ABSTRACT

One of the most important and much needed concerns shown by the researchers world over is related to energy. And there are several aspects related to energy on which researches are being focused e.g. optimal use of energy, new or renewable sources of energy and reuse or recovery of waste energy etc. The main motive of this work is to recover and use waste heat from a energy consuming system. The present study focuses on a simple thermodynamic analysis, heat transfer analysis, and exergy analysis of modified two stage vapour compression refrigeration system equipped with auxiliary condenser with various operating variables using R-717, R-134a, R-507a, R-152a, and R-1234yf as the refrigerants both ultra low GWP and zero ODP refrigerants in modified two stage vapour compression system. In the present work emphasis is on waste heat recovery from auxiliary condenser using water as a cooling fluid of a vapour compression refrigeration system utilized in air conditioning system or ice plants. The modified vapour compression system consists of a waste heat recovery device called auxiliary condenser.

A mathematical computational model has been developed for calculating coefficient of performance (COP), exergetic efficiency, exergy destruction and efficiency defects to evaluate the performance of integrated vapour compression system. Throughout the process, condenser temperature is kept at 313K and 323K. Evaporator temperature is kept in the range from 233K to 255K. Results obtained for selected condenser temperature are compared and discussed elaborately. Among the system components, evaporator shows highest efficiency defect value and compresser shows the lowest. Thus, the effects of the pressure drop and fouling in the evaporator, condenser, auxiliary condenser is neglected. Results indicate that there is an increase in overall coefficient of performance and cooling capacity using an auxiliary condenser in two stage vapour compression refrigeration system. A comparison of between the properties of refrigerants R-717, R-507a, R-134a, R-152a and R-1234yf is also presented.

The computational analysis has allowed the determination of the best energetic performance of the system. The analysis can represent a real system with a reasonable accuracy and is useful for future research work of whole system.

CHAPTER 1

INTRODUCTION

1.1. OVERVIEW

In the vapour compression cycle, first of all vapour is compressed to a super heated fluid, further cooled and condensed at constant pressure. Then refrigerant is allowed to throttle to a lower pressure by irreversible process, as a result producing a combination of liquid and vapour. Finally, at constant pressure liquid is evaporated. Heat rejected from refrigeration air conditioning plants is of low grade quality. Owing to the high costs related with the recovery of such heat and the accessibility of alternate way for requirement of low grade heat requirements, generally low grade waste heat is rejected to the atmosphere. On the other hand, the negative values that are being placed on waste or reject heat streams into the atmosphere by statutory bodies in developed countries, disposal of reject heat into the atmosphere is becoming even more costly [1].

In-spite from high temperature lift applications, there are applications where refrigerant at different temperatures is needed. For instance, dairy plant may require refrigerant at $–30^{o}C$ for producing ice cream and at $2^{o}C$ for producing chilling milk. In such cases it could be advantageous to use a multi-stage system with the low temperature evaporator working at $–30^{o}C$ and the high temperature evaporator operating at $2^{o}C$ [2].

Many experiments have been carried out for the performance improvement in multi-stage vapour compression refrigeration system by thermodynamic analysis but there may be chance arises of irreversibility destruction and losses in components of a system which causes difficulties in finding the results [3].

The second law analysis of thermodynamics is commonly used on multi-stage vapour compression refrigeration system, since it gives the idea about enhancement in coefficient of performance in addition to integration of multi-stage compression system. Such thermodynamic analysis also provides space for enhancement in the existing system [8].

The performance of a refrigeration system is evaluated in terms of COP called as first law efficiency that can be defined as the ratio of refrigeration effect to the total work input given to the system. The Coefficient of performance (COP) of a multi-stage vapour compression refrigeration system is enhanced by increasing the refrigeration effect in terms of cooling load capacity and it can also done by reducing compressor work input given to the system [14].

Commonly most of the refrigerant used in the industrial and domestic applications that have high GWP and ODP. Therefore those refrigerants have to be used must be eco-friendly and not have any environmental affect in the bad manner [15].

Simple vapour compression refrigeration system (single stage) consists of a high pressure side corresponding to condenser and a low pressure side corresponding to evaporator. These systems produce satisfactory results till the difference between evaporator and condenser temperature is not too high .There are applications where this difference can be really high. High difference may be due to low evaporator temperature or high condenser temperature. For instance, in frozen food industries the desired evaporator temperature can be as low as –40^{o}C, while in chemical industries the temperature can reach as low as –150^{o}C. On the high temperature side the required condensing temperatures can be very high if the refrigeration system is used as a heat pump for heating applications such as process heating, drying etc. because of some reasons above mentioned, as long as the temperature difference increases, the single stage systems become incompetent and not practical [22].

When I talk about two stage vapour compression cycles, the intermediate stage pressure equivalent to minimum compressor work is normally taken as the geometric mean of the evaporation pressure and refrigerant condensation. On the other hand the optimum interstage pressure is much different than the geometric mean pressure. When the flash chamber is integrated to the two-stage vapour compression system, there is substantial difference between the magnitude of geometric mean and optimum pressure values [23].

CHAPTER 2

LITERATURE REVIEW

Kaushik et al. [1] presented an investigation of the feasibility of the heat recovery from the condenser of a simple vapour compression refrigeration system through a canopus heat exchanger which acts as an auxiliary condenser between the compressor and condenser components. To recover the possible heat from the superheat of the discharged vapour and utilized it for increasing the temperature of the external fluid (water) removing heat from the condenser. The analysis is performed for the different inlet water temperature mass flow rate on the heat recovery output and its distribution over the condenser and canopus heat exchanger. These results were compared for different working fluids such as R-22, R-12, R-717, and R-500; found that a heat recovery factor of the order of 2.0 and 40% of condenser heat can be removed through the canopus heat exchanger.

Zubair and Yakub et al. [2] presented on thermodynamic analysis of vapour compression refrigeration cycle is investigated by both first and second laws of thermodynamics. Second law analysis is carried out for both two-stage and mechanical sub-cooling refrigeration cycle with refrigerants HFC-134a and CFC-22. The analysis is performed on each of the system components to determine the individual contribution to the overall system irreversible losses. In addition he showed that the optimum interstage pressure for two-stage and mechanical sub-cooling refrigeration system is very close to saturation pressure corresponding to the arithmetic mean of the refrigerant condensation and evaporation temperatures. These results were compared with the existing practices in the industry .Furthermore the theoretical results of a two-stage refrigeration system performance were also compared with experimental values for the refrigerants mentioned above.

Nikolaidis et al. [3] observed the performance of two-stage compression refrigeration having flash-chamber and water-intercooler using refrigerants R22 has been demonstrated by the thermodynamic analysis. He has taken the condenser saturation temperature was varied from 298-308 K and the evaporator saturation temperature from 238-228 K. The effects of temperature changes in the condenser and the evaporator on the plants irreversibility rate were determined. He found that the greater the temperature difference between either “the condenser and the environment” or “the evaporator and the cold room temperature”. There will be higher generation in irreversibility rate. Any reduction in irreversibility rate of the condenser gives approximately 2.40 times greater reduction in the irreversibility rate for the whole plant. Any reduction in the evaporator irreversibility rate gives a 2.87 times greater mean reduction in the irreversibility rate of the whole plant because the changes in the temperature in the condenser and the evaporator contributed significantly to the overall plant performance.

Rahman et al. [4] presented the performance of the recently developed integrated space condition system. A conventional split type air conditioner is modified to reclaim the superheated portion of the heat leaving the compressor to be utilized for the space conditioning purposes. The experimental investigation revealed that his device can cool down the room air temperature to the desired level, by using this type of energy recovery device compressor efficiency can be improved. The end result was expected to be faster cooling and prolonged compressor life.

Cabello and Torrella et al. [5] proposed the main operating variables on the energetic characteristics of the vapour compression plant. The tests were performed on a single-stage vapour compression plant using three different working fluids, R134a, R407C and R22. The variables considered are the evaporating pressure, the condensing pressure and the superheating degree at the compressor inlet. They concluded that the refrigerant mass flow rate is largely dependent on the suction specific volume and therefore on the suction conditions.

Apera and Renno et al. [6] has gone through the experimental studies of performance of a vapour compression refrigeration plant using as working fluids R22 and R417a. This type of plant was applied to a commercially available cold store, generally adopted for preservation of food stuff. They were working on regulated by on/off cycles of the compressors operating at the nominal frequency of 50 Hz imposed by the classical thermostatic control. The problem related to the replacement of the fully HCFC and of the partially HCFC have been only partially solved .As for R22, there is no unanimous solution of its substitution. In his work, the refrigerant fluid experimentally tested as a substitute for R22 is the non-azeotropic mixture R417a.

Ouadha and En-nacer et al. [7] carried out a detailed exergy analysis of a two stage vapour compression cycle by calculating its component exergetic loses. The exergy equations were used using refrigerant thermodynamic properties computed by means of two stage model of equations of states. The operated two-stage refrigeration system between a constant evaporating temperature of -30^{0}C and condensation temperatures of 30, 40 ,50 and 60^{0}C with two natural substitutes of HCFC22, namely propane (R290) and ammonia(R717) as working fluids . They found that the optimum interstage pressure for a two-stage refrigeration system is very close to the saturation pressure corresponding to the arithmetical mean of the refrigerant condensation and evaporation temperatures.

Arora and Kaushik [8] proposed a detailed exergy analysis of an actual vapour compression refrigeration (VCR) cycle. A computational model was developed for computing coefficient of performance (COP), exergy destruction, exergetic efficiency and efficiency defects for R502, R404A, and R507A. The analysis showed that for evaporator and condenser temperature in the range of -50^0 C to 0^{0}C and 40^{0}C to 55^0 C respectively. The results indicated that R507A is a better substitute to R502 than R404A. The efficiency defect in condenser is highest and lowest in liquid vapour heat exchanger for the refrigerants considered.

Torrella and Llopis et al. [12] described a second law analysis based on experimental data of a two-stage vapour compression system driven by a multi-stage compression system for medium and low capacity refrigeration unit. He used direct liquid water intercooler and flash chamber. The experimental analysis results showed that for an evaporating temperature range between -36^0 C and -20^0 C and for condensing temperature range between 30^{0}C and 47^{0}C using the refrigerant R-404a. The final results were compared with energy analysis from previous works. He found a new criterion of equivalence between the single stage vapour compression cycle and the two-stage compression cycle.

Santiago and Jose et al. [13] found that two stage cycle in case of high temperature difference between heat sink and heat source in order to overcome the high pressure ratios that deteriorate compressor volumetric and isentropic efficiency, thus it require high swept volume for giving the required cooling capacity and refrigeration effect. He proposed a model for two stage cycle with external sub cooling was developed that takes into account volumetric and isentropic efficiency of compressor dependant on the pressure ratios. First COP dependent on intermediate pressure is calculated for the standard low temperature conditions working with the refrigerants R-404a and R-717(Ammonia). Results showed that optimum intermediate pressure is close to the arithmetic mean in case of R-404a but there is a significant difference in case of ammonia.

Mishra and Jain et al. [14] worked on the replacement of R22 with several environment friendly refrigerants. They used hydrocarbon (HCF) refrigerants like R134a, R410a, R407C and M20. The developed a thermodynamic model to simulate the performance of actual system as closely as possible. Out of the above stated refrigerants R407C proved to be a potential HFC refrigerant which can replace R22 with minimum investment and efforts. It proved to be a non –ozone depleting refrigerant giving high system efficiency.

Aggarwal and Matani [15] stated that various obstacles faced in working of different refrigerants due to their environmental impact (R11, R12). Toxicity (NH_3), flammability (HC) and high pressure (CO_2) , which makes them more hazardous than other working fluids . They observed the performance of different environment friendly refrigerants and their mixtures in different proportions. They also observed the effect of working parameters like working pressure and working temperature which affect the coefficient of performance (COP) of vapour compression refrigeration system. They proved that there is a need of new and efficient refrigerants that have a low impact on global warming potential (GWP) and ozone depletion potential (ODP).

Gupta and Soni [16] provided a detailed exergy analysis for theoretical vapour compression refrigeration cycle using R404A, R407C and R410A. The equations of exergetic efficiency and exergy destruction for the main system components such as compressor, condenser, expansion devices (throttling valve), liquid-vapour heat exchanger and evaporator are developed. They showed an expression for coefficient of performance (COP)of refrigeration cycle and various results were obtained for the effect of evaporating temperatures, condensing temperatures , degree of sub-cooling and effectiveness of liquid-vapoour heat exchanger on COP.

Mishra [18] described a thermodynamic modeling of a vapour compression refrigeration system using R134a in primary circuit and Al_2O_3-water based nanofluids in the secondary circuit. In this model certain input variables like geometric characteristics, size of the nanoparticles and the compressor speed were used to calculate the output of the system. This system can be used to design various components like evaporator, condenser and throttling valve for vapour compression refrigeration systems (VCS). Simulation showed that for the same geometric characteristics of the system performance increases from 17% to 20% by application of nanofluid as a secondary fluid in VCS.

2.1. OUTLINE OF THE THESIS

The Thesis is formulated as,

- In this third chapter, titled as Theoretical analysis, theory of single stage vapour compression refrigeration system along with its applications and limitations is discussed.
- In the fourth chapter, titled as Theoretical analysis, theory of double stage vapour compression refrigeration system along with its applications and advantages over single stage vapour compression system is discussed.
- In the fifth chapter, titled as Thermodynamic modeling of modified double stage vapour compression refrigeration system along with its energy analysis and exergy analysis is discussed.
- In the sixth chapter, the results obtained after performing the energy and exergy analysis on the system and an elaborate discussion on the results obtained are presented.
- In the seventh chapter, the essence of whole thesis is presented in the form of conclusion, where the writer has concluded about the project and its future scope.

CHAPTER 3

SIMPLE VAPOUR COMPRESSION CYCLE

3.1. INTRODUCTION

Vapour compression refrigeration systems are the most commonly used refrigeration systems. In this cycle the working fluid i.e. refrigerant undergoes phase change at least during one process and refrigeration is occurred as the refrigerant evaporates at lower temperature. The net input work given to the system is in the form of mechanical energy that is used to run the compressor. Such a systems can also be called as mechanical refrigeration systems. Single stage vapour compression refrigeration systems can be applicable almost in all applications having refrigeration capacities varying from few Watts to few megawatts [17].

The Coefficient of performance (COP) of Carnot refrigeration cycle is primarily dependent on evaporator and condenser temperatures, due of this reason the same expression can be obtained for air cycle refrigeration systems working on Carnot cycle. The COP of the Carnot cycle refrigeration systems working between two constant temperature thermal reservoirs (heat source and sink) is dependent on temperature only. Carnot's theorems implies that, for the same heat, no irreversible cycle can have COP higher than that of Carnot COP for a given source and sink temperatures [19].

A single stage vapour compression refrigeration (VCR) cycle is a work absorbing cycle based on the reversed heat engine principle. It consists of mechanical components such as mechanical compressor, a condenser, an expansion device, and an evaporator as shown in Fig. 3.1.

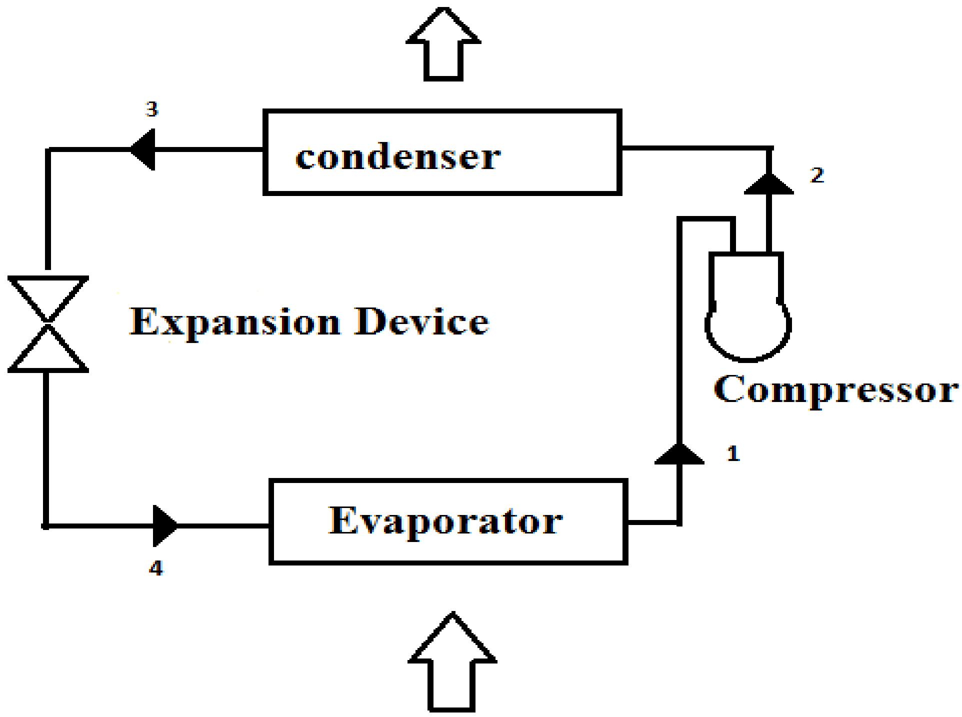

Figure 3.1: Schematic diagram of simple vapour compression refrigeration system

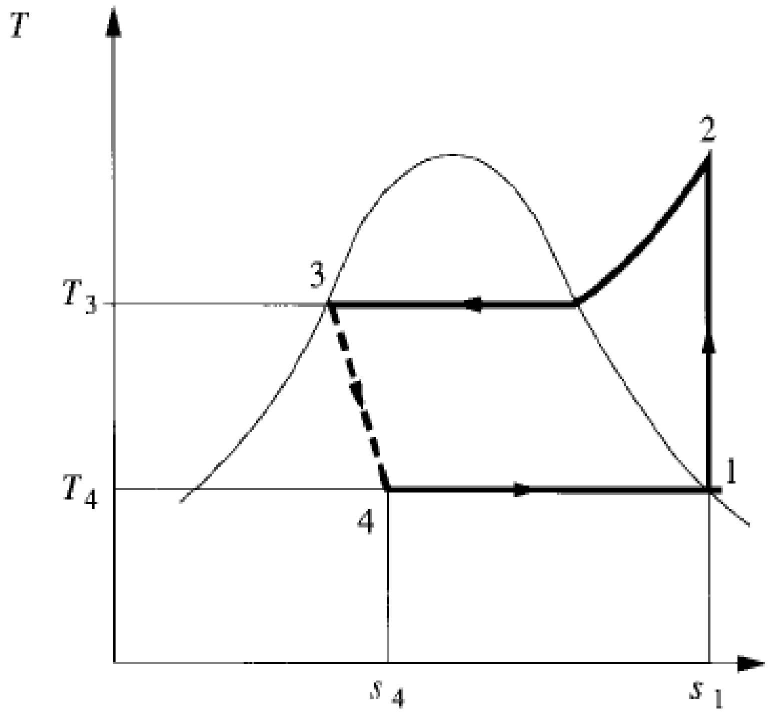

Figure 3.2: Simple vapour compression cycle presented on T-S diagram

Processes involve in the cycles shown in the figure 1 are as follows:

- 1-2 Isentropic compression of the refrigerant from compressor to condenser
- 2-3 Heat rejection occurring at constant pressure
- 3-4 Throttling of the refrigerants from condenser to evaporator
- 4-1 Heat absorption at constant pressure

3.2. Practical difficulties with Carnot refrigeration system:

1. During process 1-2, as shown in figure 3.1, a mixture consisting of liquid and vapour have to be compressed isentropically in the compressor. Such a compression is known as wet compression due to the presence of liquid.

II. Wet compression is very difficult especially with reciprocating compressors. This problem is particularly severe in case of high speed reciprocating compressors, which get damaged due to the presence of liquid droplets in the vapour. Even though some types of compressors can tolerate the presence of liquid in vapour, since reciprocating compressors are most widely used in refrigeration, traditionally dry compression (compression of vapour only) is preferred to wet compression.

III. The second practical difficulty with Carnot cycle is that using a turbine and extracting work from the system during the isentropic expansion of liquid refrigerant is not economically feasible, particularly in case of small capacity systems. This is due to the fact that the specific work output (per kilogram of refrigerant) from the turbine is given since the specific volume of liquid is much smaller compared to the specific volume of a vapour/gas, the work output from the turbine in case of the liquid will be small.

IV. One way of achieving dry compression in Carnot refrigeration cycle is to have two compressors – one isentropic and one isothermal i.e. multi-stage vapour compression system[21].

CHAPTER 4

TWO-STAGE VAPOUR COMPRESSION SYSTEM

4.1. How two-stage compression system Works

It consists of a low pressure (LP) and high pressure (HP) compressor, a water intercooler, two expansion valves, evaporator and condenser. The vapour leaving the low pressure (LP) compressor is intercooled in the water intercooler. After leaving the water intercooler, the vapour enters the flash intercooler where it is cooled by direct contact with the cold liquid refrigerant. The flash intercooler is a pressure vessel in which a constant level of liquid refrigerant is maintained by a float type expansion valve. The pressure in this vessel is intermediate pressure. The cooling of low pressure vapour is done by evaporation of liquid refrigerant in the flash chamber. Compression system employs heat and a refrigerant to produce refrigeration effect [12].

In its simplest design of multi-stage vapour compression system having flash chamber compression system consists of basic components as shown in given figure 4.1 are as follows:

1. Flash chamber

2. Condenser

3. Evaporator

4. High pressure (HP) compressor

5. Low pressure (LP) compressor

6. Expansion valve

7. Refrigerant

A two-stage compression system with flash tank for flash gas removal of refrigerants such as R 134a is shown in Figure 4.1 and Figure 4.2 shows the corresponding P-h diagram.

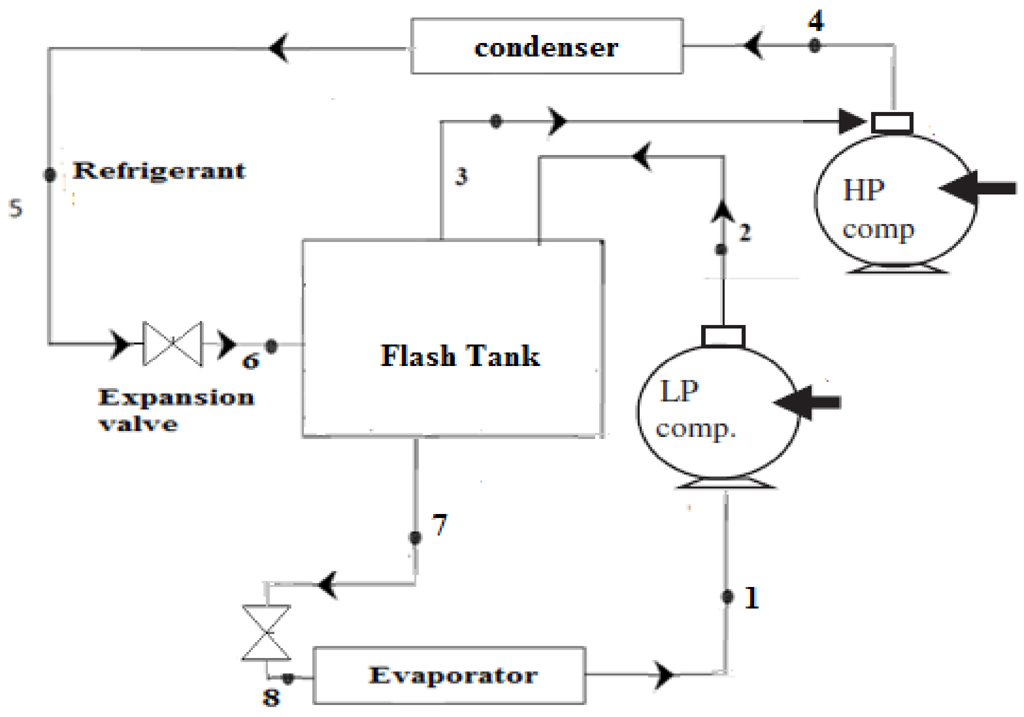

Figure 4.1: Schematic diagram of a two-stage compression system with flash chamber

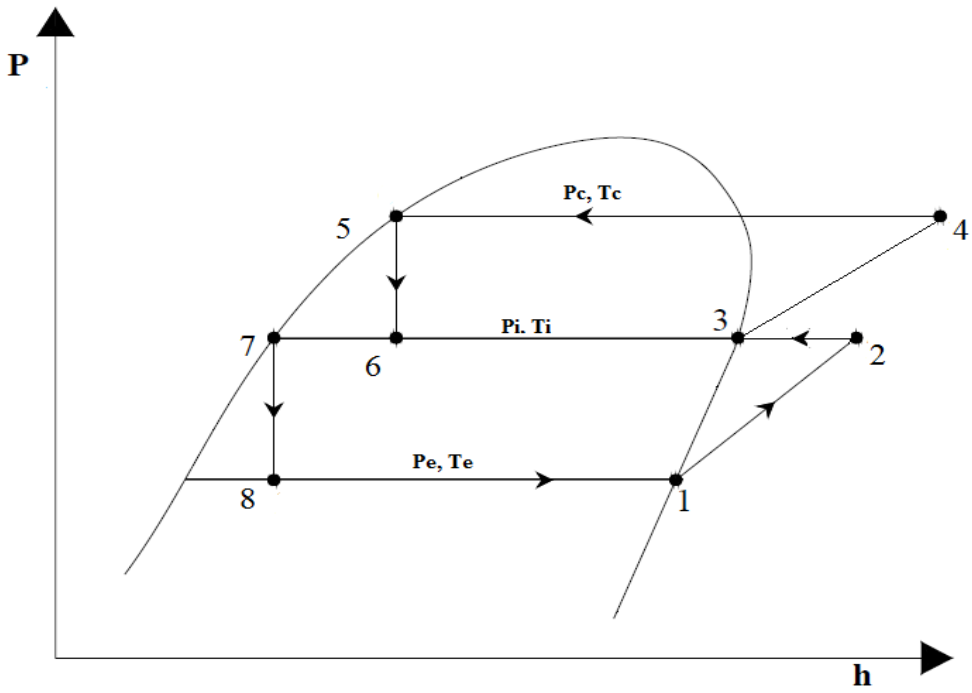

Figure 4.2: Two stage vapour compression refrigeration system with flash chamber on P-h diagram

4.2. Function of Components:

Condenser:

The function of a condenser is to condense the refrigerant. The heat gets transferred from the refrigerant vapour to the water, the refrigerant gets condensed. The condensed liquid refrigerant collects in the bottom of the condenser before travelling to the expansion device. The cooling water system is typically connected to a cooling tower. Generally, the generator and condenser are contained inside of the same shell.

Compressor:

Ay higher pressure and temperature refrigerant is compressed for condensation. Air or gas is compressed to a higher pressure and temperature. In order to reduce the temperature of the air or gas, heat is released to the surroundings at constant pressure.

Evaporator:

The principle of evaporator is to produce refrigeration effect and exchange the heat with the environment.

Expansion valve:

In an expansion system or throttling valve, Refrigerant is compressed to a higher pressure by mechanical energy. It is then cooled and expanded to a lower pressure, due the temperature of refrigerant gets lowered.

Refrigerant:

Refrigerant is the main working fluid used for absorbing and transmitting heat in a refrigeration system. Refrigerants absorb heat at a low temperature and low pressure and discharge heat at a higher temperature and pressure. Generally refrigerants undergoes phase changes during heat absorption—evaporation—and heat releasing—condensation.

Flash chamber:

In compound systems, flash coolers are used to sub-cool liquid refrigerant to the saturated temperature corresponding to the interstage pressure by vaporizing part of the liquid refrigerant. In a two-stage compound system with a flash cooler, a portion of the liquid flashed into vapour going directly to the second-stage suction inlet.

CHAPTER 5

THERMODYNAMIC MODELLING OF MODIFIED TWO STAGE VCR SYSTEM

In the present work, we studied a two stage vapour compression refrigeration system equipped with an auxiliary condenser (AuxC). The purpose of incorporating AuxC is to decrease the temperature of refrigerant coming out of the compressor exit and to superheat the saturated water flowing through the heat exchanger.

A computational thermodynamic model was developed for carrying out the analysis of the system using Engineering Equation Solver software [27]. To analyze the feasible realistic performance, a detailed thermodynamic first and second law analysis of modified two stage vapour compression refrigeration (VCR) system has been carried out.

5.1. System Description

This work is based on the modified vapour compression refrigeration (MVCR) system, where an Auxiliary condenser is incorporated between condenser and higher pressure (HP) compressor. The recovery of heat from both the condenser i.e. Auxiliary condenser (AuxC) and condenser is done through water. Refrigerants with desirable thermodynamic characteristics, such as R-717, R-134a, R-507a, R-152a, and R-1234yf are used. In the above two stage cycle, the refrigerant coming from the evaporator is drawn by low stage compressor and compressed isentropically at the cost of input work W_{comp} (process 1-2). The compressed refrigerant vapour is fed into the flash chamber (process 2-3) where saturated liquid and saturated vapour refrigerant at intermediate pressure gets separated, and then it passes through high stage compressor (process 3-4) isentropically . The compressed refrigerant at higher pressure directed into AuxC (process 4-4') where sensible heat Q_{AuxC} is recovered and then it passes through condenser where it condenses (process 4'-5) at constant pressure due to recovery of heat of condenser Q_C , with water used for cooling. The refrigerant from the condenser undergoes adiabatic expansion through high stage expansion valve, at constant enthalpy (process 5-6), after isenthalpic process it passes through again to flash chamber at intermediate pressure (process 6-7). The liquid refrigerant coming from the flash chamber undergoes adiabatic expansion through the low stage expansion valve, accompanied by a drop in pressure at constant enthalpy (process 7-8). The supposition of isenthalpic expansion, relatively than isentropic expansion is better from the point of view of realistic ease and economic constraints. Although, because of this there is a loss of energy in throttling. At reduced pressure the liquid refrigerant evaporates in the evaporator, thereby absorbing heat Q_E from the space where cooling is required. The low pressure vapour from the evaporator is compressed by the low pressure (LP) compressor and the cycle is repeated.

The rise in water temperature through the condenser is below the temperature of condensation of the refrigerant vapour. After compression, the superheated refrigerant vapour transfers a large quantity of heat to the water stream passes through the AuxC which has been given its name “Auxiliary” condenser.

To recover heat from the condenser, an additional Auxiliary condenser is incorporated, as shown in the figure 5.1. An external fluid (water) is used to recover heat from the condenser and the Auxiliary condenser. A counter-flow arrangement is employed in which the inlet water stream comes into contact with the condensing refrigerant and achieves temperature near to the condensation temperature. After leaving from the condenser, the water comes into contact with the higher temperature superheated refrigerant vapour leaving the compressor and achieves a temperature higher than the condensing temperature of the vapour. Greater heat recovery is possible by increasing the mass flow rate of water for a given inlet temperature of water [1].

Schematic line diagram of two stage vapour compression refrigeration system with auxiliary condenser is shown in figure 5.1

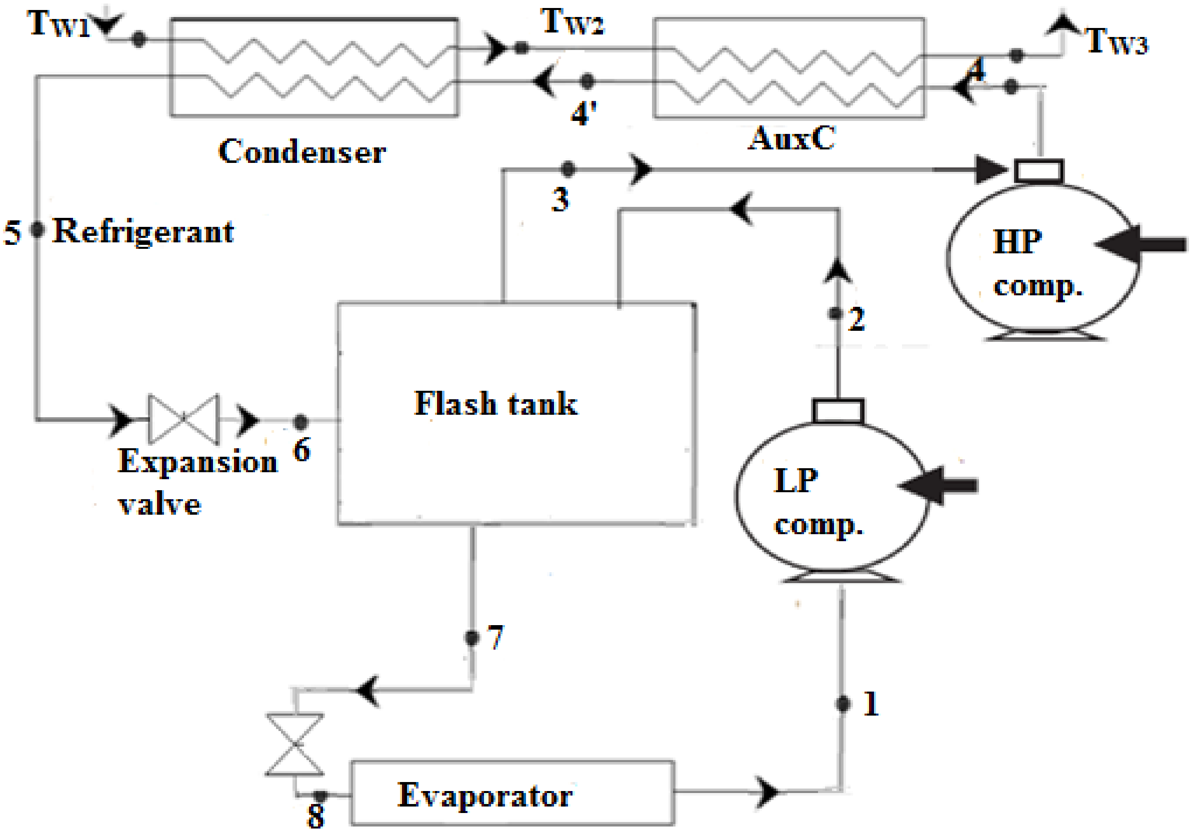

Figure 5.1: Schematic diagram two stage vapour compression refrigeration system with auxiliary condenser (AuxC).

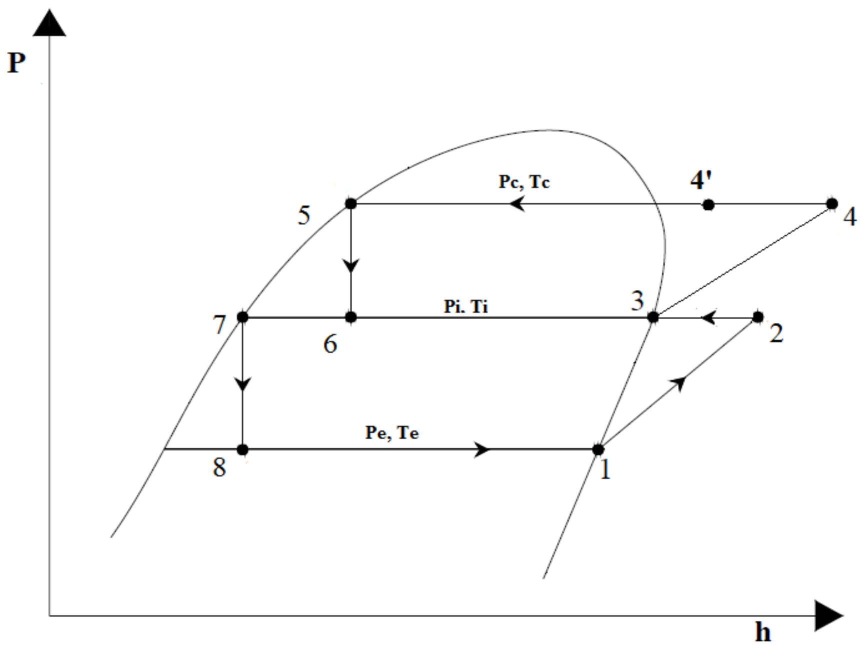

Figure 5.2: represents the Pressure (P) vs Enthalpy (h) diagram of modified VCR system

The process involved in the cycle (see figure 5.2) for present system is as follows:

- *Process* 1-2: Adiabatic compression of the vapour refrigerant at low stage compressor.
- *Process* 2-3: Heat transfer process at intermediate pressure in flash chamber.
- *Process* 3-4: Adiabatic compression of the vapour refrigerant at high stage compressor.
- *Process* 4-4': Constant pressure heat rejection process in AuxC.
- *Process* 4'-5: Constant pressure heat rejection process in Condenser.
- *Process* 5-6: Isenthalpic expansion process in high stage expansion valve.
- *Process* 6-7: Heat transfer process at intermediate pressure in Flash chamber.
- *Process* 7-8: Isenthalpic expansion process in low stage expansion valve.
- *Process* 8-1: Constant pressure heat absorption process (phase change from liquid to vapour).

5.1.1. Advantages of the modified VCR system

- The waste heat rejected by the vapour compression system is recovered through Auxiliary condenser is utilized in food processing, drying, space heating etc. Heat rejected by the condenser can be recovered to heat the working fluid.
- Recovery of waste heat further contributes in conservation of energy as well as it would be helpful in reduction of global warming.
- Since waste heat is being utilized therefore there will be increase in overall capacity of the system and increase COP.
- Air is abundant in atmosphere and doesn't need any external circuit to warm any space or thing means can directly fed to get the desired results.
- Maintenance and operating cost of air circulating circuit in less.

5.2. Thermodynamic modelling

5.2.1. Energy analysis

The thermodynamics first law (energy analysis) is related to energy and work losses, whereas the thermodynamics second law (exergy analysis) is related to entropy with irreversibility [11].

As we can consider each component of the system as open system and applying I_{st} law of thermodynamics i.e. steady flow energy equation, we will be able to obtain the energy interaction taking place in each of the component concerned.

The thermodynamics first law or energy balance for the steady flow process of an open system is given by:

$$\dot{Q}_i + W_i + \sum \dot{m}_i \left(h_i + \frac{v_i^2}{2} + gz_i \right) = \dot{Q}_e + W_e + \sum \dot{m}_e \left(h_e + \frac{v_e^2}{2} + gz_e \right) \tag{1}$$

Applying steady flow energy various components and neglecting change in kinetic energy and change in potential energy, we obtained the following expressions.

- **Evaporator**

  Heat extracted in the evaporator:

$$\dot{Q}_{8-1} = \dot{Q}_E = \dot{m}_{lp}\left(h_1 - h_8\right) \tag{2}$$

Where $\dot{m}_{lp}$ = mass flow rate of refrigerant in low stage compressor

$$\dot{m}_{lp} = \text{TR}\left[\frac{3.5167}{h_1 - h_8}\right]$$ [TR= ton of refrigeration]

- **Low Stage Compressor**

  Ideal work input to compressor:

$$\dot{W}_{lp} = W_1 = \dot{m}_{lp}\left(h_2 - h_1\right) \tag{3}$$

- **Flash Chamber**

  From energy balance of the flash tank:

$$\dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$

Isentropic efficiency of compressor*:* Ratio of ideal (isentropic) work required to the actual work required to do given task.

$$\eta_{isentropic} = \frac{W_{isentropic}}{W_{actual}} = \left(\frac{h_{4'} - h_3}{h_4 - h_3}\right) \tag{4}$$

Also, W_{actual} can also be written in the term of enthalpy as

$$\dot{W}_{actual} = \dot{W}_{hp} = \dot{m}_{hp}\left(h_4 - h_3\right) \tag{5}$$

$$W_{hp1} = \dot{m}_{hp}\left(h_{4'} - h_3\right) \tag{6}$$

$$\dot{W}_{net} = \dot{W}_{hp} + \dot{W}_{lp} \tag{7}$$

$$\dot{W}_{net1} = \dot{W}_{hp1} + \dot{W}_{lp} \tag{8}$$

The actual enthalpy h_4 can be calculated from the isentropic efficiency of compressor that can be defined as the ratio of the isentropic compression work to the actual compression work.

- **Condenser**

  Heat rejected by the condenser to the condenser to the surrounding is given by:

$$Q_C = Q_E + W_{net} \tag{9}$$

- **Expansion device**

  The expansion process is an isenthalpic (constant enthalpy) process, hence

  $h_5 = h_6$ (for high stage expansion valve)

  $h_7 = h_8$ (for low stage expansion valve)

Coefficient of performance (COP): obtained from second law of thermodynamics for two stage VCR system i.e. no heat recovery device is installed is given by:

$$COP_{actual} = \frac{Q_E}{W_{comp}} = \frac{\dot{m}_{lp}\left(h_1 - h_8\right)}{W_{net}} \tag{10}$$

Auxiliary Condenser (AuxC):

Heat removed through the auxiliary condenser at constant pressure is given by:

$$Q_{AuxC} = \dot{m}_W * C_w * \left(T_{w3} - T_{w2}\right) \tag{11}$$

For a given mass flow rate of water ($\dot{m}_W$) and inlet temperature (T_{w1}), the outlet temperature of water (T_{w2}) is given by:

$$T_{w2} = T_{w1} + \left[\frac{Q_C}{\dot{m}_W * C_W} \right] \tag{12}$$

To obtain higher water temperature than that of condenser temperature, an Auxiliary condenser of effectiveness (ε) is applied between the high pressure (HP) compressor and the condenser. The water outlet temperature through the Auxiliary condenser is given as:

$$T_{w3} = T_{w2} + \varepsilon * \left(T_4 - T_{w2}\right) \tag{13}$$

The fraction of the condenser heat removed by the Auxiliary condenser (FR) is given as:

$$\text{FR} = \frac{Q_{AuxC}}{Q_C} = \frac{\dot{m}_w * C_w * \left(T_{w3} - T_{w2}\right)}{Q_E + W_{net}} \tag{14}$$

The heat removal factor (HRF) is given by the relation:

$$HRF = \left[\frac{T_{w3} - T_{w1}}{T_{w2} - T_{w1}} \right] \tag{15}$$

Overall COP of the two-stage VCR with Auxiliary condenser for combined cooling and heating is given by the relation:

$$COP_{AUXC} = \frac{Q_E}{W_{net1}} \tag{16}$$

Total heat removed by condenser when water is flowing through it, for a given mass flow rate ($\dot{m}_W$) and at fixed inlet temperature (T_{w1}) is given by:

$$Q_{cond} = \dot{m}_{hp} * \left(h_{4'} - h_5\right) + \dot{m}_w * \left(h_{w2} - h_{w1}\right) \tag{17}$$

5.3. EXERGY ANALYSIS

The second law of thermodynamics infers the concept of exergy, a powerful tool for analysing both the quantity and quality of energy utilization. It is defined as the maximum amount of work obtainable when the stream of matter is brought from its initial state to the dead state by the processes during which the stream may interact only with the environment. The exergy balance is similar to an energy balance but has the fundamental difference that, while the energy balance is a statement of a law of conservation of energy, the exergy may be looked upon as a statement of law of degradation of energy [24].

Exergy analysis is a powerful tool in the design, optimization, and performance evaluation of energy systems. An exergy balance applied to a process or a whole plant tell us how much of the usable work potential, or exergy supplied as the inlet to the system under consideration has been consumed (irreversibly lost) by the process. The exergy destruction or irreversibility provides a generally applicable quantitative measure of process inefficiency. Analysing a multi-component plant indicates the total plant irreversibility distribution among the plant components, pinpointing those contributing most to overall plant inefficiency. [24].

Exergy analysis is useful for improving the efficiency of energy-resource use, since it quantifies the locations, types and magnitudes of losses [25].

Exergy balance for a control region undergoing a steady-state process is expressed as

$$\dot{\mathrm{X}}_i + \dot{\mathrm{X}}_{\mathrm{j}}^{\mathrm{Q}} = \dot{\mathrm{X}}_e + \dot{\mathrm{W}}_{\mathrm{j}} + \mathrm{E}\dot{\mathrm{D}}_{\mathrm{j}}$$

$$\dot{\mathrm{X}}_i = \sum_{\mathrm{IN}} \dot{\mathrm{m}}\, x$$

$$\dot{\mathrm{X}}_e = \sum_{\mathrm{OUT}} \dot{\mathrm{m}}\, x$$

$$\dot{\mathrm{X}}_{\mathrm{j}}^{\mathrm{Q}} = \sum \left[\dot{\mathrm{Q}}_{\mathrm{j}} \frac{\mathrm{T} - \mathrm{T}_{\mathrm{o}}}{\mathrm{T}}\right]$$

$$x = (\mathrm{h} - \mathrm{T}_{\mathrm{o}}\mathrm{s}) - (\mathrm{h}_{\mathrm{o}} - \mathrm{T}_{\mathrm{o}}\mathrm{s}_{\mathrm{o}})$$

where the first term on left hand and right hand side represent physical exergy (neglecting kinetic, potential and chemical exergy component) of stream of matter entering and leaving the control region respectively. The second term on left hand side and right hand side is thermal exergy flow, which gives exergy transfer rate corresponding to the heat transfer rate Q when the temperature at the control surface where heat transfer is occurring is T and exergy associated with work transfer to and from the control region. ED represents rate of exergy destruction [24].

5.3.1. Exergy Destruction(ED)

Irreversibility, such as friction, mixing, chemical reactions, and heat transfer through a finite temperature difference, unrestrained expansion, non-quasi-equilibrium compression or expansion always generate entropy, and anything that generates entropy always destroys exergy. Exergy destroyed is a positive quantity for any actual process and becomes zero for a reversible process. Exergy destroyed represents the lost work potential and is also called the irreversibility or lost work

- **Evaporator**

$$\dot{ED}_E = \dot{X}_8 + \dot{Q}_E\left(1 - \frac{T_o}{T_E}\right) - \dot{X}_1 = \dot{m}_{lp}[(h_8 - h_1) - T_o(s_8 - s_1)] + \dot{Q}_E\left(1 - \frac{T_o}{T_E}\right) \quad (18)$$

- **Compressor-1**

$$\dot{ED}_{comp\ 1} = \dot{X}_1 + \dot{W}_{comp\ 1} - \dot{X}_2 = \dot{m}_{lp}[T_o(s_1 - s_2)] \quad (19)$$

- **Condenser**

$$\dot{ED}_c = \dot{X}_{4'} - \dot{X}_5 = \dot{m}_{hp}[(h_{4'} - h_5) - T_o(s_{4'} - s_5)] + \dot{m}_w[(h_{w1} - h_{w2}) - T_o(s_{w1} - s_{w2})] \quad (20)$$

- **Auxiliary condenser**

$$\dot{ED}_{AuxC} = \left(\dot{X}_4 - \dot{X}_{4'}\right) = \dot{m}_{hp}\left[(h_4 - h_{4'}) - T_o[(s_4 - s_{4'})]\right] + \dot{m}_w[(h_{w2} - h_{w3}) - T_o(s_{w2} - s_{w3})] \quad (21)$$

- **Throttle Valve-1**

$$\dot{ED}_{t1} = \dot{m}_{hp}[(h_5 - h_6)\ T_o(s_5 - s_6)] \quad (22)$$

- **Throttle Valve-2**

$$\dot{ED}_{t2} = \dot{m}_{lp}\left[(h_7 - h_8) - T_o(s_7 - s_8)\right] \tag{23}$$

- **Compressor-2**

$$\dot{ED}_{comp\ 2} = \dot{X}_3 + \dot{W}_{comp\ 2} - \dot{X}_{4'} = \dot{m}_{hp}\left[(h_3 - h_{4'}) - T_o(s_3 - s_{4'})\right] \tag{24}$$

5.3.2. Total Exergy Destruction

It is the sum of exergy destruction in different components of the system

$$\dot{ED}_{total} = \dot{ED}_{comp1} + \dot{ED}_{comp2} + \dot{ED}_{E} + \dot{ED}_{cond} + \dot{ED}_{tv1} + \dot{ED}_{tv2} + \dot{ED}_{AuxC} + \dot{ED}_{f.chamber} \tag{25}$$

5.3.3. Exergetic Efficiency

$$\eta_{exergetic} = \frac{\text{minimum exergy required to do given task}}{\text{actual exergy consumed}} \tag{26}$$

For vapour compression refrigeration system, desired task is heat abstraction in to the evaporator from the space to be cooled at temperature T_E and minimum exergy required is the thermal exergy flow between evaporator and space to be cooled, i.e.

$$\dot{X}_E^Q = \dot{Q}_E\left|\left(1 - \frac{T_o}{T_E}\right)\right| \tag{27}$$

and actual exergy consumed is actual compressor work input, $\dot{W}_{comp}$.

Hence, exergetic efficiency is given by

$$\eta_{exergetic} = \frac{\dot{Q}_E\left|\left(1-\frac{T_o}{T_E}\right)\right|}{\dot{W}_{comp}} = \frac{COP_{vcr}}{COP_{rev}} \tag{28}$$

where COP_{rev} and COP_{vcr} are coefficient of performance of reversible refrigerator operating between T_O and T_E and of actual vapour compression cycle respectively.

5.3.4. Exergy Destruction Ratio (EDR)

EDR is defined as the ratio of total exergy destruction in the system to minimum exergy required in doing a given task and is given by

$$\mathrm{EDR} = \frac{\dot{ED}_{total}}{\dot{X}_E^Q} = \frac{COP_{vcr}}{COP_{rev}} - 1 \tag{29}$$

EDR in terms of exergetic efficiency can be written as

$$\mathrm{EDR} = \frac{1}{\eta_{exergetic}} - 1 \tag{30}$$

5.3.5. Efficiency Defect $(\boldsymbol{\delta_j})$

That fraction of the input which is lost through irreversibilities in the different components is called efficiency defect. It gives a direct casual relationship between component irreversibilities and their effect on the efficiency of the plant [24]. It is the ratio between rate of exergy destruction in j-th component to the actual exergy consumed (i.e. actual compressor work) and is given by (31).

$$\delta_j = \frac{\dot{ED}_j}{\dot{W}_{comp}} \tag{31}$$

- **Evaporator**

$$\delta_e = \frac{\dot{ED}_E}{\dot{W}_{comp}} \tag{32}$$

- **Compressor-1**

$$\delta_{comp} = \frac{\dot{ED}_{comp\ 1}}{\dot{W}_{comp}} \tag{33}$$

- **Condenser**

$$\delta_c = \frac{\dot{ED}_c}{\dot{W}_{comp}} \tag{34}$$

- **Auxiliary Condenser**

$$\delta_{AuxC} = \frac{\dot{ED}_{AuxC}}{\dot{W}_{comp}} \tag{35}$$

- **Throttle Valve-1**

$$\delta_{t1} = \frac{\dot{ED}_{t1}}{\dot{W}_{comp}} \tag{36}$$

- **Throttle Valve-2**

$$\delta_{t2} = \frac{\dot{ED}_{t2}}{\dot{W}_{comp}} \tag{37}$$

5.4. Simulation study

Simulation was performed to evaluate the integrated Refrigeration system with the following assumptions:

1. The flow through all the components is under steady states.
2. The pressure drop due to friction within the refrigeration system can be neglected, except through the expansion valve.
3. The fluid streams in the piping between the components and the heat exchangers are adiabatic.
4. Assuming the kinetic and potential energy change as zero in the system.

5.5. Current applications of refrigeration

The widely-used present applications of refrigeration are of the air-conditioning of private home and public buildings, and the refrigeration of foodstuffs in homes, restaurants and big storage warehouses. Dairy products are persistently in need of refrigeration and it was only revealed in the past few decades that eggs needed to be refrigerated during shipment rather than waiting to be refrigerated after arrival at the grocery store. Meats, poultry and fish all must be kept in climate-controlled environments before being sold. Refrigeration also maintains fruits and vegetables edible longer. In multi stage ammonia is used in central air–conditioning plant.

The theoretical analysis of the two stage VCR system has been carried out for the following applications. These are the evaporator temperature of two stage VCR system.

1) Ice cream : -20℃
2) Storage of frozen food : -25℃
3) Fisher items : -30℃
4) Frozen food industries : -40℃
5) Chemical industries : -50℃

5.6. Selection criterion for input parameters and their values

- Depending upon the five different applications of the system the evaporator temperature (T_E) has been varied in the range of -20 to -50℃ in step increment of 2 ℃.
- Normal ambient condition were the deciding factor for the temperature range of condenser (T_c), which is 40℃ and 50℃.

A mathematical computational model is developed for performing the energy and exergy analysis of the Integrated Refrigeration System as shown in figure 5.1 using EES software [27].

The input data assumed for the calculation of results shown in fig. 5.1 are:

1. Refrigerant: R-717, R-507a, R-134a, R-152a, R-123yf4
2. Effectiveness of the AuxC (ε_{AuxC}) : 0.85
3. Condenser temperature(T_c) : 40℃ and 50℃.
4. Evaporator temperature(T_E) : -20 to -50℃
5. Isentropic efficiency of compressor: (η_{comp}) = 0.9
6. Ambient state temperature (T_o) : 298K
7. Water inlet temperature flowing through the condenser (T_{w1}):283K
8. Ambient atmospheric pressure (P_{atm}):101.325Kpa
9. Tonns of refrigeration(TR) :10
10. Specific heat of cooling fluid water ($C_{w)}$:4.18 Kj/Kg-K
11. It is assumed that pressure drop in evaporator; condenser and auxiliary condenser is negligible.
12. Mass flow rate of cooling fluid (water) is varied from 0.18 to 0.85, while discussing its effect on the system performance.

CHAPTER 6

RESULTS AND DISCUSSIONIN

In order to have a numerical appreciation of heat recovery from a two stage vapour compression refrigeration system, a computer program has been developed in Engineering Equation Solver [27], using the set of input parameters given in previous chapter.

6.1. RESULTS

An extensive exergy analysis has been performed and the results obtained are arranged in the tabular form. Also the comparison between various parameters have been done e.g. after varying evaporator temperature, ambient state temperature, and mass flow rate of water, and at two different values of condenser temperature, has been represented in graphs.

6.1.1. Variation of Evaporator Temperature

The following tables from table no. 8.1 to 8.10 shows the effect of varying evaporator temperature on the various system parameters, corresponding to condenser temperature of 313K and 323K.

Table 6.1: Effect of evaporator temperature (T_E) on the different parameters for R-717

(at T_C = 313K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_W$ = 0.27 kg/s)

T_E (K)	T_C (K)	COP_{actual}	COP_{AuxC}	Q_{AuxC}	ED_{total}	$\eta_{exergetic}$	EDR	FR	HRR
233	313	2.421	2.56	42.27	19.43	0.6755	1.981	0.8506	1.413
235	313	2.52	2.663	40.46	18.53	0.6755	1.965	0.8237	1.397
237	313	2.623	2.772	38.71	17.66	0.6751	1.951	0.797	1.381
239	313	2.732	2.886	37.02	16.82	0.6743	1.938	0.7705	1.366
241	313	2.847	3.006	35.37	16.02	0.6733	1.926	0.7443	1.351
243	313	2.968	3.133	33.77	15.25	0.6718	1.916	0.7183	1.337
245	313	3.097	3.268	32.22	14.51	0.6699	1.907	0.6926	1.323
247	313	3.233	3.411	30.72	13.79	0.6675	1.9	0.6671	1.309
249	313	3.378	3.563	29.26	13.11	0.6647	1.894	0.6419	1.296
251	313	3.532	3.724	27.84	12.44	0.6614	1.89	0.617	1.283
253	313	3.697	3.896	26.46	11.81	0.6575	1.887	0.5923	1.271
255	313	3.872	4.081	25.13	11.19	0.653	1.887	0.5679	1.258

Table 6.2: Effect of evaporator temperature (T_E) on the different parameters for R-717

(at T_C = 323K, ε_{AuxC} = 0.85,η_{comp} = 0.9, $\dot{m}_W$ = 0.29 kg/s)

T_E (K)	T_C (K)	COP_{actual}	COP_{AuxC}	Q_{AuxC}	ED_{total}	$\eta_{exergetic}$	EDR	FR	HRR
233	323	2.25	2.38	55.59	22.81	0.6276	2.325	1.094	1.445
235	323	2.338	2.472	53.61	21.81	0.6267	2.313	1.068	1.428
237	323	2.43	2.569	51.69	20.85	0.6254	2.303	1.041	1.412
239	323	2.526	2.671	49.83	19.92	0.6237	2.295	1.015	1.396
241	323	2.628	2.777	48.03	19.03	0.6216	2.288	0.9893	1.38
243	323	2.736	2.89	46.28	18.17	0.6192	2.283	0.9636	1.366
245	323	2.849	3.008	44.58	17.35	0.6162	2.281	0.9382	1.351
247	323	2.968	3.133	42.92	16.56	0.6129	2.28	0.913	1.337
249	323	3.095	3.266	41.32	15.79	0.609	2.282	0.8881	1.323
251	323	3.229	3.406	39.77	15.06	0.6046	2.287	0.8634	1.31
253	323	3.371	3.555	38.26	14.35	0.5995	2.294	0.8389	1.297
255	323	3.522	3.713	36.79	13.67	0.5939	2.305	0.8148	1.284

Table 6.3: Effect of evaporator temperature (T_E) on the different parameters for R-507a

(at T_C = 313K, ε_{AuxC} = 0.85,η_{comp} = 0.9, $\dot{m}_W$ = 0.58 kg/s)

T_E (K)	T_C (K)	COP_{actual}	COP_{AuxC}	Q_{AuxC}	ED_{total}	$\eta_{exergetic}$	EDR	FR	HRR
233	313	2.195	2.326	26.19	17.85	0.6122	1.819	0.5116	1.456
235	313	2.284	2.42	26.47	17.22	0.6122	1.827	0.5234	1.438
237	313	2.378	2.518	26.74	16.62	0.6119	1.836	0.5352	1.421
239	313	2.476	2.621	27	16.03	0.6113	1.847	0.547	1.404
241	313	2.581	2.731	27.26	15.46	0.6104	1.858	0.5588	1.387
243	313	2.691	2.846	27.52	14.9	0.6091	1.872	0.5705	1.372
245	313	2.808	2.968	27.76	14.35	0.6075	1.887	0.5821	1.356
247	313	2.932	3.098	28	13.82	0.6054	1.904	0.5938	1.341
249	313	3.064	3.236	28.23	13.31	0.603	1.923	0.6053	1.326
251	313	3.204	3.383	28.46	12.8	0.6	1.944	0.6168	1.312
253	313	3.354	3.539	28.68	12.31	0.5966	1.968	0.6283	1.298
255	313	3.514	3.707	28.89	11.83	0.5926	1.995	0.6396	1.285

Table 6.4: Effect of evaporator temperature (T_E) on the different parameters for R-507a

(at T_C = 323K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_w$ = 0.6 kg/s)

T_E (K)	T_C (K)	COP_{actual}	COP_{AuxC}	Q_{AuxC}	ED_{total}	$\eta_{exergetic}$	EDR	FR	HRR
233	323	1.79	1.904	48.5	23.07	0.4994	2.352	0.8848	1.559
235	323	1.86	1.976	48.89	22.33	0.4985	2.369	0.904	1.538
237	323	1.932	2.053	49.26	21.62	0.4974	2.389	0.9231	1.517
239	323	2.009	2.133	49.63	20.93	0.4958	2.411	0.9422	1.498
241	323	2.089	2.216	49.98	20.25	0.494	2.435	0.9611	1.479
243	323	2.172	2.304	50.33	19.6	0.4917	2.462	0.98	1.46
245	323	2.261	2.397	50.66	18.96	0.489	2.493	0.9987	1.442
247	323	2.353	2.494	50.98	18.34	0.4859	2.526	1.017	1.425
249	323	2.451	2.596	51.29	17.74	0.4824	2.563	1.036	1.408
251	323	2.554	2.704	51.59	17.15	0.4783	2.605	1.054	1.391
253	323	2.663	2.818	51.87	16.58	0.4737	2.651	1.072	1.375
255	323	2.779	2.939	52.15	16.02	0.4686	2.702	1.091	1.36

Table 6.5: Effect of evaporator temperature (T_E) on the different parameters for R-134a

(at T_C = 313K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_w$ = 0.55 kg/s)

T_E (K)	T_C (K)	COP_{actual}	COP_{AuxC}	Q_{AuxC}	ED_{total}	$\eta_{exergetic}$	EDR	FR	HRR
233	313	2.376	2.518	25.9	16.6	0.663	1.692	0.5184	1.421
235	313	2.471	2.617	25.96	16.03	0.6625	1.7	0.5255	1.405
237	313	2.571	2.721	26.02	15.47	0.6616	1.709	0.5328	1.389
239	313	2.676	2.831	26.09	14.93	0.6605	1.72	0.5401	1.374
241	313	2.787	2.947	26.16	14.4	0.6591	1.731	0.5475	1.359
243	313	2.904	3.07	26.24	13.88	0.6572	1.744	0.555	1.344
245	313	3.028	3.2	26.31	13.38	0.655	1.759	0.5625	1.33
247	313	3.16	3.338	26.39	12.89	0.6524	1.775	0.5701	1.316
249	313	3.3	3.485	26.48	12.41	0.6494	1.793	0.5778	1.303
251	313	3.449	3.641	26.56	11.94	0.6458	1.813	0.5855	1.29
253	313	3.608	3.807	26.64	11.49	0.6417	1.836	0.5932	1.277
255	313	3.779	3.986	26.73	11.04	0.6372	1.862	0.6011	1.265

Table 6.6: Effect of evaporator temperature (T_E) on the different parameters for R-134a

(at T_C = 323K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_w$ = 0.58 kg/s)

T_E (K)	T_C (K)	COP_{actual}	COP_{AuxC}	Q_{AuxC}	ED_{total}	$\eta_{exergetic}$	EDR	FR	HRR
233	323	2.016	2.14	48.38	20.85	0.5625	2.125	0.9197	1.496
235	323	2.092	2.22	48.52	20.2	0.5608	2.143	0.9334	1.478
237	323	2.171	2.302	48.65	19.57	0.5588	2.162	0.9471	1.461
239	323	2.254	2.389	48.78	18.96	0.5564	2.184	0.9609	1.444
241	323	2.341	2.48	48.92	18.36	0.5536	2.208	0.9747	1.427
243	323	2.432	2.576	49.06	17.78	0.5505	2.234	0.9885	1.411
245	323	2.528	2.676	49.19	17.22	0.5469	2.263	1.002	1.396
247	323	2.629	2.782	49.33	16.66	0.5429	2.295	1.016	1.38
249	323	2.736	2.894	49.47	16.13	0.5384	2.33	1.03	1.366
251	323	2.848	3.012	49.6	15.6	0.5333	2.369	1.044	1.351
253	323	2.967	3.136	49.74	15.09	0.5278	2.413	1.058	1.337
255	323	3.093	3.268	49.87	14.59	0.5216	2.461	1.072	1.323

Table 6.7: Effect of evaporator temperature (T_E) on the different parameters for R-152A

(at T_C = 313K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_w$ = 0.4 kg/s)

T_E (K)	T_C (K)	COP_{actual}	COP_{AuxC}	Q_{AuxC}	ED_{total}	$\eta_{exergetic}$	EDR	FR	HRR
233	313	2.453	2.596	18.72	15.66	0.6844	1.596	0.3781	1.408
235	313	2.549	2.697	18.44	15.1	0.6834	1.602	0.3767	1.392
237	313	2.65	2.803	18.18	14.56	0.6821	1.608	0.3753	1.377
239	313	2.757	2.914	17.92	14.03	0.6805	1.616	0.374	1.363
241	313	2.869	3.032	17.68	13.51	0.6786	1.625	0.3728	1.349
243	313	2.988	3.157	17.45	13.01	0.6763	1.635	0.3718	1.335
245	313	3.114	3.289	17.23	12.52	0.6737	1.646	0.3708	1.321
247	313	3.248	3.428	17.01	12.04	0.6706	1.659	0.3699	1.308
249	313	3.39	3.577	16.81	11.58	0.6671	1.673	0.3691	1.295
251	313	3.541	3.735	16.61	11.12	0.663	1.689	0.3683	1.282
253	313	3.702	3.904	16.42	10.68	0.6585	1.707	0.3676	1.27
255	313	3.875	4.085	16.24	10.24	0.6534	1.728	0.367	1.258

Table 6.8: Effect of evaporator temperature (T_E) on the different parameters for R-152a

(at T_C = 323K, ε_{AuxC} = 0.85,η_{comp} = 0.9, $\dot{m}_w$ = 0.45 kg/s)

T_E (K)	T_C (K)	COP_{actual}	COP_{AuxC}	Q_{AuxC}	ED_{total}	$\eta_{exergetic}$	EDR	FR	HRR
233	323	2.107	2.233	41.14	20.04	0.5877	2.043	0.7932	1.475
235	323	2.184	2.314	40.85	19.4	0.5854	2.058	0.7967	1.458
237	323	2.264	2.398	40.57	18.78	0.5828	2.075	0.8003	1.442
239	323	2.349	2.487	40.31	18.18	0.5798	2.094	0.804	1.426
241	323	2.437	2.579	40.06	17.59	0.5764	2.115	0.8077	1.41
243	323	2.53	2.677	39.82	17.02	0.5727	2.138	0.8116	1.395
245	323	2.628	2.779	39.59	16.46	0.5685	2.164	0.8154	1.381
247	323	2.731	2.887	39.36	15.92	0.5639	2.193	0.8193	1.366
249	323	2.839	3.001	39.15	15.39	0.5588	2.224	0.8233	1.352
251	323	2.954	3.121	38.94	14.87	0.5532	2.259	0.8273	1.339
253	323	3.075	3.247	38.74	14.37	0.547	2.297	0.8314	1.325
255	323	3.204	3.382	38.55	13.88	0.5402	2.34	0.8355	1.312

Table 6.9: Effect of evaporator temperature (T_E) on the different parameters for R-1234yf

(at T_C = 313K, ε_{AuxC} = 0.85,η_{comp} = 0.9, $\dot{m}_w$ = 0.48 kg/s)

T_E (K)	T_C (K)	COP_{actual}	COP_{AuxC}	Q_{AuxC}	ED_{total}	$\eta_{exergetic}$	EDR	FR	HRR
211	313	1.511	1.61	1.493	23.55	0.6232	1.624	0.02555	1.662
213	313	1.568	1.67	2.21	22.77	0.6258	1.622	0.03836	1.638
215	313	1.627	1.732	2.903	22.01	0.6282	1.622	0.05113	1.615
217	313	1.689	1.797	3.574	21.28	0.6305	1.621	0.06383	1.592
219	313	1.753	1.864	4.224	20.57	0.6325	1.622	0.07648	1.57
221	313	1.821	1.935	4.853	19.88	0.6343	1.623	0.08907	1.549
223	313	1.891	2.008	5.463	19.21	0.6359	1.625	0.1016	1.529
225	313	1.964	2.085	6.054	18.56	0.6373	1.627	0.1141	1.509
227	313	2.041	2.166	6.628	17.93	0.6385	1.63	0.1265	1.49
229	313	2.122	2.251	7.185	17.31	0.6394	1.634	0.1389	1.471
231	313	2.207	2.339	7.725	16.72	0.64	1.639	0.1512	1.453
233	313	2.296	2.432	8.25	16.13	0.6404	1.645	0.1634	1.436

Table 6.10: Effect of evaporator temperature (T_E) on the different parameters for R-1234yf

(at T_C = 323K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_w$ = 0.5 kg/s)

T_E (K)	T_C (K)	COP_{actual}	COP_{AuxC}	Q_{AuxC}	ED_{total}	$\eta_{exergetic}$	EDR	FR	HRR
211	323	1.277	1.365	17.77	28.81	0.5267	1.987	0.2834	1.783
213	323	1.325	1.415	18.61	27.89	0.5288	1.987	0.3016	1.755
215	323	1.375	1.467	19.42	27.01	0.5307	1.989	0.3197	1.727
217	323	1.426	1.521	20.21	26.15	0.5323	1.992	0.3377	1.701
219	323	1.479	1.577	20.96	25.32	0.5337	1.996	0.3557	1.676
221	323	1.535	1.635	21.69	24.52	0.5348	2.001	0.3735	1.652
223	323	1.593	1.696	22.4	23.74	0.5356	2.007	0.3913	1.628
225	323	1.653	1.759	23.08	22.99	0.5362	2.015	0.4089	1.605
227	323	1.715	1.825	23.74	22.26	0.5365	2.023	0.4265	1.583
229	323	1.781	1.893	24.38	21.55	0.5366	2.034	0.444	1.562
231	323	1.849	1.965	25	20.86	0.5363	2.045	0.4614	1.541

6.1.2. Variation of Ambient State Temperature (T_0)

Table 6.11: Effect of ambient temperature (T_0) on the different parameters for R-717

(at T_C = 313K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_w$ = 0.27 kg/s)

T_0 (K)	T_C (K)	EDR	$\eta_{exergetic}$	ED_{total}
298	313	7.946	0.6579	115.2
301	313	7.461	0.6805	111.9
304	313	7.006	0.7032	108.6
307	313	6.58	0.7259	105.3
310	313	6.18	0.7486	102
313	313	5.803	0.7713	98.65
316	313	5.448	0.794	95.33
319	313	5.112	0.8166	92.02
322	313	4.795	0.8393	88.7
325	313	4.494	0.862	85.39
328	313	4.209	0.8847	82.07

Table 6.12: Effect of ambient temperature (T_0) on the different parameters for R-717

(at T_C = 323K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_W$ = 0.29 kg/s)

T_0 (K)	T_C (K)	EDR	$\eta_{exergetic}$	ED_{total}
298	323	7.411	0.5823	107.5
301	323	6.972	0.6024	104.6
304	323	6.561	0.6225	101.7
307	323	6.176	0.6426	98.81
310	323	5.814	0.6626	95.93
313	323	5.473	0.6827	93.05
316	323	5.152	0.7028	90.17
319	323	4.849	0.7229	87.29
322	323	4.562	0.743	84.4
325	323	4.291	0.763	81.52
328	323	4.033	0.7831	78.64

Table 6.13: Effect of ambient temperature (T_0) on the different parameters for R-507a

(at T_C = 313K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_W$ = 0.58 kg/s)

T_0 (K)	T_C (K)	EDR	$\eta_{exergetic}$	ED_{total}
298	313	1.576	0.6122	15.46
301	313	1.55	0.6405	15.91
304	313	1.526	0.6687	16.35
307	313	1.504	0.697	16.8
310	313	1.484	0.7252	17.24
313	313	1.465	0.7535	17.69
316	313	1.447	0.7818	18.13
319	313	1.431	0.81	18.58
322	313	1.416	0.8383	19.02
325	313	1.402	0.8665	19.47
328	313	1.389	0.8948	19.91

Table 6.14: Effect of ambient temperature (T_0) on the different parameters for R-507a (at T_C = 323K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_w$ = 0.6 kg/s)

T_0 (K)	T_C (K)	EDR	$\eta_{exergetic}$	ED_{total}
298	323	2.099	0.4994	20.59
301	323	2.033	0.5224	20.87
304	323	1.973	0.5455	21.14
307	323	1.918	0.5685	21.42
310	323	1.867	0.5916	21.7
313	323	1.82	0.6146	21.97
316	323	1.776	0.6377	22.25
319	323	1.735	0.6607	22.52
322	323	1.697	0.6837	22.8
325	323	1.662	0.7068	23.08
328	323	1.629	0.7298	23.35

Table 6.15: Effect of ambient temperature (T_0) on the different parameters for R-134a (at T_C = 313K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_w$ = 0.55 kg/s)

T_0 (K)	T_C (K)	EDR	$\eta_{exergetic}$	ED_{total}
298	313	1.454	0.663	14.27
301	313	1.432	0.6936	14.69
304	313	1.411	0.7242	15.12
307	313	1.392	0.7548	15.54
310	313	1.374	0.7854	15.97
313	313	1.358	0.816	16.4
316	313	1.343	0.8466	16.82
319	313	1.329	0.8772	17.25
322	313	1.316	0.9078	17.67
325	313	1.303	0.9384	18.1
328	313	1.292	0.969	18.52

Table 6.16: Effect of ambient temperature (T_0) on the different parameters for R-134a (at T_C = 323K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_w$ = 0.27 kg/s)

T_0 (K)	T_C (K)	EDR	$\eta_{exergetic}$	ED_{total}
298	323	1.884	0.5625	18.48
301	323	1.825	0.5885	18.73
304	323	1.771	0.6144	18.98
307	323	1.722	0.6404	19.23
310	323	1.676	0.6663	19.48
313	323	1.634	0.6923	19.73
316	323	1.595	0.7183	19.98
319	323	1.559	0.7442	20.23
322	323	1.525	0.7702	20.49
325	323	1.493	0.7961	20.74
328	323	1.464	0.8221	20.99

Table 6.17: Effect of ambient temperature (T_0) on the different parameters for R-152a (at T_C = 313K, ε_{AuxC} = 0.85, η_{comp} = 0.9, $\dot{m}_w$ = 0.4 kg/s)

T_0 (K)	T_C (K)	EDR	$\eta_{exergetic}$	ED_{total}
298	313	1.347	0.6844	13.22
301	313	1.33	0.716	13.65
304	313	1.315	0.7475	14.09
307	313	1.3	0.7791	14.52
310	313	1.287	0.8107	14.96
313	313	1.275	0.8423	15.39
316	313	1.263	0.8739	15.82
319	313	1.253	0.9055	16.26
322	313	1.243	0.9371	16.69
325	313	1.233	0.9686	17.13
328	313	1.225	1	17.56

Table 6.18: Effect of ambient temperature (T_0) on the different parameters for R-152a

(at T_C = 323K, ε_{AuxC} = 0.85,η_{comp} = 0.9, $\dot{m}_W$ = 0.45 kg/s)

T_0 (K)	T_C (K)	EDR	$\eta_{exergetic}$	ED_{total}
298	323	1.793	0.5877	17.59
301	323	1.74	0.6149	17.86
304	323	1.692	0.642	18.13
307	323	1.647	0.6691	18.4
310	323	1.606	0.6962	18.66
313	323	1.568	0.7234	18.93
316	323	1.533	0.7505	19.2
319	323	1.5	0.7776	19.47
322	323	1.469	0.8047	19.74
325	323	1.441	0.8319	20.01
328	323	1.414	0.859	20.28

Table 6.19: Effect of ambient temperature (T_0) on the different parameters for R-1234yf

(at T_C = 313K, ε_{AuxC} = 0.85,η_{comp} = 0.9, $\dot{m}_W$ = 0.48 kg/s)

T_0 (K)	T_C (K)	EDR	$\eta_{exergetic}$	ED_{total}
298	313	1.625	0.6359	19.21
301	313	1.601	0.6614	19.7
304	313	1.58	0.6868	20.18
307	313	1.56	0.7122	20.66
310	313	1.541	0.7377	21.14
313	313	1.524	0.7631	21.62
316	313	1.507	0.7886	22.11
319	313	1.492	0.814	22.59
322	313	1.478	0.8394	23.07
325	313	1.464	0.8649	23.55
328	313	1.451	0.8903	24.03

Table 6.20: Effect of ambient temperature (T_0) on the different parameters for R-1234yf

(at T_C = 323K, ε_{AuxC} = 0.85,η_{comp} = 0.9, $\dot{m}_w$ = 0.5 kg/s)

T_0 (K)	T_C (K)	EDR	$\eta_{exergetic}$	ED_{total}
298	323	1.987	0.5267	28.81
301	323	1.95	0.5449	29.25
304	323	1.916	0.563	29.7
307	323	1.884	0.5812	30.15
310	323	1.854	0.5993	30.6
313	323	1.826	0.6175	31.04
316	323	1.8	0.6357	31.49
319	323	1.774	0.6538	31.94
322	323	1.751	0.672	32.39
325	323	1.728	0.6902	32.84
328	323	1.707	0.7083	33.28

6.1.3. Variation of mass flow rate of water $\dot{m}_w$

Table 6.21: Effect of mass flow rate of water ($\dot{m}_w$) on the different parameters for R-717

(at T_C = 313K, ε_{AuxC} = 0.85,η_{comp} = 0.9, T_{W1} = 283K)

$\dot{m}_w$	T_{w1}	Q_{AuxC}	Q_{cond}	T_{w2}	T_{w3}	$T_{rise,AuxC}$	$T_{rise,cond}$	FR	HRF
0.18	283	19.83	104.6	353.2	379.5	26.36	70.15	0.3757	1.376
0.2076	283	29.75	104.6	343.8	378.1	34.28	60.83	0.5636	1.564
0.2394	283	41.19	104.6	335.7	376.9	41.16	52.74	0.7804	1.78
0.2762	283	54.39	104.6	328.7	375.8	47.12	45.73	1.03	2.03
0.3185	283	69.61	104.6	322.6	374.9	52.29	39.65	1.319	2.319
0.3673	283	87.17	104.6	317.4	374.1	56.77	34.38	1.651	2.651
0.4237	283	107.4	104.6	312.8	373.5	60.65	29.81	2.035	3.035
0.4886	283	130.8	104.6	308.8	372.9	64.02	25.84	2.477	3.477
0.5636	283	157.7	104.6	305.4	372.3	66.94	22.41	2.988	3.988
0.65	283	188.8	104.6	302.4	371.9	69.47	19.43	3.576	4.576

Table 6.22: Effect of mass flow rate of water ($\dot{m}_w$) on the different parameters for R-717

(at T_C = 323K, ε_{AuxC} = 0.85, η_{comp} = 0.9, T_{W1} = 293K)

$\dot{m}_w$	T_{w1}	Q_{AuxC}	Q_{cond}	T_{w2}	T_{w3}	$T_{rise,AuxC}$	$T_{rise,cond}$	FR	HRF
0.18	293	21.32	109.5	366.5	394.8	28.34	73.47	0.3857	1.386
0.2322	293	41.14	109.5	350	392.3	42.38	56.95	0.7442	1.744
0.2844	293	60.96	109.4	339.5	390.8	51.27	46.5	1.103	2.103
0.3367	293	80.78	109.4	332.3	389.7	57.4	39.28	1.461	2.461
0.3889	293	100.6	109.4	327	388.9	61.89	34.01	1.82	2.82
0.4411	293	120.4	109.4	323	388.3	65.31	29.98	2.178	3.178
0.4933	293	140.2	109.5	319.8	387.8	68.01	26.81	2.537	3.537
0.5456	293	160.1	109.5	317.2	387.4	70.19	24.24	2.895	3.895
0.5978	293	179.9	109.5	315.1	387.1	71.99	22.12	3.254	4.254
0.65	293	199.7	109.5	313.3	386.8	73.5	20.35	3.612	4.612

Table 6.23: Effect of mass flow rate of water ($\dot{m}_w$) on the different parameters for R-507a

(at T_C = 313K, ε_{AuxC} = 0.85, η_{comp} = 0.9, T_{W1} = 283K)

$\dot{m}_w$	T_{w1}	Q_{AuxC}	Q_{cond}	T_{w2}	T_{w3}	$T_{rise,AuxC}$	$T_{rise,cond}$	FR	HRF
0.4	283	2.633	108.1	315.7	317.2	1.575	32.66	0.04821	1.048
0.45	283	8.765	108.1	312	316.7	4.66	29.03	0.1605	1.16
0.5	283	14.9	108.1	309.1	316.3	7.127	26.13	0.2728	1.273
0.55	283	21.03	108.1	306.8	315.9	9.147	23.76	0.385	1.385
0.6	283	27.16	108.1	304.8	315.6	10.83	21.78	0.4973	1.497
0.65	283	33.29	108.1	303.1	315.4	12.25	20.1	0.6096	1.61
0.7	283	39.42	108.1	301.7	315.1	13.47	18.66	0.7219	1.722
0.75	283	45.56	108.1	300.4	315	14.53	17.42	0.8341	1.834
0.8	283	51.69	108.1	299.3	314.8	15.46	16.33	0.9464	1.946
0.85	283	57.82	108.1	298.4	314.6	16.27	15.37	1.059	2.059

Table 6.24: Effect of mass flow rate of water ($\dot{m}_w$) on the different parameters for R-507a

(at T_C = 323K, ε_{AuxC} = 0.85, η_{comp} = 0.9, T_{w1} = 293K)

$\dot{m}_w$	T_{w1}	Q_{AuxC}	Q_{cond}	T_{w2}	T_{w3}	$T_{rise,AuxC}$	$T_{rise,cond}$	FR	HRF
0.4	293	0.01538	116.3	0.009199	35.21	328.2	328.2	0.0002613	1
0.45	293	6.272	116.3	3.334	31.3	324.3	327.6	0.1065	1.107
0.5	293	12.53	116.3	5.995	28.17	321.2	327.2	0.2128	1.213
0.55	293	18.79	116.3	8.171	25.61	318.6	326.8	0.3191	1.319
0.6	293	25.04	116.3	9.985	23.47	316.5	326.5	0.4254	1.425
0.65	293	31.3	116.3	11.52	21.67	314.7	326.2	0.5317	1.532
0.7	293	37.56	116.3	12.84	20.12	313.1	326	0.638	1.638
0.75	293	43.81	116.3	13.98	18.78	311.8	325.8	0.7442	1.744
0.8	293	50.07	116.3	14.97	17.6	310.6	325.6	0.8505	1.851
0.85	293	56.33	116.3	15.85	16.57	309.6	325.4	0.9568	1.957

Table 6.25: Effect of mass flow rate of water ($\dot{m}_w$) on the different parameters for R-134a

(at T_C = 313K, ε_{AuxC} = 0.85, η_{comp} = 0.9, T_{w1} = 283K)

$\dot{m}_w$	T_{w1}	Q_{AuxC}	Q_{cond}	T_{w2}	T_{w3}	$T_{rise,AHE}$	$T_{rise,cond}$	FR	HRF
0.4	283	6.392	105.1	314.7	318.5	3.823	31.73	0.1205	1.12
0.45	283	12.83	105.1	311.2	318	6.819	28.2	0.2418	1.242
0.5	283	19.26	105.1	308.4	317.6	9.216	25.38	0.3631	1.363
0.55	283	25.7	105.1	306.1	317.3	11.18	23.07	0.4844	1.484
0.6	283	32.13	105.1	304.2	317	12.81	21.15	0.6057	1.606
0.65	283	38.57	105.1	302.5	316.7	14.19	19.52	0.7271	1.727
0.7	283	45	105.1	301.1	316.5	15.38	18.13	0.8484	1.848
0.75	283	51.44	105.1	299.9	316.3	16.41	16.92	0.9697	1.97
0.8	283	57.87	105.1	298.9	316.2	17.31	15.86	1.091	2.091
0.85	283	64.31	105.1	297.9	316	18.1	14.93	1.212	2.212

Table 6.26: Effect of mass flow rate of water ($\dot{m}_w$) on the different parameters for R-507a

(at T_C = 313K, ε_{AuxC} = 0.85, η_{comp} = 0.9, T_{W1} = 283K)

$\dot{m}_w$	T_{w1}	Q_{AuxC}	Q_{cond}	T_{w2}	T_{w3}	$T_{rise,AuxC}$	$T_{rise,cond}$	FR	HRF
0.5	293	16.84	110.9	319.8	327.9	8.059	26.82	0.3005	1.301
0.54	293	22	110.9	317.8	327.6	9.747	24.83	0.3925	1.393
0.58	293	27.16	110.9	316.1	327.3	11.2	23.12	0.4846	1.485
0.62	293	32.32	110.9	314.6	327.1	12.47	21.63	0.5766	1.577
0.66	293	37.48	110.9	313.3	326.9	13.58	20.32	0.6687	1.669
0.7	293	42.64	110.9	312.2	326.7	14.57	19.16	0.7607	1.761
0.74	293	47.8	110.9	311.1	326.6	15.45	18.12	0.8527	1.853
0.78	293	52.95	110.9	310.2	326.4	16.24	17.19	0.9448	1.945
0.82	293	58.11	110.9	309.4	326.3	16.95	16.35	1.037	2.037
0.86	293	63.27	110.9	308.6	326.2	17.6	15.59	1.129	2.129

Table 6.27: Effect of mass flow rate of water ($\dot{m}_w$) on the different parameters for R-152a

(at T_C = 313K, ε_{AuxC} = 0.85, η_{comp} = 0.9, T_{W1} = 283K)

$\dot{m}_w$	T_{w1}	Q_{AuxC}	Q_{cond}	T_{w2}	T_{w3}	$T_{rise,AuxC}$	$T_{rise,cond}$	FR	HRF
0.3	283	4.08	103.9	324.8	328.1	3.254	41.81	0.07783	1.078
0.35	283	12.19	103.9	318.8	327.2	8.331	35.84	0.2325	1.232
0.4	283	20.3	103.9	314.4	326.5	12.14	31.36	0.3871	1.387
0.45	283	28.4	103.9	310.9	326	15.1	27.87	0.5417	1.542
0.5	283	36.51	103.9	308.1	325.6	17.47	25.09	0.6964	1.696
0.55	283	44.62	103.9	305.8	325.2	19.41	22.81	0.851	1.851
0.6	283	52.73	103.9	303.9	324.9	21.02	20.9	1.006	2.006
0.65	283	60.83	103.9	302.3	324.7	22.39	19.3	1.16	2.16
0.7	283	68.94	103.9	300.9	324.5	23.56	17.92	1.315	2.315
0.75	283	77.05	103.9	299.7	324.3	24.58	16.72	1.47	2.47

Table 6.28: Effect of mass flow rate of water ($\dot{m}_w$) on the different parameters for R-152a (at T_C = 323K, ε_{AuxC} = 0.85, η_{comp} = 0.9, T_{W1} = 293K)

$\dot{m}_w$	T_{w1}	Q_{AuxC}	Q_{cond}	T_{w2}	T_{w3}	$T_{rise,AuxC}$	$T_{rise,cond}$	FR	HRF
0.4	293	18.61	109	11.13	32.93	325.9	337.1	0.3379	1.338
0.45	293	26.78	109	14.24	29.27	322.3	336.5	0.4864	1.486
0.5	293	34.96	109	16.73	26.34	319.3	336.1	0.6349	1.635
0.55	293	43.13	109	18.76	23.95	316.9	335.7	0.7834	1.783
0.6	293	51.31	109	20.46	21.95	315	335.4	0.9319	1.932
0.65	293	59.48	109	21.89	20.26	313.3	335.2	1.08	2.08
0.7	293	67.66	109	23.12	18.82	311.8	334.9	1.229	2.229
0.75	293	75.83	109	24.19	17.56	310.6	334.8	1.377	2.377
0.8	293	84.01	109	25.12	16.46	309.5	334.6	1.526	2.526
0.85	293	92.19	109	25.95	15.5	308.5	334.4	1.674	2.674

Table 6.29: Effect of mass flow rate of water ($\dot{m}_w$) on the different parameters for R-1234yf (at T_C = 313K, ε_{AuxC} = 0.85, η_{comp} = 0.9, T_{W1} = 283K)

$\dot{m}_w$	T_{w1}	Q_{AuxC}	Q_{cond}	T_{w2}	T_{w3}	$T_{rise,AuxC}$	$T_{rise,cond}$	FR	HRF
0.45	283	2.265	106.5	311.6	312.8	1.204	28.58	0.04213	1.042
0.4889	283	6.41	106.5	309.3	312.4	3.137	26.31	0.1192	1.119
0.5278	283	10.56	106.5	307.4	312.2	4.785	24.37	0.1963	1.196
0.5667	283	14.7	106.5	305.7	311.9	6.206	22.7	0.2734	1.273
0.6056	283	18.85	106.5	304.2	311.7	7.445	21.24	0.3505	1.351
0.6444	283	22.99	106.5	303	311.5	8.535	19.96	0.4276	1.428
0.6833	283	27.14	106.5	301.8	311.3	9.5	18.82	0.5047	1.505
0.7222	283	31.28	106.5	300.8	311.2	10.36	17.81	0.5818	1.582
0.7611	283	35.43	106.5	299.9	311	11.14	16.9	0.6589	1.659
0.8	283	39.57	106.5	299.1	310.9	11.83	16.08	0.736	1.736

Table 6.30: Effect of mass flow rate of water ($\dot{m}_w$**)** on the different parameters for R-1234yf (at T_c = 323K, ε_{AuxC} = 0.85,η_{comp} = 0.9, T_{w1} = 293K)

$\dot{m}_w$	T_{w1}	Q_{AuxC}	Q_{cond}	T_{w2}	T_{w3}	$T_{rise,AHE}$	$T_{rise,cond}$	FR	HRF
0.56	293	11.03	113.2	317.5	322.2	4.712	24.46	0.1927	1.193
0.5867	293	13.87	113.2	316.3	322	5.657	23.34	0.2423	1.242
0.6133	293	16.71	113.2	315.3	321.8	6.52	22.33	0.292	1.292
0.64	293	19.56	113.2	314.4	321.7	7.311	21.4	0.3416	1.342
0.6667	293	22.4	113.2	313.5	321.6	8.038	20.54	0.3913	1.391
0.6933	293	25.24	113.2	312.8	321.5	8.71	19.75	0.4409	1.441
0.72	293	28.08	113.2	312	321.4	9.332	19.02	0.4906	1.491
0.7467	293	30.93	113.2	311.3	321.3	9.909	18.34	0.5402	1.54
0.7733	293	33.77	113.2	310.7	321.2	10.45	17.71	0.5899	1.59
0.8	293	36.61	113.2	310.1	321.1	10.95	17.12	0.6395	1.64

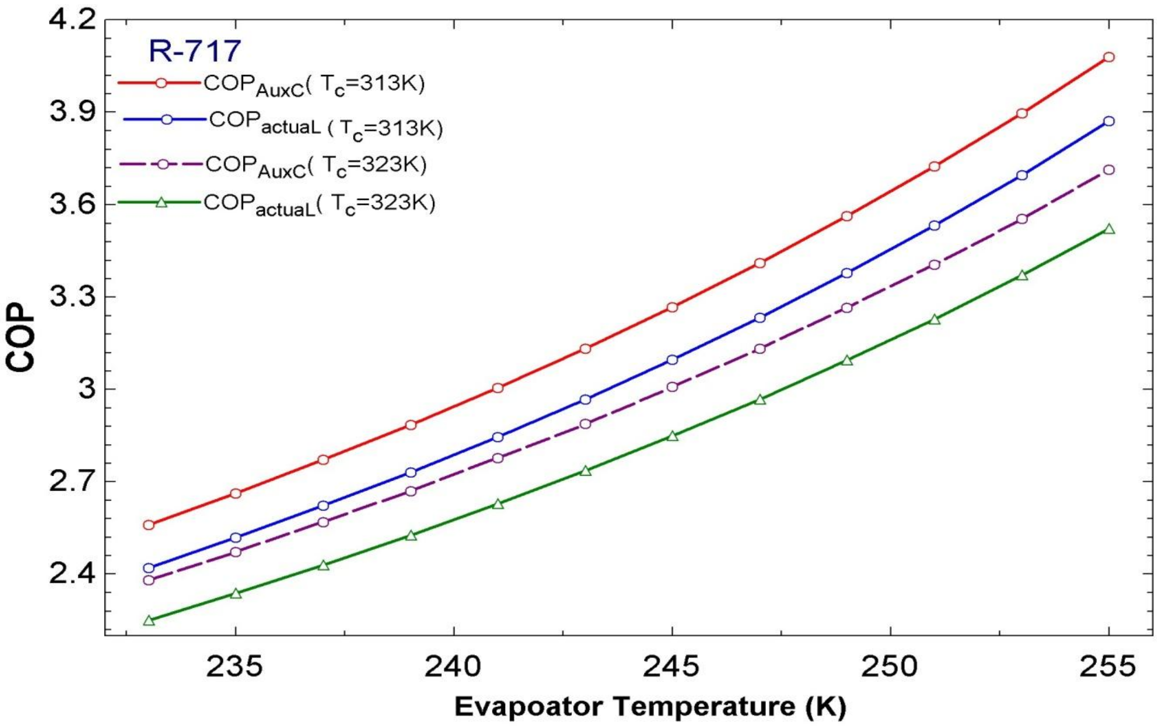

Figure 6.1 variations in COP with varying evaporator temperature (T_E), (at T_c = 313K & 323K, ε_{AuxC} = 0.85,η_{comp} = 0.9, $\dot{m}_w$ = 0.27 kg/s & 0.29 kg/s, R-717)

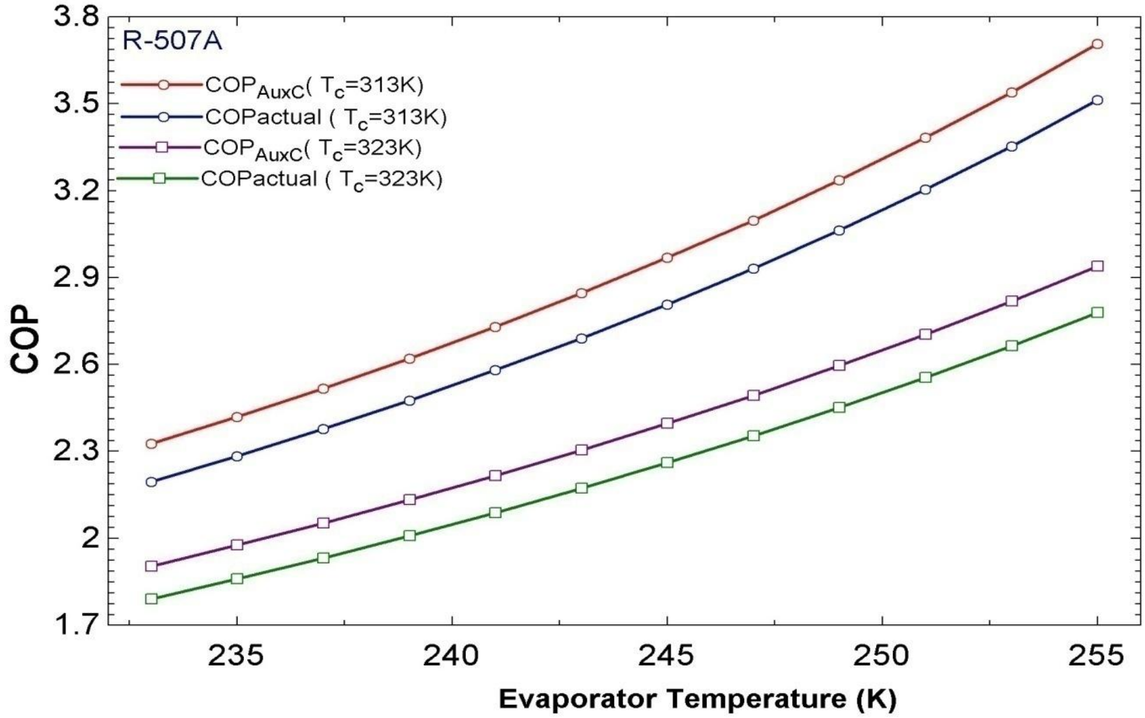

Figure 6.2 variations in COP with varying evaporator temperature (T_E), (at T_c = 313K & 323K

ε_{AuxC} = 0.85,η_{comp} = 0.9, $\dot{m}_w$ = 0.58 kg/s & 0.56 kg/s, R-507a)

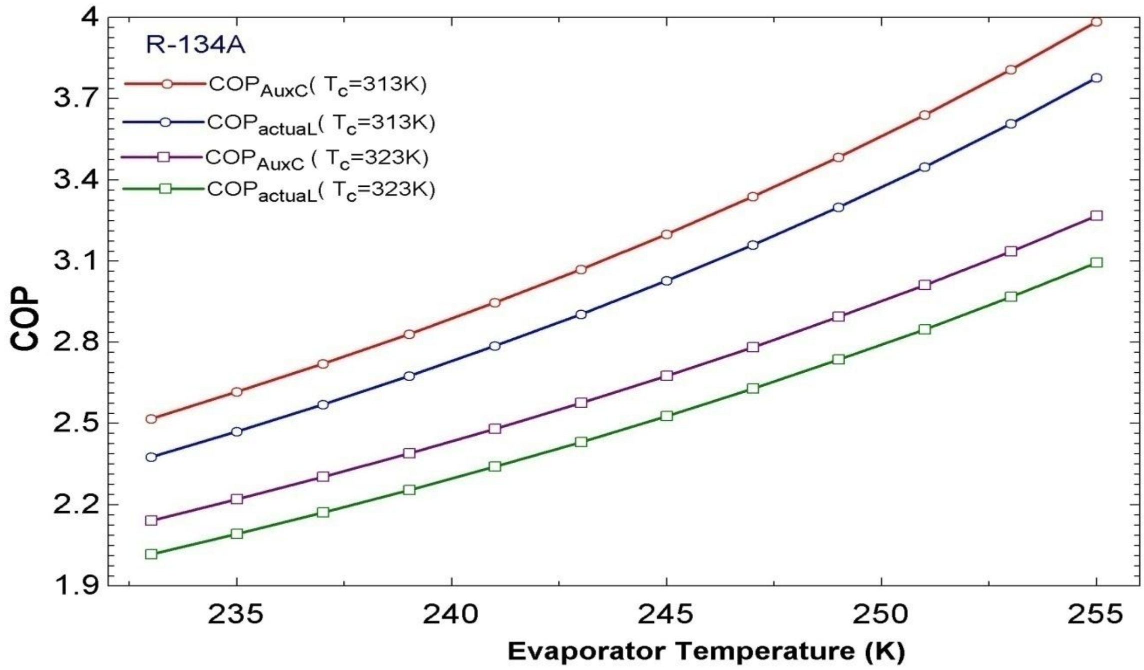

Figure 6.3 variations in COP with varying evaporator temperature (T_E), (at T_c = 313K & 323K,

ε_{AuxC} = 0.85,η_{comp} = 0.9, $\dot{m}_w$ = 0.55 kg/s & 0.58 kg/s, R-134a)

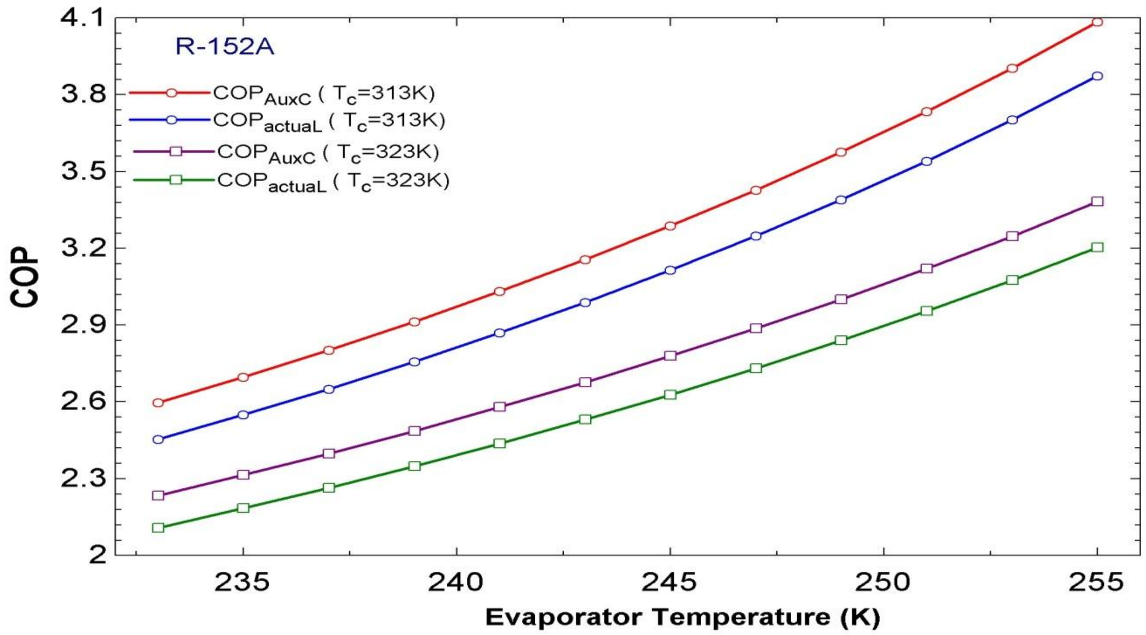

Figure 6.4 variations in COP with varying evaporator temperature (T_E). (at T_c = 313K & 323K ε_{AuxC} = 0.85,η_{comp} = 0.9, $\dot{m}_w$ = 0.4 kg/s & 0.45 kg/s, R-152a)

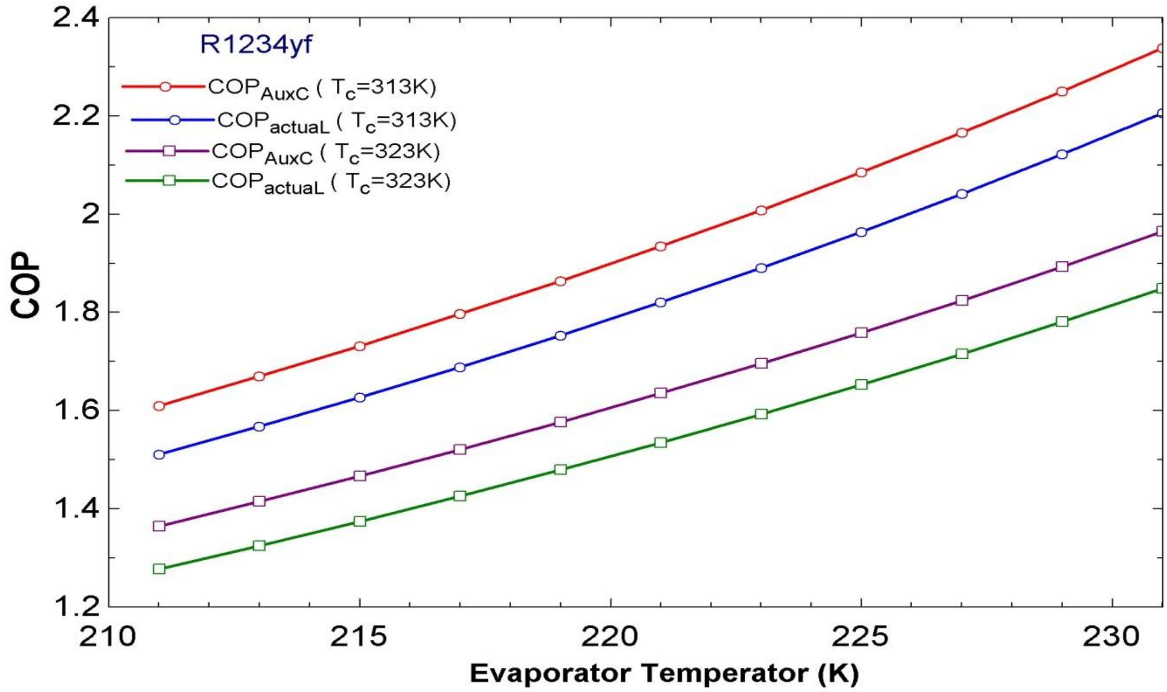

Figure 6.5 variations in COP with varying evaporator temperature (T_E), (at T_c = 313K & 323K, ε_{AuxC} = 0.85,η_{comp} = 0.9, $\dot{m}_w$ = 0.48 kg/s & 0.55 kg/s, R-1234yf)

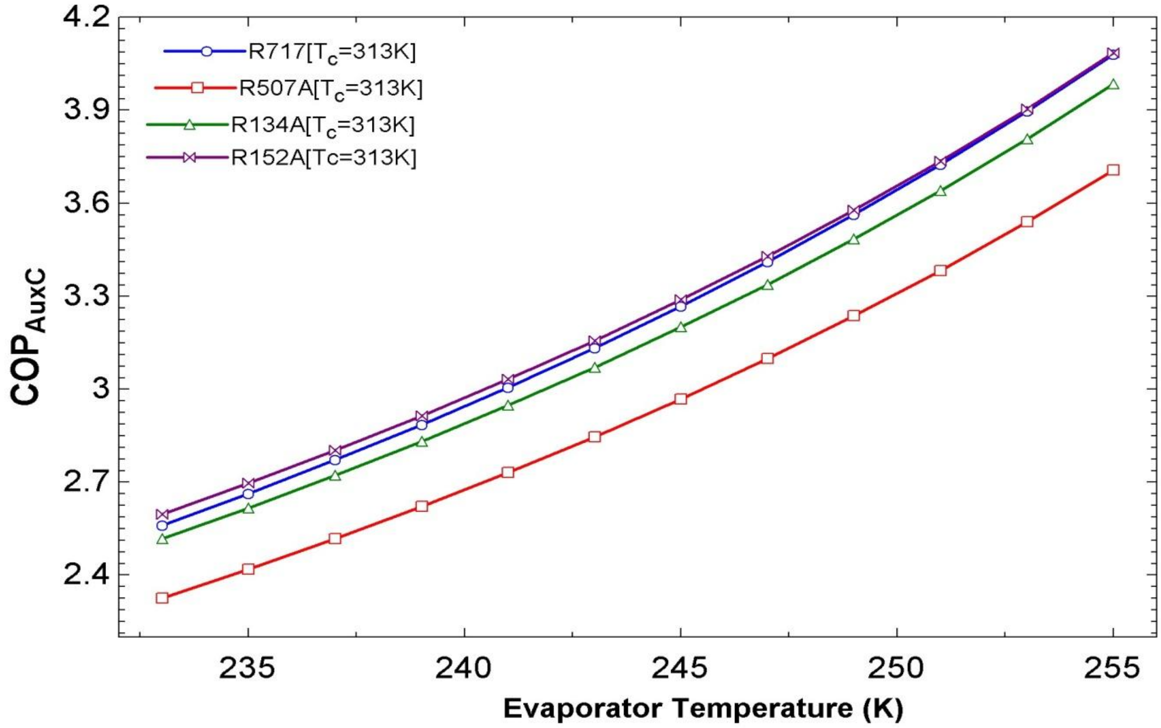

Figure 6.6 variations in COP with varying evaporator temperature (T_E), (at T_c = 313K)

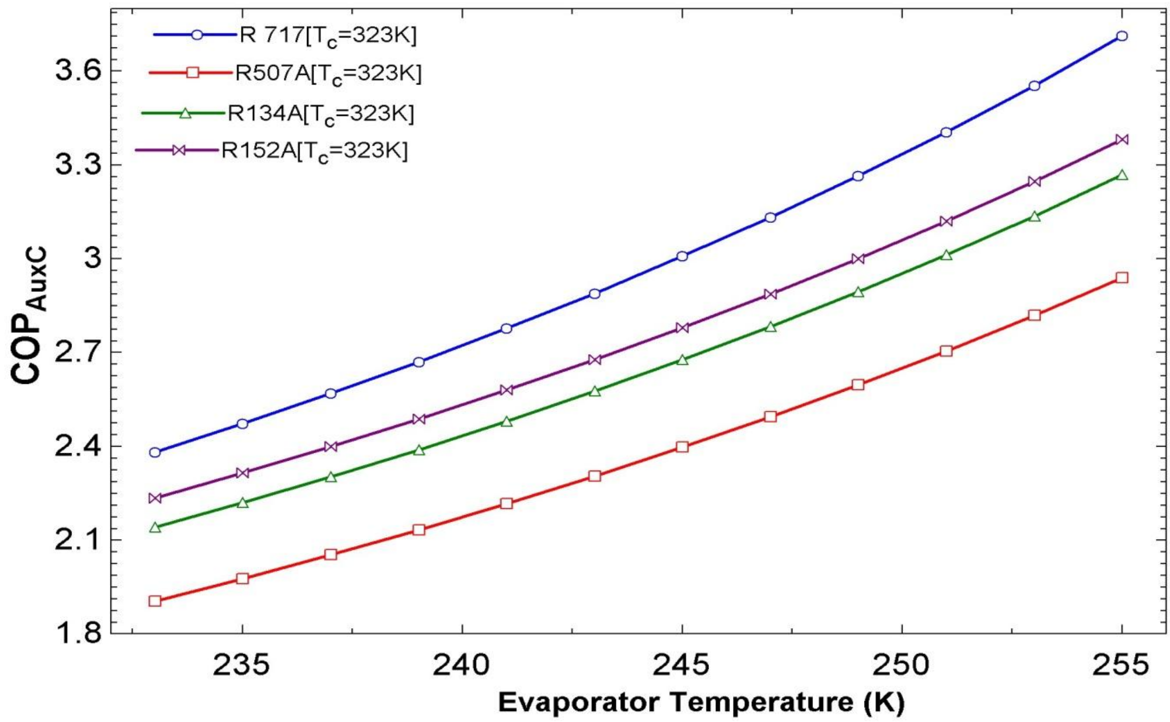

Figure 6.7 variations in COP with varying evaporator temperature (T_E), (at T_c = 323K)

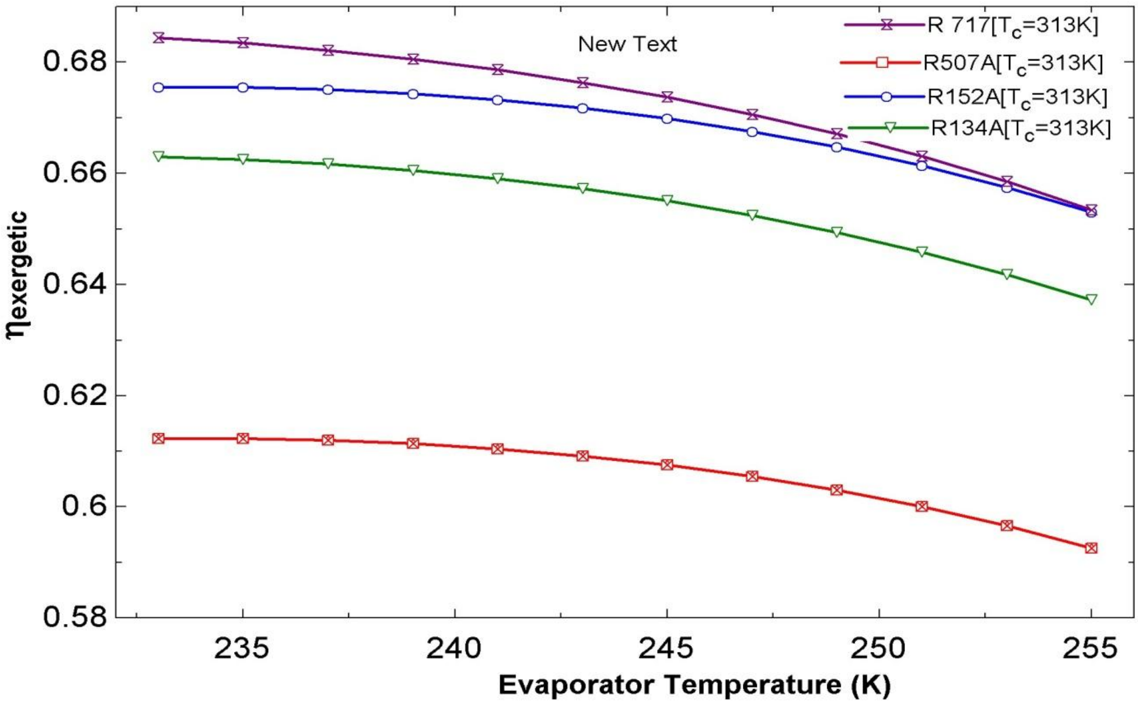

Figure 6.8 variations in exergetic efficiency ($\eta_{exergetic}$) with varying evaporator temperature (T_E).,(at T_c = 313K)

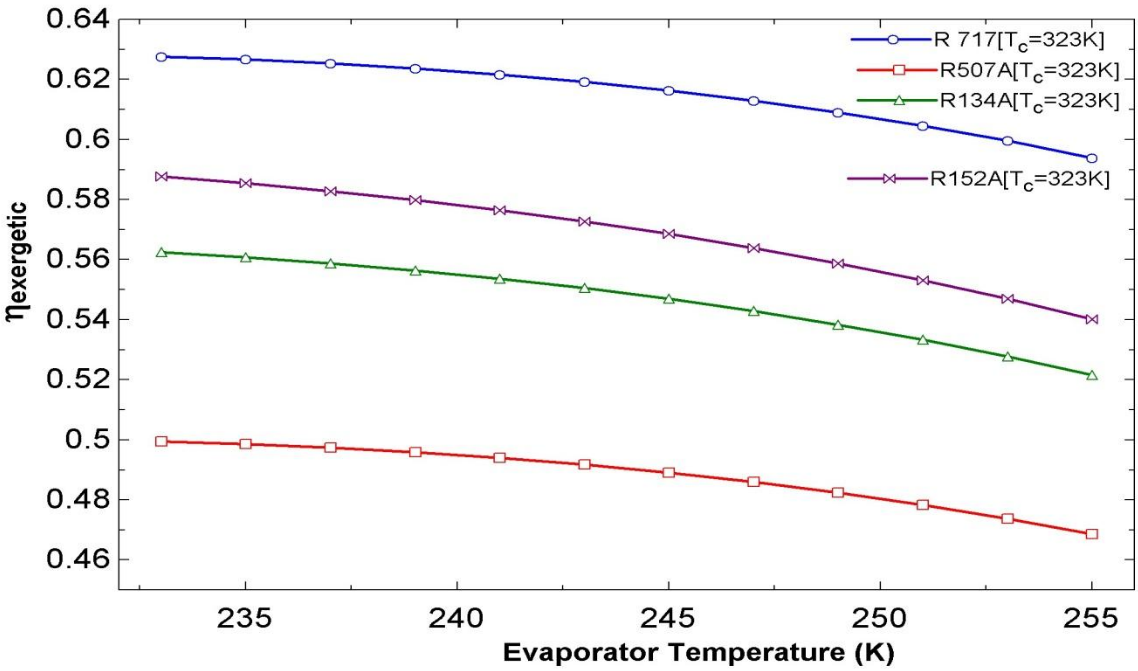

Figure 6.9 variations in exergetic efficiency ($\eta_{exergetic}$) with varying evaporator temperature (T_E)., (at T_c = 313K)

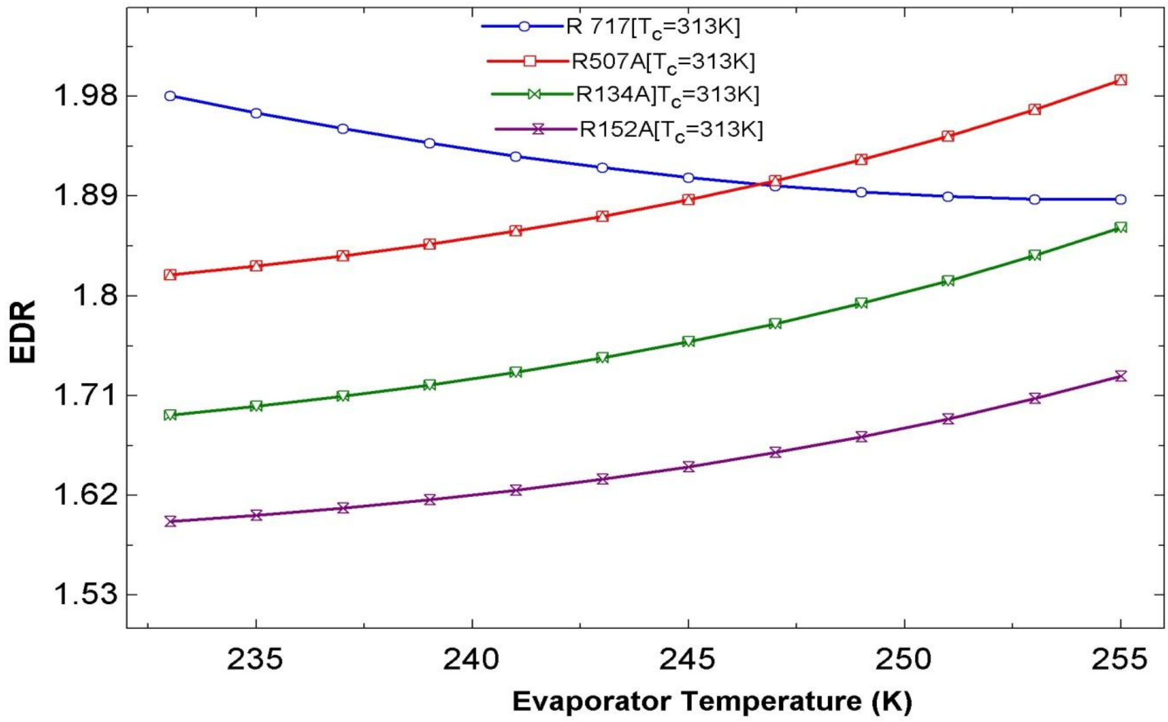

Figure 6.10 variations in Exergy destruction ratio (EDR) with varying evaporator temperature (T_E), (at T_c = 313K)

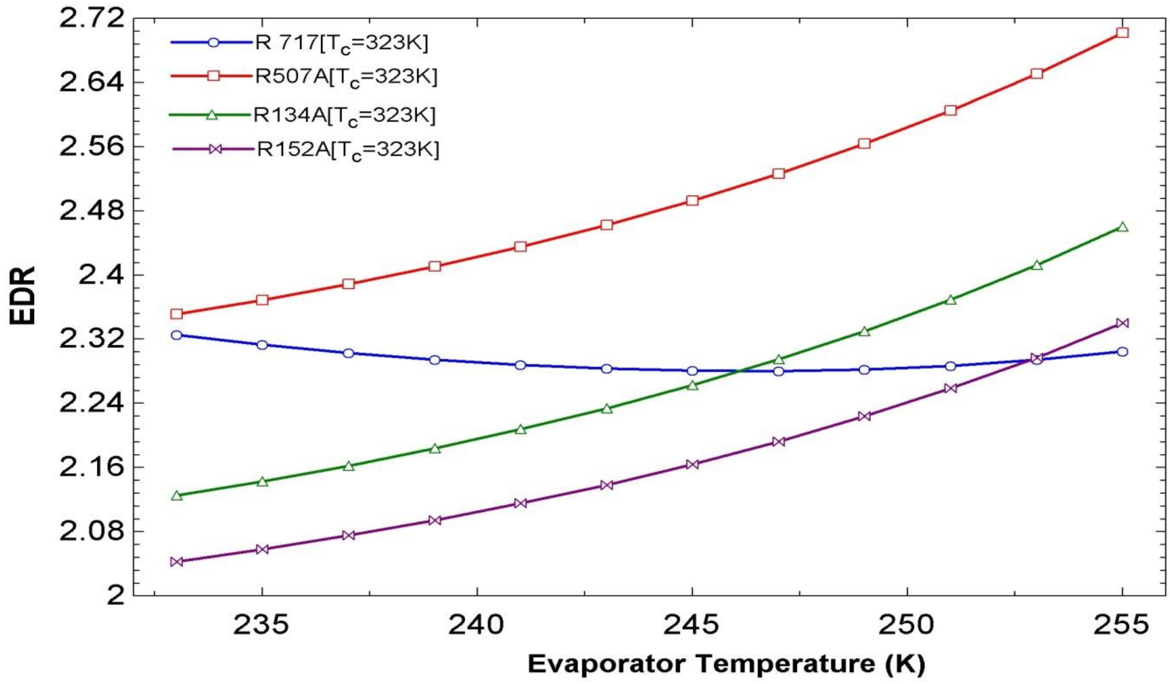

Figure 6.11 variations in Exergy destruction ratio (EDR) with varying evaporator temperature (T_E), (at T_c = 323K)

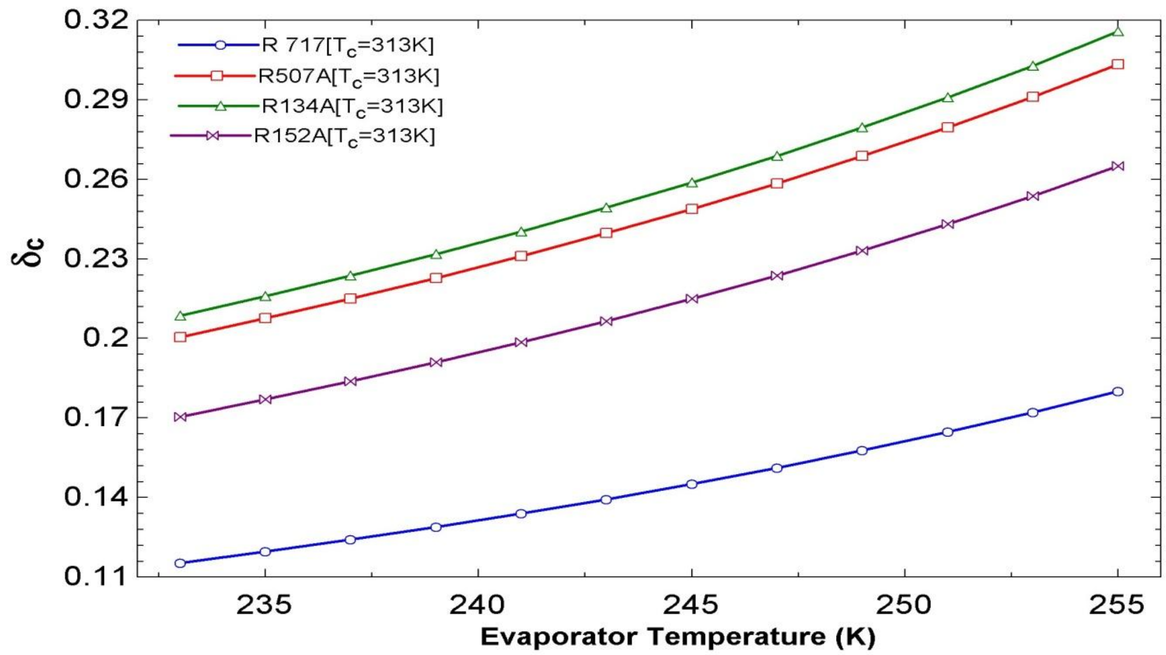

Figure 6.12 variations in efficiency defect in condenser (δ_C) with varying evaporator temperature (T_E), (at Tc=313K)

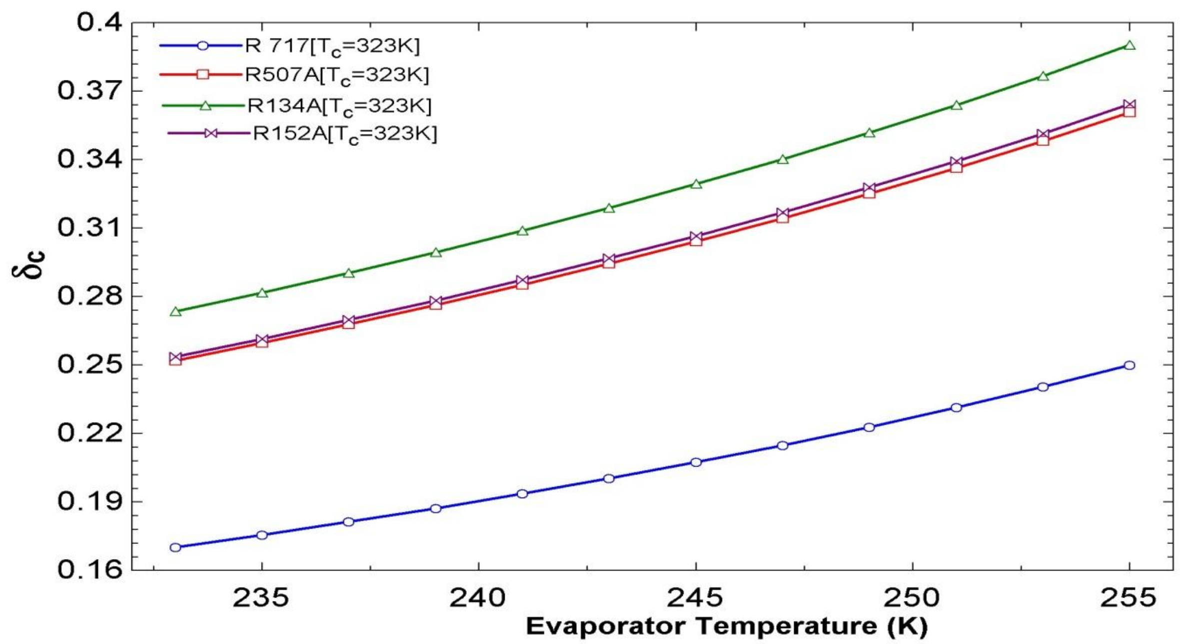

Figure 6.13 variations in efficiency defect in condenser (δ_C) with varying evaporator temperature (T_E), (at Tc=323K)

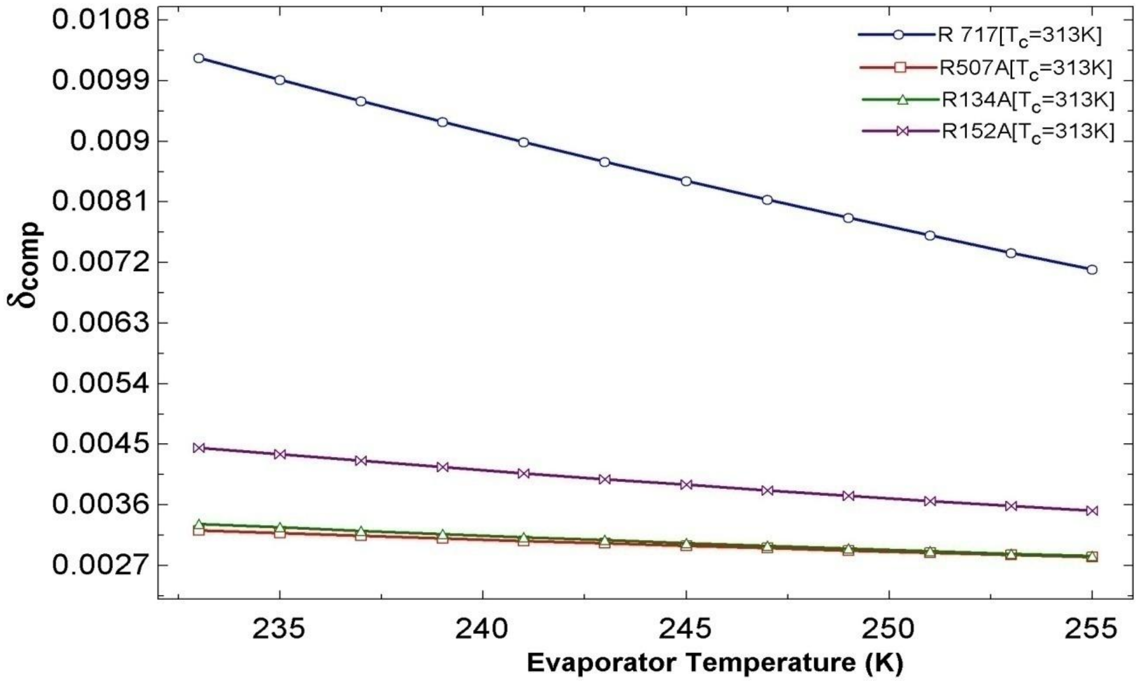

Figure 6.14 variations in efficiency defect in compressor (δ_{comp}) with varying evaporator temperature (T_E), (at Tc=313K)

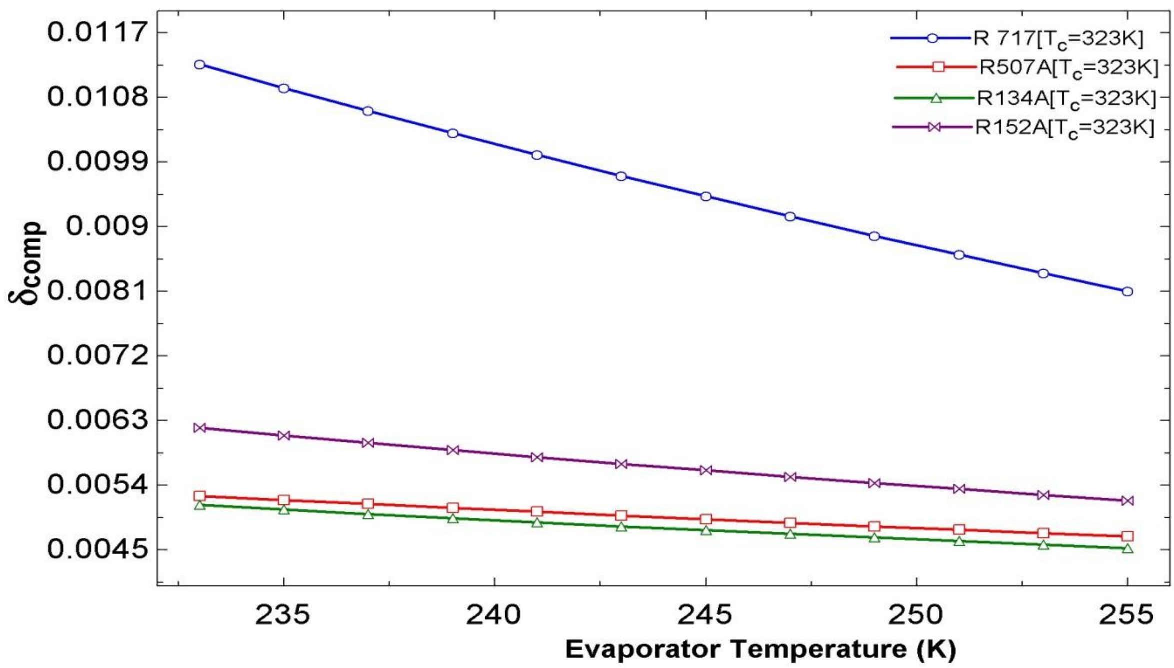

Figure 6.15 variations in efficiency defect in compressor (δ_{comp}) with varying evaporator temperature (T_E), (at Tc=323K)

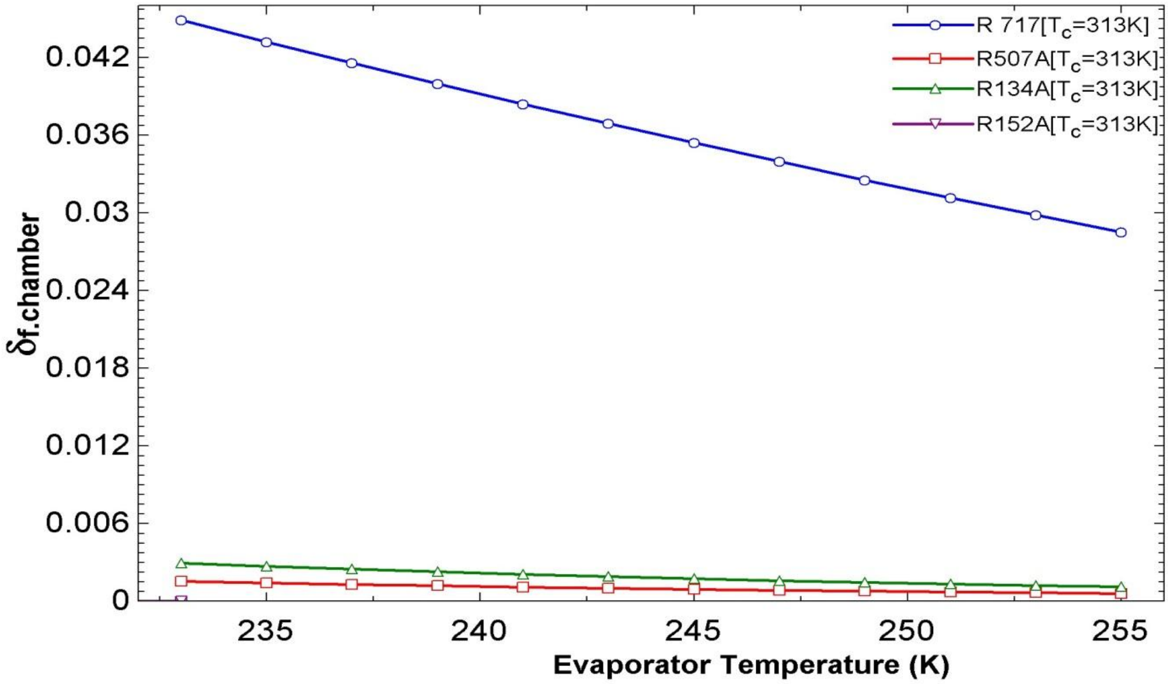

Figure 6.16 variations in efficiency defect in flash chamber ($\delta_{f.chamber}$) with varying evaporator temperature (T_E), (at Tc=313K).

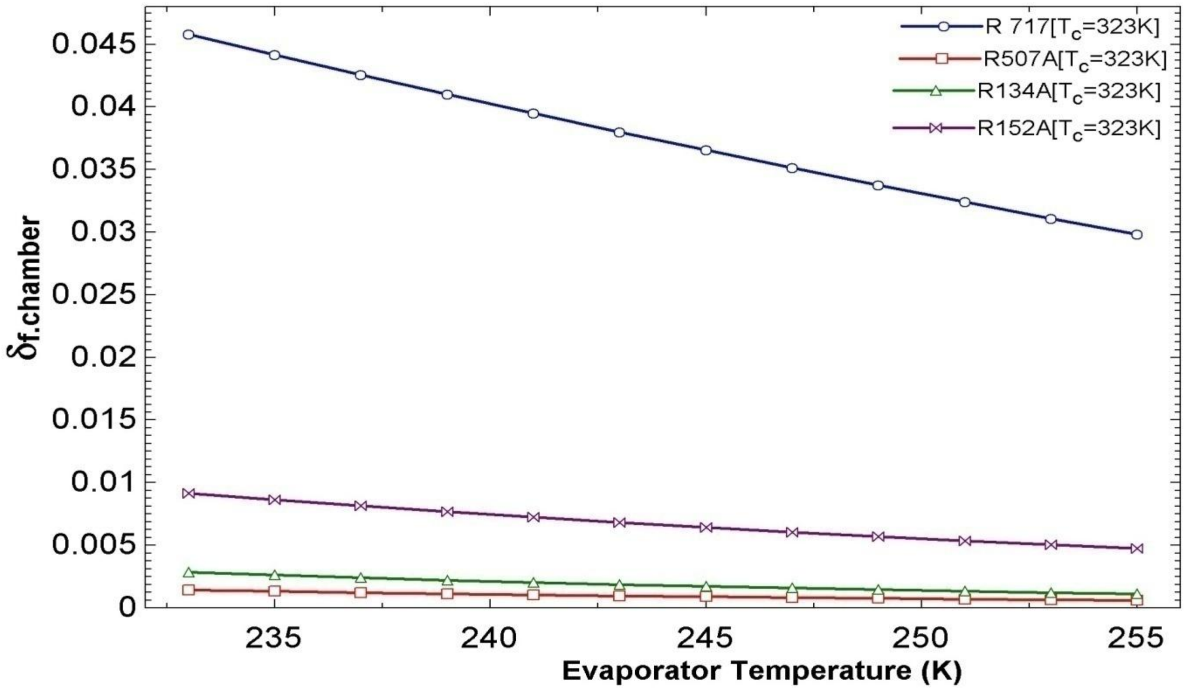

Figure 6.17 variations in efficiency defect in flash chamber ($\delta_{f.chamber}$) with varying evaporator temperature (T_E), (at Tc=323K).

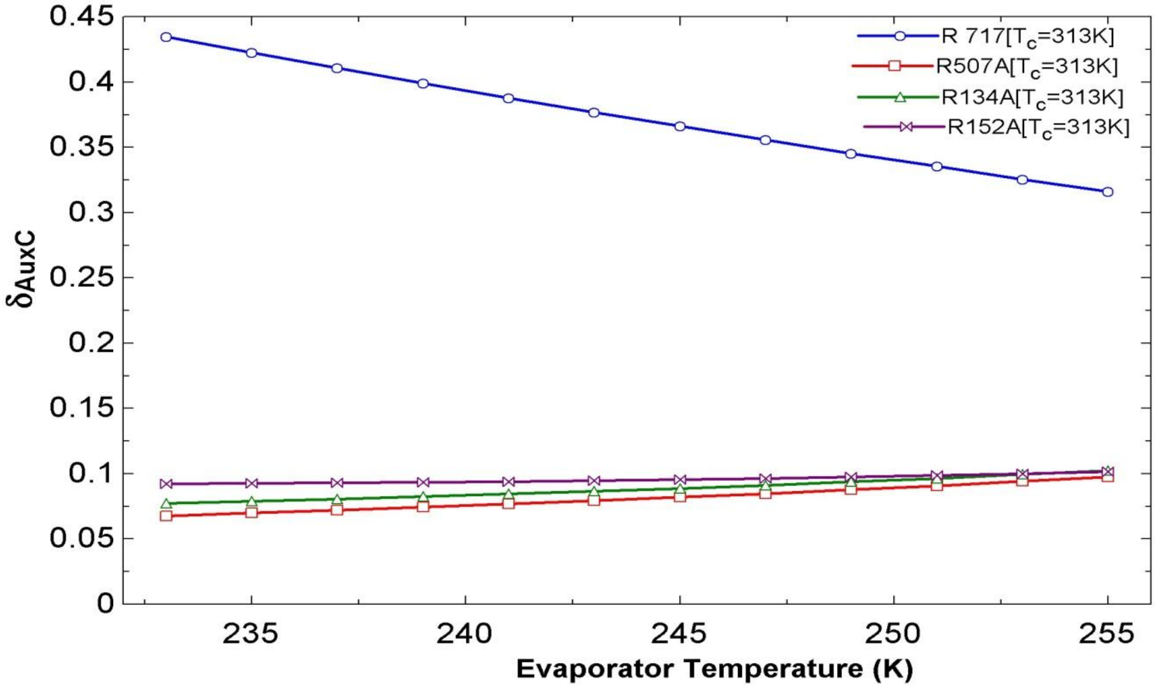

Figure 6.18 variations in efficiency defect in auxiliary condenser (δ_{AuxC}) with varying evaporator temperature (T_E), (at Tc=313K).

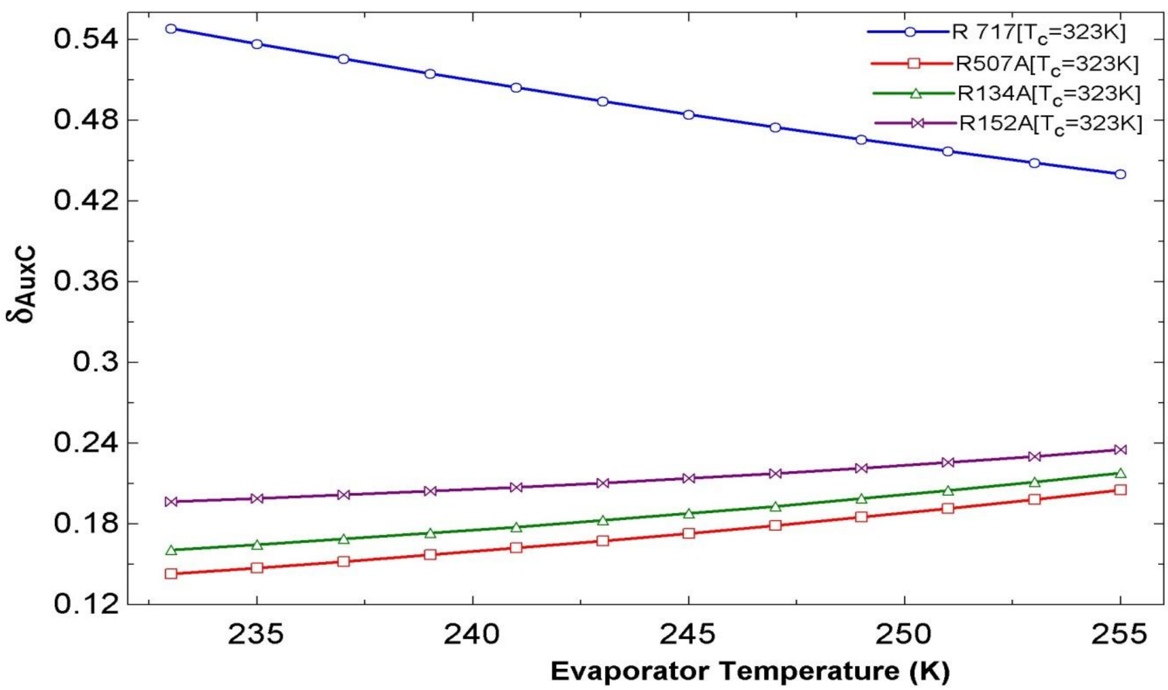

Figure 6.19 variations in efficiency defect in auxiliary condenser (δ_{AuxC}) with varying evaporator temperature (T_E), (at Tc=323K).

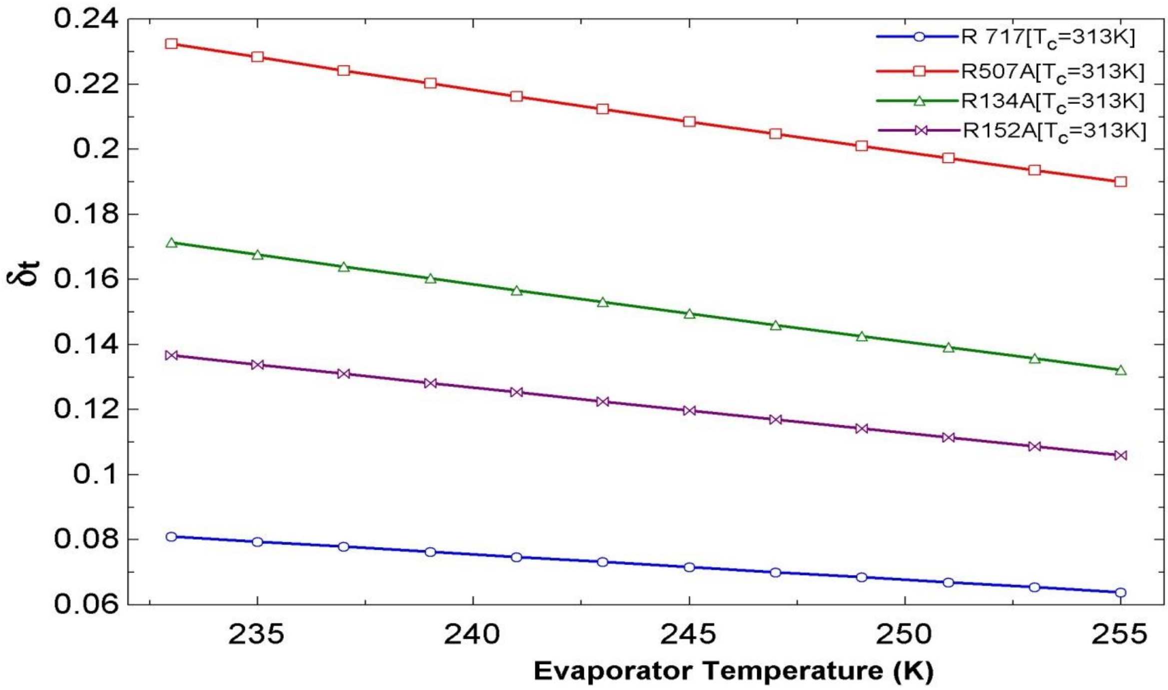

Figure 6.20 variations in efficiency defect in throttling valve (δ_t) with varying evaporator temperature (T_E), (at Tc=313K).

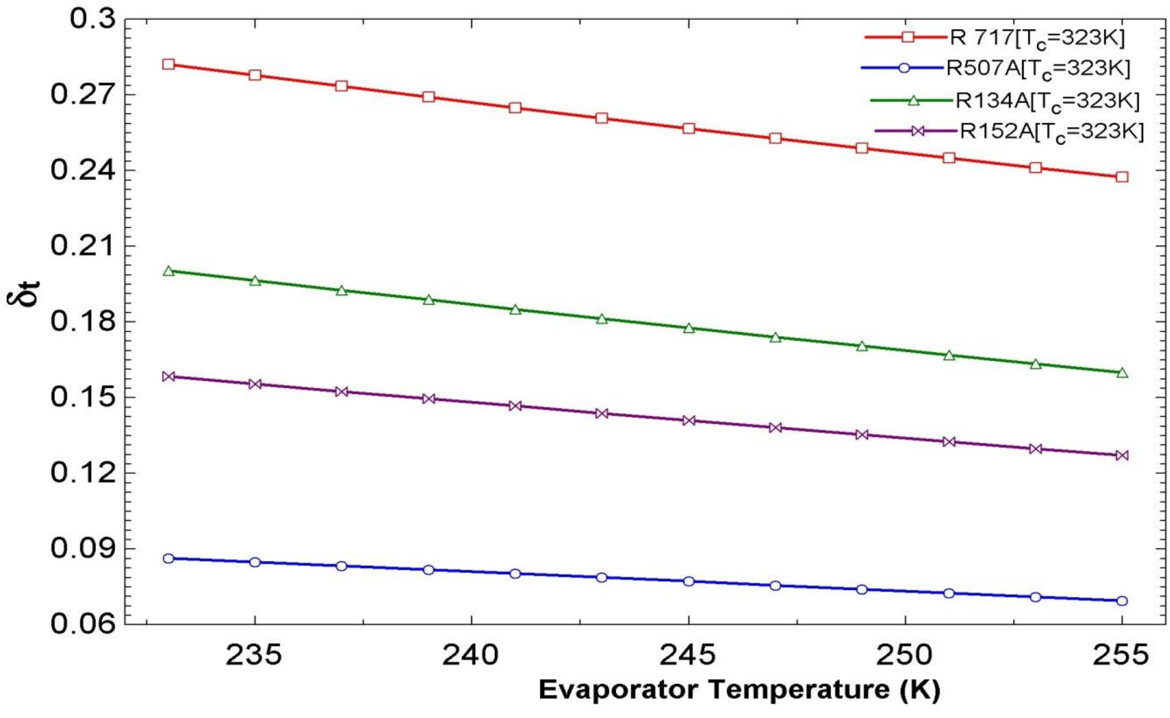

Figure 6.21 variations in efficiency defect in throttling valve (δ_t) with varying evaporator temperature (T_E), (at Tc=323K).

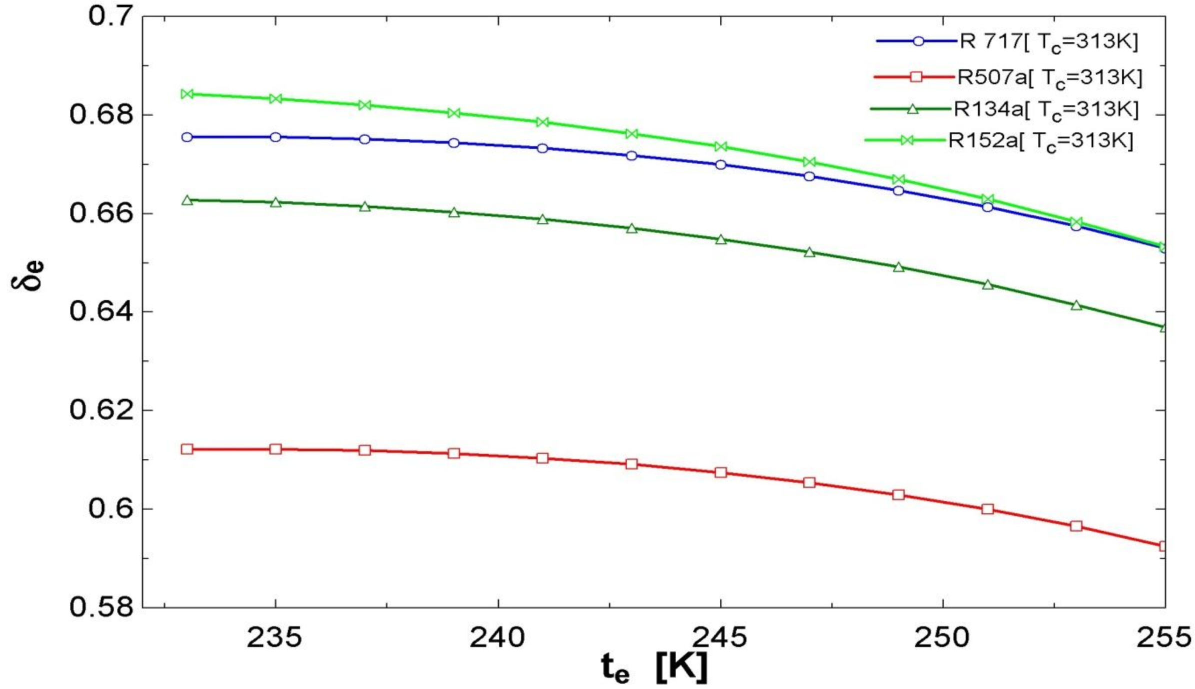

Figure 6.22 variations in efficiency defect in evaporator (δ_e) with varying evaporator temperature (T_E), (at Tc=313K).

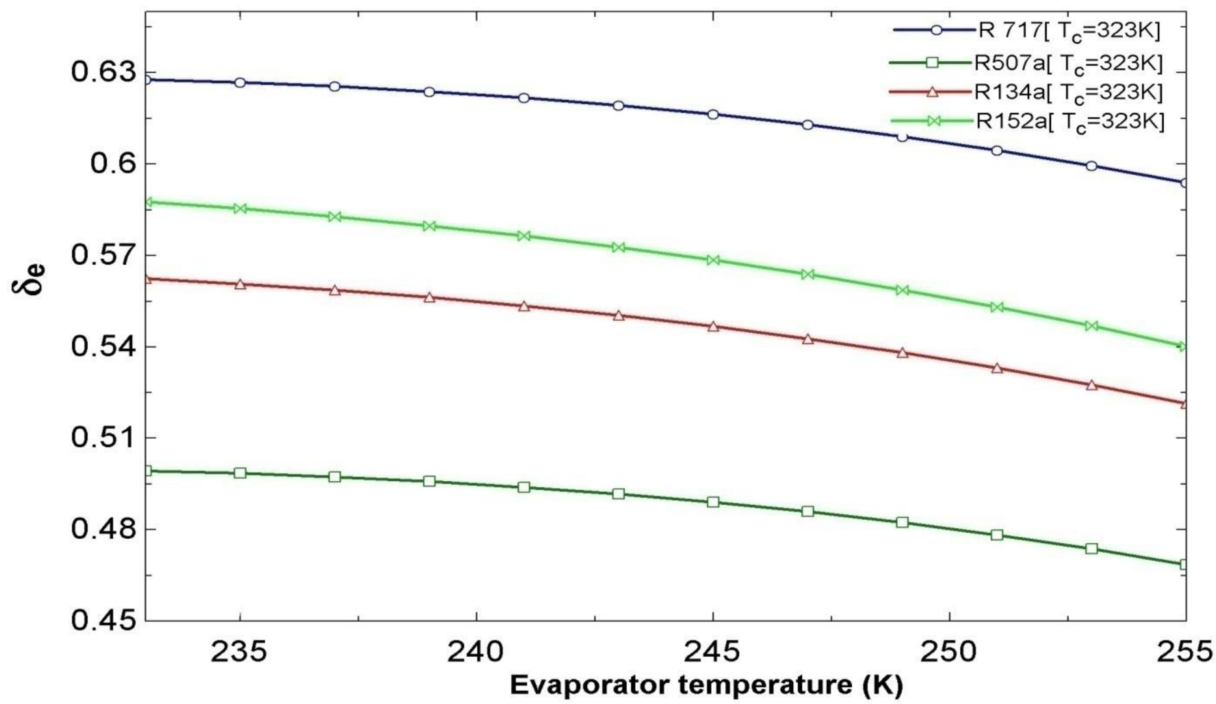

Figure 6.23 variations in efficiency defect in evaporator (δ_e) with varying evaporator temperature (T_E), (at Tc=323K).

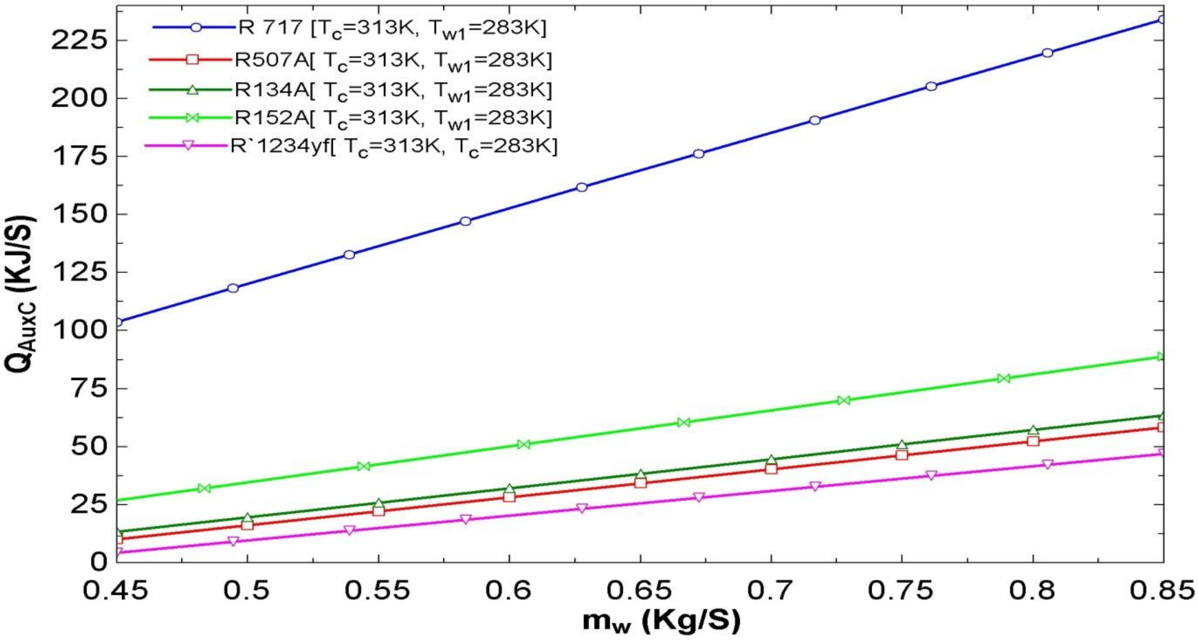

Figure 6.24 variations in Q_{AuxC} with varying mass flow rate of water ($\dot{m}_w$), (at T_c = 313K,ε_{AuxC} = 0.85,η_{comp} = 0.9,T_{w1} = 283K)

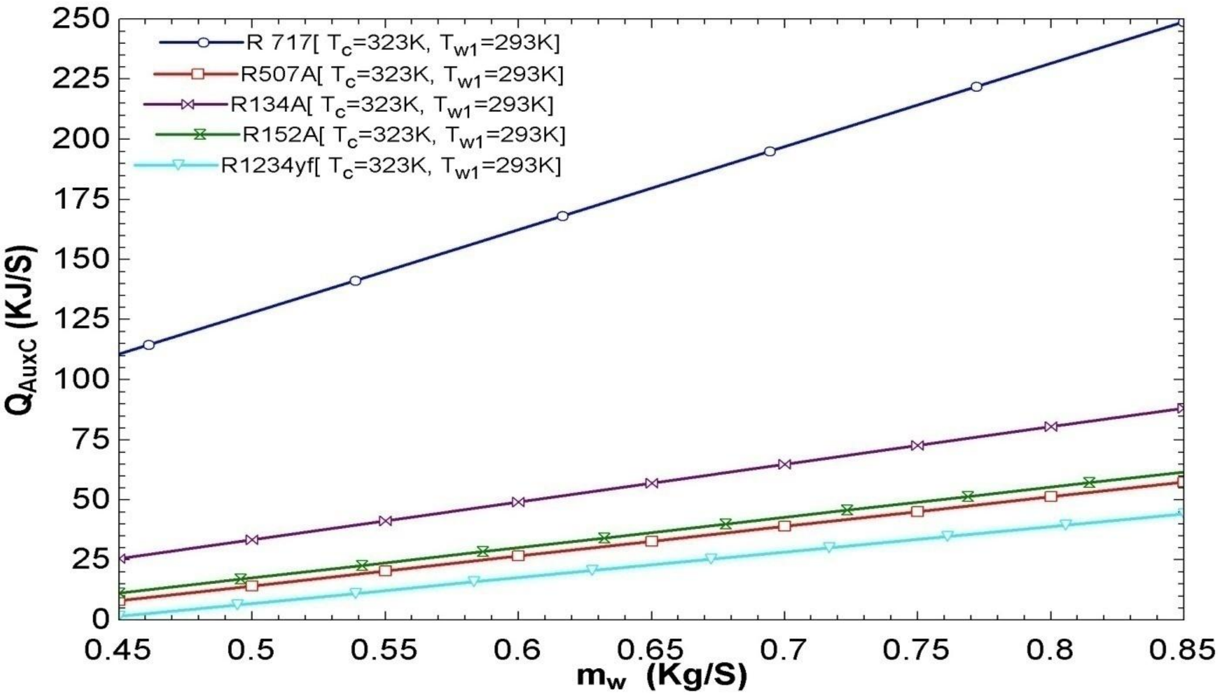

Figure 6.25 variations in Q_{AuxC} with varying mass flow rate of water ($\dot{m}_w$), (at T_c = 323K,ε_{AuxC} = 0.85,η_{comp} = 0.9,T_{w1} = 293K)

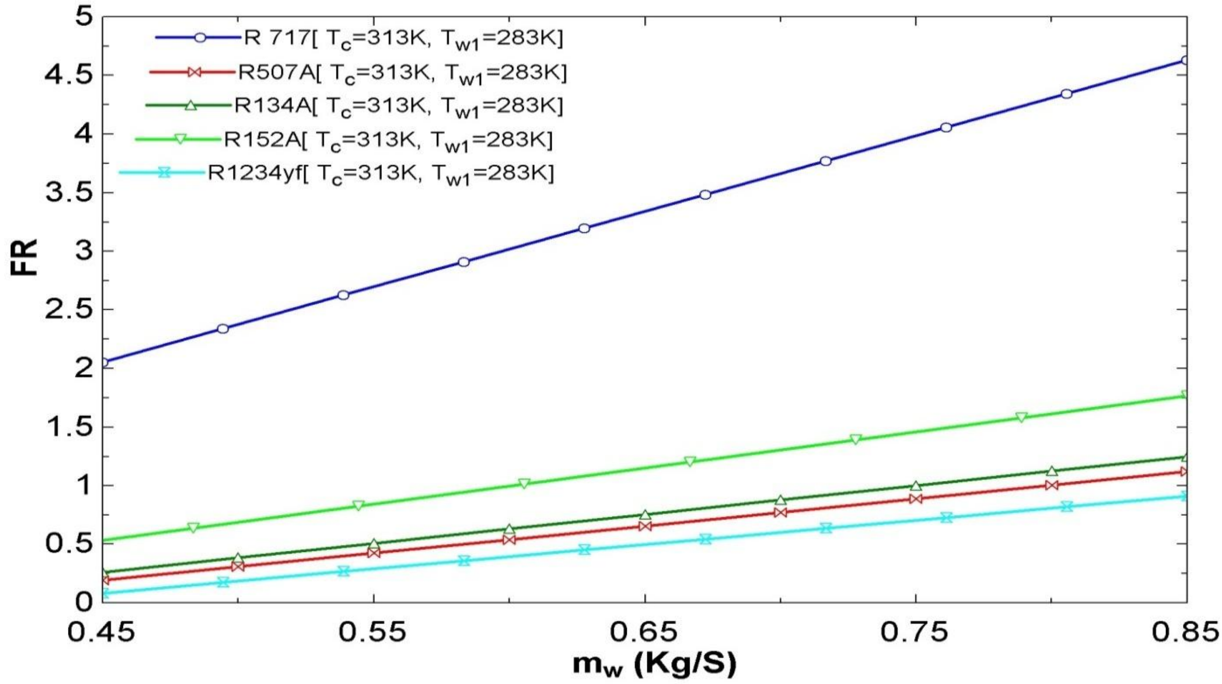

Figure 6.26 variations in FR with varying mass flow rate of water ($\dot{m}_w$), (at T_c = 313K,ε_{AuxC} = 0.85,η_{comp} = 0.9,T_{w1} = 283K).

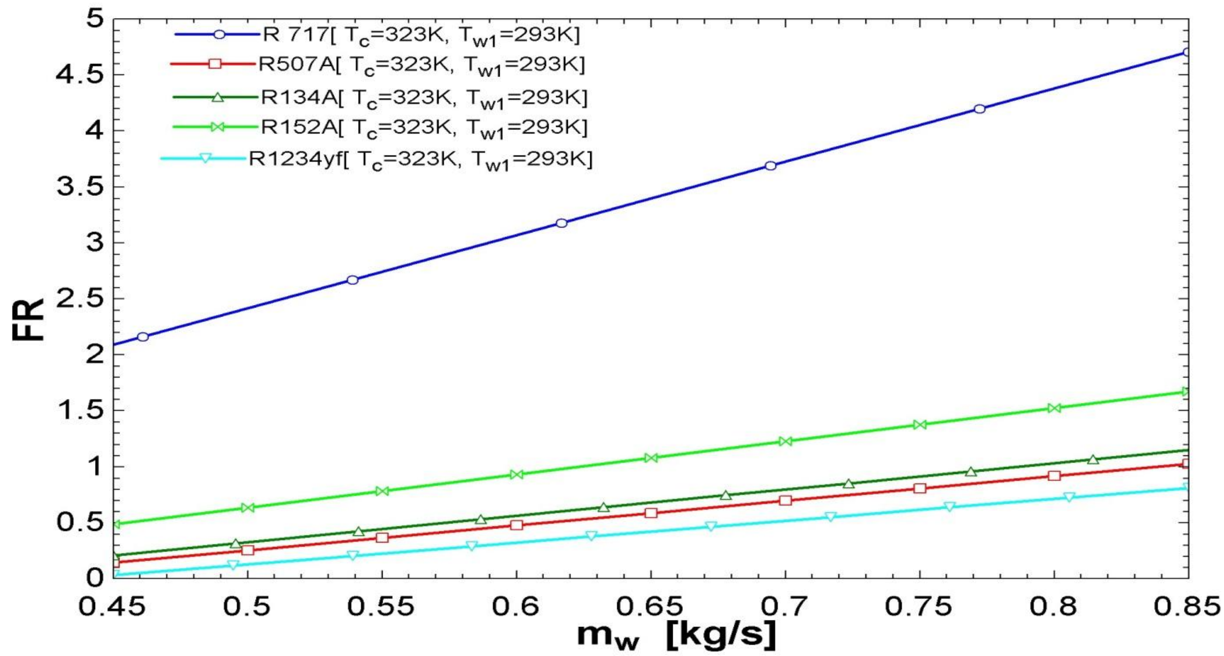

Figure 6.27 variations in FR with varying mass flow rate of water ($\dot{m}_w$), (at T_c = 323K,ε_{AuxC} = 0.85,η_{comp} = 0.9,T_{w1} = 293K).

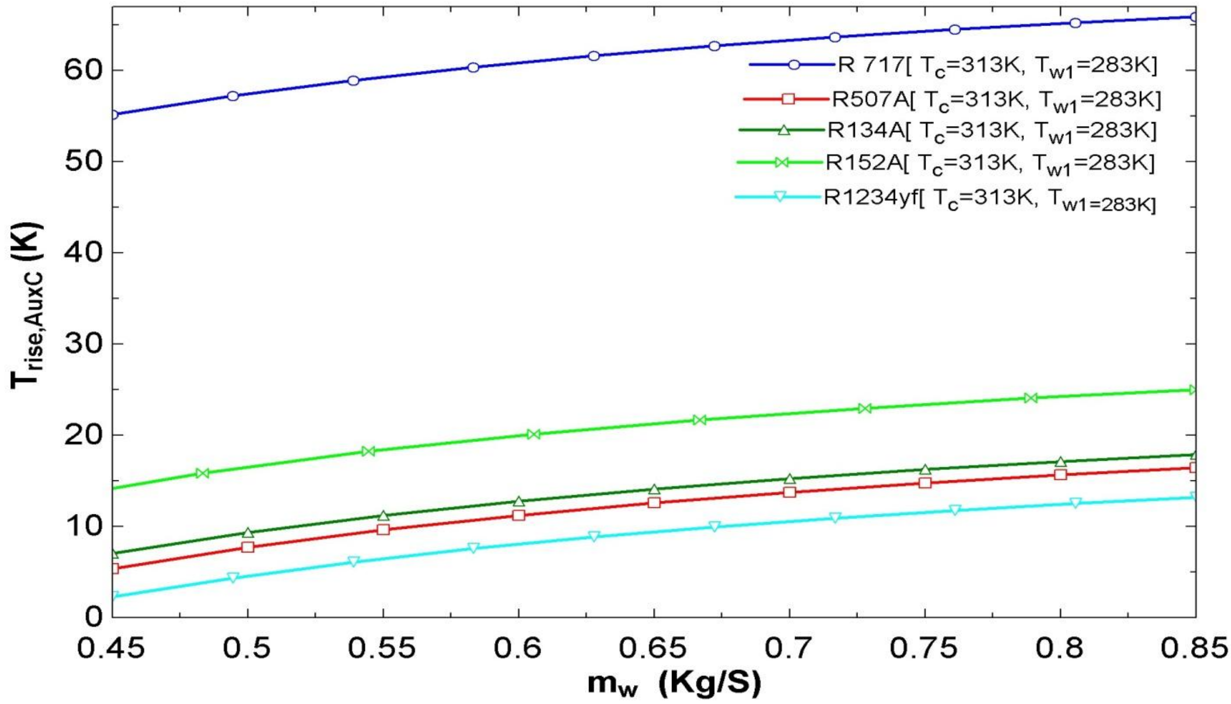

Figure 6.28 variations in rise in temperature in auxiliary condenser ($T_{rise,AuxC}$) with varying mass flow rate of water ($\dot{m}_W$), (at T_c = 313K,ε_{AuxC} = 0.85,η_{comp} = 0.9,T_{w1} = 283K).

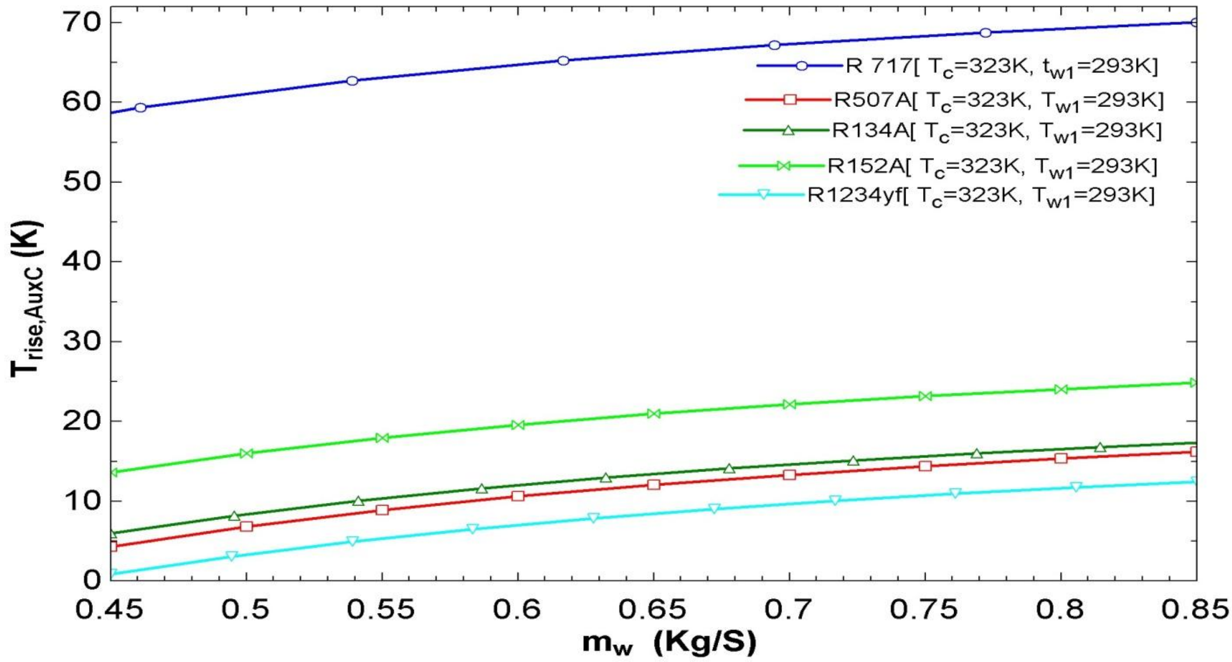

Figure 6.29 variations in rise in temperature in auxiliary condenser ($T_{rise,AuxC}$) with varying mass flow rate of water ($\dot{m}_W$), (at T_c = 323K,ε_{AuxC} = 0.85,η_{comp} = 0.9,T_{w1} = 283K).

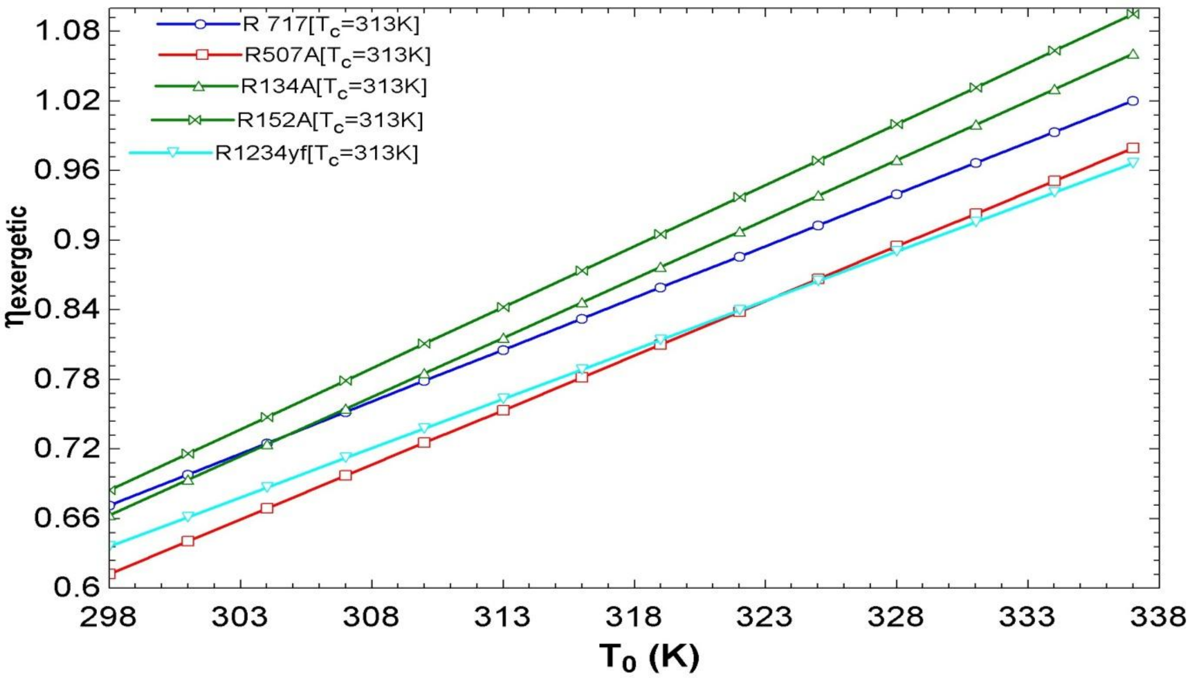

Figure 6.30 variations in exergetic efficiency ($\eta_{exergetic}$) varying with ambient temperature(T_O), (at T_c = 313K,ε_{AuxC} = 0.85,η_{comp} = 0.9,T_{w1} = 283K).

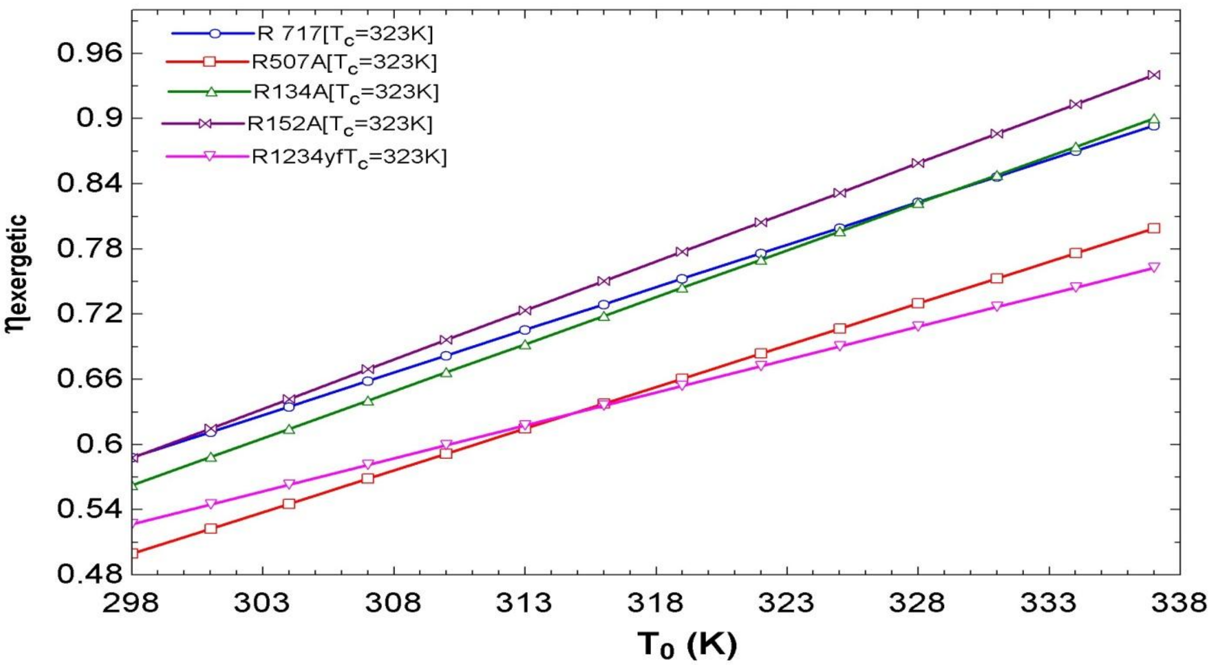

Figure 6.31 variations in exergetic efficiency ($\eta_{exergetic}$) varying with ambient temperature(T_O), (at T_c = 323K,ε_{AuxC} = 0.85,η_{comp} = 0.9,T_{w1} = 293K).

6.2 DISCUSSION

In the present study first of all observed that it is more relevant to use R-717, R-507a, R-134a, R-152a and R-1234yf as the refrigerant in modified two stage vapour compression system as value of ODP and GWP for these refrigerant is much less as compared to other refrigerants like R-407c, R-410a, R-32, R-22. From thermodynamic point of view, ammonia is also one of the best options as its heat transfer coefficient is higher than R-22, R-11, R-12 and R502 also from the point of availability, economic and environment, ammonia is better suited, but due to its toxicity and flammability it cannot be used for house hold applications.

The effect of evaporator temperature and condenser temperature on COP, heat recovery by auxiliary heat exchanger, exergetic efficiency, exergy destruction ratio, efficiency defect, rise in temperature in auxiliary heat exchanger and mass flow rate with constant effectiveness of the auxiliary condenser on various heat recovery parameters are presented in table 6.1-6.30.

It reveals that:

Fig.6.1 shows variation of COP with evaporator temperature at condenser temperature 313K & 323K for R-717 and it can be easily inferred that as the evaporator temperature increases, pressure ratio decreases causing compressor work to reduce and specific refrigerating effect to increase and hence COP increases. Two stage modified VCR system shows COP greater than that of two stage VCR system at a particular condenser temperature. As the condenser temperature increases, pressure ratio increases causing compressor work to increase and specific refrigerating effect to decrease, hence decrease in COP. The similar trend has been shown by the refrigerants corresponding to the condenser temperature of 323K.

Fig.6.2 shows variation of COP with evaporator temperature at condenser temperature 313K & 323K for R-507a and it can be easily inferred that as the evaporator temperature increases, pressure ratio decreases causing compressor work to reduce and specific refrigerating effect to increase and hence COP increases. Two stage modified VCR system shows COP greater than that of two stage VCR system at a particular condenser temperature. As the condenser temperature increases, pressure ratio increases causing compressor work to increase and specific refrigerating effect to decrease, hence decrease in COP. The similar trend has been shown by the refrigerants corresponding to the condenser temperature of 323K.

Fig.6.3 shows variation of COP with evaporator temperature at condenser temperature 313K & 323K for R-134a and it can be easily inferred that as the evaporator temperature increases, pressure ratio decreases causing compressor work to reduce and specific refrigerating effect to increase and hence COP increases. Two stage modified VCR system shows COP greater than that of two stage VCR system at a particular condenser temperature. As the condenser temperature increases, pressure ratio increases causing compressor work to increase and specific refrigerating effect to decrease, hence decrease in COP. The similar trend has been shown by the refrigerants corresponding to the condenser temperature of 323K.

Fig.6.4 shows variation of COP with evaporator temperature at condenser temperature 313K & 323K for R-152a and it can be easily inferred that as the evaporator temperature increases, pressure ratio decreases causing compressor work to reduce and specific refrigerating effect to increase and hence COP increases. Two stage modified VCR system shows COP greater than that of two stage VCR system at a particular condenser temperature. As the condenser temperature increases, pressure ratio increases causing compressor work to increase and specific refrigerating effect to decrease, hence decrease in COP. The similar trend has been shown by the refrigerants corresponding to the condenser temperature of 323K.

Fig.6.5 shows variation of COP with evaporator temperature at condenser temperature 313K & 323K for R-1234yf and it can be easily inferred that as the evaporator temperature increases, pressure ratio decreases causing compressor work to reduce and specific refrigerating effect to increase and hence COP increases. Two stage modified VCR system shows COP greater than that of two stage VCR system at a particular condenser temperature. As the condenser temperature increases, pressure ratio increases causing compressor work to increase and specific refrigerating effect to decrease, hence decrease in COP. The similar trend has been shown by the refrigerants corresponding to the condenser temperature of 323K.

Fig.6.6 - 6.7 shows variation of COP with evaporator temperature and it can be easily inferred that as the evaporator temperature increases, pressure ratio decreases causing compressor work to reduce and specific refrigerating effect to increase and hence COP increases. R-717 shows highest COP among all the refrigerants, followed very closely by R-152a which shows almost same COP. R-507a shows lesser COP then R-134a. As the condenser temperature increases, pressure ratio increases causing compressor work to

increase and specific refrigerating effect to decrease, hence decrease in COP. The similar trend has been shown by the refrigerants corresponding to the condenser temperature of 323K.The COP of R-717 is 4.0-8% and 3.7-6% higher than R-507a for T_C = 313K & 323K respectively, having minimum value at higher end of evaporator temperature, and 6.2% and 4.4% higher than R-134a for T_C = 313K & 323K respectively, which diminishes at high value of evaporator temperature.

Fig.6.8 - 6.9 represents the variation of exergetic efficiency with evaporator temperature. Exergetic efficiency is decreasing as evaporator temperature increases, and this can be attributed to two parameters (refer 26). First parameter is thermal exergy flow in the evaporator i.e. $\dot{Q}_E \left|\left(1 - \frac{T_o}{T_E}\right)\right|$, with increase in evaporator temperature, refrigerating effect $\dot{Q}_E$ increases, however the term $\left|\left(1 - \frac{T_o}{T_E}\right)\right|$ reduces since T_E approaches To and second parameter is compressor work, which reduces with the increment in the evaporator temperature. The effect of $\dot{Q}_E$ and $\dot{W}_{comp}$ is to increase the exergetic efficiency as opposite to the decreasing effect of $\left|\left(1 - \frac{T_o}{T_E}\right)\right|$. The combined effect of these two parameters is to increase the exergetic efficiency till it reaches the maximum point and the evaporator temperature corresponding to this efficiency is optimum evaporator temperature, beyond which the combined effect is to decrease the exergetic efficiency. Exergetic efficiency of R-152a is 0.69-11.6% higher than R-507a for T_C = 313K and 0.59-15.25% for T_C = 323K, having minimum difference at the higher end of evaporator temperature and 4.4%, 5.1% higher than R-134a at lower end of evaporator temperature, corresponding to T_C = 313K & 323K respectively, which is negligible at higher end of evaporator temperature.

Fig.6.10-6.11 represents the variation of exergy destruction ratio (EDR) with evaporator temperature, and it has been observed that trend shown by curves of EDR is reverse of that shown by curves of exergetic efficiency except for ammonia i.e. decreasing and can be validated by (27). Due to this reason ammonia as a refrigerant in multi stage vapour compression system is recommended to be used with intercooler. EDR of R-507a is higher than R-152a and this difference decreases in the range 1.84-15.5% for T_C = 313K and 2.35-13.2% for T_C = 323K as the evaporator temperature increases.

Fig. 6.12-6.23 represents variation of efficiency defect in compressor, condenser, auxiliary condenser, throttle valve, flash chamber and evaporator. Efficiency defect shows that what fraction of the input is lost through irreversibility in different components of the system. Thus it assists in determining the worst component in a system. Flash chamber showing minimum efficiency defect is the most efficient component of a system. In the descending order of efficiency defect the components are arranged as evaporator, auxiliary condenser, throttle valve, condenser, compressor and flash chamber. From the perspective of refrigerants it has been observed that loss of input energy due to irreversibility in compressor and throttle valve is maximum for R-507a and minimum for R-717 and the difference is decreasing as the evaporator temperature is increasing, however with increase in condenser temperature, efficiency defect in compressor reduces whereas it increases for throttle valve. The trend shown by the curves for the two condenser temperature is same. In condenser and evaporator, R-134a and R-152a gives maximum efficiency defect value respectively, which increases for the condenser and decreases for the evaporator as the condenser temperature increases. R-717 gives minimum value at lower end of evaporator temperature and at higher end of evaporator temperature for condenser. Total of efficiency defects in different components is observed to be minimum for R-717, followed by R-507a, R-152a and R-134a in ascending order.

Fig.6.24-6.25 represents the variation of heat removed in auxiliary condenser (Q_{AuxC}) with mass flow rate of water, and it has been observed that trend shown by curves of Q_{AuxC} is increasing as the mass flow rate of water increases. Q_{AuxC} is observed to be maximum for R-717 followed by R-152a, R-134a, R-507a and R-234yf in descending order. Q_{AuxC} of ammonia is 100-90% higher than R-1234yf for T_C = 313K and T_{w1} = 283K and 115-95% for T_C = 323K and T_{w1} = 293K, having maximum difference at the higher end of evaporator temperature and 80%, 85% higher than R-134a at lower end of evaporator temperature, corresponding to T_C = 313K & 323K respectively.

Fig.6.26-6.27 represents the variation of the fraction of the condenser heat removed by the Auxiliary condenser (FR) with mass flow rate of water, and it has been observed that trend shown by curves of FR is increasing as the mass flow rate of water increases. FR is observed to be maximum for R-717 followed by R-134a, R-152a, R-507a and R-234yf in descending order. FR of R-717 is 2-90% higher than R-1234yf for T_C = 313K and T_{w1} = 283K and 2.2-92% for T_C = 323K and T_{w1} = 293K, having maximum difference at the higher end of evaporator temperature and 80%, 85% higher than R-134a at lower end of evaporator temperature, corresponding to T_C = 313K & 323K respectively.

Fig.6.28-6.29 represents the variation of the temperature rise through auxiliary condenser ($T_{rise,AuxC}$) with mass flow rate of water, and it has been observed that trend shown by curves of $T_{rise,AuxC}$ is increasing as the mass flow rate of water increases. $T_{rise,AuxC}$ is observed to be maximum for R-717 followed by R-152a, R-134a, R-507a and R-234yf in descending order. $T_{rise,AuxC}$ of ammonia is 55-92.72% higher than R-1234yf for T_C = 313K and T_{w1} = 283K and 58-96.6% for T_C = 323K and T_{w1} = 293K, having maximum difference at the lower end of evaporator temperature and 83.6%, 87.93% higher than R-134a at lower end of evaporator temperature, corresponding to T_C = 313K & 323K respectively.

Fig.6.30-6.31 shows the effect of ambient state temperature on exergetic efficiency with increase in ambient state temperature, exergetic efficiency increases because of the increment in the term $\left|\left(1-\frac{T_o}{T_E}\right)\right|$, while the term Q_E and W_{comp} remains constant as can be observed from (18) and (20). This can also be attributed to the reason that with increase in the ambient state temperature, the irreversibility due to finite temperature difference decreases and hence exergetic efficiency increases. R-1234yf and R-507a shows the similar trends and their curves for exergetic efficiency is overlapping. R-1234yf shows lesser values for exergetic efficiency as compared to R-717 and R-134a for the ambient temperature range considered. As the condenser temperature increases, reduction in exergetic efficiency is being observed.

Using the computer program, we have calculated the results for a particular set of data. The effect of input parameter on condenser heat output and its heat distribution over the condenser and auxiliary condenser, the fraction of condenser heat available through AuxC, the available water temperature and the heat recovery factor have been studied by changing each of the operating parameters while keeping the others constant. The parametric variations of the input variables have been undertaken to study their effects on the condenser heat recovered through the AuxC as a function of evaporator temperature for different working fluids, namely R717, R-507a, R-134a, R-152a, and R-1234yf.

CHAPTRTR 8

CONCLUSION

Present study relates to a modified vapour compression system and focuses on waste heat recovery from superheat form of vapor compression system. A detailed thermodynamic analysis of the modified VCR system has been performed. The analysis has been done according to the applications of refrigeration. First of all results carried out by varying evaporator temperature of modified VCR system for most appropriate refrigerants R-717, R-507a, R-134a, R-152a, R-1234yf.

During this extensive energy and exergy analysis of the above mentioned refrigerants in a modified two stage vapour compression cycle, following conclusions are summarized below.

1. COP and exergetic efficiency of R-717 and R-152a is almost same having a difference of 3.5%, which increases with the increase in evaporator temperature, whereas it is 15.4% higher than R-507a. Hence R-507a can be a good ‘drop-in’ replacement of R-717 at higher value of evaporator temperature and R-134a can be a good replacement after certain modification.
2. From the irreversibility or exergy destruction viewpoint, worst component is evaporator followed by auxiliary condenser, throttle valve, condenser, flash chamber and compressor.
3. Increase in ambient state temperature has a positive effect on exergetic efficiency and EDR, i.e. EDR reduces and exergetic efficiency increases.R-1234yf gives lesser values of exergetic efficiency whereas R-507a gives approximately similar value.

Hence, it can be concluded the available waste heat can be utilized for domestic purposes and in drying or food processing sector etc. by simply introducing an Auxiliary condenser with existing system i.e. two stage VCR system. The novelty of this auxiliary condenser is that it does not affect the COP of the system rather than increasing overall COP of the system. Hence, heat recovery through an auxiliary condenser is feasible and can be maximized by selecting optimum water flow rate, inlet water temperature, suitable operating conditions, and working fluid.

On comparison of results for all five refrigerants, it is concluded that for the same operating conditions, the R-717 yield better performance as compared to R-134a, R-507a, R-152a, and R1234yf. An auxiliary condenser options for heat recovery for a given evaporator temperature R-717 giving higher heat removal factor. But few of the results are in contrary to this conclusion which may be indicating the use of an inter-cooler for an overall positive outcome while using R-717 in a multi-stage vapour compression refrigeration system as suggested in many literatures.

CHAPTER 9

RECOMMENDATION FOR FUTURE WORK

Although the multi-stage vapour compression refrigeration system enjoys certain advantages over the conventional method of single stage vapour compression, it is observed that the temperatures to which a refrigerant is to subjected in a multi-stage system are limited. For going beyond the temperature limits and to further enhance the performance of the system and its utility , other performance enhancing systems/ devices can be employed in conjunction with this system like; cascading it with ejector refrigeration system, adsorption refrigeration system etc.

1. Exergy analysis should be carried out for this modified vapour compression refrigeration system.
2. Thermo-economic optimization can be performed for optimal design of the system.
3. The analysis could be done using other better refrigerants of further low GWP and thermodynamic properties than R-717, R-507a, R-134a, R152a, and R-1234yf.
4. The cooling fluid with higher specific heat gives better result i.e. the amount of heat removed in auxiliary condenser and condenser will be large.

CHAPTER 10

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