0 = 25$°C).

For a respective cooling capacity, evaporation and space temperature, the exergy efficiency may not remain the same as it is also affected by dead state temperature which do not remain constant due to seasonal variations throughout the year. The effect of dead state temperature on exergy efficiency is presented in figure 4.46 for 4^0C evaporating and 40^0C condensing temperature. It is seen that exergy efficiency increases with increase in dead state temperature. At any dead state temperature R22 and R422D has highest and lowest exergy efficiency.

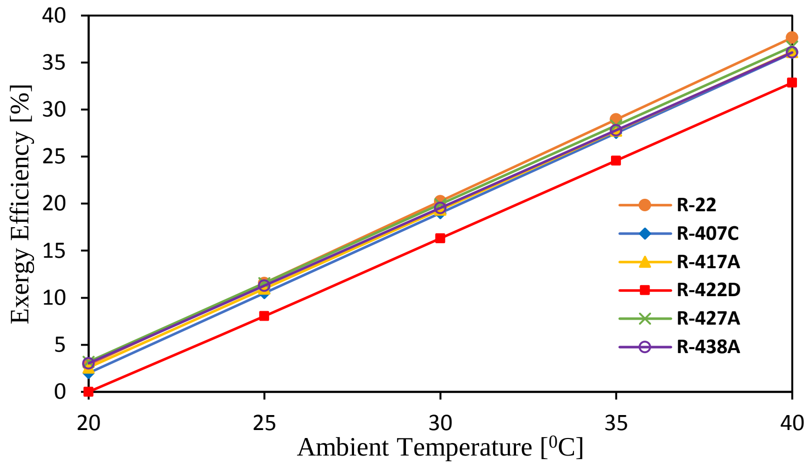

Figure 4.46 Effect of ambient temperature on exergy efficiency ($T_e = 4$℃ , $T_k = 40$℃).

From the above analysis we conclude that in terms of exergy destruction, from operational point of view, for the considered alternate refrigerants except for R417A, the compressor and evaporator has similar criticality as for R22. Due to lower losses, the condenser exergy performance of all the alternate refrigerants is better than R22. Expansion valve emerged as the most critical component when using alternate refrigerants because of their higher losses relative to R22. We see for any refrigerant the major contribution in total exergy destruction is of condenser and evaporator. Among the considered refrigerants, all have lower exergy efficiency than R22. The worst exergy efficiency is of R422D while the exergy efficiency of other candidates are close to each other. For any refrigerant the largest exergy destroyed is in the condenser and evaporator which contribute to about 65% of the total exergy destroyed. Compressor and expansion valve have comparatively less exergy destroyed. The figures below represents the average exergy destroyed in components.

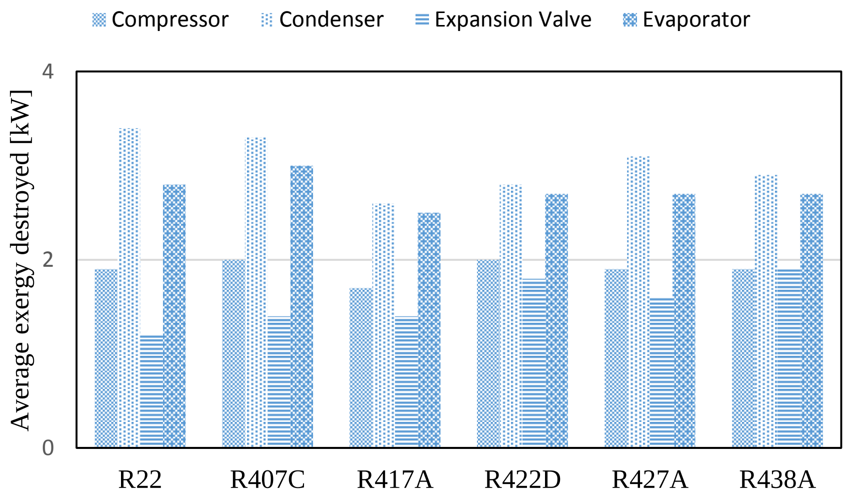

Figure 4.47 Average exergy destroyed in each component.

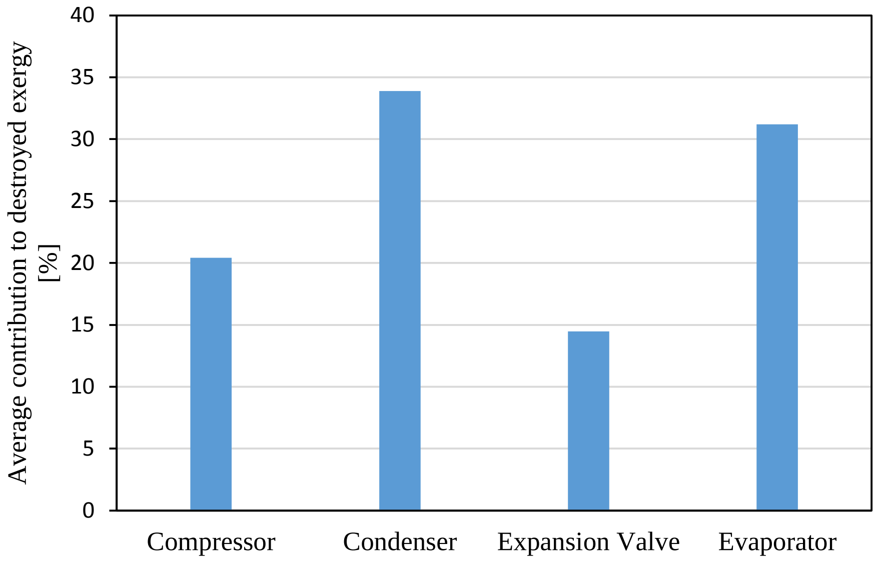

Figure 4.48 Average contribution in exergy destruction.

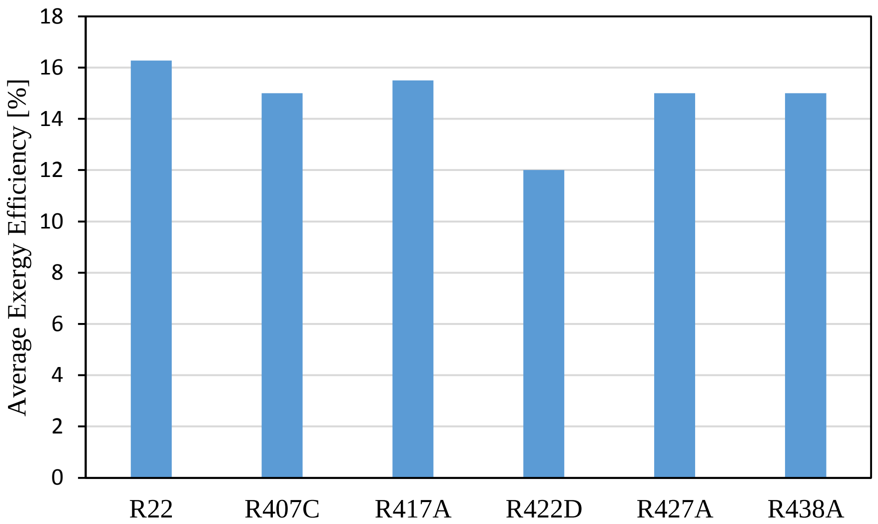

Figure 4.49 Average exergy efficiency of refrigerants.

Previous work of researchers on performance of alternate refrigerants relative to R22, which may also include our considered refrigerants, claims that the exergy destruction in R22 is less than any of the alternate refrigerant. This is particularly true when the system is thermodynamically accessed from the view point that R22 and alternate refrigerants will provide same cooling capacity, without taking into consideration the inherent constraints of the system. We also independently verified that if constant capacity is considered then the exergy destroyed of R22 is less. However our case is based on retrofitting the existing system in which candidate refrigerants are to perform under similar constraints as R22 and as a result cooling capacity changes. But both the methods gives same exergy efficiency, a parameter of prime importance. R407C, R417A, R427A and R438A are equivalent in terms of exergy performance indicating a reduction of about 7.8% lower relative to R22. We noted that R422D in terms of exergy performance R422D emerged as worst candidate having exergy efficiency which is 26.2% lower than R22.

4.2.3 Other Considerations

The suction and discharge pressure of substitutes are slightly lower and higher than R22. So slight adjustment in existing pressure settings will be needed after retrofitting the system. This also means that the existing pressure and temperature switches/controls can be used with retrofitted system. The sealing material adaptability with new refrigerant is important to be considered to prevent leakage as seal may swell or shrink.

R407C and R427A are not compatible with mineral oil which is used with R22. There is a problem in oil return when these are used because they are not miscible with mineral oil. 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. Moreover retrofitting becomes time consuming when the mineral oil is entirely to be flushed out. On the other hand R438A is compatible with both mineral oil and polyolester oil which gives it an added advantage.

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. The focussed parameters were compressor discharge temperature, pressure ratio, cooling capacity, compressor power, coefficient of performance, heat rejection, volumetric cooling capacity, exergy destroyed in components and exergy efficiency. For energy analysis method based on constant volume flow and volumetric efficiency is used. For the considered refrigerants and evaporating temperature range it is observed that constant volume flow may result an error of 2.1% to 7.7% in mass flow, a controlling factor of other parameters. However both the methods are regarded as appropriate when pressure ratios are not very high. The results obtained for each of the candidate refrigerant were compared against R22 in order to arrive at most suitable alternate.

It is concluded that the system retrofitted with any of the selected refrigerants will have low energy performance than the system originally designed for R22. The COP values suggests that R407C, R417A, R427A and R438A are good potential candidates whose respective COP are 2.7% to 3.8%, 4.8% to 5.9%, 4.8% to 5.9%, 4.8% to 5.9% are lower than R22 but low cooling capacity of R417A which is about 15.8% to 22.6% lower than R22 makes it least desirable option. R422D has comparatively lower COP than others and has high mass flow, being about 28% to 35% higher than R22 which may not be acceptable for existing piping and auxiliaries. Both these facts make R422D as not a suitable choice. On the basis of energy performance R407C, R427A and R438A are most appropriate substitutes for retrofitting.

It is also concluded that the system retrofitted with any of the selected refrigerants will have low exergy performance than the system originally designed for R22. From exergy point of view, for alternate refrigerants expansion valve emerged as the worst component. The exergy efficiency of all the selected refrigerants is lower than R22. R407C, R417A, R427A and R438A have similar exergy efficiencies being 7.5% lower while the lowest exergy efficiency is recorded for R422D which is about 26% lower than R22. From both energy and exergy point of view R407C, R427A and R438A are satisfactory options to substitute R22.

Retrofitting with alternate refrigerants results in lower compressor discharge temperatures than R22, thus enhancing the overall reliability of compressor. The values of electrical power, heat rejected, pressure ratio and VCC indicated that no major modifications in system components are required 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. R438A has an advantage over R407C and R427A because it is compatible both with mineral oil and polyolester oil. In current system if R438A is used for retrofitting and the system is running on 4^0C evaporating and 40^0C condensing temperature it is expected the cooling capacity will be reduced from 58.2kW to 53kW. Compressor power will be reduced from 11.5kW to 11kW. COP will be change from 5.1 to 4.8. Discharge temperature will be 56^0C instead of 69^0C. Mass flow will be increased from 0.36kg/sec to 0.4kg/sec. Heat rejection will be reduced from 69.8kW to 64.6kW. Pressure drop across expansion valve will be increased from 967kPa to 1013kPa. Exergy efficiency will drop from 12% to 11.3% and lubricating oil do not need to be changed.

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