# THERMOECONOMIC ANALYSIS OF REFRIGERATION SYSTEMS WITH ALTERNATE REFRIGERANTS

**THESIS**

***Submitted in fulfillment of the requirement of the degree of***

## DOCTOR OF PHILOSOPHY

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***J.C. BOSE UNIVERSITY OF SCIENCE & TECHNOLOGY, YMCA***

***by***

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**FEBRUARY, 2019**

# DEDICATION

**Dedicated**

**to**

**My Parents, Wife and Son Ayaan**

# CANDIDATE'S DECLARATION

I hereby declare that this thesis entitled **THERMOECONOMIC ANALYSIS OF REFRIGERATION SYSTEMS WITH ALTERNATE REFRIGERANTS** *by* **GAURAV** being submitted in fulfillment of the requirements for the Degree of Doctor of Philosophy in DEPARTMENT OF MECHANICAL ENGINEERING under Faculty of Engineering & Technology of J.C. Bose University of Science & Technology, YMCA, Faridabad during the academic year 2018-19, is a bonafide record of my original work carried out under guidance and supervision of **DR. RAJ KUMAR, PROFESSOR, Department of Mechanical Engineering** and has not been presented elsewhere.

I further declare that the thesis does not contain any part of any work which has been submitted for the award of any degree either in this university or in any other university.

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# CERTIFICATE

This is to certify that this Thesis entitled **THERMOECONOMIC ANALYSIS OF REFRIGERATION SYSTEMS WITH ALTERNATE REFRIGERANTS** by **GAURAV** submitted in fulfillment of the requirement for the Degree of Doctor of Philosophy in **DEPARTMENT OF MECHANICAL ENGINEERING** under Faculty of **Engineering & Technology** of J.C. Bose University of Science & Technology, YMCA Faridabad, during the academic year **2018-19**, is a bonafide record of work carried out under my guidance and supervision.

I further declare that to the best of my knowledge, the thesis does not contain any part of any work which has been submitted for the award of any degree either in this university or in any other university.

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Dated:

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# ABSTRACT

Global warming is one of the challenges facing our world today. Earth's temperature is increasing every year by some degree because of this. Leakage of refrigerants from refrigerators & air conditioners into the atmosphere is one of the factors contributing to global warming and the reason behind the increase in global warming is both direct (i.e. leakage of refrigerant into the atmosphere) and indirect (i.e. through energy consumption). Refrigerants such as HFCs and HCFCs are contributing towards the increase in global warming. The protocol linked with the banning of refrigerants which contribute to global warming is Kyoto Protocol. Recently, Kigali Amendment to the Montreal Protocol provides a great opportunity to further reduce both direct and indirect refrigerant emissions from refrigeration and air conditioning sector. Scientists found that the world can avoid up to 0.4ºC of global warming this century through implementation of the Kigali Amendment. India is also bound by the protocol and its amendment but the country gets more time to get rid of such refrigerants as compared to window available to developed countries.

HFCs refrigerants like R134a are contributing towards the increase in global warming because of the presence of fluorine atoms in their structure. Currently used refrigerants in place of HFCs are HCs due to the absence of fluorine atoms and performance is comparable with HFCs. But, HCs find limited application in refrigeration and air conditioning industry because of flammability issues related to them. Recently, proposed refrigerants by researcher to overcome the issues related with HFCs and HCs are HFOs. HFOs contains a double bond in their structure and perform exceptionally well as compares to HFCs and HCs. HFOs come under the category of mild flammability (i.e. A2L) and presently these are very costly. Thus, further research move towards the mixture of HFC/HFO.

R134a is one of the main refrigerants used in a large number of applications of refrigeration and air conditioning. So, there is a requirement for finding a suitable drop-in substitute for it from an environment point of view. The present thesis worked on this problem. A total of thirty refrigerants are selected as a replacement for R134a form literature review. Exergy and thermoeconomic analysis are performed on a domestic refrigerator working on a vapour compression system. The various

parameters computed are pressure ratio, mass flow rate, relative volumetric cooling capacity, relative coefficient of performance, cooling capacity, exergetic efficiency, exergy destruction, efficiency defect, cost of operating, cost of exergy destruction, levelized electricity cost, energy efficiency ratio, exergoeconomic factor and cost importance. A computational model is made for exergy and thermoeconomic analysis of the system into Engineering Equation solver and the properties of refrigerants have been calculated using REFPROP. The results are validated experimentally.

After conducting all evaluations, it has been observed that among all tested refrigerants, R600a has the lowest value of cost of operation, cost of exergy destruction, levelized electricity cost, and energy efficiency ratio whereas R134a/R1234yf (10%/90%) has the largest value except for energy efficiency ratio. Refrigerant having a mixture of 40%R134a/22%R1234yf/38%R1234ze has the relative volumetric cooling capacity and other parameters nearly similar to that of R134a. The value of cost importance and exergy destruction is highest for compressor and lowest for expansion valve. The value of exergoeconomic factor for compressor & electric motor is within the prescribed limit but for condenser and evaporator, it is lower than the prescribed limit. Compressor has the largest value of cost importance & exergy destruction and whereas expansion valve has lowest value of cost importance & exergy destruction. Reduced quantity of R1234yf refrigerant with changes in lower compression ratio, with efficient condenser and evaporator, will maintain earlier COP and refrigerating effect. It will also reduce flammability as the temperature and pressure encountered in the system is lesser. Optimization of capillary tube parameters provided improvement up to 5.5% in COP of HFO/HFC mixture. Based upon thermodynamic and thermoeconomic analysis, refrigerant mixture having composition 40%R134a/22%R1234yf/38%R1234ze with GWP around 600 is found to be the best drop-in replacement for R134a among all selected refrigerant.

# TABLE OF CONTENTS

**CHAPTER 2**



**CHAPTER 3**

# LIST OF TABLES

# LIST OF FIGURES

# LIST OF ABBREVIATIONS

| | |
|---|---|
| A | Annuity |
| c | Average unit cost ($/kJ) |
| C | Cost |
| CCR | Compressor Clearance Ratio |
| CFC | Chlorofluorocarbon |
| CFCs | Chlorofluorocarbons |
| $CO_2$ | Carbon dioxide |
| COP | Coefficient of Performance |
| $\dot{C}$ | Cost rate ($/s) |
| CRF | Capital Recovery Factor |
| DNA | Deoxyribonucleic Acid |
| EDR | Exergy Destruction Ratio |
| EER | Energy Efficiency Ratio |
| EES | Engineering Equation Solver |
| EXCEM | Exergy, Cost, Energy and Mass |
| $\dot{E}$ | Exergy (kW) |
| f | Exergoeconomic factor |
| F | Future value |
| F-P-L | Fuel Product Loss |
| ft | Feet |
| g | Gram |
| GWP | Global Warming Potential |
| h | Specific enthalpy (kJ/kg) |
| HC | Hydrocarbon |
| HCFC | Hydrochlorofluorocarbon |
| HCFCs | Hydrochlorofluorocarbons |
| HCs | Hydrocarbons |
| HFC | Hydrofluorocarbon |
| HFCs | Hydrofluorocarbons |
| HFO | Hydrofuoroolefin |

| | |
|---|---|
| HFOs | Hydrofuoroolefins |
| HOC | Heat of combustion |
| i | Interest rate |
| $i_{eff}$ | Effective cost or value of money |
| $\dot{j}$ | Amortization factor |
| kJ | Kilojoule |
| kW | Kilowatt |
| LEL | Lower flammability limit |
| LPG | Liquefied Petroleum Gas |
| m | Number of compounding periods in one year |
| MAC | Mobile Air Conditioning |
| $\dot{m}$ | Mass Flow Rate (kg/s) |
| MATLAB | Matrix Laboratory |
| N | Compressor speed in rpm |
| n | System life |
| n.d. | Not defined |
| N_r | Compressor speed in rps |
| $NH_3$ | Ammonia |
| NIST | National Institute of Standards and Technology |
| ODP | Ozone Depletion Potential |
| OEL | Occupational exposure limit |
| *P* | Present value |
| P | Pressure (bar) |
| $P_o$ | Constant value at the beginning of first year |
| POE | Polyolester |
| ppm | Parts per million |
| PR | Pressure ratio |
| Q | Cooling Capacity (kW) |
| $\dot{Q}$ | Heat flow rate (kW) |
| r | Relative cost difference |
| REFPROP | REFerence fluid PROPerties |
| refrigerant$ | Enter the name of refrigerant under consideration |
| $r_{inf}$ | Inflation rate |

| | |
|---|---|
| $r_n$ | Nominal or apparent escalation rate |
| rpm | Revolution per minute |
| rps | Revolution per second |
| $r_r$ | Real escalation rate |
| s | Specific entropy (kJ/kgk) |
| t | Operating time (s/yr) |
| T | Temperature (ºC) |
| TEWI | Total Equivalent Warming Index |
| $T_r$ | Space temperature (ºC) |
| USA | United States of America |
| v | specific volume ($m^3$/kg) |
| W | Power input (kW) |
| $\dot{W}$ | Rate of doing work or Power (kW) |
| $\dot{Z}$ | Capital investment or Sum of capital and operation & maintenance cost ($/s) |
| ZC | Cost importance |

Greek Symbols

| | |
|---|---|
| $\eta$ | Efficiency |
| $\delta$ | Efficiency defect |

Superscripts

| | |
|---|---|
| Ch | Chemical |
| Ph | Physical |

Subscripts

| | |
|---|---|
| 0 | Ambient condition or Restricted dead state |
| any ref | any refrigerant |
| cl | clearance |
| comp | compressor |
| cond | condenser |
| D | Destruction |
| elec | electric |

| | |
|---|---|
| electric | Electricity |
| evap | evaporator |
| ex | Exergetic |
| exp | expansion valve |
| F | Fuel |
| in | Entering a component or system |
| is | isentropic |
| k | $k^{th}$ component |
| L | Levelized |
| L | Loss |
| lvhx | liquid vapour heat exchanger |
| out | Leaving a component or system |
| P | Product |
| r | refrigerant |
| ref | Reference value |
| rel | relative |
| rr | reversible refrigerator |
| sat | saturated |
| st | stroke |
| tot | total |
| vol | volumetric |
| W1 | Associated with motor |
| W2 | Associated with compressor |

# CHAPTER 1
# INTRODUCTION

## 1.1 BACKGROUND

Refrigeration is defined as a process in which heat is removed from a reservoir at low temperature and then transferred to a reservoir at high temperature. In refrigeration, cooling of fluids or bodies takes place that results in the lower temperature as compared to surrounding at a particular time and space. The application of refrigeration includes food processing and preservation, comfort air conditioning, industrial air conditioning, automobile air conditioning, medical treatment and preservation, cold treatment of metals etc.

The refrigeration can be achieved by a natural or artificial process. The natural refrigeration processes are nocturnal cooling (cooling of a building roof by night sky cooling), evaporative cooling, cooling by salt solution, use of ice transported from cooler regions, use of ice harvested in winter and stored in ice houses. The demerits associated with the natural refrigeration are its dependency on local conditions and uncertain nature. The credit for the earlier attempt on artificial refrigeration goes to Scottish Professor William Cullen (1755) who successfully produced small quantities of ice by evaporation of ether under vacuum. The artificial refrigeration processes are vapour compression refrigeration system, vapour absorption refrigeration system, solar energy based refrigeration system, air cycle refrigeration system, steam vapour jet refrigeration system, vortex tube and thermoelectric refrigeration system. Among all artificial refrigeration systems, the most famous and commonly used refrigeration system in the refrigerator is a vapour compression system. Today, refrigeration industry is facing major challenges like lower energy efficiency and environmental & safety concerns. In order to meet these challenges, it is necessary to replace the commonly used HFC with potential refrigerant explore through both simulation and experimentation.

## 1.2 VAPOUR COMPRESSION REFRIGERATION SYSTEM

The concept of vapour compression refrigeration system was given by American Engineer Oliver Evans in 1805. He developed a closed cycle for producing refrigeration in a continuous manner. The credit for building the actual vapour compression system goes to Jakob Perkins (1835) for successfully patenting the system using ethyl ether as a refrigerant. Even though this system worked very well but not achieved commercial success because of toxic nature of ethyl ether. In 1850, Alexander Twining received a British patent for a vapour compression system by use of ether, $NH_3$ and $CO_2$. The credit for making practical vapour compression refrigeration system goes to James Harrison. He used various refrigerants such as ether, alcohol & ammonia and also received a patent for his work in 1856. Eastman Kodak first installed the air conditioning for storage of photographic film and domestic air conditioning system in 1894. The first mechanical domestic refrigerator was introduced by General Electric, USA in 1911.

In vapour compression cycle, refrigerant enters as a vapour at low pressure in compressor and leaves as a superheated vapour at high temperature and pressure. The vapour in superheated form then enters the coils or tubes of the condenser where it is cooled by the fluid surrounding the coils. The cooling of refrigerant in condenser can be natural or forced cooling. The refrigerant exits the condenser as a high pressure liquid having a temperature slightly more than that of the surrounding. This liquid refrigerant having high pressure enters a reducing area via an expansion device where it becomes a low pressure and low temperature liquid vapour mixture. In expansion device nearly half portion of the liquid refrigerant undergoes explosive-like flash evaporation due to the sudden decrease in pressure and latent heat required for this evaporation is obtained from still liquid refrigerant in the nearby area. This process is known as auto-refrigeration. The low pressure partially vaporizes cold refrigerant from expansion valve flows through the evaporator coils and takes up the heat from air surrounding of the evaporator coil (space to be cooled). The air surrounding the coil is usually blown by a fan. The fully vaporized refrigerant exit from the evaporator and again, go inside the compressor and refrigeration cycle goes on. Figure 1.1 depicts a simple vapour compression system [58].

## 1.3 REFRIGERANTS

Refrigerant is a substance that acts as a cooling medium by extracting heat from a reservoir at low temperature and passes it to a reservoir at high temperature. The selection of suitable refrigerant is very important for the design of any refrigeration system. We can also say that it is the first step for the design of refrigeration systems. Even though the theoretical COP of the refrigeration systems mainly determined by operating temperature but the selection of suitable refrigerant decides many important parameters like initial & operating costs, system design aspects, size & weight of the system, serviceability, reliability and safety issues.

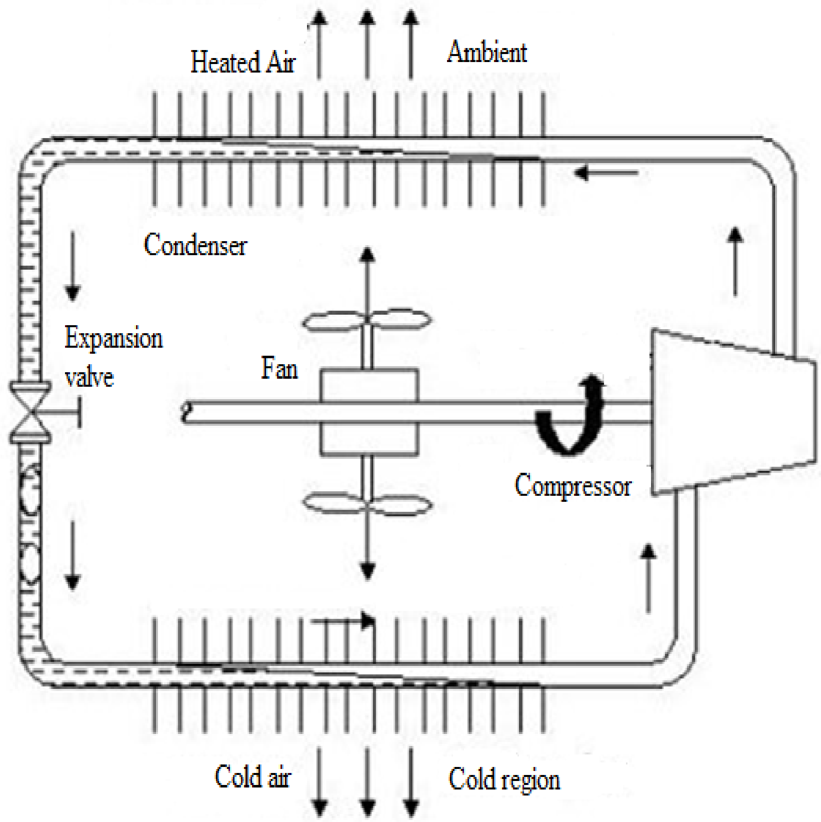



Figure 1.1 Vapour compression refrigeration system

### 1.3.1 History of Refrigerants

The first refrigerant used was ethyl ether by Jakob Perkins and others in an early refrigeration machine. Ethyl ether as a refrigerant was selected because it exists as a liquid at ambient condition and was easier to handle. But several problems were also associated with the use of ethyl ether like to operate the system at a lower temperature, it was required to maintain the vacuum for the evaporation and its toxic nature. In order to overcome the shortcomings of ethyl ether, the requirement of new refrigerants felt. Some of the refrigerants found and used in the refrigeration system are shown in Table 1.1. Hydrocarbons (HCs) were also tried earlier as a refrigerant but not much used because of flammability issue. Due to the problem associated with earlier refrigerants, the refrigerant industry felt that in order to make progress it was required to develop refrigerants that were safe, non-toxic, non-flammable etc. and this led to the invention of chlorofluorocarbons (CFCs). CFCs entered the market in the

1930s under the trade name of Freons. Freon-12 was the first commercial CFC refrigerant in 1931.

Table 1.1 Refrigerants used during 1835-1885

| Sr. No. | Refrigerant | Year | Used by | Issues |
|---|---|---|---|---|
| 1 | Ethyl ether | 1835 | Jakob Perkins | Toxicity |
| 2 | Dimethyl ether | 1864 | Charles Tellier | Toxicity |
| 3 | Sulphur dioxide | 1874 | Raoul Pictet | Chemical stability |
| 4 | Ammonia | 1877 | Carl von Linde | Toxicity, Material compatibility |
| 5 | Carbon dioxide | 1885 | Windhausen | Operating pressure |

Freon-11 came out from the study of Thomas Midgley Jr. in 1932. The rapid growth of Freon group in the refrigeration industry existed due to the safe nature of CFCs. Among all earlier refrigerants, only ammonia survived the Freon magic. Hydrochlorofluorocarbons (HCFCs) like HCFC-22 were also used in the refrigeration industry along with CFC. Theory of ozone layer depletion was proposed by Rowland and Molina in 1974. They found that CFCs contribute to ozone layer depletion due to the release of chlorine and bromine atoms from CFCs into the atmosphere. Montreal Protocol started banning the substances which are responsible for ozone depletion such as CFCs in 1987. CFCs were also found as one of the major factors in contributing to global warming. So, there is a need to replace CFCs and their alternative refrigerant should have low GWP with zero ODP. Hydrofluorocarbons (HFCs) were introduced to replace CFCs and HCFCs. Some of earlier refrigerants like carbon dioxide and HCs also make a comeback. The main problem arises with the use of HFCs is global warming due to the presence of fluorine atoms in it. As per the Kyoto agreement in 1997, it is mandatory for all the developed countries to control the release of gases which produce global warming and almost 100 developing countries, including India, were exempted from it. During Doha conference in December 2012, Kyoto Protocol was put forth to 2020. Under Kyoto Protocol, it was decided to phase out the refrigerants like HFCs and HCFCs which contribute to global warming. The alternatives of HFCs and HCFCs currently in use are HCs and refrigerant mixtures. Hydrocarbons as refrigerant found limited application in refrigeration because of flammability issue. Currently, proposed alternatives of HFCs are HFOs and their mixtures. In order to designate all the refrigerants symbol R followed by a unique number is used. The developments of refrigerants with their demerits are shown in Figure 1.2.

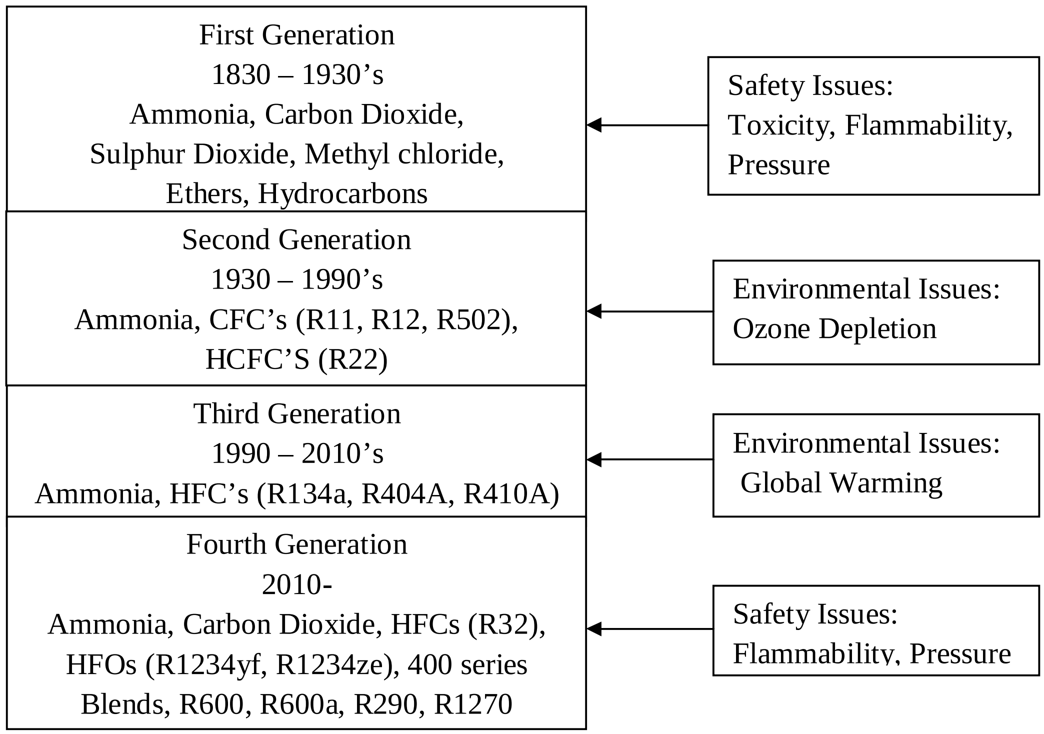



Figure 1.2 Development of refrigerants

**1.3.2 Properties of an Ideal Refrigerant**

It is very difficult to find out a refrigerant which has all the properties of ideal refrigerant but still researchers are trying to find a refrigerant with properties similar to an ideal refrigerant. The following are the properties which an ideal refrigerant must possess:

**1.3.2.1 Physical Properties**

It includes thermodynamic as well as thermophysical properties like low freezing point, low condensing pressure, high evaporator pressure, high critical pressure, high vapour density/low specific volume, high dielectric strength, high latent heat of vapourisation, high heat transfer coefficient, high thermal conductivity, low viscosity, low isentropic index of compression, miscibility with lubricating oil, low water solubility, non reactive with materials used in refrigeration and chemical stability.

**1.3.2.2 Environmental Properties**

It includes eco-friendly environmental properties like the ease of leak detection, non-flammable, value of ozone depletion potential must be zero, global warming potential value as low as possible, low total equivalent warming index (TEWI) and non toxic.

#### 1.3.2.3 Economic properties

Refrigerant must be inexpensive and easily available.

### 1.3.3 Ozone Depletion Potential

It is defined as the relative amount of ozone layer degradation caused by a refrigerant with respect to standard refrigerant like R11 of the same mass. The ODP value of R11 is being fixed at 1 because of the presence of three chlorine atoms. Brominated substances have a range of ODP usually between 5-15, for example, R13B1 with a value 10. The presence of chlorine and bromine atoms in refrigerants is the reason behind ozone depletion. HFCs, HCs and HFOs have zero value of ODP. The ozone layer depletion causes skin cancer, cataracts, weakened immune system and damage DNA structure in human beings and have an adverse effect on crop yield and aquatic ecosystems. According to Montreal Protocol, refrigerants having non zero ODP will be phased like R22 or already being phased out like R12.

### 1.3.4 Global Warming Potential

It is termed as the measure of the contribution of a refrigerant to global warming with respect to carbon dioxide having the same mass. The GWP value of carbon dioxide is being fixed at 1. CFCs have a very high value of GWP followed by HFC. HCs and HFOs have a value of GWP less than that of 10. From the environmental point of view as well as protocol given in Kyoto, a refrigerant must have GWP value as small as possible. The impact of global warming will be as follows:

- Rise in temperature of earth’s surface
- Change in climate condition over a long period of time and melting of glaciers
- Intensity of heat waves and droughts will be increased
- Intensity of hurricanes will be increased with the increase in global warming
- Rise in sea level and it is expected to rise in  feet in coming 100 years

### 1.3.5 Total Equivalent Warming Index

It is a measure used to express the contribution of both indirect (through energy consumption) and direct (due to release into the atmosphere) effect of refrigerants into the atmosphere. From global warming point of view, a refrigerant with low value of TEWI is preferred.

### 1.3.6 Engineering Equation Solver and REFPROP

A program that is used to solve thousands of coupled non-linear algebraic and differential equation numerically is termed as EES. A significant feature of EES is high reliability database of hundred of fluids for finding their thermodynamic and transport property. REFPROP is a descriptor of REFerence fluid PROPerties, which was originated by NIST. It is used to find out the thermodynamic and transport features of fluids and their mixtures which are industrially significant. Tables and plots can be used to display these properties through the graphical user interface.

### 1.3.7 Refrigerant Mixtures

Refrigerant mixtures are the blend of two or more refrigerants. The properties like flammability, toxicity, GWP, ODP and oil miscibility etc., can be controlled by varying composition in the mixtures. Thus, refrigerant mixtures find wide application in refrigeration and air conditioning. These can be classified into the following two types:

#### 1.3.7.1 Azeotropic Refrigerant Mixtures

It is a blend of two or more refrigerants that boil and condense at the constant temperature under constant pressure and behave like a single component. Refrigerants in the azeotropic mixture have the same composition in the liquid and vapour phase. However, there is no ideal azeotropic mixture exists in refrigeration but for most of the azeotropic mixture change in composition is small. It is designated by the 500 series. It is used in a large number of refrigeration applications such as R502 is used for cold storage purpose which is a mixture of R22 (48.8%) and R115 (51.2%).

#### 1.3.7.2 Zeotropic Refrigerant Mixtures

It is a blend of two or more refrigerants that shows temperature variation during boiling and condensation processes under constant pressure and behaves like individual components. It shows different composition in liquid and vapour phases at equilibrium. Hence, if there is any leakage from the system then there will be a difference between original composition and remaining refrigerant composition. So, the use of zeotropic mixtures requires greater care in comparison to azeotropic mixtures. It is designated by the 400 series. It is also used in a large number of refrigeration applications such as R407C is used residential air conditioner as an alternative of R22 and it has a composition of 23%R32/R125 25%/52%R134a52%.

## 1.4 STANDARD TEN STATE POINT CYCLE

This cycle represents the standard rating cycle for domestic refrigerators. It approximates the operating and design conditions of a domestic refrigerator. This cycle is designed according to ISI Code No. 1476-1979. Ambient temperature corresponds to 32ºC is considered in this cycle. Subcooling of liquid refrigerant takes place from 43ºC to 32ºC in the regenerator whereas superheating of vapour refrigerant takes place to 32ºC in the suction line. Figure 4.1 in Chapter 4 explains various parameters of the standard ten point cycle. The ten state points correspond to test conditions are as follows:

- The pressure drop in the evaporator is 0.1 bar i.e. $P_8$-$P_9$
- The pressure drop on the compressor suction side is 0.1 bar i.e. $P_{10}$-$P_1$
- The pressure drop on the compressor delivery side is 0.25 bar i.e. $P_{2n}$-$P_{3n}$ or $P_{2s}$-$P_{3s}$
- The condition of vapour refrigerant at the compressor shell outlet/condenser inlet is $T_{3n} = T_{2n}$ or $T_{3s} = T_{2s}$ and condenser pressure = compressor shell outlet pressure
- The saturated vapour refrigerant state in the condenser is corresponded to 55 ºC i.e. $T_4$
- The saturated liquid refrigerant state in the condenser is corresponded to 55 ºC i.e. $T_5$
- The subcooled liquid refrigerant is leaving the condenser at 43ºC i.e. $T_6$
- The subcooled liquid refrigerant leaving the regenerator/entering the expansion device is at 32ºC i.e.$T_7$
- The evaporator temperature corresponds to -25ºC which represents the liquid vapour refrigerant mixture temperature leaving the expansion device/entering the evaporator
- The superheated refrigerant leaving the regenerator/entering the compressor at 32ºC i.e.$T_{2s}$ or $T_{2n}$

## 1.5 SOLAR EJECTOR–JET REFRIGERATION SYSTEM

One of the simplest approaches for using thermal energy in cooling is by the use of ejector cycles [44]. Ejector cycles are very reliable, low maintenance, free from vibration, simple and have limited applications in the field of air-conditioning. However, their energy efficiency is about 0.2 or less. It is expected that combining the ejector cycle with solar energy can facilitate higher energy efficiency. At present, the vapour compression system is mainly used in the air-conditioning and refrigeration. It is desirable to combine the solar energy-driven system with the vapour compression system for energy saving as well as to provide a comfortable environment. Environmental sustainability concerns such as global warming and the depletion of fossil fuels have led to shifting the focus on the use of renewable energy, which has a very little impact on the environment. As far as combined solar ejector–jet vapor compression refrigeration system is concerned, the most suitable application is in air conditioning because of the relatively higher evaporator temperatures required than for refrigeration applications. There is a requirement of new refrigerant like R1234yf for the replacement of existing such as R134a with negligible impact on the environment.

An ejector is a device in which a higher pressure fluid (called primary fluid) is used to induce a lower pressure fluid (called secondary fluid) into the ejector for mixing. Fluids from these two streams mix together and discharge to a pressure that lies in between the pressures of primary fluid and secondary fluid as shown in Figure 1.3. In an ejector refrigeration cycle, the compressor used in a vapour compression system is replaced by the ejector and a pump. The refrigerant R1234yf is cooled down in the condenser and separated into two streams. One stream goes into the evaporator through the expansion valve and the other is pumped back to the generator. Ejector refrigeration system has the inherent advantage of no moving parts and hence less maintenance needed during operation. The suffix g, e, c refers to generator, evaporator and condenser respectively as shown in Figure 1.4. It has been demonstrated in the present work that the refrigerant R1234yf as working fluids provides a better environmental option for solar ejector-jet compression systems, although they will require carefully developed safety protocols due to their flammability.

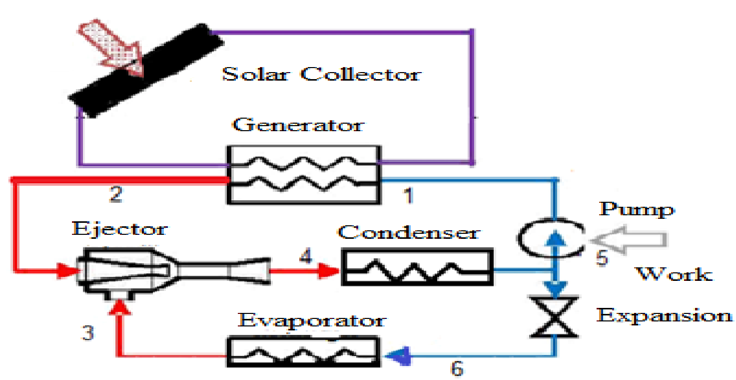



Figure 1.3 Solar ejector-jet refrigeration system

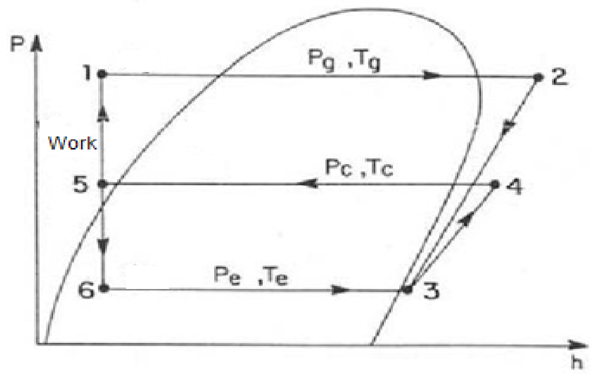



Figure 1.4 ph diagram of a solar ejector-jet refrigeration system

The results come out of the published paper during this thesis work are as follows:

- It is not necessary that an increase in evaporator temperature will result in an increase in refrigerating effect and COP. An optimum evaporator temperature exists at which refrigerating effect and entrainment efficiency provide the best COP.
- Solar energy requirement decreases with decrease in generator temperature in a solar ejector-jet refrigeration system.
- R1234yf as refrigerant provides better energy efficiency and environmental option for solar ejector-jet compression systems and its cycle simulation demonstrated that the proposed ejector-jet cycle can provide energy savings if solar energy is supplied.
- When the generator temperature decreases, the COP of solar ejector-jet refrigeration system increases.

It was analyzed using ejector-jet cycle parameters and established that there is new eco-friendly refrigerant R1234yf for the replacement of existing such as R134a with negligible impact on the environment.

**1.6 EXERGY ANALYSIS**

The concepts of energy, energy balance, energy transfer by work and heat, entropy, entropy balance and calculation of thermodynamic properties at equilibrium are provided in classical thermodynamics. According to the thermodynamic first law, energy cannot be created nor be destroyed but it can be transferred from one form to another. The second law of thermodynamics complements and enhances the energy balance given in the first law of thermodynamics by enabling calculation of the true value of energy carrier and the real inefficiencies as well as losses from the processes and systems. Thus, the second law of thermodynamics introduced the concept of exergy. Exergy is defined as the maximum possible reversible work obtainable in bringing the state of the system to equilibrium with that of the environment and it is an extensive property. From the analysis and design point of view of a thermal system, we are interested in finding something that can be destroyed and that quantity is exergy. Thus, exergy is the quantity that can compare the usefulness of say, 1 kJ of electricity generated from a power plant with that of 1 kJ of energy from the cooling water stream plant. Clearly, electricity has greater economic value and quality. Therefore, to analysis and design thermal system, the tool which can be used is exergy. The earliest contribution to the availability of energy started from 1868 by Clausius, Tait, Thomson, Maxwell and Gibbs. However, Rant's in 1956 had given the term exergy which has replaced various terms used in different languages such as usable energy, availability, available energy and work capability. Thermodynamic analysis includes calculating of the coefficient of performance (COP), work input and cooling capacity whereas exergy analysis includes exergetic efficiency, exergy loss, exergy destruction and its ratio in the overall system along with each component of the system. In all the real processes, a part of total exergy is irreversibly destroyed when it is supplied to a system which is not so in case of energy. The part of exergy connected to energy stream rejected to the environment is the exergy loss and the other part of exergy connected to the irreversibility of the system is the exergy destruction. Thus, the exergy destruction is due to irreversibility within the system and exergy loss is the exergy of a stream that is not used in the system.

## 1.7 THERMOECNOMIC ANALYSIS

In 1932, J.H. Keenan introduced the idea of exergy costing and suggested to use exergy in a cogeneration plant for allocating costs to the steam and electric power produced. M. Benedict presented a lecture on exergy destruction costing and obtaining optimal design with the help of these costs in 1949. The thermoeconomic was first coined by M. Tribus and R. B. Evans in 1950s. They had assigned exergy costs to the streams in desalination processes and apply cost balances in the same system at the component level. The exergy analysis provides all the necessary information required for performance and design analysis of an energy system as per thermodynamic point of view. However, there is a need to know how much it costs to plant operator when exergy destruction and exergy loss take place and these costs are calculated in the thermoeconomic analysis. The economic analysis gives us the methodology to find the cost allied with the investment, operating without considering fuel, fuel and maintenance of the system under consideration. Thermoeconomic is that field of engineering which brings together exergy analysis and economic principles to give system designers or operators with the availability of the information not provided by conventional energy analysis and economic evaluations but such evaluations are important for designing and operating a cost-effective system. It can be termed as exergy aided cost minimization [24]. So, by applying thermoeconomic analysis we will be able to understand the process of cost formation and the flow of cost in a given system. Thermoeconomic evaluation includes in examining cost of exergy destruction, cost of exergy loss, levelized electricity cost, relative cost difference, cost importance, exergoeconomic factor, rate of exergy destruction, exergy destruction ratio and exergetic efficiency.

**1.7.1 Cost of Refrigerator Components**

Most refrigerators have a common arrangement of components inside the refrigerator cabinet and their percentage cost is shown in Figure 1.5 and Table 1.2. The data were collected from Refrigeration and Air-conditioning companies in India.

Table 1.2 Cost of refrigerator components

| Items | Percentage cost |
|---|---|
| Compressor | 37.5 |
| Evaporator | 8.125 |
| Condenser | 6.875 |
| Expansion device | 1.875 |
| Plastic items, body and insulation | 21.875 |
| Assembling cost | 14.375 |
| Miscellaneous (Electrical wiring accessories drier, refrigerant etc.*)* | 8.75 |

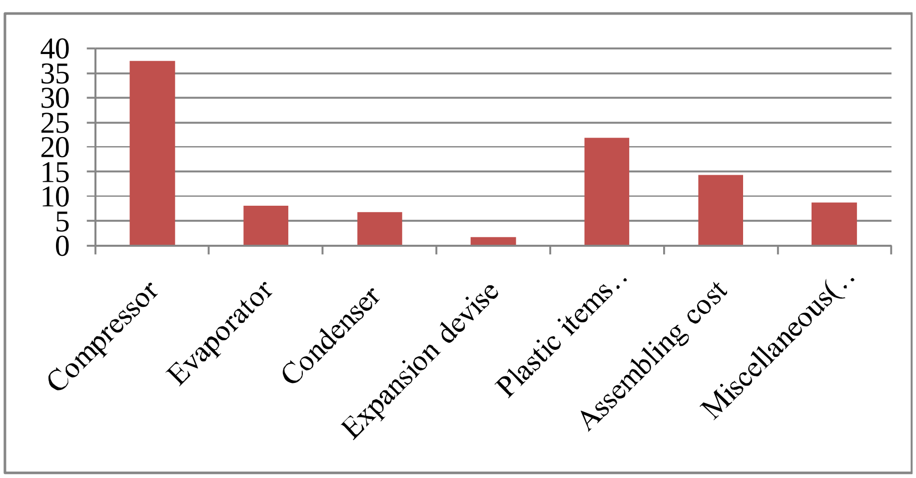



Figure 1.5 Percentage cost of parts of a refrigerator

### 1.7.2 Copper and Aluminium Coils

Cost is the first thing to consider while choosing among the copper and aluminium coil. Copper is better than aluminium as regard to heat transfer rate, toughness and it can be easily repaired. Aluminium is cheaper than copper and it can be employed economically in refrigerators while choosing material among copper and aluminium coils in condensers and evaporators. In the present work aluminium alloy storage device, evaporator and condenser are suggested for cost reduction.

### 1.7.3 Thermal Conductivity of Material

In refrigerators, materials which are used to make cabinet and door can be aluminium or steel sheet metal that is painted sometimes. Like the outer cabinet, inner cabinet of the refrigerator can also be made by using sheet metal/plastic and the insulation which is used to fill the gap between inner and outer cabinets is composed of polyfoam. The compressor, condenser, coils, fins are made of aluminium, copper, or an alloy. To ensure the metal ductility and bending ability without any breakage, copper is usually used for tubing. The tubing is usually copper because of its metal ductility and its ability to bend without breaking. Interior areas are made up of plastic or glass. It is advisable that storage rack, water bottles, storage baskets, utensils, ice trays are of good conducting material to increase heat transfer rates and energy saving. Thermal conductivity of various materials is given in table 1.3.

Table 1.3: Thermal conductivity of various materials

| Material | Acrylic Glass | Plastic, Fiber Reinforced | Stainless Steel | Copper Commercial | Aluminium Alloy |
|---|---|---|---|---|---|
| Thermal conductivity ($W\ m^{-1}K^{-1}$) | 1.9-2 | 0.7-1.06 | 16.3-24 | 360-423 | 72.8-135.6 |

## 1.8 OBJECTIVE OF THE PRESENT WORK

R134a is one of the most widely used refrigerants in refrigeration and air conditioning industry. R134a is a hydrofluorocarbon refrigerant. It has zero value of ozone depletion potential but high value of global warming potential around 1430. Thus, it is not affecting the ozone layer but contributing to global warming. Global warming is one of the biggest challenges faced by the world in the current scenario. Thus, an alternative to R134a is required. In October 2016, the Kigali amendment under the Montreal Protocol introduces the phase down HFCs production and consumption. This amendment was adopted by around 200 countries. It was decided under this amendment that countries will cut down the production and consumption of HFCs by 80% over a period of 30 years. This phase down is equivalent to cut of 80 billion metric tons of $CO_2$ by 2050 and this would avoid 0.4ºC warming of the earth's atmosphere. This amendment also helps countries financially for the transition towards eco-friendly refrigerants.

Present work concentrates on finding various alternative refrigerants of R134a. Alternative to HFCs refrigerants are HCs with no fluorine content. HCs are the refrigerants which occur naturally and perform outstandingly in terms of energy efficiency, critical point, solubility, transport, heat transfer properties and are environmentally sound. Some common examples of HCs are propane, n-butane and isobutene. The major concern related to HCs is their flammability. Therefore, to overcome the issue associated with HCs, two more refrigerants HFO1234yf and HFO1234ze introduced with GWP similar to hydrocarbons and less flammability. A lot of research has been performed on HFCs, HCs and HFOs but limited research work has been found with a refrigerant mixture of HFC and HFO. It has been found that HFOs are better than hydrocarbons but are flammable in comparison to R134a. Thus, present study considered the refrigerants HFCs, HCs, HFOs and mixture of refrigerants HFC/HFO to replace R134a in the vapour compression system. Exergy analysis will be performed for the variously considered refrigerants on a vapour compression system followed by thermoeconomic analysis. Results obtained from the analysis will be validated experimentally as well as with the results obtained from the various papers published in journals. Through these analyses, the best suitable refrigerant among all considered refrigerants will be found out. We can summaries our objective as follows:

- Refrigerant R134a is not harmful to the ozone layer but contribute to global warming. So, there is a requirement to discover an alternate refrigerant from the environmental point of view
- Comparison between different considered alternate refrigerants using conventional thermodynamic and thermoeconomic analysis of a vapour compression system
- Determination of suitable alternate refrigerant

## 1.9 ORGANIZATION OF PROPOSED THESIS

The thesis work is divided into six chapters. Each chapter focus on achieving the objectives of the research work. The organization of various chapters described in the present thesis work is as follows:

**CHAPTER 1: INTRODUCTION**

This chapter starts with an introduction to refrigeration systems, especially the vapour compression refrigeration system and solar ejector-jet refrigeration system. Afterwards, in depth detail of refrigerants, their history, properties of an ideal refrigerant, ozone depletion potential, global warming potential, refrigerant mixtures and standard ten point cycle are discussed. The component chosen for the present research is the finding of alternative refrigerants which can replace R134a. Refrigerant R134a has widespread application in domestic refrigerators and automotive air conditioning. Exergy and thermoeconomic analysis are also briefly discussed. Subsequently, this chapter discusses the objectives of the present research work and thesis organization.

**CHAPTER 2: LITERATURE REVIEW**

This chapter starts with literature available in the area of low GWP refrigerants, energy efficiency and various refrigerants for replacing R134a in the vapour compression system. Through this literature review, the various possible alternatives for R134a are selected. Afterwards this chapter focus on what research has been carried out in exergy and followed by thermoeconomic analysis of vapour compression system. Based on the literature survey, the identified research gaps have been presented.

**CHAPTER 3: ANALYSIS OF THERMAL SYSTEM**

This chapter discusses the methodology used for the present research work. Importance of energy, exergy and thermoeconomic analysis for the design of the thermal system are presented in this chapter. The formulas used to present study for the exergy and thermoeconomic analysis are described in details. Properties of all the considered refrigerants such as latent heat of vaporization, GWP, ODP and safety group etc. are discussed. This chapter also enlightens the process of defining the fuel-product-loss definition. The experiential procedure, safety aspect and refrigerant mixtures are discussed in detail.

**CHAPTER 4: ENERGY AND EXERGY ANALYSIS OF VAPOUR COMPRESSION SYSTEM USING HFCs, HCs, HFOs AND THEIR MIXTURES**

This chapter represents a simulation analysis of vapour compression system to find a drop-in replacement of R134a. A standard ten-state-points cycle is considered for the present work. A total of thirty refrigerants are selected for finding a substitute for R134a. The various parameters computed are pressure ratio, mass flow rate, relative volumetric cooling capacity, and relative coefficient of performance, cooling capacity, exergetic efficiency, exergy destruction and efficiency defect. The results obtained from the refrigeration process are validated on a vapour compression test rig. A program is made to analyze energy and exergy of the system into an Engineering Equation Solver and REFPROP is used to calculate various thermodynamic properties of refrigerants. This chapter also discusses the optimization of capillary tube parameters using environmentally friendly refrigerant.

**CHAPTER 5: THERMOECONOMIC ANALYSIS OF VAPOUR COMPRESSION SYSTEM USING ENVIRONMENT-FRIENDLY REFRIGERANTS AND THEIR MIXTURES**

In this chapter, thermoeconomic analysis of a simple domestic refrigerator is done. The refrigerants taken in the present study for replacing R134a are R152a, R600a, R1234yf, R1234ze, R134a/R1234yf (10%/90%) and R134a/R1234yf/R1234ze (40%/22%/38%). A computational model is made for thermoeconomic analysis of the system into an Engineering Equation Solver in order to find the best alternative of R134a among all considered refrigerants. The various parameters computed are operating cost, cost of exergy destruction, levelized electricity cost, energy efficiency ratio, exergetic efficiency, exergy destruction, pressure ratio, cooling capacity, power consumption, exergoeconomic factor and cost importance of component followed by validation of results. This chapter also describes the analysis of environmental friendly refrigerant from the thermal and economic point of view.

**CHAPTER 6: CONCLUSIONS AND SCOPE FOR FUTURE WORK**

In this chapter, the outcomes of the research work are discussed. This chapter tells us various conclusions, limitations and scope for future research works which can be drawn from present research work.

# CHAPTER 2
# LITERATURE REVIEW

## 2.1 OVERVIEW

As discussed in the previous chapter, the vapour compression process is the most commonly used refrigeration and air conditioning system. Refrigerant R134a is responsible for the increase in global warming and it has major applications in refrigerator and automobile air conditioning. Refrigerant R134a used in vapour compression system is harming our environment directly as well as indirectly. Directly it affects the environment by its leakage from the system and indirectly by energy consumption. To reduce the energy consumption we have to increase the value of the coefficient of performance whereas to reduce the effects of refrigerant on the environment we have to search for a new refrigerant which has zero value of ODP and negligible GWP without compromising the performance of refrigeration system. Therefore, from an environmental way of looking there is a requirement to discover an alternative to R134a. This chapter focuses on finding the alternatives of R134a from the literature review and then studying the exergy and thermoeconomic analysis of vapour compression system available in the literature. The literature survey is divided into three parts followed by finding a research gap as follows:

- Alternatives of R134a refrigerant
- Exergy Analysis
- Thermoeconomic Analysis

## 2.2 ALTERNATIVE OF HFC134a REFRIGERANT

HFC refrigerants are used in all major applications of refrigeration but these refrigerants are contributing towards global warming. By banning CFC refrigerants, ozone depletion is controlled but as the earth's temperature is increasing there is a need to find alternatives of HFC refrigerants. Thus, this section is based on reviewing the literature in the field of finding alternatives of refrigerant HFC134a used in vapour compression system.

**Devotta and Gopichand (1992)** compared the theoretical performance of R134a with R22, R134, R152a, R142b, R12 and R124 to find an alternative to R12. The performance parameters investigated and the energy input was determined lowest for R152a followed by R142b, R22, R134, R12, R124 and R134a respectively against the variation in evaporator temperatures. The performance of R134a and R152a was found very close to R12 and the system charged with R134a and R152a does not require much modification in place of R12.

**Bodio et al. (1993)** performed a thermodynamic analysis of butane-propane mixture and observed that it can be a possible alternative to replace R12 in a domestic refrigerator. It was found that no change in the construction of the domestic refrigerator was required when it was charged with hydrocarbon and power input to the system was also comparable. They showed through experimental investigation that the same lubricant can be used with proposed mixture and the flammability issues related to hydrocarbons refrigerants were also addressed in this paper.

**Richardson and Butterworth (1995)** investigated the hydrocarbon performance to replace R12 in the vapour compression system. Through investigation, it was elaborated that under same operating condition propane and propane/isobutane mixtures give better value of COP and do not require any modification in existing system. Mixture 50%R290/50%R600a exhibit similar performance as that of R12 and the value of COP was increased by increasing the proportion of propane in the mixture.

**Radermacher and Kim (1996)** suggested that there is a need for finding an alternative of CFC like R12 in domestic refrigerators due to their high value ozone depletion potential. Through their study, they suggested R134a and HCs as an alternative for R12 in domestic refrigerators because they are not harmful to the ozone layer. The result showed that R134a has got more advantage than HCs because of flammability issues related to HCs and global warming associated with HFCs was not much concern at that time.

**Alsaad and Hammed (1998)** carried out an experimental study to determine the performance of liquefied petroleum gas (LPG) in a domestic refrigerator. LPG is a mixture of HCs like 56.4% butane, 24.4% propane and 17.2% isobutane and it is also environmentally friendly. They coined that refrigerator without any modification worked satisfactorily with LPG as a refrigerant and produce no problem by the use of the same compressor and lubricating oil as that for R12. The mass flow rate of LPG showed an increasing tendency with an increase in evaporator temperature.

**Jung et al. (2000)** examined the performance of propane/isobutane mixture in a domestic refrigerator by varying mass fraction of propane in the range of 0.2 to 0.6. The mixtures with varying composition of propane/isobutane were tested theoretically as well as experimentally by comparing with R12. The result obtained through the analysis indicated that the 0.6 mass fraction of propane in the mixture of propane/isobutane has more energy efficiency and cooling rate than R12 and the proposed mixture has lesser compressor on-time.

**Granryd (2001)** presented an overview of hydrocarbons as a refrigerant. It was found that hydrocarbons as refrigerant possess excellent environmental properties like negligible GWP and zero ODP but it has limited scope as a refrigerant due to their flammability issue. He suggested that there is a need to make safety standards for their safe use and there is a need to develop technology from an economic point of view in a number of applications such as domestic refrigerator etc. Hydrocarbons were found to be more compatible than that of HFC134a in mineral oil.

**Tashtoush et al. (2002)** performed an experimental study of HFC/HCs mixture to replace CFC12 in a domestic refrigerator. The refrigerant mixture taken for the study was butane/propane/R134a in different compositions. The results showed that mixture having composition 31.25%propane/31.25%butane/37.5%R134a as a mass fraction can be used without any change in lubricating oil and condenser. It was observed from the results that with the decrease in the proportion of R134a in the mixture, the value of COP was increased.

**Calm (2002)** studied the emission impact from the air conditioning and refrigeration on the environment. He suggested that there is a need to find a refrigerant which has zero ODP, negligible GWP, safe to use, compatibility, low cost and most important low energy consumption. Many issues in the currently used alternative of R123 in centrifugal chillers were noticed such as increasing global warming due to their higher energy consumption. Thus, there is a need to take care of all prospective related to environmental benefits before changing an existing refrigerant.

**McMullan (2002)** presented the issues and future strategies in solving a problem related to the environment due to refrigeration. This paper discussed the need for replacement of currently used CFC and HCFC refrigerants with R134a and R407C etc. due to their impact on the ozone layer and global warming impact. This paper was focused to educate industry, institute and public awareness for finding and using the suitable alternative of CFC and HCFC. This paper also presented various alternatives of CFC and HCFC refrigerants.

**Naer and Rozhentsev (2002)** discussed the use of HCs mixtures in cryogenic and small refrigerating machines. In such cases, the amount of HCs mixtures used is between 50-80 gram and it is safe to use. Cryogenic small coolers (temperature range -73 to -183 ºC ) with HC mixture as a refrigerant was reported as an alternative to another cooling system.

**Akash and Said (2003)** performed the experimental study on LPG having mass fraction composition as 30% propane, 15% iso-butene and 55% n-butane. This study was performed on a domestic refrigerator to find alternatives for CFC12. It was found that among all mass charges of LPG, best result was obtained for mass charge of 80 g. The value of COP increased at a higher rate with an increase in evaporator temperature for 80 g charge than that of other mass charges.

**Mao-Gang et al. (2005)** analyzed the performance of refrigerant mixture HFC152a/HFC125 for a domestic refrigerator. The outcomes showed that the optimum value of mass proportion was 85% HFC152a/15 % HFC125 and amount of charge were 97g. The energy consumption in refrigerator showed a decreasing tendency with an increase in charge value up to an optimum value than it again starts increasing.

**Wongwises and Chimres (2005)** worked on HC mixtures to find a replacement of HFC134a in a domestic refrigerator. The HCs under investigation were propane, isobutane, butane and their mixtures. The experiment was performed under no load condition having a temperature of 25ºC. The best alternative of R134a obtained from the study was HC mixture having a composition of 60% propane/40% butane.

**Fatouh and Kafafy (2006) [48]** evaluated the performance of LPG having composition 60% propane and 40% commercial butane in comparison with R134a on a domestic refrigerator. The experiment was performed by varying the capillary length in the range 4 m to 6 m for LPG whereas in case of R134a it was kept constant at 4 m. The results indicated that LPG can be used in the existing refrigerator as an alternative of R134a except for a change in initial charge and capillary tube length.

With the increase in mass charge of LPG, it was found that the value of discharge temperature, mass flow rate, and energy input increase irrespective of capillary length.

**Mohanraj et al. (2007)** compared the performance of HC mixture 45% HC290/55% HC600a consisting of different mass charges 40 g, 50 g, 70 g and 90 g with HFC134a. HC mixture having mass charge 70 g in domestic refrigerator was found as best alternative of R134a among all considered mass charges. It has shown better COP, lower discharge temperature and energy consumption than R134a. It was observed that with the increased charge of refrigerant the value COP and energy input increased.

**Mani and Selladurai (2008)** conducted an experimental analysis for finding an alternative of refrigerants R12 and R134a by using HC mixture in weight proportion as 68% R290/32% R600a. The proposed HC mixture was found to have nearly same discharge pressure and temperature as that of R12 and R134a. Refrigerating capacity and COP was higher for HC mixture as compared to R12 and R134a. So, it was concluded that the proposed mixture can be used as a drop-in replacement. The refrigeration efficiency in refrigerator elevates with rising evaporator temperature up to an optimum value than it starts decreasing.

**Jwo et al. (2009)** analyzed a refrigerator to find the performance of refrigerant mixture having a composition of 50% R290/50% R600a with varying mass charge as compared to R134a. The result showed that considered HC mixture had a better refrigerating effect, lower energy consumption and the amount of mass charge required was reduced by 40%. The value of lower mass charge of HC mixture was found to have a higher value of COP as compared to higher mass charge.

**Mohanraj et al. (2009) [88]** presented a study on halogenated refrigerants to finds its alternative from environment point of view. Halogenated refrigerants were mostly used in developing countries but they have an adverse impact on the environment due to their high value of global warming potential and ozone depletion potential. This paper reviewed various alternative refrigerants such as HC, HFC mixtures and HC/HFC mixtures, in addition, their technical difficulties and future challenges.

**Mohanraj et al. (2009) [90]** discussed the phase down of HFC134a under Kyoto protocol due to the high value of GWP. They proposed an HC refrigerant mixture having composition 45.2%R290/54.8%R600a by weight percentage in place of HFC134a. The result indicated that proposed refrigerant has higher COP, lower energy consumption, lower discharge temperature and pull down time. The maximum value of COP was obtained at the capillary length of 5 m when it is varied in the range 4-6 m.

**Dalkilic and Wongwises (2010)** studied the vapour compression system using various mixtures of HFCs and HCs refrigerants. This study was conducted to find alternatives for R12, R22 and R134a. The efficiency of the system was enhanced by using superheating and subcooling. The results showed that among all considered refrigerants, the alternative of R12 and R22 comes out to be 40%R290/60% R600a and 20%R290/80%R1270 by weight respectively. It was found that with rising evaporator temperature there was a rise in the values of COP and refrigerating effect whereas values of energy input and pressure decrease.

**Bolaji (2010) [29]** performed an experimental study using R152a and R32 in a refrigerator to find an alternative to R134a. Both the refrigerants showed better COP than that of R134a but less energy consumption was obtained by using R152a only. It was observed that with extending the value of refrigerant charge the value of discharge pressure also increases. The discharge pressure of R152a was found to be nearly same as that of R134a whereas in case of R32 it was much higher than R134a. Thus, R152a can be used in a domestic refrigerator as a substitute for R134a.

**Bolaji (2011)** presented a paper on the methodology for the selection of alternative refrigerants from environment point of view. This paper focused on shortcoming that must be taken into consideration so that proposed alternative refrigerants are accepted worldwide. The various alternative refrigerants discussed in this paper were HC, azeotropic & zeotropic mixtures, carbon dioxide and HFC. It was found using matrix triangles that emerging refrigerants from the derivatives of Methane were R23 and R32 whereas from the derivatives of ethane were R134a, R143a, R124 and R152a. These proposed refrigerants are environment-friendly, non toxic and have lower flammability value than HC.

**Ooi (2012)** compared the compressor performance using R134a and R1234yf as a refrigerant. In this paper two compressors were used out of which one is same as that of used in R134a and second is designed especially for R1234yf. The results showed that compressor worked better with R1234yf at low condenser pressure and vice versa at low evaporator and high condenser temperature. At a lower value of condenser temperature R1234yf performed better than R134a.

**Karber et al. (2012)** carried out a research work using refrigerants R1234yf and R1234ze as an alternative to R134a. R1234yf and R1234ze are HFO refrigerants having a low value of GWP and zero ODP. Although results showed that R1234ze has less energy consumption than R1234yf but still it cannot be used as a substitute to R134a because of its lesser capacity. Thus, R1234yf comes out to be a future alternative of R134a in a domestic refrigerator.

**Rasti et al. (2012)** performed an experimental study to find out the performance of R436A in a domestic refrigerator and compare its performance with R134a. R436A is a mixture of 56%R290/44%R600a by weight. The energy consumption in refrigerator decreases with an increase in charge value (range between 45-80 g) up to an optimum value than it starts increasing. Results indicated that energy consumption and ON time ratio were reduced in comparison with R134a and also there was a reduction of 45% in charge value for optimum value. There was an enhancement in the value of energy efficiency index with the use of R436A.

**Benhadid-Dib and Benzaoui (2012)** presented a study on the need to find alternatives of HFCs and HCFCs due to their impact on the environment. HFCs and HCFCs contribute to global warming and ozone layer depletion because of fluorine and chlorine content. So, there is a need to find a refrigerant from an environmental point of view and which can be socio-economically accepted. Thus, natural refrigerants come out as one of the solutions to this problem.

**Sanaye and Dehgandokht (2012)** studied the performance of mini-channel type evaporator and compared it with laminated evaporator used in MAC. Mini channel type evaporator showed higher pressure drops and cooling effects. Reduced fuel consumption was obtained by the use of mini-channel due to its lower outer air temperature and enthalpy.

**Lee and Jung (2012)** examined the performance of R1234yf in a bench tester in order to replace R134a in MAC due to its high GWP. The test was performed under winter and summer condition. System equipped with R1234yf showed a lower value of COP, capacity, charge required and discharge temperature at the outlet of the compressor. Thus, R1234yf can be used as an alternative to R134a in MAC from an environmental way of looking.

**Navarro-Esbri et al. (2013)** performed an analysis to find an alternative of R134a in a vapour compression system. The alternate refrigerant taken for the present study was HFO1234yf due to its excellent environmental properties. The effect of increased in compressor ratio value on compressor volumetric efficiency was also presented. Experimental tests showed that system R1234yf has a lower value of cooling capacity and COP in contrast to R134a but this difference in values was considerably reduced by the use of internal heat exchanger.

**Mohanraj (2013)** compared the performance of R403A with R134a in a domestic refrigerator. The analysis was performed for condenser temperatures having values 40ºC, 50ºC and 60ºC with evaporator temperatures having a range from -30ºC to 0ºC. The results coined that same compressor size can be used for both the refrigerants due to their similar volumetric cooling capacity. The maximum value of COP was obtained for R403A at 40 ºC condenser temperature under the variation in evaporator temperature. Also, TEWI of R403A was found to be lower than R134a. So, it can be used in place of R134a from environment and energy perspective.

**Lee et al. (2013)** evaluated the performance of R134a, R1234yf and their mixtures having a proportion of R134a as 5%, 10% and 15%. The test was performed under summer as well as winter condition. The test revealed that R1234yf and its mixture with R134a have a similar value of the coefficient of performance, discharge temperature and capacity as that of R134a. Therefore, the mixture having 10-11% of R134a can be used as a substitute to R134a, as it is non-flammable and has GWP less than 150.

**Mota-Babiloni et al. (2014)** studied the performance of HFO1234yf and HFO1234ze(E) as a drop-in replacement for R134a. Tests were conducted with and without an internal heat exchanger. The variation of volumetric efficiency with compressor ratio was presented in the paper. The parameters evaluated were volumetric efficiency, COP and cooling capacity. It was found that with the use of internal heat exchangers there was an increment in the value of parameters.

**Wang and Amrane (2014)** investigated the various alternative refrigerants from a global warming point of view. They tested thirty-eight low global warming refrigerants for a variety of applications to replace R134a, R22, R404A and R410A and several refrigerants come out as an alternative for existing refrigerants with varying GWP.

**Deveccioglu and Oruc (2015)** described the properties of trial stage low GWP refrigerants. Various new generations of refrigerants were discussed and refrigerants like R1234yf, L40, DR5, and R444B were selected as an alternative to R134a, R404A, R410A and R22 respectively. The properties of the refrigerants mentioned in this paper were found out by using the software REFPROP.

**Righetti et al. (2015)** compared the performance of R1234yf, R1234ze(E) and R600a with R134a for a domestic refrigerator. The analysis revealed that all considered refrigerants have low GWP and can be a good substitute for R134a but there is a requirement of a change in compressor displacement volume. The result coined that with an increase in mass flow rate the refrigerating capacity also increases and the maximum value was found for R134a at an evaporator temperature of -20ºC. For same mass flow rate, R1234yf showed the performance similar to that of R134a. Thus, it can be taken as a direct substitute for R134a in refrigerators.

**LIopis et al. (2015)** empathized the need for finding the low GWP refrigerants due to F-Gas regulation and in the product in which leakage rate is high. HFOs and natural refrigerants can be a good alternative for high GWP refrigerants. It was suggested to use two-stage refrigeration systems with low GWP refrigerants. With increasing temperature difference in the cascade heat exchanger, the value of COP will decrease. The result showed that for future low GWP refrigeration system, there is a need to compromise with issues like flammability and toxicity.

**Mota-Babiloni et al. (2016) [95]** presented a review paper on the use of R1234ze(E) as a refrigerant and stressed on the need to phase out high GWP HFCs refrigerants. This paper considered R1234ze(E) as an alternative refrigerant and discussed its various properties. It was found from the study that some modification in the system was required to replace R134a with R1234ze(E) for matching the performance. It was suggested that if R1234ze (E) is mixed with other refrigerants, then GWP value will be reduced and it can replace HFCs refrigerants.

**Aprea et al. (2016)** investigated the study of substituting R134a with R1234yf in domestic refrigerators. The analysis revealed that there was a decrease in pull down time, duty cycle and an increase in compression ratio. The condenser pressure was observed higher in R1234yf. The value of cooling capacity and energy consumption was found nearly same as that of R134a. Thus, refrigerant R1234yf can be used as a drop-in replacement for R134a

**Aprea et al. (2016) [14]** studied the performance of R1234ze when it is used as a replacement for R134a in a domestic refrigerator. The direct and indirect global warming effect of R1234ze was very less as compared to R134a found through experiments. The lower values of LCCP and TEWI for R1234ze indicated that it can be an excellent substitute for R134a.

**Mota-Babiloni et al. (2017) [94]** performed the modification in the system in order to increase the cooling capacity of the system working on refrigerant R1234ze(E). R134a is going to be phased out from most of the refrigeration and air conditioning application due to its high GWP value. R1234ze(E) was proposed as an alternative to R134a but results revealed that it has low cooling capacity than R134a. In order to achieve the required cooling, the system was charged with 43% higher charge and also an internal heat exchanger was used.

**Mota-Babiloni et al. (2017)** presented a study on low GWP synthetic refrigerants as an alternative for HFCs refrigerants. There is a requirement of finding an alternative to HFCs due to Kigali's amendment. Due to new regulation imposing on the use of HFCs there is a need for new refrigerants which might be a mixture of HFOs and HFCs. The new refrigerants are expected environmental safe, possess low flammability, economical and non toxic.

**Devecioglu and Oruc (2017)** compared the performance of R1234yf, R444A and R445A as low GWP refrigerants in MAC systems. It was found that cooling capacities of R444A and R445A were higher than that of R1234yf but vice versa in case of COP. The value of volumetric flow rate was observed higher at lower evaporator temperature. It was concluded that R444A can be used in heavy motor vehicles due to its low value of flammability.

**Aprea et al. (2017)** investigated the effect of using R1234yf and its mixture with

R134a having proportion 90% R1234yf/10%R134a by weight in a domestic refrigerator. The proposed mixture comes out as an excellent drop-in replacement for R134a due to its similar performance with respect to pressure and temperature. When the proposed mixture was used at optimum charge condition it has resulted in nearly 14% reduction in energy consumption.

**Fang et al. (2017)** studied the ejector refrigeration cycle to replace R134a with proposed HFOs refrigerants R1234yf and R1234ze(E). It was observed that the value of ejector entrainment ratio increase with an increase in the proportion of R134a in the mixture of R134a/R1234yf and R134a/R1234ze. The results indicated that there was a requirement of some modification in the system when R1234ze(E) used in the system. It was found that the value of COP and cooling capacity decrease with the use of proposed refrigerants although R1234yf performance is better than R1234ze(E).

**Borokinni et al. (2018)** analyzed the domestic refrigeration system when it is retrofitted with refrigerants R510A and R600a. It was found that with elevating evaporator temperature discharge pressure decrease. The considered refrigerants showed better performance in terms of COP and refrigerating effect but R510A indicated a higher value for volumetric cooling capacity. Refrigerant R510A performed better than R600a to replace R134a.

**Aprea et al. (2018)** conducted pull down and one day electric consumption tests to replace R134a with R1234ze(E) in domestic refrigerators. The pull down test was performed to find an optimum charge required and the system showed 9% improvement in energy consumption after 24 hours of working. Thus, R1234ze(E) can be used as a replacement for R134a due to its excellent environmental properties.

## 2.3 EXERGY ANALYSIS

Energy analysis is not enough to find the irreversibility at the component level and overall system. It tells us about only the performance of the system in terms of COP and energy input but there is a requirement of finding the term which is destroyed in the system. The exergy is the term which is destroyed or lost in a system and from the system to the surrounding. The exergy analysis is very helping in understanding the flow process and finding the worst component so that more research can be performed for increasing the performance of the system. Thus, exergy analysis is more helping in understanding and designing of system than that of energy analysis. This section is based on a review of literature in the field of the exergy analysis of the environment-friendly alternatives of HFC134a in a vapour compression system.

**Fratzscher (1997)** discussed the applications of exergy and results obtained through it. In this paper, a description of various terms associated with exergy was defined and the differences between energy and exergy analysis were discussed. Importance of exergy in solving problems related to industrial engineering was described. Exergy is found to be one of the most important tools in understanding a given system.

**Nikolaidis and Probert (1998)** used R22 as a refrigerant and apply exergy analysis on a refrigeration system based on vapour compression for finding the effect of a change in temperature of evaporator and condenser on the irreversibility of the system. It was found that the condenser efficiency defect was elevating with the rise in condenser saturation temperature. The study revealed that irreversibility of the plant can be minimized by declining the temperature difference between condenser & environment and also between cold room & evaporator.

**Chen and Prasad (1999)** analyzed an actual cycle through computer simulation for R12 and R134a in a vapour compression system. The value of exergy loss increased with decrease in the value of evaporator temperature. The results showed that R134a has a slightly higher value of exergy loss, lower value of the coefficient of performance and it consumed higher energy input as compared to R12.

**Aprea and Greco (2002)** studied the exergy analysis to substitute R22 with refrigerant mixture R407C. It was found that exergetic performance of R22 was better than R407C and then the exegetic performance of different components carried out in order to find the worst component from exergy point of view. The compressor was found to contribute most towards irreversibility and then methods of its improvement also discussed.

**Yumrutas et al. (2002)** investigated the impact of the change in temperature of condenser and evaporator on the performance of the refrigeration system through exergy analysis. It was found that by declining the temperature difference between condenser & environment and also between cold room & evaporator, the value of the coefficient of performance, second law efficiency increased whereas the value of total exergy loss for the system decreased.

**Aprea et al. (2003)** carried out the exergy analysis of refrigeration system working on different refrigerants R22, R407C, R417A and R507 by varying the revolution in the compressor from 30-50 Hz. The best performance was obtained for R22 followed by R407C. Expansion valve showed least exergy destruction among all the main components of the system. With increase in speed of compressor, there was a decrease in the value of exergetic efficiency and COP. It was also found that the R407C can be used as a substitute of R22 among all considered refrigerants.

**Srinivasan et al. (2003)** investigated the performance of $CO_2$ in a vapour compression system using exergy analysis. Through this analysis, optimum operating conditions were identified for obtaining the maximum value of COP and exergetic efficiencies. This paper presented various equations for the calculation of the performance of the system.

**Sencan et al. (2006)** applied exergetic analysis for comparing the performance of various environment-friendly refrigerants in vapour compression system using an artificial neural network. The exergetic efficiency decreased with increase in the value of pressure ratio up to an optimum value then it again starts decreasing. System irreversibility and COP were strongly affected by the temperature of evaporator and condenser. The performance of R134a & R407C was found similar whereas it was different for R410A.

**Arora et al. (2007)** analysed the performance of the refrigeration system for different refrigerants by applying exergetic analysis through a computational model. The results showed that the performance of R22 was higher as compared to R407C and R410A. The optimum value of evaporator temperature was found out at different temperatures of the condenser to find minimum exergy destruction. It was found that the total irreversibility of the system was decreased with increasing the value of compressor efficiency and increasing difference between condenser & evaporator temperature.

**Lior and Zhang (2007)** empathized on the importance of exergy related terms and the difference between second law efficiency and energy. Various equations were used to describe the various terms of exergy. This paper also focused on understanding terms relating to energy like COP.

**Kabul et al. (2008)** carried out a study to find the performance of R600a in refrigeration system from energy and exergy point of view. The results indicated that compressor had the highest exergy destruction followed by the condenser, expansion valve, evaporator and heat exchanger respectively. The performance parameters like COP, exergy destruction and efficiency ratio etc. were affected by the change in temperature of evaporator and condenser temperature.

**Arora and Kaushik (2008)** presented a computational model on exergetic analysis of refrigeration system working on R502, R404A and R507A. The analysis was carried out using different ranges of temperature for condenser and evaporator. The most effective component from exergy point of view was liquid vapour heat exchanger followed by the evaporator, an expansion device, compressor and condenser respectively. It was observed that efficiency defect in expansion valve was decreased with an increase in evaporator temperature. The results revealed that R507A is a better substitute for R502.

**Arora and Sachdev (2009)** compared the performance of R422 series refrigerants for finding the alternative of R22 using energy and exergy analysis. Energy Equation Solver software was used to execute the program for the analysis and properties of refrigerants were found from REFPROP. The most effective component from efficiency defect point of view was evaporator followed by compressor, expansion device and condenser respectively. R422B showed better performance among all the considered refrigerants of R422 series.

**Dopazo et al. (2009)** investigated the energy and exergy analysis of the environmental friendly natural refrigerants like $CO_2$ and $NH_3$ for a cascade vapour compression refrigeration system. The optimum value of condenser temperature with $CO_2$ as refrigerants obtained by applying the optimization process and this value increases with increasing the temperature of $NH_3$ condenser in the designed cascade system. The results showed that the value of isentropic efficiency for compressor greatly affect the optimum condition and COP.

**Shilliday et al. (2009)** analyzed the performance of R744, R404A and R290 for a vapour compression system. R404A and R290 showed a higher value of COP than that of R744 in single stage refrigeration system. The compressor has highest value of exergy destruction ratio for R404A and R290 whereas in case of R744 it was for throttle valve. It was observed from the results that R744 has a maximum value of exergy destruction ratio over the range of condenser and evaporator temperatures.

**Karkri et al. (2009)** applied the energy and exergetic analysis of R410A in refrigeration system using EES software. This paper emphasized the importance of evaporator and condenser temperatures in analyzing the vapour compression system. The study revealed that both these temperatures strongly affect the coefficient of performance, exergy loss, exergetic efficiency compressor and heat exchanges of the system.

**Bolaji (2010) [28]** studied the performance of R12, R134a and R152a in the home refrigerator by applying first and the second law of thermodynamic analysis. It was found that the value of COP for R152a was similar to that of R12 and a lesser value obtained for R134a. The overall efficiency defect was found lowest for R152a among all considered refrigerants. The exergetic efficiency was observed highest for R152a and its value increase with an increase in evaporator temperature for all considered refrigerants. Thus, R152a can be used as an alternative for R12.

**Padilla et al. (2010)** compared the performance of R413A when it was retrofitted in refrigeration system working on R12 by applying exergy analysis. System working on R413A was found to consume less energy input and has higher exergetic efficiency than that of R12. The system working on R413A takes the shortest time for achieving the desired value of temperature. Thus, R413A can be retrofitted in the existing system without any modification.

**STANCIU et al. (2011)** applied exergy analysis for finding the impact of parameters affecting the performance of a vapour compression system using refrigerants R22, R134a, R717, R507A and R404A. Effects of parameters like compression ratio and superheating on exergy destruction, refrigerating effect and COP were discussed in this paper. The outcome depicts that COP value decrease with increasing the value of compression ratio for all the refrigerants. It was found that exergy destruction increased with increase in the value of compression ratio.

**Ahamed et al. (2011)** studied the prospect of HCs like R290 and R600a as an alternative refrigerant for HFC from exergy point of view. Exergy loss was found highest in the compressor among all the components of vapour compression system and it was found lesser for R290 & R600a as compare to R134a. Exergetic efficiency showed a tendency of increasing with increase in the value of evaporator temperature. Exergetic efficiency was highest for R600a and value of COP was nearly the same among all considered refrigerants.

**Ozgur et al. (2012)** elaborated the energy and exergy analysis of R1234yf as a substitute to R134a. The rate of exergy destruction was observed higher in R134a as compared to R1234yf in the compressor of vapour compression system. With raising the value of condenser temperature the rate of irreversibility for both the refrigerants was found increasing. The performance of both the refrigerants was found comparable. Thus, R1243yf can be used as an alternative if the flammability issue of the refrigerant has been addressed.

**Ahamed et al. (2012)** analyzed the performance of domestic refrigerator using hydrocarbon refrigerants. Exergy loss was found lower at higher evaporator temperature and it was highest for the compressor. R600 and R600a showed a lower value of exergy loss as compared to R134a. Sustainability index of hydrocarbons was determined more than that of R134a and its value was highest for R600 among all considered refrigerants.

**Reddy et al. (2012)** analyzed a refrigeration system using refrigerants R134a, R152a, R143a, R404A, R407C, R410A, R502 and R507A. The results revealed that among all considered refrigerants R134a showed best performance and R407C showed poor performance. The values of COP and exergetic efficiency were affected by evaporator and condenser temperatures. The highest value of exergetic efficiency was found for R134a among considered refrigerates whereas highest fluctuation in exergetic efficiency value of nearly 10.34% was found for R407C for selected condenser temperature range.

**Aprea et al. (2013)** carried an exergetic analysis to replace R134a with natural refrigerant like carbon dioxide ($CO_2$). The overall performance of R134a was better than that of R744. It was revealed from the analysis that the thermodynamic performance of the system was improved with an increase in evaporator temperature, There was a marginal improvement in the system performance with the use of internal heat exchanger in the transcritical cycle.

**Ansari et al. (2013)** presented an exergetic analysis on environment-friendly refrigerants like R1234yf and R1234ze which can be used as a substitute to R134a in a refrigeration system. The value of COP and exergetic efficiency was found highest for R1234yf. The most efficient component from exergy destruction point of view was liquid vapour heat exchanger followed by evaporator, expansion device, compressor and condenser. Thus, results revealed that R1234yf can be used as a drop-in substitute for R134a but R1234ze required some modification in the system.

**Saravanakumar and Selladurai (2013)** compared the thermodynamic performance of HC mixtures as a substitute for R134a in the home refrigerator. The proposed HC mixture for the analysis was 45% R290/55% R600a. The exergetic analysis revealed that the value of COP and exergetic efficiency was better for mixture than that of R134a and also efficiency defect value was lower for mixture and the highest value of exergy destruction was found in the compressor. The results coined that the value of exergetic efficiency decreases with an increase in the value of evaporator temperature for both the refrigerants and decline in its value was more at a higher temperature of the evaporator for R134a.

**Menlik et al. (2013)** performed a thermodynamic analysis of R407C and R410A to find an alternative to R22 in a refrigeration system. The analysis was carried out by varying the values of temperatures of the evaporator, condenser, superheating/subcooling and dead state. The overall values of coefficient of performance and exergetic efficiency were higher for R407C than that of R410A and it was also having lesser exergy destruction ratio. Thus, R407C was found a better alternative for R22.

**Murthy et al. (2013)** investigated the performance of R134a, R290 and R407C as an alternative of R22 in the vapour compression system. Although R290 performed better in terms of COP and irreversibility than that of all considered alternative refrigerants but still it was not considered as best alternative in the window air conditioner due to its flammability issue. So, R407C comes out as one of the alternative of R22 among all considered refrigerants.

**Yataganbaba et al. (2015)** carried out exergetic analysis of HFO1234yf and HFO1234ze in a refrigeration system to a find an alternative of R134a. A computer program on EES was also developed for the analysis. It was observed from the result that performance parameters of R1234yf were comparable for R134a. Thus, it can be used as an alternative to R134a but R1234ze required some modification in the system. The compressor was found worst component from exergy destruction point of view.

**Cho and Park (2016)** conducted a study on variable speeds of the compressor in an automotive air conditioner with HFO1234yf. The system with HFO1234yf was tested with & without internal heat exchanger and also compared with R134a. The highest value of discharge pressure was observed for R134a in comparison to R1234yf with and without a heat exchanger. The value of COP and exergetic efficiency was improved with internal heat exchanger and can be compared with R134a. Exergy destruction ratio of R1234yf with internal heat exchanger was found lower than that of R134a at compressor speed of 2500 rpm.

**Golzari et al. (2016)** compared the performance of an automotive air conditioner using R1234yf as an alternative to R134a. A computer program was prepared on EES for exergetic analysis. It was revealed from the analysis that HFO performed better in terms of exergetic efficiency and exergy destruction. The compressor was found to have the highest value of exergy destruction. The value of exergy loss and exergetic efficiency was found to be decreased with increase in the value of evaporator temperature.

**Ahamed et al. (2017)** studied the exergetic analysis of domestic split air conditioner to replace refrigerant HCFC 22 with a mixture of HCFC and HC. Refrigerant mixtures prepared by weight percentage were 90% R22/10% R290 and 85% R22/15% R290, and properties were taken from REFPROP software. The value of exergy loss was found higher for 85% R22/15% R290 as compared to 90% R22/10% R290 and R22. The condenser was found to have the highest value of exergy destruction and exergetic loss was found affected with change in evaporator temperature.

**Morad et al. (2018)** carried out the thermodynamic investigation of an advanced refrigeration system using refrigerants such as R22, R134a, R404A and R410A. It was found that subcooling affects the thermodynamic performance of the system. The exergy destruction of the system was found declining with expanding evaporator temperature and its highest value was for the condenser. The system performed best with R22 followed by R134a, R410A and R404A. The value of mass flow rate and power input was observed to decrease with an increase in evaporator temperature.

## 2.4 THERMOECONOMIC ANALYSIS

Through exergy analysis, we are able to understand the flow process and performance of components. But from the overall design point of view, use of engineering economic is equally important as that of exergy analysis. Thermoeconomic analysis is an approach that joins exergy analysis with that of engineering economics. With the help of thermoeconomic, we are able to assign a cost to the input exergy stream and output exergy stream. It enables us to understand the process of cost formation within the system and the technique of reducing the total cost of a system which includes capital investment and operating cost. This section is based on the literature review in the field of thermoeconomic analysis of vapour compression refrigeration system using HFC refrigerants and its alternative from environmental aspects.

**Tsatsaronis (1993)** discussed the thermoeconomic and exergetic analysis application in the energy system. Thermoeconomic is a method that is defined as a combination of exergy and engineering economics for the design and analysis of the system. This paper described the history related to exergy and thermoeconomic. It also empathized on the importance of analysis of a system using thermoeconomic. This paper explained methods of exergy analysis, economic principles, exergy costing, thermoeconomic evaluation and optimization.

**Lozano and Valero (1993)** described the process of cost estimation in thermal systems. Mathematical formulations related to thermoeconomic analysis were explained and the cost of each stream in the system was calculated. This paper optimized the system as a whole and also at the local level.

**Lozano et al. (1993)** compared the methodology of optimization using the theory of exgertic cost and the method of Lagrange multipliers for thermoeconomic optimization. The results obtained from both the methods do not show much difference although it was found that the theory of exergetic cost was a much easier method compared to others.

**Massimo et al. (1998)** applied the thermoeconomic technique for the optimization of a refrigeration system for reducing overall and optimization cost. This paper used theory of exergetic costing for optimization and also defined a cost model for various parts of refrigeration plant. The results showed that if more investment can be made on the plant, evaporator and electric motor then cost related to electricity consumption can be decreased.

**Dingec and Ileri (1999)** carried out the optimization of refrigerators by selecting the compressor efficiency and areas of evaporator & condenser as the decision variables. Two cases are considered in this paper for thermoeconomic optimization: in the first case compressor efficiency was considered constant & optimized values of areas were determined and in the second case optimized values of compressor efficiency and areas of evaporator & condenser were determined.

**EI-Sayed (1999)** explained the concept of thermoeconomic through examples of sea water distillation process. This paper used costing equations for optimization of given systems along with the process of cost formation. This paper analyzed the given systems using the first and second law of efficiency.

**Hasan and Darus (2003)** optimized a home air conditioning system by choosing exergetic efficiency as the decision variable for the main component of the system. Exergetic analysis of the considered system showed that evaporator and compressor were the worst components from exergy point of view. There was an improvement of 11 % in cost saving due to thermoeconomic optimization.

**Misra et al. (2003)** applied thermoeconomic analysis for decreasing the operating and amortization cost in an air conditioning system working on vapour absorption system. The fuel product loss table was presented for the analysis of the considered system. Firstly, the thermodynamic analysis was performed followed by thermoeconomic optimization. It was revealed from the results that improvements in COP, exergetic efficiency and reduction in exergetic destruction in cost were 10.419%, 10.423% and 11.5% respectively. This improvement was achieved at the expense of 3.14% increase in investment cost.

**Al-Otaibi et al. (2004)** developed cost models for the thermoecomic analysis of refrigeration system. The efficiency of the main component of the system was chosen as the decision variable for the study. The paper showed the effect of variation of efficiency of the component on component cost. A program was made in MATLAB for the study of the system and its experimental validation of results was done.

**Vincent and Heun (2006)** applied exergetic and thermoeconomic analysis on a domestic refrigerator. Cost equations were formulated on the various main components of the refrigerator. The various parameters determined were exergy destruction, levelized electricity cost, levelized purchased cost, a variation of cost of exergy destruction with energy efficiency rating and exergoeconomic factor. The results showed that if we decreased the cost of the compressor without compromising in its efficiency then purchase cost and operating cost will be decreased.

**Selbas et al. (2006)** carried out the thermoeconomic analysis of refrigeration system using refrigerants R22, R134a and R407C. This paper determined optimized values of temperatures of subcooling & superheating and areas of evaporator & condenser for the variation of evaporator and condenser temperatures. Thermodynamic properties equations were formulated for selected refrigerants using an artificial neural network (ANN).

**Hepbasli (2007)** carried out EXCEM methodology for the exergetic and thermoeconomic analysis of domestic refrigerators. The worst component from exergy destruction point of view was a compressor and best component was superheating coil. With the rise in reference temperature, the value of exergetic efficiency was increased while exergy loss rate to capital cost ratio was decreased.

**Rosen (2008)** presented a review paper on the relation between exergy & economic and various methods that combine these two terms. This paper provided the importance of exergy based economic rather than energy base economics. The various methods discussed in the paper that combines exergy and economic analysis was exergy based cost allocation and pricing, EXCEM, loss-cost analysis, thermoeconomic & exergoeconomic and exergy & environmental economics.

**Hasanuzzaman et al. (2008)** conducted experiments and found that the different variable affects the energy consumption of a household refrigerator. Their results showed that there is a great influence of opening the door on energy consumption of the refrigerator.

**Khir et al. (2008)** presented the exergoeconomic cost optimization for the evaporator of a refrigerator. The objective functions for the optimization process taken were capital cost and exergy destruction cost. Hook and Jeeves method was applied for the analysis. The results showed that the optimum conditions in thickness, air velocity and tube diameter for obtaining lowest annual cost were 0.1 mm, 3 m/s and ½ inch respectively.

**Bereche et al. (2009)** compared the performance of direct fired and hot water driven system used in vapour absorption through thermoeconomic analysis. The exergy cost of the product was found lower for hot water driven system. It was observed from the results that the difference between exergetic costs in a single effect using direct fired and hot water driven system was higher as compared to direct fired in single effect and double effect steam driven system.

**Parekh et al. (2011)** optimized cascade vapour compression system based on refrigerant pair R404A-R508B. Thermoeconomic optimization of considered system results in a decreased value of investment cost by 19.71% whereas there was an improvement of 13.7634% and 16.20% in the COP and exergy efficiency respectively.

**Kabul (2011)** performed an analysis of the vapour compression system using various environment-friendly refrigerants. The methodology adopted for the thermoeconomic analysis was the coefficient of the structural bond. The analysis was performed for different temperature values of evaporator, condenser, subcooling and superheating. The results revealed that the lowest value of heat exchanger areas was obtained for R152a followed by R600A and R410A when operating at the same temperatures of heat exchangers.

**Rezayan and Behbahaninia (2011)** investigated the performance of the cascade vapour compression system using exergy and thermoeconomic analysis. The annual cost was selected as a variable which needs to be minimized for the optimization. The worst component from the exergy destruction point of view was condenser and best component was expansion device. Using optimized value, the annual cost was reduced to 9.34%.

**Sahebi and Motallebi (2011)** applied thermoeconomic and exergetic analysis in a heat pump. The optimization of the system was done using GEMS software. The system analysis was performed by using optimized values of the efficiencies of evaporator, compressor, electric motor and condenser were 80%, 80%, 90% and 84% respectively. In this paper, component costs were found as a function of efficiencies and price of electricity.

**Mitishita et al. (2013)** derived a simulation model to optimize a domestic refrigerator for minimizing the energy consumption and cost through thermoeconomic analysis. It was observed from the results that there was a reduction of 18% in energy consumption without compromising on cost. The difference between simulation and experimental results was not more than 9%. This paper also discussed the performance of the system using different types of compressors.

**Abbaspour and Saraei (2015)** carried out the thermoeconomic optimization of vapour absorption system. The genetic algorithm was used for the design optimization and it was found that the exergetic efficiency was improved with the increase in hot water temperature at the generator inlet. The optimization results in an increment of COP, exergetic efficiency and total cost of the plant were 74.5%, 47% and 12% respectively.

**Jain et al. (2015)** worked on the thermoeconomic analysis of cascade system working in combination with vapour compression and vapour absorption system. R410A was used in compression side and water-LiBr was used in absorption side. The system was designed for the water chilling application. The aim of the study was to reduce size and cost by using nonlinear programming. The results indicated that a reduction of 11.9% can be achieved with 22.4% less investment cost and these values were reduced with increased life span and operating time.

**Yildiz (2016)** compared the performance of diffusion absorption refrigeration system working on LPG and electricity. It was found that energy and exergy input in LPG system was more than that of electricity although both were having nearly the same values of COP and exergetic efficiency. Thermoeconomic cost of LPG operated system was nearly 64% more than the other system. Thus, the electric system performs better than LPG system.

**Khatwani and Maheshweri (2016)** developed a software TAESS for the thermoeconomic analysis of water-Libr based vapour absorption system. Fuel product table was used for the cost formation and diagnosis. In fuel product, table row represents product and column represents the fuel.

**Yoo et al. (2018)** performed a thermoeconomic diagnosis of an air conditioning system. Modified productive structure methodology was applied to diagnose the system. The fouling at the heat exchanger and leakage from compressor valve were determined due to malfunction and lost cost due to normal and real operation. It was revealed from the results that the cost of producing refrigeration effects will increase whenever there was an occurrence of a malfunction in the component of the selected system.

## 2.5 RESEARCH GAPS

From the literature survey related to the different alternative refrigerants of CFC12, it can be revealed that the most successful refrigerant which is used in most of the practical application in refrigerator throughout the world is R134a. It is found in the literature review that HFC134a contributes to global warming. Hence, there is a need for finding an alternative to HFC. The alternatives of the HFC that were found from literature review are HFC152a, HC290, HC600a, R403A, R404A, R407C, R410A, HFO1234yf, HFO1234ze(E), mixtures of HCs, mixtures of HFC/HC, mixtures of HFOs and mixtures of HFC/HFO. The different protocol found in the literature review regarding the banning of CFC, HCFC and HFC refrigerants:

- Montreal Protocol: Banning of CFC and HCFC refrigerants which contribute to ozone layer depletion
- Kyoto Protocol: Prohibited the use of HFC refrigerants as they contribute to global warming
- Kigali amendment under Montreal Protocol: Prohibited the use of HFC refrigerants as they contribute to global warming

The future alternative of R134a from research point of view lies in the field of mixtures of HFC/HFO/HC but a limited work is found in the literature review with regard to exergy analysis and thermoeconomic analysis.

From the literature review, the importance of the use of exergy analysis in the design of thermal was revealed in the comparison of energy analysis. The exergy analysis of various alternatives of HFC in vapour compression system was performed. The various terms used in exergy analysis were exergetic destruction, exergy loss, exergetic efficiency, efficiency defect and exergy destruction ratio. But limited research is found in the exergy analysis of vapour compression system using mixtures of HFC/HFO for reducing GWP.

The literature reviews of thermoeconomic analysis emphasis its importance in the overall design of the vapour compression system. The methodology of thermoeconomic analysis is used for cost minimization and improving the performance of the system. The various technique of thermoeconomic analysis was reviewed but limited work has been done in the field of thermoeconomic analysis of vapour compression system especially using mixtures of refrigerants for reducing GWP.

## 2.6 SUMMARY

Currently used alternative refrigerant for HFC is HC (hydrocarbon) which has a very low value of GWP. The flammability issue related with HC refrigerant has put a cap on its use and further research move towards HFO (hydrofluoroolefin) which has similar GWP but are less flammable as compared to HC. The presence of a double bond in HFO refrigerants (R1234yf and R1234ze) imparts a property of quicker breakdown in the atmosphere, yet stable in the system. Although HFO1234yf is better than hydrocarbons still they are mild flammable and much costlier in comparison to R134a. Thus, further approach moves towards the refrigerant mixtures like mixtures of HCs, mixtures of HFC/HC, mixtures of HFOs and mixtures of HFC/HFO. It can be concluded from the literature review that the limited work has been done regarding thermoeconomic analysis with alternatives of R134a and especially using HFOs based low GWP refrigerant mixtures. The detailed analysis of refrigerants selected for finding the alternatives of HFC134a with energy efficiency, low GWP and thermoeconomic consideration is needed.

# CHAPTER 3
# ANALYSIS OF THERMAL SYSTEM

## 3.1 INTRODUCTION

This chapter explains the basic concepts and methodology for the finding the eco-friendly thermoeconomic refrigerant. The refrigeration system uses refrigerants which possess high value of GWP and non zero value of ODP. The ozone depletion and global warming are the major concern from the environmental point of view. The reason behind high ODP in refrigerants is due to the presence of chlorine and bromine content in them thus, after 90's use of CFC refrigerants had been restricted. HFC refrigerants used nowadays posses zero ODP and lower GWP in comparison to CFC but still much higher as per MAC directive (2006/40/EG). Maintaining a high COP was not as important at that time, because energy prices were relatively low. Today, high COP is much more important for two reasons, first is overall energy prices are considerably higher than during the last refrigerant change and another is COP is indirectly affecting GWP. COP is the ratio of cooling effect to the net work given to the system. Thus, it can be improved either by enhancing the cooling effect or by minimizing work input given to the system.

## 3.2 RESEARCH METHODOLOGY

The objectives of the present work are as follows:

- Refrigerant R134a contributes to global warming but do not cause any harm to the ozone layer. Therefore, from an environment point of view, there is a need to find an alternative to it.
- Different alternate refrigerants will be compared using approaches of conventional thermodynamic and thermoeconomic analysis of vapour compression system.
- Finding of suitable alternate refrigerant.

The following methodology has been adopted stepwise for achieving the above objectives:

Exhaustive literature survey, finding of gaps in the literature survey and the existing system models were analyzed. The refrigerants data were collected, safety aspects were studied and the refrigerant mixtures were analyzed using REFPROP software. A detailed exergy analysis was done for internal irreversibility to calculate exergy destruction and analysis of exergy loss at the component level of the energy conversion system. At the component level of an energy conservation system, the thermoeconomic analysis was conducted to analyse exergy costing and exergoeconomic evaluation of each system component. Further, an experimental investigation was carried for finding alternate refrigerants mixtures in place of the existing refrigerant R134a. Finally, the results were validated and suitable alternate refrigerant with low GWP was determined. This is explained in detail in further sections of this chapter.

**3.3 ENERGY, EXERGY AND THERMOECONOMIC ANALYSIS**

Energy is defined as the capacity for doing work. Energy exists in different forms like thermal energy, motion energy, chemical energy etc. First law of thermodynamics gives the principle of conservation of energy. It states that energy can neither be created nor be destroyed but one form of energy can be converted into another form of energy. In thermal system, energy analysis is performed in order to find out its performance and comparing it with another system. It plays a vital role in designing and development of the thermal system. It describes all types of energy transfer in the evaluation of operating parameters. In energy analysis of refrigeration systems various terms used for measuring the performance are as follows:

- Coefficient of Performance (COP)
- Energy Efficiency Ratio (EER)
- Refrigerating Effect
- Energy Input

In order to improve the performance of refrigeration system researchers always tries to increase the value of the coefficient of performance, energy efficiency ratio, refrigerating effect and to decrease the value of energy input.

The main deficiency with energy analysis is that energy can neither be created nor destroyed but we want a quantity that can be destroyed for measuring the quantity and quality of energy. The term that can be destroyed is exergy. Basically, it is a concept of the second law of thermodynamics which states that 100% energy conversion is not possible. Exergy is associated to incoming and outgoing stream in the system. Exergy analysis is very important in order to determine the behaviour of system, especially at the component level. It also has an application in comparing the performance of various systems. It tells us which component required more attention so that overall performance of the system can be increased. Through exergy analysis, we can find the irreversibility within the system. The various terms associated with exergy analysis are as follows:

- Exergetic Efficiency
- Exergy Destruction
- Exergy Loss
- Efficiency Defect
- Exergy Destruction Ratio

The performance of the system will be improved if we increase the value of exergetic efficiency and decrease the value of exergy loss, exergy destruction and its ratio at the component level or overall system.

The complete behaviour of a thermal system can be understood if we apply exergy analysis and engineering economics. As true thermodynamic values are measured by using exergy so, it makes sense to use exergy for allocating cost to the system. Thermoeconomic is the word used to the define combination of exergy and economic principles. In thermoeconomic analysis, each exergy term is assigned with a cost. Through thermoeconomic analysis, researchers will be able to understand cost formation process in the system. The main aim of thermoeconomic analysis is to minimize the whole cost (i.e. capital cost and operating cost) of the system and improvement in the performance of the system. Thermoeconomic analysis helps the researchers in determining the components which need more attention to improve the performance of the system. The various terms associated with the thermoeconomic analysis are as follows:

- Cost of Exergy Destruction
- Cost of Exergy Loss
- Levelized Electricity Cost
- Relative Cost Difference
- Cost Importance
- Exergoeconomic Factor

## 3.4 FORMULAS IN ENERGY, EXERGY AND THERMOECNOMIC ANALYSIS

The various formulas used for the analysis work in the present thesis are as follows:

### 3.4.1 Refrigerating Effect

It is stated as the amount of heat extracted by the refrigerant from the space to be cooled i.e. evaporator. From the performance point of view higher value of refrigerating effect is required.

$$\text{Refrigerating Effect} = \text{Mass of Refrigerant} \times \text{Change in Enthalpy of Refrigerant} \qquad (3.1)$$

### 3.4.2 Work Input

It is defined as the amount of work supplied by the compressor in order to increase the pressure and temperature of the refrigerant. From the performance point of view lower value of work input is required.

$$\text{Work Input} = \text{Mass of Refrigerant} \times \text{Change in Enthalpy of Refrigerant} \qquad (3.2)$$

### 3.4.3 Coefficient of Performance

It indicates the performance of the refrigerating machine. Higher the value of the coefficient of performance better performance of system will be. It is stated as the ratio of amount of heat extracted by the refrigerant from low temperature reservoir to the work input into the refrigeration system.

$$\text{Coef icient of Performance (COP)} = \frac{\text{Refrigeration Effect}}{\text{Work Input}} \qquad (3.3)$$

It is a dimensionless quantity. A system will require less work input if it has a higher value of COP. Sometimes, energy efficiency ratio is used in place of the coefficient of performance. Energy Efficiency Ratio (EER) is defined as the ratio of refrigerating effect to the input electrical energy supplied.

$$\text{Energy Ef iciency Ratio (EER)} = \frac{\text{Refrigeration Effect}}{\text{Input Electrical Energy}} \qquad (3.4)$$

### 3.4.4 Exergy

It is defined as the maximum possible reversible work obtainable in bringing the state of the system to equilibrium with that of the environment. Exergy is an extensive property. When the system is considered to be at rest relative to the environment and magnetic, electrical, nuclear and surface tension effects are absent then the total exergy of a system ($\dot{E}$) can be grouped into two components physical exergy ($\dot{E}^{Ph}$) and chemical exergy ($\dot{E}^{Ch}$).

$$\dot{E} = \dot{E}^{Ph} + \dot{E}^{Ch} \qquad (3.5)$$

The physical exergy component is associated with the work obtainable in bringing a stream of matter from its initial state to a state that is in thermal and mechanical equilibrium with the environment. Mathematically,

$$\dot{E}^{Ph} = \dot{m}\left[\left(h - h_0\right) - T_0\left(s - s_0\right)\right] \tag{3.6}$$

where $\dot{m}$, $h$, $s$ and $T$ denotes mass flow rate (kg $s^{-1}$), specific enthalpy (kJ $kg^{-1}$), specific entropy (kJ $kg^{-1}$) and temperature at the specified state whereas $h_0$, $s_0$ and $T_0$ are the values of the same properties when the system is at the restricted dead state respectively.

### 3.4.5 Exergy Destruction

It indicates the irreversibility within the system. It is a measure of the amount of exergy destroyed within the system. Overall, in a control volume the steady-state exergy balance can be used to express exergy destruction as:

$$\dot{E}_D = \Sigma\dot{E}_{in} - \Sigma\dot{E}_{out} - \dot{Q}\left(1 - \frac{T_0}{T}\right) - \dot{W} \tag{3.7}$$

where $\dot{E}_{in}$, $\dot{E}_{out}$, $\dot{Q}$, $\dot{W}$ and $\dot{E}_D$ denotes the exergy flow rate of stream entering a component or system (kW), exergy flow rate of stream leaving a component or system (kW), heat flow rate (kW), work flow rate or power (kW) and exergy destruction (kW) respectively.

Exergy loss is the amount of exergy loss from the system to the surrounding. It represents the external irreversibility. The relationship between exergy of fuel, exergy of product, exergy destruction and exergy loss can be described as follows:

$$\text{Exergy of Fuel} = \text{Exergy of Product} + \text{Exergy Destruction} + \text{Exergy Loss} \tag{3.8}$$

### 3.4.6 Exergy Destruction Ratio

It is the measure of the inefficiency of the system. It is defined as the ratio of the exergy destruction of a system to the exergy of the fuel provided to the overall system.

$$\text{Exergetic Destruction Ratio (EDR)} = \frac{\text{Exergy Destruction}}{\text{Exergy of Fuel provided to the Overall System}} \tag{3.9}$$

### 3.4.7 Exergetic Efficiency

It indicates the true measure of the performance of a refrigeration system. A system will perform better if it has a higher value of exergetic efficiency. It is also known as second law of efficiency. Before describing the exergetic efficiency there is a need to define fuel and product of the system under consideration.

$$\text{Exergetic Ef iciency} = \frac{\text{Exergy of Product}}{\text{Exergy of Fuel}} \tag{3.10}$$

It shows how much percentage of fuel exergy is found in product exergy. It is a dimensionless quantity.

### 3.4.8 Cost Balance

The cost rate balance for $k^{th}$ component of a refrigeration system using thermoeconomic analysis can be determined out by using cost rate associated with the fuel, product and exergy loss as follows:

$$\dot{C}_{P,k} = \dot{C}_{F,k} - \dot{C}_{L,k} + \dot{Z}_{k} \tag{3.11}$$

$$c_{P,k} * \dot{E}_{P,k} = (c_{F,k} * \dot{E}_{F,k}) - \dot{C}_{L,k} + \dot{Z}_{k} \tag{3.12}$$

where $\dot{C}_{P,k}$, $\dot{C}_{F,k}$, $\dot{C}_{L,k}$, $\dot{Z}_{k}$, $c_{P,k}$, $c_{F,k}$, $\dot{E}_{P,k}$ and $\dot{E}_{F,k}$ denotes the cost rate of product ($/s), cost rate of fuel ($/s), cost rate of exergy loss ($/s), sum of capital and operation & maintenance cost of component ($/s), average unit cost of product ($/kJ), average unit cost of fuel ($/kJ), exergy rate associated with product (kW) and exergy rate associated with fuel (kW) respectively.

### 3.4.9 Cost of Exergy Destruction

It is a hidden cost. There is no cost term directly related to exergy destruction. A component having a high value of cost of exergy destruction needs attention for improving its exergetic efficiency. It is given for $k^{th}$ component as follows:

- When exergy rate ($\dot{E}_{P,k}$) associated with the product is fixed

$$\dot{C}_{D,k} = c_{F,k} * \dot{E}_{D,k} \tag{3.13}$$

- When exergy rate ($\dot{E}_{F,k}$) associated with fuel is fixed

$$\dot{C}_{D,k} = c_{P,k} * \dot{E}_{D,k} \tag{3.14}$$

Similarly, we can determine the cost of exergy loss as:

- When exergy loss is covered through the supply of additional fuel $\dot{E}_{F,k}$

$$\dot{C}_{L,k} = c_{F,k} * \dot{E}_{L,k} \tag{3.15}$$

- When exergy loss results in reduction of product $\dot{E}_{P,k}$

$$\dot{C}_{L,k} = c_{P,k} * \dot{E}_{L,k} \tag{3.16}$$

### 3.4.10 Cost of Operating

It is the cost associated with the production of refrigeration effect over the lifetime of the refrigerator. From an economical point of view, refrigerator must have a low value of the cost of operating. It is given by following formulae

$$Cost_{operating} = \sum \left( Cost_{annual}[i] * 1[yr], i = 1, n \right) \tag{3.17}$$

$$Cost_{annual}[i] = C_{electricity}[i] * \dot{W}_{elec} * cylcle_{duty} \tag{3.18}$$

$$C_{electricity}[i] = C_{electric} * (1 + Inflation * (i - 1)) \tag{3.19}$$

where n, $\dot{W}_{elec}$, $cylcle_{duty}$ and $C_{electric}$ denotes the system life, power consumption, run time of compressor per year and electricity cost respectively.

### 3.4.11 Levelized Electricity Cost

It levelize the escalation in electricity costs over the lifetime of the refrigerator to an annual constant value.

$$C_{electric,\ L} = \frac{\left( \left( Cost_{annual}[1] * 1[yr] \right) * A / P_0 \right)}{1 + Inflation} \tag{3.20}$$

where A is annuity and $P_0$ is the constant value at the beginning of the first year.

### 3.4.12 Relative Cost Difference

It indicates the relative increases in the average cost per exergy unit between fuel and product of the component. It is very useful in evaluating and optimizing a system component. For k$^{th}$ component of refrigeration system it is termed as follows:

$$r_k = \frac{c_{P,k} - c_{F,k}}{c_{F,k}} \tag{3.21}$$

### 3.4.13 Exergoeconomic Factor

It is the ratio of the contribution of the non-exergy related cost to the total cost increase. A higher value of exergoeconomic suggests that less money could be spent on the component at the expense of exergetic efficiency whereas its low value suggests that cost saving of the overall system can be achieved by increasing capital investment on the component for increasing its exergetic efficiency. For k$^{th}$ component of refrigeration system the exergoeconomic factor is defined as follows:

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + \left( c_{F,k} * \left( \dot{E}_{D,k} + \dot{E}_{L,k} \right) \right)} \tag{3.22}$$

### 3.4.14 Cost Importance

It is defined as the cost rate associated with non-exergy as well as exergy the related term. It is one of the most important terms used in the thermoeconomic analysis. A component which has the largest value of cost importance needs the highest attention for the improvement. For k$^{th}$ component of refrigeration system the cost importance is defined as follows:

$$ZC_k = \dot{Z}_k + \dot{C}_{D,k} \tag{3.23}$$

### 3.5 THERMOECONOMIC EVALUATION OF REFRIGERATION SYSTEM

The cost-effectiveness of various components of a thermal system can be improved by applying thermoeconomic evaluation. The methodology of thermoeconomic evaluation can be understood with the help of the following procedure:

- Arrange the components in a declining sequence of cost importance.
- Components with highest value of cost importance will be considered first for the design change.
- Component having larger value of relative cost difference required particular attention especially when the cost of exergy destruction and sum of cost related to capital investment and operation & maintenance are high.
- The major cost source can be identified using exergoeconomic factor and its value has following meaning.
  a) If the value of exergoeconomic factor for a component is high then the system can be made cost-effective by reducing the capital investment of the component at the expense of exergetic efficiency.
  b) If the value of exergoeconomic factor for a component is low then there is a requirement of expanding more on the capital investment of the component to improve its exergetic efficiency.
- Normal values of an exergoeconomic factor are already predefined for components such as below 55% for heat exchangers, between 35-75% for compressors and turbines.
- Subprocesses which are responsible for increasing the value of exergy destruction or loss and not contributing towards a reduction in fuel costs or capital investment of other components then it needs to be eliminated.
- If a component is having a low value of exergetic efficiency or high value of exergy destruction or loss then it needs to improve.

### 3.6 FUEL PRODUCT LOSS (F-P-L) DEFINITION

For the analysis of thermal system by using thermoeconomic it is important to identify fuel product loss definition for the system. The resources utilized for the generation of product is defined as fuel and it is not necessary that it means to be an actual fuel like oil, coal etc. The desired result obtained from the system is defined as a product. Exergy is used to express the fuel and product whereas loss represents exergy loss of the system. The guidelines for identifying fuel product loss definition are as follows:

- Identifying the fuel as the sum of all exergy inputs and the product as the sum of all exergy outputs can result in misleading conclusions for single plant components.
- The definition of exergy efficiency should be meaningful for both the thermodynamic and economic viewpoints. The purpose of owning and operating a component determines the product of a thermal system.
- When a stream crosses the boundary of a system twice with no change in chemical composition, generally only the difference in the exergy values of the stream should be considered in the calculation of the fuel or product. That is, the net exergy supplied by such a stream would be identified with the fuel and net exergy supplied to such a stream would be identified with the product.
- While evaluating the overall system efficiency, exergy losses associated with material streams should be considered and it is not required while evaluating the efficiencies of the system components.
- Special consideration should be given to the components like throttling valves and heat exchangers (coolers) because a product is not readily defined when such components are considered singly.

### 3. 7 PROPERTIES OF SELECTED REFRIGERANTS

Refrigerants considered from the literature review for finding the substitute of HFC134a in the present thesis work are HFC152a, HC290, HFC600a, HFO1234yf, HFO1234ze(E), mixture of HCs and mixtures of HFC/HFO. The main refrigerant properties from the environmental way of looking are ODP, GWP and atmospheric life. Properties other than environment related to refrigerants are toxicity, flammability, latent heat of vaporization, normal boiling point, molecular weight, specific volume, and critical temperature etc. All these properties of considered refrigerants are presented (Table 3.1). The alternative refrigerant which we are searching must have zero ODP, low GWP value than R134a, non toxic, non-flammable, economical and other properties comparable with that of R134a.

Table 3.1 Properties of considered refrigerants [33, 74 and 124]

| Characteristics | HFC 134a | HFC 152a | HFC 290 | HFC 600a | HFO 1234yf | HFO 1234ze |
|---|---|---|---|---|---|---|
| Chemical Formula | $CH_2FCF_3$ | $CH_3CHF_2$ | $CH_3CH_2CH_3$ | $(CH_3)_3CH$ | $CH_2$=$CFCF_3$ | CHF=$CHCF_3$ |
| Molecular Weight [g/mol] | 102.03 | 66.05 | 44.10 | 58.12 | 114.04 | 114.04 |
| Normal Boiing Point [°C] | -26.10 | -24.00 | -42.10 | -11.70 | -29.50 | -19.00 |
| Critical Temperature [°C] | 101.10 | 113.30 | 96.40 | 134.70 | 94.70 | 109.40 |
| Latent Heat at -30°C [kj/kg] | 219.50 | 335.20 | 412.40 | 380.72 | 180.51 | 201.5 |
| Latent Heat at +40°C [kj/kg] | 163.00 | 260.00 | 307.10 | 311.52 | 132.27 | 154.60 |
| Specific Volume at -30°C [$m^3$/kg] | 0.2259 | 0.3824 | 0.2586 | 0.7284 | 0.1708 | 0.2817 |
| GWP (100 yrs) | 1430 | 140 | 11 | 11 | 4 | 6 |
| ODP | 0 | 0 | 0 | 0 | 0 | 0 |
| Atmospheric Life (yr) | 13.40 | 1.50 | 0.041 | 0.016 | 0.029 | 0.045 |
| OEL (Occupational Exposure Limit) [PPMv] | 1000 | 1000 | 1000 | 1000 | 500 | 1000 |
| LEL (Lower Flammability Limit) [%] | None | 4.8 | 2.1 | 1.6 | 6.2 | 7.6 |
| HOC (Heat of Combustion) [Mj/kg] | 4.2 | 17.4 | 50.4 | 49.4 | 10.7 | n.d. |

It may be noted that the refrigerant R1234yf ( saturated vapour density at -10.6$^0$C=12.296 Kg/$m^3$) has 25% more mass than R134a (saturated vapour density at -10.6$^0$C=9.816 Kg/$m^3$) but it may not produces an equivalent increase in cooling capacity because the latent heat of vaporization of the refrigerant R1234yf is about 18% lower than that of the R134a.

### 3.8 SAFETY ASPECT OF REFRIGERANTS

As per ANSI/ASHRAE standard [11] regarding the safety of the refrigerants and collecting values of GWP from published papers, the various important characteristics are shown (Table 3.2 and Table 3.3). At present refrigerant classified as A2 and A2L, requires appropriate design measures in order to ensure their safe use in the refrigeration system.

Table 3.2 Refrigerants safety groups

| Flammability | Toxicity | |
|---|---|---|
| | Lower Toxicity | Higher Toxicity |
| No Flame Propagation | A1 | B1 |
| Mild Flammable | A2L | B2L |
| Lower Flammability | A2 | B2 |
| Higher Flammability | A3 | B3 |

Table 3.3 GWP of refrigerants along flammability and toxicity

| Refrigerant | GWP | Flammability | Remarks |
|---|---|---|---|
| R134a | 1430 | A1 | Safe to use |
| R1234yf | 4 | A2L | Mildly flammable |
| R1234ze | 6 | A2L | Mildly flammable |
| R152a | 140 | A2 | Flammable |
| R600a | 11 | A3 | Highly Flammable |
| HFC134a/HFO1234yf/HFO1234ze (40%/22%/38%) | Approx. 610 | A1 | Safe to use |
| R1234yf /R134a (90%/10%) | Approx. 133 | A1 | Safe to use |
| R134a/R600a/R290 (91%/4.93%/4.07% ) | Approx. 1310 | A1 | Safe to use |
| R600a/R134a (91%/9%) | Approx. 138 | A3 | Highly Flammable |

Refrigerant R1234yf has global warming potential (GWP) of 4, so it satisfies MAC Directive (GWP below 150) passed in July 2006. R1234yf has nearly similar value of molecular weight and normal boiling point, making R1234yf a good replacement of R134a. The slight flammability of R1234yf (A2L safety) can be overcome with safety design measures. The low GWP refrigerants R1234yf are promising candidates for replacing conventional HFC refrigerants but it is costly. To eliminate flammability issue and to ensure good performance with GWPs in range of 0 to 1000, refrigerant manufacturers are looking at different mixtures of HFO and HFC.

Since R152a has higher flammability concern than R1234yf, so it has not been much discussed in the present work although GWP of R152a is also lower than R134a. The same refrigeration system is adapted with limited modification of the expansion valve while replacing R134a with R1234yf. While heat exchanger process and design development appear to be less critical, the main concern of flammability is with the compressor.

A key question to be clarified is that if the flammability of the refrigerants can be suppressed by R744. It is known that R744 ($CO_2$) is a good flame suppressant. One pressurized gas container with R744 may be fitted near compressor which will automatically explode in case of accidents or any rupture to supply R744.

Environmental impact of fluid shall be discussed along with flammability and safety. Two main conditions which are responsible for happening of accidents are a flammable mixture of air & gas and ignition source at certain energy level and temperature. As a safety precaution, maximum refrigerant charge is set to be 150 g and ignition risk is very low for approx. 8 g/m$^3$, for a standard kitchen [57]. R600a is used in small quantities in refrigerator (30-70 g); therefore, it must be used with safety precautions.

### 3.9 EXPERIMENTATION

A conventional two chamber household refrigerator with internal cabin volume 190 litre and selected refrigerant/refrigerant mixtures was charged for the present study (as shown in Figure 3.1). AS/NZS test standard was used with ambient temperature 32$^\circ$C and relative humidity 65-68%. The experimental set-up comprised of a refrigerator, thermocouple and energy meter. Energy consumption was measured by the energy meter with an accuracy of ±0.2% of reading.

Figure 3.1 Domestic refrigerator test rig

The experiments were done on test rig (as shown in Figure 3.2) for measuring cooling capacity and work input. Equipment used for measuring cooling capacity were energy meter for evaporator heater, temperature sensor, PID glycol temperature controller, evaporator glycol tank of 40 litres with ethylene glycol 8 litres. Capillary tube (0.05 inch diameter x 5 ft length x 4 number) was used as an expansion device. When the test rig is operating, glycol gets cooled and an equivalent amount of heat is supplied through heaters. The heat energy supplied is equivalent to cooling capacity and is recorded by the difference in the energy meter reading.

Figure 3.2 Refrigerator test rig

$E_1$ is initial energy meter reading and $E_2$ is final energy meter reading, then ($E_2$-$E_1$) is the difference in the energy meter reading and corresponds to cooling capacity. Work input to the compressor was recorded by the difference in energy meter reading for achieving the cooling capacity in the evaporator consisting of glycol. $E_3$ is initial energy meter reading for work input in the compressor and $E_4$ is final energy meter reading for work input in the compressor, then ($E_4$-$E_3$) is the difference in the energy meter reading and corresponds to compressor work input. The error of the power input data and cooling capacity measuring was ± 4 %.

### 3.9.1 Refrigerant Mixtures Preparation

The cylinders were cleaned and flushed with R134a thrice. Refrigerant R134a is about 30 times cheaper than R1234yf therefore, it is appropriate to use R134a for flushing. Cylinders were evacuated to a pressure of 0.2 mbar. To avoid contamination in cylinders while filling they usually kept in a low temperature bath. Initially, the appropriate amount of HC refrigerant was used to fill in the cylinder and a weighing balance was used having an accuracy of 0.1 g. To maintain the required mass percentage in the total filled quantity the required mass of R 134a was calculated and filled. To have an accurate charging by weight, the mixtures were prepared in small cylinders with one Kg capacity and each cylinder was properly labelled. The mixture quantity was prepared sufficiently to maintain the 10% level and it should not fall below 10% level while charging the system.

### 3.9.2 Material Compatibility

Majurin et al. (2014) conducted the material compatibility exposures tests in Parr pressure vessels with R1234yf, with seals and polymers, which included elastomers, flat gaskets, polymeric materials and motor materials, which incorporated articles used in hermetic compressor motors such as phase insulations, motor varnishes, magnet wires, a tie cord, and a polymeric connector block material. Exposures were conducted in 100% refrigerant, 50% refrigerant: 50% lubricant, and 100% lubricant to encompass the range of refrigerant and lubricant compositions that may be present in different areas of operating systems. Commonly used lubricant, a polyolester (POE) was evaluated in the study. The study suggests that many of the seal and structural polymer materials currently used are suitable for use with next generation low GWP refrigerants R1234yf.

Rohatgi et al. (2012) conducted a two phase study to determine the thermal and chemical stability of HFO1234yf with lubricants and their long-term material compatibility with motor materials commonly used in stationary air-conditioning and refrigeration systems. With HFO1234yf, refrigerant decompositions were small (< 100 ppm fluoride ions) in ISO 32 mixed and branched acid lubricants when there was no air present.

### 3.10 SUMMARY

This chapter explains various steps, techniques for finding the eco-friendly thermoeconomic refrigerant. It is clear that all considered alternative refrigerants are non toxic, have negligible GWP and zero ODP but these are flammable (varying from A2L-A3) and costly (especially HFO1234yf and HFO1234ze) as compared to R134. Thus, refrigerant mixtures HFC/HFO and HCs are also discussed in the present thesis. It may be concluded that working fluid selection for the refrigeration and air conditioning applications is based on the factors such as flammability and safety, environmental impact (ODP and GWP), energy efficiency and cost effectiveness.

# CHAPTER 4

# ENERGY AND EXERGY ANALYSIS OF VAPOUR COMPRESSION SYSTEM USING HFCs, HCs, HFOs AND THEIR MIXTURES

## 4.1 INTRODUCTION

Due to the Kyoto Protocol and Kigali amendment under the Montreal Protocol, HFC refrigerants are going to be phased out in the coming years because they contribute to global warming. Thus, there is a requirement to replace R134a. To find a drop-in replacement of R134a, a simulation analysis of vapour compression system is represented in this section. As per ISI Code No. 1476-1979, Figure 4.1 shows a standard ten-state-points cycle for testing refrigerators [22]. The ten-state-points cycle is already discussed in Chapter 1 of the present thesis. Different parameters calculated are pressure ratio, mass flow rate, relative volumetric cooling capacity, relative coefficient of performance, cooling capacity, exergetic efficiency, exergy destruction and efficiency defect. For the present analysis, refrigerator has a 0.04 compressor clearance ratio and 6.64 $cm^3$ stroke volume [47]. A total of thirty refrigerants are selected as shown in Table 4.1 for finding a substitute for R134a. The refrigerants in Table 4.1 include HFC, HC, HFO, mixtures of HFC/HFO and HCs.

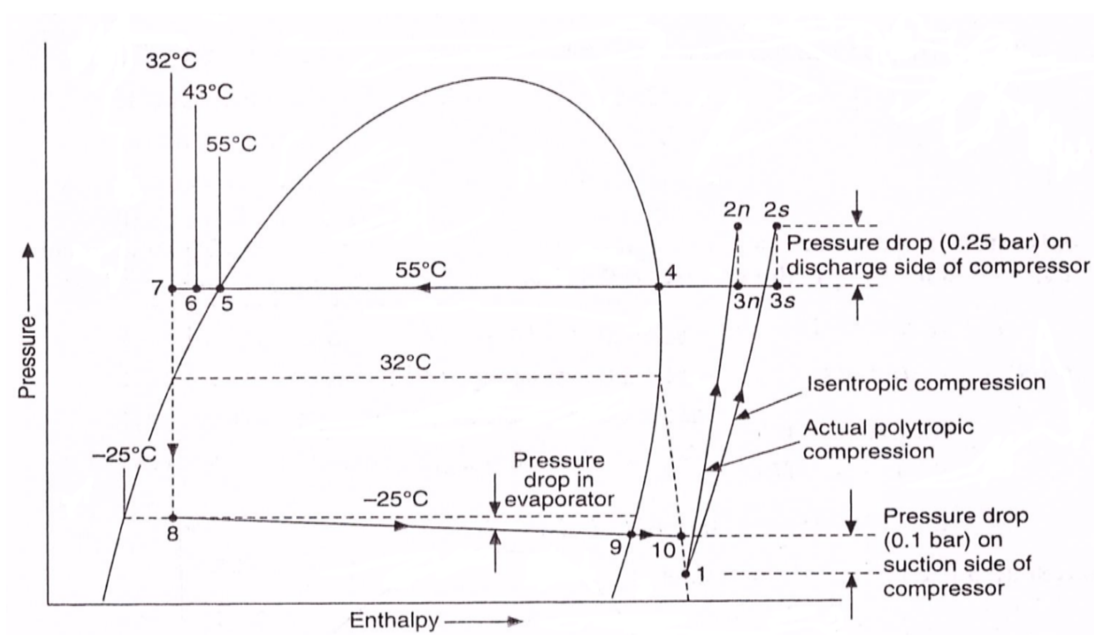



Figure 4.1 Standard ten state point cycle

Table 4.1 Properties of all the selected refrigerants [33, 124]

| Sr. No. | Refrigerant | Naming | Type | GWP (100 yrs) | ODP | Safety Group |
|---|---|---|---|---|---|---|
| 1 | R134a | R1 | HFC | 1430 | 0 | A1 |
| 2 | R152a | R2 | HFC | 140 | 0 | A2 |
| 3 | R290 | R3 | HC | 11 | 0 | A3 |
| 4 | R600a | R4 | HC | 11 | 0 | A3 |
| 5 | R1234yf | R5 | HFO | 4 | 0 | A2L |
| 6 | R1234ze | R6 | HFO | 6 | 0 | A2L |
| 7 | R134a/R1234yf (90%/10%) | R7 | Blend | 1287.4 | 0 | A1 |
| 8 | R134a/R1234yf (80%/20%) | R8 | Blend | 1144.8 | 0 | A1 |
| 9 | R134a/R1234yf (70%/30%) | R9 | Blend | 1002.2 | 0 | A1 |
| 10 | R134a/R1234yf (60%/40%) | R10 | Blend | 859.6 | 0 | A1 |
| 11 | R134a/R1234yf (50%/50%) | R11 | Blend | 717 | 0 | A1 |
| 12 | D-4y [R134a/R1234yf (40%/60%)] | R12 | Blend | 574 | 0 | A1 |
| 13 | R134a/R1234yf (30%/70%) | R13 | Blend | 431.8 | 0 | n.d. |
| 14 | R134a/R1234yf (20%80%) | R14 | Blend | 289.2 | 0 | n.d. |
| 15 | R134a/R1234yf (10%/90%) | R15 | Blend | 146.6 | 0 | n.d. |
| 16 | XP-10 [R134a/R1234yf (44%/56%)] | R16 | Blend | 631 | 0 | A1 |
| 17 | N-13a [R134a/R1234yf/R1234ze (42%/18%/40%)] | R17 | Blend | 604 | 0 | A1 |
| 18 | N-13b [R134a/R1234ze (42%/58%)] | R18 | Blend | 604 | 0 | A1 |
| 19 | ARM-41a [R32/R134a/R1234yf (6%/63%/31%)] | R19 | Blend | 943 | 0 | A1 |
| 20 | R134a/R1234yf/R1234ze (40%/5%/55%) | R20 | Blend | 575.5 | 0 | A1 |
| 21 | R134a/R1234yf/R1234ze (40%/10%/50%) | R21 | Blend | 575.4 | 0 | A1 |
| 22 | R134a/R1234yf/R1234ze (40%/15%/45%) | R22 | Blend | 575.3 | 0 | A1 |
| 23 | R134a/R1234yf/R1234ze (40%/20%/40%) | R23 | Blend | 575.2 | 0 | A1 |
| 24 | R134a/R1234yf/R1234ze (40%/25%/35%) | R24 | Blend | 575.1 | 0 | A1 |
| 25 | R134a/R1234yf/R1234ze (40%/30%/30%) | R25 | Blend | 575 | 0 | A1 |
| 26 | R134a/R1234yf/R1234ze (40%/35%/25%) | R26 | Blend | 574.9 | 0 | A1 |
| 27 | R134a/R1234yf/R1234ze (40%/40%/20%) | R27 | Blend | 574.8 | 0 | A1 |
| 28 | R134a/R1234yf/R1234ze (40%/45%/15%) | R28 | Blend | 574.7 | 0 | A1 |

| | | | | | | |
|---|---|---|---|---|---|---|
| 29 | R134a/R1234yf/R1234ze (40%/50%/10%) | R29 | Blend | 574.6 | 0 | A1 |
| 30 | R134a/R1234yf/R1234ze (40%/55%/5%) | R30 | Blend | 574.5 | 0 | A1 |
| 31 | R134a/R1234yf/R1234ze (40%/22%/38%) | R31 | Blend | 575.16 | 0 | A1 |

### 4.2 IMPORTANT PERFORMANCE CHARACTERISTICS

To find an alternative of R134a, few specific performance characteristics needed to consider which are as follows:

#### 4.2.1 Volumetric Cooling Capacity ($Q_{vol}$)

It is the measure of the size of the compressor. With the increase in the value of volumetric cooling capacity, the size of the compressor also increases. It is given by the following formula:

$$Q_{vol} = \frac{(h_9 - h_8) * \eta_{vol,cl}}{v_{10}} \quad (4.1)$$

where $h_8$ and $h_9$ are the specific enthalpy of the refrigerant at the evaporator inlet & exit respectively and $v_{10}$ is the specific volume of the refrigerant at the compressor inlet. The clearance or ideal volumetric efficiency $\eta_{vol,cl}$ is given by

$$\eta_{vol,cl} = 1 - CCR(PR^{1/n} - 1) \quad (4.2)$$

where $n$ is the polytropic index, PR is the pressure ratio and CCR is the compressor clearance ratio.

$$PR = \frac{P_{2n}}{P_1} \quad (4.3)$$

where $P_1$ and $P_{2n}$ are vapour pressure of refrigerant before compression begins and after compression ends respectively.

#### 4.2.2 Relative Volumetric Cooling Capacity

It is described as the ratio of volumetric cooling capacity of any refrigerant to the volumetric cooling capacity of baseline refrigerant i.e. R134a. It compares the volumetric cooling capacity of any refrigerant with respect to volumetric cooling capacity of R134a. It is a dimensionless quantity. Mathematically, it is given as

$$Q_{vol,rel} = \frac{Q_{vol,any\ ref}}{Q_{vol,R134a}} \quad (4.4)$$

### 4.2.3 Coefficient of Performance

$$COP = \frac{Q_{evap}}{W_{comp}} \tag{4.5}$$

where $Q_{evap}$ is the cooling capacity and $W_{comp}$ is the power input to the compressor.

$$Q_{evap} = \dot{m}_r * (h_9 - h_8) \tag{4.6}$$

$$W_{comp} = \dot{m}_r * \frac{(h_{2n} - h_1)}{\eta_{is}} \tag{4.7}$$

$$\eta_{is} = 0.874 - (0.0135 * PR) \tag{4.8}$$

where $h_1$ and $h_{2n}$ are the specific enthalpy of refrigerant before compression begins and after compression ends respectively, $\eta_{is}$ is the isentropic efficiency of compressor [32] and $\dot{m}_r$ is the mass flow rate of refrigerant which is given by

$$\dot{m}_r = \frac{V_{st} * N * \eta_{vol,cl} * 10^{-6}}{v_{10} * 60} \tag{4.9}$$

where $N$ is the compressor speed and $V_{st}$ is stroke volume.

### 4.2.4 Relative Coefficient of Performance

It is stated as the ratio COP of any refrigerant to the COP of baseline refrigerant i.e. R134a. It compared the COP of any refrigerant with respect to COP of R134a. It is a dimensionless quantity. Mathematically, it is given as

$$COP_{rel} = \frac{COP_{any\ ref}}{COP_{R134a}} \tag{4.10}$$

### 4.2.5 Exergetic Efficiency

It is given by the following formula

$$\eta_{ex} = \frac{COP}{COP_{rr}} \tag{4.11}$$

where $COP_{rr}$ is the coefficient of performance of a reversible refrigerator and it is defined between restricted dead state temperature ($T_0$) and space temperature ($T_r$) as

$$COP_{rr} = \frac{T_r}{T_0 - T_r} \tag{4.12}$$

### 4.2.6 Exergy Destruction Ratio

It is given by the following formula

$$\mathrm{EDR} = \left(\frac{1}{\eta_{ex}}\right) - 1 \tag{4.13}$$

### 4.2.7 Exergetic Destruction

For various components, it is given by the following formula

a) Compressor:

$$\dot{E}_{D,comp} = \dot{m}_r * (T_0 * (s_{2n} - S_{10}) \tag{4.14}$$

where $s_{2n}$ and $s_{10}$ are the specific entropy of refrigerant before entry and after compression ends in compressor respectively.

b) Condenser:

$$\dot{E}_{D,cond} = \dot{m}_r * (((h_{3n} - (T_0 * s_{3n})) - (h_6 - (T_0 * s_6))) \tag{4.15}$$

where $h_{3n}$ , $s_{3n}$ , $h_6$ and $s_6$ are the specific enthalpy and specific entropy of refrigerant before entry and after exit from the condenser respectively.

c) Expansion Valve:

$$\dot{E}_{D,exp} = \dot{m}_r * (T_0 * (s_8 - s_7) \tag{4.16}$$

where $s_7$ and $s_8$ are the specific entropy of refrigerant before entry and after exit from the expansion valve respectively.

d) Evaporator:

$$\dot{E}_{D,evap} = (\dot{m}_r * (((h_8 - (T_0 * s_8)) - (h_9 - (T_0 * s_9)))) + (Q_{evap} * (1 - (T_0/T_r))) \tag{4.17}$$

where $h_8 \& s_8$, are specific enthalpy and specific entropy of refrigerant before entry and $h_9$ & $s_9$ are specific enthalpy and specific entropy of refrigerant after exiting from evaporator respectively.

e) Liquid Vapour Heat Exchanger:

$$\dot{E}_{D,lvhx} = \dot{m}_r * ((h_6 - h_7 + h_9 - h_{10}) - (T_0 * (s_6 - s_7 + s_9 - s_{10}))) \tag{4.18}$$

where $h_6$ and $h_{10}$ are the specific enthalpy of refrigerant after exiting from the condenser and before entry in compressor respectively.

f) System Exergy Destruction:

$$\dot{E}_{D} = \dot{E}_{D,comp} + \dot{E}_{D,cond} + \dot{E}_{D,exp} + \dot{E}_{D,evap} + \dot{E}_{D,lvhx} \quad (4.19)$$

**4.2.8 Efficiency Defect**

For various components, it is given following formula

a) Compressor:

$$\delta_{comp} = \frac{\dot{E}_{D,comp}}{W_{comp}} \quad (4.20)$$

b) Condenser:

$$\delta_{cond} = \frac{\dot{E}_{D,cond}}{W_{comp}} \quad (4.21)$$

c) Expansion Valve:

$$\delta_{exp} = \frac{\dot{E}_{D,exp}}{W_{comp}} \quad (4.22)$$

d) Evaporator:

$$\delta_{evap} = \frac{\dot{E}_{D,evap}}{W_{comp}} \quad (4.23)$$

e) Liquid Vapour Heat Exchanger:

$$\delta_{lvhx} = \frac{\dot{E}_{lvhx}}{W_{comp}} \quad (4.24)$$

f) System Efficiency Defect:

$$\delta = \delta_{comp} + \delta_{cond} + \delta_{exp} + \delta_{evap} + \delta_{lvhx} \quad (4.25)$$

**4.3 RESULTS AND DISCUSSION**

REFPROP [74] is used to calculate thermodynamic properties of refrigerants and EES [70] is used to make a computational model for energy and exergy analysis of the system as shown in Appendix-I. Present analysis corresponds to the following assumptions:

- Steady state condition in all the components
- Kinetic, as well as potential energy and exergy losses, are not considered
- Pressure losses in the pipelines are neglected

- The difference between evaporator and space temperature ($T_r - T_e$): 15°C [19]

Figure 2 depicts the variation of pressure ratio with a change in refrigerant. The maximum value of pressure ratio is for refrigerant R4 (i.e. R600a) and the lowest value is for R3 (i.e. R290). The value of pressure ratio decreases as we increase the mass percentage of R5 (i.e. R1234yf) in the mixture of R1/R5 up to R13 (i.e. R134a/R1234yf (20%80%)) then start increasing whereas when we increase the mass percentage of R5 in the mixture of R1/R5/R6 (keeping the mass percentage of R1 constant) the value of pressure ratio decreases.

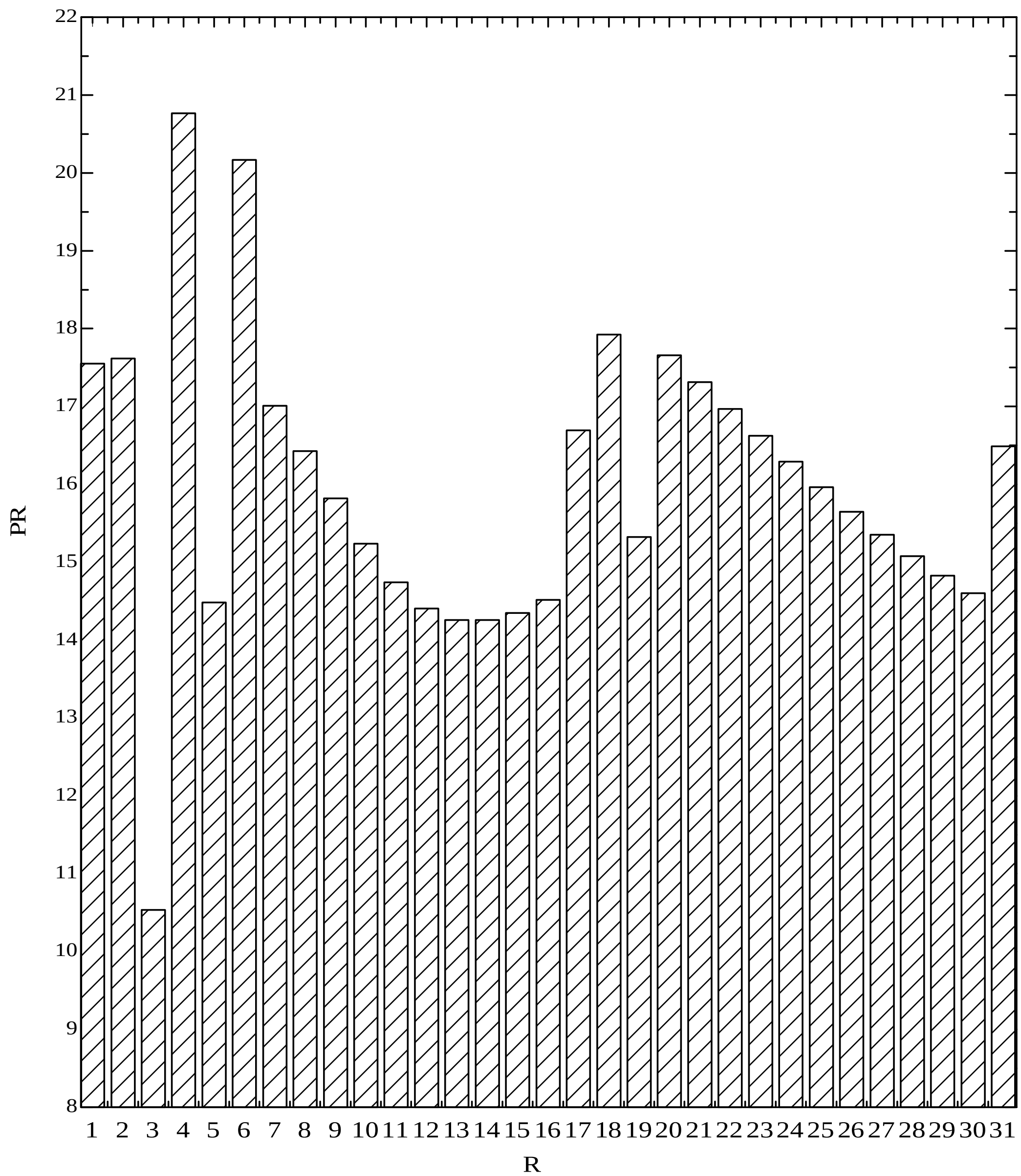



Figure 4.2 Variation of pressure ratio with refrigerant

Figure 4.3 shows the deviation of mass flow rate with change in refrigerant The value of mass flow rate increases as we increase the mass percentage of R5 (i.e. R1234yf) in the mixture of R1/R5 up to R13 (i.e. R134a/R1234yf (20%80%)) then start decreasing whereas when we increase the mass percentage of R5 (i.e.. in the mixture of R1/R5/R6 (keeping the mass percentage of R1 constant) the value of mass flow rate. The maximum value of mass flow rate is for refrigerant R13 (i.e. R134a/R1234yf (20%80%)) and the lowest value is for R4 (i.e. R600a).

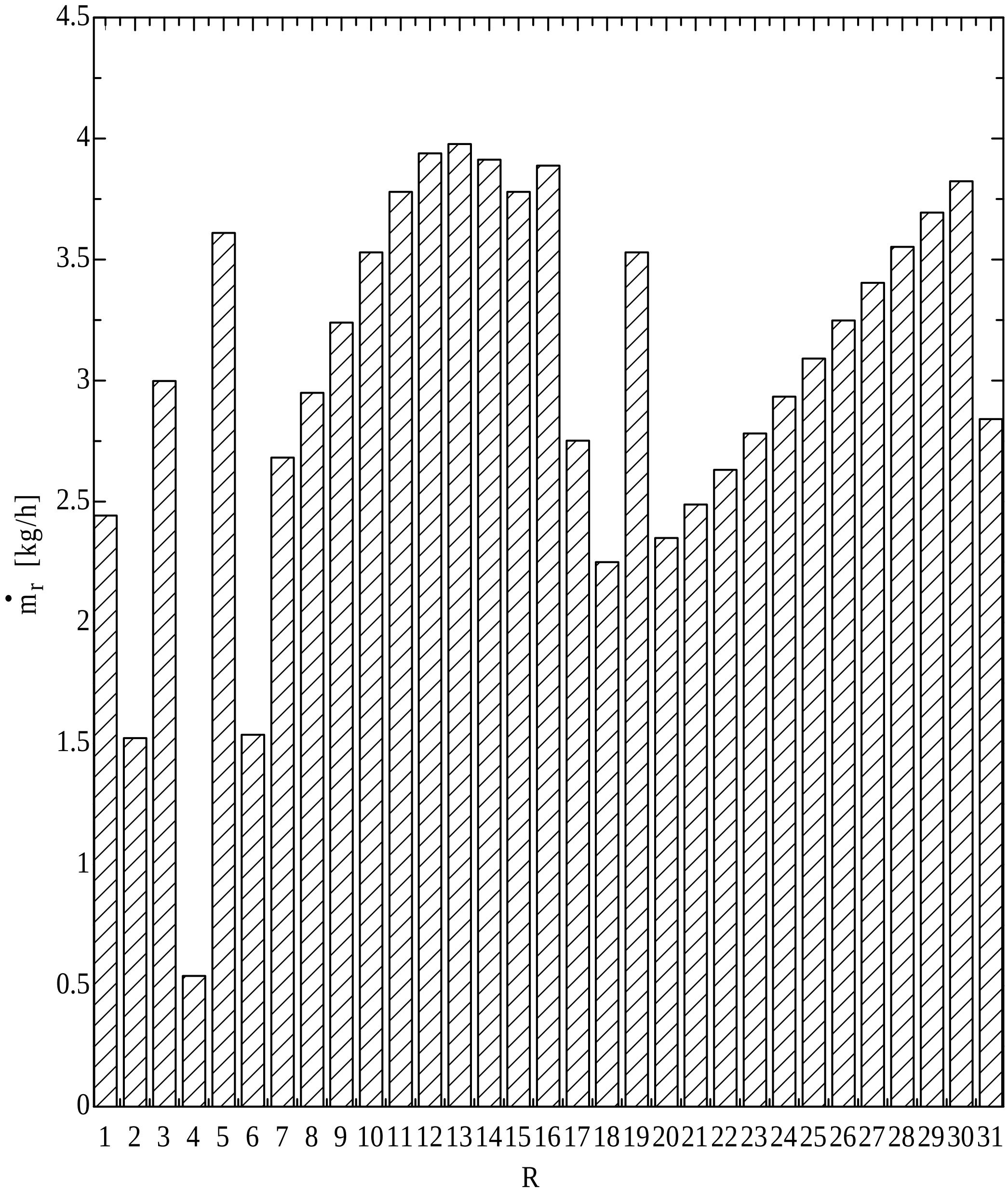



Figure 4.3 Variation of mass flow rate with refrigerant

Figure 4.4 represents a variation of relative volumetric cooling capacity with the change in refrigerant. From the viewpoint of compressor size, refrigerants with similar volumetric cooling capacity require no change in compressor [47]. It may be noted that refrigerant R31 (i.e. R134a/R1234yf/R1234ze (40%/22%/38%)) has nearly the same volumetric cooling capacity as that of the R1 (R134a baseline refrigerant).

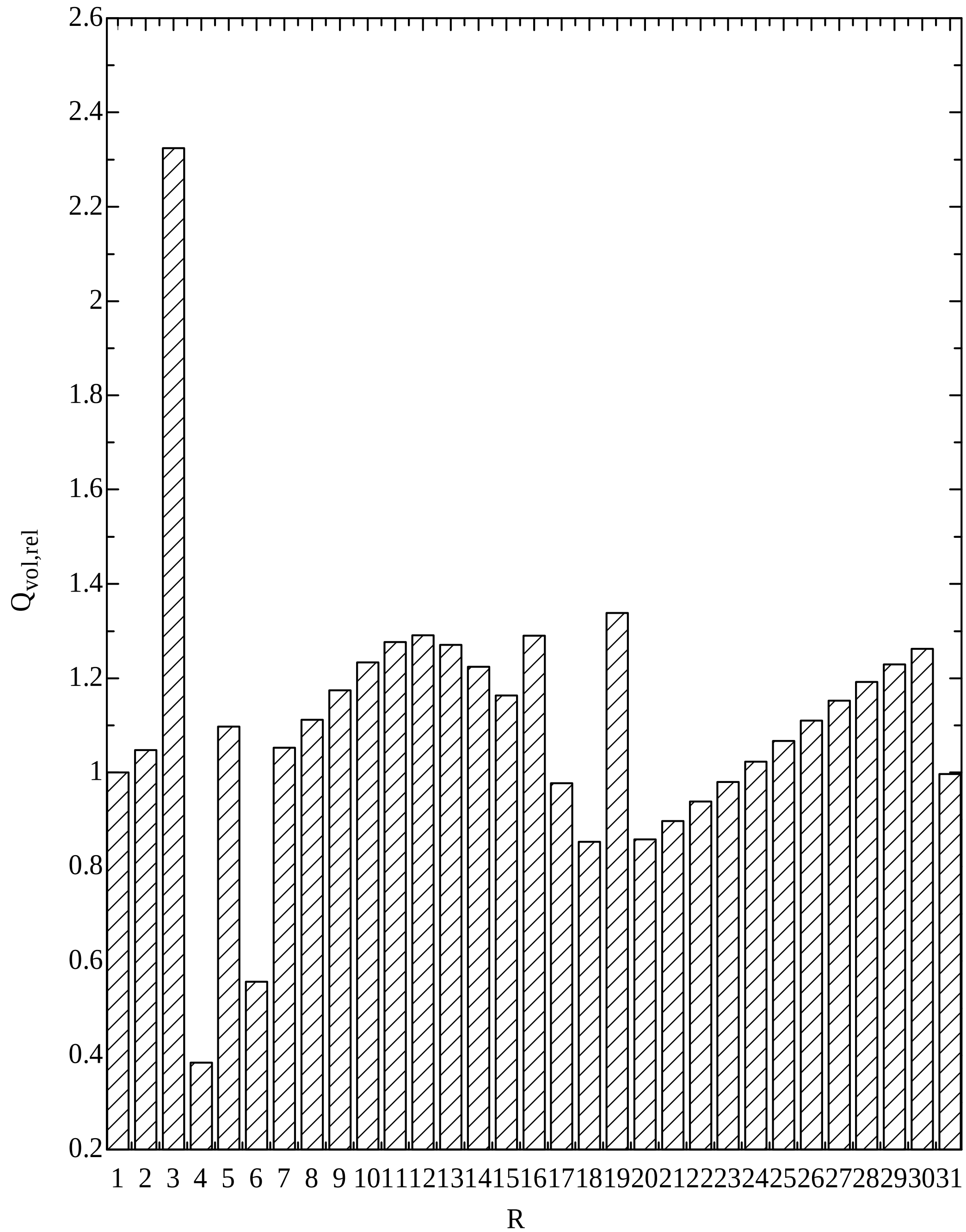



Figure 4.4 Variation of relative volumetric cooling capacity with refrigerant

Figure 4.5 represents a variation of cooling capacity with the change in refrigerant. It indicates that refrigerant R31 has nearly the same cooling capacity as that of the R1 (baseline refrigerant). With the increase in the mass percentage of R5 (i.e. R1234yf) in the mixture of R1/R5 the value of cooling capacity increases up to R12 (i.e. R134a/R1234yf (40%/60%)) then start decreasing. With the increase in the mass percentage of R5 in the mixture of R1/R5/R6 (keeping the mass percentage of R1 constant) the value of cooling capacity increases. The largest value of cooling capacity is for R3 (i.e. R290) and the lowest value is for R4 (i.e. R600a).

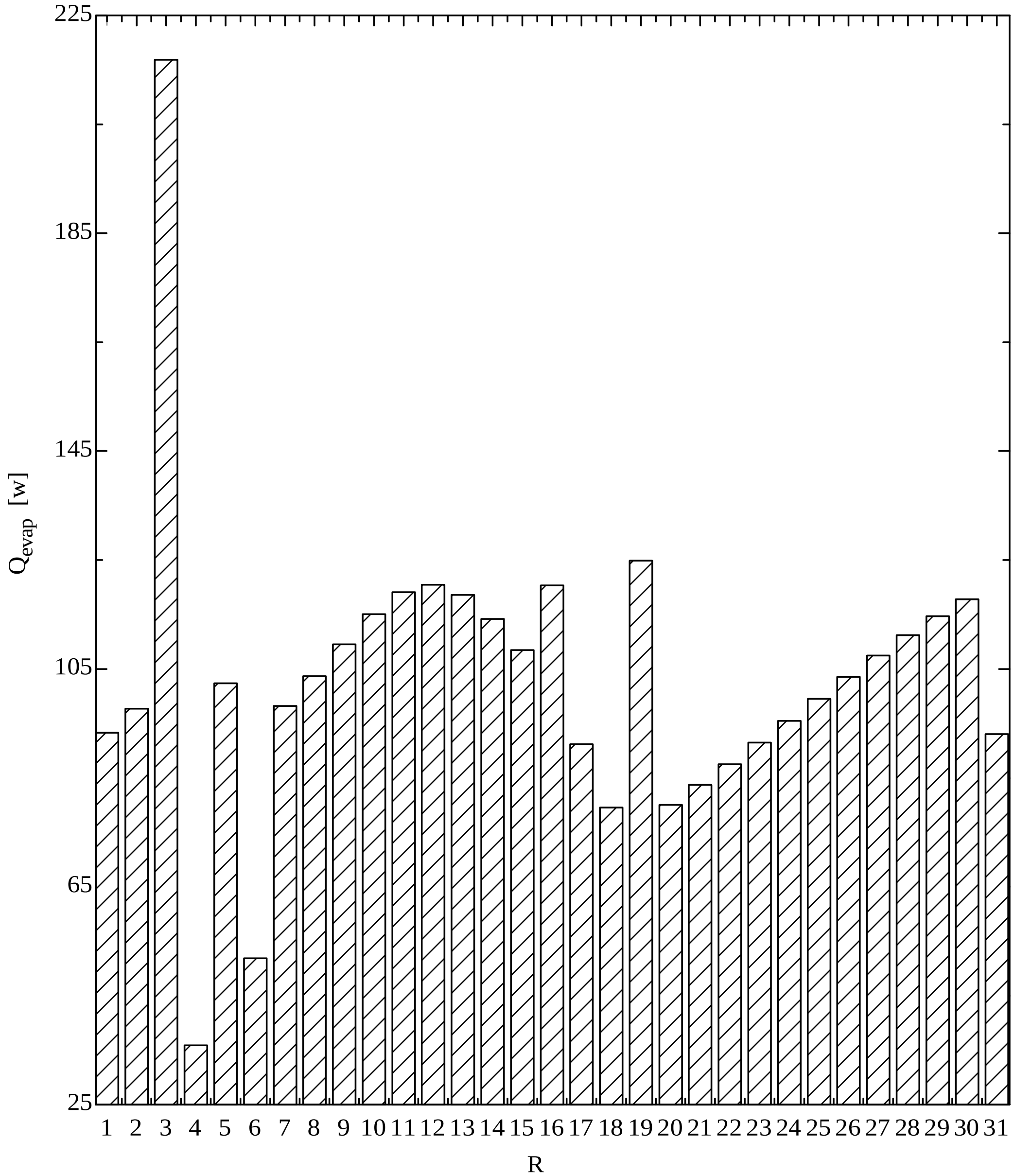



Figure 4.5 Variation of cooling capacity with refrigerant

Figure 4.6 shows the variation of relative coefficient of performance ($COP_{rel}$) with the change in refrigerant. For a low value of energy consumption, a refrigerant must have a high value of the COP. Refrigerant R3 (i.e. R290) has the highest value and R6 (i.e. R1234ze) has the lowest value of the COP among all the tested refrigerants as depicted. With the increase in the mass percentage of R5 (i.e. R1234yf) in the mixture of R1/R5 the value of COP increases up to R12 (i.e. R134a/R1234yf (40%/60%)) then start decreasing. With the increase in the mass percentage of R5 in the mixture in the mixture of R1/R5/R6 (keeping the mass percentage of R1 constant) the value of COP is not much affected much.

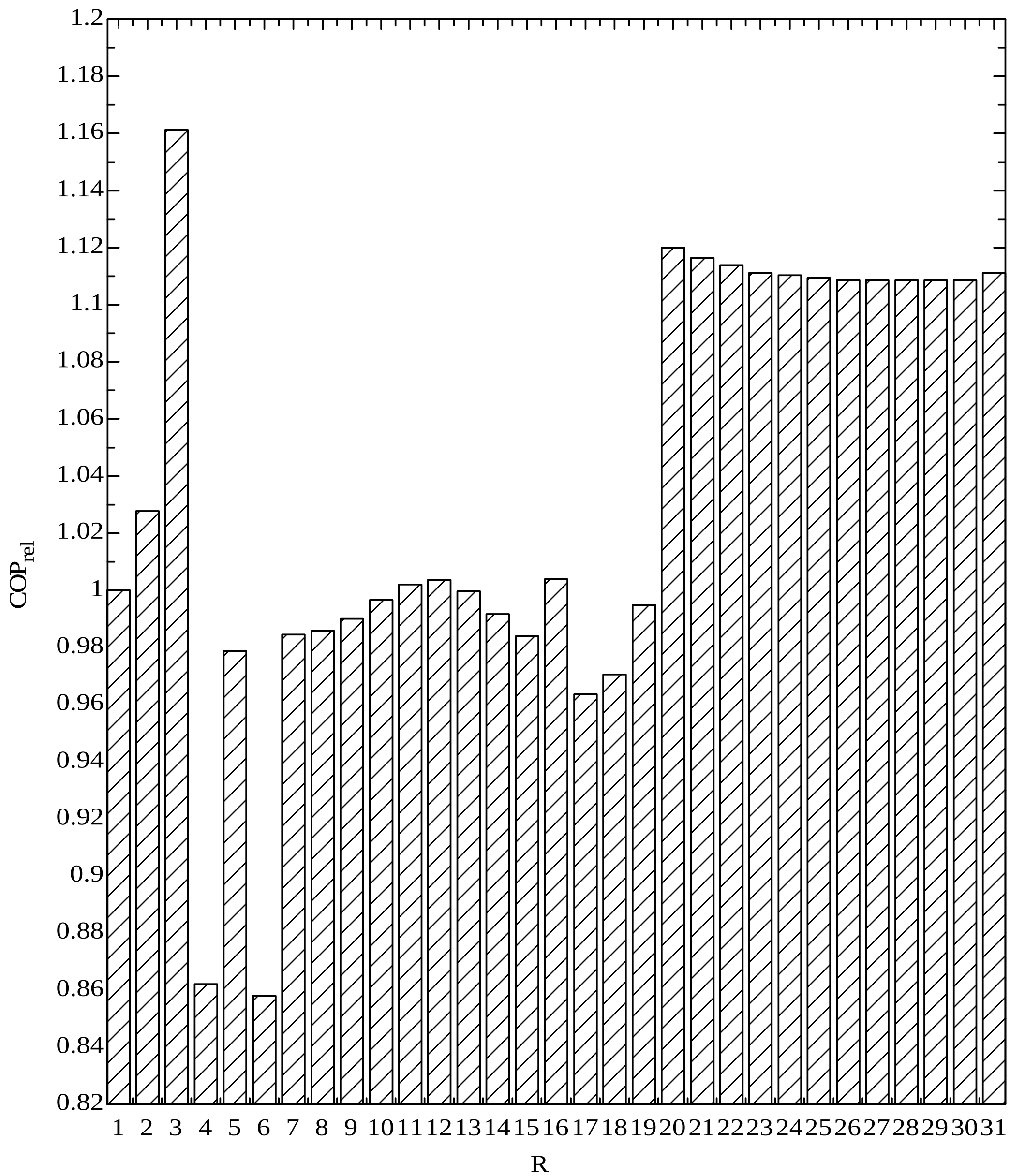



Figure 4.6 Variation of relative COP with refrigerant

The variation of exergetic efficiency with the change in the refrigerant is shown in Figure 4.7. The exergetic efficiency shows what percentage of fuel exergy can be found in the product exergy [24]. R3 (i.e. R290) presents maximum exergetic efficiency among all refrigerants. With the increase in the mass percentage of R5 (i.e. R1234yf) in the mixture of R1/R5 the value of exergetic efficiency increases up to R12 (i.e. R134a/R1234yf (40%/60%)) then start decreasing whereas if we increase the mass percentage of R5 in the mixture of R1/R5/R6 (keeping the mass percentage of R1 constant) the value exergetic efficiency slightly changes.

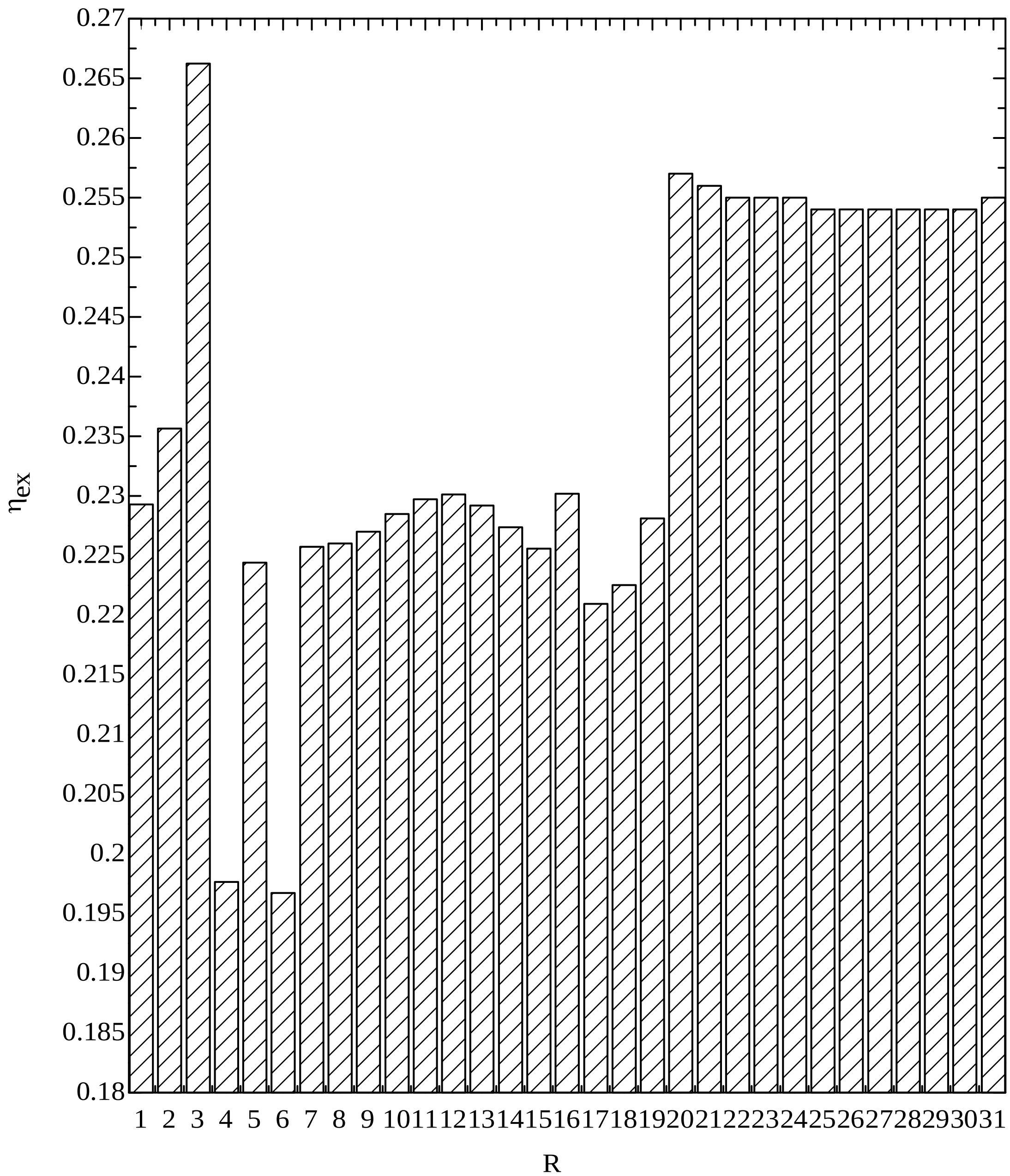



Figure 4.7 Variation of exergetic efficiency with refrigerant

Figure 4.8 represents a variation of exergy destruction ratio with the change in refrigerant. It indicates that refrigerant R6 (i.e. R1234ze) has the highest and R3 (I.e. R290) has the lowest value of the exergy destruction ratio among all considered refrigerants. With the increase in the mass percentage of R5 (i.e. R1234yf) in the mixture of R1/R5 the value of cooling capacity increases up to R12 (i.e. R134a/R1234yf (40%/60%)) then start decreasing. The value of exergy destruction ratio decreases with the increase in the mass percentage of R5 in the mixture of R1/R5 up to R12 (i.e. R134a/R1234yf (40%/60%)) then start increasing whereas if we increase the mass percentage of R5 in the mixture of R1/R5/R6 (keeping the mass percentage of R1 constant) the value of exergy destruction ratio slightly change.

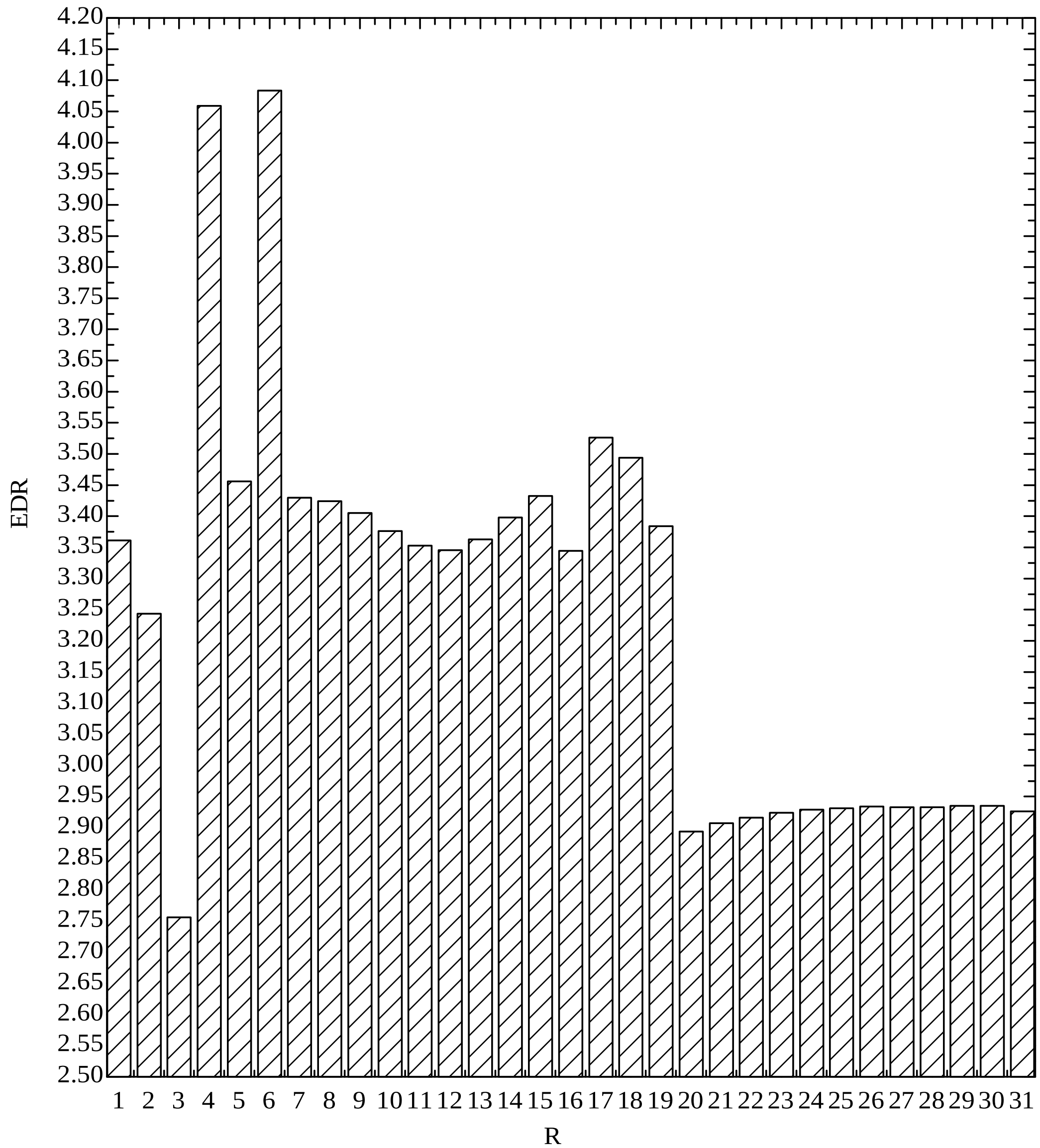



Figure 4.8 Variation of exergy destruction ratio with refrigerant

The variation of exergy destruction with the change in the refrigerant is shown in Figure 4.9. The exergy destruction is the amount of exergy that is lost to the environment and cannot be used anywhere [24]. R4 (i.e. R600a) has the least value and R3 (i.e. R290) has the highest value of exergy destruction. The value of exergy destruction increases with the increase in the mass percentage of R5 (i.e. R1234yf) in the mixture of R1/R5 up to R12 (i.e. R134a/R1234yf (40%/60%)) then start decreasing whereas if we increase the mass percentage of R5 in the mixture of R1/R5/R6 (keeping the mass percentage of R1 constant) the value of exergy destruction increases.

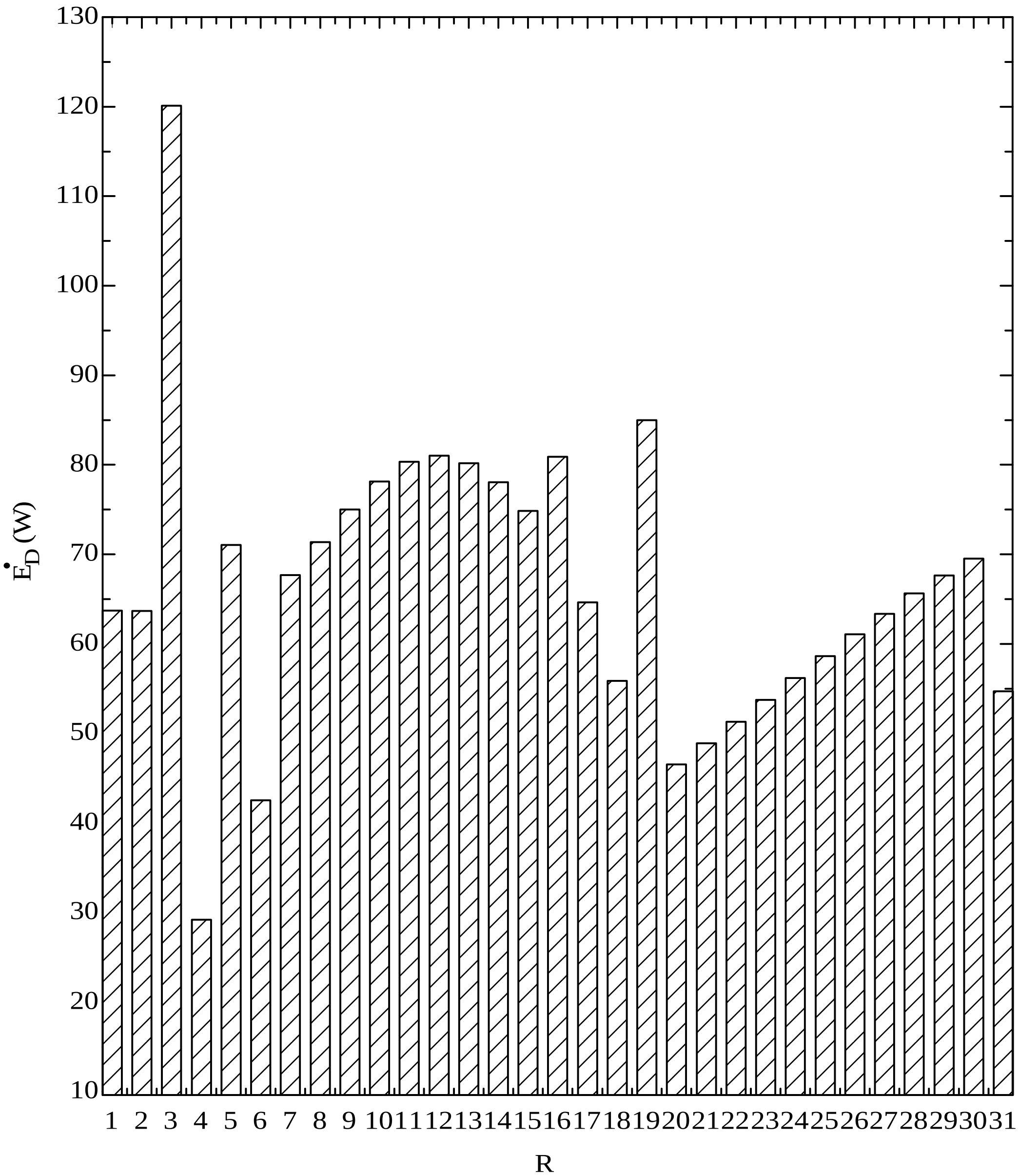



Figure 4.9 Variation of exergy destruction with refrigerant

The variation of efficiency defect with the change in the refrigerant is shown in Figure 4.10. The efficiency defect is the fraction of the input that is lost through irreversibility [71]. Among all considered refrigerant the lowest value of efficiency defect is for R3 (i.e. R290). The value of efficiency defect decreases with the increase in the mass percentage of R5 (i.e. R1234yf) in the mixture of R1/R5 up to R12 (i.e. R134a/R1234yf (40%/60%)) then start increasing whereas if we increase the mass percentage of R5 in the mixture of R1/R5/R6 (keeping the mass percentage of R1 constant) the value of efficiency defect slightly changes.

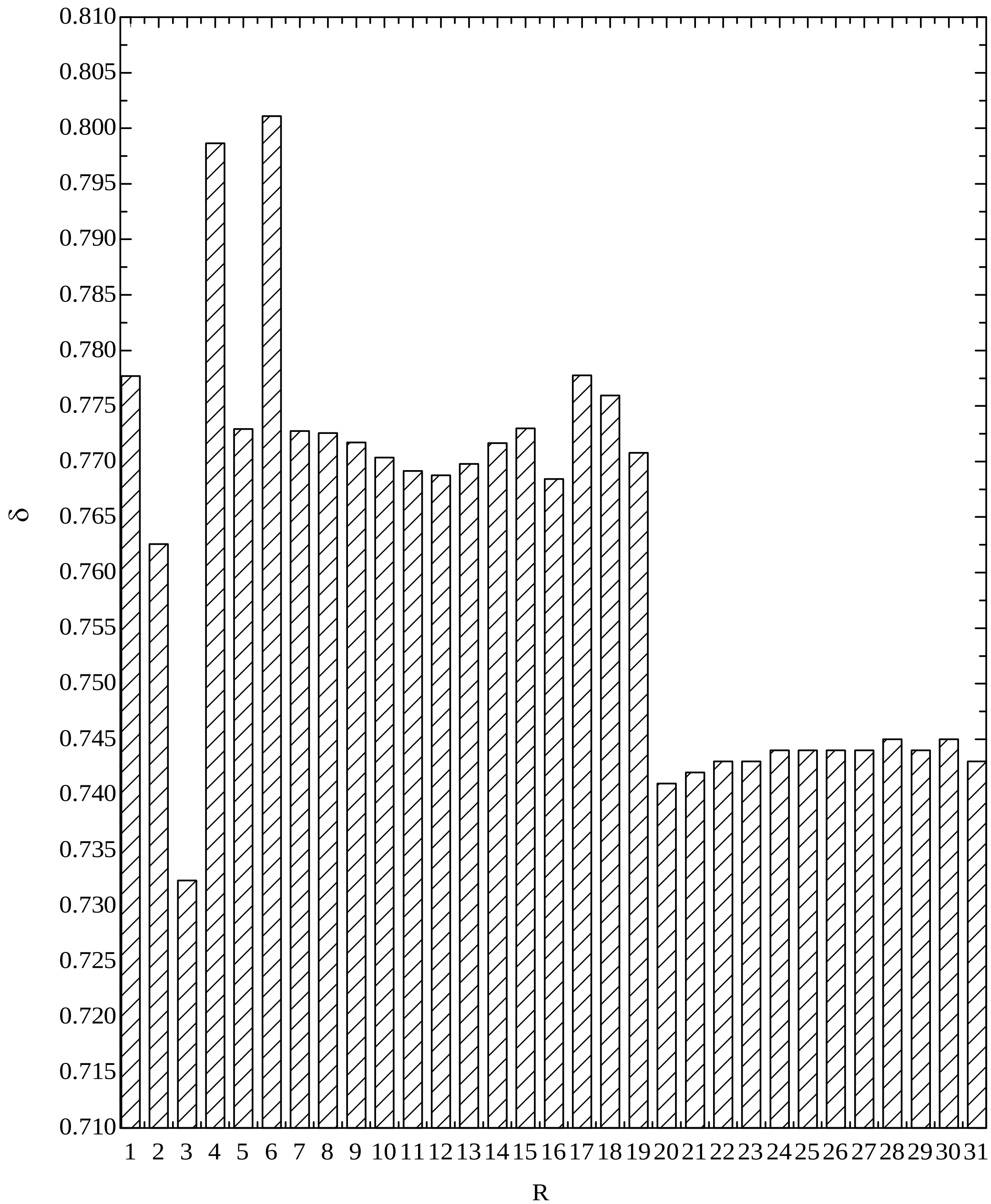



Figure 4.10 Variation of efficiency defect with refrigerant

Figure 4.11 depicts a variation of exergy destruction with components for R31 (i.e. R134a/R1234yf/R1234ze (40%/22%/38%)). It represents irreversibility within the component. It is clear from the figure that exergy destruction is maximum for compressor followed by the condenser, expansion valve, evaporator and liquid vapour heat exchanger respectively. Thus, the compressor is a component which requires maximum attention for improvement.

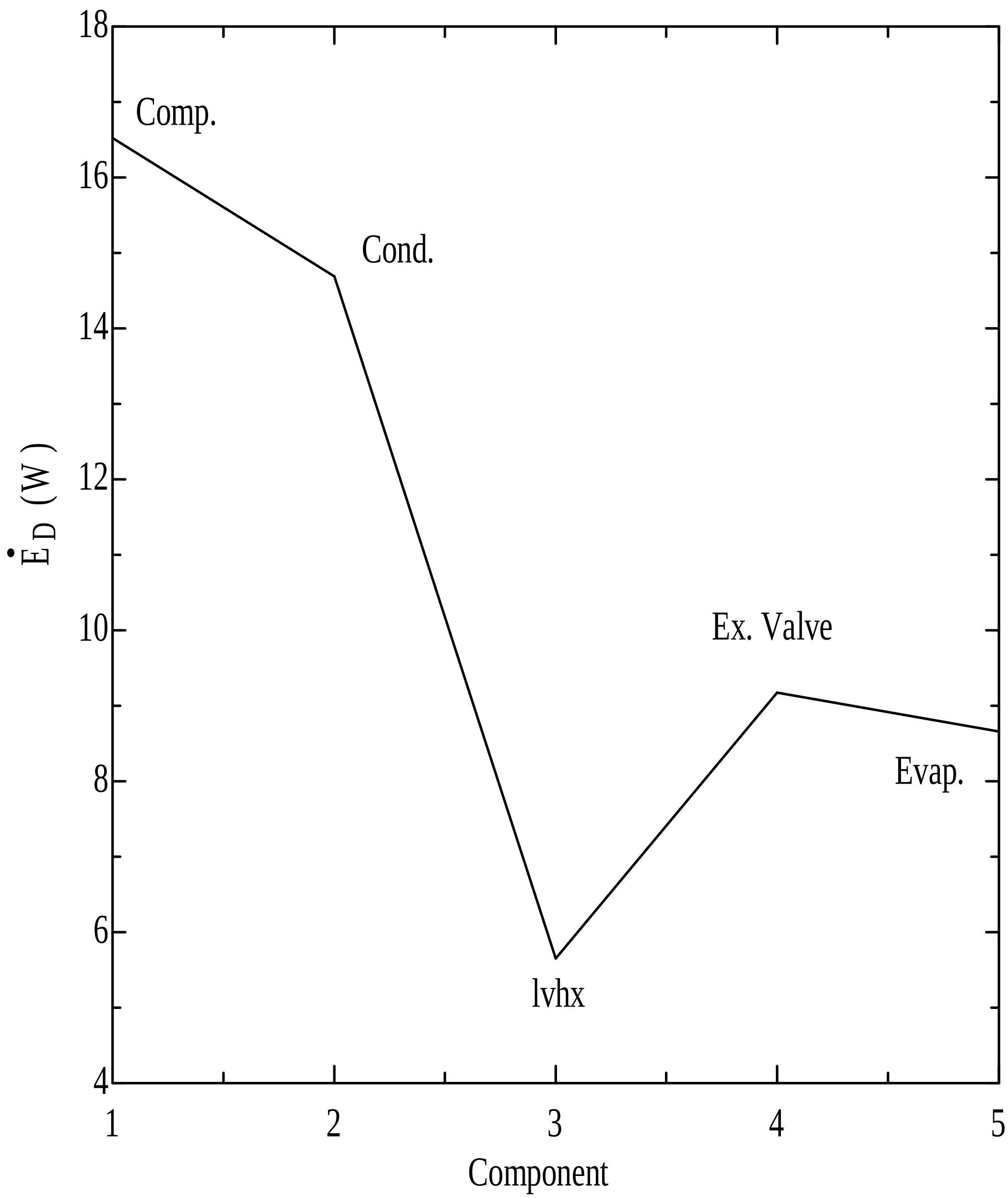



Figure 4.11 Variation of exergy destruction with component for R31 refrigerant

Figure 4.12 represents a variation of efficiency defect with components for R31 (i.e. R134a/R1234yf/R1234ze (40%/22%/38%)). It is clear from the figure that efficiency defect is highest for compressor followed by the condenser, expansion valve, evaporator and liquid vapour heat exchanger respectively. Thus, the compressor is the component which requires maximum attention for improvement.

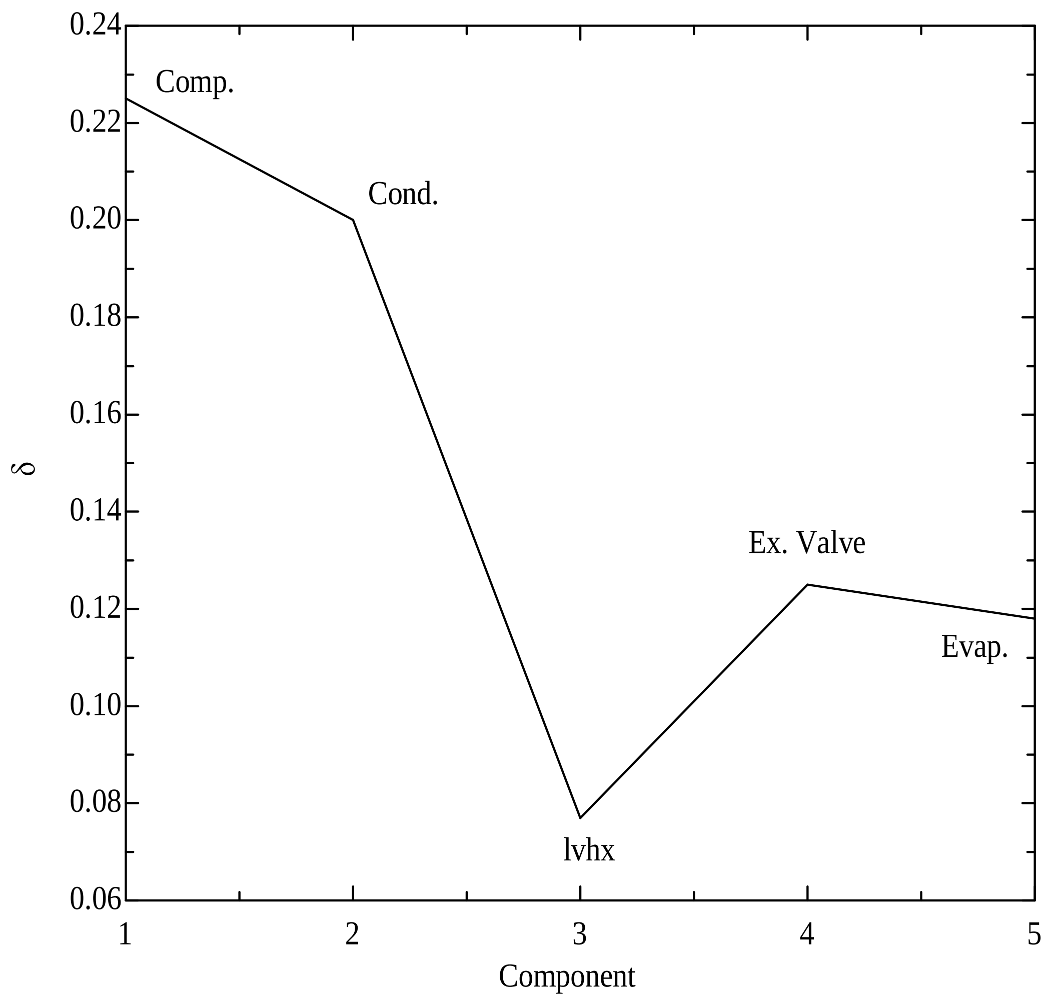



Figure 4.12 Variation of efficiency defect with a component for R31 refrigerant

Table 4.2 shows the value of performance parameters of refrigerants which can be used as an alternative to R134a based on results given by present analysis. The value of pressure ratio is maximum for R152a and lowest for R134a/R1234yf/R1234ze (40%/22%/38%) as compared to R134a whereas vice versa in case of mass flow rate. The value of relative volumetric cooling capacity for R134a/R1234yf/R1234ze (40%/22%/38%) is almost equal to 1 that means R134a/R1234yf/R1234ze (40%/22%/38%) is having volumetric cooling capacity value almost nearly same as that of R134a so it can be used with same compressor size as that of R134a. The value of exergetic efficiency and relative COP is highest for R134a/R1234yf/R1234ze (40%/22%/38%) whereas it has the lowest value of exergetic destruction and efficiency defect. The value of cooling capacity is highest for R134a/R1234yf (90%/10%) and lowest for R134a/R1234ze (42%/58%). It may be observed that refrigerant mixture HFC134a/HFO1234yf/HFO1234ze (40%/22%/38%) is the best drop-in replacement of HFC134a among available alternatives.

Table 4.2 Performance parameters of refrigerants

| Refrigerant | PR | $m_r$ (kg/hr) | $Q_{vol,rel}$ | $COP_{rel}$ | $Q_{eva}$ (W) | $\eta_{ex}$ | $\dot{E}_D$ (W) | $\delta$ |
|---|---|---|---|---|---|---|---|---|
| R1 (R134a) | 17.55 | 2.442 | 1 | 1 | 93.288 | 0.229 | 63.721 | 0.778 |
| R2 (R152a) | 17.61 | 1.523 | 1.047 | 1.028 | 97.682 | 0.236 | 66.665 | 0.763 |
| R7 (R134a/ R1234yf (90%/10%)) | 17.01 | 2.681 | 1.052 | 0.985 | 98.177 | 0.226 | 67.687 | 0.773 |
| R17 (R134a/R1234yf/R1234ze (42%/18%/ 40%)) | 16.70 | 2.752 | 0.977 | 0.964 | 91.146 | 0.221 | 64.619 | 0.778 |
| R18 (R134a/ R1234ze (42%/58%)) | 17.92 | 2.249 | 0.853 | 0.970 | 79.552 | 0.223 | 55.870 | 0.776 |
| R31 (R134a/R1234yf/R1234ze (40%/22%/ 38%)) | 16.49 | 2.841 | 0.997 | 1.111 | 93.039 | 0.255 | 54.692 | 0.743 |

### 4.4 EXPERIMENTAL VERIFICATION OF RESULTS

Two refrigerants HFC134a and HFC134a/HFO1234yf/HFO1234ze (40%/22%/38%) were tested and compared to the equivalent weight of the refrigerant (Table 4.2). The validation was done on refrigeration vapour compression test rig already discussed in chapter 3. The heat energy supplied was equivalent to cooling capacity and was recorded by the difference in the energy meter reading.$E_1$ is initial energy meter reading and $E_2$ is final energy meter reading, then ($E_2$-$E_1$) is the difference in the energy meter reading and corresponds to cooling capacity. Work input to the compressor was recorded by the difference in energy meter reading for achieving the cooling capacity in the evaporator consisting of glycol. $E_3$ is initial energy meter reading for work input in the compressor and $E_4$ is final energy meter reading for work input in the compressor, then ($E_4$-$E_3$) is the difference in the energy meter reading and corresponds to compressor work input. The error of the power input data and cooling capacity measurement was ± 4 %. Two refrigerants R134a and HFC134a/HFO1234yf/HFO1234ze (40%/22%/38%) were charged alternatively through compressors with the setting of the same parameters. In both cases, parameters were $5^0$C evaporator temperature, $50^0$C condensing temperature, $0^0$C subcooling and $10^0$C superheating. The coefficient of performance was calculated from the relation defined as cooling capacity divided by work input. The relative COP ($COP_{rel}$) is the COP with respect to refrigerant R1. The results obtained for these two refrigerants are shown in Table 4.3. The values relative COP ($COP_{rel}$) are near to each other (Table 4.2 and Table 4.3). The COP of R31 is slightly lesser as compared to R1 and variation in the value of the COP is 5.5%.

Table 4.3 Experimental results for R1 and R31

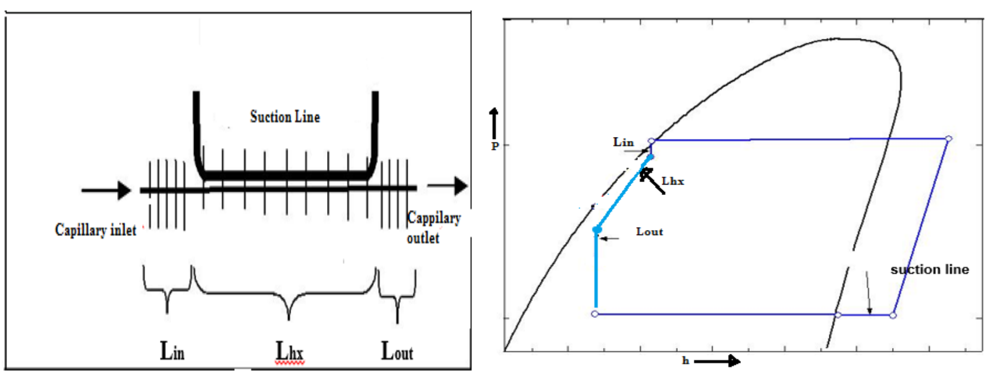
| Sr. No. | Refrigerant | Average Cooling Capacity (W) | Power Input (W) | COP | $COP_{rel}$ |
|---|---|---|---|---|---|
| 1 | R1: R134a | 113.0 | 51.36 | 2.20 | 1 |
| 2 | R31: HFC134a/HFO1234yf/ HFO1234ze (40%/22%/38%) | 113.8 | 54.70 | 2.08 | 0.945 |

### 4.5 OPTIMIZATION OF CAPILLARY TUBE PARAMETERS

Among all the main factors which are responsible for energy loss in vapour compression cycle performance, one is expansion process because refrigerant flashes to vapour on entering the capillary tube, reducing the cooling capacity in the evaporator and as a result the size of evaporator increases. This issue can be resolved by adopting multi-stage expansion with the flash chamber in which the flash vapour is removed after each stage of expansion. The effect of refrigeration can also be enhanced by passing the refrigerant through subcooler after the condenser. Although a lot of work can be found on refrigerants R134a and R1234yf for various thermodynamic properties and compressor performance, very limited work on optimization of a capillary dimension such as length and diameter for refrigerant R1234yf. So, there is a need for finding capillary tube parameters when the system is charged with mixture HFC134a/HFO1234yf/HFO1234ze. The coefficient of performance of a system can be improved either by minimizing work input given to the system or by increasing the refrigeration effect The present work is a systematic comparison of the cooling capacity, COP, capillary tube length and diameter for optimizing the refrigerating effect of mixture HFC134a/HFO1234yf/HFO1234ze.

### 4.5.1 Material and Method

The basic refrigeration and air-conditioning system consist of the evaporator, compressor, condenser and capillary tube. Capillary tube has no moving parts and leak-proof system. The capillary tube is widely used to provide required pressure drop between the condenser & evaporator and also to regulate the mass flow rate through the system. Selection of the capillary tube is based upon the required pressure drop, compressor specifications, mass flow rate, and cooling capacity required. R134a vapour compression cycle efficiency benefits greatly from the suction line heat exchange. The suction line heat exchanger comprises an adiabatic inlet section, a heat exchanger section and an adiabatic outlet section. The refrigerant after leaving the condenser flashes in the adiabatic inlet section and enters the heat exchanger section of the capillary tube (as shown in Figure 4.13 a & b), where it rejects heat to the cold suction line downstream of the evaporator and enters the adiabatic outlet section pressure.




(a) (b)

Figure 4.13 a. Capillary tube length parameters b. P-h diagram

In deciding the performance of a system the major role is played by the capillary dimension. All combinations of capillary tube diameter and lengths which can give a stable and efficient performance are required to be identified, as the heat exchanger length is maximized to utilize whatever suction line length is accessible. The size of the capillary tube is fairly critical. Unlike orifices, the pressure drop in expansion device such as capillary tubes depends on their length as well as their diameter. The relationship between these two factors has been illustrated. Three capillary tubes of copper metal of different inner diameter 0.78mm, 0.81mm and 0.83 and same outer diameter 2.1 mm were selected. A 3.600 m length was selected from the standard chart [56] available for the refrigerant R134a and increment of 10% length was given in two stages for the new refrigerant mixture. Nine observations were recorded for each refrigerant using various combinations of length and diameter. S, M, L stands for small, medium and large size respectively. In two letters for capillary tube-like SM: the first letter stands for the inner diameter of the capillary tube and second letter stands for the length.

### 4.5.2 Results and Discussion

After the evaluation of R134a, a perfect vacuum was created in the system and a similar procedure was then repeated for the refrigerant mixture as both have similar working pressure range and allows application of the same refrigerant oil (POE oil). The validation was done on refrigeration vapour compression test rig as mentioned in chapter 3.

Optimum length and diameter of capillary tube were estimated through an experimental investigation by replacing R134a with the refrigerant mixture. For comparison, firstly the results were recorded using R134a (as shown in Table 4.4). With this diameter and length of capillary tube, the results obtained are refrigerating effect 217 W and COP 2.20.

Table 4.4 Effect of capillary tube size on various properties of R134a

| Sr. No. | Capillary Tube Specification | Capillary Tube Length (m) | Inner Diameter (mm) | Average Refrigerating Effect (W) | Average COP |
|---|---|---|---|---|---|
| 1 | SS | 3.600 | 0.78 | 217 | 2.20 |

For refrigerant mixture R31, when the diameter of the capillary tube is kept constant and the capillary tube length is increased, the COP increases due to increase in refrigerating effect as shown in table 4.5. The diameter of capillary tube was increased to 0.81 mm and 0.83 mm for refrigerant mixture R31. This provides flexibility to the designer in selecting a capillary tube matching various parameters. Capillary tube ML for the refrigerant mixture R31 with the internal diameter 0.81mm, length 4.360 m provides highest refrigerating effect 217 W and COP 2.18.

Table 4.5 Effect of capillary tube size on various properties of refrigerant mixture R31

| Sr. No. | Capillary Tube Specification | Capillary Tube Length (m) | Inner Diameter (mm) | Refrigerating Effect (W) | COP |
|---|---|---|---|---|---|
| 1 | SS | 3.600 | 0.78 | 207 | 2.08 |
| 2 | SM | 3.960 | 0.78 | 208 | 2.09 |
| 3 | SL | 4.356 | 0.78 | 208 | 2.09 |
| 4 | MS | 3.600 | 0.81 | 210 | 2.11 |
| 5 | MM | 3.960 | 0.81 | 213 | 2.14 |
| 6 | ML | 4.356 | 0.81 | 217 | 2.18 |
| 7 | LS | 3.600 | 0.83 | 206 | 2.06 |
| 8 | LM | 3.960 | 0.83 | 207 | 2.08 |
| 9 | LL | 4.356 | 0.83 | 209 | 2.10 |

It was revealed that for the same compressor work, the cooling capacity of R1234yf decreases although R1234yf had about 25% more saturation vapour density than R134a because latent heat of vaporization of R1234yf was about 18% lower than that of the R134a as shown in Table 4.6.

Table 4.6 Comparison of R134a and R1234yf properties

| Property | R134a | R1234yf | ((R134a-R1234yf)/(R134a))*100 |
|---|---|---|---|
| Saturated Vapour Density ($kg/m^3$) | 9.816 | 12.296 | -25.26% |
| Latent Heat of Vaporization (kJ/kg) | 206.40 | 169.81 | +17.73% |
| Saturated Pressure @ -10.6ºC (kPa) | 195.90 | 216.92 | -10.73% |
| Saturated Pressure @ 54.4ºC (kPa) | 1469.80 | 1444.50 | +1.72% |

Over the range of operational conditions tested, the difference in terms of COP is less than 5.5% while the difference between cooling capacity for SS is 4.6% for R134a and refrigerant mixture R31. Thus, this analysis provides the following information:

- Flexibility to the designer in selecting capillary tube parameters for the environmentally friendly refrigerant mixture R31. In the present work, capillary tube ML for the refrigerant mixture R31 with the internal diameter 0.81 mm and length 4.360 m provides the maximum refrigerating effect and the optimum value of the COP.
- When the diameter of the capillary tube is kept constant and the capillary tube length is increased using refrigerant mixture R31, COP increases due to increase in refrigerating effect.
- As the diameter of capillary tube increases using refrigerant mixture R31 it results in increased mass flow rate. It is evident that the change in diameter on a percentage basis can change the flow more than an equal change in length.
- Over the range of operational conditions tested, the maximum difference in terms of cooling capacity and COP is less than 5.5 %.
- With the modifications in capillary tube parameters, refrigerant mixture R31 provides the substitute of R134a in terms of comparable COP.

**4.6 SUMMARY**

This chapter provides us with the drop-in substitute alternative of R134a in a vapour compression system by applying energy and exergy analysis. The proposed refrigerant HFC134a/HFO1234yf/HFO1234ze (40%/22%/38%) is having zero ODP, GWP around 600, performance parameter similar to that R134a but COP value slightly lesser. Optimization of capillary tube parameters provides up to 5.5% improvement in COP of refrigerant mixture R31 and it will result in COP comparable with that of R134a. The proposed refrigerant mixture is lesser costlier than HFO1234yf, safer to use and provide refrigeration properties similar to R134a.

# CHAPTER 5

# THERMOECONOMIC ANALYSIS OF VAPOUR COMPRESSION SYSTEM USING ENVIRONMENTAL FRIENDLY REFRIGERANTS AND THEIR MIXTURES

## 5.1 INTRODUCTION

This chapter discusses thermoeconomic analysis of refrigeration system for finding a drop-in substitute for an alternative refrigerant of R134a. The alternative refrigerants taken for the present study are R152a, R600a, R1234yf, R1234ze, R134a/R1234yf (10%/90%) and R134a/R1234yf/R1234ze (40%/22%/38%). Each of these refrigerants are tested among various parameters such as cost of operating, cost of exergy destruction, levelized electricity cost, energy efficiency ratio, volumetric cooling capacity, exergetic efficiency, exergy destruction, exergoeconomic factor and cost importance. Schematic diagram of the household domestic refrigerator is shown in Figure 5.1 for the present analysis [86]. Refrigerants selected for the analysis are shown in Table 5.1.

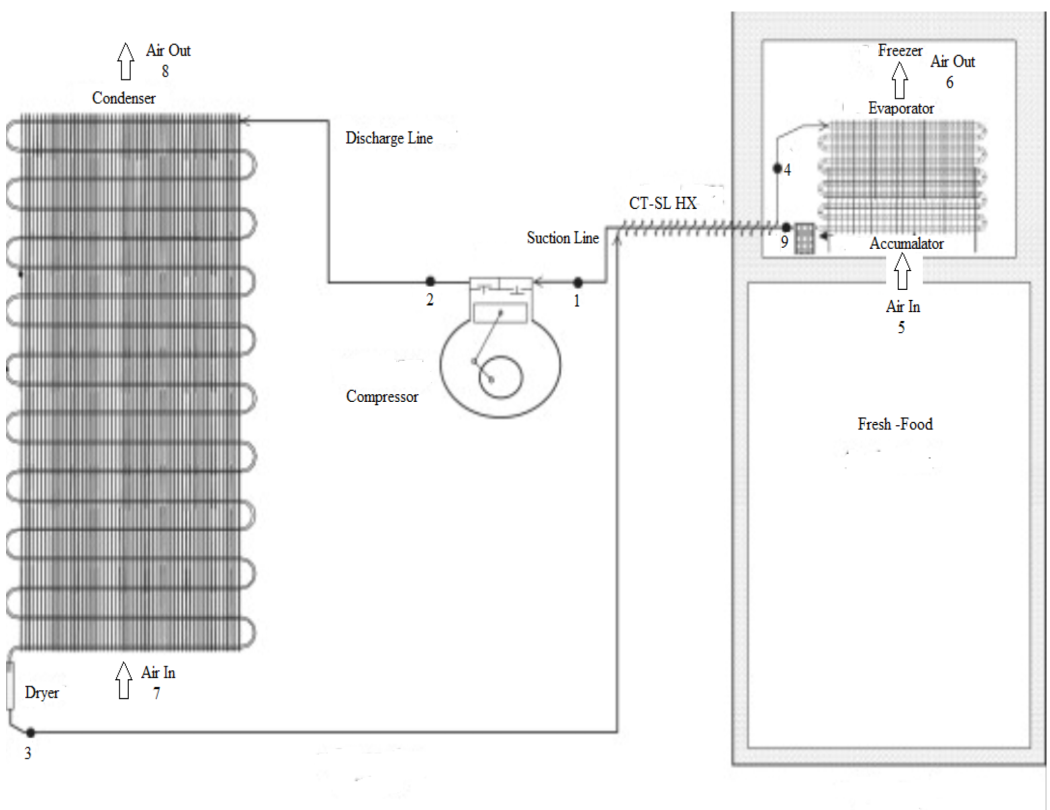



Figure 5.1 Household domestic refrigerator

Table 5.1 Properties of all considered refrigerants [33, 124]

| Sr. No. | Refrigerant | Naming | Type | GWP (100 yrs) | ODP | Safety Group |
|---|---|---|---|---|---|---|
| 1 | R134a | R1 | HFC | 1430 | 0 | A1 |
| 2 | R152a | R2 | HFC | 140 | 0 | A2 |
| 3 | R600a | R3 | HC | 11 | 0 | A3 |
| 4 | R1234yf | R4 | HC | 4 | 0 | A2L |
| 5 | R1234ze | R5 | HFO | 6 | 0 | A2L |
| 6 | R134a/R1234yf (10%/90%) | R6 | Blend | 146.6 | 0 | n.d. |
| 7 | R134a/R1234yf/R1234ze (40%/22%/38%) | R7 | Blend | 575.16 | 0 | A1 |

## 5.2 IMPORTANT PERFORMANCE CHARACTERISTICS

For the present analysis, refrigerator has a 0.04 compressor clearance ratio and 6.64 $cm^3$ stroke volume [47]. Some important performance characteristics need to be considered in order to find an alternative to R134a and theses are:

### 5.2.1 Volumetric Cooling Capacity

$$Q_{vol} = \frac{(h_9 - h_4) * \eta_{vol,cl}}{v_1} \tag{5.1}$$

where $h_9$ and $h_4$ are the specific enthalpy of the refrigerant at the evaporator inlet & exit respectively and $v_1$ is the specific volume of the refrigerant at the compressor inlet. The clearance or ideal volumetric efficiency $\eta_{vol,cl}$ is given by

$$\eta_{vol,cl} = 1 - CCR(PR^{1/n} - 1) \tag{5.2}$$

where $n$ is the polytropic index, PR is pressure ratio and CCR is compressor clearance ratio.

$$PR = \frac{P_{sat,,cond}}{P_{sat,evap}} \tag{5.3}$$

where $P_{sat,cond}$ and $P_{sat,evap}$ are the vapour pressure of refrigerant at saturated condenser and evaporator temperature respectively.

### 5.2.2 Relative Volumetric Cooling Capacity

$$Q_{vol,rel} = \frac{Q_{vol,any\,ref}}{Q_{vol,R134a}} \tag{5.4}$$

### 5.2.3 Energy Efficiency Rating

$$EER = \frac{Q_{evap}}{W_{elec}} \tag{5.5}$$

where $Q_{evap}$ is the cooling capacity and $W_{elec}$ is the electric power supply to the system.

$$Q_{evap} = \dot{m}_r * (h_9 - h_4) \tag{5.6}$$

$$W_{elec} = W_{motor} + W_{fan} \tag{5.7}$$

$$W_{fan} = W_{fan,\,ref} * \left( \left( \frac{\dot{m}_{air,\,evap}}{\dot{m}_{air,\,evap,\,ref}} \right)^3 \right) \tag{5.8}$$

$$W_{fan,\,ref} = 0.007\ [kW] \tag{5.9}$$

$$\dot{m}_{air,evap,ref} = 0.01766 [kg/s] \tag{5.10}$$

$$W_{motor} = \frac{W_{comp}}{\eta_{motor}} \tag{5.11}$$

where $W_{motor}$ , $W_{fan}$ , $W_{comp}$ , $W_{fan,\,ref}$ , $\dot{m}_{air,evap}$ and $\dot{m}_{air,evap,ref}$ denotes electric power input to motor, electric power input to fan, power input to compressor, reference value of power consumption of evaporator fan [86], mass flow rate of air in the evaporator and reference value of mass flow rate of air due to evaporator fan respectively [86].

$$W_{comp} = \dot{m}_r * \frac{(h_{2s} - h_1)}{\eta_{is}} \tag{5.12}$$

$$\eta_{is} = 0.874 - (0.0135 * PR) \tag{5.13}$$

where $h_1$ and $h_{2s}$ are the specific enthalpy of refrigerant before compression begins and after compression ends, $\eta_{is}$ is the isentropic efficiency of compressor [32], $\dot{m}_r$ is the mass flow rate of refrigerant (kg/s) and is given by

$$\dot{m}_r = \frac{V_{st} * N * \eta_{vol,cl} * 10^{-6}}{v_1 * 60} \tag{5.14}$$

where $N$ is the compressor speed and $V_{st}$ is stroke volume.

### 5.2.4 Exergetic Destruction

For various components, it is given by the following formula

a) Compressor:

$$\dot{E}_{D,comp} = \dot{E}_1 + W_{comp} - \dot{E}_2 \tag{5.15}$$

where $\dot{E}_1$ and $\dot{E}_2$ are the exergy rate of refrigerant before entry and after compression ends in compressor respectively.

b) Condenser:

$$\dot{E}_{D,cond} = \dot{E}_2 + \dot{E}_7 - \left(\dot{E}_3 + \dot{E}_8\right) \tag{5.16}$$

where $\dot{E}_3, \dot{E}_7$ and $\dot{E}_8$ denotes exergy rate of refrigerant at exit of condenser, exergy rate of air before entry in condenser and exergy rate of air after exiting from condenser respectively.

c) Expansion Valve:

$$\dot{E}_{D,exp} = \dot{E}_3 - \dot{E}_4 \tag{5.17}$$

where $\dot{E}_4$ is the exergy rate of refrigerant at the exit of the expansion valve.

d) Evaporator:

$$\dot{E}_{D,evap} = \dot{E}_4 + \dot{E}_5 - \left(\dot{E}_9 + \dot{E}_6\right) \tag{5.18}$$

where $\dot{E}_5, \dot{E}_6$ and $\dot{E}_9$ are exergy rate of air before entry in the evaporator, exergy rate of air after exit from evaporator and exergy rate of refrigerant after exiting from evaporator respectively.

e) Motor:

$$\dot{E}_{D,motor} = W_{motor} - W_{comp} \tag{5.19}$$

f) System Exergy Destruction:

$$\dot{E}_D = \dot{E}_{D,comp} + \dot{E}_{D,cond} + \dot{E}_{D,exp} + \dot{E}_{D,evap} + \dot{E}_{D,motor} \tag{5.20}$$

### 5.2.5 Exergetic Efficiency

For various components it is given by following formulae:

a) Compressor:

$$\eta_{ex,comp} = \frac{\dot{E}_2 - \dot{E}_3}{W_{comp}} \tag{5.21}$$

b) Condenser:

$$\eta_{ex,cond} = \frac{\dot{E}_3}{\dot{E}_2} \tag{5.22}$$

c) Expansion Valve:

$$\eta_{ex,exp} = \frac{\dot{E}_4}{\dot{E}_3} \tag{5.23}$$

d) Evaporator:

$$\eta_{ex,evap} = \frac{\dot{E}_6 - \dot{E}_5}{\dot{E}_4 - \dot{E}_9} \tag{5.24}$$

e) Motor:

$$\eta_{ex,motor} = \frac{W_{comp}}{W_{motor}} \tag{5.25}$$

f) System Exergy Destruction:

$$\eta_{ex} = 1 - \frac{\dot{E}_D + \dot{E}_{loss}}{\dot{E}_{fuel,system}} \tag{5.26}$$

where $\dot{E}_{loss}$ and $\dot{E}_{fuel,system}$ are the exergy loss of the system and exergy of the fuel supply to the system respectively.

$$E_{loss} = \dot{E}_8 - \dot{E}_7 \tag{5.27}$$

$$\dot{E}_{fuel,system} = W_{elec} \tag{5.28}$$

### 5.2.6 Exergy Destruction Ratio

For various components it is given by following formulae:

a) Compressor:

$$y_{D,comp} = \frac{\dot{E}_{D,comp}}{W_{elec}} \tag{5.29}$$

b) Condenser:

$$y_{D,cond} = \frac{\dot{E}_{D,cond}}{W_{elec}} \tag{5.30}$$

c) Expansion Valve:

$$y_{D,exp} = \frac{\dot{E}_{D,exp}}{W_{elec}} \tag{5.31}$$

d) Evaporator:

$$y_{D,evap} = \frac{\dot{E}_{D,evap}}{W_{elec}} \tag{5.32}$$

e) Motor:

$$y_{D,motor} = \frac{\dot{E}_{D,motor}}{W_{elec}} \tag{5.33}$$

**5.2.7 Cost of Operating**

$$Cost_{operating} = \sum \left( Cost_{annual}[i] * 1[yr], i = 1, n \right) \tag{5.34}$$

$$Cost_{annual}[i] = C_{electricity}[i] * W_{elec} * cylcle_{duty} \tag{5.35}$$

$$C_{electricity}[i] = C_{electric} * \left(1 + Inflation * (i - 1)\right) \tag{5.36}$$

where n, $cylcle_{duty}$ and $C_{electric}$ denotes the system life, run time of compressor per year and electricity cost respectively.

**5.2.8 Levelized Electricity Cost**

$$C_{electric,\ L} = \frac{\left(\left(Cost_{annual}[1] * 1[yr]\right) * A/P_0\right)}{1 + Inflation} \tag{5.37}$$

where A is annuity and $P_0$ is the constant value at the beginning of first year.

$$\frac{A}{P_0} = \frac{k * (1 - k^n)}{(1 - k) * CRF} \tag{5.38}$$

where CRF is capital recovery factor.

$$CRF = \frac{i_{eff} * \left(1 + i_{eff}\right)^n}{\left(1 + i_{eff}\right)^n - 1} \tag{5.39}$$

$$k = \frac{1 + r_n}{1 + i_{eff}} \tag{5.40}$$

$$r_n = ((1 + r_r) * (1 + r_{inf})) - 1 \tag{5.41}$$

where $r_n$, $r_r$ and $r_{inf}$ denotes nominal or apparent escalation rate, real escalation rate and inflation rate respectively.

### 5.2.9 Fuel Cost

It is the ratio of cost rate of fuel associated with a component to the exergy rate of fuel associated with that particular component.

For various components it is given by following formulae:

a) Compressor:

$$c_{fuel,comp} = \frac{\dot{C}_{W2}}{W_{comp}} \tag{5.42}$$

b) Condenser:

$$c_{fuel,cond} = \frac{\dot{C}_2}{\dot{E}_2} \tag{5.43}$$

c) Expansion Valve:

$$c_{fuel,exp} = \frac{\dot{C}_3}{\dot{E}_3} \tag{5.44}$$

d) Evaporator:

$$c_{fuel,evap} = \frac{\dot{C}_4 - \dot{C}_9}{\dot{E}_4 - \dot{E}_9} \tag{5.45}$$

e) Motor:

$$c_{fuel,motor} = \frac{\dot{C}_{W1}}{W_{motor}} \tag{5.45}$$

### 5.2.10 Cost of Exergy Destruction ($\dot{C}_D$)

For various components it is given by following formulae:

a) Compressor:

$$\dot{C}_{D,comp} = c_{fuel,comp} * \dot{E}_{D,comp} \tag{5.46}$$

b) Condenser:

$$\dot{C}_{D,cond} = c_{fuel,cond} * \dot{E}_{D,cond} \tag{5.47}$$

c) Expansion Valve:

$$\dot{C}_{D,exp} = c_{fuel,exp} * \dot{E}_{D,exp} \tag{5.48}$$

d) Evaporator:

$$\dot{C}_{D,evap} = c_{fuel,evap} * \dot{E}_{D,evap} \tag{5.49}$$

e) Motor:

$$\dot{C}_{D,motor} = c_{fuel,motor} * \dot{E}_{D,motor} \tag{5.50}$$

f) System Cost of Exergy Destruction:

$$\dot{C}_{D} = \dot{C}_{D,comp} + \dot{C}_{D,cond} + \dot{C}_{D,exp} + \dot{C}_{D,evap} + \dot{C}_{D,motor} \tag{5.51}$$

where $c_{fuel,component}$ denote the fuel cost associated with the component.

### 5.2.11 Exergoeconomic Factor

For various components it is given by following formulae:

a) Compressor:

$$f_{comp} = \frac{\dot{Z}_{comp}}{\dot{Z}_{comp} + \dot{C}_{D,comp}} \tag{5.52}$$

b) Condenser:

$$f_{cond} = \frac{\dot{Z}_{cond}}{\dot{Z}_{cond} + (c_{fuel,cond} * (\dot{E}_{D,cond} + \dot{E}_{loss}))} \tag{5.53}$$

c) Expansion Valve:

$$f_{exp} = \frac{\dot{Z}_{exp}}{\dot{Z}_{exp} + \dot{C}_{D,exp}} \tag{5.54}$$

d) Evaporator:

$$f_{evap} = \frac{\dot{Z}_{evap}}{\dot{Z}_{evap} + \dot{C}_{D,evap}} \tag{5.55}$$

e) Motor:

$$f_{motor} = \frac{\dot{Z}_{motor}}{\dot{Z}_{motor} + \dot{C}_{D,motor}} \tag{5.56}$$

where $\dot{Z}_{component}$ stands for sum of capital and operation & maintenance cost of component [82].

### 5.2.12 Cost Importance

For various components it is given by following formulae:

a) Compressor:

$$ZC_{comp} = \dot{Z}_{comp} + \dot{C}_{D,comp} \tag{5.57}$$

b) Condenser:

$$ZC_{cond} = \dot{Z}_{cond} + \dot{C}_{D,cond} \tag{5.58}$$

c) Expansion Valve:

$$ZC_{exp} = \dot{Z}_{exp} + \dot{C}_{D,exp} \tag{5.56}$$

d) Evaporator:

$$ZC_{evap} = \dot{Z}_{evap} + \dot{C}_{D,evap} \tag{5.57}$$

e) Motor:

$$ZC_{motor} = \dot{Z}_{motor} + \dot{C}_{D,motor} \tag{5.58}$$

## 5.3 RESULTS AND DISCUSSION

REFPROP [74] is used to calculate thermodynamic properties of refrigerants and EES [70] is used to make a computational model for thermoeconomic analysis of the system as shown in Appendix-II. The fuel-product-loss definition is shown in Table 2. The set of data defining the base operating conditions for the present analysis are:

- Steady state condition in all the components
- Kinetic as well as potential energy losses are not considered
- Pressure losses in the pipelines are neglected
- Number of compounding periods in one year is one
- System life 10 years [93]
- Interest rate 13%
- Inflation rate 4% [123] and real escalation rate 2% [123]
- Effective cost or value of money 12% [24]
- Operation time of the system is 8 hours per day
- Ambient temperature 32ºC
- Saturated evaporator and condenser temperature are -20ºC and 45ºC respectively
- Degree of superheating and subcooling are 5.8ºC and 3.5ºC respectively [99]

- Cost of electricity consumption 0.085 \$/kW-hr (Rs.6/kW-hr) [123]

Table 5.2 Fuel Product Loss definition

| Component | Fuel | Product | Loss |
|---|---|---|---|
| Electric Motor | $\frac{W_{com}}{\eta_{elec}}$ | $W_{com}$ | - |
| Compressor | $W_{com}$ | $\dot{E}_2 - \dot{E}_1$ | - |
| Condenser | $\dot{E}_2$ | $\dot{E}_3$ | $\dot{E}_8 - \dot{E}_7$ |
| Expansion valve | $\dot{E}_3$ | $\dot{E}_4$ | - |
| Evaporator | $\dot{E}_4 - \dot{E}_9$ | $\dot{E}_6 - \dot{E}_5$ | - |

In Figure 5.2, the variation in the value of operating cost with a change in the refrigerant is shown. A good refrigerant must have a lower value of operating cost. The lowest cost of operating is for refrigerant R3 (i.e. R600a) and maximum value is for R6 (i.e. 10%R134a/90%R1234yf).

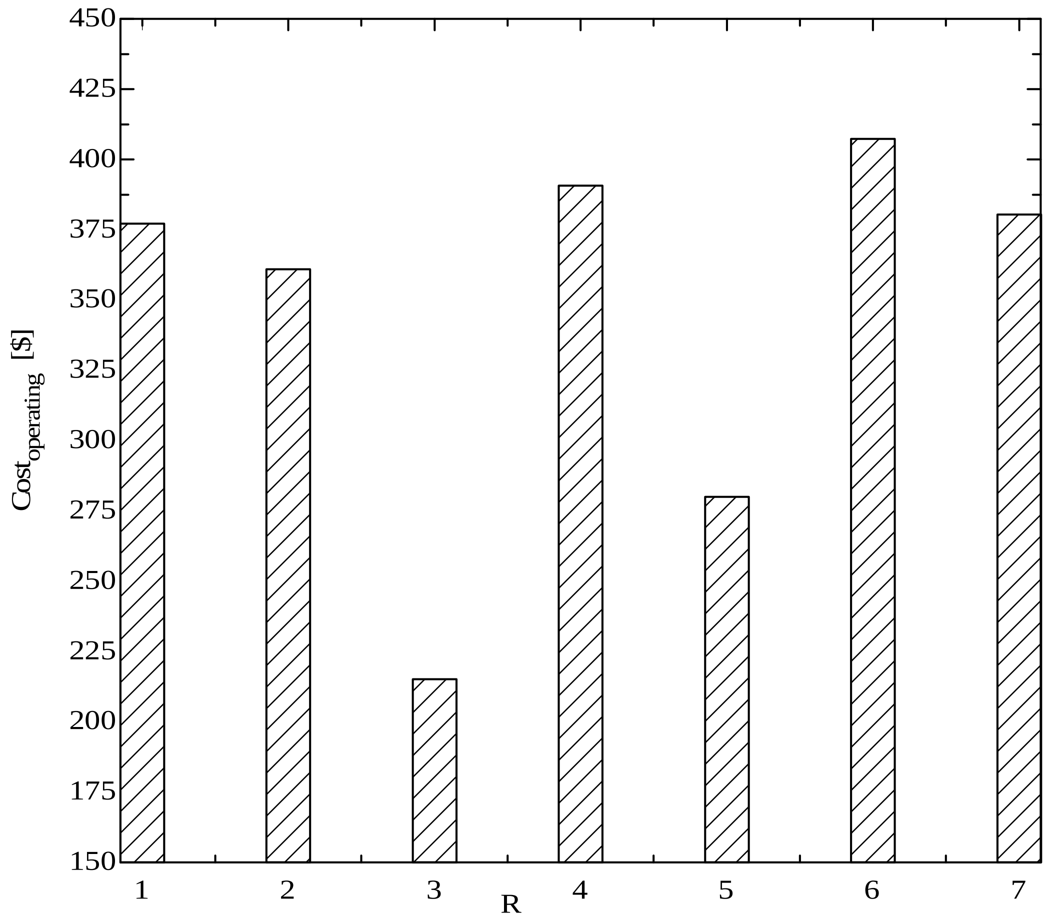



Figure 5.2 Variation in cost of operating with refrigerant

In figure 5.3, the variation in levelized cost with a change in the refrigerant is shown. A good refrigerant must have a lower value of levelized electricity cost. The lowest levelized cost is for refrigerant R3 (i.e. R600a) and maximum is for R6 (i.e. 10%R134a/90%R1234yf).

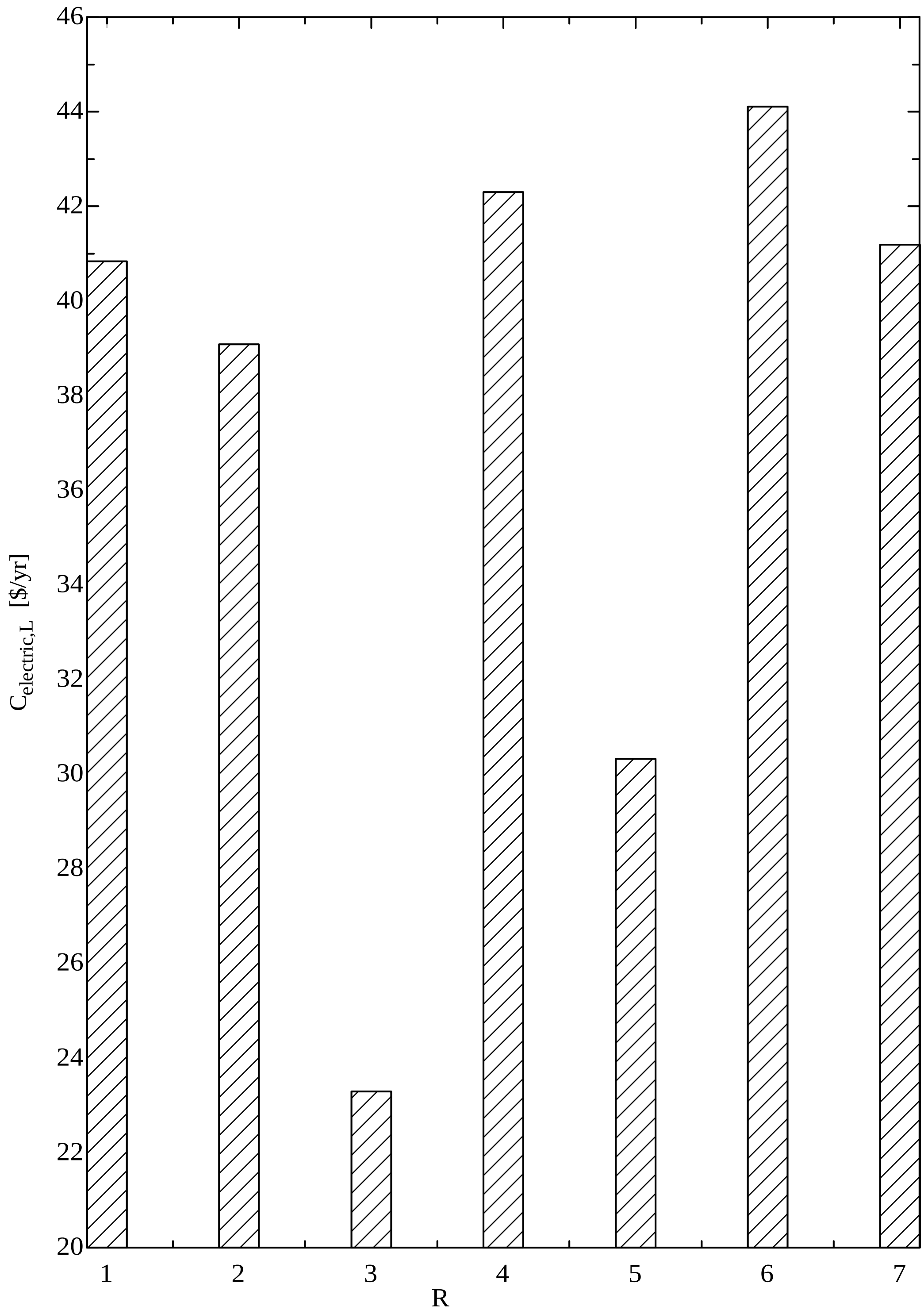



Figure 5.3 Variation of levelized cost with refrigerant

Figure 5.4 shows the cost of exergy destruction per year variation with change in refrigerant. Lower the value of the cost of exergy destruction associated with the refrigerant, lesser will the loss of cost due to exergy destruction. Refrigerant R3 (i.e. R600a) has the lowest value and R6 (i.e. 10%R134a/90%R1234yf) has the highest value of cost of exergy destruction.

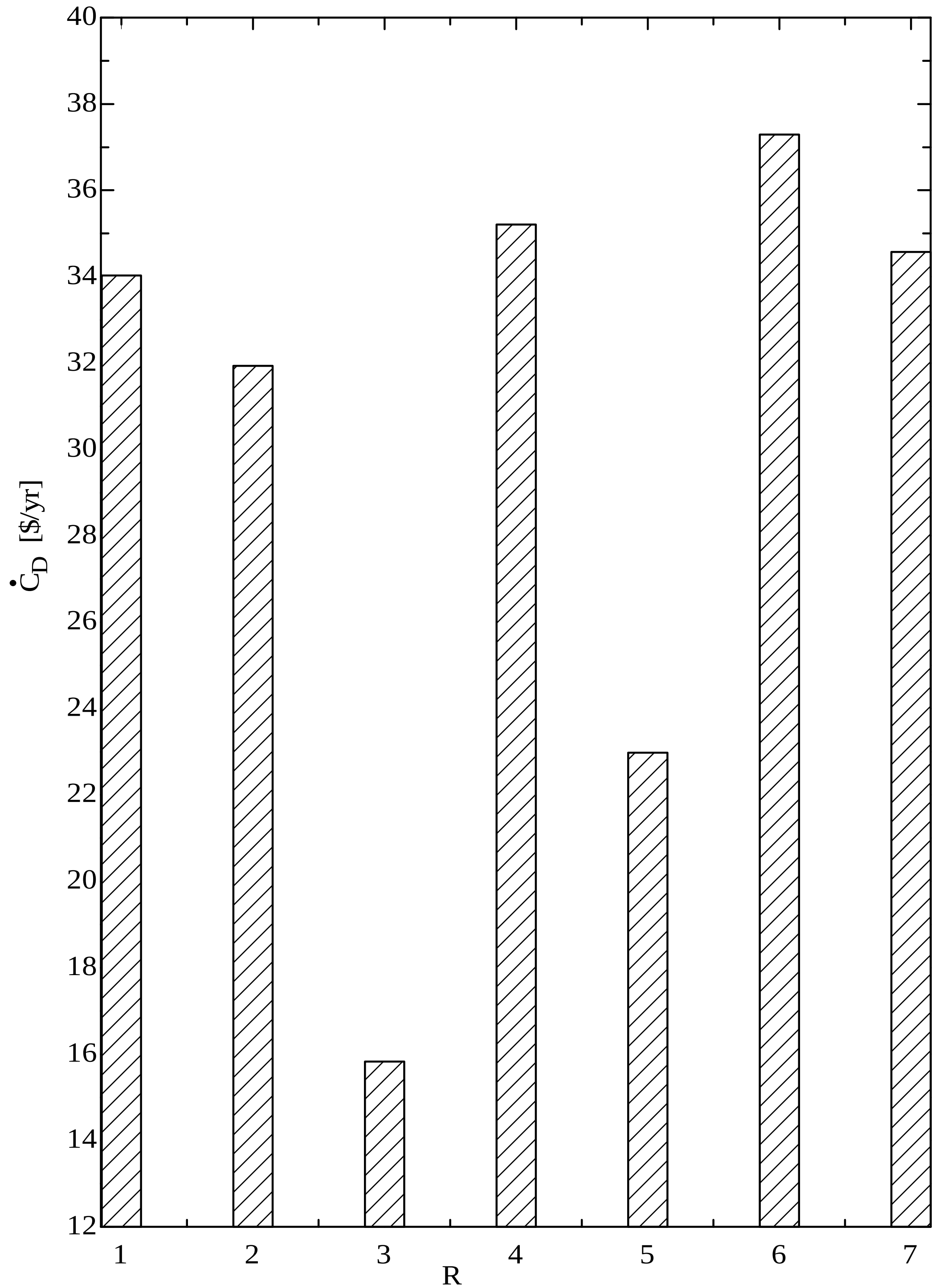



Figure 5.4 Variation of cost of exergy destruction with refrigerant

Figure 5.5 shows the exergetic efficiency variation with change in refrigerant. Exergetic efficiency is the ratio of exergy of product to the exergy of fuel. The maximum value of exergetic efficiency is for refrigerant R3 (i.e. R600a) and lowest value is for R6 (i.e. 10%R134a/90%R1234yf).

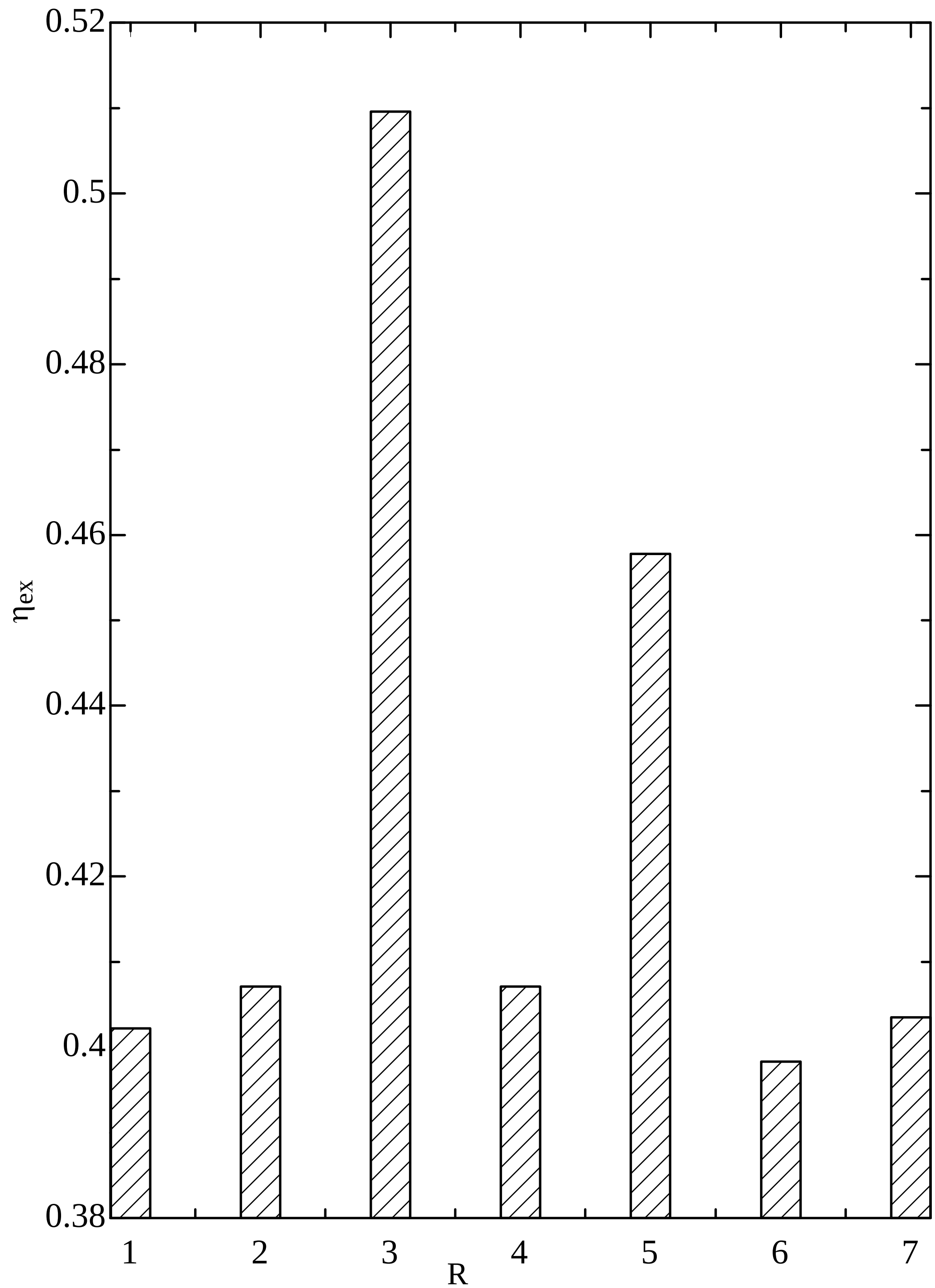



Figure 5.5 Variation of exergetic efficiency of system with refrigerant

Figure 5.6 represents a variation of energy efficiency ratio (EER) with a change in refrigerants. EER indicates the performance of the refrigerating machine. A refrigerator with high EER will have a lesser consumption of energy. Among all considered refrigerates R2 (i.e. R152a) has the highest value and R3 (i.e. R600a) has the lowest value of EER.

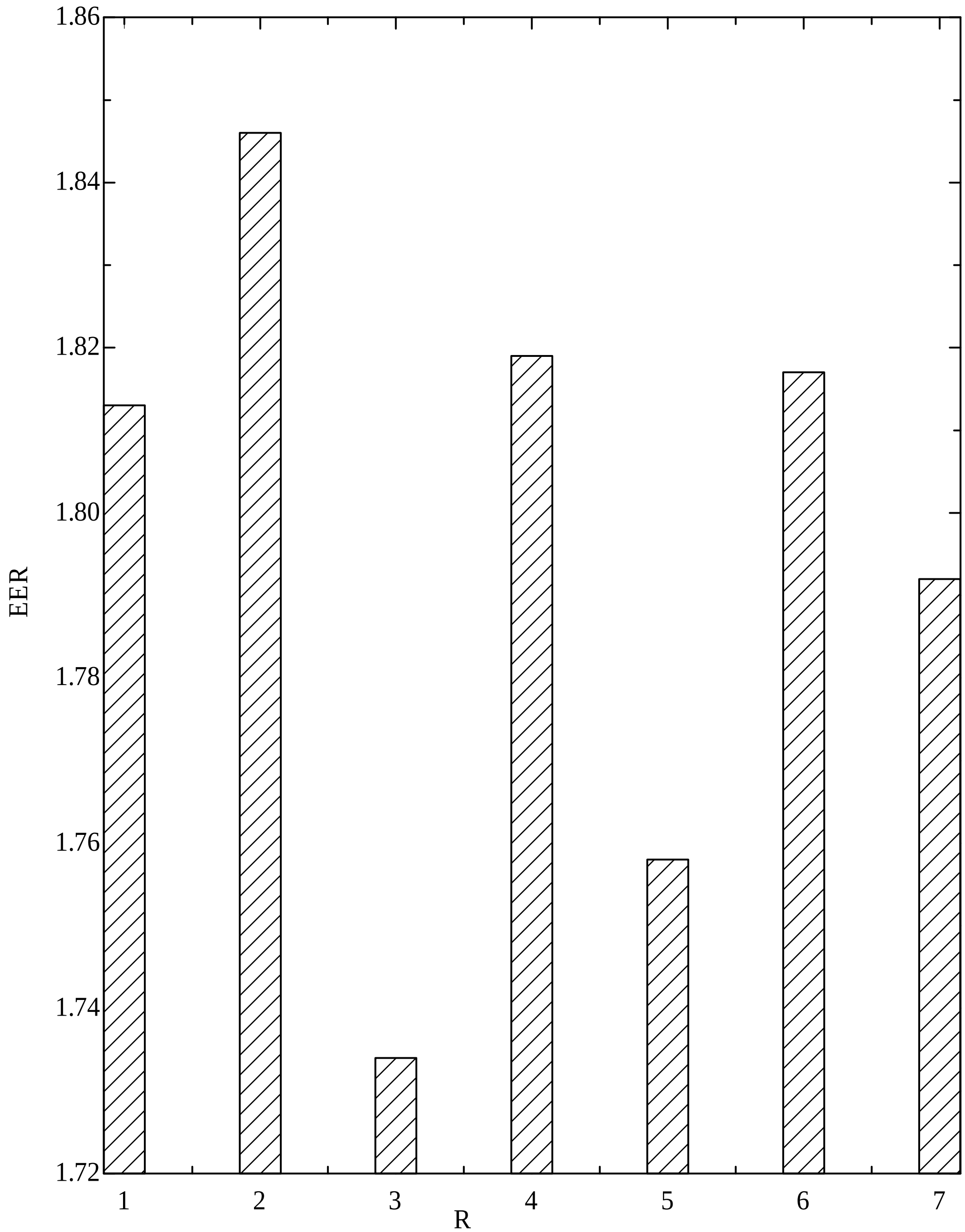



Figure 5.6 Variation of EER with refrigerant

Figure 5.7 shows the variation of relative volumetric cooling capacity with a change in refrigerants. No change in refrigerator compressor size is required if refrigerants are having same volumetric cooling capacity. It was observed that R7 (i.e. 40%R134a/22%R1234yf/38%R1234ze) is having nearly the same value of volumetric cooling capacity as that of R1 (i.e. R134a). Hence, if we replace R1 with refrigerant R7 then there is no change in refrigerator size is required.

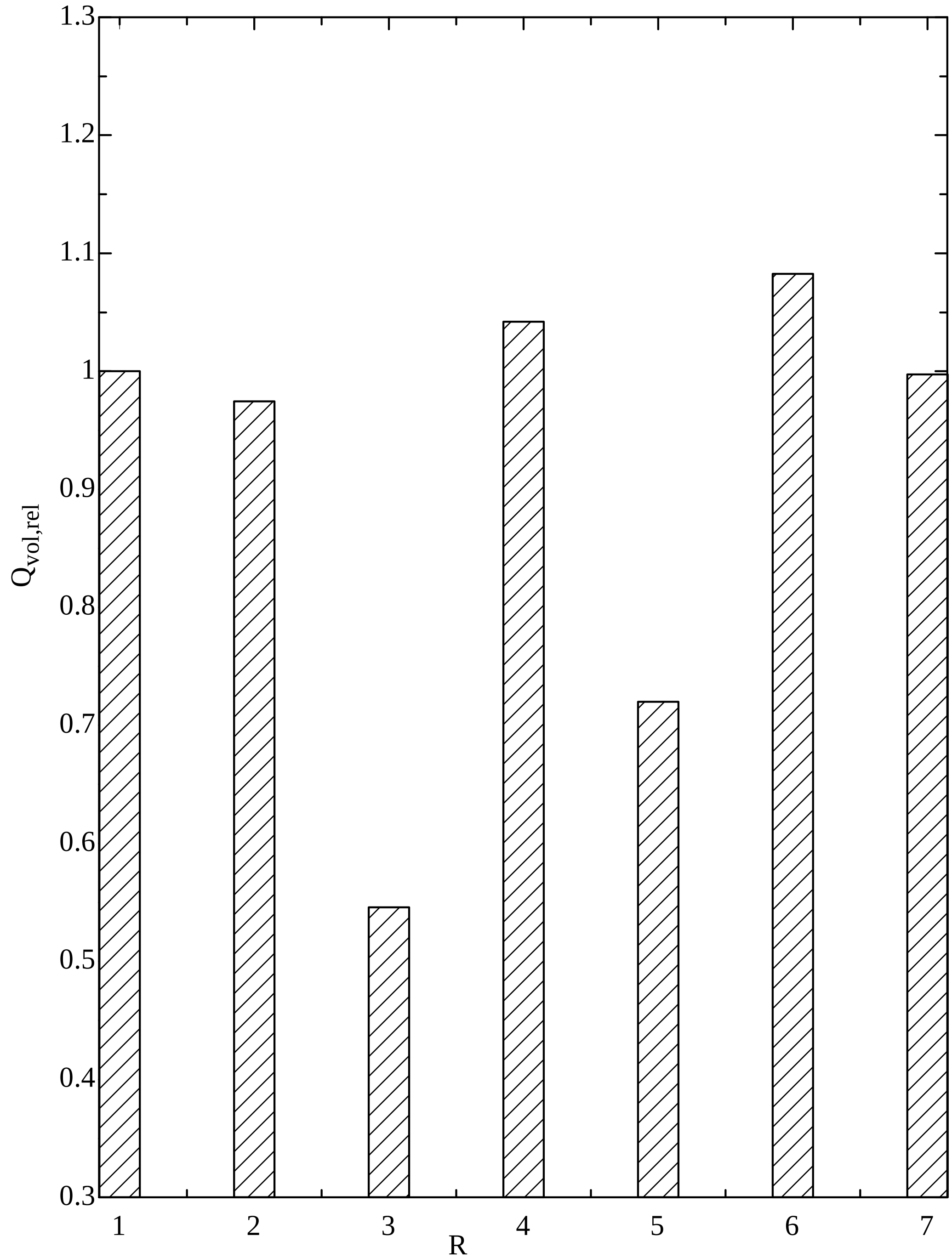



Figure 5.7 Variation of relative volumetric cooling capacity with refrigerant

Figure 5.8 depicts a variation of exergy destruction of components for the refrigerant R7 (i.e. 40%R134a/22%R1234yf/38%R1234ze). Exergy destruction represents the internal irreversibility of the system. It was observed that the compressor has the largest value of exergy destruction followed by evaporator, condenser, motor and expansion valve respectively.

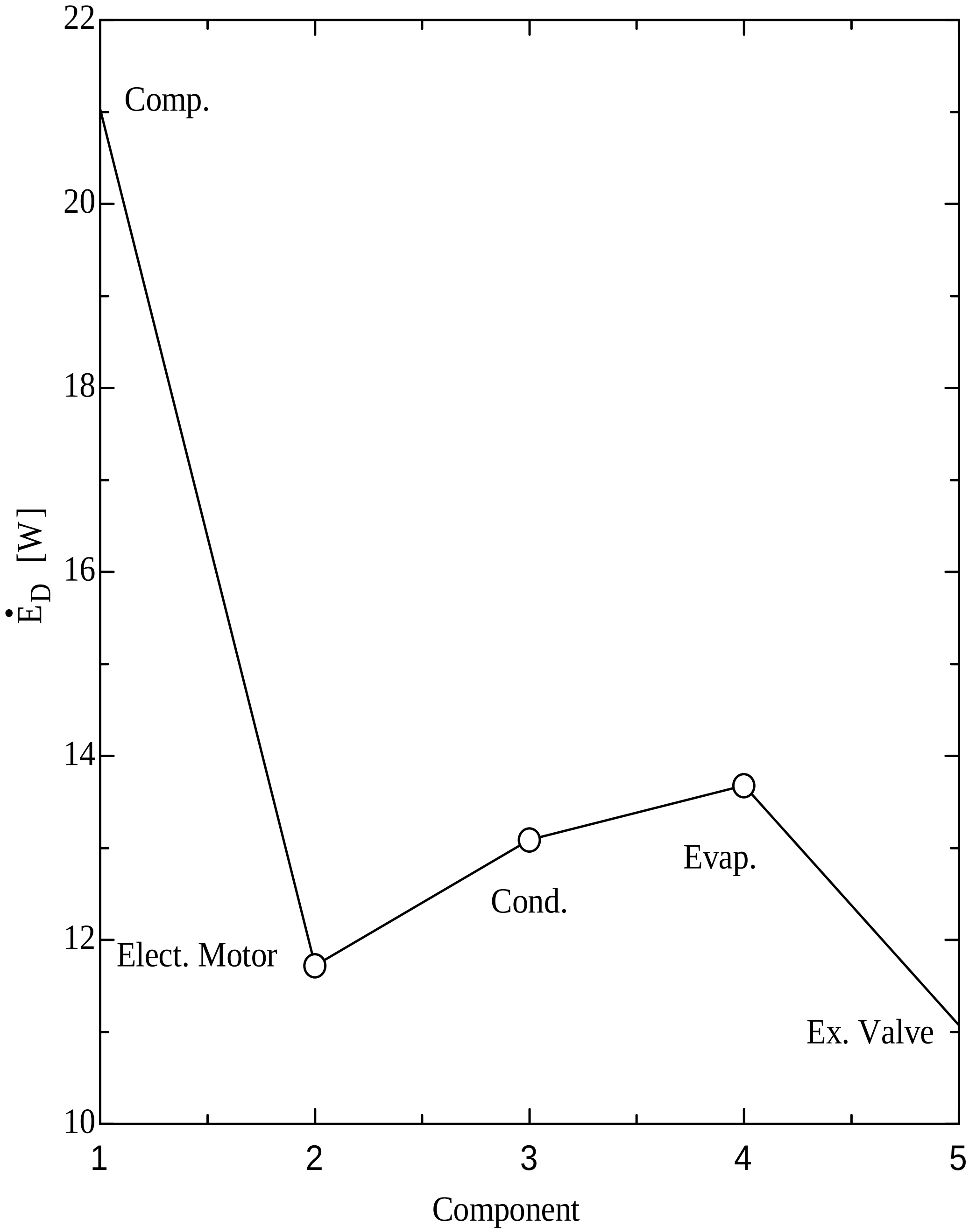



Figure 5.8 Variation of exergy destruction with component for R7 refrigerant

Figure 5.9 depicts a variation of cost of exergy destruction of components for the refrigerant R7 (i.e. 40%R134a/22%R1234yf/38%R1234ze). Cost of exergy destruction is the cost rate associated with exergy destruction. It was observed that the evaporator has largest value of cost of exergy destruction followed by evaporator, condenser, expansion valve and motor respectively.

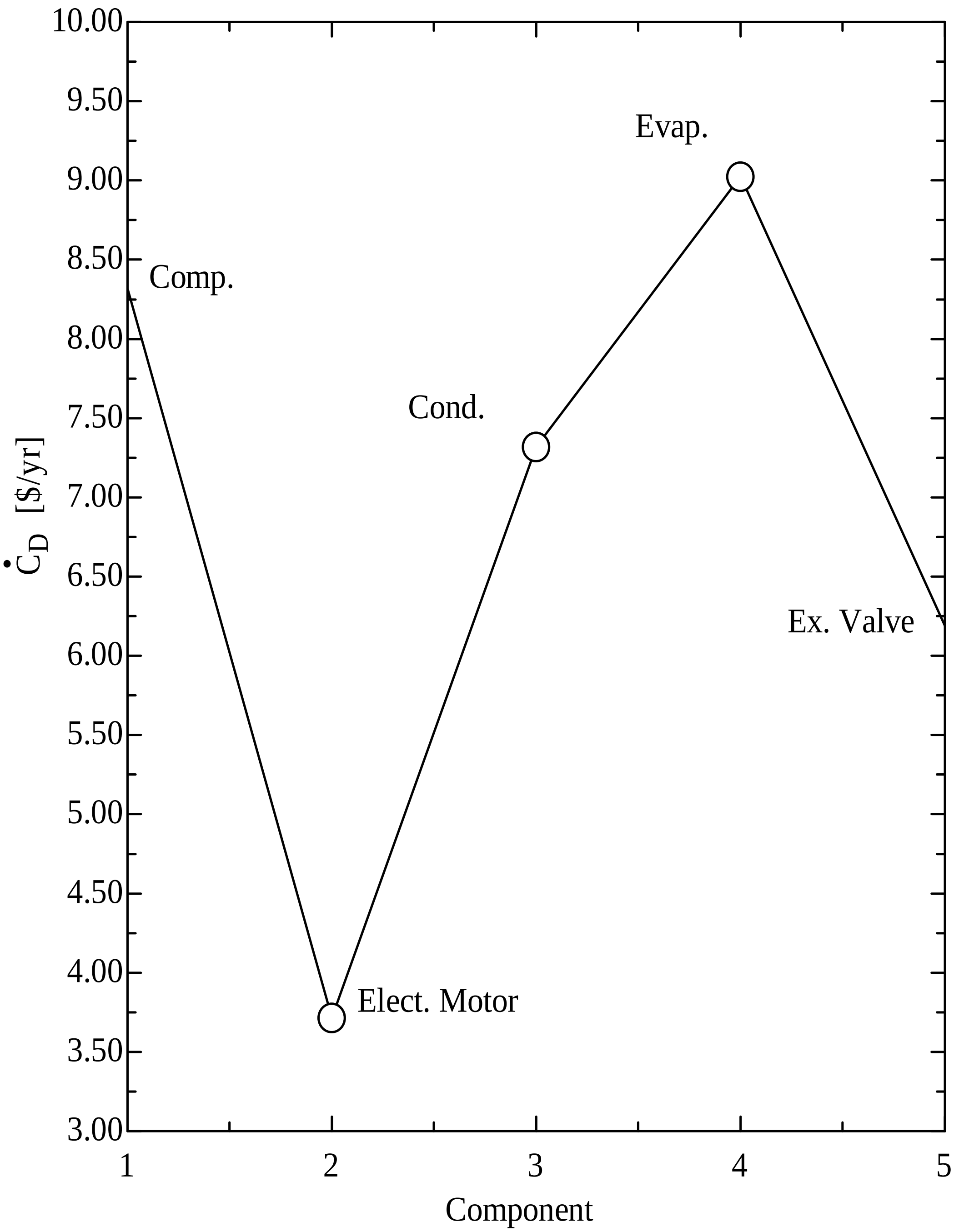



Figure 5.9 Variation of cost of exergy destruction with component for R7 refrigerant

Figure 5.10 shows the variation of an exergoeconomic factor of components for the refrigerant R7 (i.e. 40%R134a/22%R1234yf/38%R1234ze). Typical range of exergoeconomic factors for compressor and turbine is between 35 to 75% whereas for heat exchanger is lower than 55% [24]. It was observed that compressor and motor are within a typical range of 35 to 75% but condenser and evaporator value is lower than that specified. A low value of exergoeconomic value indicates low initial investment and high exergy destruction cost. So, more money could be spent.

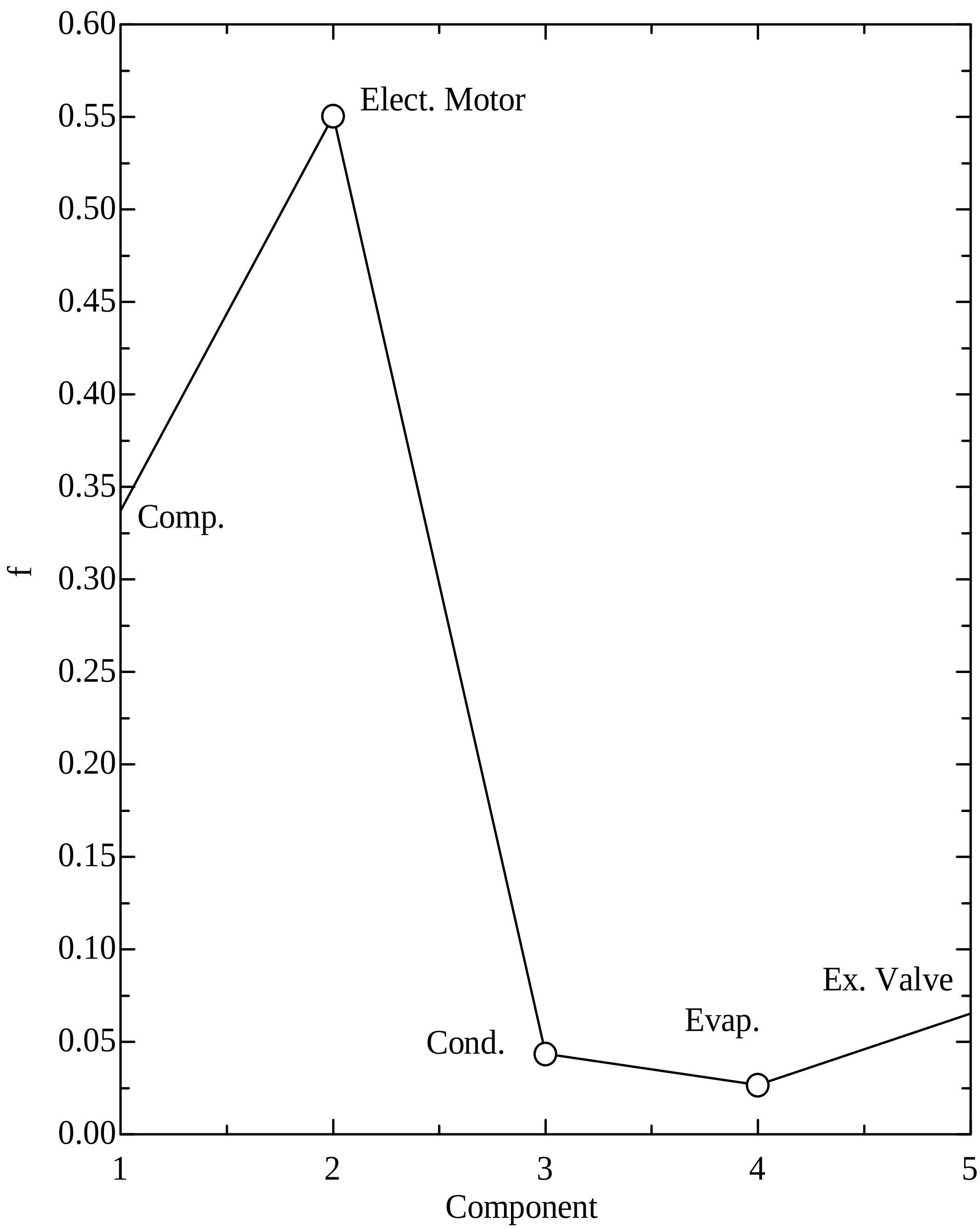



Figure 5.10 Variation of exergoeconomic factor with component for R7 refrigerant

Figure 5.11 depicts a variation in cost importance of components for R31 (i.e. R134a/R1234yf/R1234ze (40%/22%/38%)). Cost importance is one of the most important parameters in thermoeconomic evaluation. The decreasing order of cost importance obtained from Figure 5.11 is compressor followed by evaporator, motor, condenser and expansion valve respectively. Thus, compressor is the component which required maximum attention for improvement.

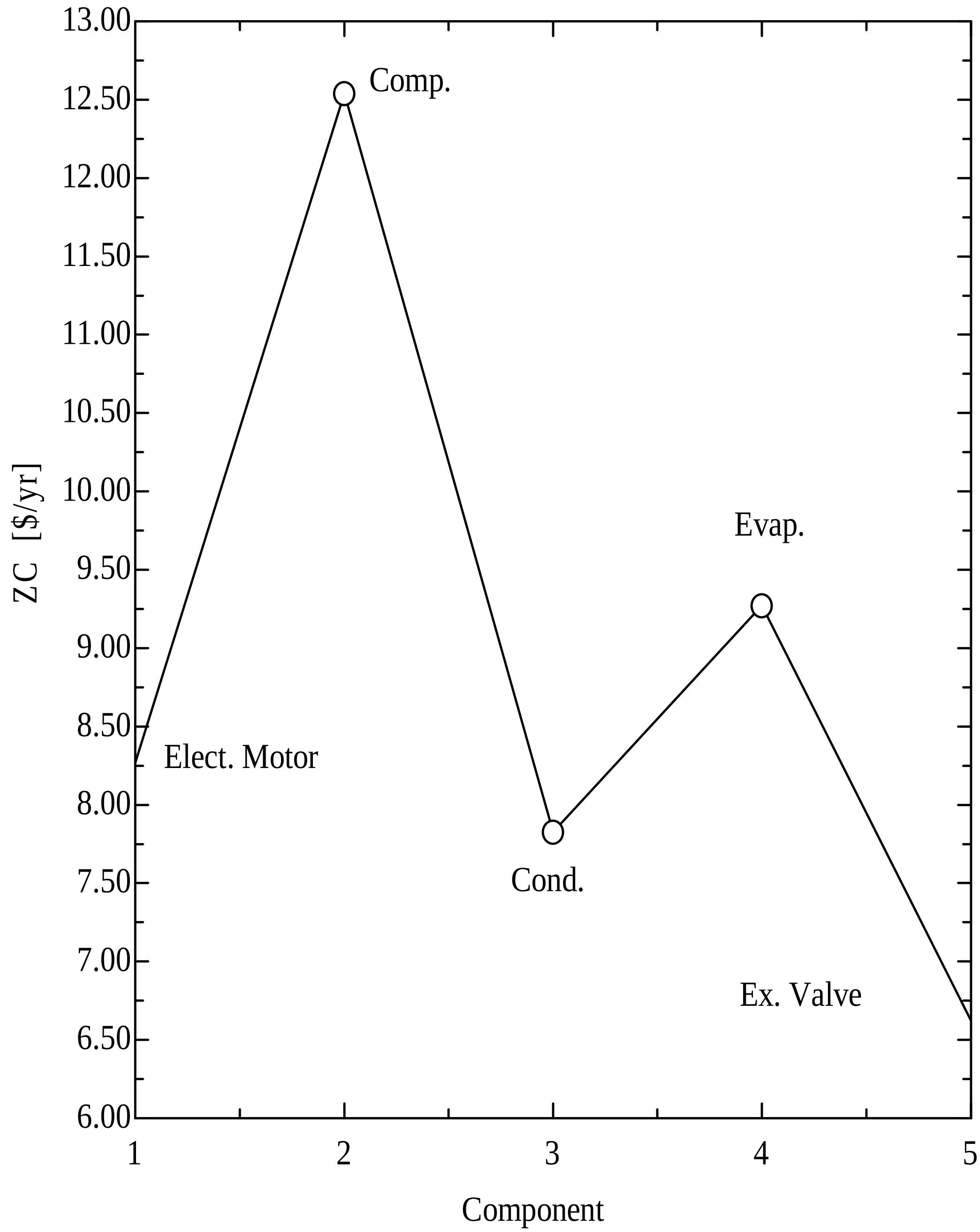



Figure 5.11 Variation of cost importance with component for R7 refrigerant

Figure 5.12 represents a variation of exergy destruction ratio with components for R7 (i.e. R134a/R1234yf/R1234ze (40%/22%/38%)). It is evident from the figure that exergy destruction ratio is highest for compressor followed by the evaporator, condenser, motor, and expansion valve respectively.

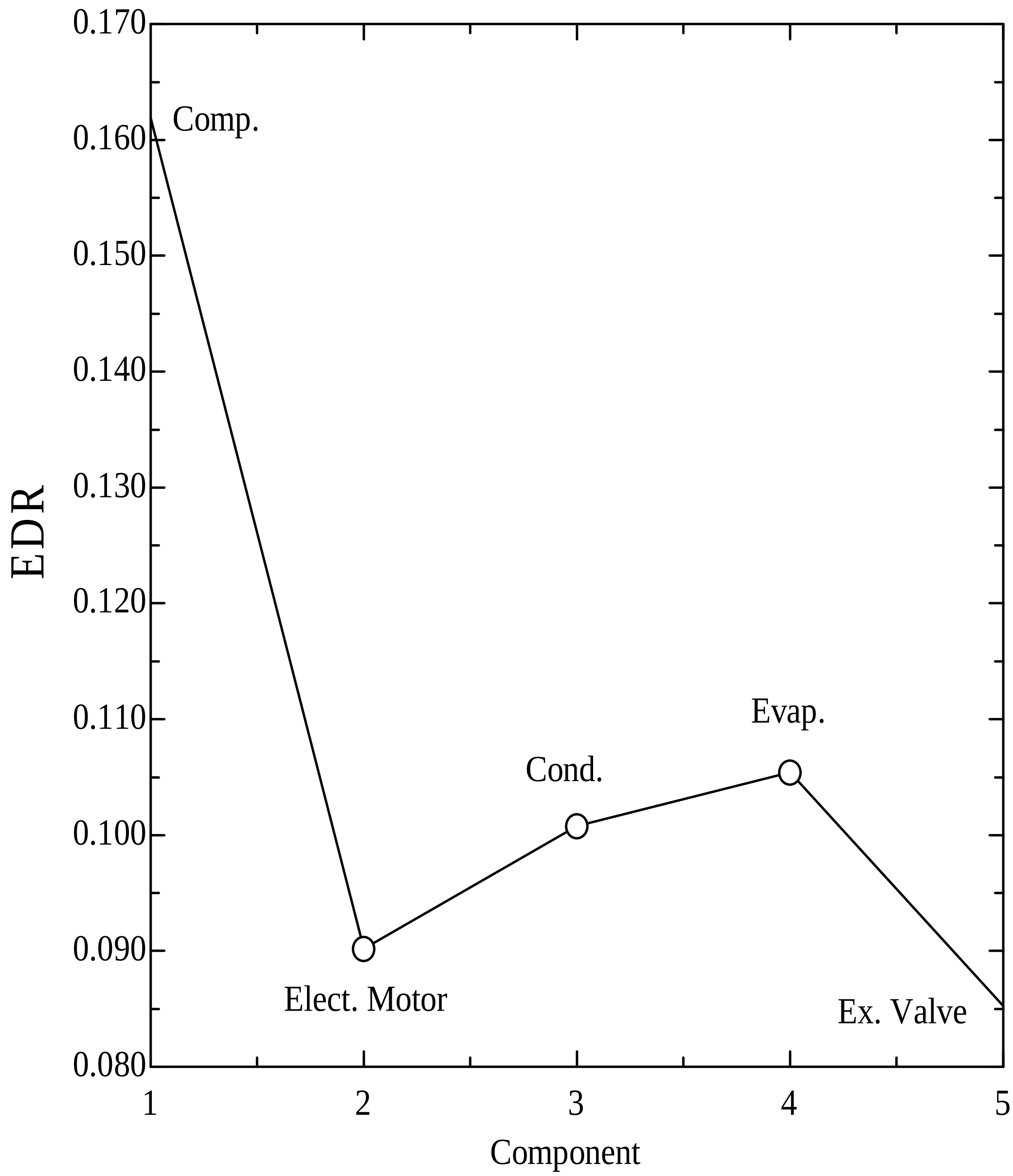



Figure 5.12 Variation of exergy destruction ratio with component for R7 refrigerant

The methodology to improve the cost effectiveness of a thermal system having several components [24] suggest that the compressor is the component which required maximum attention due to its highest value of cost importance and exergy destruction followed by the evaporator, condenser, motor and expansion valve respectively. Table

5.3 shows the performance parameter of refrigerants. Refrigerant R7 (i.e. R134a/R1234yf/R1234ze (40%/22%/38%)) has nearly same value of cost of operating, levelized electricity cost, cost of exergy destruction, exergetic efficiency, energy efficiency ratio and volumetric cooling capacity as that of R1 (i.e. R134a) among all considered refrigerants.

Table 5.3 Performance parameters of refrigerants

| Refrigerant | Naming | $Cost_{Operating}$ ($) | $C_{electric,L}$ ($/yr) | $\dot{C}_D$ ($/yr) | $\eta_{ex}$ (%) | EER | $Q_{vol.rel}$ |
|---|---|---|---|---|---|---|---|
| R134a | R1 | 377.10 | 40.84 | 34.03 | 40.22 | 1.81 | 1 |
| R152a | R2 | 361 | 39.09 | 31.94 | 40.71 | 1.84 | 0.9746 |
| R600a | R3 | 215.20 | 23.30 | 15.83 | 50.96 | 1.73 | 0.5458 |
| R1234yf | R4 | 390.70 | 42.30 | 35.21 | 40.71 | 1.82 | 1.0420 |
| R1234ze | R5 | 280 | 30.33 | 22.98 | 45.78 | 1.76 | 0.7199 |
| R134a/R1234yf (10%/90%) | R6 | 407.30 | 44.11 | 37.29 | 39.83 | 1.82 | 1.0827 |
| R134a/R1234yf/ R1234ze (40%/22%/38%) | R7 | 380.40 | 41.19 | 34.57 | 40.35 | 1.79 | 0.9974 |

Thermoeconomic analysis of the performance parameters of refrigerants was done to provide the drop-in substitute alternative of R134a in the vapour compression system. The proposed refrigerant HFC134a/HFO1234yf/HFO1234ze (40%/22%/38%) is having zero ODP, GWP around 600 and various performance parameters from thermoeconomic point of view are similar to that for R134a.

**5.4 VALIDATION OF RESULTS**

Theoretical and experimental results were validated through various published papers references on refrigeration. No work was reported for the refrigerant R134a/R1234yf/R1234ze (40%/22%/38%), but the author of the thesis has given the reference to his own work. Table 5.4 compares theoretical relative COP with experimental relative COP and it is found within the acceptable range.

**5.5 ANALYSIS OF ENVIRONMENTAL FRIENDLY REFRIGERANT MIXTURES FOR REPLACEMENT OF R134a**

The development of suitable refrigerants is one of the main parameters in defining the success of vapour compression refrigeration systems. They have zero ODP (ozone depleting potential) and very low GWP (global warming potential), but some of them are mildly flammable and vast research is going on to overcome this problem as they

Table 5.4 Validation of results

| Refrigerant | GWP (100 yrs) | Safety Group | $COP_{rel,theo}$ | $COP_{rel,exp}$ | Reference validation with respect to R134a |
|---|---|---|---|---|---|
| R134a | 1430 | A1 | 1 | 1 | - |
| R152a | 140 | A2 | 1.0233 | 1.0470 | Variation of 2.31% (Ref. Bolaji B. O., 2010, pp. 3796) [29] |
| R600a | 11 | A3 | 1.0429 | 1.0360 | Variation of 0.66 % (Ref. Borokinni F. O. et al., 2017, pp. 16) [31] |
| R1234yf | 4 | A2L | 0.9995 | 0.9621 | Variation of 3.47 % (Ref. Lee Yohan et al.,2012, pp. 242) [72] |
| R1234ze | 6 | A2L | 1.0080 | 0.9552 | Variation of 5.23 % (Ref. Mota-Babiloni Adrián et al., 2014, pp. 262) [93] |
| R134a/R1234yf (10%/90%) | 146.6 | | 0.9946 | 0.9828 | Variation of 1.19 % (Ref. Lee Yohan et al., 2013, pp. 1206) [73] |
| R134a/R1234yf/ R1234ze (40%/22%/38%) | 575.16 | A1 | 0.9879 | 0.9450 | Variation of 4.34 % (Ref. Gaurav et al., 2018) [50] |

are anticipated as next generation refrigerants. For the last few decades, work on finding new environmentally friendly alternative refrigerant is going on. However, the limited research work has been done on energy saving and safety aspects using refrigerant HFO1234yf and HC600a. Refrigerant R1234yf is an eco-friendly refrigerant which has a lower GWP value of 4 but it is costly and COP is found to be slightly lower than R134a. Refrigerant, R600a (GWP less than 20) has lower power consumption than R134a and average power consumption for refrigerants R1234yf is higher than R134a. In the present work, various relevant mixtures of refrigerants have been experimentally tested to establish a relationship between energy consumption, COP and cooling capacity.

**5.5.1 Material and Methods**

A conventional household refrigerator is shown in Figure 3.1 has been used for the present study, which was originally designed for R134a refrigerant with 95g charge. Joybari et al. (2013) illustrated that total exergy destruction in optimum condition with R600a was 45.05% of the base refrigeration R134a. Therefore, 43g of R600a is selected. It is also important to note that if R1234yf is used as a direct drop-in, the system will perform adequately without modification to any of the R134a baseline components and the system pressures will stay within acceptable limits and hence the same quantity of refrigerant as selected for R134a is taken for experimentation. Economically fixed cost of refrigerant R600a as obtained from Table 5.5 is Rs. 27.95 and it is lowest among the entire three refrigerants. Its ODP value is zero and GWP value is very less. All experiments are conducted in a test room, under standard conditions (32$^0$C ambient temperature and 50% relative humidity). Temperatures in 08 arbitrary points of the refrigerator are monitored and recorded continuously. Time was measured from the stopwatch and energy consumption is monitored and measured by an energy meter. In addition, the consumed voltage, current, working time and ON time of the compressor are recorded.

Table 5.5 General comparison of R134a, R600a and R1234yf

| | R1234yf | R600a | R134a |
|---|---|---|---|
| Quantity (g) for 190 L Refrigerator | 95 | 43 | 95 |
| Price Rs. Per Kg | 15500 | 650 | 450 |
| Charge Cost (Rs./Refrigerator) | 1472.5 | 27.95 | 42.75 |
| ODP | 0 | 0 | 0 |
| GWP | 4 | 11 | 1430 |

This study is carried out with the objective of reducing costs, flammability, improving thermal properties and reducing energy consumption using refrigerant mixtures. This research work explores the feasibility of the R134a/HC blend in an existing R134a refrigerator. Kim et al. (1994) worked on a compatible mixture for the refrigerants R134a/R600a/R290 (91/4.93/4.07 mass percentage). The mixture becomes miscible with the conventional mineral oil because of hydrocarbon presence and it also enhances the thermo-physical properties of R134a. This blend has proved to be a better retrofit refrigerant and improves the thermo-physical properties of R134a. Agarwal (1998) experimentally evaluated the HC mixture of R134a/R600a and found that the R600a/R134a exhibits higher system capacity than R134a. Douglas et al. (1996) proposed that flammable refrigerants can be mixed with non-flammable refrigerants to produce a non-flammable mixture. R1234yf has similar thermodynamic properties to the R134a however, it is expensive. Barroso-Maldonado et al. (2015) carried out an energy simulation to replace R134a with a mixture of R134a and R1234yf. Finally, the model is carried out to an energy simulation in order to predict the behaviour of different mass fractions of R1234yf and used R1234yf with a value of 0.9 so that GWP of the mixture is 150. Based on a literature survey and in order to lower GWP equal to 150 (or less) and thermo economic aspect, the following refrigerant mixtures are tested:

Reco1: Mixture of 90% R1234yf and 10% R134a in mass percentage

Reco2: Mixture of 91% of R600a and 9% R134a in mass percentage

Reco3: Mixture of R134a/R600a/R290 in 91/4.93/4.07 in mass percentage

**5.5.2 Results and Discussion**

As it was predicted, at the zero time (when the compressor starts) the power consumption for all three refrigerants was high but soon it starts decreasing. When interior cabin reaches around a preset temperature, the compressor is turned OFF and finally stops. The average power consumption during the compressor ON time, determine refrigerator energy consumption. It may be revealed that energy consumption slightly decreases as compared with R1234yf in case of Reco1 mixture. As R1234yf is costly, therefore, the cost of refrigerant is reduced by the addition of R134a while it maintains GWP mixture less than 150. Energy consumption is higher for Reco2 as compared to R600a (as shown in Table 5.6). Refrigerant Reco2 has an area under the curve equal to Reco3 which means same energy consumption (as shown in Figure 5.13). Isobutane (R600a) is found to be a viable additive due to its better thermo-physical properties, but its NBP is -11.73$^0$C. In the mixture of R600a and R134a, when R600a is added to R134a (NBP = -26.15$^0$C), the more volatile R134a will evaporate first and leave the mineral oil lubricant behind, as it is not miscible and the refrigerant R600a is less volatile in the evaporator. However, energy consumption of a mixture of R134a/R600a/R290 (91/4.93/4.07) in mass percentage is lower than R134a (as shown in Table 5.6). Two essential preconditions required for accidents are flammable mixture of gas & air and ignition source of energy/temperature. As a safety precaution, maximum refrigerant charge is set to be 150 g and ignition risk is very low for approx. 8 g/m$^3$, for a standard kitchen [49]. R600a is generally used in small quantities in refrigerator (20-50 g); therefore, it may be used with safety precautions. Flammable refrigerants can be mixed with non-flammable refrigerants to reduce the flammability tendency of mixture. If added in very small quantity in non-flammable refrigerant like in Reco3, the mixture will be inflammable. Thus this analysis provides the following information:

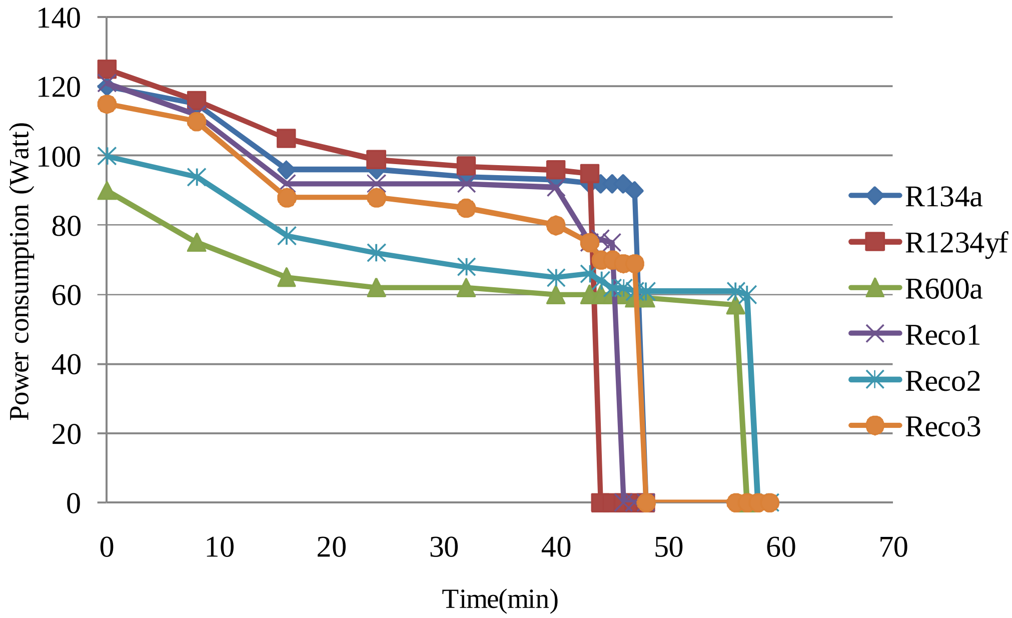



Figure 5.13 Power consumption for refrigerant mixture

Table 5.6 Comparison of various mixtures of R134a, R600 and R1234yf

| Property | R134a | Reco3 | R1234yf | Reco1 | R600a | Reco2 |
|---|---|---|---|---|---|---|
| COP | 2.04 | 2.10 | 1.98 | 2.05 | 2.28 | 2.12 |
| Energy consumption (Wh) | 75.7 | 65.30 | 76.90 | 68.23 | 59.23 | 65.35 |

- Refrigerant R600a is the most thermo economical refrigerant among the selected three refrigerants but it is flammable.
- The maximum difference in terms of COP and cooling capacity is less than 3% for R1234yf as compared to R134a. Therefore R1234yf is used as a direct drop-in, the system will perform adequately without modification to any of the R134a baseline components and the system pressures will stay within acceptable limits.
- Refrigerant R600a power consumption is 21.75 % lower than R134a. Average

power consumption for refrigerants R1234yf is 1.59 % higher than R134a.

- Amount of R600a used in refrigerator is less as compared to other alternative refrigerants. So, it may be used in a household refrigerator with safety precautions.
- It may be revealed that energy consumption slightly decreases as compared with R1234yf in case of Reco1 mixture. As R1234yf is costly, therefore, the cost of refrigerant is reduced by the addition of R134a while it maintains GWP of the mixture equal to 150. Energy consumption is higher for Reco2 as compared with R600a. Refrigerant Reco2 has an area under the curve equal to Reco3 which means same energy consumption. However, energy consumption of mixture of R134a/R600a/R290 (91/4.93/4.07) in mass percentage is lower than R134a.

## 5.6 SUMMARY

This chapter provides us with the drop-in substitute alternative of R134a in a vapour compression system by applying thermoeconomic analysis. The proposed refrigerant HFC134a/HFO1234yf/HFO1234ze (40%/22%/38%) is having zero ODP, GWP around 600, performance parameter from energy, exergy and thermoeconomic point of view are similar as that R134a but COP value slightly lesser. A study is carried out on an environment-friendly refrigerant with the objective of reducing costs, flammability, improving thermal properties and reducing energy consumption. The quantity of R600a, R600a in Reco2 mixture and R600a in Reco3 mixture used in the refrigerator is lesser as compared to other alternative refrigerants. So, these may be used safely in a household refrigerator with safety precautions.

# CHAPTER 6

# CONCLUSIONS AND SCOPE FOR FURTHER WORK

In this chapter, the contributions of the present research are discussed. The study focused on finding an alternative of HFC134a from energy, exergy and thermoeconomic point of view followed by experimental verification. The refrigerants used are HFC, HCs, HFOs and their mixtures. On the basis of present work, various conclusions, limitations and scope for future work are being presented in this chapter.

## 6.1 CONCLUSIONS

The research work dealt with the problems which are faced due to high GWP value of HFC, flammability of HC and high cost & mild flammability of HFOs. The conclusions of this work remove the demerits associated with earlier refrigerants by finding a suitable refrigerant. The following section will summarize the major findings in the research work:

- Based on performance parameters, refrigerants which come out as an alternative to R134a are: R152a, R134a/R1234yf (90%/10%), R134a/R1234yf/R1234ze (42%/18%/40%), R134a/R1234ze (42%/58%) and R134a/R1234yf/R1234ze (40%/22%/38%).
- R290 has largest value of COP, exegetic efficiency and lowest value of efficiency defect but still it cannot be used as a drop-in replacement of R134a because of its high value of volumetric cooling capacity and flammability.
- Refrigerant mixture R134a/R1234yf/R1234ze (40%/22%/38%) has nearly the same volumetric capacity and cooling capacity as that of baseline refrigerant R134a and it has low flammability.
- With the increase in mass percentage of R1234yf in the mixture R134a/R1234yf and R134a/R1234yf/R1234ze (keeping mass percentage of R134a constant), the value of mass flow rate, volumetric cooling capacity, cooling capacity and exergy destruction increases whereas pressure ratio decreases.
- R134a/R1234yf (10%/90%) has the lowest value of exergetic efficiency of the system whereas R600a has largest value.
- R600a has the lowest value of cost of operation, cost of exergy destruction, levelized electricity cost, energy efficiency ratio, total exergy destruction of the

system, power consumption and cooling capacity whereas R134a/R1234yf (10%/90%) has largest value except for energy efficiency ratio.

- As a safety precaution, maximum refrigerant charge is set to be 150 g and ignition risk is very low for approx. 8 $g/m^3$, for a standard kitchen. R600a is used in small quantities in refrigerator; therefore, it must be used with safety precautions.
- R600a/R134a (91%/9%) provides thermoeconomical solution with GWP less than 150.
- The value of exergoeconomic factor for compressor & electric motor is within the prescribed limits but for condenser and evaporator, it is lower than the prescribed limit.
- The compressor has the largest value of cost importance & exergy destruction whereas the expansion valve has lowest value of cost importance & exergy destruction.
- R1234yf and Reco1 (i.e. 10%R134a/90%R1234yf) refrigerants are costly and mildly flammable and hence find limited use at present but these are future refrigerants with low GWP.
- R134a/R1234yf (10%/90%) has the lowest value of pressure ratio whereas R1234ze has largest value.
- Reduced quantity of R1234yf refrigerant with changes in lower compression ratio, with efficient condenser and evaporator, will maintain earlier COP and refrigerating effect. It will also reduce flammability as the temperature and pressure encountered in the system is lesser.
- Optimization of capillary tube parameters provided improvement up to 5.5% in COP of HFO/HFC mixture.
- The methodology to improve the cost effectiveness of a thermal system having several components suggest that compressor is the component which required maximum attention due to its highest value of cost importance and exergy destruction followed by the evaporator, condenser, motor and expansion valve respectively.
- A study has been carried out on environmental friendly refrigerants with the objective of reducing costs, flammability, improving thermal properties and reducing energy consumption. Quantity of R600a, R600a in Reco2 mixture and R600a in Reco3 mixture used in refrigerator is lesser as compared to other

alternative refrigerants. So, these may be used safely in a household refrigerator with safety precautions.

- Based upon thermodynamic and thermoeconomic analysis, the refrigerant mixture HFC134a/HFO1234yf/HFO1234ze (40%/22%/38%) is the drop-in replacement of HFC134a with GWP around 600.

## 6.2 RESEARCH LIMITATIONS

The research limitations of the present research work can be described as:

- The refrigerant pressure drop in heat exchanger and pipe are not considered in the present research work.
- Steady state condition is assumed in all the components.
- Kinetic as well as potential energy and exergy losses are not considered.

## 6.3 SCOPE FOR FUTURE WORK

The study work could be extended in the following direction for future work:

- The refrigerant pressure drop in heat exchanger can be incorporated with the help of enhanced model.
- A forced convection type condenser can be considered in which heat is transferred with the help of a fan.
- Cost structure used in the present work is taken from information available in open literature but a refined cost structure can be considered for obtaining more accurate optimized results.
- The research can be extended to implement in large tonnage systems for thermoeconomic performance.
- Thermoecnomic optimization for newly finds refrigerant mixture HFC134a/HFO1234yf/HFO1234ze (40%/22%/38%) can be performed.

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**APPENDIX-I**

# PROGRAM ON ENERGY AND EXERGY ANALYSIS OF VAPOUR COMPRESSION SYSTEM IN EES

```
T_evap = 248.15 [k]
P_evap = P_sat(refrigerant$, T=T_evap)
P_9 = P_evap-10
h_9 = enthalpy(refrigerant$, P=P_9, x=1)
s_9 = entropy(refrigerant$, P=P_9, x=1)
P_10 = P_9
T_lvhx = 305.15 [K]
h_10 = enthalpy(refrigerant$, P=P_10, T=T_lvhx)
s_10 = entropy(refrigerant$, P=P_10, T=T_lvhx)
Nu_10 = volume(refrigerant$, P=P_10, T=T_lvhx)
P_1 = P_10-10
h_1 = enthalpy(refrigerant$, P=P_1, T=T_lvhx)
s_1 = entropy(refrigerant$, P=P_1, T=T_lvhx )
C_p = CP(refrigerant$, P=P_1, T=T_lvhx)
C_v = CV(refrigerant$, P=P_1, T=T_lvhx)
Gamma = C_p/C_v
T_c = 328.15 [k]
P_c = P_sat(refrigerant$, T=T_c)
P_2s = P_c+25
s_2s = s_1
h_2s = enthalpy(refrigerant$, P=P_2s, s=s_2s)
T_2s = temperature(refrigerant$, P=P_2s, s=s_2s)
Eta_is = 0.874-(0.0135*PR)
h_2n = h_1+((h_2s-h_1)/ Eta_is)
P_2n = P_2s
PR = P_2n/P_1
CCR = 0.04
Eta_vol_cl = 1-CCR*(PR^(1/Gamma)-1)
N = 3000 [rpm]
V_st = 0.00000664 [m^3]
```

```
m_dot_r = (V_st)*N*Eta_vol_cl/ (Nu_10*60)
T_2n = temperature(refrigerant$, P=P_2n, h=h_2n)
s_2n = entropy(refrigerant$, P=P_2n, h=h_2n )
h_3s = enthalpy(refrigerant$, P=P_c, T=T_2s)
s_3s = entropy(refrigerant$, P=P_c, T=T_2s )
h_3n = enthalpy(refrigerant$, P=P_c, T=T_2n)
s_3n = entropy(refrigerant$, P=P_c, T=T_2n )
h_4 = enthalpy(refrigerant$, P=P_c, x=1)
h_5 = enthalpy(refrigerant$, P=P_c, x=0)
T_6 = 316.15 [k]
h_6 = enthalpy(refrigerant$, P=P_c, T=T_6)
s_6 = entropy(refrigerant$, P=P_c, T=T_6)
h_7 = enthalpy(refrigerant$, P=P_c, T=T_lvhx)
s_7 = entropy(refrigerant$, P=P_c, T=T_lvhx)
h_8 = h_7
s_8 = entropy(refrigerant$, P=P_evap, h=h_8)
W_comp = m_dot_r *1000*(h_2n-h_1)
Q_evap = m_dot_r *1000*(h_9-h_8)
Q_vol = ((h_9-h_8)*Eta_vol_cl )/Nu_10
COP = Q_evap/W_comp
T_0 = T_6
T_r = T_evap+15
COP_rr = -(1/(1-(T_0/T_r)))
Eta_ex = COP/COP_rr
EDR = (1/ Eta_ex) -1
E_dot_D_comp = m_dot_r *1000*(T_0*(s_2n-s_10))
Delta_comp = E_dot_D_Comp/W_comp
E_dot_D_exp = m_dot_r *1000*((h_7-(T_0*s_7))-(h_8-(T_0*s_8)))
Delta_exp = E_dot_D_exp/W_comp
E_dot_D_cond = m_dot_r *1000*((h_3n-(T_0*s_3n))-(h_6-(T_0*s_6)))
Delta_cond = E_dot_D_cond/W_comp
E_dot_D_evap = (m_dot_r *1000*((h_8-(T_0*s_8))-(h_9-(T_0*s_9))))+(Q_evap *
                                        (1-(T_0/T_r)))
Delta_evap = E_dot_D_evap/W_comp
```

E_dot_D_lvhx = m_dot_r *1000*((h_6-h_7+h_9-h_10)-((T_0)*(s_6-s_7+s_9-s_10)))

Delta_lvhx = E_dot_D_lvhx/W_comp

Delta = Delta_comp+Delta_exp+Delta_cond+Delta_evap+Delta_lvhx

E_dot_D = E_dot_D_comp+E_dot_D_exp+E_dot_D_cond+E_dot_D_evap+
E_dot_D_lvhx

## APPENDIX-II

## PROGRAM ON THERMOECONOMIC ANALYSIS OF VAPOUR COMPRESSION SYSTEM IN EES

```
function FPRatio(i, m, n)
   "A = Annuity
    P = Present value
    F = Future value"
    FPRatio = (1+i/m)^(m*n)
end function
function AFRatio(i, m, n)
    AFRatio = (i/m)/((1+i/m)^(n*m)-1)
end function
function APRatio(i, m, n)
     APRatio = ((i/m)*(1+i/m)^(m*n))/((1 + i/m)^(m*n)-1)
end function
function AP0Ratio(n, k, CRF)
    AP0Ratio = (k*(1 - k^(n))/(1 - k))*CRF   "Constant-Escalation Levelization
                                                                Factor"
end function
function EscalationRate(r_r, r_inf)
    EscalationRate = (1+r_r)*(1+r_inf)-1   "Solving for r_n, the nominal or apparent
                                                          escalation rate"
end function
function kcalc(r_n, i_eff)
      kcalc = (1+r_n)/(1+i_eff)
end function
“! Economic Constants"
m = 1   "Number of compounding periods in one year"
n = 10 [yrs]   "Period of payment or system life"
i = 0.13   "Interest rate or discount rate"
“! Economic Constants"
r_r = 0.02   "Real escalation rate"
r_inf = 0.04   "Inflation rate"
i_eff = 0.12   "Effective cost or value of money"
r_n = EscalationRate(r_r, r_inf)
```

```
k = kcalc(r_n, i_eff)
CRF = (i_eff*((1+i_eff)^(n)))/(((1 +i_eff)^(n)) - 1)   "Capital Recovery Factor"
t = 10512000 [s/yr]   "Operation time"
evapfluid$ = 'air_ha'
condfluid$ = 'air_ha'
"! Boundary Conditions"
T[0] = 32 [C]
P[0] = 101.325 [kPa]
P_atm = P[0]
T[7] = T[0]
P[5] = P[0]
P[6] = P[5]
P[7] = P[0]
P[8] = P[7]
T_sat_evap = -20 [C]
T_sat_cond = 45 [C]
DELTAT_superheat = 5.8 [C]
DELTAT_subcool = 3.5 [C]
T[9] = T_sat_evap+DELTAT_superheat
T[3] = T_sat_cond-DELTAT_subcool
Epsilon_lvhx = 0.55   "Effectiveness of liquid vapour heat exchanger"
T[1] = (Epsilon_lvhx*T[3])+((1-Epsilon_lvhx)*T[9])
P_sat_cond = pressure(refrigerant$, T=T_sat_cond, x=0.5)
P_sat_evap = pressure(refrigerant$, T=T_sat_evap, x=0.5)
P[1] = P_sat_evap; P[2] = P_sat_cond; P[3] = P_sat_cond; P[4] = P_sat_evap; P[9] =
P_sat_evap
h[1] = enthalpy(refrigerant$, P=P[1], T=T[1])
h[3] = enthalpy(refrigerant$, P=P[3], T=T[3])
h[9] = enthalpy(refrigerant$, P=P[1], T=T[9])
h[4] = h[3] -h[1]+h[9] "Expansion valve"
T[4] = temperature(refrigerant$, P=P[4], h=h[4])
s[1] = entropy(refrigerant$, P=P[1], h=h[1])
s_s[2] = s[1]
h_s[2] = enthalpy(refrigerant$, P=P[2], s=s_s[2])
```

```
T_s[2] = temperature(refrigerant$, P=P[2], h=h_s[2])
w_s = h_s[2]-h[1]
PR = P_sat_cond/P_sat_evap
Eta_is = 0.874-(0.0135*PR)
N_r = 50 [rps]   "Compressor speed"
V_st = 0.00000664 [m^3]   "Swept volume"
Nu_1 = volume(refrigerant$, T=T[1], P=P[1])
C_p = CP(refrigerant$, T=T[1], P=P[1])
C_v = CV(refrigerant$, T=T[1], P=P[1])
Gamma = C_p/C_v
CCR = 0.04   "Compressor Clearance Ratio"
Eta_vol_cl = 1-CCR*(PR^(1/Gamma)-1)   "Clearance volumetric efficiency"
m_dot_r = (V_st*N_r*Eta_vol_cl)/(Nu_1)   "Refrigerant mass Rate"
Q_vol = ((h[9]-h[4])*(Eta_vol_cl))/(Nu_1)
W_comp = (m_dot_r*w_s)/Eta_is
Q_evap = m_dot_r*(h[9] - h[4])
h[2] = h[1] +((h_s[2]-h[1])/Eta_is)
T[2] = temperature(refrigerant$, P=P[2], h=h[2])
duplicate i = 2,4
        s[i] = entropy(refrigerant$, P=P[i], h=h[i])
end duplicate i
s[9] = entropy(refrigerant$, P=P[1], h=h[9])
Q_dot_cond = W_comp+Q_evap
m_dot_air_cond = 0.02602 [kg/s]
C_p_air_cond = specheat(condfluid$, T=T[7], P = P[7])
Q_dot_cond = m_dot_air_cond*C_p_air_cond*(T[8] - T[7])
C_p_air_evap = specheat(evapfluid$, T=T[5], P = P[5])
C_min_evap = m_dot_air_evap*C_p_air_evap
UA_evap = 0.020 [kW/k]
m_dot_air_evap = 0.02155 [kg/s]
NTU_evap = UA_evap/C_min_evap
epsilon_evap = 1 - exp(-NTU_evap)
Q_evap = epsilon_evap*C_min_evap*(T[5]- T_sat_evap)
Q_evap = m_dot_air_evap*C_p_air_evap*(T[5] - T[6])
```

```
Eta_motor = 0.9
W_motor = W_comp/Eta_motor
“! Air Side of Heat Exchangers: Thermal Properties"
“! Determine Evaporator Air Entropies"
“! Determine Evaporator Air Enthalpies"
duplicate i = 5,6
	h[i] = enthalpy(evapfluid$, T=T[i], P=P[i])
end duplicate i
“! Determine Condenser Air Enthalpies"
duplicate i = 7,8
	h[i] = enthalpy(condfluid$, T=T[i], P=P[i])
end duplicate i
“! Determine Evaporator Air Entropies"
duplicate i = 5,6
	s[i] = entropy(evapfluid$, T=T[i], P=P[i])
end duplicate i
“! Determine Condenser Air Entropies"
duplicate i = 7,8
	s[i] = entropy(condfluid$, T=T[i], P=P[i])
end duplicate i
“! Exergy"
"! Refrigerant exergy"
h_ref[0] = enthalpy(refrigerant$,T = T[0], P=P[0])
s_ref[0] = entropy(refrigerant$, T = T[0], P=P[0])
duplicate i = 1,4
	e[i] = (h[i]-h_ref[0])-converttemp(C,K,T[0])*(s[i] - s_ref[0])
	E_dot[i] = m_dot_r*e[i]
end duplicate i
e[9] = (h[9] - h_ref[0])-converttemp(C,K,T[0])*(s[9]-s_ref[0])
	E_dot[9] = m_dot_r*e[9]
"! Evaporator exergy"
h_evap[0] = enthalpy(evapfluid$,T=T[0], P=P[0])
s_evap[0] = entropy(evapfluid$, T=T[0], P=P[0])
duplicate i = 5,6
```

```
    e[i] = (h[i]-h_evap[0])-converttemp(C,K,T[0])*(s[i]-s_evap[0])
    E_dot[i] = m_dot_air_evap*e[i]
end duplicate i
"! Condenser exergy"
h_cond[0] = enthalpy(condfluid$,T = T[0], P = P[0])
s_cond[0] = entropy(condfluid$, T = T[0], P = P[0])
duplicate i = 7,8
    e[i] = (h[i]-h_cond[0])-converttemp(C,K,T[0])*(s[i]-s_cond[0])
    E_dot[i] = m_dot_air_cond*e[i]
end duplicate i
"! Exergy Destruction"
"Compressor"
E_dot_D_comp = E_dot[1]+W_comp-E_dot[2]
"Condenser"
E_dot_D_cond = E_dot[2]+E_dot[7]-(E_dot[3]+E_dot[8])
"Expansion Valve"
E_dot_D_exp = E_dot[3]-E_dot[4]
"Evaporator"
E_dot_D_evap = E_dot[4]+E_dot[5]-(E_dot[9]+E_dot[6])
"Motor"
E_dot_D_motor = W_motor-W_comp
"Total"
E_dot_D = E_dot_D_evap+E_dot_D_exp+E_dot_D_cond+E_dot_D_comp
                                              +E_dot_D_motor
"! Exergetic Efficiency of the component"
"Eta_ex_component = (E_dot_product/E_dot_fuel)"
Eta_ex_motor = W_comp/W_motor   "Motor"
Eta_ex_comp = (E_dot[2]-E_dot[1] )/W_comp   "Compressor"
Eta_ex_cond = E_dot[3]/E_dot[2]   "Condenser"
Eta_ex_exp = E_dot[4]/E_dot[3]   "Expansion Valve"
Eta_ex_evap = (E_dot[6]-E_dot[5])/(E_dot[4]-E_dot[9])   "Evaporator"
W_dot_fan_ref = 0.007 [kW]   "Reference value of power consumption of evaporator
                                                                           fan"
m_dot_air_evap_ref = 0.01766 [kg/s]   "Reference value of mass flow rate of air due
                                                             to evaporator fan"
W_dot_fan = W_dot_fan_ref*((m_dot_air_evap/m_dot_air_evap_ref)^3)
```

```
W_elec = W_motor+W_dot_fan
"! Exergy Destruction Ratios"
E_dot_fuel_system = W_elec
y_d_motor = E_dot_D_motor /E_dot_fuel_system   "Motor"
y_d_comp = E_dot_D_comp/E_dot_fuel_system   "Compressor"
y_d_cond = E_dot_D_cond/E_dot_fuel_system   "Condenser"
y_d_exp = E_dot_D_exp/E_dot_fuel_system   "Expansion Valve"
y_d_evap = E_dot_D_evap/E_dot_fuel_system   "Evaporator"
"! Exergetic Efficiency of the whole system"
E_dot_loss = E_dot[8]-E_dot[7]
Eta_ex = 1 - ((E_dot_D+E_dot_loss)/E_dot_fuel_system)
EER = Q_evap/W_elec   "Energy Efficiency Rating"
COP = Q_evap/W_comp
" !Costs"
C_electric =  0.085 [$/kW-h]
Z_o_comp = 12000 [$/ kW]   "Capital investment of compressor"
P_o_comp = 100 [kW]
P_comp = E_dot[2]-E_dot[1]   "Product of compressor"
n_comp = 0.5
m_comp = 1
J_dot = CRF/t   "Amortization factor"
Z_dot_comp = Z_o_comp*((P_comp/P_o_comp)^(m_comp))*((Eta_is/(0.9-Eta_is))
                                                    ^(n_comp ))*J_dot
Z_o_motor = 150 [$/kW]   "Capital investment of motor"
P_o_motor = 10 [kW]
P_motor = W_comp
m_motor = 0.87
Z_dot_motor = Z_o_motor*((P_motor/P_o_motor)^(m_motor))*(Eta_motor/(1-
                                                  Eta_motor))*J_dot
Z_o_cond= 450 [$/kW]   "Capital investment of condenser"
UA_o_cond = 0.0150 [kW/k]
T_cond = ((h[2]-h[3])/(s[2]-s[3]))-273.15
P_cond = E_dot[3]
Epsilon_cond = (T[8]-T[7])/(T_cond-T[7])
Z_dot_cond = Z_o_cond*( (Q_dot_cond/(UA_o_cond))*(-ln(1-Epsilon_cond)))
```

```
                                                   *(P_cond/(T[0]+273.15))*J_dot
Z_o_evap = 1140 [$/kW]   "Capital investment of evaporator"
UA_o_evap = 0.0200 [kW/k]
P_evap = E_dot[6]-E_dot[5]
Z_dot_evap = Z_o_evap*( (Q_evap/(UA_o_evap))*(-ln(1-Epsilon_evap)))*
                                                 (P_evap/(T[0]+273.15))*J_dot
Z_o_exp = 37 [$/kW]   "Capital investment of expansion device"
P_exp = E_dot[4]
Z_dot_exp = Z_o_exp*P_exp * J_dot
cycle_duty = t/3600 [h/yr]
inflation = 0.04   "Average nominal escalation rate of fuel"
" ! Time Value of Money Cost to Operate"
duplicate i = 1, n
    C_electricity[i] = C_electric*(1+inflation*(i-1))
    Cost_annual[i] = C_electricity[i]*W_elec*cycle_duty
    Cost_annual1[i] = C_electricity[i]*W_motor*cycle_duty
end duplicate i
Cost_operating = sum(Cost_annual[i]*1[yr],i = 1, n)
"! Ratios"
FP = FPRatio(i, m, n)
AF = AFRatio(i, m, n)
AP = APRatio(i, m, n)
AP[0] = AP0Ratio(n, k, CRF)
C_electric_L = ((Cost_annual[1]*1[yr])*AP[0])/(1+inflation)
C_electric_L_motor = ((Cost_annual1[1]*1[yr]))*AP[0]/(1 + inflation)
f_opendoor = 0.1
"! Cost Balances"
duplicate i = 1, 9
      C_dot[i] = c[i]*E_dot[i]
end duplicate i
c[4] = c[9]   "any 'unused' refrgierant leaving has same ability to do work as entering
                                                                      refrigerant"
c[1] = c[9]   "unused principle"
c[2] = c[3]   "unused principle"
c[7] = 0 [$/kJ]   "the air coming in is free b/c no work has been done to it"
```

```
c[5] = ((2.3e-5 [$/kJ]) / EER)*f_opendoor
C_dot_W1= convert($/h, $/s)*(C_electric_L_motor/(cycle_duty*1[yr]))
C_dot[4]+C_dot[5]+Z_dot_evap = C_dot[9]+C_dot[6]   "Evaporator"
C_dot[2]+C_dot[7]+Z_dot_cond = C_dot[3]+C_dot[8]   "Condenser"
C_dot[3]+Z_dot_exp = C_dot[4]   "Expansion Valve"
C_dot_W2 = C_dot_W1+Z_dot_motor   "Motor"
C_dot[1]+C_dot_W2+Z_dot_comp = C_dot[2]   "Compressorr"
"! Fuel Costs"
c_fuel_motor = C_dot_W1/W_motor   "Motor"
c_fuel_comp = C_dot_W2/W_comp   "Compressor"
c_fuel_cond = C_dot[2]/E_dot[2]   "Condenser"
c_fuel_exp = C_dot[3]/E_dot[3]   "Expansion valve"
c_fuel_evap = (C_dot[4]-C_dot[9])/(E_dot[4]-E_dot[9])   "Evaporator"
"! Product Costs"
c_prod_motor =  C_dot_W2/W_comp   "Motor"
c_prod_comp = (C_dot[2]-C_dot[1])/(E_dot[2]-E_dot[1])   "Compressor"
c_prod_cond = C_dot[3]/E_dot[3]   "Condenser"
c_prod_exp = C_dot[4]/E_dot[4]   "Expansion valve"
c_prod_evap = (C_dot[6]-C_dot[5])/(E_dot[6]-E_dot[5])   "Evaporator"
"! Cost of Exergy Destruction in the component"
C_dot_D_motor = c_fuel_motor*E_dot_D_motor   "Motor"
C_dot_D_comp = c_fuel_comp*E_dot_D_comp   "Compressor"
C_dot_D_cond = c_fuel_cond*E_dot_D_cond   "Condenser"
C_dot_D_exp = c_fuel_exp*E_dot_D_exp   "Expansion valve"
C_dot_D_evap = c_fuel_evap*E_dot_D_evap   "Evaporator"
C_dot_D_tot = C_dot_D_comp+C_dot_D_cond+C_dot_D_exp+C_dot_D_evap+
                                                        C_dot_D_motor
"! Exergoeconomic Factor"
f_motor = Z_dot_motor/(Z_dot_motor + C_dot_D_motor)   "Motor"
f_comp = Z_dot_comp/(Z_dot_comp+ C_dot_D_comp)   "Compressor"
f_cond = Z_dot_cond/(Z_dot_cond + (c_fuel_cond*(E_dot_D_cond+E_dot_loss)))
                                                              "Condenser"
f_exp = Z_dot_exp/( Z_dot_exp+C_dot_D_exp)   "Expansion valve"
f_evap = Z_dot_evap/(Z_dot_evap+C_dot_D_evap)   "Evaporator"
"! Cost Importance"
```

```
ZC_motor = Z_dot_motor+C_dot_D_motor   "Motor"
ZC_comp = Z_dot_comp+C_dot_D_comp   "Compressor"
ZC_cond = Z_dot_cond+C_dot_D_cond   "Condenser"
ZC_exp = Z_dot_exp+C_dot_D_exp   "Expansion valve"
ZC_evap = Z_dot_evap+C_dot_D_evap   "Evaporator"
```

# BRIEF PROFILE OF THE RESEARCH SCHOLAR

Mr. Gaurav is a research scholar in the Mechanical Engineering Department of J. C. Bose University of Science & Technology, YMCA, Faridabad, Haryana, India. He has completed his M.Tech. in Thermal Engineering from NIT Kurukshetra, Haryana, India and B.Tech. from CITM, Faridabad, Haryana, India. He has teaching experience of more than 8 Years. Presently, he is working as an Assistant Professor in Mewat Engineering College established by Haryana Waqf Board. His areas of interest include Heat Exchangers and Refrigeration & Air Conditioning. He has published 14 research papers in various National & International Journals and Conferences. He has also organized workshops in various fields like Solar and Robotics.

# LIST OF PUBLICATIONS OUT OF THESIS

**List of Published Papers**

| **Sr. No.** | **Title of Paper** | **Name of Journal** | **No.** | **Volume & Issue** | **Year** | **Pages** |
|---|---|---|---|---|---|---|
| 1 | Computational Energy and Exergy Analysis of R134a, R1234yf, R1234ze and their Mixtures in Vapour Compression System | Ain Shams Engineering Journal, Elsevier Publication, Indexed: ESCI, Scopus, UGC Approved | ISSN 2090-4479 | Vol. 9, Issue 4 | 2018 | 3229-3237 |
| 2 | Thermo Economic Analysis of Environmental Friendly Refrigerant Mixtures for Replacement of R134a | ARPN Journal of Engineering and Applied Sciences, Indexed: Scopus, UGC Approved | ISSN 1819-6608 | Vol. 13, Issue 4 | 2018 | 1357 – 1363 |
| 3 | Sustainability of Alternative Material of R-134a in Mobile Air-Conditioning System: A Review | Journal: Materials Today Proceedings, Elsevier Publication, Indexed: Scopus, UGC Approved | ISSN 2214-7853 | Vol. 4, Issue 2 | 2017 | 112-118 |
| 4 | Optimization of Capillary Tube Parameters in Vapour Compression System using Environmentally Friendly Refrigerant R1234yf | International Journal of Engineering and Technology, Engg Journals Publications, Indexed: Google Scholar | ISSN 0975-4024 | Vol. 9, Issue 1 | 2017 | 243-248 |

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| 5 | Environmental Sustainability of Automobile Air-Conditioning System with Refrigerant R1234yf | International Journal of Advance Research and Innovation, Indexed: Google Scholar | ISSN 2347-3258 | Vol. 3, Issue 4 | 2015 | 653-656 |
| 6 | Performance Analysis of Household Refrigerator with Alternate Refrigerants | International Journal of Innovative Research in Science, Engineering and Technology, Indexed: Google Scholar | ISSN 2319-8753 | Vol. 3, Issue 4 | 2014 | 11397-11405 |
| 7 | Analysis of Solar Ejector –Jet Refrigeration System Using Eco-Friendly Material R1234yf | International Conference on CAD/CAM, Robotics and Factories of the Future, Springer Publication, Indexed: Scopus, UGC Approved | ISSN 2195-4364 | | 2016 | 831-838 |
| 8 | Energy and Exergy Analysis of Hydrocarbon mixtures in Domestic Refrigerators | National Conference on Roles of Science and Technology towards ‘Make in India’, YMCA University of Science & Technology, Sponsored by DST Haryana and DRDO, New Delhi | | Paper ID ME03 | 2016 | |

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|---|---|---|---|---|---|---|
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