CHAPTER 1

INTRODUCTION

There are two categories of thermal power plants. These are:

i. Work producing plants i.e. thermal power plants.
ii. Work consuming plants i.e. refrigeration or heat pump plants.

The work producing plants (heat engines) convert heat into work. The work consuming plants lead to the flow of heat from a low temperature body to a high temperature body. Common examples of work consuming plants are: Cold storage, food freezing, central air conditioning, room air conditioners, ice plants, freeze-drying plants, domestic refrigerator etc.

A refrigerating machine or work consuming device is a device which either cool or maintain a temperature of the body below that of the surrounding. Heat must be flow from low temperature of the body to the surrounding which is at higher temperature.

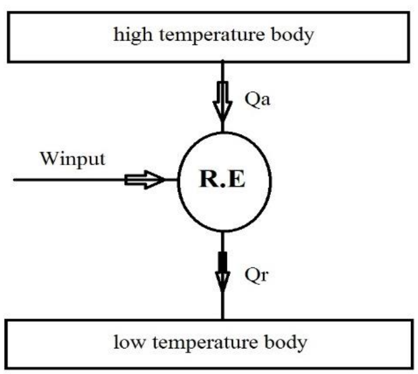

Fig. 1: A thermodynamic cycle involving heat transfer

According to Clausius statement of second law of thermodynamics: it is impossible for a device to work in a cycle and transfer heat from a low temperature to a high temperature without any external work input. Hence the refrigeration system requires work input W to run the system.

The basic refrigeration system works on the Reversed Carnot Cycle which is also known as Vapor Compression cycle. There are four process in this cycle:

1. Compression
2. Condensation
3. Expansion
4. Evaporation

In a vapor compression refrigeration system, a working fluid (Refrigerant) evaporates at low temperature and pressure. The temperature remains constant while the enthalpy of evaporation or latent heat is absorbed from the space or product to be cooled. The vapor is compressed to a high pressure so that heat can be rejected to the surrounding and the vapor condensed and cycle back to absorb heat.

For food freezing industries and milk storage plants, Refrigeration is essential to freeze the food, making of ice and store milk at a lower temperature. The basic purpose of using refrigeration in our life is such as:

1. For the food products, fruits, vegetables and milk, refrigeration is very important. It produces lower temperature than the surrounding temperature. Storing of fruits requires lower temperature at about -6^{o}C to 10^{o}C and fishes are freeze at about -15^{o}C to -30^{o}C.
2. Refrigeration is also required for making of many products like fruit cream, butter, cheese etc.
3. Refrigeration is also essential for making of ice (at about -25^{o}C) and storage of food products at low temperature ie. Fishes, chicken, vegetables, fruits etc.
4. Refrigeration is also required in the refrigerated vehicles to transport the food products, so that the quality of the food products should not be degrade.

1.1 ENVIRONMENTAL IMPACTS

Green House gas (GHG) emissions from fossil fuel combustion for power generation and emission of halogenated refrigerants from vapor compression based refrigeration , air conditioning and heat pump systems contribute significantly to global warming. A reduction in GHG emissions can only be achieved by using environment friendly and energy efficient refrigerants. The high environmental impacts due to halogenated refrigerant emissions lead to identifying a long-term alternative to meet all the system requirements including system performance, refrigerant lubricant interaction, energy efficiency, safety and service. Halogenated refrigerants are extensively used in the refrigeration and air conditioning industries over many decades due to their excellent thermodynamic and thermo-physical properties. As per the Montreal Protocol 1987, developing countries like India, with a per capita consumption less than 0.3 kg of ozone depletion substance have been categorized as Article-5 countries. These countries are required to phase out all Chloro -Fluro carbons (CFCs) by 2010 and all Hydro-Chloro-Fluro carbons (HCFCs) by 2040. HFC refrigerants are considered as one among the six targeted green-house gas under Kyoto protocol of United Nations Framework Convention on Climate Change (UNFCCC) in 1997.

1.2 OZONE LAYER DEPLETION

The first major environmental impact that struck the refrigeration based industries is Ozone Depletion Potential (ODP) due to manmade chemicals into the atmosphere. About 90% of the ozone exists in the stratosphere between 10 and 50 km above the earth surface. Molena and Rowland (1974) give in detail that chlorine based refrigerants are stable enough to reach the stratosphere, where the chlorine atoms act as catalyst to destroy the stratospheric ozone layer which protects the earth surface from direct ultra violet rays.

1.3 GLOBAL WARMING POTENTIAL (GWP)

The second major environmental impact is GWP, which is due to the absorption of infrared emissions from the earth, causing an increase in global earth surface temperature. While solar radiation at 5800 K and 1360 W/m^2 arrives the earth , more than 30% is reflected back into space and most of the remaining radiation passes through the atmosphere and reaches the ground. This solar radiation heats up the earth, which is approximately as a black body radiating energy with a spectral peak in the infrared wavelength range. This infrared radiation cannot pass through the atmosphere because of absorption by GHG including the halogenated refrigerants.

1.4 BLENDING

Blends are made up of two or more single component refrigerants. One of two situations will occur, depending on how strong the different molecules are attracted to each other:

a) Azeotrope: a blend that behaves like a single component refrigerant. When a blend forms an azeotrope it displays unique and unexpected properties.
b) Zeotrope: a blend that behaves like a mixture of the individual components. Zeotrope have predictable properties based on combinations of the pure components' properties.

In addition, it is sometimes helpful to classify some blends as Near-Azeotropes. These blends have predictable blend properties, however the difference between these properties and what is observed for single component refrigerants is not that significant. Many of the problems associated with system operation using a blend will not be noticeable with near-azeotropes.

Two new properties (to be explained soon) are Fractionation and Temperature Glide. We can split the zeotropic blends into Low Fractionation Potential, which also show Low Temperature Glide, and High Fractionation Potential, which also show High Temperature Glide. (Generally speaking blends with less than 5ºF glide are considered near-azeotropes).

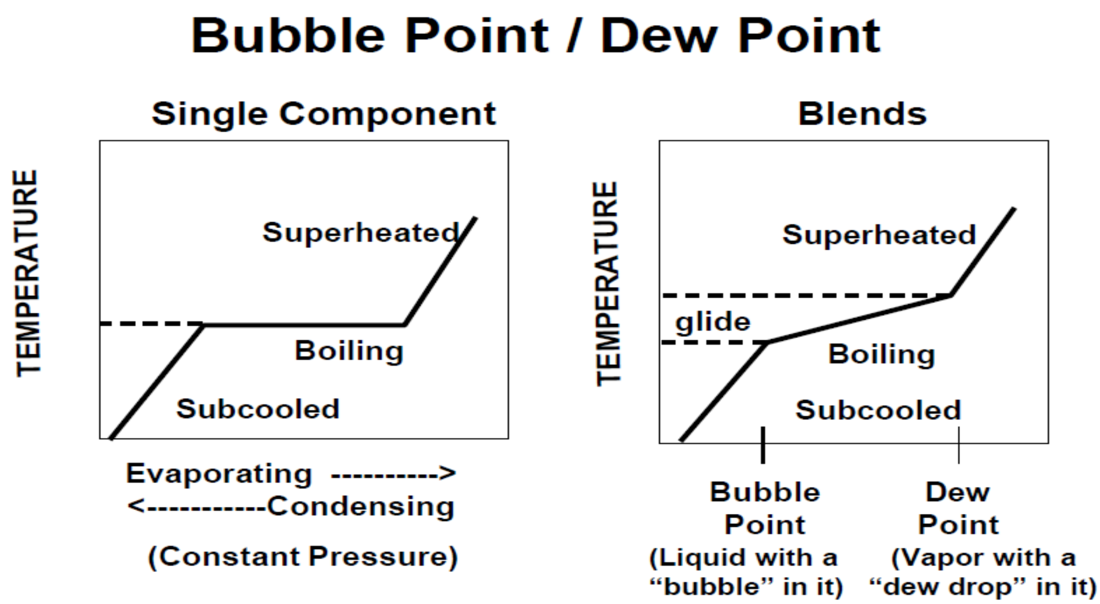

Fig. 2: Comparison of evaporation and condensation of pure refrigerant (single component) and mixed refrigerant (blends)

The process of phase change (boiling or condensing) is the same for blends as it is for pure refrigerants:

i. Boiling: liquid reaches a temperature where bubbles form, then the liquid boils to vapor. When the last drops of liquid disappear, any additional heat input causes the vapor to superheat.
ii. Condensing: vapor cools to a temperature where liquid drops start to form, then the vapor condenses to liquid. When the last of the vapor disappears, any additional removal of heat causes the liquid to sub-cooled.

When these phase changes occur to a pure refrigerant, at constant pressure, the temperature stays constant at what we normally call the "boiling point."

For blends the process is the same, but the shift in composition during phase change causes the temperature glide to occur. The vapor will still superheat, and the liquid will still sub-cooled, however the Saturated Vapor temperature and the Saturated Liquid temperature are not the same like they were for pure refrigerants. We now must know the particular saturated temperatures at the ends of the temperature glide for a given pressure.

Saturated Liquid = Bubble Point (Liquid with bubbles starting to form)

Saturated Vapor = Dew Point (Vapor with dew drops starting to form)

1.5 SCOPE FOR PRESESNT THESIS

In this present work, Multi evaporator refrigeration system is considered in which there are multiple evaporator having individual compressor and expansion devices, also there is flash chamber, in this system. Earlier R12 was used as a working fluid in the refrigerator but now R134a is used. In this present thesis, an exergy analysis is done using a mixture of R290 and R600a with different proportion for food freezing plant or household refrigerator applications. The multi-evaporator system with evaporator-1 temperature is -30^{o}C with capacity of 2 TR and evaporator-2 temperature is 2^{o}C with the capacity of 10 TR is assumed.

CHAPTER 2

LITERATURE SURVEY

A brief literature survey of work which is done by using alternative refrigerants in simple vapor compression system or in multi evaporator vapor compression refrigeration system during last few years.

2.1 LITERATURE

Alptug Yataganbaba [1] carried out the exergy analysis on a two evaporator vapor compression refrigeration system using R1234yf, R1234ze and R134a as refrigerants. The effects of the evaporator and condenser temperatures on the exergy destruction and exergy efficiency of the system were investigated. R1234yf and R1234ze, which are good alternatives to R134a concerning

environmentally friendly properties and this is the most significant finding emerging from this study.

Arora et al. [2] performed the exergy analysis of actual vapor compression system using the different refrigerants (R22, R410a and R407c) and calculated the performance parameters ie COP, EDR and exergetic efficiency. Results show that exergetic efficiency of R22 are higher than R410A and R407C.

Nikolaidis and Probert [3] conducted exergy analysis of two stage vapor compression system with a flash inter cooing using R22. The effects of evaporator's temperature which varied from -35^{o}C to -45^{o}C and the effects of condenser's temperature which varied from 25^{o}C to 35^{o}C, on the performance parameters.

Halimic et al. [4] carried out theoretical and graphical analysis to find out the best alternative refrigerant among R290, R134a and R134a for the replacement of R12 in simple vapor compression system. R134a can be used in the system due to its similar properties and performance compare to R12. R290 has the zero GWP (global warming potential) should be considered as alternative refrigerant for the replacement of R12.

Dalkilic and Wongwises [5] did a theoretical performance study on a traditional vapor-compression refrigeration system with refrigerant mixtures based on HFC134a, HFC152a, HFC32, HC290, HC1270, HC600, and HC600a was done for various ratios and their results are compared with CFC12, CFC22, and HFC134a as possible alternative replacements.

Bolaji et al. [6] analyzed the performances of the three ozone Hydro fluorocarbon (HFC) refrigerants (R32, R134a and R512a) in a vapor compression refrigeration system were investigated experimentally and compared. The results obtained showed that R32 yielded undesirable characteristics, such as high pressure and low co-efficient of performance (COP). Comparison among the investigated refrigerants confirmed that R152a and R134a have approximately the same performance, but the best performance was obtained from the used of R152a in the system. As a result, R152a could be used as a drop-in replacement for R134a in vapor compression refrigerant system. The COP of R152a obtained was higher than those of R134a and R32 by 2.5% and 14.7% respectively. Also, R152a offers the best desirable environmental requirements; zero ozone potential and very low global warming potential.

Arora and Kaushik [7] performed exergy analysis and computational modeling on actual vapor compression refrigeration (VCR) system. A computational modeling has been done for computing COP, exergy destruction, exergetic efficiency and efficiency defects for R502, R404A and R507A. The present investigation has been done for evaporator and condenser temperatures in the range of -50 ^{o}C to 0 ^{o}C and 40 ^{o}C to 55 ^{o}C, respectively. The results indicate that R507A is a better substitute to R502 than R404A. The efficiency defect in condenser is highest, and lowest in liquid vapor heat exchanger for the refrigerants considered.

Mishra [8] has done thermodynamic analysis of multi-evaporators system at different temperatures with single compressor and single expansion valve using liquid vapor heat exchanger. This analysis for vapor compression refrigeration systems has been done in terms of performance parameter for R507a, R125, R134a, R290, R600, R600a, R1234ze, R1234yf, R410a, R407c, R707, R404a and R152a refrigerants. The theoretical analysis has been done for both the systems. It was also observed that performance of both systems using R717 is higher but R600 and R152a nearly matching same values under the accuracy of 5% can be used in the above system .But difficulties using R152a, R600, R290 and R600a have flammable problems therefore safety measures are required using these refrigerants. Therefore R134a refrigerant is recommended for commercial applications although it has slightly less thermal performance than R152a which is not widely used refrigerant for domestic and industrial applications

Kumar et al. [9] performed the exergy analysis in terms of COP, exergy destruction and exergetic efficiency to find out the alternative refrigerant for R134a because R134a has the more GWP so it is not eco-friendly refrigerant. Results shows that the best refrigerant which is used as alternative refrigerant of R134a is blend of R290 and R600a, also results show that this refrigerant has higher COP and exergetic efficiency than that of refrigerant 134a.

Padilla et al. [10] studied the energy analysis of the impact of direct replacement of R12 with the zeotropic mixture R413a on the performance of a domestic vapor compression refrigeration system. They found that the evaporator outlet temperature ranged from 15^{0}C to -10^{0} C. the overall energy performance of the system working with R413a is consistently better than that of R12.

Kumar et al. [11] has worked on the simple vapor compression (VCR) cycle to performed the thermodynamic analysis using refrigerant 12 and refrigerant 11 and compare both of the refrigerants in terms of coefficient of performance, exergetic efficiency and exergy destruction.

Cabello et al. [12] conducted the theoretical analysis of simple vapor compression (VCR) system and developed a parametric table which shows the variation of the input parameters with the performance parameter using R407c, R22 and R134a. Results show that R407c has the highest COP among these refrigerants for the higher compression ratio.

Han et al. [13] performed an experimental investigation in a simple vapor compression (VCR) system using R407c and blend of R32/R125/R161 at different conditions. After the evaluation of the performance parameters it can be concluded that this blend has the higher COP at low pressure ratio and it can be used as a drop in refrigerant for R407c.

Mohan Raj et al. [14] conducted an experimental investigation with hydrocarbon refrigerant mixture (blend of R290 and R600a in the ratio of 45.2:54.8 by weight) as an alternative to R134a in a 200*l* single evaporator domestic refrigerator. The results showed that the hydrocarbon mixture has lower values of energy consumption with 3.25-3.6% higher coefficient of performance (COP). The above hydrocarbons refrigerant mixture could be the best long term alternative to phase out R134a.

Getu and Bansal [15] has worked on the cascade refrigeration system to optimize the design of the system and operating parameter using R744-R717. Optimization is done for the temperature difference in heat exchanger, evaporator temperature, condensing temperature and superheating and sub cooling temperature. This optimization is done by the regression analysis.

Spartz and Motta [16] conducted an experimental analysis for the replacement of refrigerant 12 which is used in the household refrigerator. A thermodynamic analysis has been done using R404a, R290 and R12. By the experiment, it is found that R290 give the best result than the others, so this refrigerant can be considered as a drop in refrigerant for R12.

Ahamed et al. [17] performed the energy and exergy analysis of vapor compression refrigeration system using hydro carbons (HC) refrigerants. In this research Nano fluid is also used with the result of exergy destruction of the system should be minimized.

Xuan and Chen [18] did an experimental analysis of simple vapor compression (VCR) system using two refrigerants ie. R502 and HFC-161. Results show that at a higher evaporating temperature HFC-161 is better than R404a. It can be concluded that HFC-161 is an alternative refrigerant which is used as a drop in refrigerant of R502.

2.2 CONCLUSION AND GAPS

Now a days, in the household refrigerator and food industries refrigeration system works on the Vapor compression refrigeration cycle which is also known as Reversed Carnot Cycle, is used to achieve and maintain required low temperature. In this refrigeration system, R12 and R134a are used which are CFCs and HFCs refrigerants respectively. CFCs and HFCs are not eco-friendly. Chlorine containing CFCs cause Ozone depletion and all of them cause global warming by greenhouse effect. Eventually, these will also have to be replaced with alternative refrigerants with low ODP and GWP.

Based on literature survey, it can be said that researchers did experimental and theoretical analysis to increase COP by using alternative refrigerants and to increase exergetic efficiency for the simple vapor compression refrigeration system in which there is only one evaporator. Exergy analysis or thermodynamic analysis is not done by researchers in the following systems:

- Multi evaporator vapor compression refrigeration system using individual compressor and multiple expansion device in that system.
- Use of blend refrigerants or mixed refrigerants in multi evaporator refrigeration system.
- Use of mixed refrigerant in multi evaporator refrigeration system with a flash tank and compare with pure refrigerants.
- To find out the alternative refrigerant which give the best result for COP and exergetic efficiency and which is also eco friendly.

Refrigerants	R12	R134a	R290	R600a	R436a
Name	Dichlorodifluoromethane	Tetrafluoroethane	Propane	Iso-Butane	R290/R600a (.056/.044)
Molecular Mass (g/mol)	120.91	102.03	44.1	58.12	54.56
Boiling Point (^oC)	-29.8	-26.3	-42.16	-11.7	-34.3
Freezing Point (^oC)	-157.7	-103.3	-188	-159.6	-
Critical Temperature (^oC)	101.06	112	96.7	134.71	130.1
Critical Pressure(bar)	40.59	41.36	42.49	36.5	33.9
Density (kg/m^3)	4.25	1.49	2.01	2.4787	1.2
GWP	10900	1430	3	3	< 3

Table 1: Properties of Refrigerants

2.3 PROBLEM FORMULATION

In the present study, thermodynamic analysis of multi evaporator vapor compression refrigeration system using individual compressor and multiple expansion device with a flash chamber is carried out using pure refrigerants and mixed refrigerants.

It is proposed to inspect the variation of following parameters with the performance perameter of multi evaporator refrigeration systems and find out better alternative refrigerant which is eco-friendly and give the best result, in this present work for the application of food freezing industries (fishes, fruits and vegetables).

- Effect of varying first evaporator's temperature on Work Input, COP, Exergy Destruction, Exergy Destruction Ratio and Exergetic Efficiency.
- Effect of varying second evaporator's temperature on Work Input, COP, Exergy Destruction, Exergy Destruction Ratio and Exergetic Efficiency.
- Effect of varying condenser temperature on Work Input, COP, Exergy Destruction, Exergy Destruction Ratio and Exergetic Efficiency.

CHAPTER-3

THERMODYNAMIC ANALYSIS

In this chapter, the thermodynamic analysis has been done on multi evaporator vapor compression refrigeration system with individual compressor and multiple expansion devices and also using flash tank.

3.1 SYSTEM DESCRIPTION

Mostly the refrigeration system which are used in the refrigerators, food freezing plant, cold storages and milk chilling plants, works on the simple vapor compression refrigeration (VCR) cycle which is also known as Reversed Carnot Cycle. This system can be a single stage compression or multi stage compression. A single stage vapor compression system works only with low temperature lift (the difference between condenser temperature and evaporator temperature). However, there are many applications where the temperature lift can be quite high. The temperature lift can become large either due to the requirement of very low evaporator temperatures and/or due to the requirement of very high condensing temperatures. For example, in frozen food industries the required evaporator can be as low as -40^{O}C, while in chemical industries temperatures as low as -150^{O}C may be required for liquefaction of gases. When the evaporating temperature decreases, the effects are

i. expansion losses increase
ii. refrigeration effect decreases
iii. Superheat losses increase
iv. Discharge temperature of compressor increases
v. Quality of the vapor increases at the inlet to the evaporator

As a result of this, the refrigeration capacity decreases and work of compression increases, so it is recommended to work with a high temperature lift in a single stage vapor compression refrigeration (VCR) system. For the solution of this problem multi stage refrigeration system are used. For example Refrigerant 22 is used in a single stage system up to -30^{o}C, in a two stage system up to -50^{o}C and in a three stage system up to -65^{o}C.

Apart from high temperature lift applications, multi-stage systems are also used in applications requiring refrigeration at different temperatures. For example, in a food freezing plant refrigeration may be required at -30^{O} C for storing of fishes and at 5^{O}C for storing fruits. In such cases it may be advantageous to use a multi-evaporator refrigeration system with first evaporator operating at a low temperature of -30^{O}C and the second evaporator operating at a high temperature of 2^{O}C.

A multi-stage system is a refrigeration system with two or more low-side pressures. Multi-stage systems can be classified into:

a. Multi-compression systems
b. Multi-evaporator systems
c. Cascade systems, etc.

Fig.3 represents a Multi-evaporator refrigeration system with individual compressors using a flash tank and multiple expansion devices. First evaporator (evaporator-1) is at a temperatures of T_{e1} and second evaporator (evaporator-2) is at a temperature of T_{e2}, and the refrigerating capacity for evaporator-1 is Q_{e1} (2TR) and for evaporator-2 is Q_{e2} (10TR). W_{c1} is the work input for the compressor-1 and W_{c2} is the work input for the compressor-2.

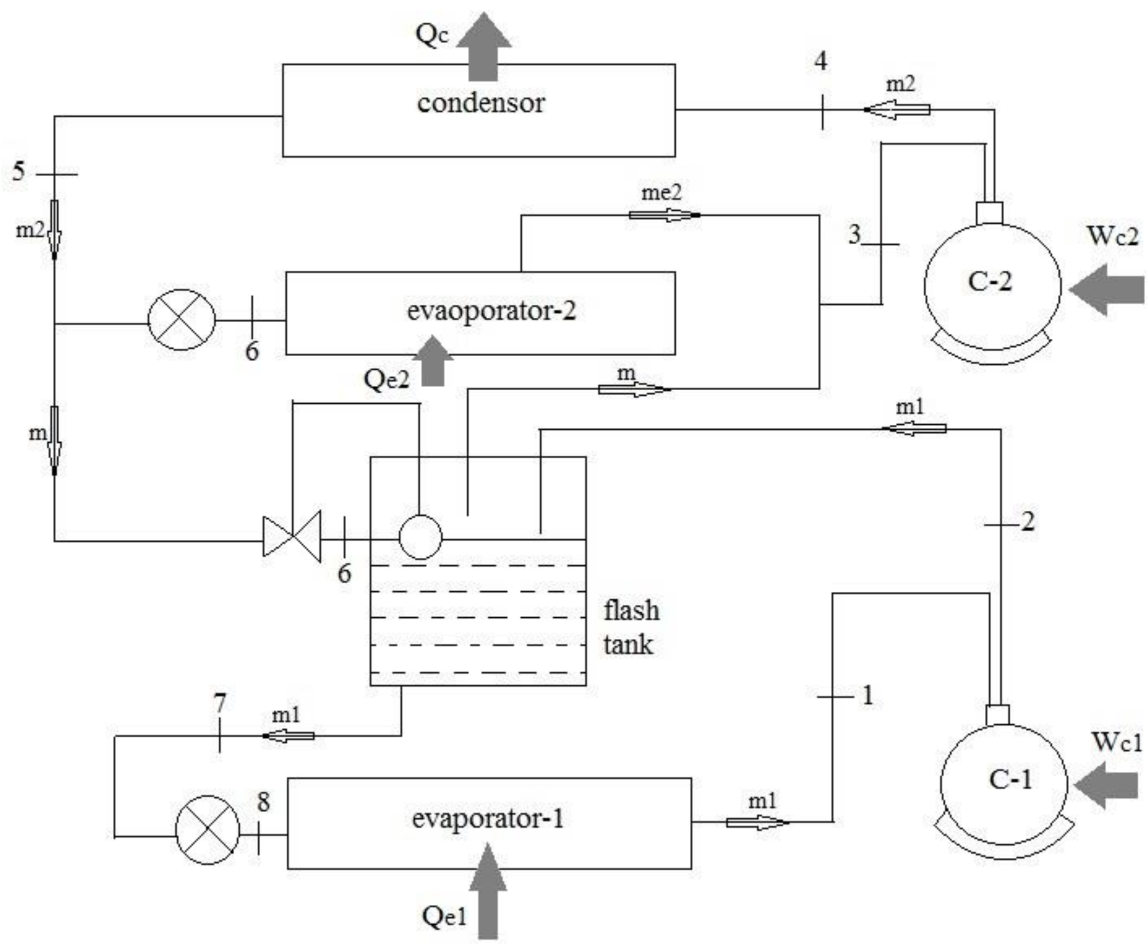

Fig. 3: System Diagram of Multi-evaporator refrigeration system using individual compressor with a flash tank

In this system, the vaporized refrigerant is compressed by the compressor at point 4 to the condenser temperature. This superheated refrigerant is cooled in the condenser, turning into liquid under high pressure at point 5. Then this refrigerant flows from the condenser to the evaporator-2 through expansion valve at point 6.

Also, after passing the flash chamber at point 7, the liquid refrigerant flows through the expansion valve at point 8 to the evaporator-1. Where, it takes heat from the system which we want to cool or freeze. Then it flows from evaporator-1 to compressor-1 and it compress the refrigerant to point 2 and comes into the flash tank. After the flash tank at point 3, the refrigerant flows into the compressor-1. The function of the flash tank is to separate the vapor and liquid from the refrigerant.

3.2 ENERGY ANALYSIS

In this present work, thermodynamic analysis is done to determine the performance parameters with the variation of evaporating temperature and condensing temperature of multi evaporator system.

The following assumptions are to be taken for this analysis:

1. All process in the system are steady state process and all components are taken as to be a steady flow.
2. All the pressure loss either in condenser or in evaporator and in suction and discharge line, are assumed to be negligible.
3. All expansion devices (throttling) are isenthalpic.
4. There is no superheating of the vapor in the evaporator.
5. There is no sub cooling of the liquid in the condenser.
6. Effects of changes in the frictional energy, kinetic energy and potential energy are assumed to be negligible.

Based on following assumptions, the basic thermodynamic relations which are used for the thermodynamic analysis of the system are as follow:

Mass Balance for all components

$$\sum_{in} \mathrm{m} = \sum_{out} \mathrm{m}$$

Energy Balance for all components

$$\mathrm{Q} - \mathrm{W} + \sum_{in} m\, h - \sum_{out} m\, h = 0$$

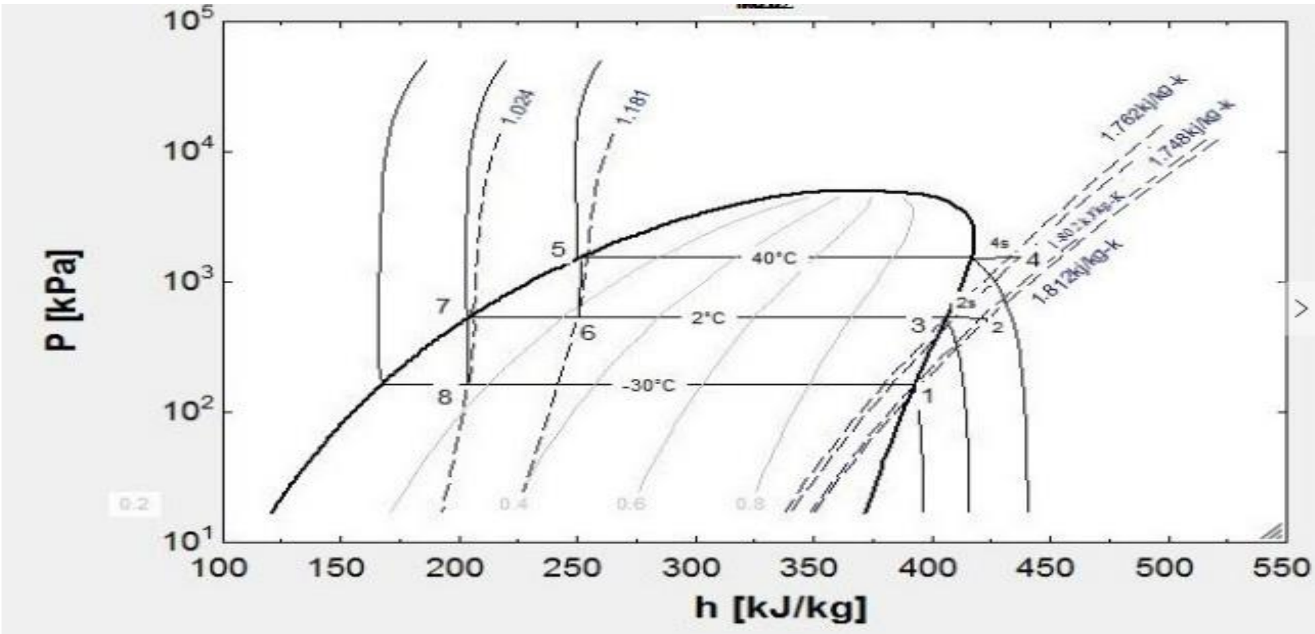

Fig. 4: p-h diagram of the multi evaporator refrigeration system for R12

Energy analysis for each component of multi evaporator system is as follows:

Evaporator: Evaporator is the component of a refrigeration system in which heat is removed from air, water or any other body required to be cooled by the eveporating refrigerant.

Refrigerating capacity of evaporator-1 is given as:

$Q_{e1} = m_1 (h_1 - h_4)$

Refrigerating capacity of evaporator-2 is given as:

$Q_{e2} = m_{e2} (h_3 - h_6)$

Compressor: Compressor is used to compress the working fluid (refrigerant) from a low temperature to a high temperature which requires work according to Clausius statement of second low of thermodynamic.

Isentropic (ideal) work for the compressor-1 as:

$W_1 = m_1 (h_{2s} - h_1)$

Isentropic efficiency of the compressor-1 as:

$\eta_1 = (h_{2s} - h_1) / (h_2 - h_1)$

Actual compressor-1 work is specified by

$W_1 = m_1 (h_2 – h_1)$

Isentropic (ideal) work for the compressor-2 as:

$W_2 = m_2 (h_{4s} – h_3)$

Isentropic efficiency of the compressor-2 as:

$\eta_2 = (h_{4s} – h_3) / (h_4 - h_3)$

Actual compressor-2 work is specified by

$W_2 = m_2 (h_4 – h_3)$

Condenser: Condenser is device which is used to condense vapor into liquid state, by cooling it. The rejected heat as:

$Q_C = m_2 (h_4 - h_5)$

Expansion Devices: An expansion device in a refrigeration system normally serves two purposes. One is the thermodynamic function of the expanding the liquid refrigerant from condenser pressure to evaporator pressure. The other is the control function which may involve the supply of the liquid to the evaporator at the rate at which it is evaporated.

In expansion device-1:

$h_5 = h_6$

In expansion device-2:

$h_7 = h_8$

Flash Tank: A system with a flash tank, eliminates the undesirable throttling of the vapor generated at the intermediate pressure. The energy balance for the flash tank is such as:

$m\, h_6 + m_1\, h_2 = m\, h_3 + m_1\, h_7$

Coefficient of Performance: The performance of a heat engine is describe by its thermal efficiency. The performance of a refrigerating machine or a heat pump is expressed by the ratio of useful heat (refrigerating effect) to work (compressor work),

COP = Refrigerant Effect / Total Work Input

3.3 EXERGY ANALYSIS

Exergy Destruction in Evaporator

In Evaporator-I:

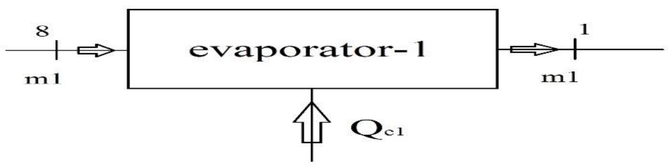

Fig. 5: System diagram of evaporator-1

$ED_{e1} = m_1 (h_8 - T_o s_8) + Q_{e1} (1 - T_o/T_{r1}) - m_1 (h_1 - T_o s_1)$

In Evaporator-II:

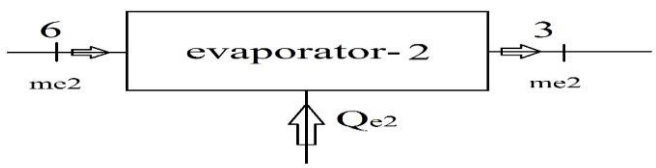

Fig. 6: System diagram of evaporator-2

$ED_{e2} = m_{e2} (h_6 - T_o s_6) + Q_{e2} (1 - T_o/T_{r2}) - m_{e2} (h_3 - T_o s_3)$

Exergy Destruction in Compressor

In compressor-1:

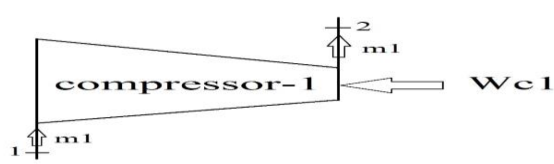

Fig. 7: System diagram of compressor-1

$ED_{c1} = m_1 (T_o (s_2 - s_1))$

In compressor-2:

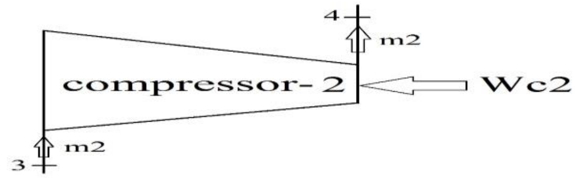

Fig. 8: System diagram of compressor-2

$ED_{c2} = m_2 (T_o (s_4 - s_3))$

Exergy Destruction in Condenser

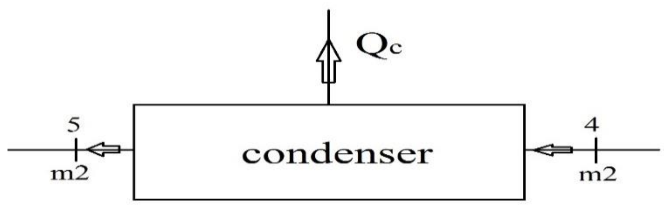

Fig. 9: System diagram of condenser-1

$ED_{cond} = m_2 (h_4 - T_o s_4) – m_2 (h_5 - T_o s_5)$

Exergy Destruction in Expansion device

In expansion device-1:

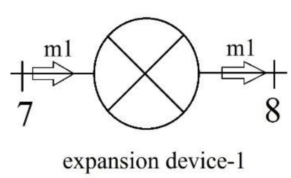

Fig. 10: System diagram of expansion valve-1

$ED_{t1} = m_1 (T_o (s_8 - s_7))$

In expansion device-2:

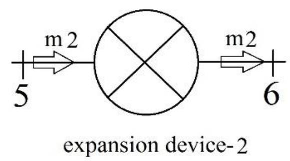

Fig. 11: System diagram of expansion valve-2

$ED_{t2} = m_2 (T_o (s_6 - s_5))$

Exergy Destruction in Flash Tank

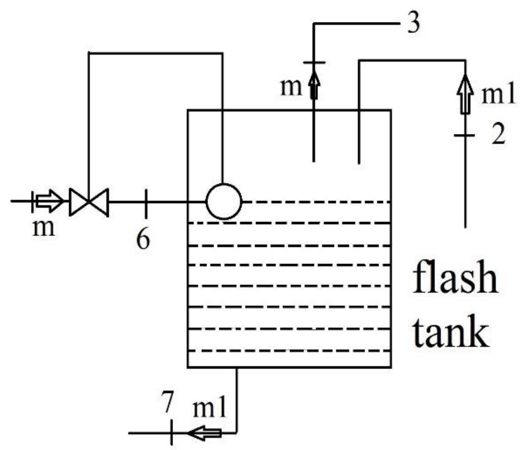

Fig. 12: System diagram of flash tank

$ED_{flash} = m (h_6 - T_o s_6) + m_1 (h_2 - T_o s_2) - m_1 (h_7 - T_o s_7) – m (h_3 - T_o s_3)$

Total Exergy Destruction

$ED = ED_{e1} + ED_{e2} + ED_{c1} + ED_{c2} + ED_{cond} + ED_{t1} + ED_{t2} + ED_{flash}$

Exergy of Product

$EP = Q_{e1} (1 - T_o/T_{r1}) + Q_{e2} (1 - T_o/T_{r2})$

Exergetic Efficiency

η_{ex}= EP / W_{total}

Exergy Destruction Ratio

EDR= ED / EP

3.4 Input Parameters

This present work is for the application of the food freezing plant (fruits, vegetables, fishes etc) of small capacity. Frozen fish products should be stored at different temperature according to their species. Some products are stored at higher than the -30^{o}C which are for a short period. Temperature of -18^{o}C for lean fish like cod and haddock and -24^{o}C for fatty species like herring. Mostly fruits are stored at between temperatures of -6^{o}C to 10^{o}C. Practically the condenser temperature should be taken more than 5^{o}C to environment temperature. In India, generally the average temperature of environment is 38^{o}C in summer and 20^{o}C in winter.

The input parameters for the thermodynamic analysis are given below:

Evaporator-I Temperature	T_{e1} = -30^{0}C to -15^{0}C
Evaporator-II Temperature	T_{e2} = -6^{0}C to 10^{0}C
Condenser Temperature	T_c = 25^{0}C to 44^{0} C
Refrigeration Capacity of Evaporator-1	Q_1 = 2 TR
Refrigeration Capacity of Evaporator-2	Q_2 = 10 TR
Isentropic Efficiency for Compressor-1	η_1 = 0.9
Isentropic Efficiency for Compressor-2	η_2 = 0.89
Refrigerants Used	R12, R134a, R436a(0.56R290/0.44R600a), R436b(0.52R290/0.48R600a)

By taking these parameters, a theoretical analysis is carried out of the Multi-Evaporator Refrigeration System. The properties are calculated for pure refrigerants and mixed refrigerants using REFPROP and EES (Engineering Equation Solver) and a parametric analysis is done to see the effect of performance parameters with the variation of these inputs.

CHAPTER 4

RESULTS AND DISCUSSION

A thermodynamic analysis is done with the help of Engineering Equation Solver (EES) and REFPROP of Multi-Evaporator System with individual compressor using Flash Chamber. The input parameters for this analysis has mentioned earlier in chapter 3 except the evaporator temperatures and condenser temperature, whose effects on performance parameters are shown in plots.

In this analysis, we can conclude that for a multi evaporator refrigeration system using blend of R290/R600a at evaporator-1 temperature of -30^oC, evaporator-2 temperature of 2^oC and condenser temperature of 40^oC, the best result is found when the proportion is between 0.5 (R290) to 0.6 (R290). The performance parameters are calculated for the five proportion of blend of R290 and R600a which is shown in Table 2.

RATIO(R290/R600a)	0.7/0.3	0.6/0.4	0.56/0.44	0.52/0.48	0.5/0.5
W_{total} (kW)	9.649	9.542	9.501	9.466	9.45
COP	4.373	4.423	4.442	4.458	4.465
ED_{e1} (kW)	0.4965	0.4854	0.4802	0.4756	0.473
ED_{e2} (kW)	1.897	1.859	1.831	1.83	1.819
ED_{c1} (kW)	0.1192	0.1194	0.1184	0.1181	0.117
ED_{c2} (kW)	0.8298	0.8222	0.822	0.8178	0.8135
ED_{cond} (kW)	2.862	2.86	2.858	2.865	2.868
ED_{t1} (kW)	0.1283	0.1257	0.1255	0.1242	0.1242
ED_{t2} (kW)	1.27	1.229	1.228	1.197	1.197
ED_{flash} (kW)	0.01366	0.01962	0.02368	0.02202	0.0223
ED (kW)	7.616	7.521	7.488	7.449	7.435
EDR	3.696	3.65	3.634	3.615	3.608
eta_ex	0.2135	0.2159	0.2169	0.2177	0.218

Table 2: comparison of different proportion of blend of R290 and R600a

4.1 Total Work Input for the Compressor

Fig. 13 to Fig. 27 show the variation of total work input for the system for different refrigerants against evaporator-1, evaporator-2 and condenser temperatures respectively. Fig. 17, 22 and 27 show the comparison of variation of total work input for the system for different refrigerants.

Among all the refrigerants used in the system, R436b has the least total work input which is 6% less than the total work input of R134a and R436a has the second least total work input which is 4% less than that of R134a.

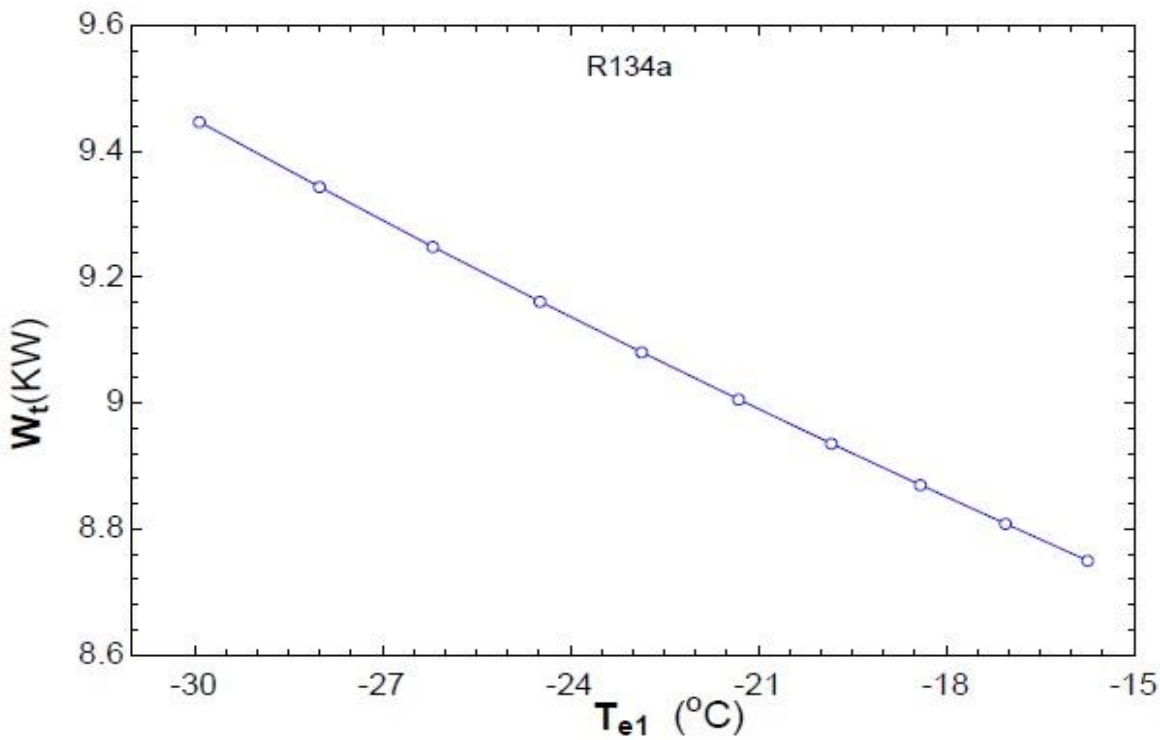

Fig. 13: Variation of total work input with Evaporator Temperature-1 using R134a

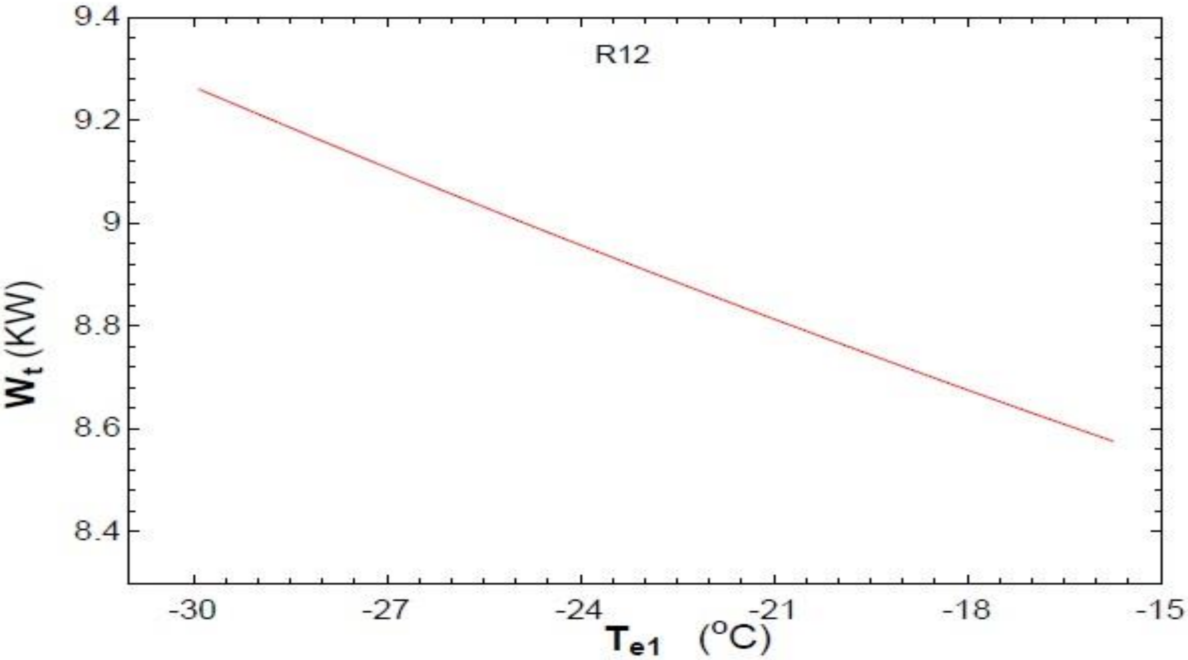

Fig. 14: Variation of total work input with Evaporator Temperature-1 using R12

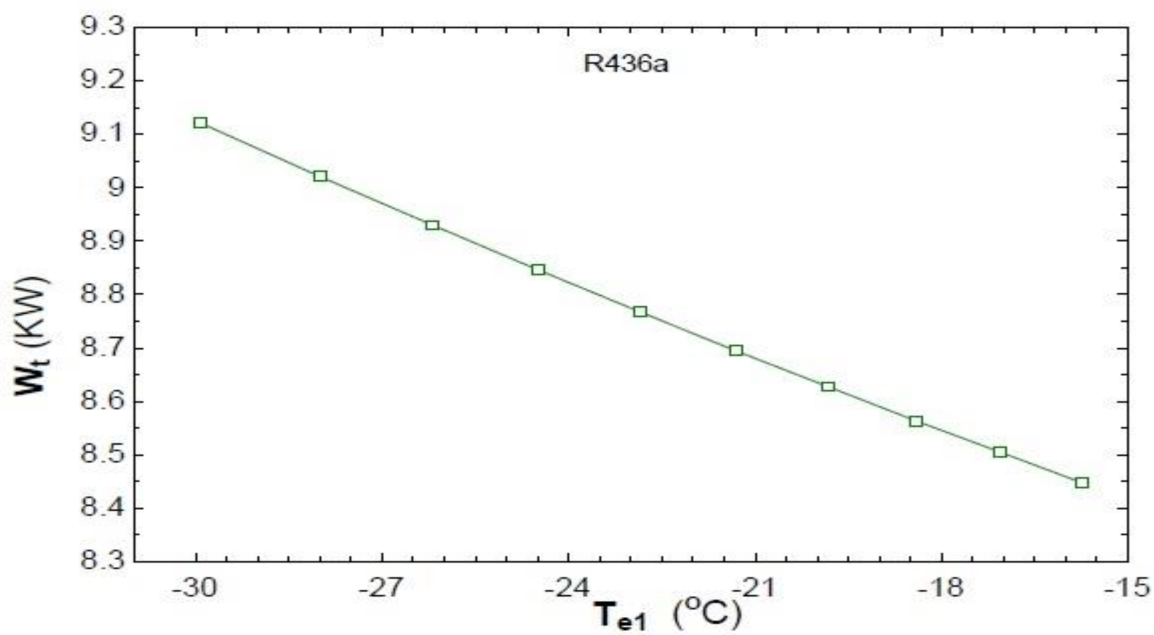

Fig. 15: Variation of total work input with Evaporator Temperature-1 using R436a

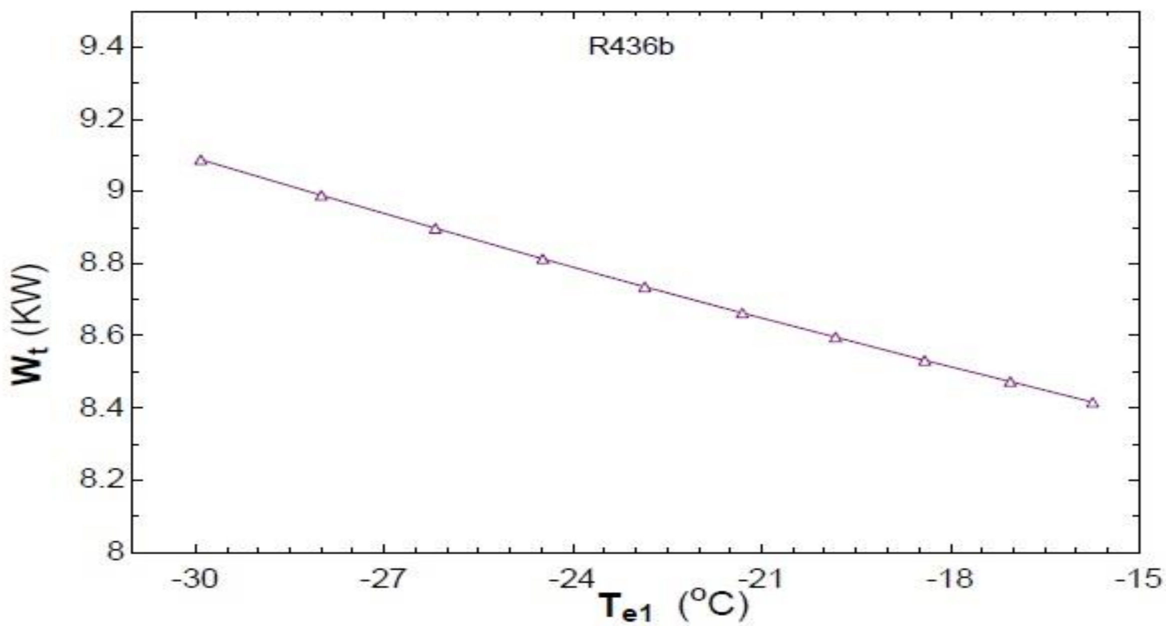

Fig. 16: Variation of total work input with Evaporator Temperature-1 using R436b

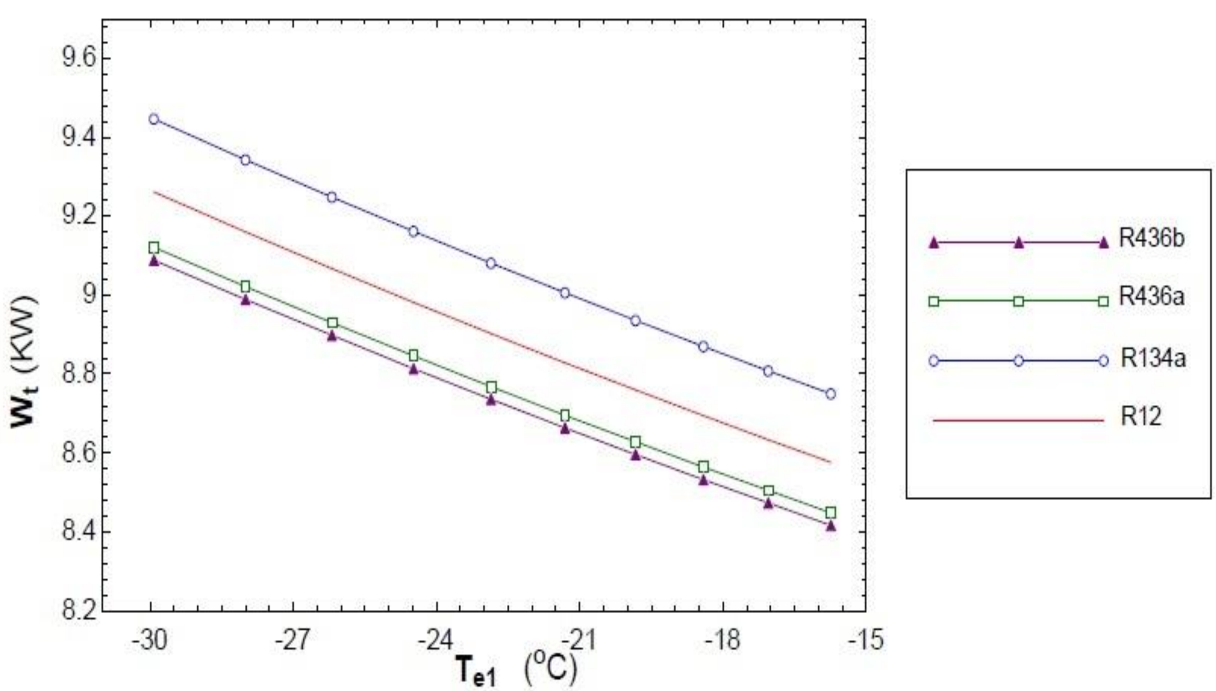

Fig. 17: Comparison of Variation of total work input with Evaporator Temperature-1

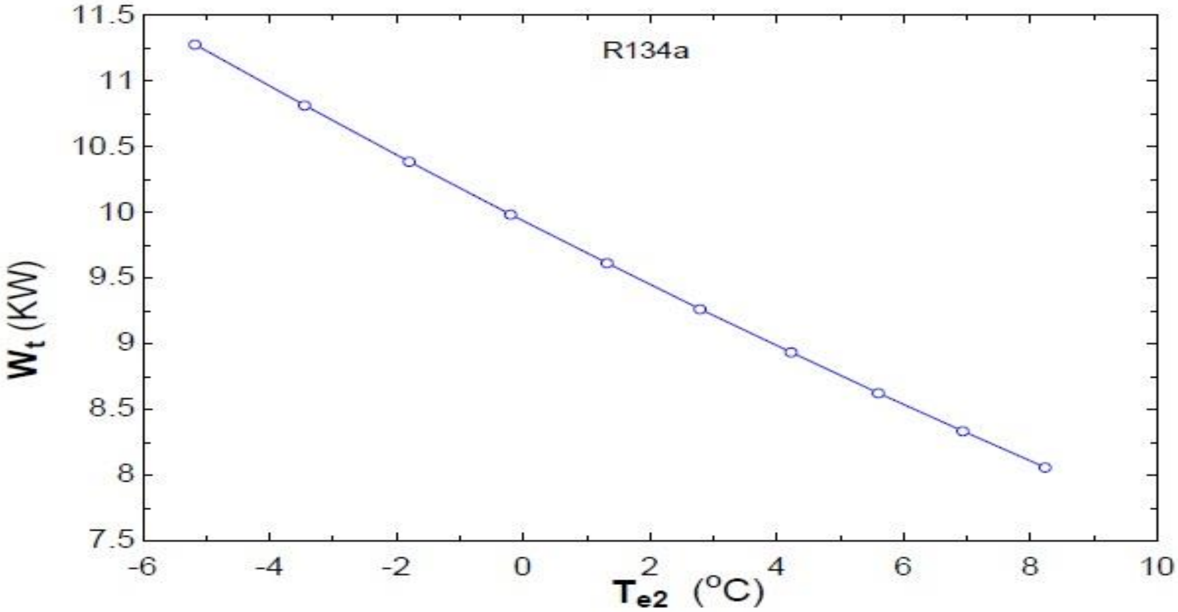

Fig. 18: Variation of total work input with Evaporator Temperature-2 using R134a

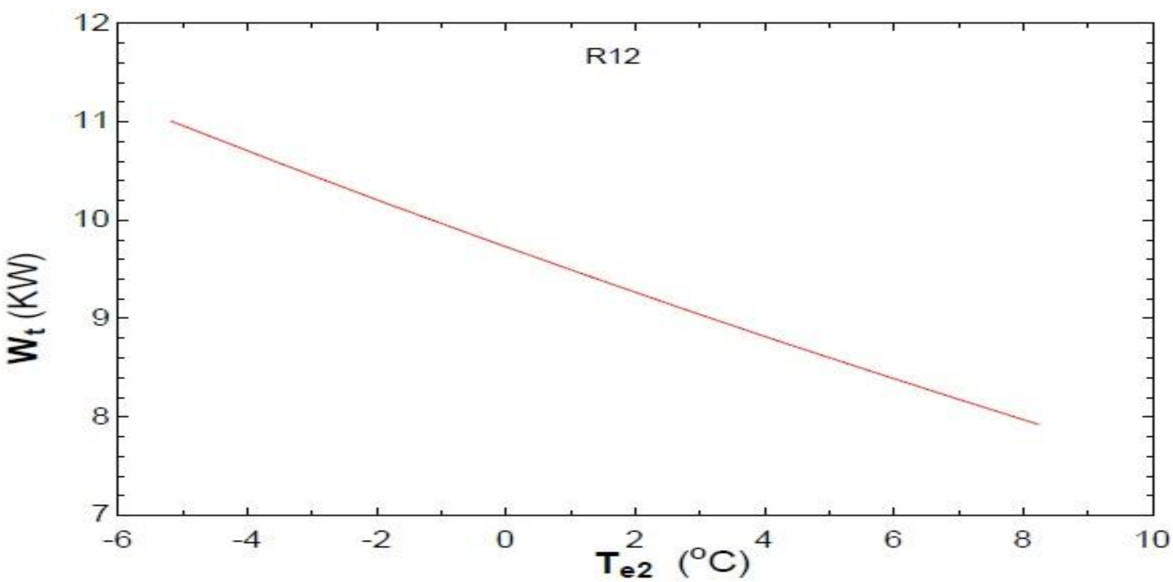

Fig. 19: Variation of total work input with Evaporator Temperature-2 using R12

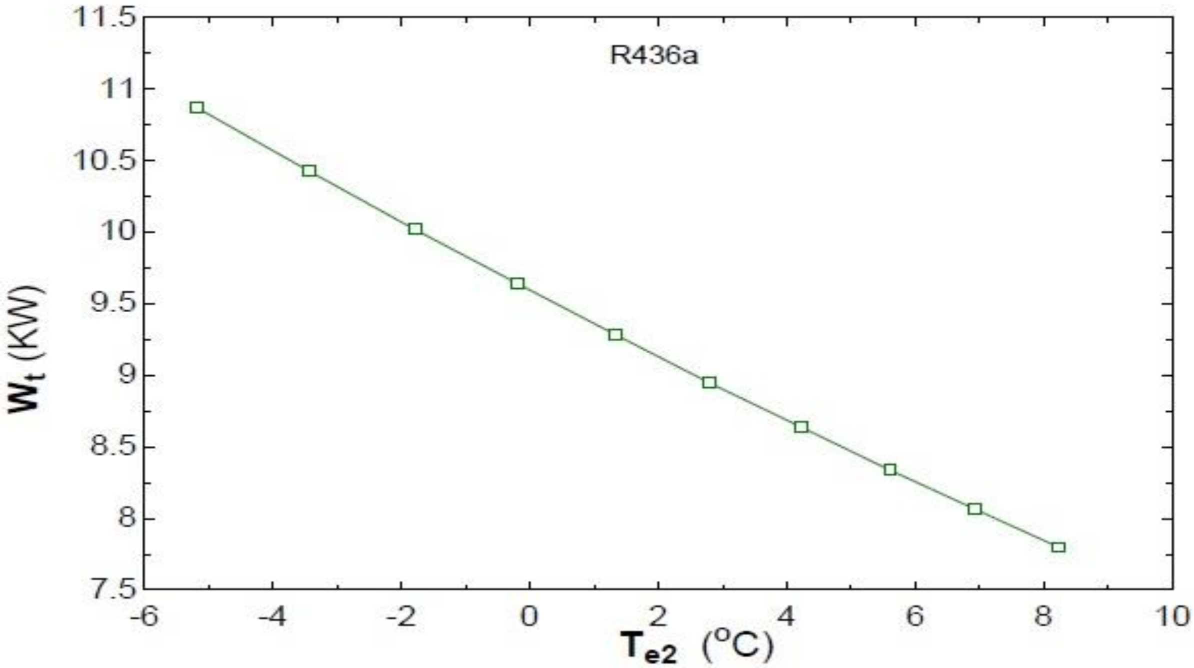

Fig. 20: Variation of total work input with Evaporator Temperature-2 using R436a

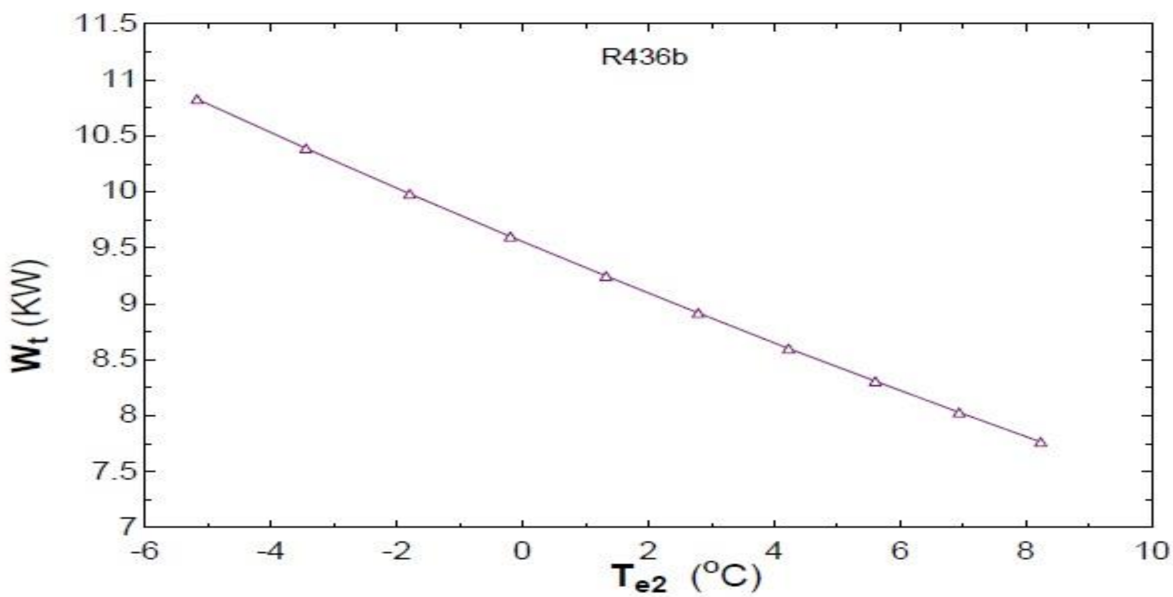

Fig. 21: Variation of total work input with Evaporator Temperature-2 using R436b

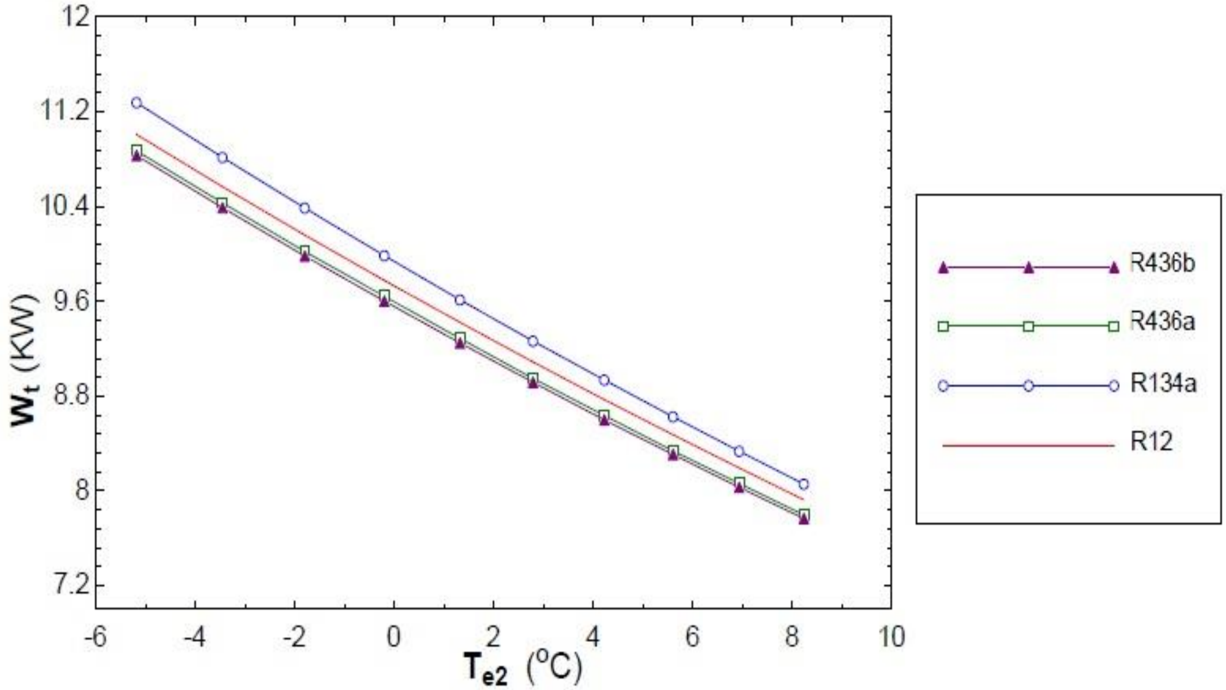

Fig. 22: Comparison of Variation of total work input with Evaporator Temperature-2

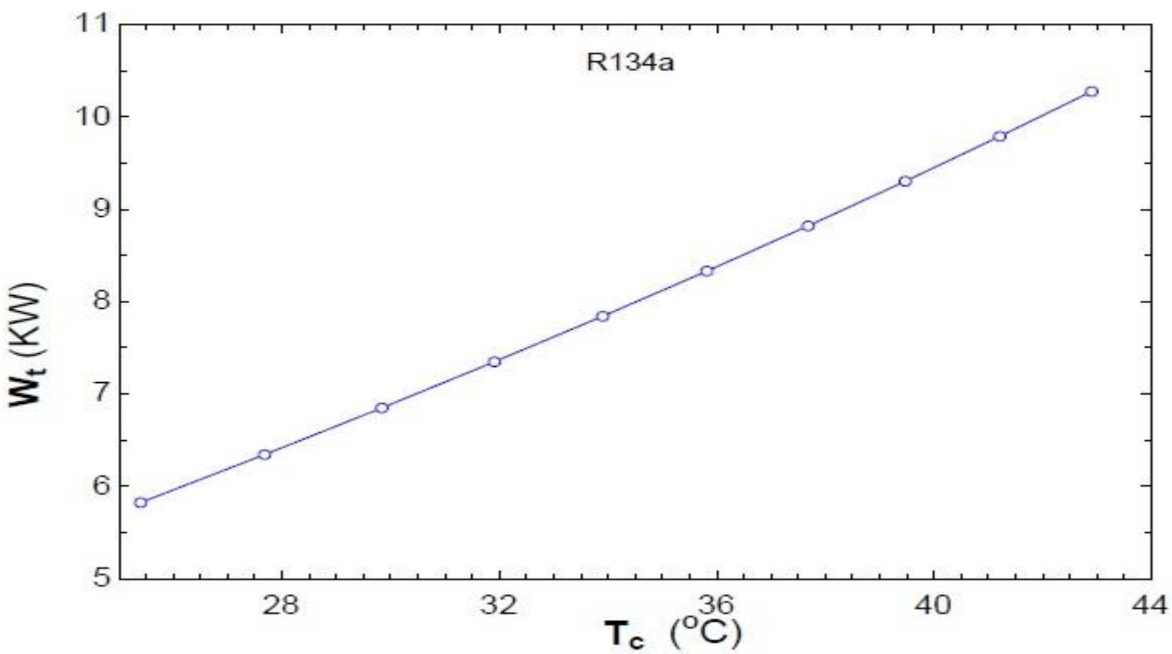

Fig. 23: Variation of total work input with Condenser Temperature using R134a

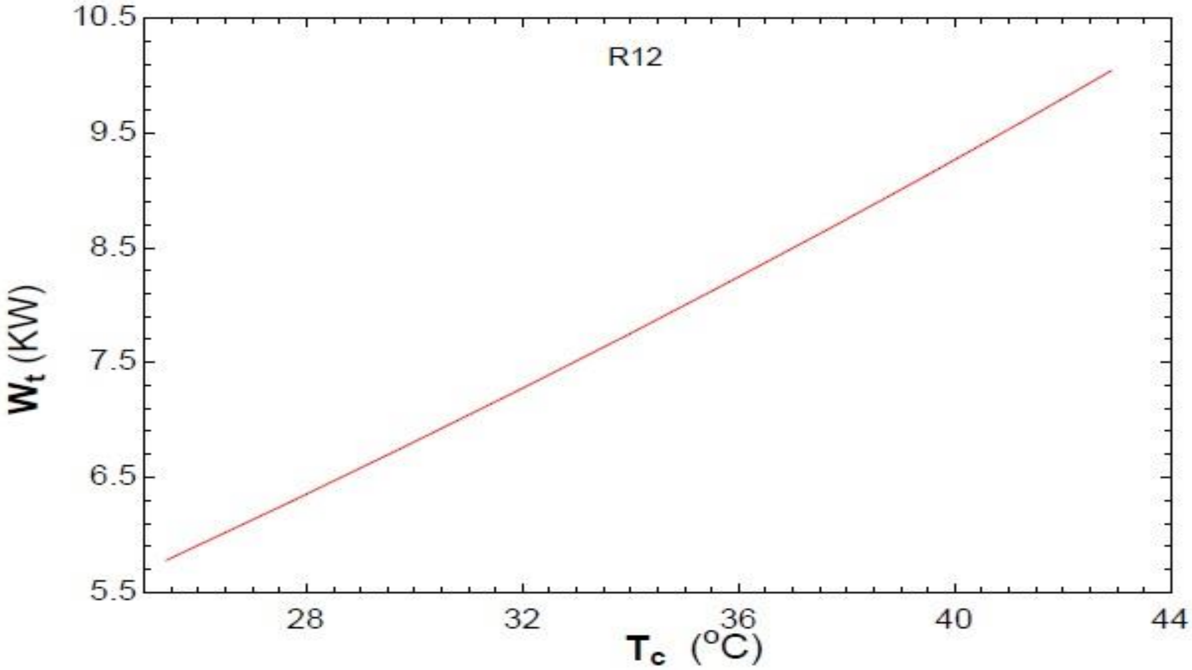

Fig. 24: Variation of total work input with Condenser Temperature using R12

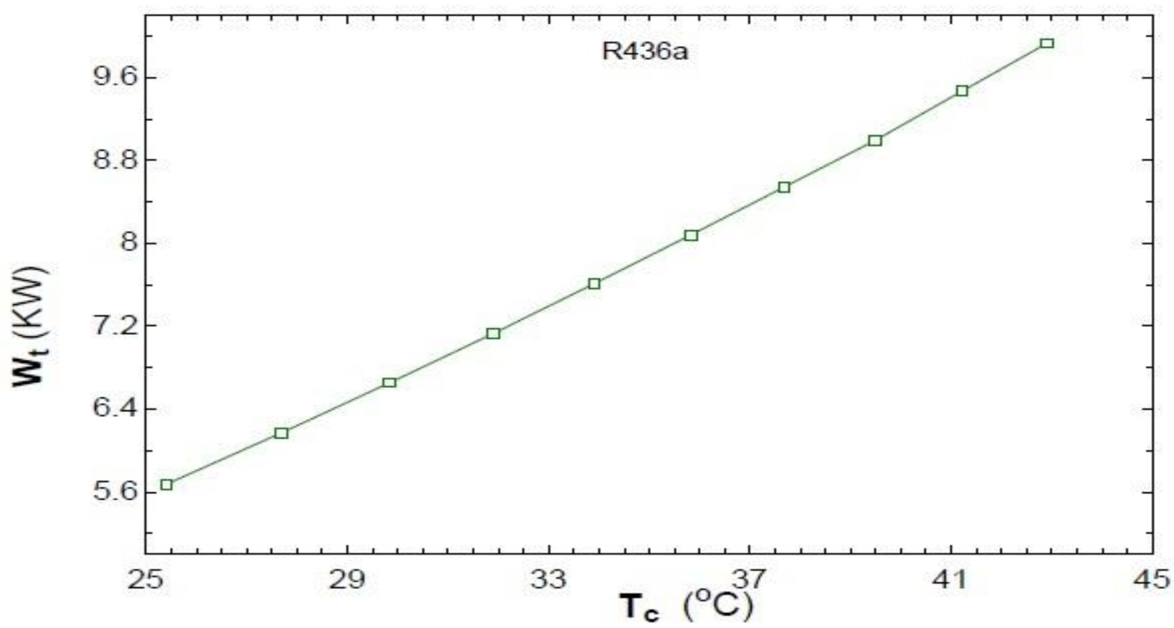

Fig. 25: Variation of total work input with Condenser Temperature using R436a

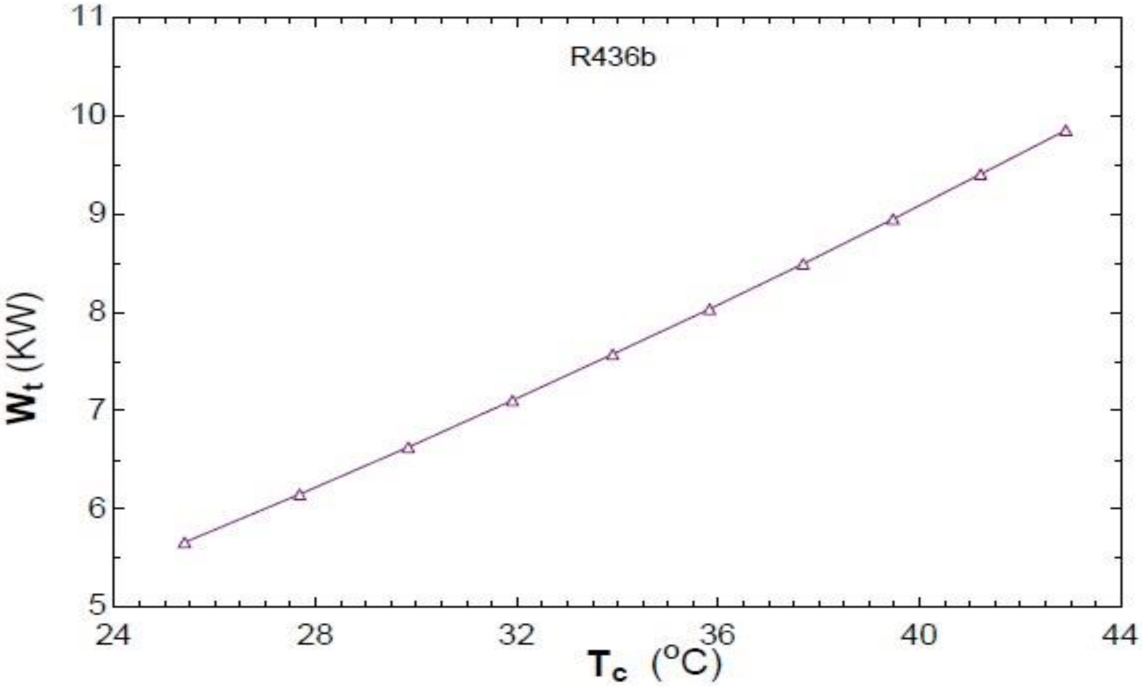

Fig. 26: Variation of total work input with Condenser Temperature using R436b

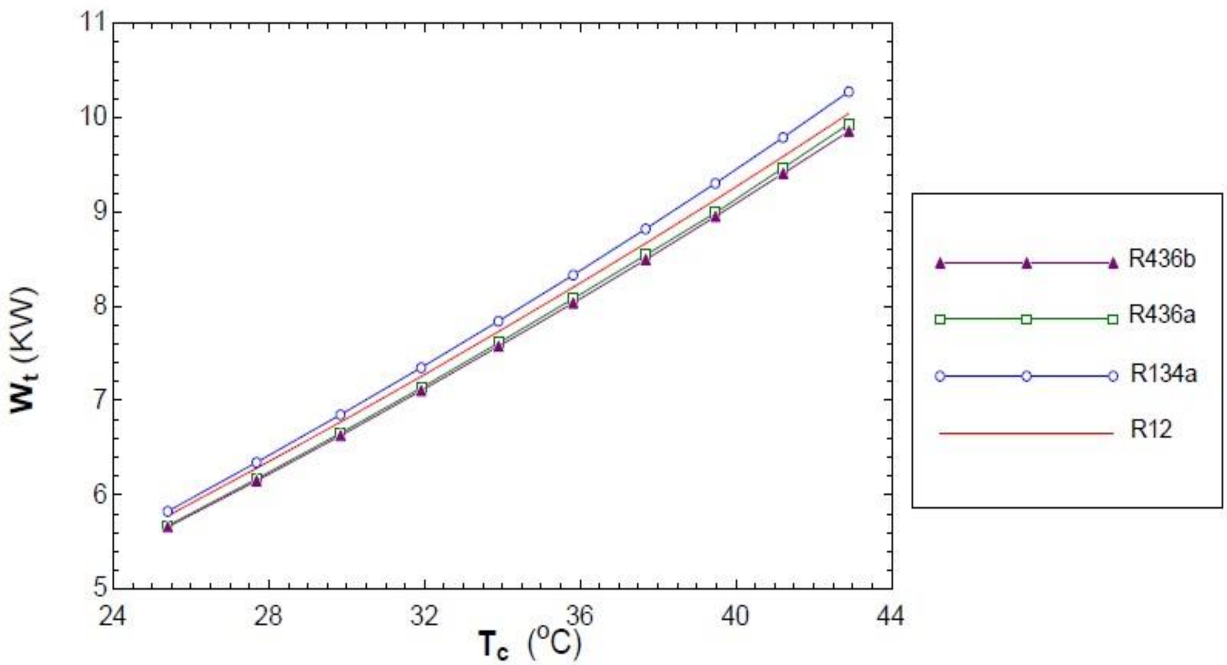

Fig. 27: Comparison of Variation of total work input with Condenser Temperature

4.2 Coefficient of Performance

Fig. 28 to Fig. 42 show the variation of coefficient of performance using different refrigerants against evaporator-1, evaporator-2 and condenser temperatures respectively. Fig. 32, 37 and 42 show the comparison of variation of COP for different refrigerants with different evaporating and condensing temperatures. If we increase the evaporator temperatures, ultimately refrigerating effect is increased but when we go on increasing condenser temperature, refrigeration effect decreases, thus the coefficient of performance also decreases.

Among all the different refrigerants, R436b has the maximum COP among all refrigerants which is 4.0% more than COP of R134a corresponding to condenser temperature considered and R436a has second best COP among all refrigerant which is 3.5% more than COP of R134a. R134a has least COP compared to all selected refrigerants which is used as a working fluid in the refrigerator.

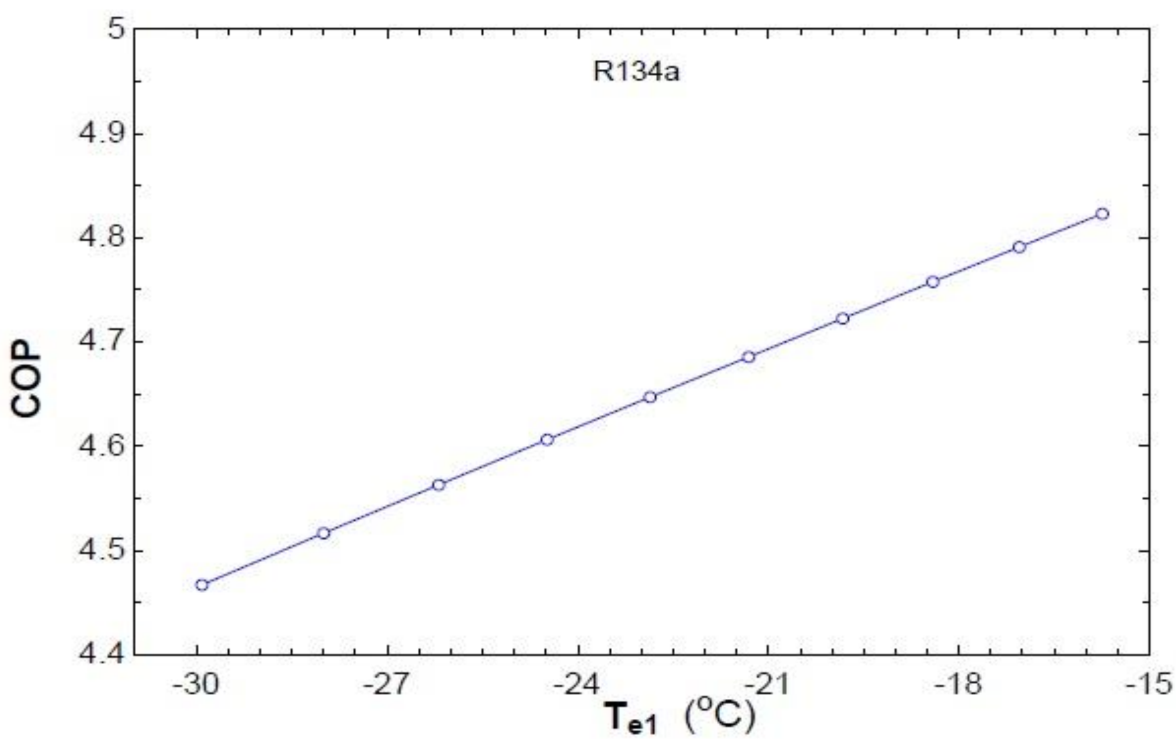

Fig. 28: Variation of COP with Evaporator-I Temperature using R134a

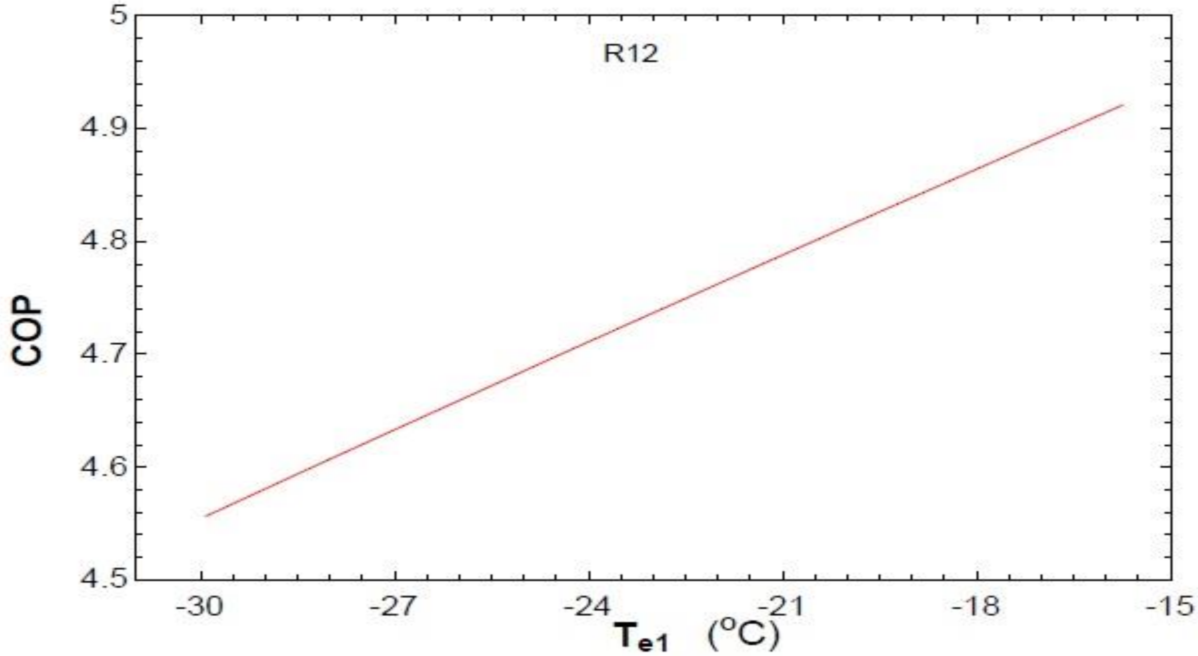

Fig. 29: Variation of COP with Evaporator-I Temperature using R12

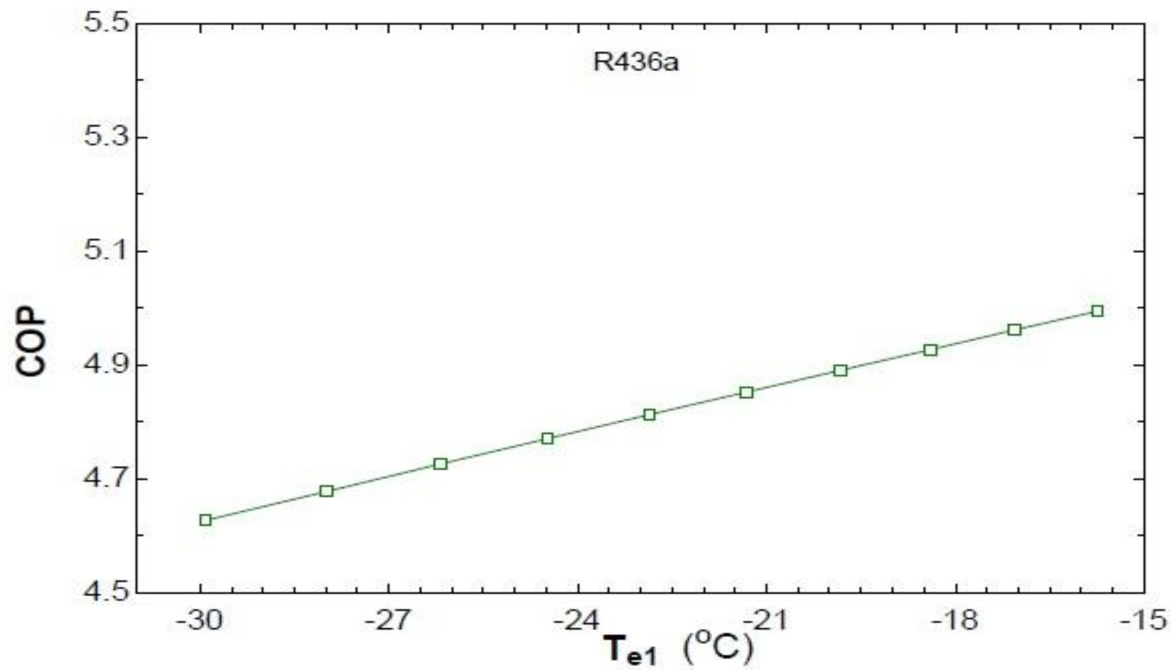

Fig. 30: Variation of COP with Evaporator-I Temperature using R436a

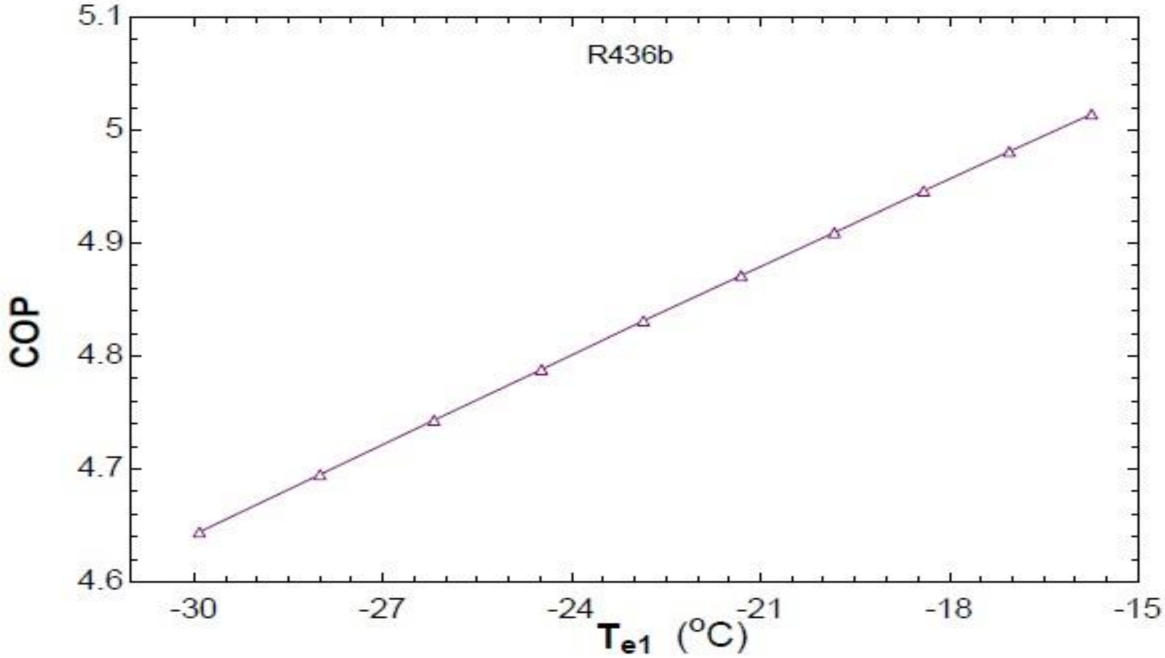

Fig. 31: Variation of COP with Evaporator-I Temperature using R436b

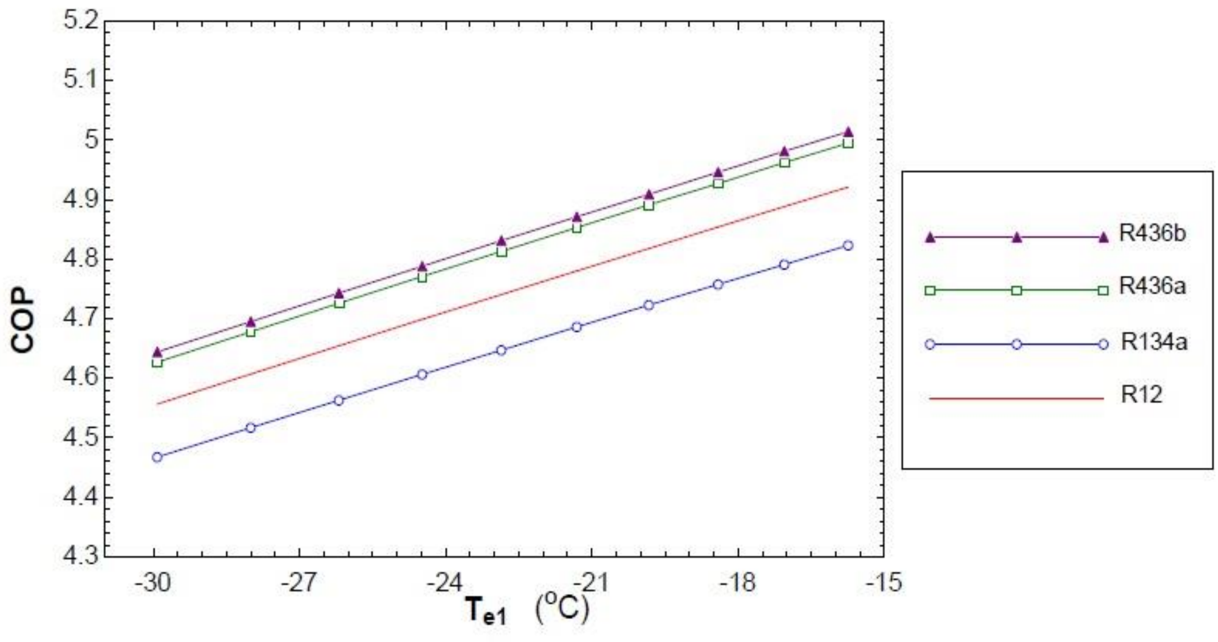

Fig. 32: Comparison of Variation of COP with Evaporator-I Temperature

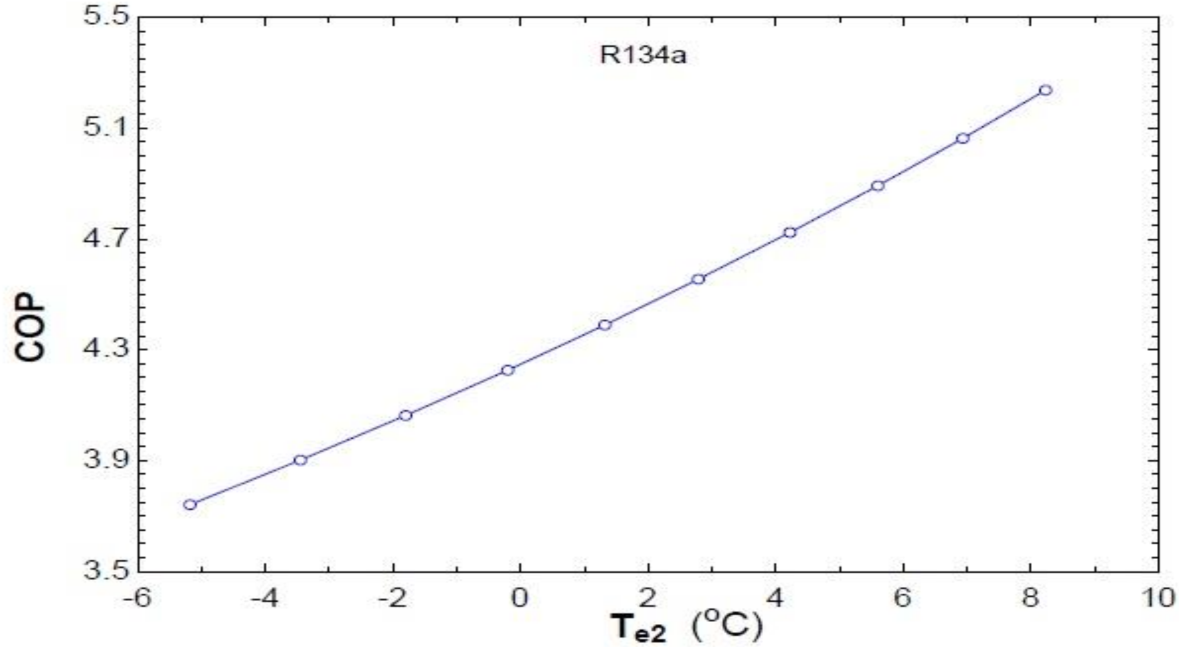

Fig. 33: Variation of COP with Evaporator Temperature-2 using R134a

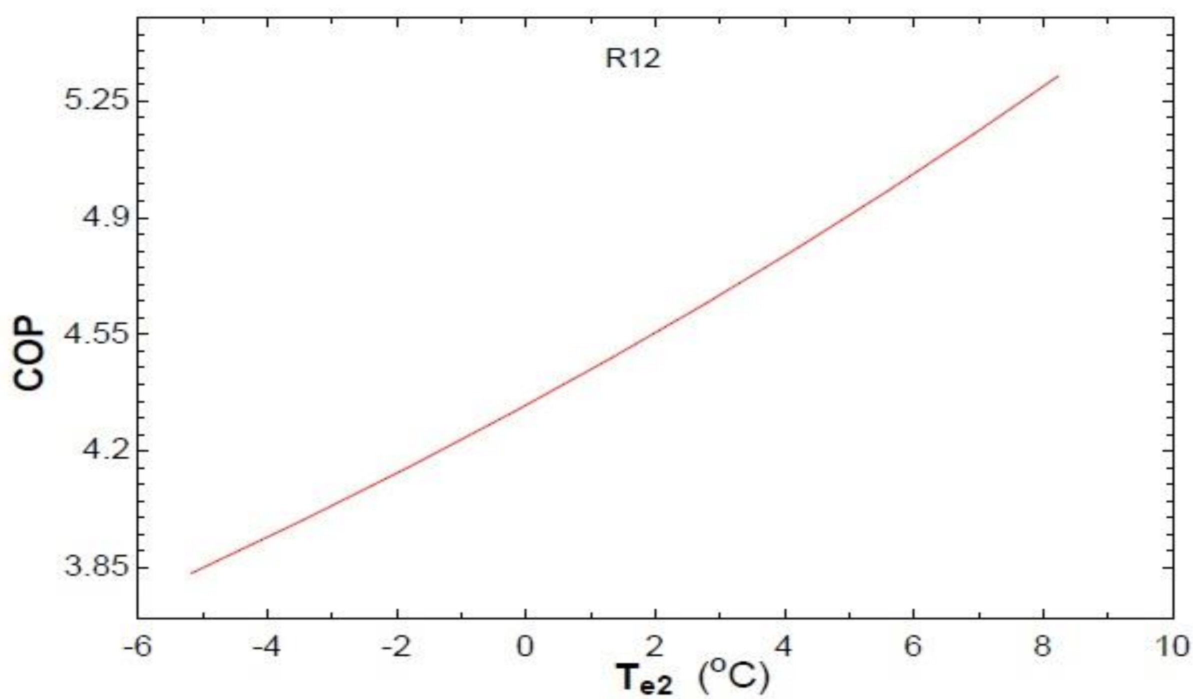

Fig. 34: Variation of COP with Evaporator Temperature-2 using R12

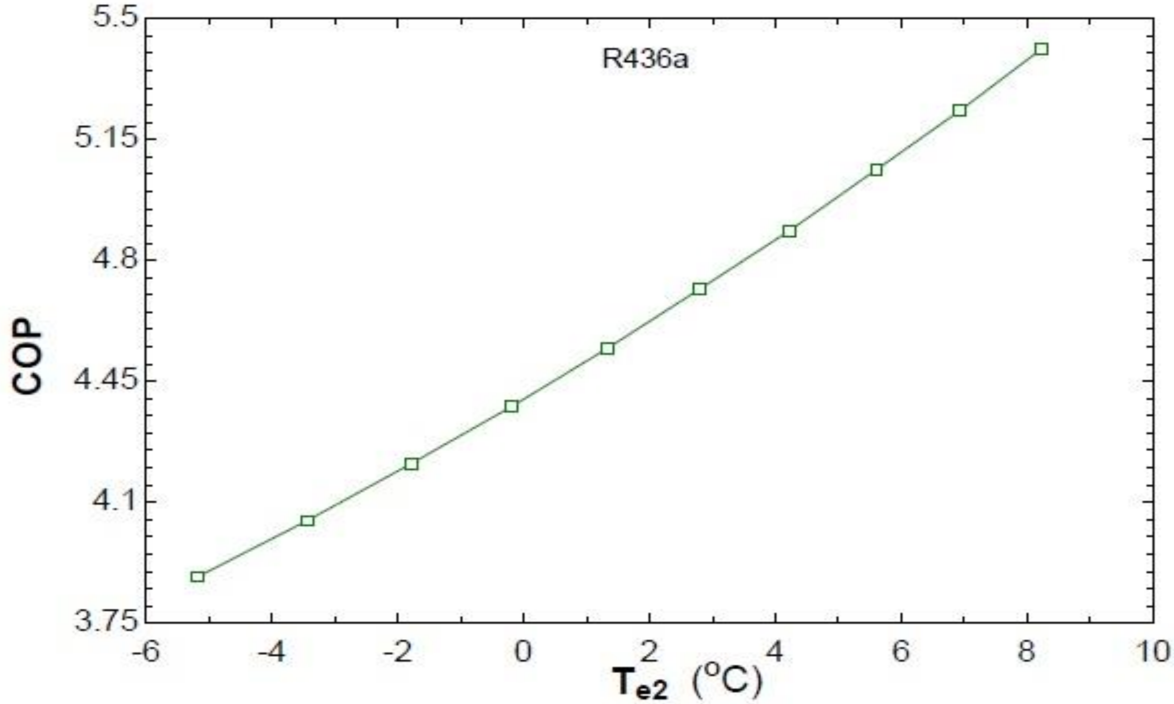

Fig. 35: Variation of COP with Evaporator Temperature-2 using R436a

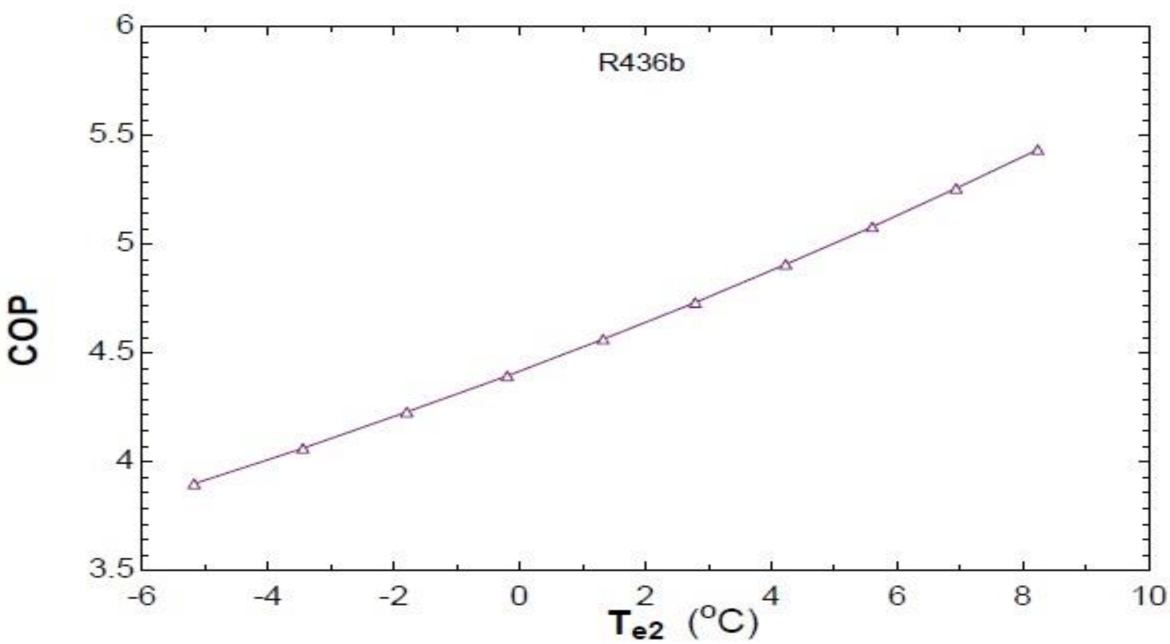

Fig. 36: Variation of COP with Evaporator Temperature-2 using R436b

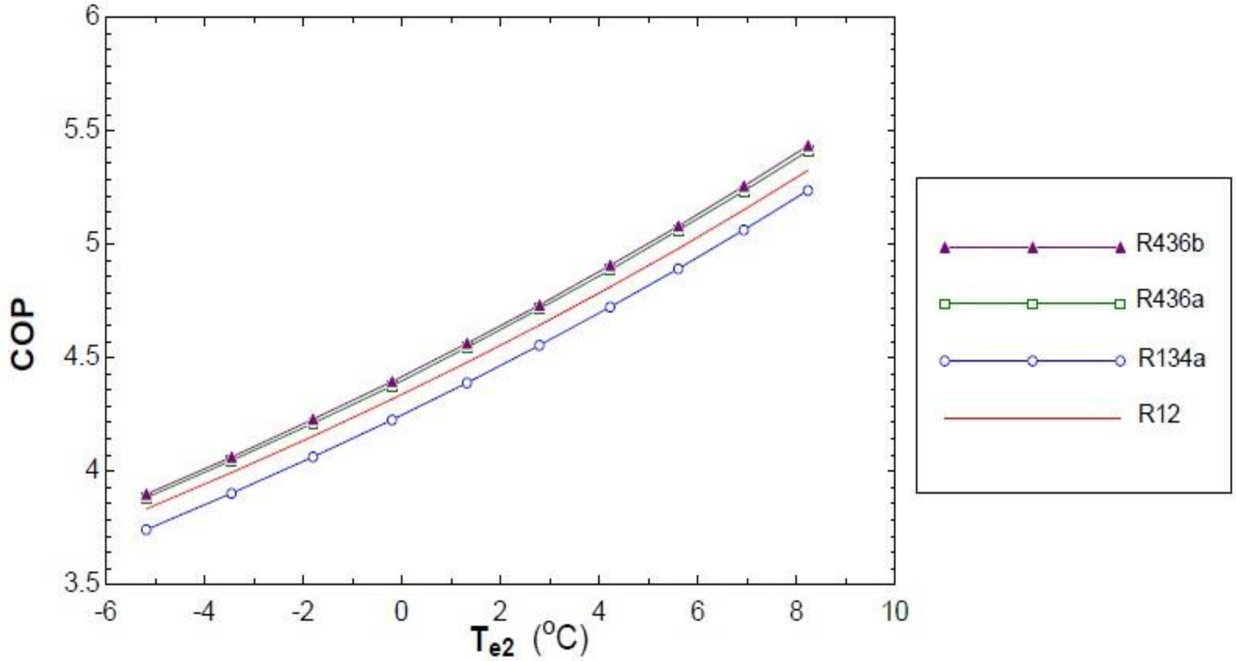

Fig. 37: Comparison of Variation of COP with Evaporator Temperature-2

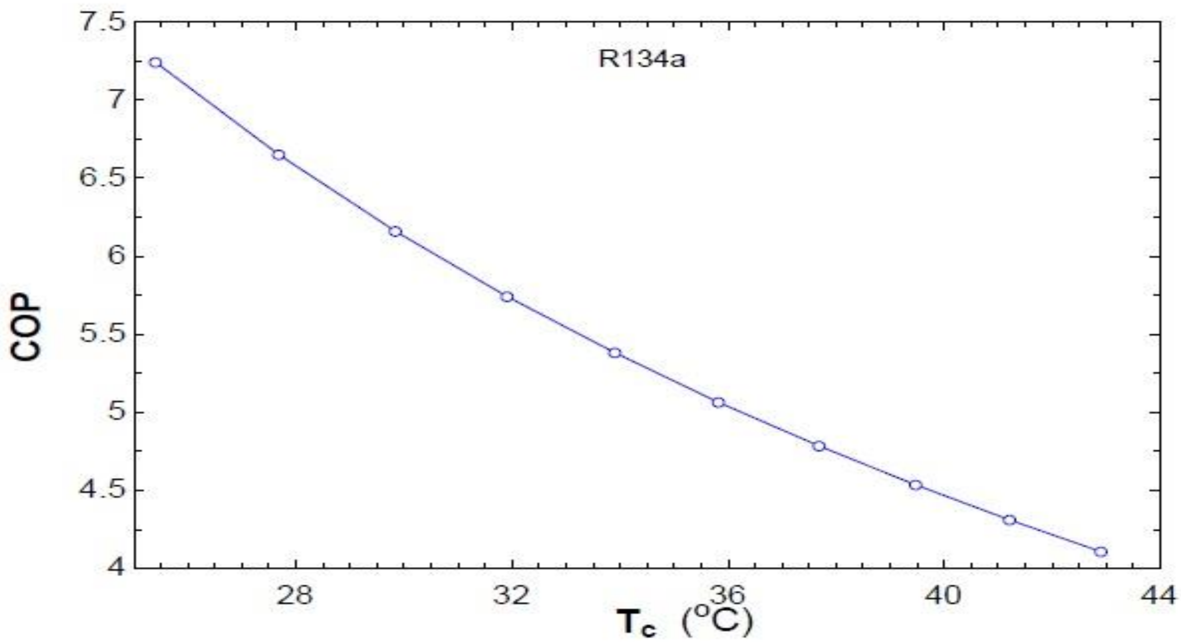

Fig. 38: Variation of COP with Condenser Temperature using R134a

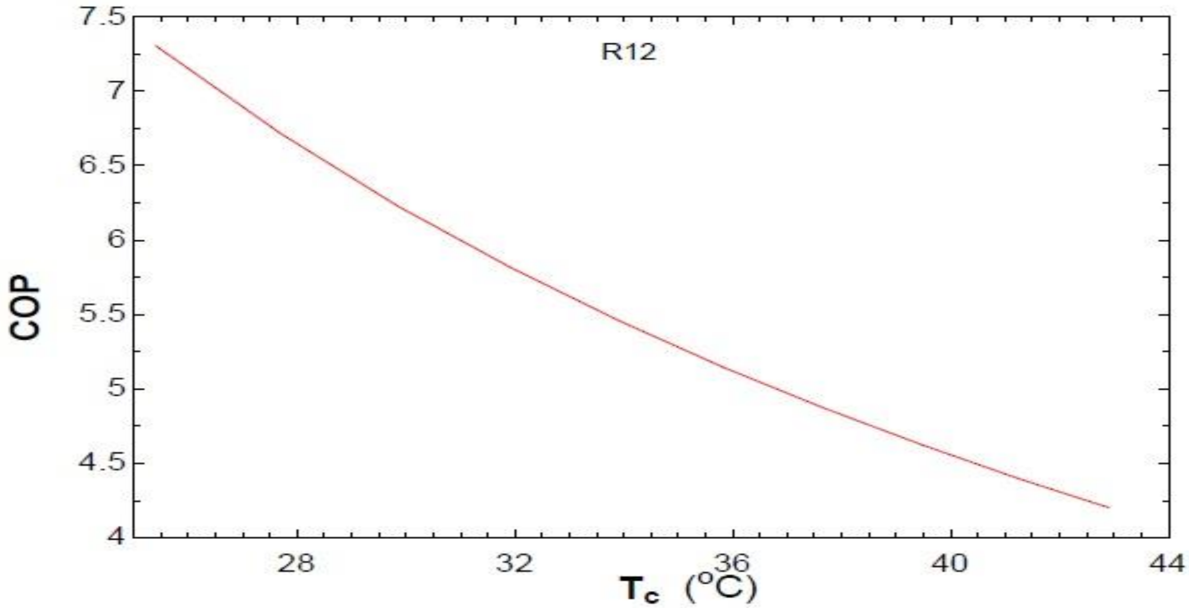

Fig. 39: Variation of COP with Condenser Temperature using R12

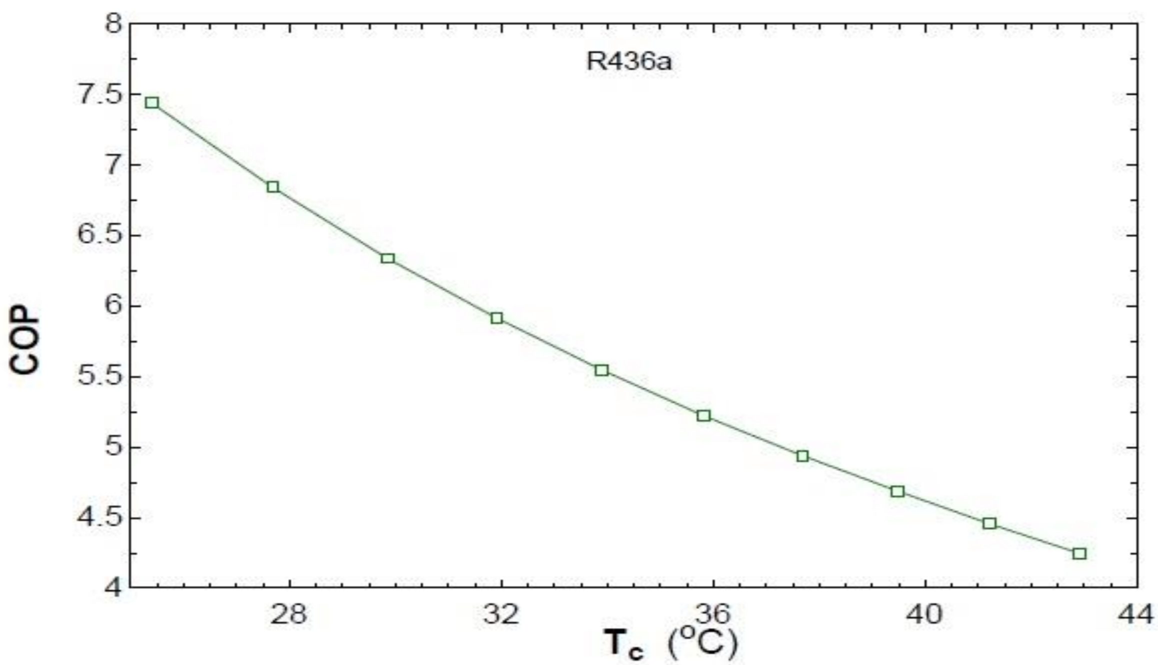

Fig. 40: Variation of COP with Condenser Temperature using R436a

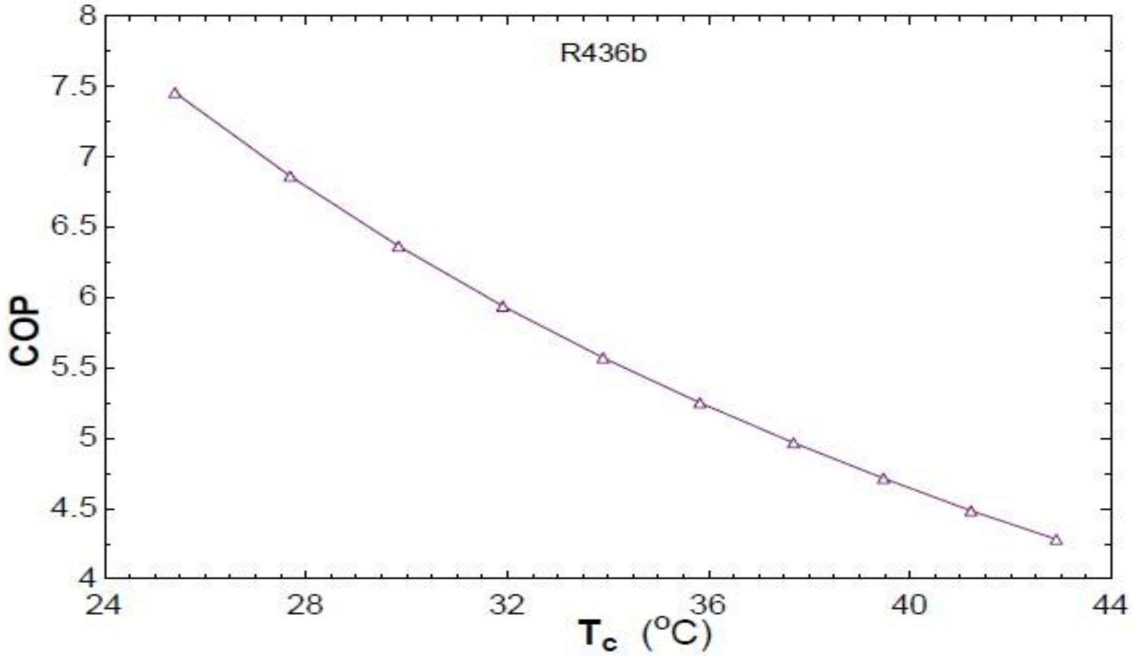

Fig. 41: Variation of COP with Condenser Temperature using R436b

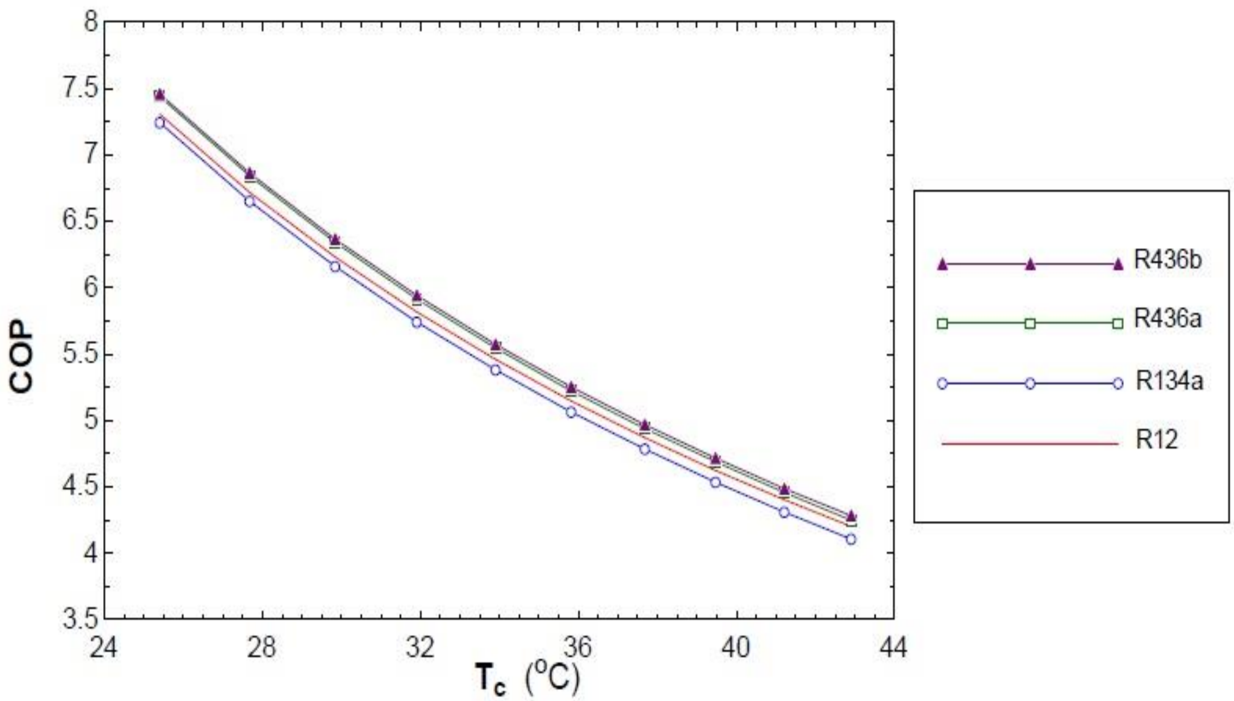

Fig. 42: Comparison of Variation of COP with Condenser Temperature

4.3 Exergy Destruction for the System

Fig. 43 to Fig. 57 show the variation of exergy destruction using different refrigerants alternatively against evaporator-1, evaporator-2 and condenser temperatures respectively. Fig. 47, 52 and 57 show the comparison of variation of exergy destruction for different refrigerants. Exergy destruction is directly related to the irreversibility. It tells us how much irreversibility is there in the system.

Among all the different refrigerants, R436b has the least ED among all refrigerants which is 4.1% less than ED of R134a corresponding to condenser temperature considered and R436a has second least ED among all refrigerant which is 3.7% less than ED of R134a. R134a has highest ED compared to all selected refrigerants.

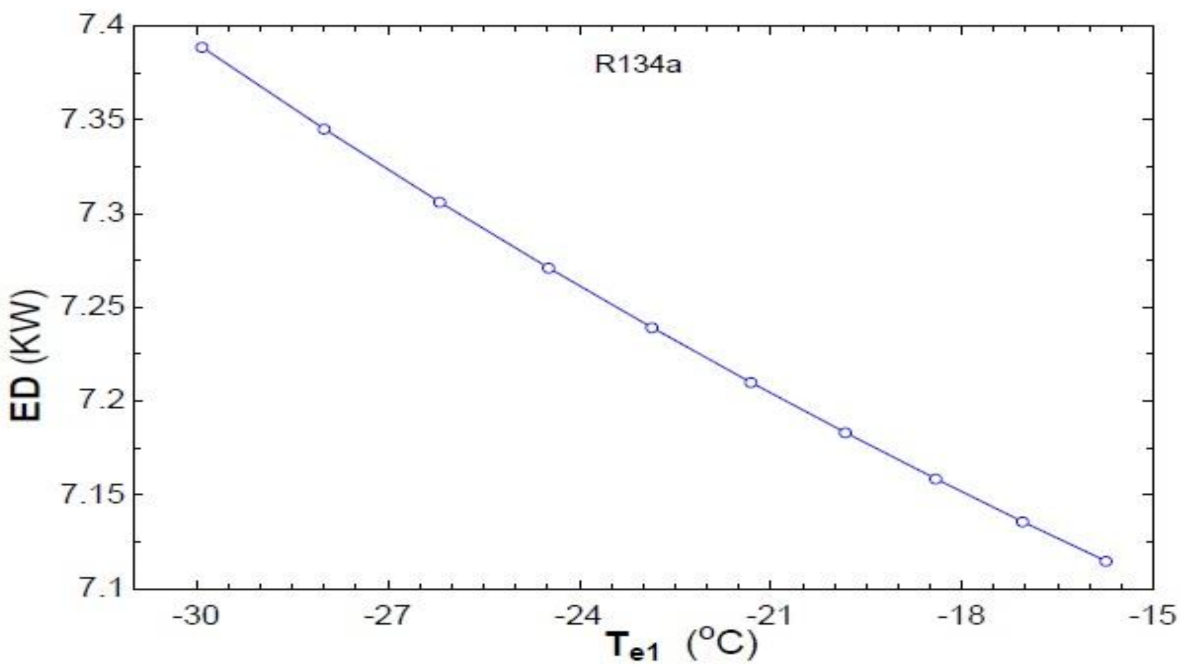

Fig. 43: Variation of Exergy Destruction with Evaporator temperature-1 using R134a

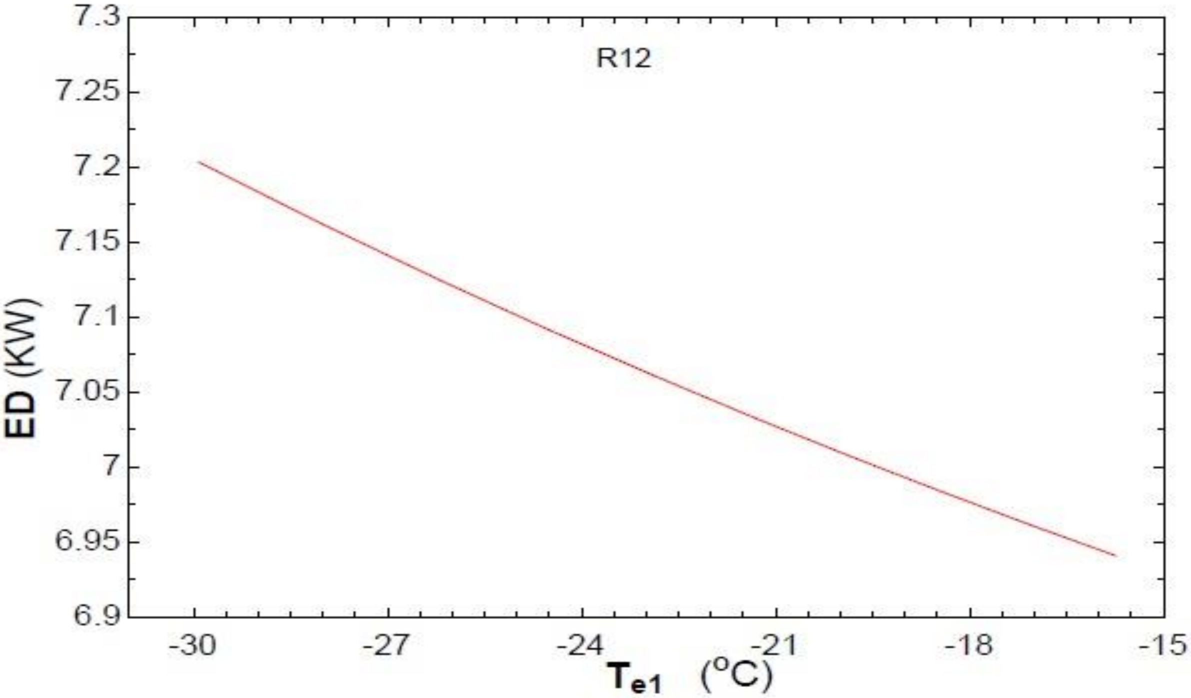

Fig. 44: Variation of Exergy Destruction with evaporator temperature-1 using R12

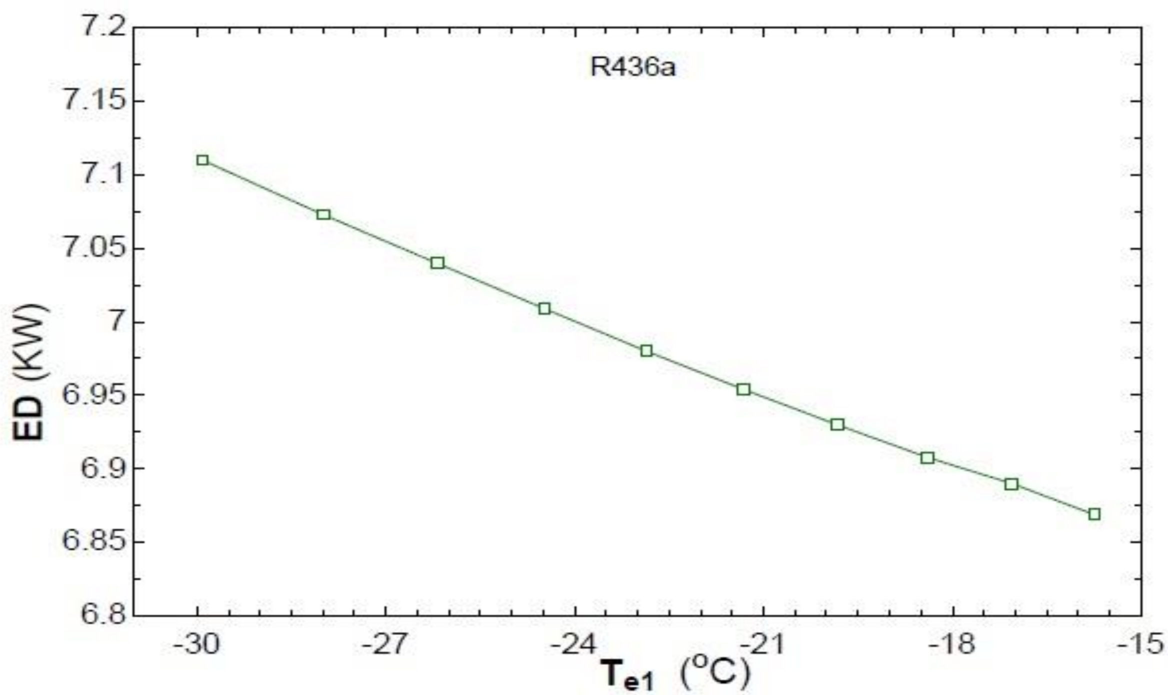

Fig. 45: Variation of Exergy Destruction with evaporator temperature-1 using R436a

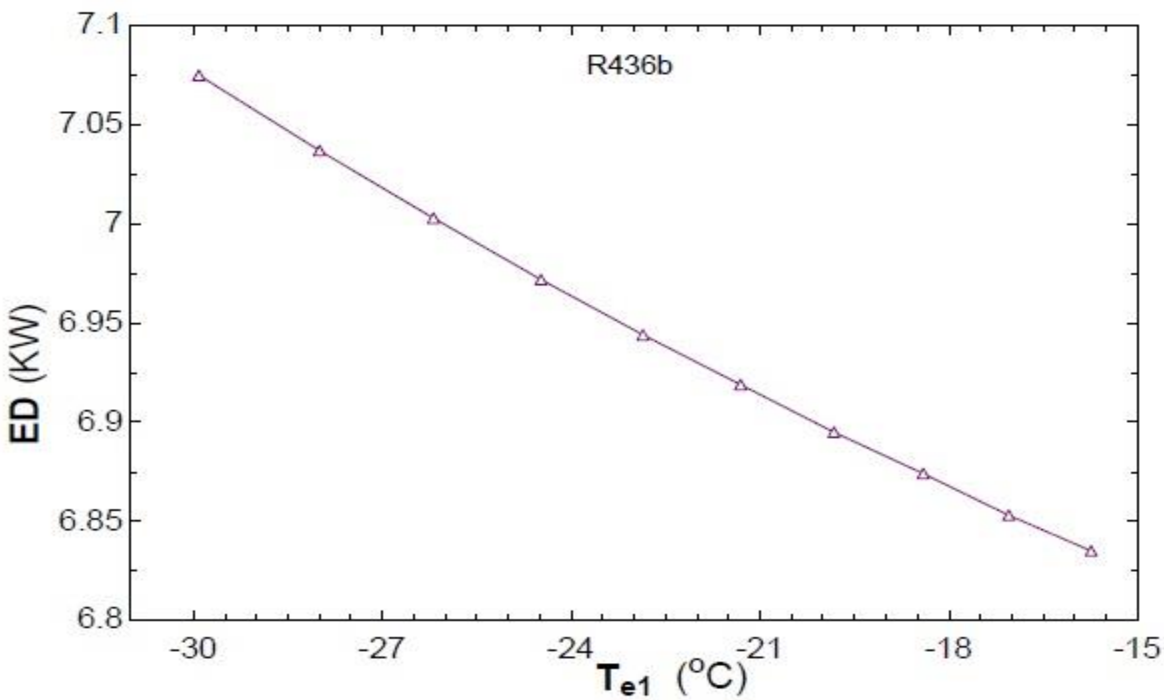

Fig. 46: Variation of Exergy Destruction with evaporator temperature-1 using R436b

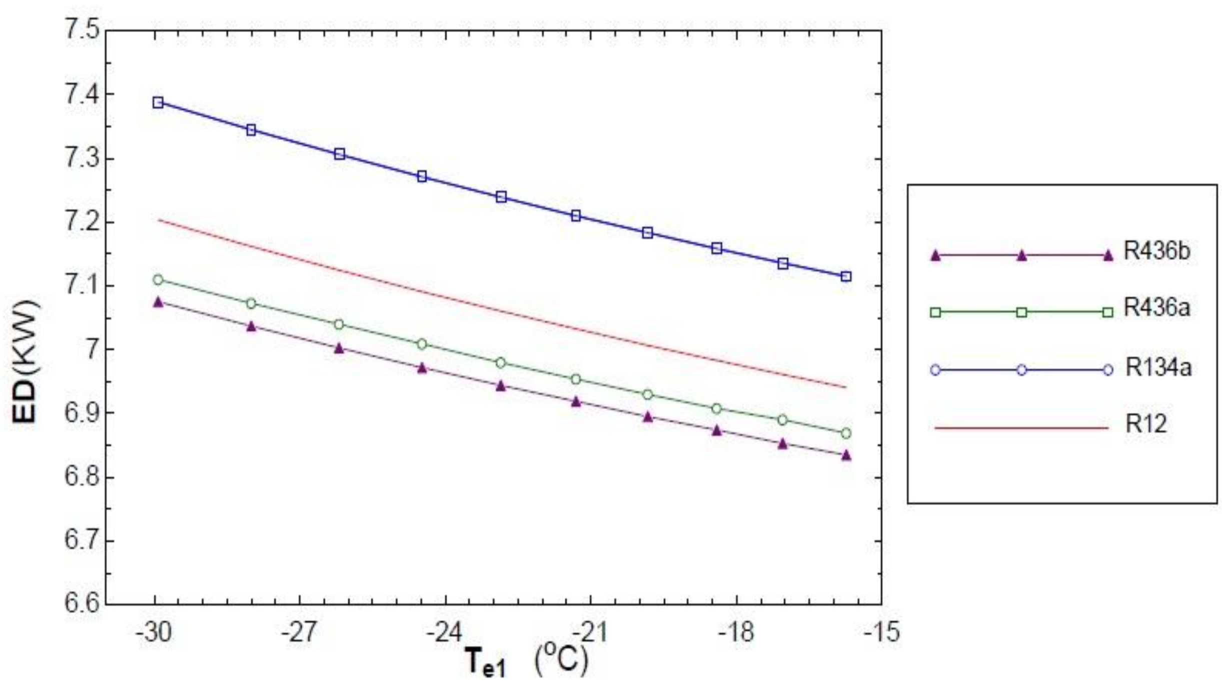

Fig. 47: Comparison of Variation of Exergy Destruction with evaporator temperature-1

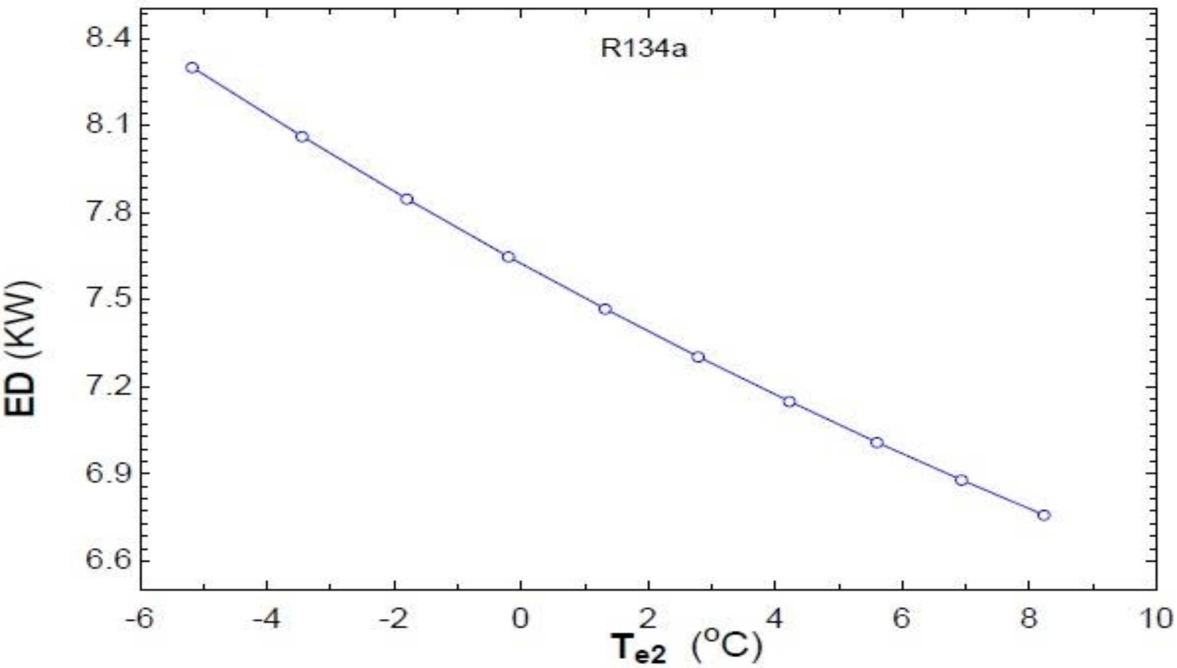

Fig. 48: Variation of Exergy Destruction with evaporator temperature-2 using R134a

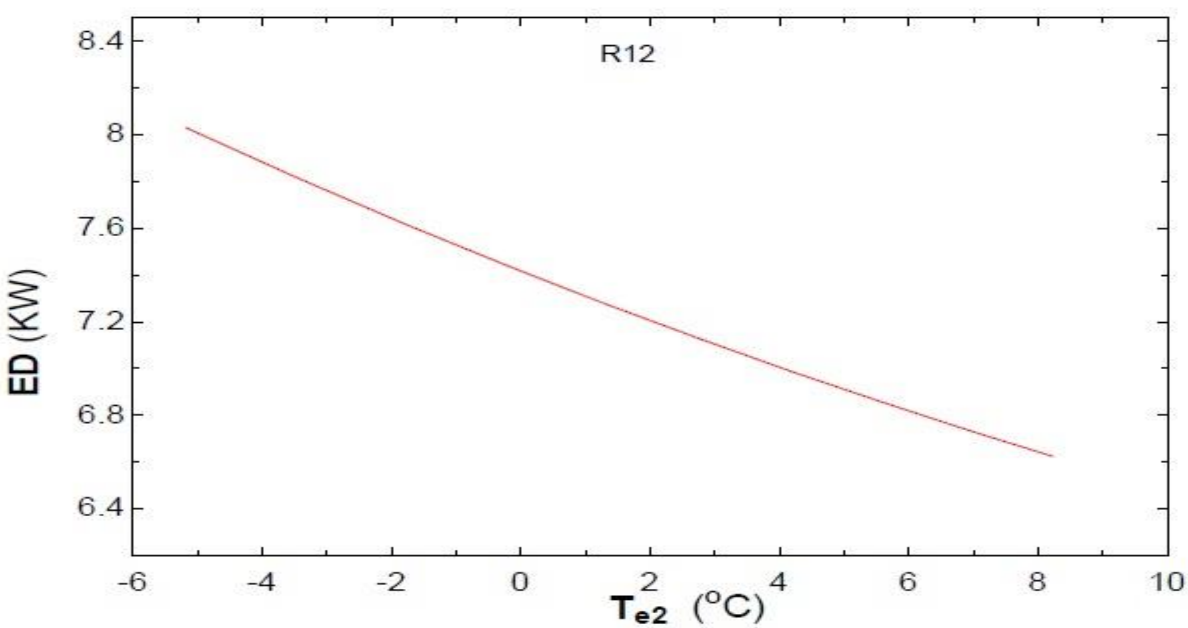

Fig. 49: Variation of Exergy Destruction with evaporator temperature-2 using R12

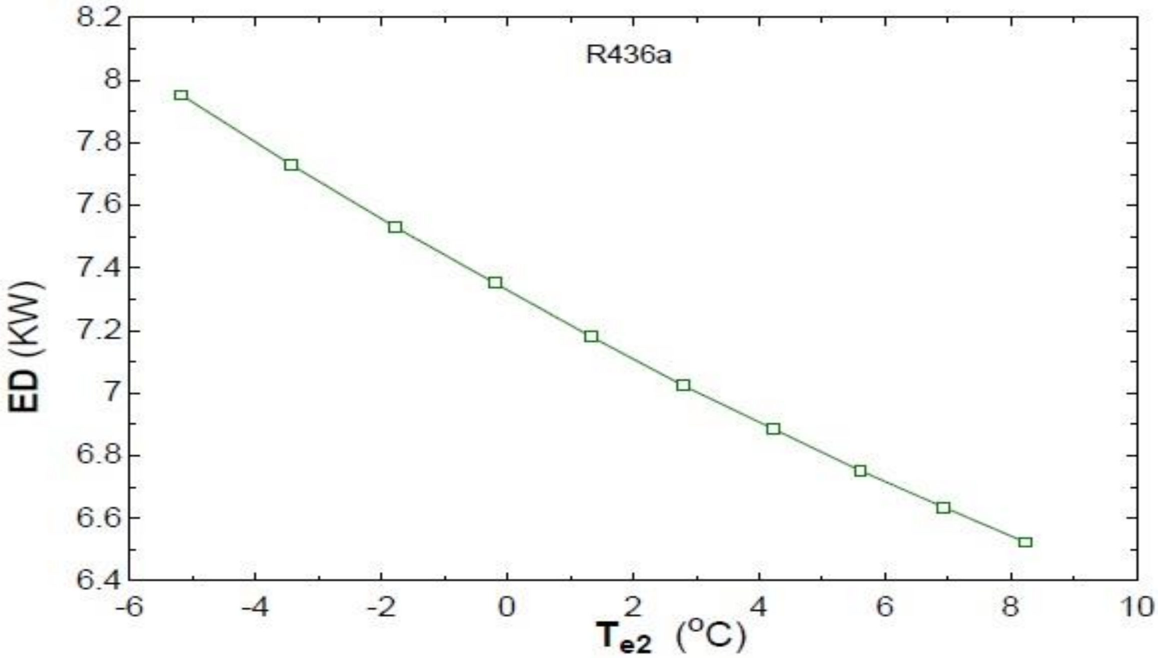

Fig. 50: Variation of Exergy Destruction with evaporator temperature-2 using R436a

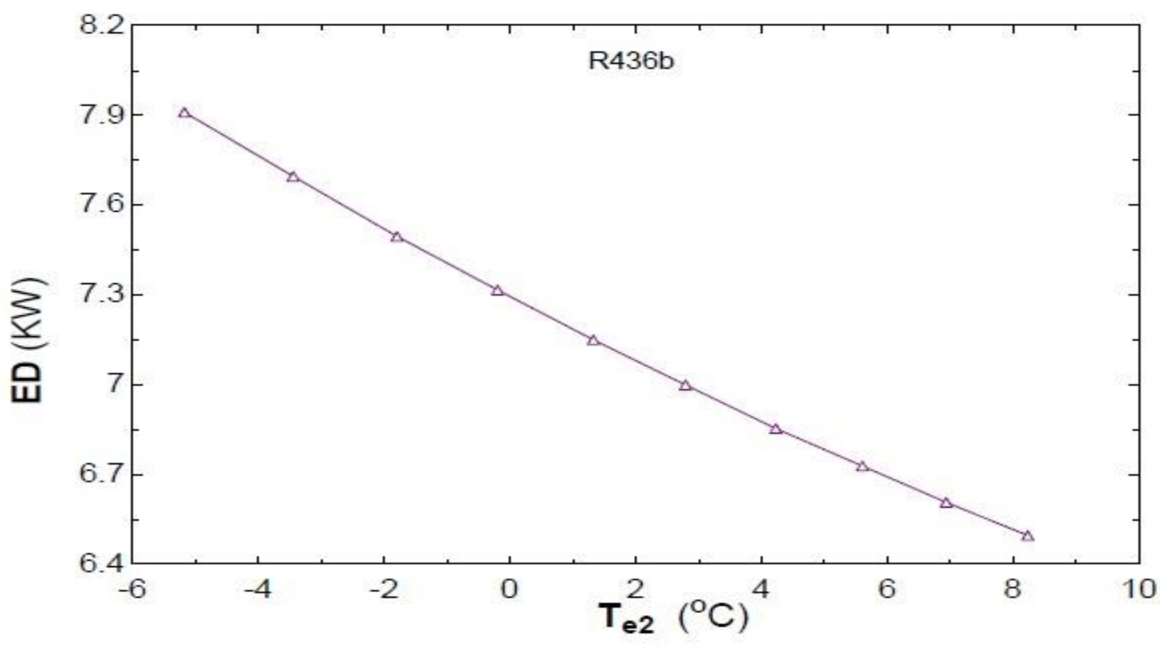

Fig. 51: Variation of Exergy Destruction with evaporator temperature-2 using R436b

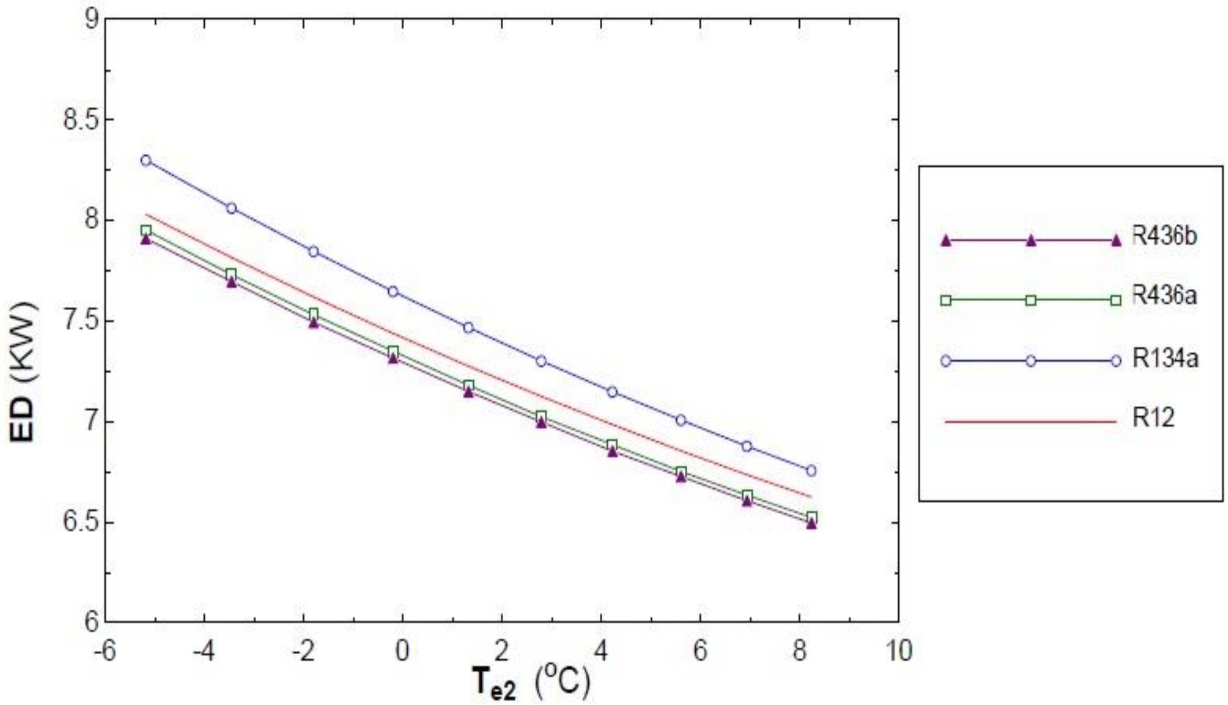

Fig. 52: Comparison of Variation of Exergy Destruction with evaporator temperature-2

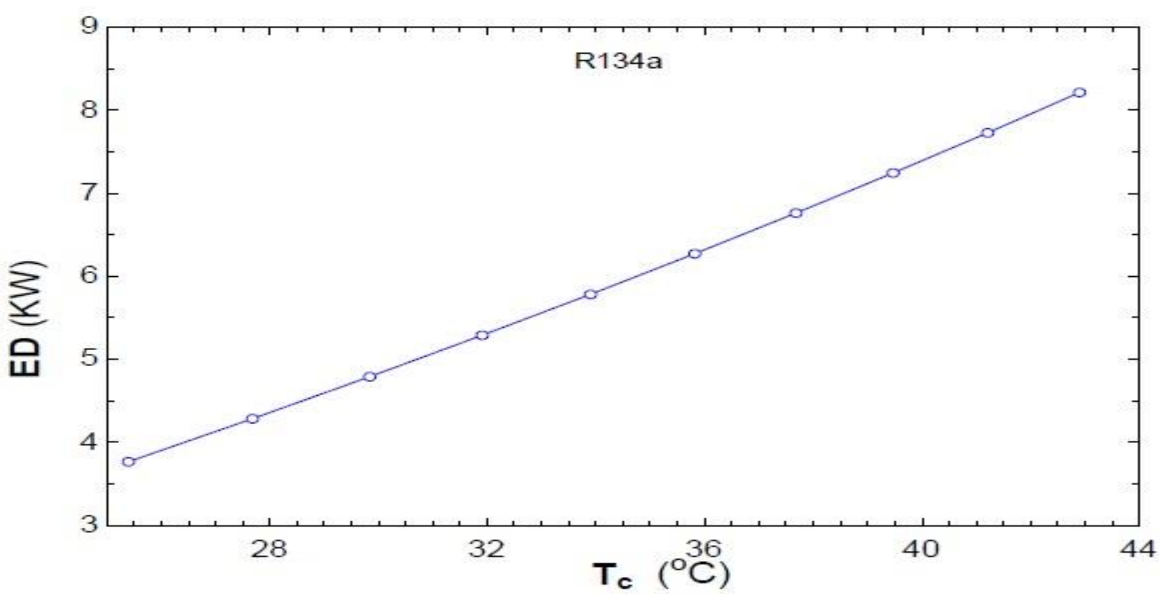

Fig. 53: Variation of Exergy Destruction with condenser temperature using R134a

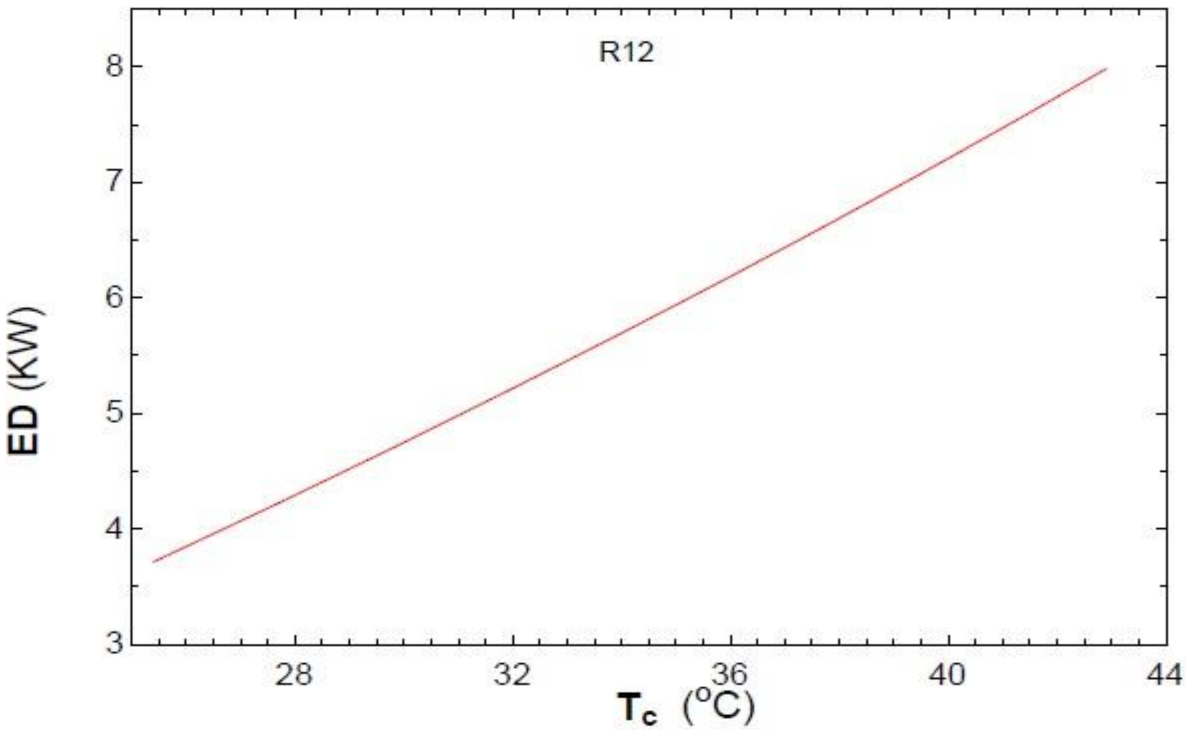

Fig. 54: Variation of Exergy Destruction with condenser temperature using R12

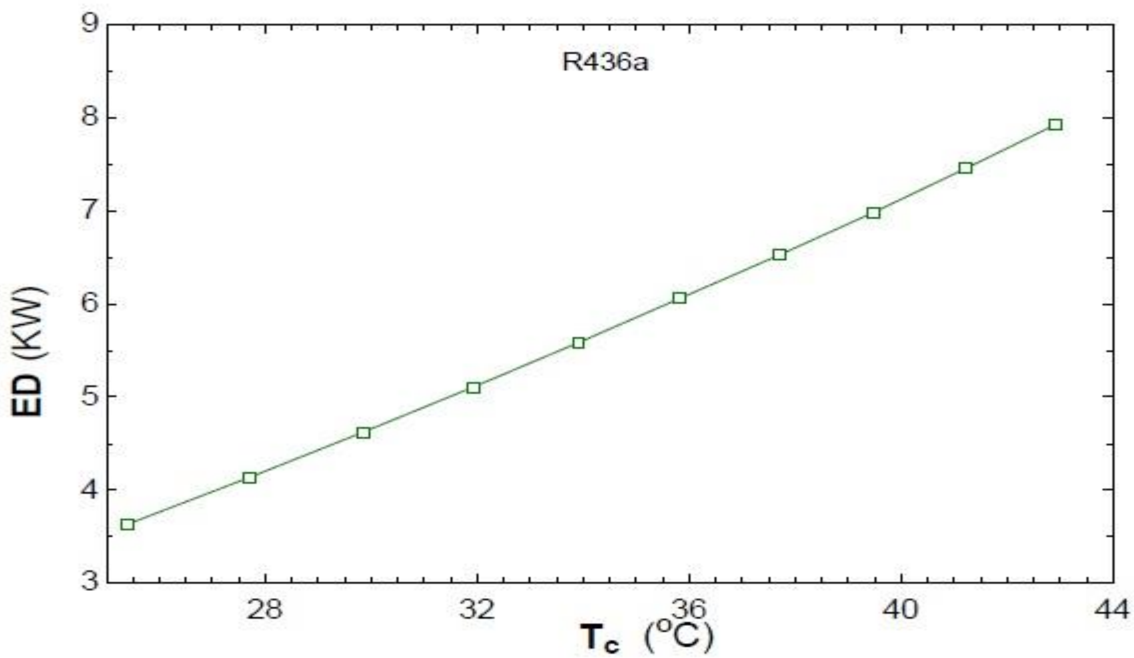

Fig. 55: Variation of Exergy Destruction with condenser temperature using R436a

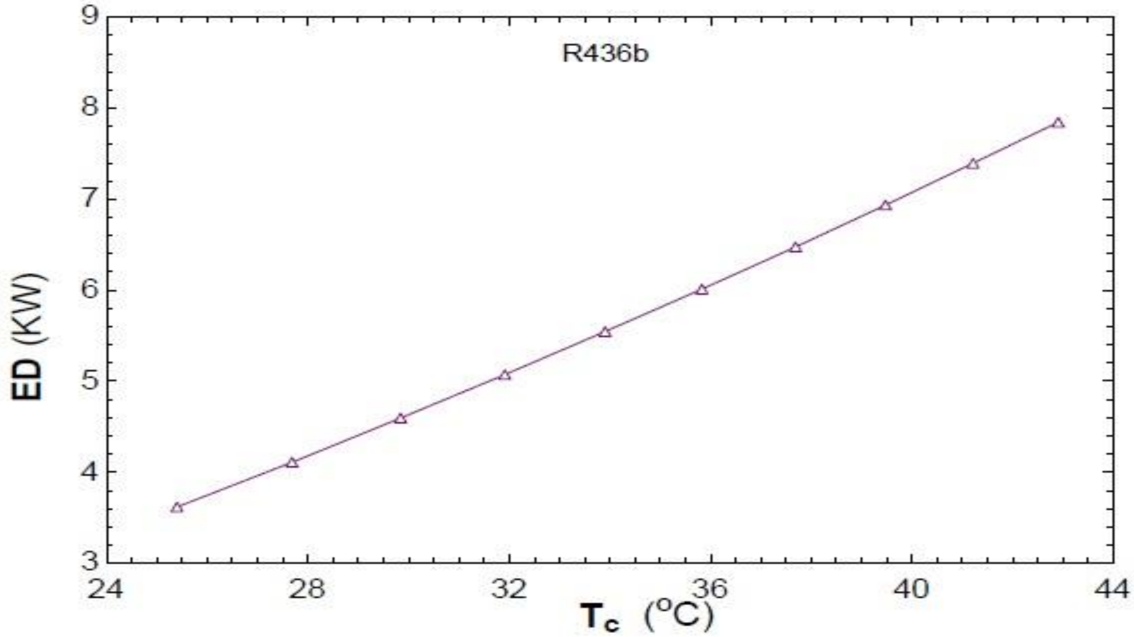

Fig. 56: Variation of Exergy Destruction with condenser temperature using R436b

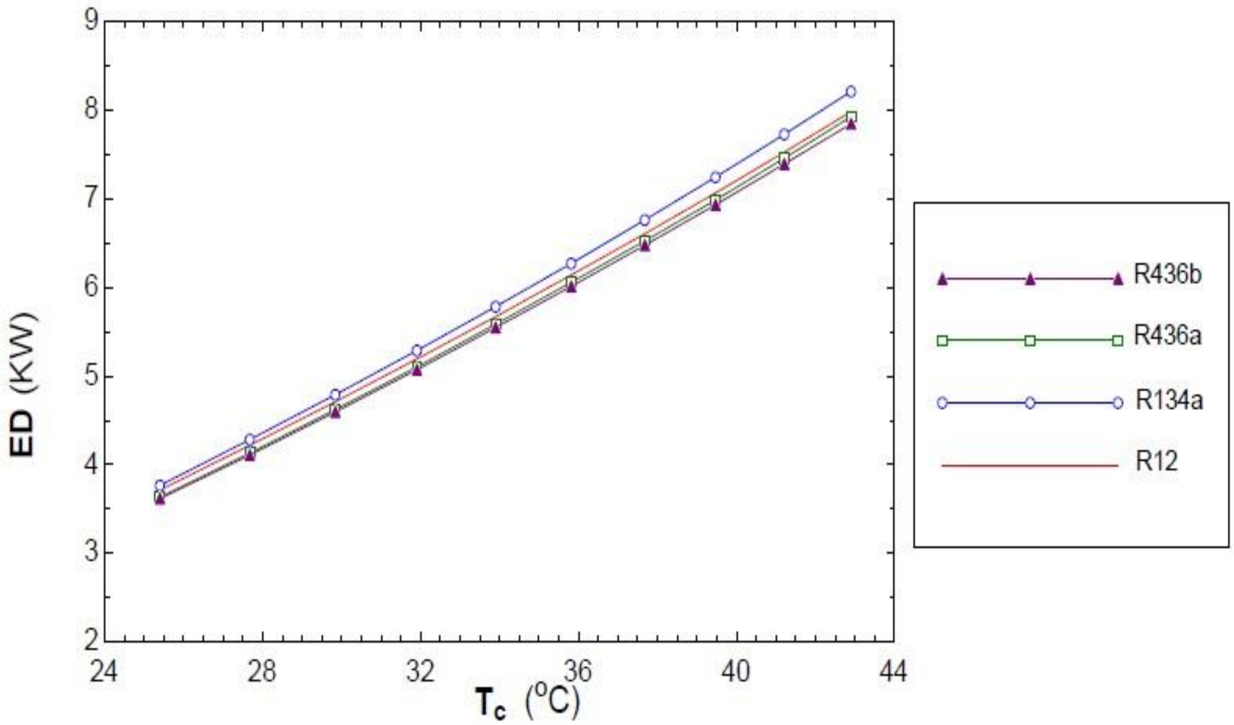

Fig. 57: Comparison of Variation of Exergy Destruction with condenser temperature

4.4 Exergy Destruction Ratio

Fig. 58 to Fig. 72 show the variation of Exergy Destruction using different refrigerants alternatively against evaporator-1, evaporator-2 and condenser temperatures respectively. Fig. 62, 67 and 72 show the comparison of variation of Exergy Destruction for different refrigerants.

Exergy destruction ratio is inversely proportional to exergetic efficiency. A least value of the exergy destruction ratio gives the maximum exergetic efficiency and that system is best. R134a has the more EDR than the other refrigerants. R436b has least EDR which is 4.5% less than EDR of R134a, among all refrigerants. R436a has second least EDR which is 4.0% less than EDR of R134a, compared to all selected refrigerants.

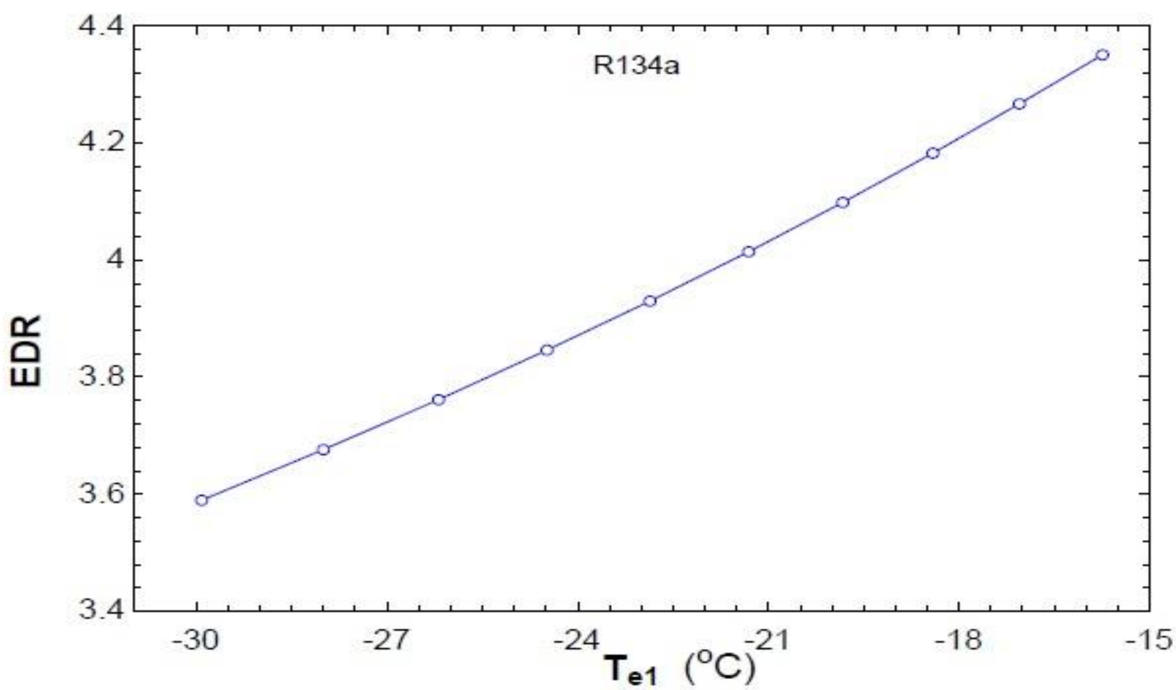

Fig. 58: Variation of EDR with Evaporator Temperature-1 using R134a

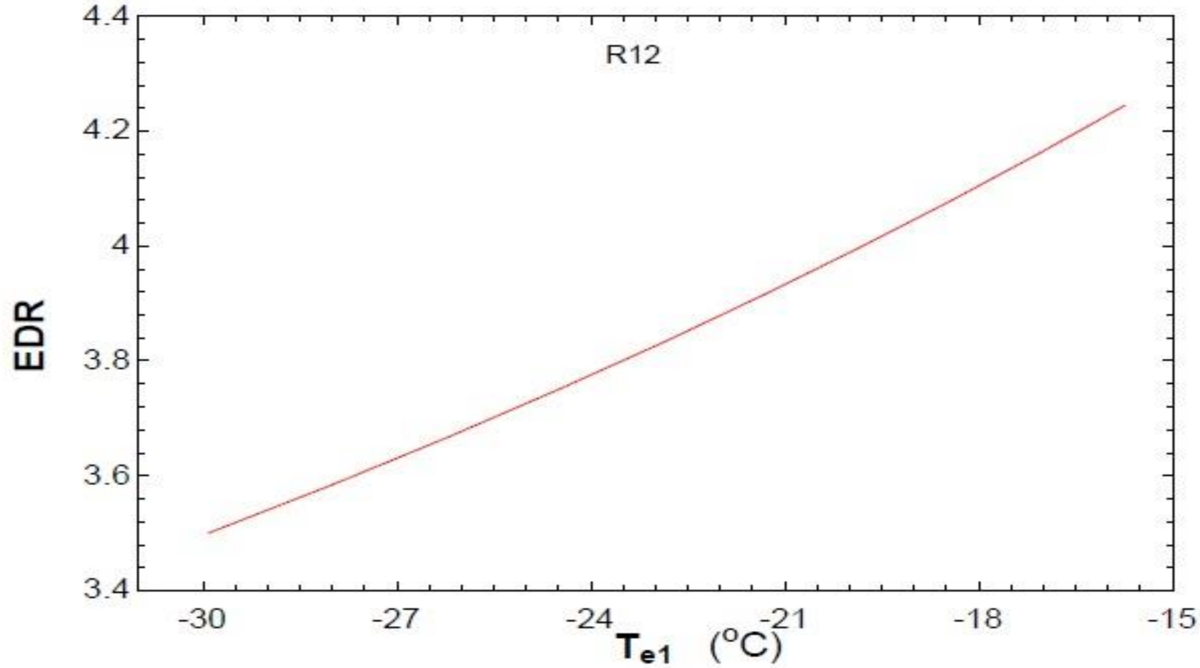

Fig. 59: Variation of EDR with Evaporator Temperature-1 using R12

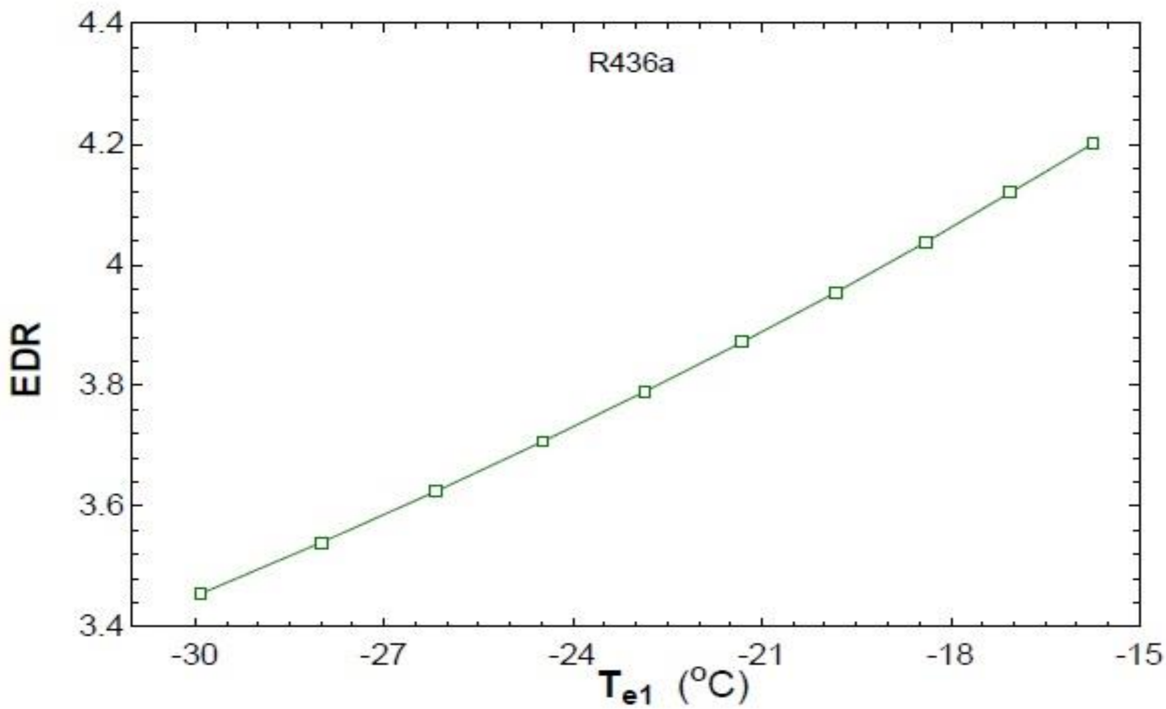

Fig. 60: Variation of EDR with Evaporator Temperature-1 using R436a

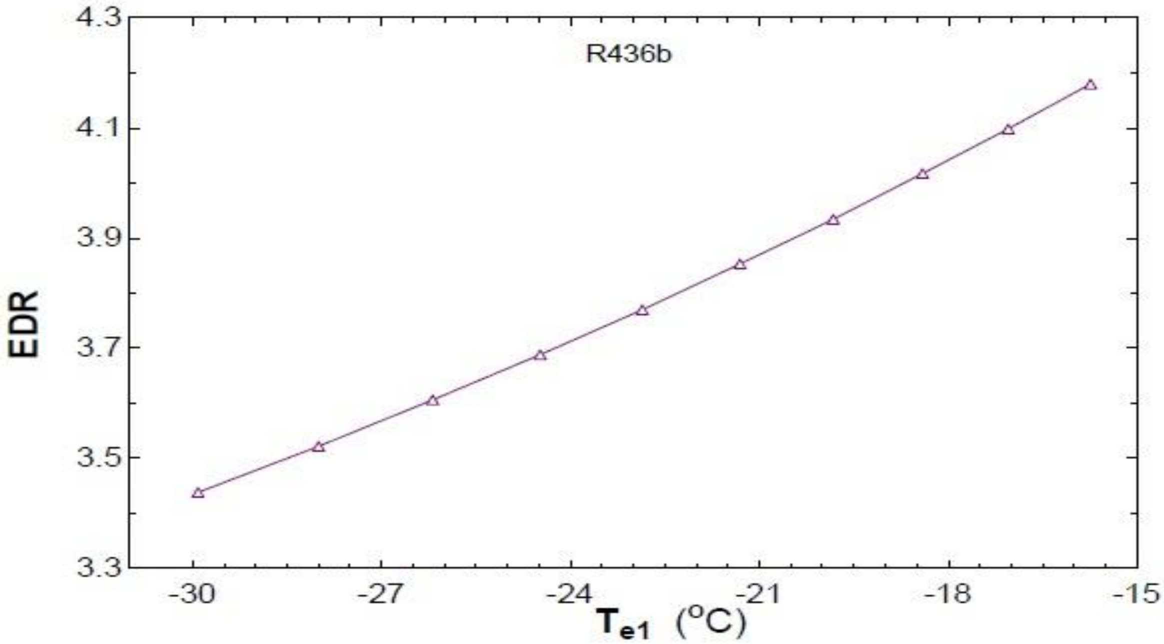

Fig. 61: Variation of EDR with Evaporator Temperature-1 using R436b

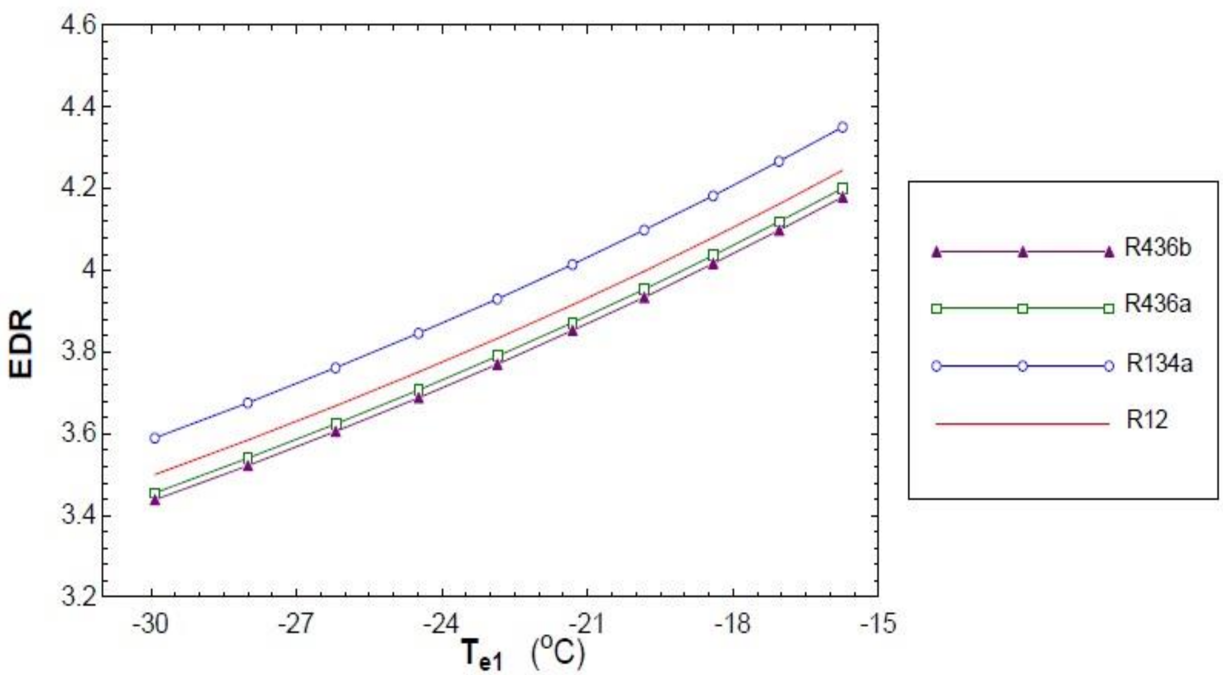

Fig. 62: Comparison of Variation of EDR with Evaporator Temperature-1

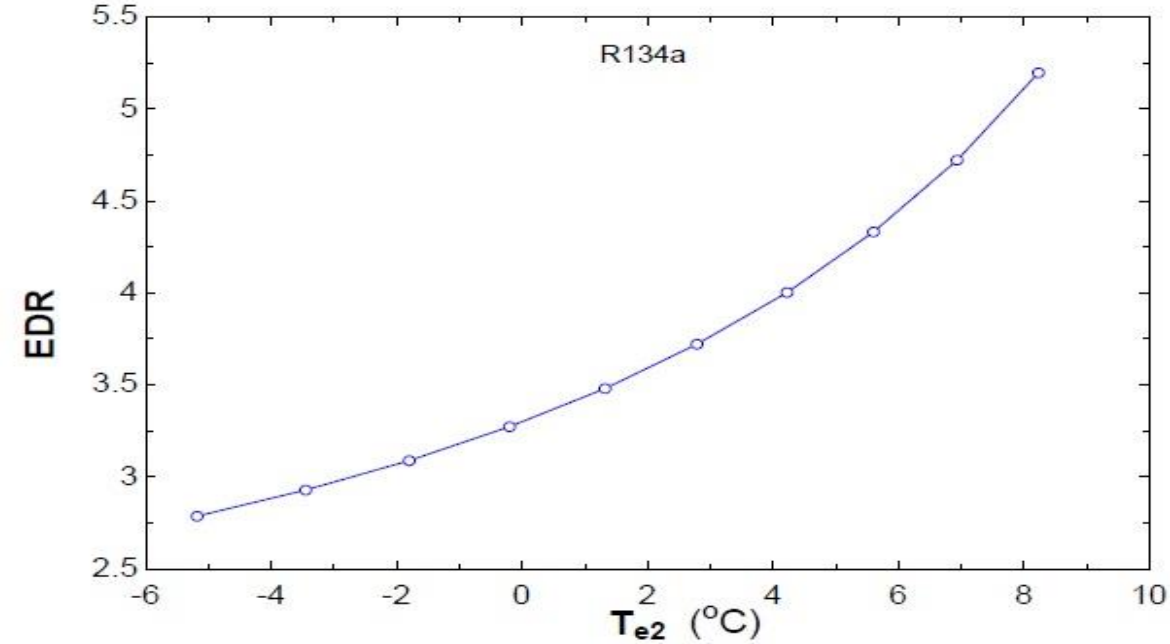

Fig. 63: Variation of EDR with Evaporator Temperature-2 using R134a

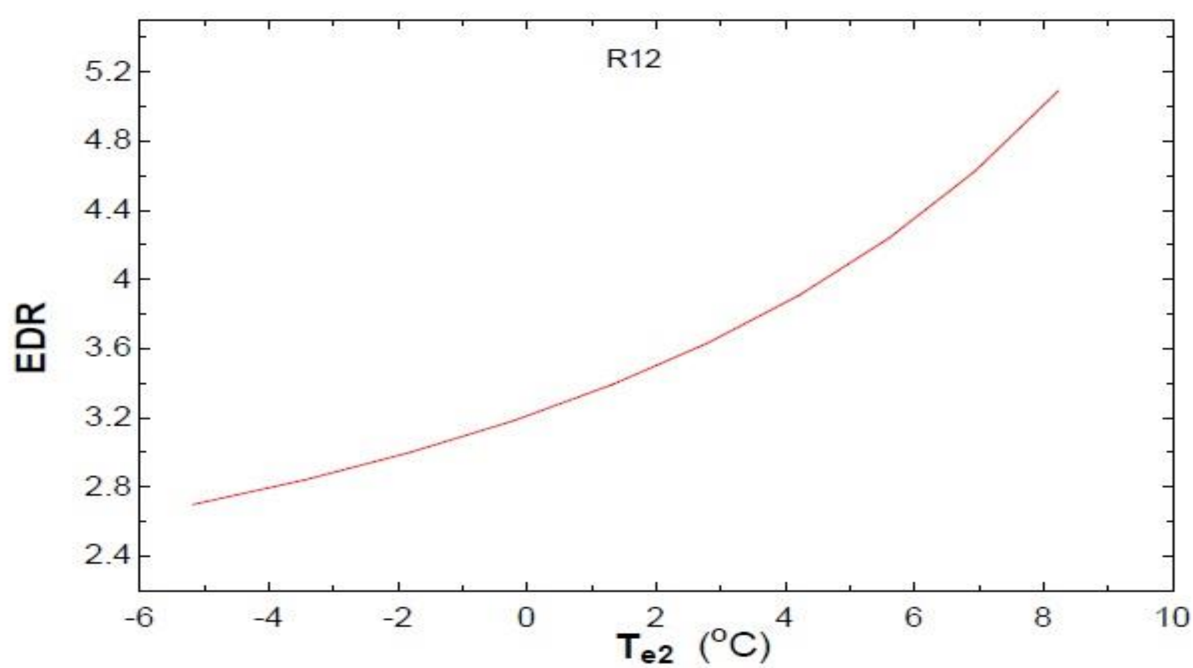

Fig. 64: Variation of EDR with Evaporator Temperature-2 using R12

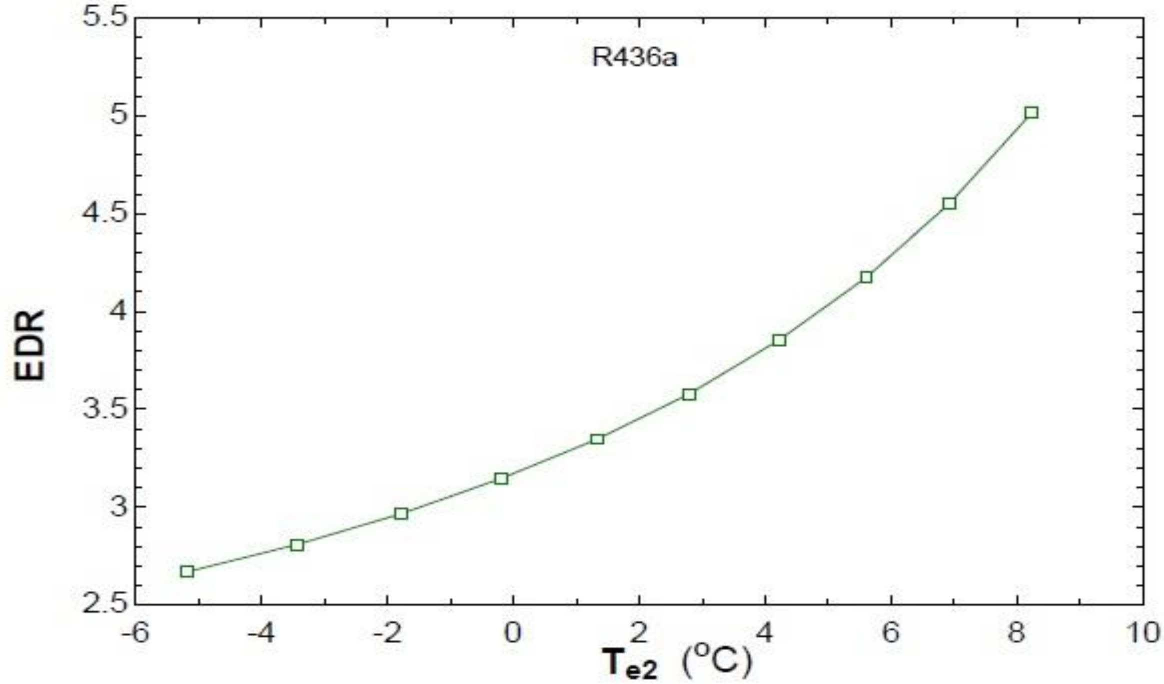

Fig. 65: Variation of EDR with Evaporator Temperature-2 using R436a

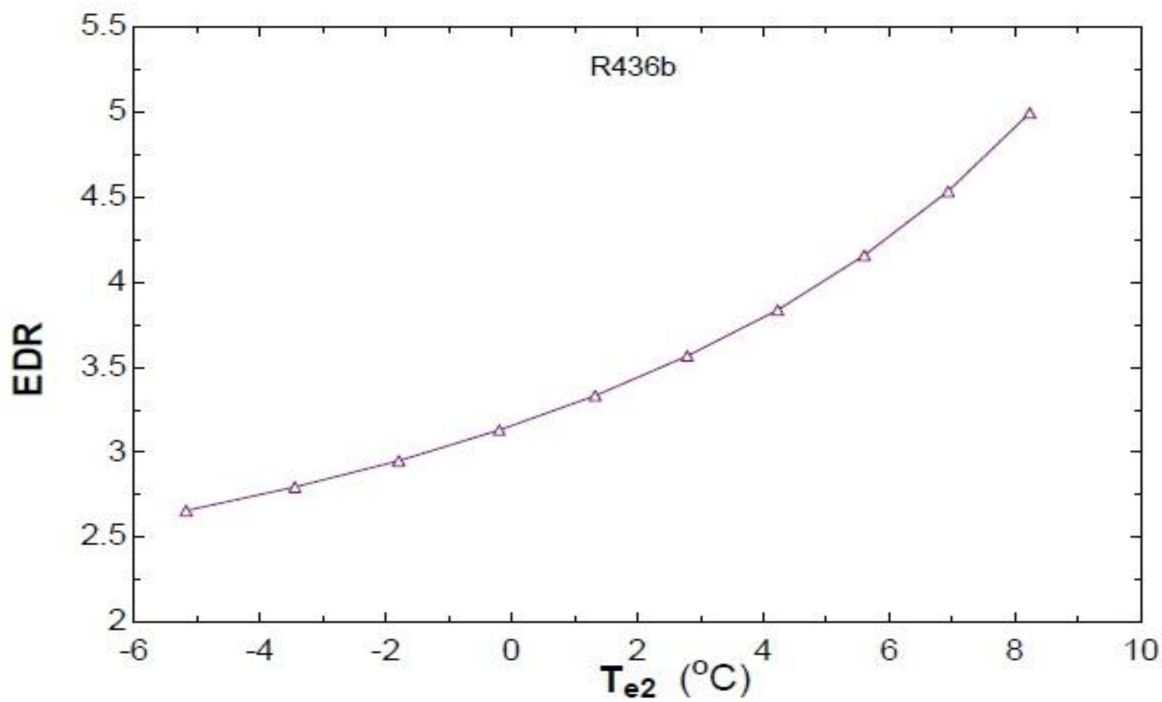

Fig. 66: Variation of EDR with Evaporator Temperature-2 using R436b

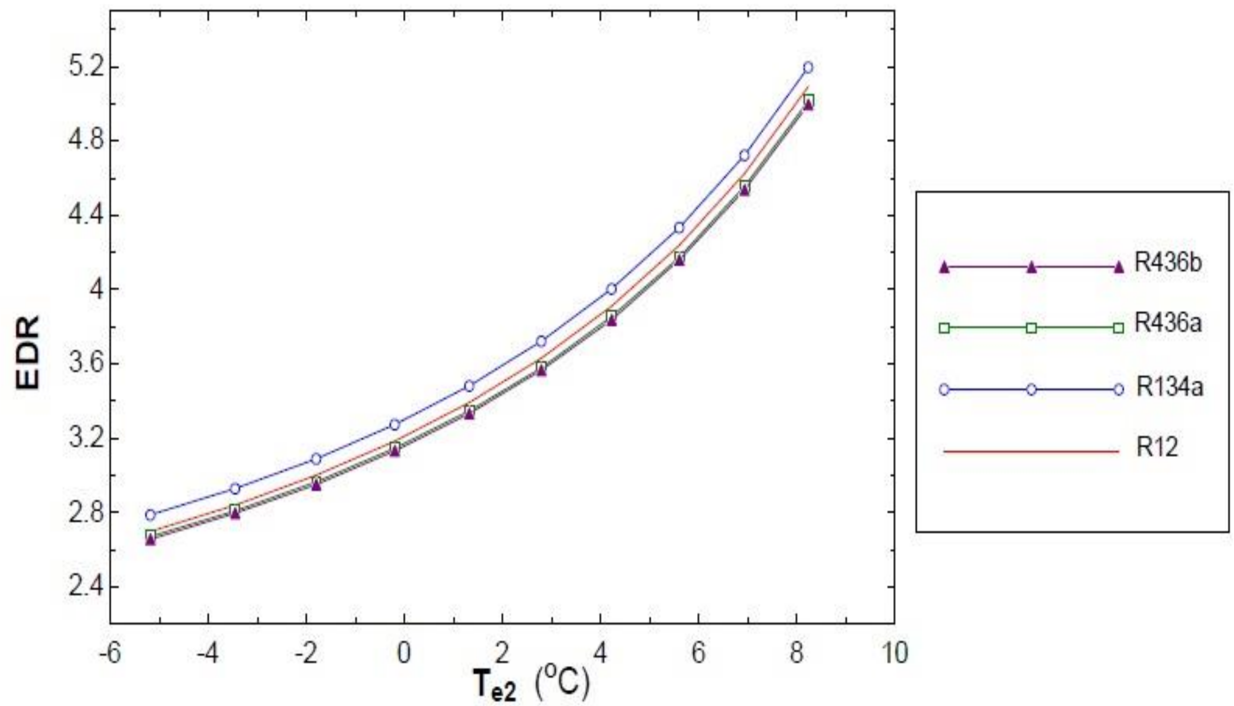

Fig. 67: Comparison of Variation of EDR with Evaporator Temperature-2

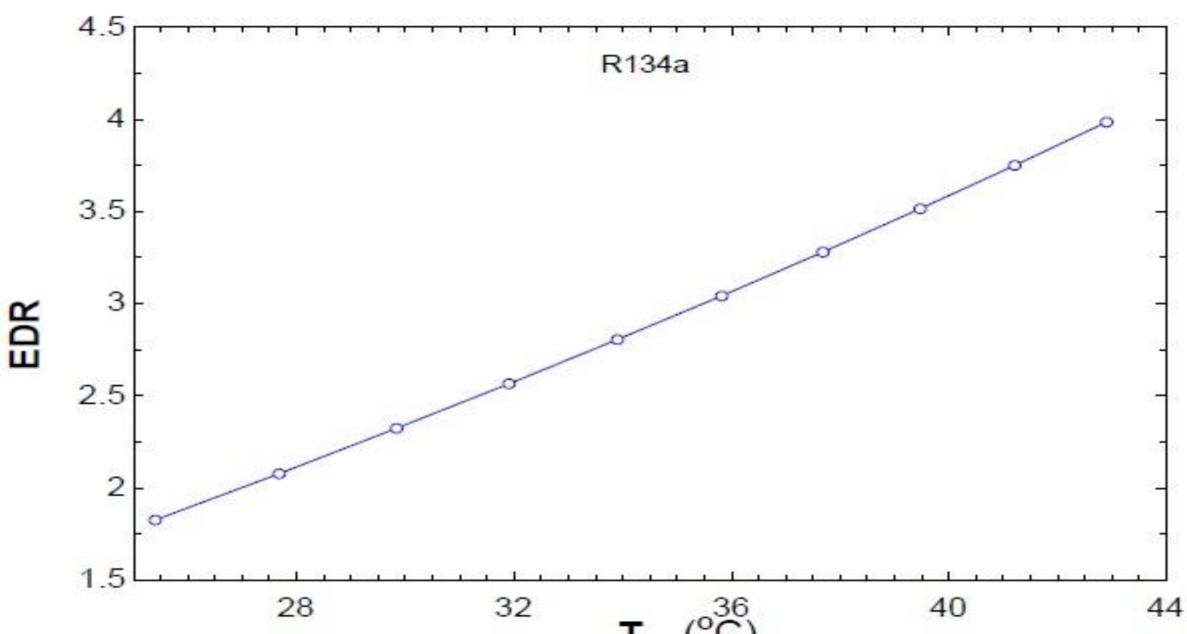

Fig. 68: Variation of EDR with condenser temperature using R134a

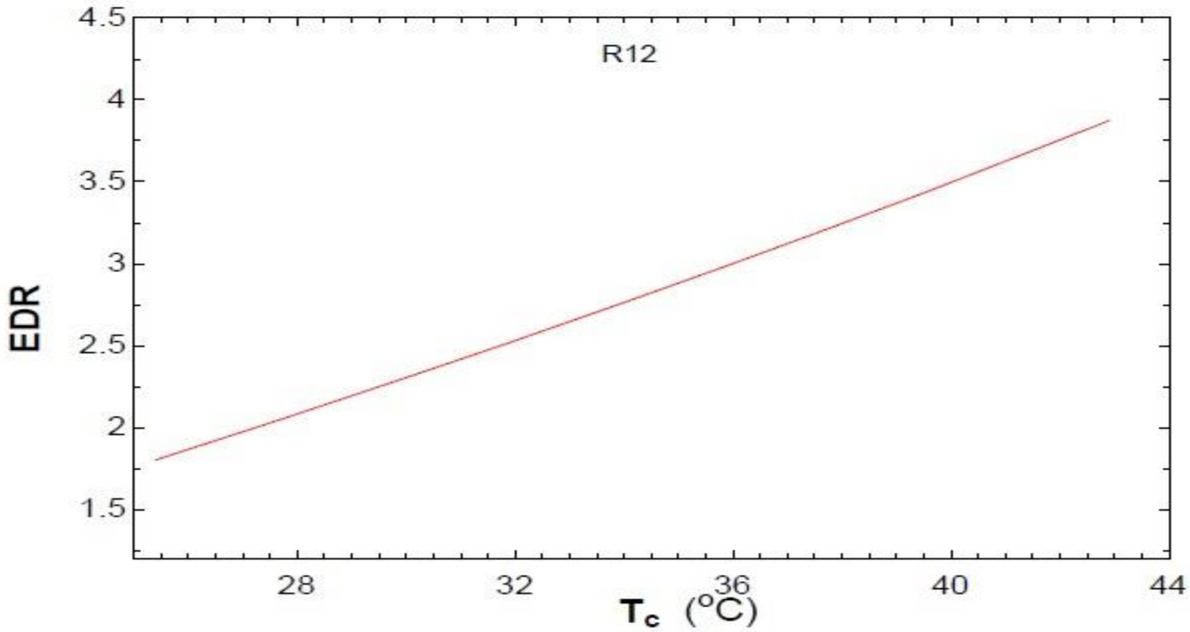

Fig. 69: Variation of EDR with condenser temperature using R12

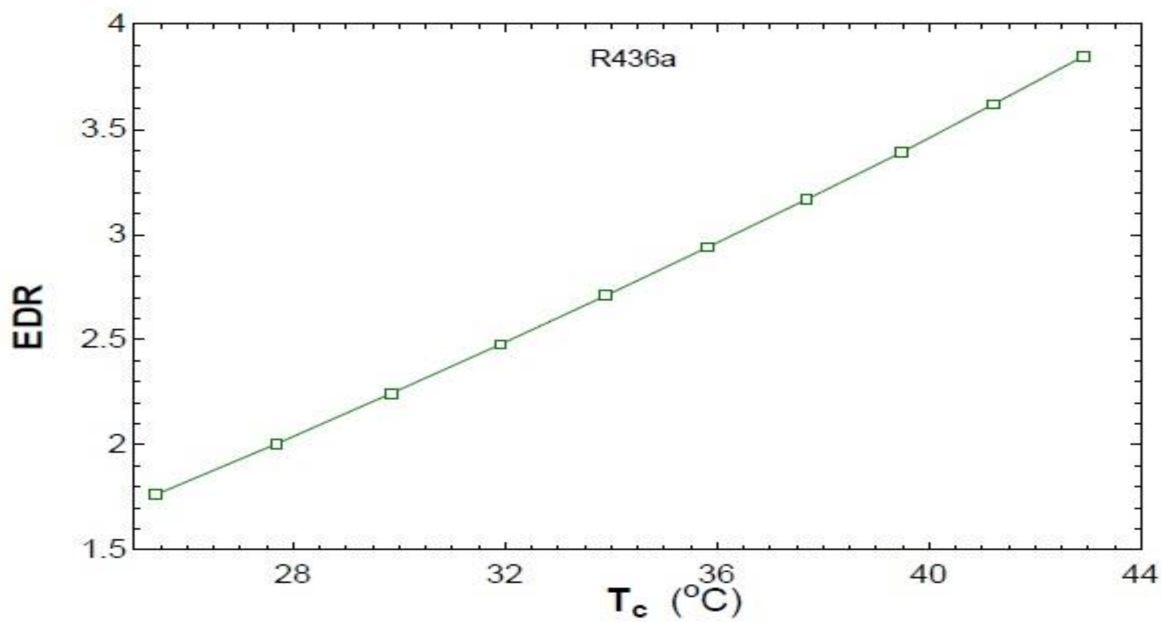

Fig. 70: Variation of EDR with condenser temperature using R436a

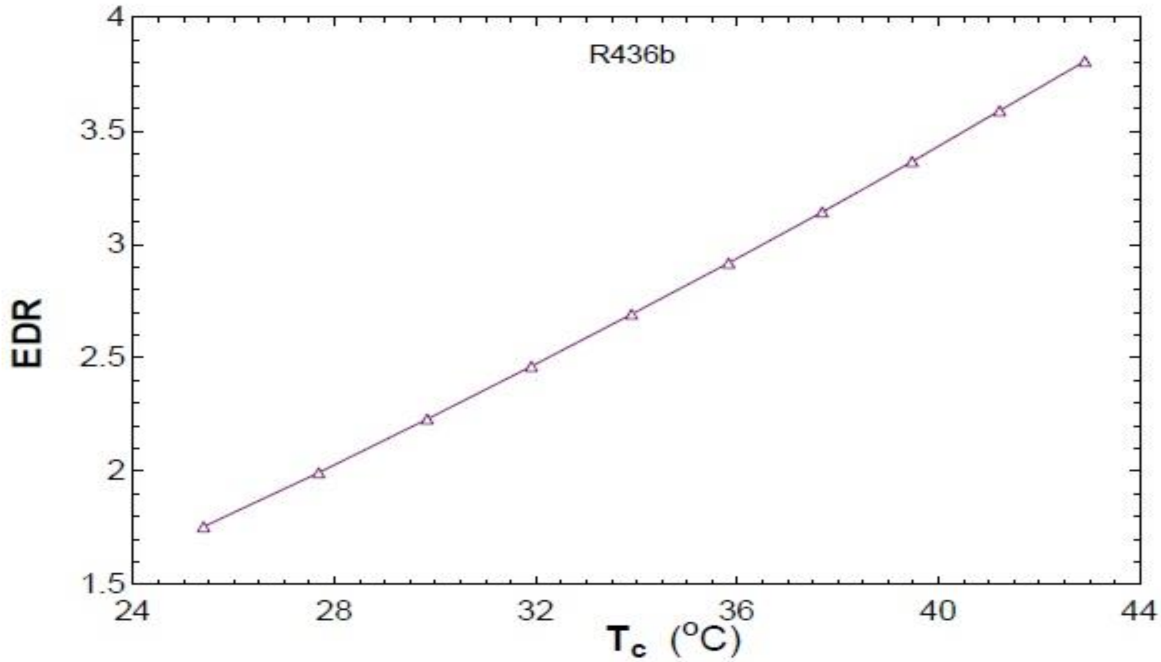

Fig. 71: Variation of EDR with condenser temperature using R436b

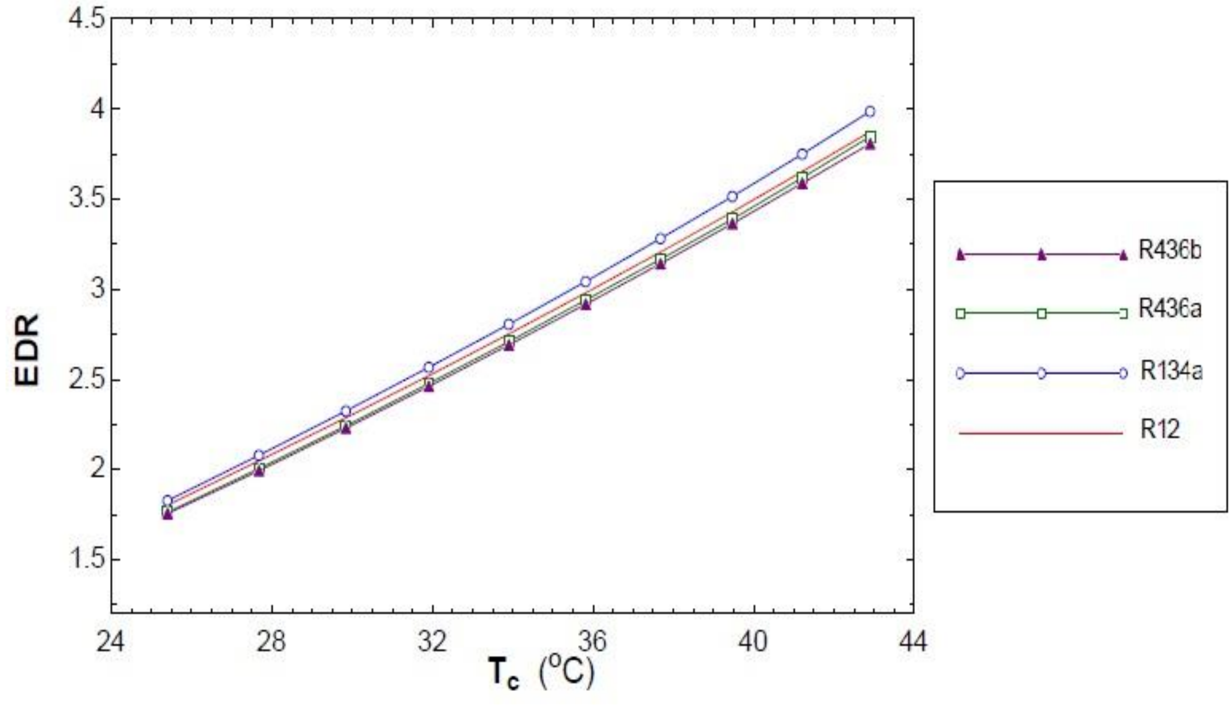

Fig. 72: Comparison of Variation of EDR with condenser temperature

4.5 Exergetic Efficiency

Fig. 73 to Fig. 87 show the variation of exergetic efficiency using different refrigerants alternatively against evaporator-1, evaporator-2 and condenser temperatures respectively. Fig. 77, 52 and 87 show the comparison of variation of exergetic efficiency for different refrigerants.

Among all the refrigerants which are taken, R436b has the maximum exergetic efficiency. R436a has second highest exergetic efficiency among all the refrigerants. R134a has least exergetic efficiency among all the selected refrigerants.

Among all the four refrigerants, R436b has 4.0% more exergetic efficiency than R134 and R436a has 3.6 more exergetic efficiency than exergetic efficiency of R134a.

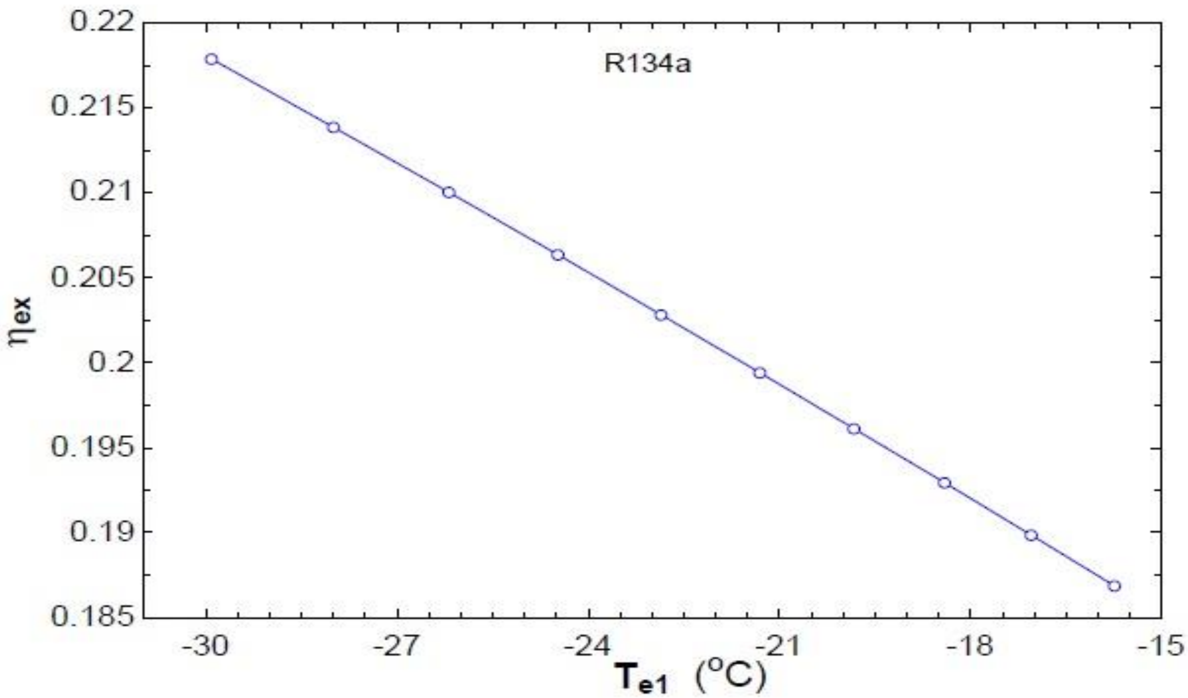

Fig. 73: Variation of Exergetic Efficiency with Evaporator Temperature-1 using R134a

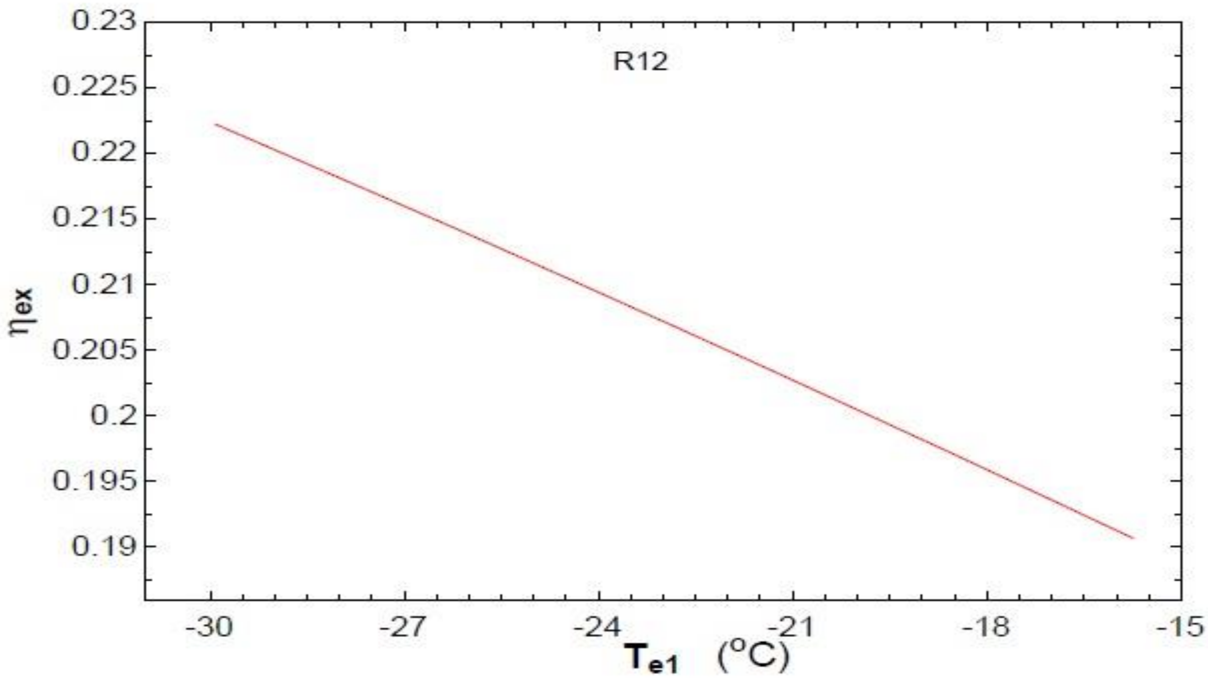

Fig. 74: Variation of Exergetic Efficiency with Evaporator Temperature-1 using R12

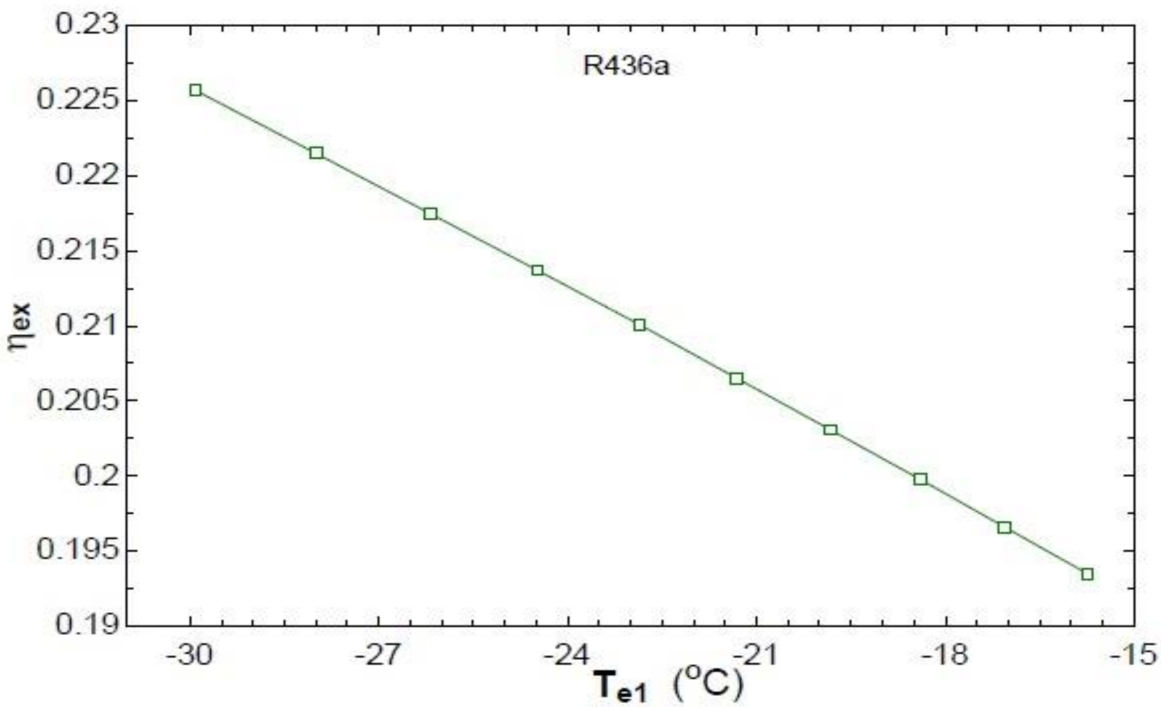

Fig. 75: Variation of Exergetic Efficiency with Evaporator Temperature-1 using R436a

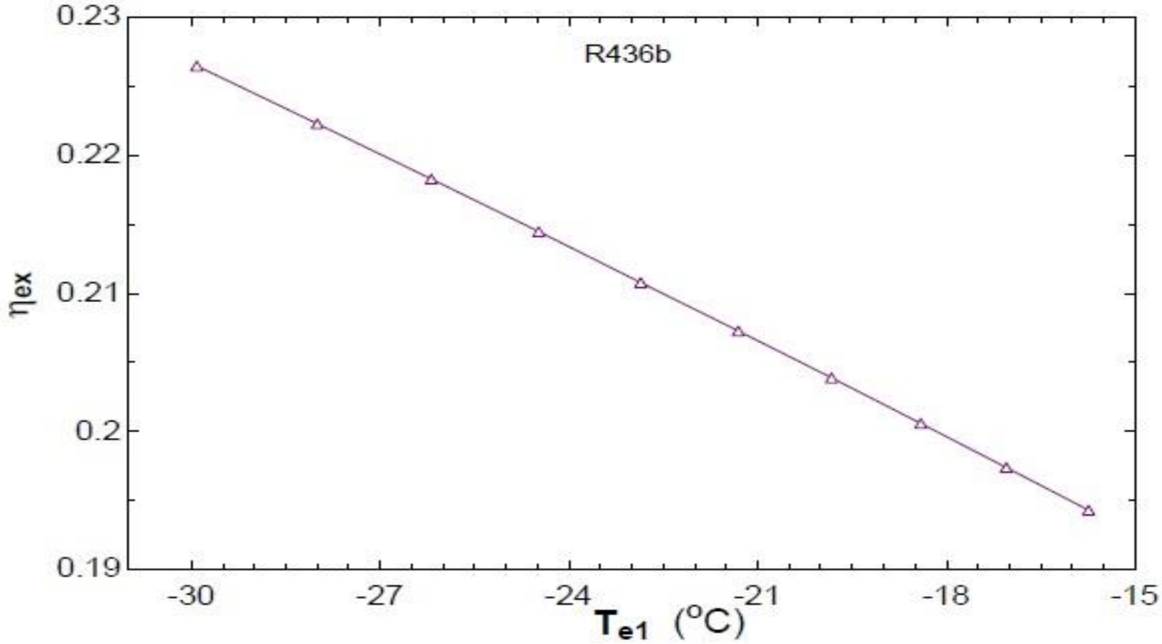

Fig. 76: Variation of Exergetic Efficiency with Evaporator Temperature-1 using R436b

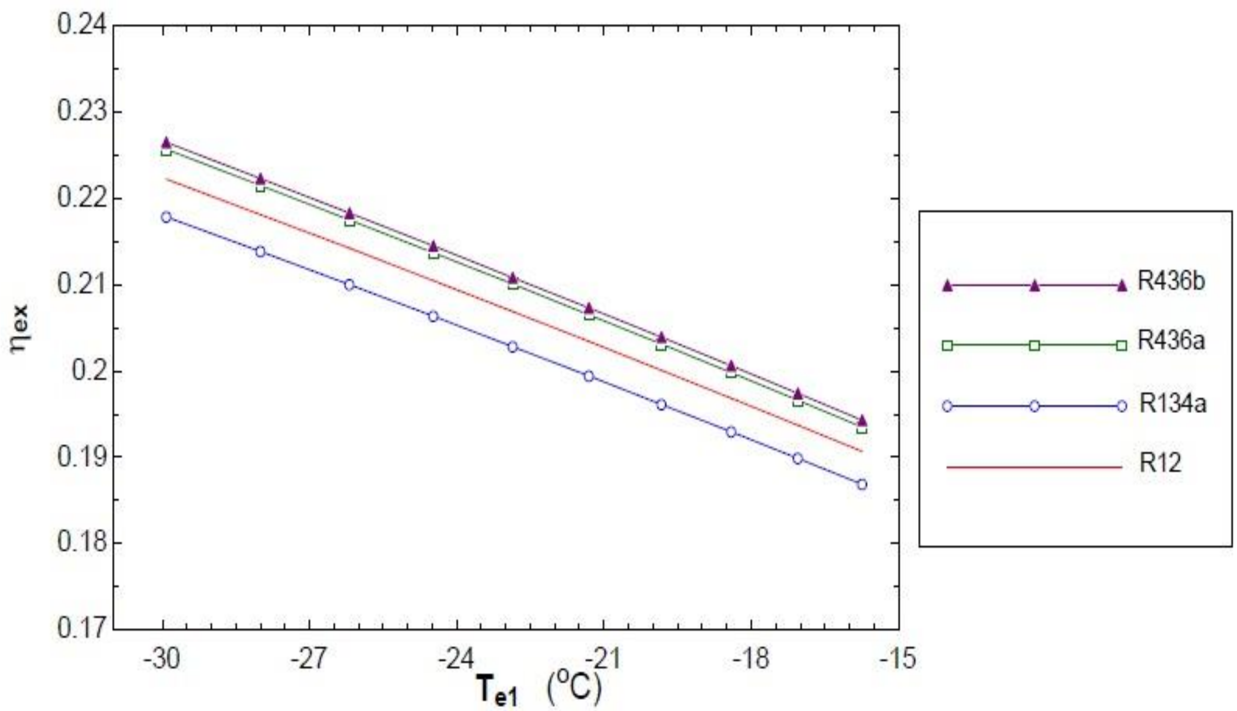

Fig. 77: Comparison of Variation of Exergetic Efficiency with Evaporator Temperature-1

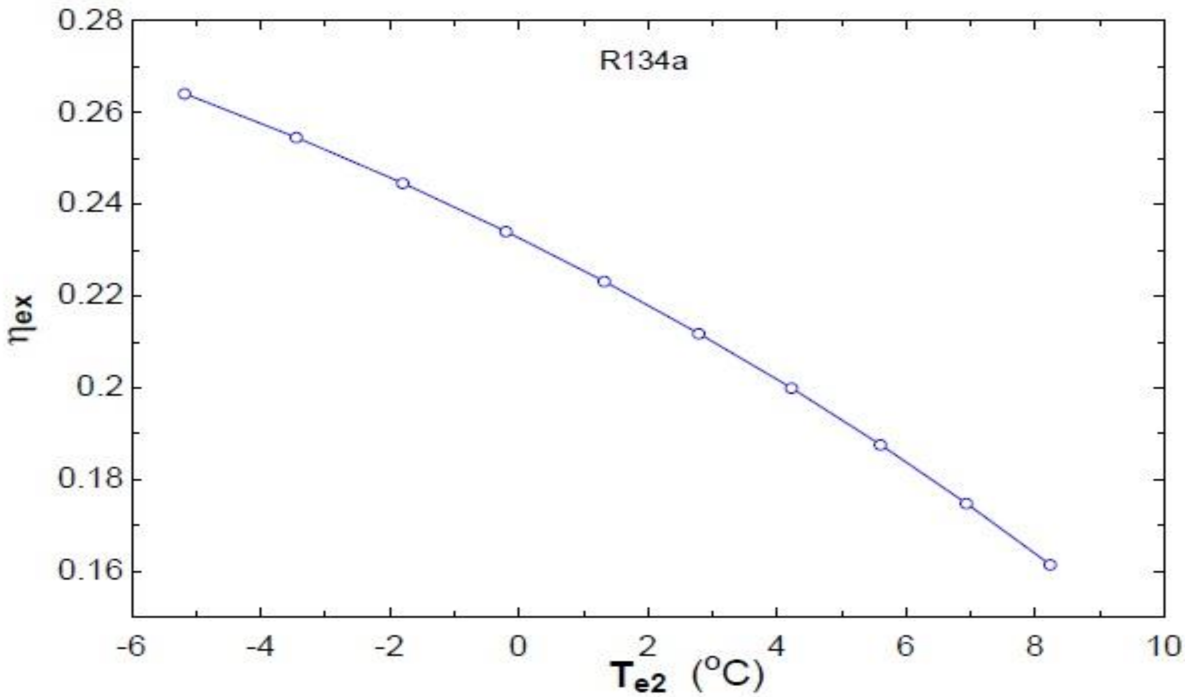

Fig. 78: Variation of Exergetic Efficiency with Evaporator Temperature-2 using R134a

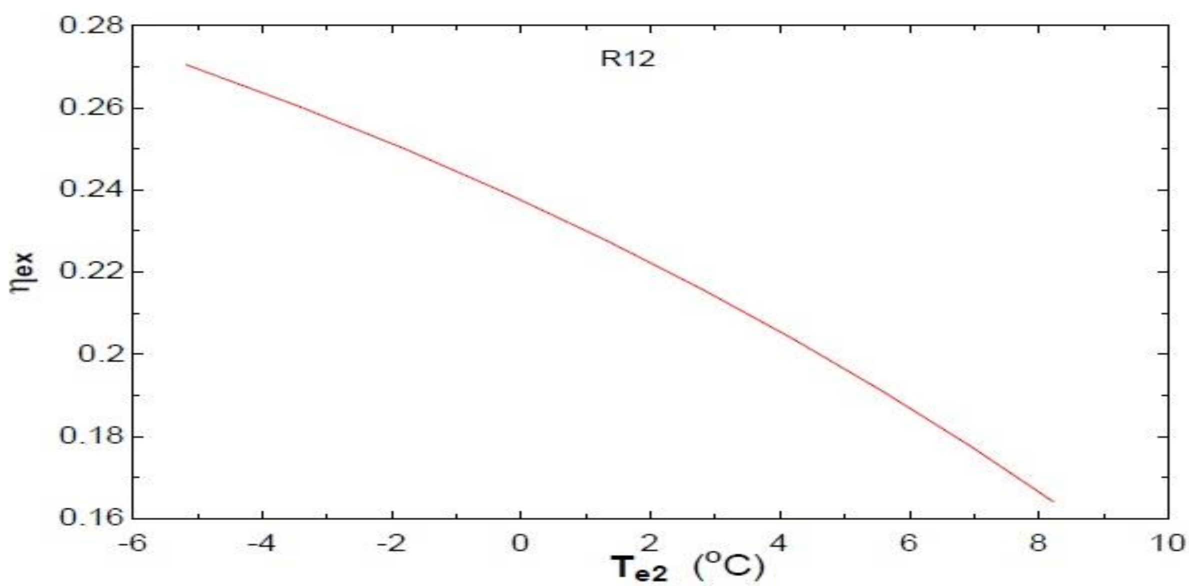

Fig. 79: Variation of Exergetic Efficiency with Evaporator Temperature-2 using R12

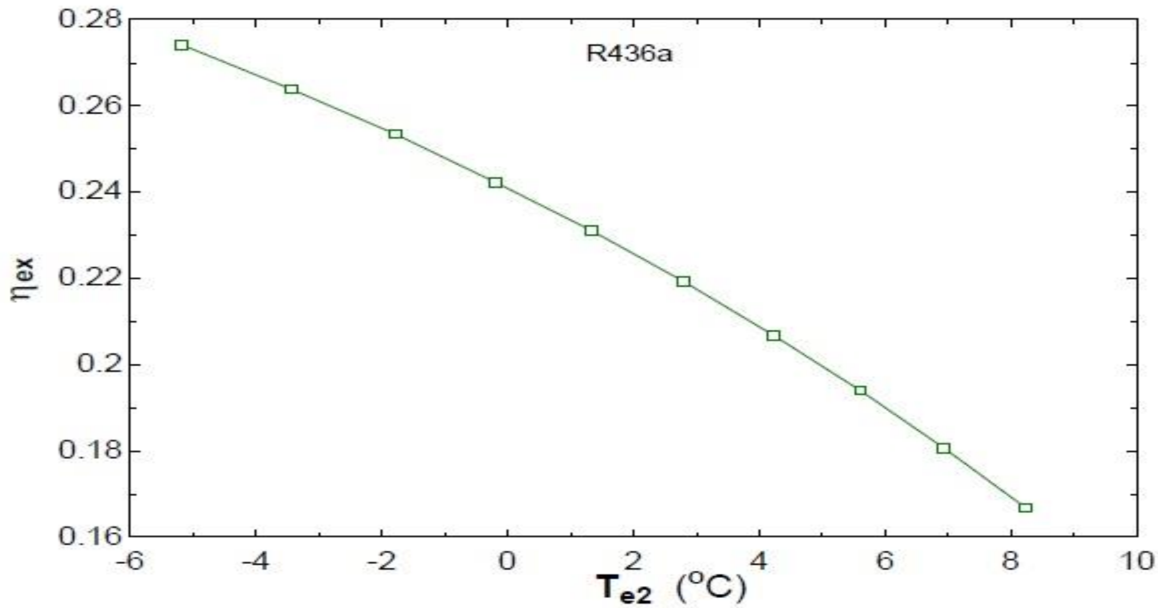

Fig. 80: Variation of Exergetic Efficiency with Evaporator Temperature-2 using R436a

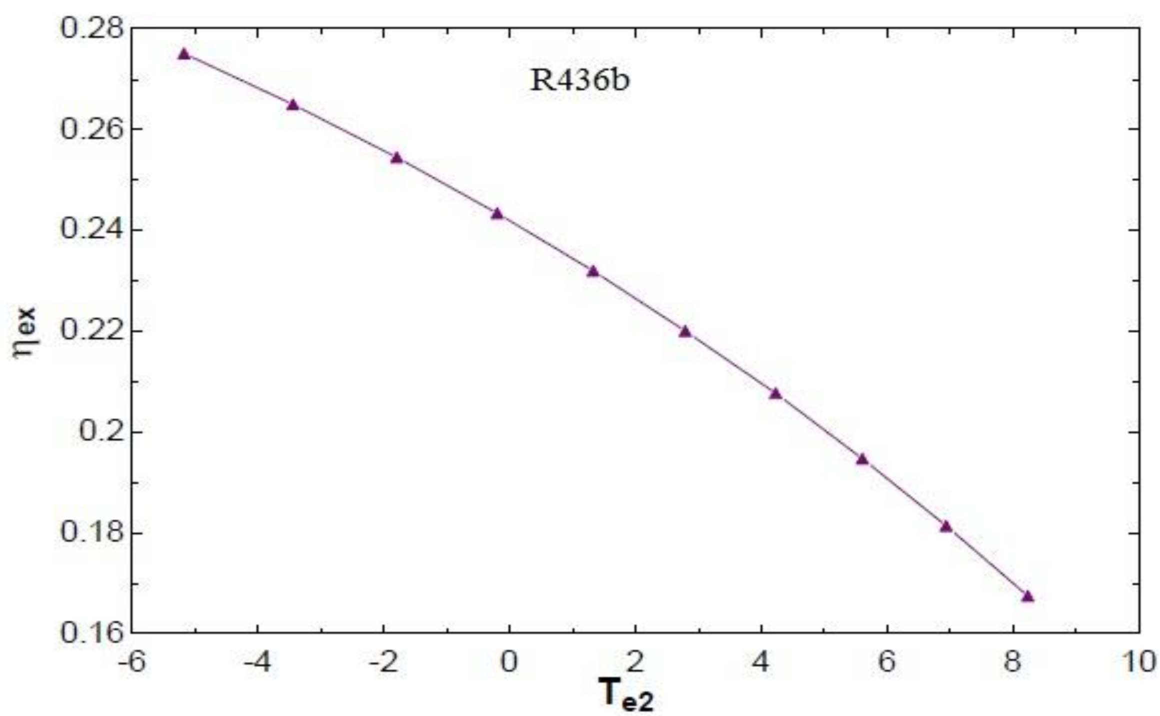

Fig. 81: Variation of Exergetic Efficiency with Evaporator Temperature-2 using R436b

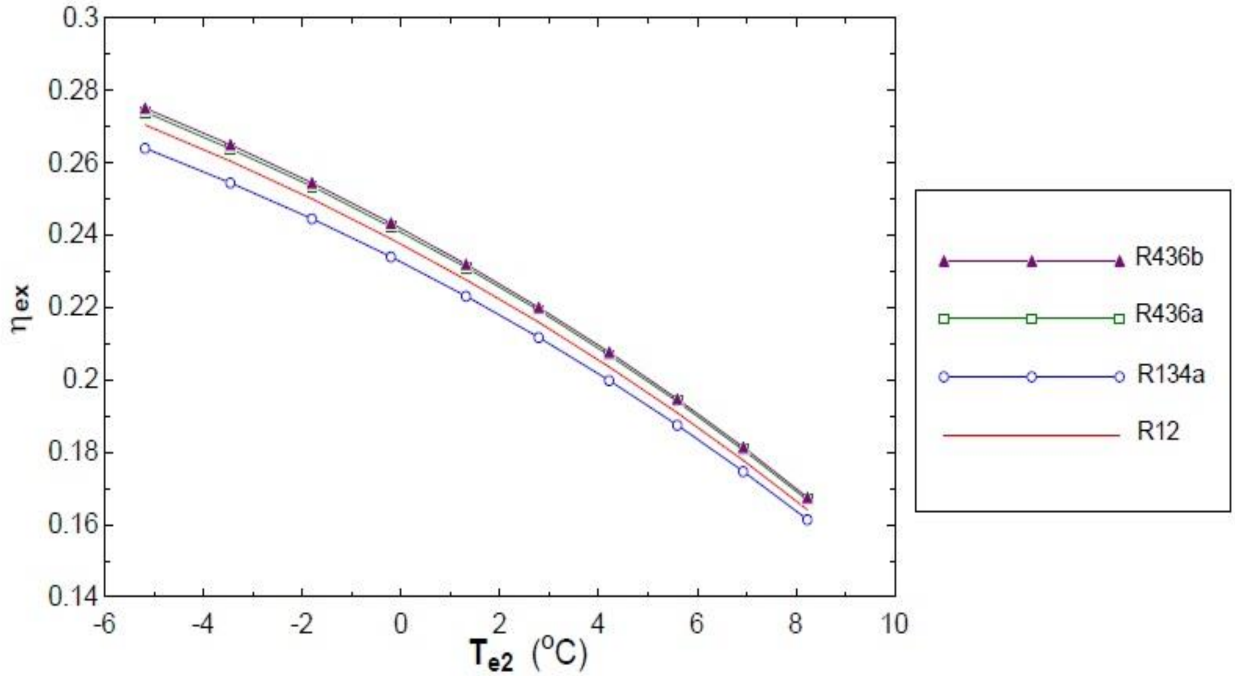

Fig. 82: Comparison of Variation of Exergetic Efficiency with Evaporator Temperature-2

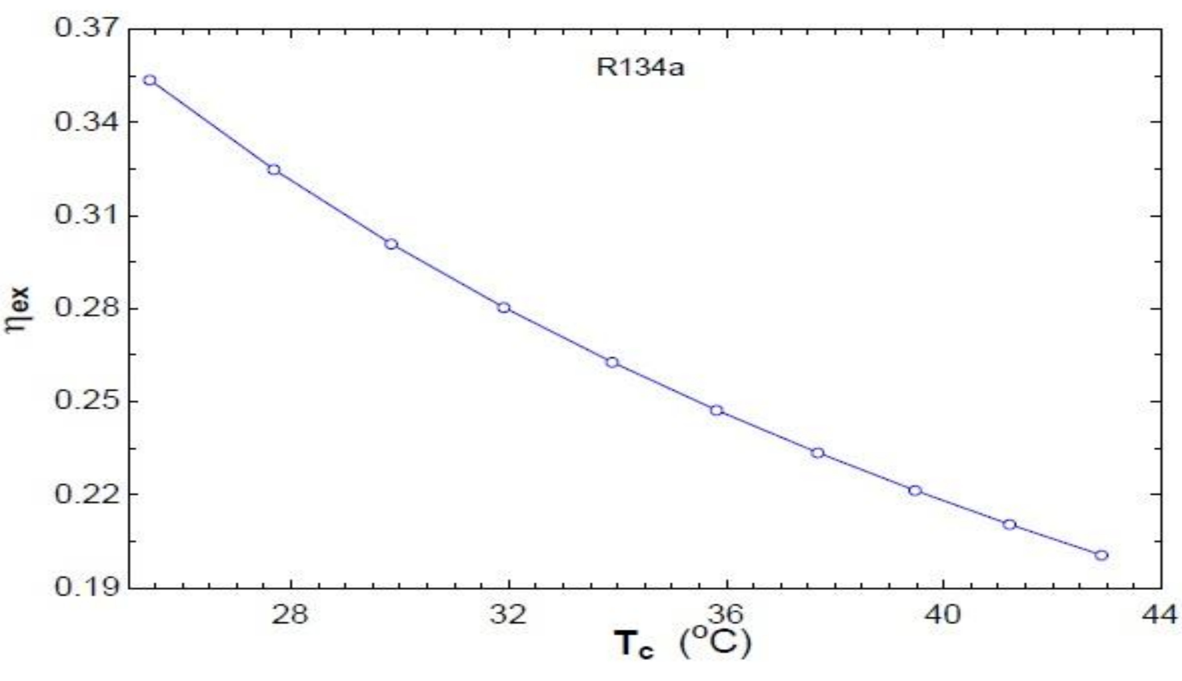

Fig. 83: Variation of Exergetic Efficiency with Condenser Temperature using R134a

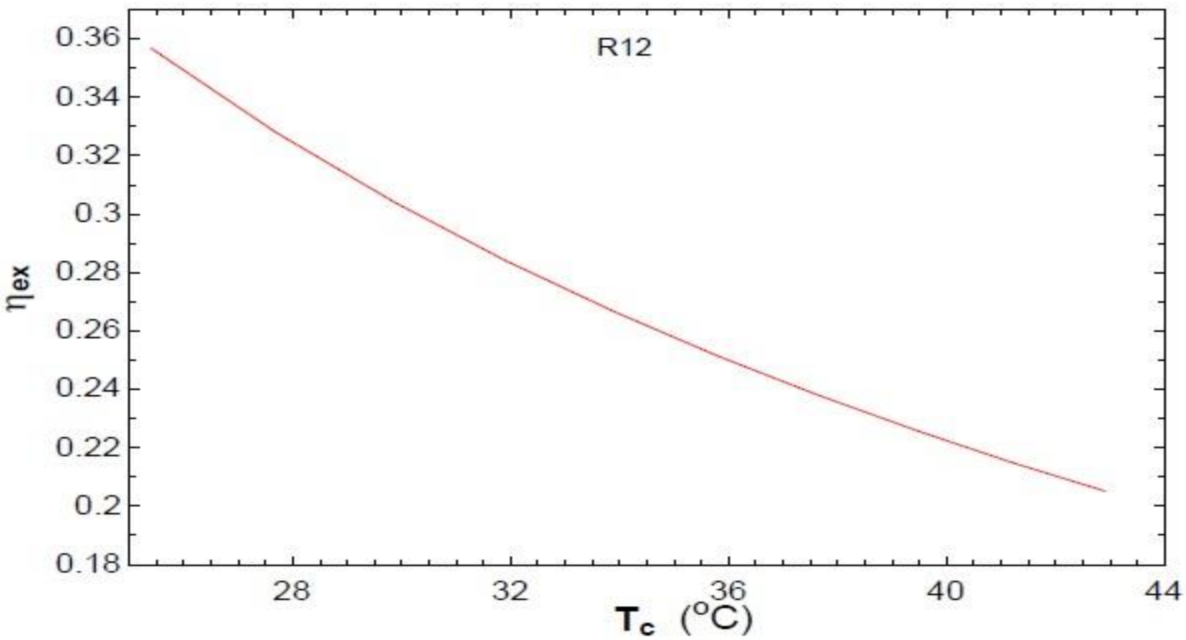

Fig. 84: Variation of Exergetic Efficiency with Condenser Temperature using R12

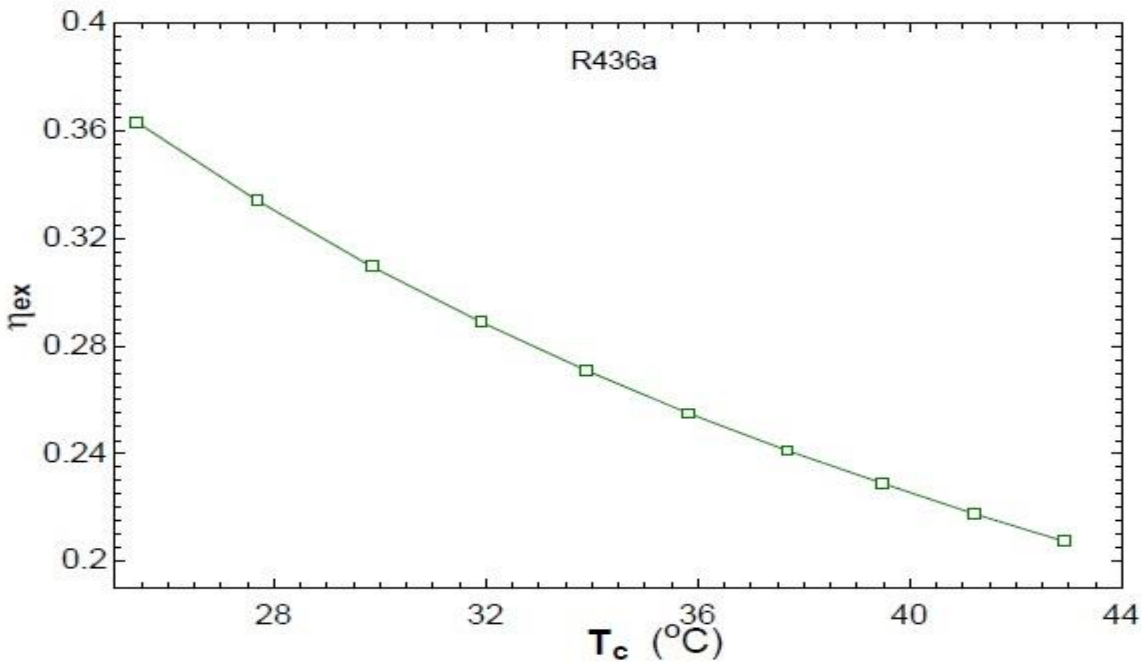

Fig. 85: Variation of Exergetic Efficiency with Condenser Temperature using R436a

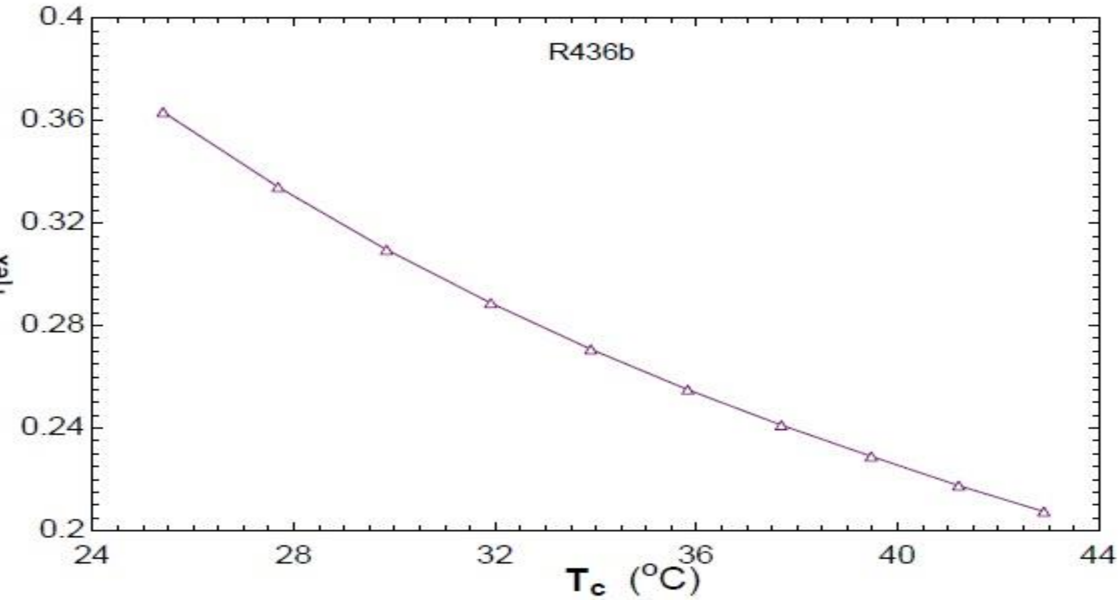

Fig. 86: Variation of Exergetic Efficiency with Condenser Temperature using R436b

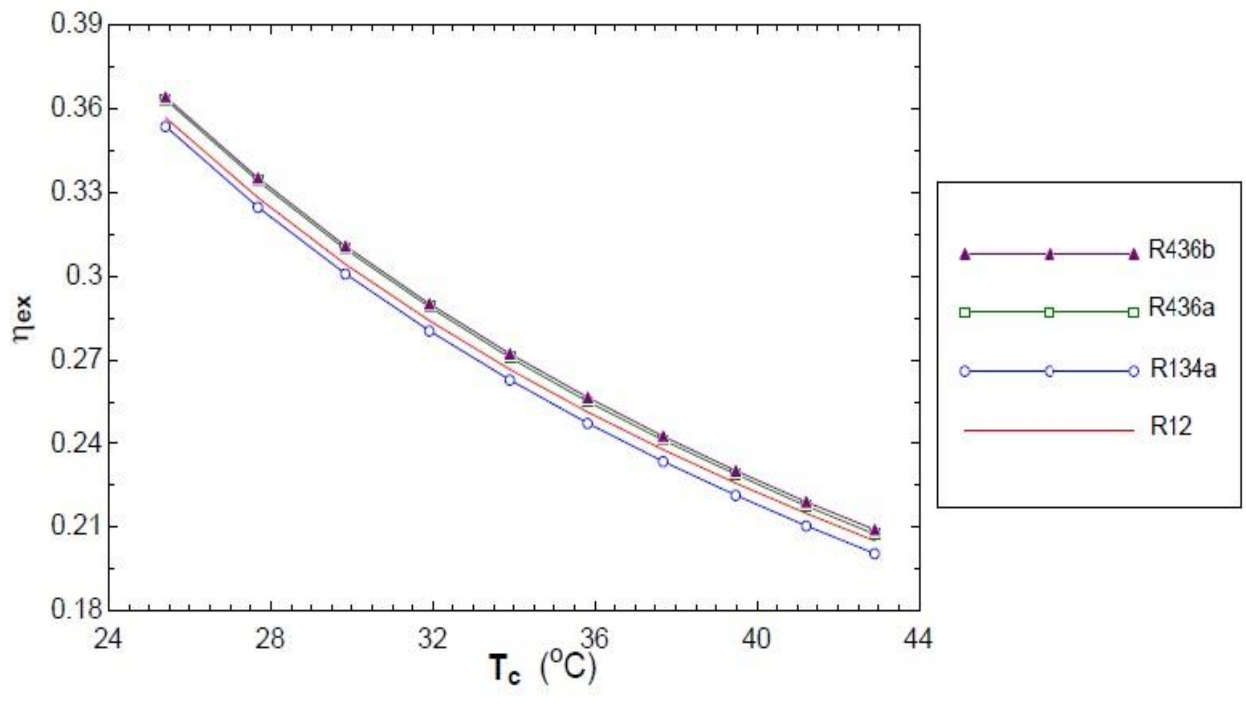

Fig. 87: Comparison of Variation of Exergetic Efficiency with Condenser Temperature

CHAPTER 5

CONCLUSION

In this thesis, thermodynamic analysis is done which is based on first law (energy) and second law (exergy) analysis of R12, R134a, R436a(0.52R290 / 0.48R600a) and R436b(0.56R290 / 0.44R600a) refrigerants in multi-evaporator vapor compression cycle with individual compressor using flash chamber is presented. Conclusions of this analysis are summarized as follows:

1. R436b refrigerant has the least total work input for the given system and R436a refrigerant has the second least total work input for the given system but R134a refrigerant has the maximum total work input.
2. R436b refrigerant has the highest COP and exergetic efficiency for the given system but R134a has least COP and exergetic efficiency which is used as a working fluid (refrigerant) in the household refrigerator and food freezing industries.
3. R134a refrigerant has higher EDR among all selected refrigerants but R436b has least EDR for the system.
4. EDR is inversely proportional to the exergetic efficiency. The system which has the least value of EDR gives the highest exergetic efficiency.
5. Among all the refrigerants which are taken, R436b has the least ED and has maximum exergetic Efficiency but R134a has the maximum ED and has least exergetic Efficiency.

R436b can be taken as better option in multi-evaporator refrigeration system with individual compressor and multiple expansion devices using flash chamber. R436b has least Exergy Destruction for the whole system among all the refrigerants which are taken. R436b refrigerant has highest COP as well as highest exergetic efficiency in this system and also it has zero GWP, so it is recommended to use as a working fluid (refrigerant) in the household refrigerator with multiple cabin and food freezing plant or milk chilling plant.

CHAPTER 6

SCOPE FOR FUTURE WORK

1. Further analysis should be done for other best alternative refrigerants which should be eco-friendly in this system.
2. Further investigation should be taken for using the blend of R290 and R600a (due to its flammable property, it cannot be used in air conditioning) as a working fluid (refrigerant) in air conditioning application.
3. Actual Vapor Compression Refrigeration Cycle should be studied for this system by considering the pressure drops in evaporators and condenser and also in the suction and discharge pipe.
4. Superheating in evaporator and sub cooling in condenser should also be considered in multi evaporator refrigeration system.

CHAPTER 6

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