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Thermodynamic Analysis of Cascade Refrigeration System for Low Temperature Applications

Thesis
B.Sc. in Mechanical Engineering

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Candidate's Declaration

We hereby declare that this thesis titled "*Thermodynamic Analysis of Cascade Refrigeration System for Low Temperature Applications*" is a genuine report of our study carried out as a requirement for the award of degree *Bachelor of Science in Mechanical Engineering* at Islamic University of Technology, Gazipur, Dhaka, under the supervision of Sayedus Salehin and Mohammed Raihan Uddin, MPE, IUT during January 2020 to February 2021. This thesis has never been submitted in whole or in part to any other university/college for the award of any other degree or diploma.

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Certificate of Research

The thesis titled **'Thermodynamic Analysis of Cascade Refrigeration System for Low Temperature Applications'** by Md Walid Faruque (160011042) and Mahdi Hafiz Nabil (160011030) has been accepted as satisfactory in partial fulfilment of the requirement for the Degree of Bachelor of Science in Mechanical Engineering on March 2021.

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Dedication

We would like to dedicate this thesis to our loving parents . . .

Abstract

In recent years, concern about global warming and ozone layer depletion has been growing. Increasing attention is being given to refrigerants used in refrigeration and air-conditioning systems as they have a significant contribution towards these environmental issues. As industries and different chemical plants are increasing day by day, it is becoming a challenge for researchers to find suitable refrigerants that will be safe for the environment and maintain the desired performance. In the case of different ultra-low temperature (-100^{o}C to -150^{o}C) applications (e.g., LNG, LPG liquefaction, cryogenic engineering, food, and medicine storage), it is even more challenging to satisfy environmental conditions and performance. This study aims to find suitable environment-friendly refrigerants and implement them in a triple cascade refrigeration system (TCRS) for low-temperature application (-100^{o}C to -150^{o}C). For this purpose, a simulation model for the system was developed and validated. Based on low global warming potential, and low ozone depletion potential; eco-friendly hydrocarbon (HC) refrigerants were selected for implementation in the triple cascade refrigeration system (TCRS). In the lower temperature circuit (LTC), 1-butene was used as working fluid, and in the higher temperature circuit (HTC), m-Xylene was used. For the mid-temperature circuit (MTC), trans-2-butane, n-heptane, n-butane, cis-2-butane were used. To assess the performance of TCRS, energy analysis and exergy analysis were conducted, and analyzed. The results show that 1-butene/heptane/m-Xylene pair gives the best performance in terms of 1st law efficiency (COP) and 2nd law efficiency (exergy destruction) for

low-temperature applications (lower than -100°C). The results obtained from the simulation model suggests that exergy destruction mainly occurs at the condenser, hence further studies can be carried out on the condenser to increase the overall COP. Further experimental studies can be carried out to assess the feasibility of utilizing the proposed refrigerants pair for low-temperature applications.

Keywords

Cascade refrigeration system; Low-temperature application; Hydrocarbon refrigerants; Thermodynamic analysis; Thermodynamic system modeling; Energy analysis; Exergy analysis

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Chapter 1

Introduction

1.1 Introduction

Achieving temperature lower than the atmosphere is done by employing refrigeration system. Among traditional refrigeration systems like vapor compression system are widely used for household refrigeration systems and different storages such as food, medicine, cryogenic products etc. Ideally, Using simple single stage vapor compression refrigeration system lowest evaporation temperature that can be achieved is around -25°C. This temperature is a bit higher in actual system due to irreversibility. Which can be achieved using refrigerant having normal boiling point temperature around this range. Kilicarslan et al. [32] states that single stage vapor compression systems are not suitable for low temperature applications due to difficulty in compression process of the refrigerant in between large pressure ratio and solidification temperature of refrigerants. Johnson et al. [27] listed two major limitations of single stage vapor compression refrigeration system . Firstly, if pressure goes below certain limit, the compression process becomes too expensive and secondly, after compression the refrigerant must have pressure below its critical pressure to ensure two-phase condensation process. In industrial processes like natural gas liquefaction, preservation of complex medicine and vaccine [e.g. COVID-19 vaccine preservation needs around -70°C], cryogenic process and other very low temperature applications employing single stage vapor compression refrigeration system is not a feasible option thermodynamically or economically. This problem calls for a need of different approach to achieve the desired level of cooling effect at very low temperature. Widely used refrigeration technologies for low temperature application can be separated in the two major category based on working fluid types used [59] which are the systems using pure working fluids and systems using mixed working fluids which is generally the combination of different organic compounds. Systems that employ single refrigerant as working fluid are either simple vapor compression system or another innovative system that uses multiple stages of vapor compression systems or combination of absorption system and vapor compression system on top of another named 'Cascade Refrigeration System'. The system which has several refrigeration cycles above each other, each cycle ranging in different temperature and pressure level, exchanging heat through a common heat exchanger in between, generating a huge cooling effect at very low temperature is defined as the cascade refrigeration system. The first usage of cascade refrigeration system is found in 1877 which was built by Cailletet and Pictet [52] who used ethylene and SO_2/CO_2 separately to build the system. Then later on, in 1908 helium was liquefied using precooling by cascade refrigeration cycles with air and hydrogen [52]. Since then, a trend of using one refrigeration cycle above another was introduced. This is where the term 'Cascade Refrigeration System' first occurred.

1.2 Related Literature Review

After the first use of cascade refrigeration system, it has been a topic of interest for the researchers. Analysis and optimization of different cascade systems for different applications are found throughout the available literatures. Specially two-stage cascade refrigeration system had the spotlight of research interest throughout the late

1990s and early 2000s.

Kilicarslan et al. [31] conducted experimental investigation on different types of vapor compression refrigeration system. He investigated the effect of mass flow rate and cooling load on coefficient of performance of single and two-stage refrigeration system using R134a. He also compared the compressor's work, exit pressure, refrigerant mass flow rate and showed better result compared to the traditional single stage system.

Thermodynamic analysis of performance of two stage cascade using three different pairs (R744 and R1270 or R744 and R717 or R744 and RE170) were compared by Massuchetto and others [39]. This paper shows R744 and RE170 obtained the best COP of 2.34, where the exergy efficiency had been increased up to 30 %.

A twin-screw compressor was employed by Bingming and Huagen et al. [7] using natural refrigerants in two stages of cascade system. NH_3 and CO_2 was used in high temperature and low temperature circuit respectively. The result showed that among other alternatives, the dual-stage with different refrigerants in high and low circuits proved to be more efficient when the system had evaporation temperature below -40^{o}C.

Kilicarslan nad Hosoz [32] conducted energy and irreversibility investigations on a cascade refrigeration system employing R152a–R23, R290–R23, R507–R23, R234a–R23, R717–R23 and R404a– R23 using a computer simulation. From the experiment following conclusions were derived-

1. Increase evaporation temperature or polytropic efficiency of compressor reduces irreversibility using any pair of refrigerants.

2. Coefficient of performance increases as polytropic efficiency and evaporator temperature rises.
3. As temperature of evaporation decreases the effectiveness of single stage vapor compression system decreases.

Lee et al. [34] conducted thermodynamic analysis on a cascade refrigeration system to find the optimal temperature of condenser while using CO_2 and NH_3 as refrigerant pair. He concluded that for condensation temperature of -15°C the maximum coefficient of performance of 1.15 can be achieved. It was also observed that maximum coefficient of performance can be enhanced by using higher evaporator temperature, on the other hand performance decreases with increasing condensation temperature.

Bhattacharyya et al. [6] tried to maximize the exergy efficiency of a endo-reversible cascade refrigeration system using numerical method employing trans-critical CO_2-C_3H_8 . An optimum intermediate temperature was obtained. Also. Heat reservoir temperatures were optimized. Furthermore, Generalized analysis on single fluid cascade refrigeration system was carried out by Ratts and Brown [54] using entropy generation minimization method. Employing R134a in each stage the analysis provided insight on mechanical dissipation and thermal dissipation.

Prapainop et al. [50] analyzed the exergy and refrigerant properties of cascade refrigeration system for natural gas liquefaction plant. It was concluded that the combination of low viscosity, high vapor density, latent heat, thermal conductivity, low moleculat weight can increase the overall energy and exergy performance. For critical temperature, trade off between performance and cooling capacity is suggested. It was also observed that, system performance significantly depends on refrigerant's critical temperature, thermal conductivity, viscosity and vapor density.

Application of cascade refrigeration system in olefin plants was analyzed by Mafi et al. [37]. In this study an expression for minimum work requirement for olefin plant was developed. Exergy efficiency of a typical cascade refrigeration system in olefin plants was determined 30.88 % which showed the potential of improvement in different components of the system. Also mixed refrigerant composition was anlyzed in this work.

Murthy and Murthy [47] investigated the energy and exergy performance of cascade system operating in vapor compression heat pump using R12 refrigerant and that of a cascade refrigeration system with R11. Result showed the coefficient of performance to be the range of 1.2-1.7 and exergitic efficiency was 0.1-0.15 in heat pump and refrigeration system. Which indicates better cascade system performance.

Theoretically analysis can be found of a cascade refrigeration system done by Dopazo et al. [1] using CO_2 and NH_3 refrigerant in both cycles. Study was conducted based on evaporator and condesing temperature, temperature difference between common heat exchanger fluids and irreversibility analysis of the compressor. Also, Cascade system in an automobile air conditioning system employing R134a in both cycles was investigated by Hosoz et al. [24] by employing neural network.

Usage of renewable energy in refrigeration system is found in absorption refrigeration system and solar PV refrigeration systems. Absorption system replaces traditional compressor system of classic vapor compression system. Input work is decreased as the pump does not work in the vapour region. Literatures of two stage cascade refrigeration systems are found employing absorption cycle as upper cycle and vapor compression cycle as lower cycle. This system configuration has been getting spotlight for some time now. Auto-cascade refrigeration which uses single compressor for both cycles is another innovative cascade refrigeration system, which is also proven to be capable of obtaining very low evaporator temperature.

Absorption cascade refrigeration system was first patented in 1969 by Bearint et al. [4]. Despite being patented during mid 1900s, implementation in cascade absorption refrigeration systems started in later in nineteenth century. In recent years, cascade absorption systems have been analyzed and optimized many literatures.

LiBr-H_2O compression-absorption system was analyzed by Cimsit and Ozturk [13]. Coefficient of performance of 0.592 was obtained which was 33 % better than system with NH_3-H_2O. Later on optimization was carried out on this system using computer simulation [14] which increased the COP by 7.3 % and second law efficiency by 3.3 %. It was concluded that this system has potential to reduce consumption of electricity by 50 % using solar, geothermal and waste heat, compared to vapor compression system.

Hybrid absorption-compression refrigeration system with ammonia-water binary mixture was proposed by Han et al. [20] using mid-temperature waste heat. Subsystems of the setup shared same evaporator and compressor. The waste heat was utilized to generate power for compression subsystem and also to preheat the strong solution before rectification. 46.7 % improvement based on cooling effect over classic absorption system was obtained.

In natural gas liquefaction, replacement of classic vapor compression system by a novel mix fluid cascade absorption system was simulated by Mehdi et. al. [41]. Using Aspen HYSYS the simulation showed he system COP to be 0.48 with reduction in power consumption by 30 % by reducing heat transfer area upto 31 %. Another novel cascade absorption-compression system achieving very low temperature was simulated and also experimentally validated by Xu et al. [60]

Evaporator temperature -40°C was produced by employing novel cascade absorption refrigeration system consisting two absorption refrigeration (AR) system of ammonia-water and LiBr-water systems [62]. The maximum coefficient of performance was obtained to be 0.19 and second law efficiency of 9.71 % was observed. This system was used for recovering low-grade waste heat energy. Thermodynamic analysis of cascade compression-absorption system by Jain et al. [26] using R401A, R407C and R134A showed power consumption reduction and COP improvement over classic VCR system were 61 % and 155 % respectively. These refrigerants were shown as potential replacement of R22.

Cul et al. [16] conducted a multi-objective optimization of cascade absorption refrigeration system. This study confirmed the contradicting position of economical efficiency and thermodynamic efficiency. This study shows that the trade off between two efficiency must be kept in mind while designing the system.

Exergy, economic and environmental analysis on cascade absorption-compression refrigeration system was conducted by Mousavi and Mehropooya [46]. Simulated system reached exergy efficiency of 69 % with highest irreversibility in rectifier. According to economic analysis again the rectifier had the highest investment cost of about 2.817 USD per hour. Environmental analysis makred rectifier and pressure pump to be the most suitable equipment.

Cascade Absorption refrigeration systems are found in waste heat recovery cogeneration systems. Farnandez et al. [18] showed the adaptability between power supply and power requirement in a cascade absorption system following the spanish cogeneration regulations. Cascade absorption system powered by waste heat from organic rankine cycle was optimized by Patel et al. [49]. The energitc efficiency of the whole system with cooling was found to be 22.3 % and 79 % with cooling and heating during cogeneration. Cogeneration of Steam Rankine cycle and cascade absorption system was also investigated by Liang et al. [35] which showed, recovery of expansion work in absorption system increases the electricity generation of steam rankine cycle. Waste heat recovery cascade absorption system was optimized by Yang et al. [61] using LiBr and H_2O.

Kai et al. [17] conducted studies on characteristics of an auto-cascade refrigeration system using zeotropic mixture as refrigerant. Effect of gas liquid separation, heat transfer and refrigerant mixture had significant impact on performance of the system. It was also showed that using 30 % charging concentration by weight can increase system coefficient of performance.

He and Chen [22] conducted an experimental study on absorption refrigeration system for low temperature applications. R23, R32, R134a and DMF was used as refrigerants. It was concluded that lower temperature than absorption system can be obtained using auto cascade refrigeration system.

In recent times, another type of refrigeration system is getting attention in the field of low temperature cascade refrigeration system. This system includes three stage of refrigeration circuits on top of another similar to two stage cascade system, which may include same refrigerants or different in different stages. For liquefaction of different natural gas, petroleum and for cryogenic applications and in pharmaceutical storages temperature as low as -170°C or below might be required, which is difficult to achieve using traditional approach without compromising the economical aspects and complexity. Due to being a relatively new concept in the field of cascade refrigeration systems, there are very few lieratures are available analyzing the performance and application of three-stage cascade refrigeration system. A three-stage cascade refrigeration system applied in cryogenic system was designed by Johnson et al. [27] . This literature presented a quantitative design of three-stage cascade refrigeration system. The System achieved evaporator temperature of -158°C using methane , ethylene and propylene in three stages and used it in cryogenic distillation column to separate carbon monoxide and methane.

The Application of three stage cascade refrigeration in LNG liquefaction was analyzed by Najibullah et al. [48]. By obtaining optimal operating condition using simulation this paper presents the enhanced efficiency of the system. Cho et al. [11] simulated natural gas liquifeaction process for propane, a three stage cascade refrigeration system was used at -40°C. Two stage system was used for ethylene.

Another application of triple Cascade refrigeration in natural gas liquefaction using propane, nitrogen monoxide and nitrogen was presented by Yoon et. al. [63].Comparison was shown with another setup using refrigerant combination propane, ethylene, methane. Significant performance upgrade was noticed using this system by 25 % above single stage cycle and optimized triple cascade system with second combination of refrigerants. COP was observed 1.68 with lowest temperature of the system keeping at -150°C.

Sivkumar and Sosmasundaram [56] used zeotropic mixture of two combination of five environment friendly refrigerants in three stage auto cascade refrigeration system. The system was employed for -97°C. The COP was found to be 0.253 and exergic efficiency was found to be 22.6 % improved over conventional system.

Thermal modeling of three stage cascade refrigeration system was developed by Mishra [44] for semen preservation. The system was developed using R1234yf and R1234ze in the high and mid-temperature circuit and R134a or R404a in the low-temperature circuit. Best thermal performance was obtained using a combination of R1234ze-R134a-R404a.

Sun et al. [58] optimized three stage cascade refrigerant system for different pairs of refrigerants on the basis of low global warming potential. Based on the simulation result this paper suggested using R717, R152a and R161 for high temperature cycle, R41 and R170 in intermediate cycle. R1150 was recommended for lowest circuit for evaporator temperature -120°C to -80°C range.

For last few decades environmentalists have been concerned about the global warming and ozone layer destruction as they are posing great threat to the proper existence of ecosystem of the word. As it's known that working fluid of different thermodynamic systems, specially the fluids from refrigeration and air conditioning systems are mainly responsible for global warming and ozone layer depletion, researchers are trying to find environment friendly working fluids for existing systems. While looking for suitable alternative it's important to keep in mind that the performance of different refrigeration system must not be compromised. Harby [21] mentioned the role of refrigeration system in impacting environment. This paper suggests using hydrocarbons and their mixture instead of halogenated refrigerants as their global warming potential is close to zero. Faraz et al. [3] employed environment friendly organic refrigerants with low global warming potential in rankine cycle and and compared the performance of the system in different seasons. Zhai et al. [64] analyzed organic rankine cycles and suggested two indicators for screening refrigerant with lower GWP.

1.3 Scope of this work

Previous studies indicate widespread use of two stage cascade refrigeration systems. Specially, cascade absorption system has been employed in many applications. This study aims to analyze three stage cascade refrigeration system (TCRS) for low temperature applications(e.g., -40^{o}C to -150^{o}C). Due to ongoing environmental concerns regarding refrigerants current study also focuses on finding suitable organic refrigerant's combinations applied in three stage cascade refrigeration system without compromising the performance of the TCRS. Different refrigerant pairs were employed in the system. Best refrigerant combination has been selected based on COP of the system. Energy and exergy analysis of the system was done using python program, which was validated from [58] . CoolProp 6.4 [5] was used for finding thermodynamic properties during the analysis.

Chapter 2

Literature review

In this section, the refrigeration system's fundamentals will be discussed along with the necessary thermodynamic analysis parameters. Briefly, the types of different refrigeration systems will be discussed. Besides, modifications for the improvement of the refrigeration system will be discussed.

2.1 Refrigeration System

There are mainly two types of thermodynamic systems: (1) work producing plant (e.g. Power Plants) and (2) work consuming systems (e.g. refrigeration and air conditioning systems). The refrigeration system is a work consuming device. A refrigeration system is a work consuming device as it uses external power input to generate a cooling effect using combinations of different components and a heat transfer fluid. The refrigeration system, serving the purpose of cooling a space by heat rejection, is widely used in almost all industries and households worldwide. This device either cools or maintains the temperature of storage below the temperature of the environment. This operation can be executed by implementing the Carnot cycle in reverse, which is not feasible in real life as it's an ideal concept. The iso-thermal processes of heat absorption and rejection cannot be achieved. In real life, the refrigeration cycle is executed by applying the well-known vapor compression refrigeration cycle.

2.2 Development of Refrigeration Systems

In the past, ice was used for preservation purposes, which was not enough to store larger amounts, and ice does not last long, leading to inventing devices that can hold larger storage and sustain longer. That led the scientists to research and development of refrigerators. Refrigeration systems seen today had to go through many research and developments to reach this point. However, further possibilities of improvements leading researchers to conduct studies and innovate new systems.

A refrigerator with or without a freezer is a major appliance designed for cooling and preserving foods. One of the essential advancements toward the invention of the refrigerator came when the refrigerated coil was invented by Ibn Sina [30] in the 11th century. But the first research that can be found about the refrigeration system was back in 17th century by Sir F. Bacon. Later on more researches were done during the 18th century by M. Lomonoso [32]. The Vapor Compression Refrigeration system we know now was first attempted to be built by W. Cullen [33] in 1755. But the term ‘Refrigerator’ was mentioned later on in 19th century. Around 1800 American engineer, T. Moore made a box insulated with rabbit fur, filled with ice, surrounded by a sheet metal container. In 1834 the first patent of vapor compression refrigeration (VCR) was obtained by J. Perkins. The vapor absorption type refrigeration we know now was first introduced by E. Carre who used water and sulphuric acid in 1850 [33]. Nine years later his brother F Carre built the first NH_3/H_2O based absorption type refrigeration system[12]. Around the same time, the first commercial refrigerator in the world was built by J. Gorrie and patent was granted in 1851. This field kept on improving from the first introduction till today and will continue to improve.

The main goal of the refrigeration system is to lower the temperature of substances below atmospheric temperature. A refrigeration system is used in different applications e.g. preserving the food, medicine, blood bank etc. Also, in gas liquefaction and air conditioning systems. They might be used in different complex systems like chemical processing and refining industries, separation of chemical components of a mixture by condensation, and other processes. Different systems need different cooling requirements.

The simple vapor compression cycle is the most widely used refrigeration cycle. Due to its simplicity, it is suitable for household use. However, for larger industrial applications, the main goal is to get higher efficiency and varying cooling loads according to need, so new modification is frequently done on the related cycles. The new industries have cooling requirements different than typical applications. liquefaction of LNG, LPG, air, and component separation from a mixture need lower temperature than -100oC. Especially in cryogenic engineering, the temperature needed is around -233^{o}C [25] which is not achievable by conventional refrigeration systems. To overcome this challenge, cascade refrigeration system was introduced. The first usage of cascade refrigeration system is found in 1877 which was built by Cailletet and Pictet [52] who used ethylene and SO_2/CO_2 separately to build the system. Then later on, in 1908 helium was liquefied using pre-cooling by cascade refrigeration cycles with air and hydrogen. Since then, a trend of using one refrigeration cycle on top of another like electrical series connection was introduced. This is where the term ‘Cascade Refrigeration System’ first occurred. So, in a way we can define cascade refrigeration system as, the system which has several refrigeration cycles above each other, each cycles operating in different temperature and pressure level, exchanging heat through a common heat exchanger in between, generating a huge cooling effect is called a cascade refrigeration system.

2.2.1 Vapour Compression System

The vapor compression cycle is the most commonly used in the commercial refrigeration systems. It has a high coefficient of performance. Replacing wet compression of reversed Carnot cycle by dry compression and isentropic expansion by throttling process we can execute this system.

Instead of the iso-thermal process of heat addition and rejection, the isobaric process is used. An ideal VCR cycle consists of four components that execute four processes. These processes are applied using a heat transfer fluid named refrigerant flowing through different sections of the system. The refrigerant evaporates taking heat from the cold body in refrigerated space, which then goes through a compression process and gets condensed by rejecting heat to the environment. Finally, it gets throttled to evaporating pressure.

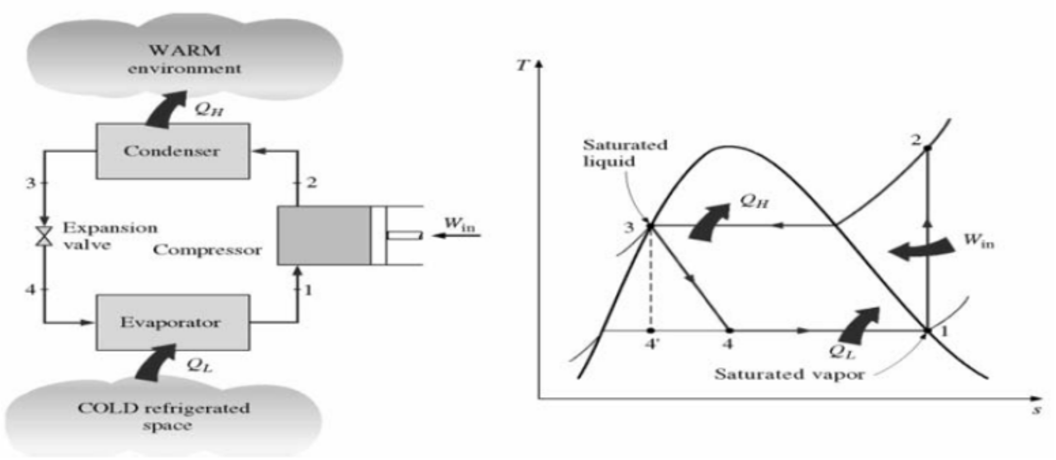


Figure 2.1: Vapor Compression Refrigeration cycle [51]

Processes involved are listed below-

1. Compression.
2. Condensation.

3. Expansion.
4. Evaporation.

Figure 2.1 shows the execution of four processes in four different components. Before diving into the details of the processes the states of the working fluid at the entry point of each component should be noted. At state 1 refrigerant is in saturated vapor condition before entering compression. At state 2 refrigerant enters the condenser as a high-pressure superheated fluid and condenses into a saturated liquid at state 3. After the expansion process, it enters the evaporator as a low-quality mixture of liquid and gaseous fluid at point 4. Thermodynamic processes involved in this system are discussed below.

Compression

From point 1 to 2 compression process occurs in a compressor, which takes electrical input from external source to execute this process. This is the system input. It is done isentropically, although there might be some deviation due to system losses inside the compressor. Refrigerant enters the compressor as saturated vapor at a very low pressure and leaves as a high pressure and high temperature fluid. As a result, the specific volume of the refrigerant reduces significantly after compression. As compressor generally handles gaseous fluid for safety of the device, its better to make sure the state of point 1 is fully gaseous. In that regard sometimes the refrigerant is superheated to some extant to ensure the gaseous fluid input into the compressor. No heat transfer is involved and the work input provided by the electrical compressor will raise the enthalpy of the refrigerant. Which can be expressed,

$$W_{in} = h_2 - h_1 \; [\mathrm{kJ\,s^{-1}}] \tag{2.1}$$

Condensation

After refrigerant leaves the compressor it is sent to the condenser, which is a heat exchanger where hot fluid is refrigerant and cold fluid is generally atmospheric air. In the condenser fluid undergoes phase change. Rejection of both sensible and latent heat to the environment will convert the refrigerant into saturated liquid at the exit of the condenser. This process is executed in constant pressure (isobaric process).

No work is done in this process. Only heat is rejected to the environment, which can be expressed,

$$Q_C = h_2 - h_3 \ [\mathrm{kJ\,s^{-1}}] \tag{2.2}$$

Expansion

To return the refrigerant to it's initial low pressure state, expansion is done. Due to expansion the refrigerant will have it's pressure reduced to the initial one. This expansion is done in an expansion valve, which expands the fluid in isenthalpic manner. After expansion the refrigerant reaches a condition where it falls under the mix region. No heat transfer is involved in this process. As expansion is done in isenthalpic manner, the enthalpy remains unchanged,

$$h_4 = h_3 \tag{2.3}$$

Evaporation

This process is also executed inside an evaporator. The hot substance is stored inside the refrigerated space. The refrigerant takes the latent heat of vaporization from the hot substance and evaporates while the loss of heat cools the hot substance down. In order to keep the pressure constant, the refrigerant is converted into saturated gas. The latent heat of vaporization needed is supplied by the refrigerated space, which becomes cooler due to heat loss. This heat is the desired output from the system. The exit of the evaporator is then again connected to the compressor. No work interaction occurs in the heat exchanger. The heat gained by the refrigerant is,

$$Q_E = h_1 - h_4 \ [\mathrm{kJ\,s^{-1}}] \tag{2.4}$$

COP of the System

Coefficient of performance (COP) is a measure of effectiveness of the refrigeration system with respect to the input energy provided. From the above processes it's clear that the system input is given in the compressor which finally results in heat of evaporation. Which is the desired effect from the system, in other words it's the desired output. Basic definition of efficiency or coefficient of performance is expressed as,

$$COP = \frac{h_1 - h_4}{h_2 - h_1} \tag{2.5}$$

2.2.2 Multistage Compression

In normal vapor compression refrigeration system, there's only two pressure levels. In many practical uses of refrigeration system, the pressure difference is too high for the compressor to handle. In those case multiple compressors can be used to reduce the load on one individual compressor. When single compressor is used the entropy change across the compressor pushes the constant entropy line further away from the saturated vapor line, which causes more irreversibility and finally reduction of COP. So, using multi stage compression is one of the ways to encounter this issue and keep the entropy line closer to saturation vapor line. This system is implemented when pressure ratio between the condenser and evaporator is greater than 4 or 5 [2]. Increasing pressure ratio decreases the volumetric efficiency of the compressor.

So, using multiple compressors will divide the pressure ratio among two or more compressors to get the volumetric efficiency higher.

The intermediate pressure level has to be selected properly to meet the optimum COP for this system. Pressure ratio between each stage is first determined using following equation,

$$r = (\frac{P_k}{P_o})^{\frac{1}{n}} \tag{2.6}$$

This equation gives optimum ratio of pressure with desired number of stages. One other parameter, the intermediate pressure is determined using this equation below,

$$Pi = (P_k \times P_o)^{\frac{1}{2}} \tag{2.7}$$

This equation is only acceptable for complete inter-cooling to the initial temperature which is not possible in real refrigeration system [2]. Following equation gives more optimum intermediate pressure.

$$Pi = (P_k \times P_o \times \frac{T_k}{T_o})^{\frac{1}{2}} \tag{2.8}$$

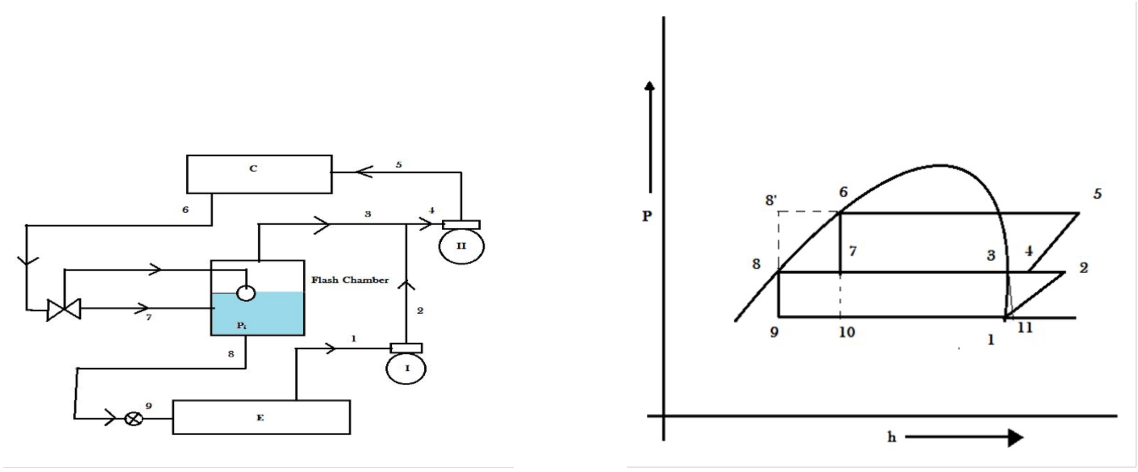
In these equations subscript k represents highest operating pressure and o represents lowest operating pressure of the system.

When two or more stages are employed in a refrigeration system, cooling between stages are necessary. This is executed in two ways,

1. Flash Gas Removal
2. Water Intercooling

Flash Gas Removal

When refrigerant goes through the condenser it doesn't turn into complete liquid. Gas doesn't exchange heat in the evaporator which reduces the overall cooling effect. This result in reduction of COP. Hence flash removal is introduced operating in the optimum intermediate pressure mentioned previously. Function of this device is to separate the liquid refrigerant and the uncondensed flash gas and send to the compressor to condense it in the condenser. The liquid is sent to the evaporator through expansion device, expanding it to evaporator pressure.



(a) Schematic diagram

(b) P-H diagram

Figure 2.2: Flash removal system ([2]).

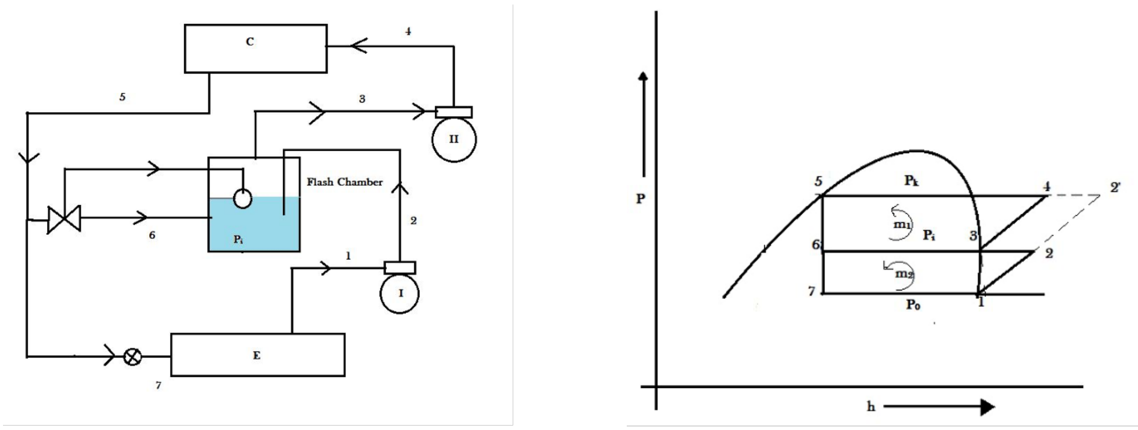
From the 2.2a point 6 is expanded to the optimum intermediate pressure and the exit flow sent into the flash chamber. Which is a chamber with exit at bottom and at the top. As liquid stays at the bottom side, saturated refrigerant leaves the chamber as point 8 and throttled to evaporator pressure. On the other hand, the vapor leaves the chamber from the top side as it's lighter. Evaporated gas at point 1 is compressed to intermediate pressure by compressor I and mixed with the flash gas removed from the flash chamber and sent to condenser II.

From the 2.2b it's evident that if the flash chamber didn't exist the condensate from point 6 would've been expanded to point 10. Which reduces the cooling effect from 9-10. Which is a huge loss of energy.

Another way of flash removal is done using liquid sub-cooler which is slightly different than a flash chamber. After the refrigerant is condensed, the flow is separated into two splits, one of which is expanded to intermediate pressure and another split is put through a pipe line through the expanded liquid refrigerant from first split. This results in subcooled liquid at the bottom of the chamber at point 8' shown on p-h diagram. These two methods to obtain results for same cooling load.

Flash Intercooling

In case of intercooling, instead of putting all the condenser exhaust liquid into the flash chamber, a portion of it is sent directly for evaporation through the expansion valve. The fraction which is directed towards the flash chamber will be expanded to the intermediate pressure. Inside the flash chamber it cools the exit gas from lower cycle compressor and itself gets evaporated to the saturated gaseous state. This saturated gas enters the upper cycle compressor.



(a) Schematic diagram

(b) P-H diagram

Figure 2.3: Flash removal system with intercooling.

The advantage of using this system is, it brings the second stage of compression near the saturation curve. Which reduces the overall work of compression which is evident from the P-h diagram 2.3b.

It is possible to implement both the flash gas removal and flash inter-cooling in single system. In flash gas removal we get more cooling effect and in inter-cooling we get reduced work of compression. Employing both in a system both the large cooling effect and reduced compressor work can be achieved. This leads to higher efficiency of the system. This system is shown in 2.4.

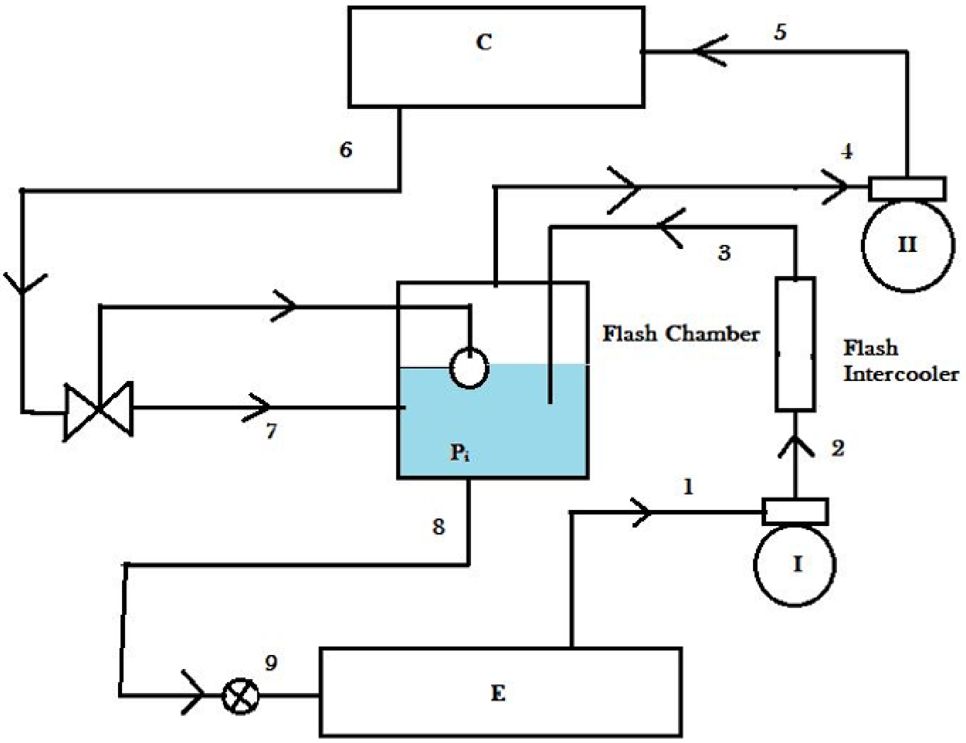


Figure 2.4: Combined flash Gas Removal and Inter-cooling

2.2.3 Multi Evaporator System

In large storage system one cooling chamber with fixed cooling load is needed. But situation might occur where multiple cooling load is needed from one system. In those case multiple evaporators is used. These systems may contain single compressor to compress merged flows from the evaporators or use separate compressors to take the separate flows to the highest system pressure state.

Single Compressor Multi Evaporator System

This type of system uses single compressor to handle the exit flows of multiple evaporators. Before compression the flows coming from the evaporators are mixed together in the lowest pressure level. To match the pressure level of both flow, high pressure stream is expanded after evaporator to the lowest pressure of the system. The single compressor system can be employed in two ways. They are discussed in the following section.

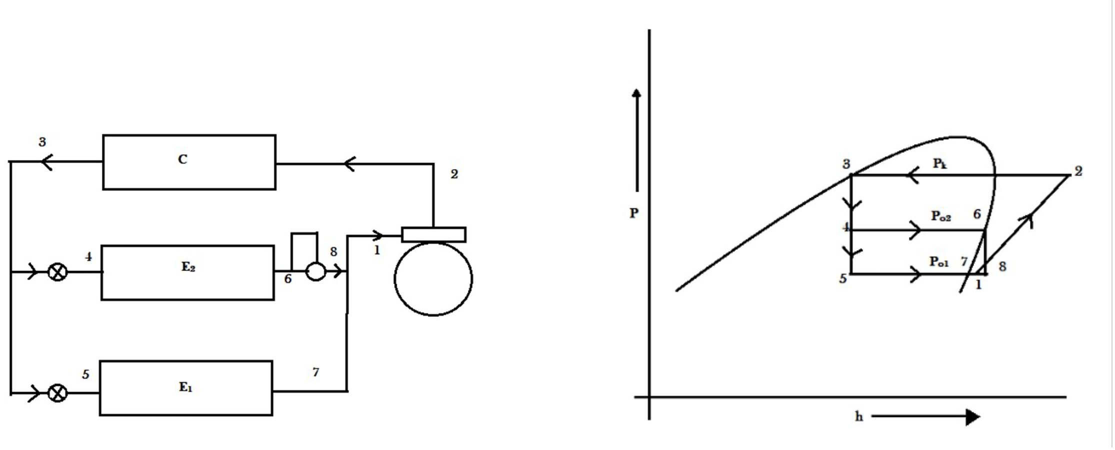
Individual expansion devices before evaporation

When different compartment of equal cooling effect is needed from different pressure levels, individual expansion devices are used before each stream enters respective evaporators. The streams are separated after the condenser, as shown in figure 2.7. Separated streams are expanded to different pressure levels and then evaporated before mixing again. Mixed flow is compressed using a single compressor to the highest pressure.

Fig:2.5b confirms that before evaporation the enthalpy state of both stream 4 and 5 are same and after evaporation they exits the evaporator at saturated vapor state in respective pressure level.

2.2.4 Expansion after Condenser

When different level of cooling effect is necessary in different levels, this type of system is employed. Figure 2.9 shows that after condenser the flow is expanded to an intermediate pressure. After expansion the flow is split into two streams one of which is directly evaporated in the intermediate evaporator. Other split is further

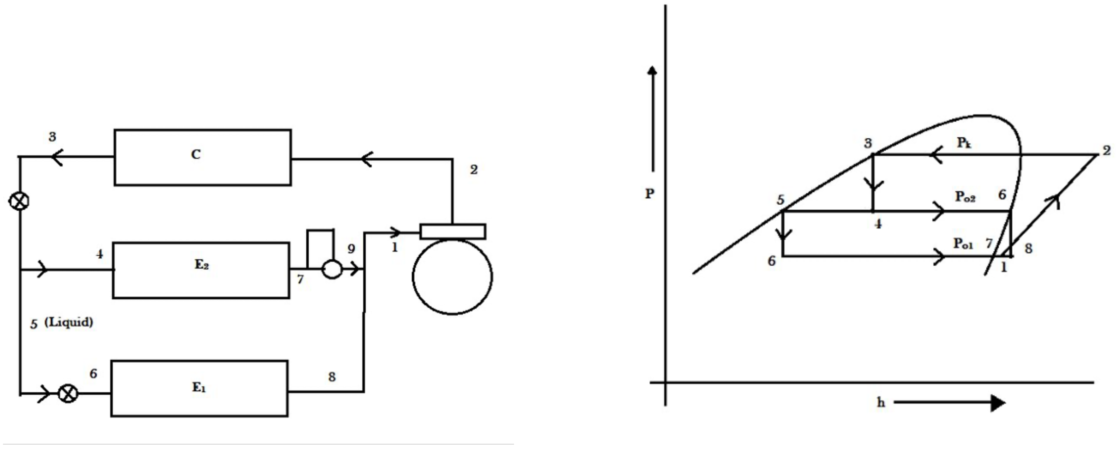


(a) Schematic diagram (b) P-H diagram

Figure 2.5: Individual expansion devices before evaporation

expanded to the lowest pressure of the system and then evaporated in the evaporator.

2.6b shows that the cooling load is different in two evaporators, which meet the purpose of the system.

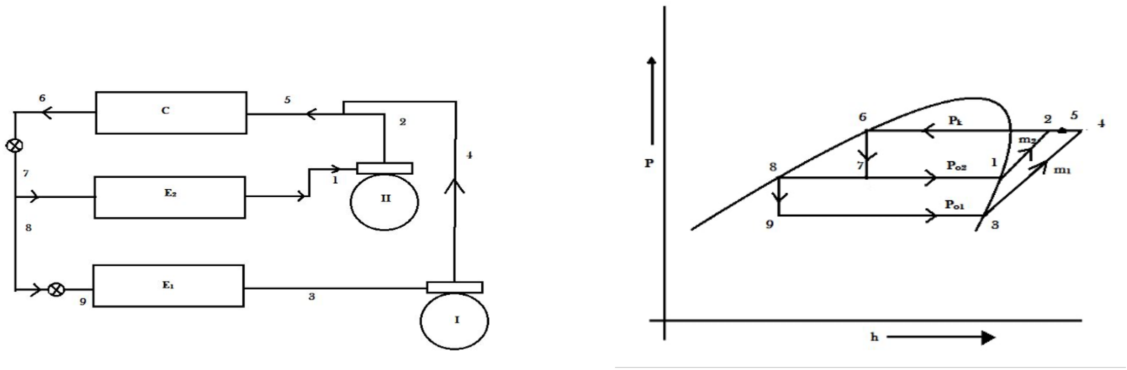


(a) Schematic diagram (b) P-H diagram

Figure 2.6: Individual expansion devices after compressor.

Individual Compressor Multi Evaporator System

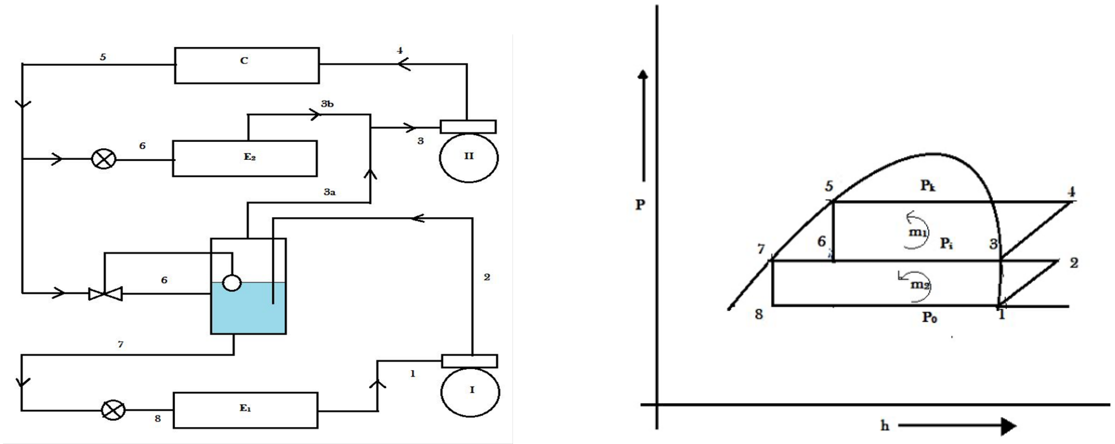
Previously mentioned systems of individual expansion device and multiple expansion device can be modified by using individual compressors to compress the exit



(a) Schematic diagram (b) P-H diagram

Figure 2.7: Individual expansion devices after compressor.

flows from the evaporators instead of mixing it before compression. Use of individual compressors reduces the load on single compressor. The power required also reduces in this method. 2.7b illustrates the system with individual compressor and multi expansion device. This system's efficiency can be increased using previously mentioned flash removal chamber and intercooler. 2.8a shows this system.



(a) Schematic diagram (b) P-H diagram

Figure 2.8: Individual Compressor Multi Evaporator System with Flash chamber and intercooling.

2.2.5 Absorption Refrigeration System

When there is a low-cost heat source, an absorption refrigeration system can be used. This system is economically feasible and environmentally friendly. Most of the refrigeration system that's out there uses Chloro-Fluro Carbon (CFC) as refrigerants. The emission of these chemical compounds is very harmful to the ozone layer. They also contribute significantly to global warming. The absorption refrigeration system provides an alternative approach to this problem using environmentally-friendly refrigerants as a binary solution with a carrier fluid. In 1859, Ferdinand Carre [12] invented NH_3/H_2O absorption system and got patent in 1860. Later on, lithium bromide and water mixture was introduced for industrial purpose. The binary solution that is used as a refrigerant consists of refrigerant and absorber. The binary solution that is used as a refrigerant consists of refrigerant and absorbent. This system mainly dependent on the thermodynamic and chemical properties of the working fluid [8]. Chemical stability is must for this type of binary working fluids.

The absorption refrigeration system is almost same as classic vapor compression system. The difference occurs as the compression process is replaced by a complex combination of some components. The replacements of compressors are a pump, a generator, an absorber, a valve and a rectifier.

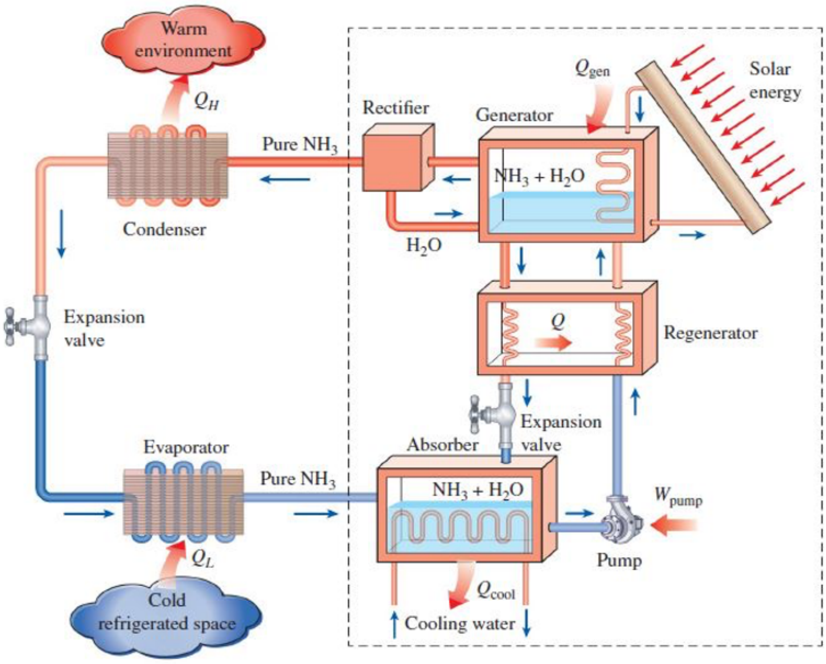


Figure 2.9: Absorption Refrigeration System [10]

Figure 2.9 illustrates the absorption refrigeration system. Except the boxed portion rest of the system is same as vapor compression system. That portion consists of a condenser, an evaporator and an expansion valve. After generating cooling effect in the evaporator ammonia enters the absorber. Generally, water is used as absorbent. In the absorber water dissolves ammonia and releases heat as it's an exothermic reaction. The solubility of ammonia in water is inversely proportional to temperature i.e., if temperature increases the solubility of ammonia in water will decrease. So, in the absorbent the temperature has to be as low as possible to ensure maximum absorption of ammonia. Thus, the ammonia rich binary mixture of water and ammonia enters a pump which is sent to a generator. The generator receives heat energy from different sources. Such as, geothermal energy, solar energy, waste heat from other cycles etc. The heat is transferred to the mixture. As a result, some of the mixture vaporizes. This ammonia rich vapor mixture goes through the rectifier, which separates water from the mixture. The separated ammonia enters the condenser and gets condensed and through expansion valve reaches evaporator again. There's a regenerator that utilizes the heat of high temperature water to preheat the mixture of ammonia and water before they enter the generator.

The COP of this system is expressed as,

$$COP = \frac{Q_L}{Q_{gen} + W_{pump}} \tag{2.9}$$

Generally, pump work can be ignored as it is very small in quantity,

$$COP = \frac{Q_L}{Q_{gen}} \tag{2.10}$$

There are different absorption refrigeration systems available now. Some of them are-

1. Single-effect absorption system[19].

2. Half-effect absorption refrigeration cycle[38]

3. Double-effect absorption system[19].
4. Multi-effect absorption system[53].
5. Absorption heat transformer[23].
6. Absorption refrigeration cycle with GAX[40].
7. Absorption refrigeration cycle with an absorber-heat-recovery[28].
8. Combined ejector-absorption refrigeration cycle[57]

There are many more absorption systems available and they are being improved continuously.

2.2.6 Cascade Refrigeration System

In the basic vapor compression refrigeration system, there's only one stage and one refrigerant used. During very low-temperature applications(e.g. below -40^{o}C) . Production of low temperature is limited by the following reasons[32]-

1. The temperature at which refrigerant solidifies.
2. If the refrigerant has a very high boiling point, a high suction volume will be encountered at the inlet of the compressor due to very low pressure of evaporation.
3. High boiling point refrigerant will have very high condensation pressure.
4. High-pressure ratio results in a lower COP.
5. Hard to achieve lower evaporation temperature.

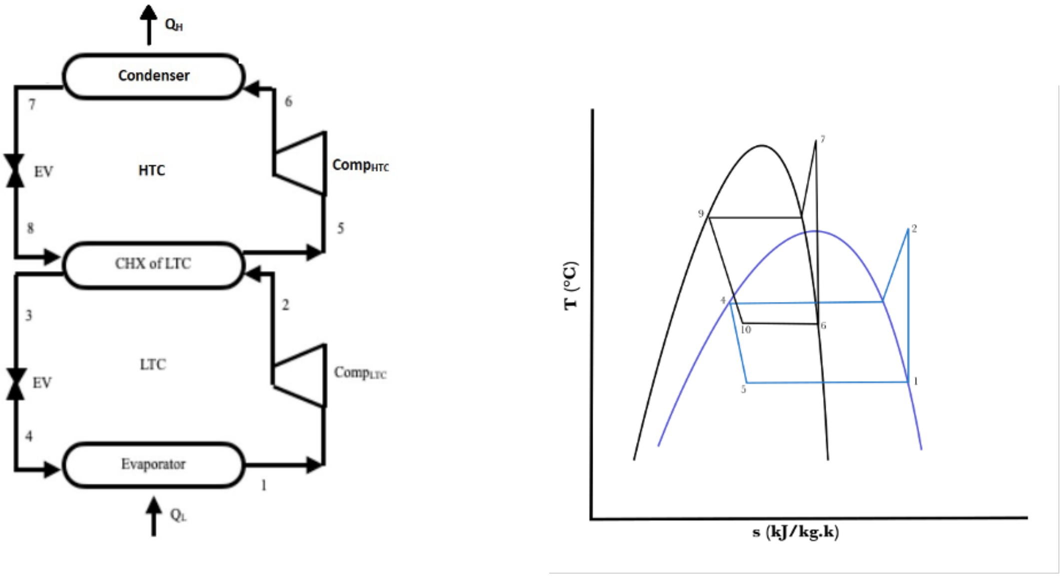
To counter these limitations the cascade refrigeration system is introduced. Cascade refrigeration system is the series connection of multiple refrigeration cycles with refrigerants having a progressively lower boiling point. As there are different cycles combined separately, there's an option to vary refrigerants in different stages according to cooling requirements.

The first usage of the cascade refrigeration system is found in 1877 which was built by Cailletet and Pictet [27] who used ethylene and SO_2/CO_2 separately to build the system. Then later on, in 1908 helium was liquefied using pre-cooling by cascade refrigeration cycles with air and hydrogen.

There are different types of cascade refrigeration systems. Most commonly used systems are, two stage cascade, three stage cascade, cascade absorption system, etc.

Two Stage Cascade Refrigeration System

The simple vapor compression refrigeration system discussed in section 2.2.1 is the most used refrigeration cycle as it's adequate for most of the common applications. Sometimes the required cooling effect can't be achieved using one cycle.



(a) Schematic diagram

(b) T-s diagram

Figure 2.10: Two-stage cascade Refrigeration system

In different processes, the difference between the two working pressure is large. This generates a problem for the compressor due to some limitations. For these reasons, the simple VCR system is modified with a double stage with two different VCR systems connecting them with a common heat exchanger, which is used as a condenser for the lower cycle and evaporator for the higher cycle. This system is especially recommended for industrial purposes. Messineo et al. [42] stated that, when the temperature difference between condensation and evaporation is too large for any single available refrigerant to execute the cycle, the cascade refrigeration system has to be used. In a cascade refrigeration system the same refrigerant can be used in both stages. Also, different refrigerants might be used according to the requirements.

In figure 2.10a, a schematic diagram of a two-stage cascade refrigeration system is shown. There are two VCR cycles used in series connecting them with a common heat exchanger in between. In this heat exchanger, the hot fluid comes from the lower circuit and the cold fluid comes from the higher circuit. There's a certain temperature difference restriction that has to be maintained to get effective heat transfer in the heat exchanger. The rest of the equipment works just like the simple vapor compression system. The lower circuit evaporator is in contact with the refrigerating space. On the other hand, the condenser is exposed to the environment.

For overall circuit, heat absorbed,

$$Q_L = m_L \times (h_1 - h_4)\ [\mathrm{kJ\,s^{-1}}] \tag{2.11}$$

Total Work,

$$W_L = W_{comp,LTC} + W_{comp,HTC}\ [\mathrm{kJ\,s^{-1}}] \tag{2.12}$$

Overall COP,

$$COP = \frac{Q_L}{W_{Total}} \tag{2.13}$$

Three Stage Cascade Refrigeration System

Typically, low-temperature systems of chemical, food, pharmaceutical industries have an evaporating temperature of -40°C to -100°C [27]. Other applications like cryogenics, gas liquefaction, etc. need evaporating temperature around -130°C. These low-temperature refrigeration systems will have very high-pressure ratios, which is not suitable for single/double cascade refrigeration system with limited compressor capabilities. To achieve these temperature range three stage cascade refrigeration system is a feasible system. The configuration is almost same as simple vapor compression refrigeration system, but instead of one cycle, three are connected in series. In between each cycle a heat exchanger is used, similar to double cascade refrigeration system.

In a standard refrigeration cycle, there is a compressor, a condenser, an evaporator, and an expansion valve are used. In the three-stage cascade refrigeration system(TCRS), three standard refrigeration cycles are joined together by cascade heat exchanger(CHX). Lower-temperature cycle (LTC) takes heat from the cooling load (Qe) from the cold space. Heat is transferred from LTC to medium-temperature cycle(MTC) through the CHX. For MTC, cooling load is the heat obtained from the CHX. Then again, heat is transferred from MTC to a higher-temperature cycle(HTC) through another CHX.

And finally, the heat is transferred from the HTC to the surroundings. Schematic diagram of a TCRS is given in figure: 2.11a.

For overall circuit, heat absorbed,

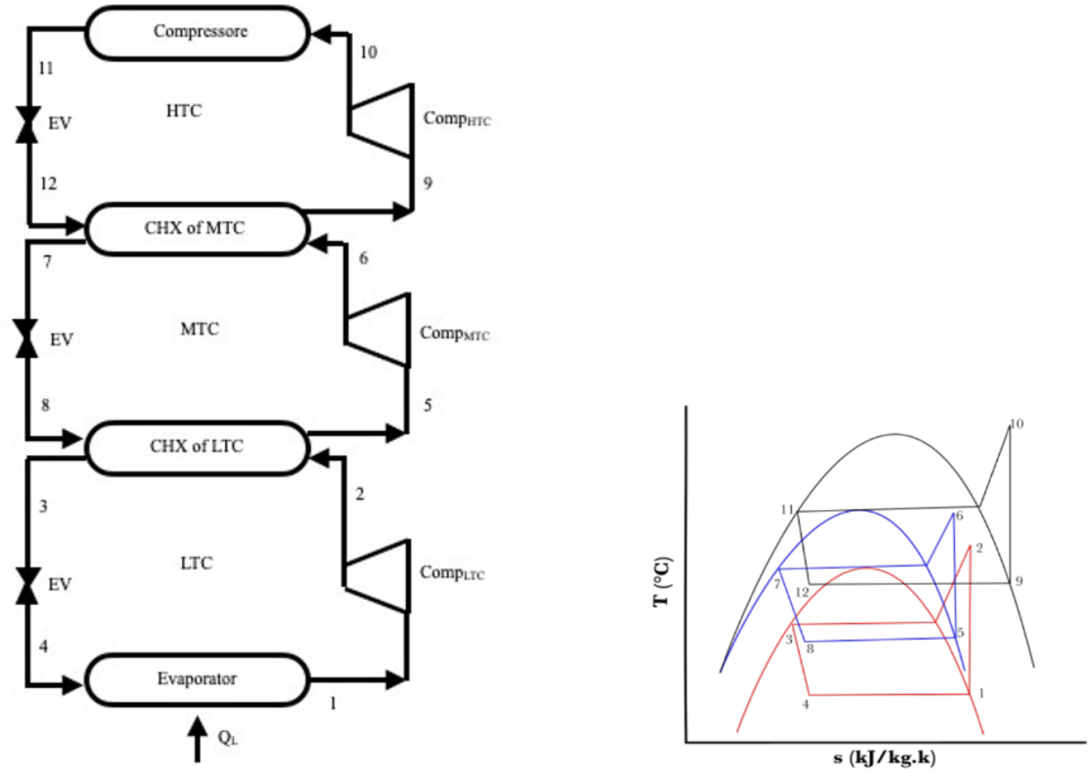
$$Q_L = m_L \times (h_1 - h_4)\ [\mathrm{kJ\,s^{-1}}] \tag{2.14}$$

Total Work,

$$W_L = W_{comp,LTC} + W_{comp,MTC} + W_{comp,HTC}\ [\mathrm{kJ\,s^{-1}}] \tag{2.15}$$

Overall COP,

$$COP = \frac{Q_L}{W_{Total}} \tag{2.16}$$



(a) Schematic diagram

(b) T-s diagram

Figure 2.11: Three-stage cascade Refrigeration system

2.3 Refrigerants

Refrigerants are the working fluids used as a heat transfer medium in refrigeration system. Performance of refrigeration system depends significantly on the thermo-physical and chemical properties of refrigerants. First refrigerant ether was employed in the hand operated vapor compression system by Perkins. In earlier days, ethyl chloride and ammonia were mostly used as refrigerant. Also, sulphur dioxide, methyl chloride and carbon dioxide were used around 1880s [2]. Around 1910-1930 N2O3, CH4, C2H6, C3H8 etc. were used for medium and low temperature systems[2].

Freon was developed in the 1930s by E.I. du Pont de Nemours and Co. Freon consists of hydrocarbon with fluorine or chlorine replacing hydrogen atoms. Using different combinations of carbon, hydrogen, chlorine, fluorine different refrigerants were developed later who are known as chloro-fluoro-carbon. They can be used for wide range of applications. Calm et. al. divided the refrigerant's development progression into four categories[9].

(i) First-generation – Whatever available

Common refrigerants of first generation were generally different solvents and volatile fluids. As the early refrigeration system's refrigerants were only selected on the basis of cooling effect, most of them were toxic and flammable. Also, they were highly reactive with the system materials. Some of the first-generation refrigerants are ether, CO_2, $HCOOCH_3$, SO_2, NH_3, CCl_4 etc. After some setbacks with these refrigerants (environmental and performance reasons) researches were initiated in search of alternative refrigerants.

(ii) Second-generation – Safety and Durability

After property table for different chemicals was developed in early 1900s, refrigerants were searched on the basis of toxicity and flammability with stable chemical properties. In 1930, Midgley and Henne showed the effect of varying amount of chlorination and fluorination in hydrocarbon-based refrigerants[43]. Midgley et. al. described the variation of boiling points, flammability and toxicity with the varying number of fluorine and chlorine in hydrocarbon-based refrigerants. Some second-generation refrigerants are, CFCs, HCFCs, HFCs, H_2O etc.

(iii) Third-generation - Ozone Protection

In the 2nd half of 20th century, new problem was detected. Leakage of CFC refrigerants from refrigeration systems was directly or indirectly influencing the depletion of ozone layer of the earth's atmosphere. Damage to this ozone layer around the earth is a huge environmental concern as it protects life on earth from different harmful rays coming from the sun. Later on, in Vienna convention and Montreal protocol strictly recommended the elimination of these types of refrigerants from being used in refrigeration systems. This protocol initiated the search for "Nutral Refrigerants". Phasing out ozone depleting refrigerants eventually provided a significant positive outcome, after the worst year in 1998[9].

(iv) Fourth-generation – Global Warming

When the ice of poles started melting, resulting in sea level rise, global warming became a major concern of environmentalists worldwide. There are different reasons behind this phenomenon. However, one of the most significant reasons is related to refrigerants. Restrictions were imposed due to governments and engineering manufacturers' environmental concerns in selecting refrigerants that have high global warming potential values. Synthetic refrigerants are being removed from usage in systems worldwide and are being replaced [55]. Some low global warming potential refrigerants shown by Calm et. al.[9] are, R-32, R-152a, R-161, R-31I1, R-600, R-290 etc.

2.3.1 Properties of Refrigerants

The performance of a refrigeration system mostly depends on the chemical and thermodynamic properties of refrigerant. The most important property of a refrigerant is boiling point temperature on which most of the other properties depend. Rotchana et. al. described the refrigeration system performance variation with the properties of refrigerant[50]. Some properties of refrigerants are described below-

1. Boiling Point

Boiling point is the temperature at which a chemical compound starts to change it's phase from liquid to gas. Higher boiling point of refrigerant will have higher latent heat of vaporization and on the contrary, lower boiling point will result in low heat of vaporization. Again, high boiling point refrigerants can have higher pressure ratios which is suitable for larger systems. So, boiling point is desired higher for refrigerant as they can provide more cooling load during operations.

2. Critical Temperature and Pressure

Critical temperature of a substance is the temperature above which it can't be converted to liquid. The pressure at this point indicates the critical pressure. For high coefficient of performance generally higher critical temperature of refrigerant is desired. Critical pressure should be low to ensured to maintain low condensing pressure.

3. Freezing Point

Freezing point is the temperature at which a fluid is solidify. As refrigerants are fluid flowing throughout the system, they must maintain their fluidity. So, during refrigerant selection it must be ensured that the refrigerant is working above it's freezing point which must be lower than the system temperature.

4. Flammability

Flammability of the refrigerant used must be lower. The ignition temperature of refrigerants must be very high. Otherwise, accidents may occur if system temperature exceeds the limit of flame generation of refrigerant.

5. Toxicity

Sometimes leakage may occur in the system. This might lead to different hazardous situations. So, during refrigerant selection toxicity level must be very low.

6. Reactivity

Refrigerants should not react with the component materials of the refrigeration system. This will result in corrosion and eventually leakage of fluid and failure of the system.

7. Viscosity

Lower viscosity is desired for refrigerants. Higher viscosity will result in resistance in flow inside the tubing.

8. Global Warming Potential

Global warming is a significant threat to the existence of our ecosystem. Global warming potential (GWP) indicates a substance's capability to trap heat inside itself, which contributes to increasing the earth's average temperature. The higher the value of GWP, the more it is a threat to the environment. Keeping that in mind, according to restrictions imposed by the environmental organizations , refrigerants should be selected based on low global warming potential.

9. Ozone Depletion Potential

Ozone depletion potential (ODP) is the ability of a substance to provide elements that will damage the ozone layer protecting the earth. Some refrigerants are highly responsible for harming the ozone layer around the earth. If the higher ODP substances are used, the ozone layer will get breached faster, which will lead to the infiltration of harmful rays inside the atmosphere. So, zero or low ozone depletion potential refrigerants should be used whenever possible.

2.3.2 Selection of Refrigerants

The refrigerants must be selected primarily on the basis of system temperature that has to be maintained. There are some characteristics that defines the refrigerant usage for a system. These are normal boiling point temperature and the critical temperature of the refrigerant. The system temperature range must be within the critical temperature and normal boiling point of the refrigerant[55], where evaporating temperature must be higher than normal boiling point temperature of the refrigerant. Luyben mentioned during refrigerant selection, the condenser pressure has to be kept less than 50% [36] of the selected refrigerant to get better output from the system. For design purpose viscosity, leakage property, reactivity should be kept as low as possible to keep the system sustainable and hazard free. For environmental concern the global warming potential and ozone depletion potential must be kept as low as possible.

Chapter 3

Methodology

When a very low-temperature (e.g. -40^{o}C to -150^{o}C) system needs to be handled, the pressure ratio becomes very high. This pressure ratio is challenging to handle using single or double refrigeration stages. Also, sequential cooling of refrigerant is needed to achieve this kind of range of temperature. Among different alternatives found in the literature, the triple cascade refrigeration system is the most suitable. The three compression stages can handle a vast pressure ratio and reach below -140^{o}C.

A triple cascade refrigeration system is a three-stage vapor compression system combined by two cascade heat exchangers. This specific type of refrigeration system is mostly applicable to achieve extremely low evaporation temperature (e.g. -40^{o}C to -150^{o}C). This system combines three VCR systems to achieve high COP for the low-temperature application rather than a single-stage VCR to achieve the same cooling using lower COP. This system is a complex refrigeration system in which two cascade heat exchangers play the most vital roles.

3.1 Thermodynamic Cycle

The triple cascade refrigeration system consisting of three VCR systems, combined by two CHX shown in fig. 3.1. In the lower temperature circuit, at state 1, the LTC working fluid is superheated vapour with the saturation pressure of the evaporator. This fluid goes into the compressor of LTC to be compressed to the pressure of the lower part of the heat exchanger. This compression process raises the temperature of the LTC working fluid by a large amount. At state 2, the superheated vapour enters the lower part of the cascade heat exchanger which acts as a condenser for the LTC. Heat is rejected to the MTC and thus, LTC refrigerant becomes saturated liquid at state 3. From state 3 to state 4, it undergoes a throttling process in an expansion valve and reaches evaporation pressure. At state 4, it is a saturated mixture of low quality which goes into the evaporator and absorbs heat from the refrigerated space. After absorbing the heat, LTC refrigerant starts to evaporate with some degree of superheating and it becomes superheated vapour at state 1. It enters the LTC compressor again and the cycle is repeated.

In the mid temperature circuit, the CHX between LTC and MTC acts as an evaporator for the MTC. Heat is absorbed by the MTC refrigerant and with some degree of superheating, it becomes superheated vapour at state 5. It enters the MTC compressor and is compressed to state 6 which is the pressure of the MTC condensation. At state 6, the refrigerant enters the 2nd CHX and it is condensed to the saturated liquid off state 7 by rejecting heat to the HTC. From state 7 to 8, it goes through a throttling process in an expansion valve and reaches the evaporation pressure of the MTC. From state 8 to 5, it starts to absorb heat again from the 1st CHX and the cycle continues.

In the high temperature circuit, the CHX between MTC and HTC acts as an evaporator for the HTC. Heat is absorbed by the HTC refrigerant and with some degree of superheating, it becomes superheated vapour at state 9. It enters the HTC compressor and is compressed to state 10 which is the pressure of the HTC condensation. At state 10, the refrigerant enters the condenser and it is condensed to the saturated liquid off state 11 by rejecting heat to the surroundings. From state 11 to 12, it goes through a throttling process in an expansion valve and reaches the evaporation

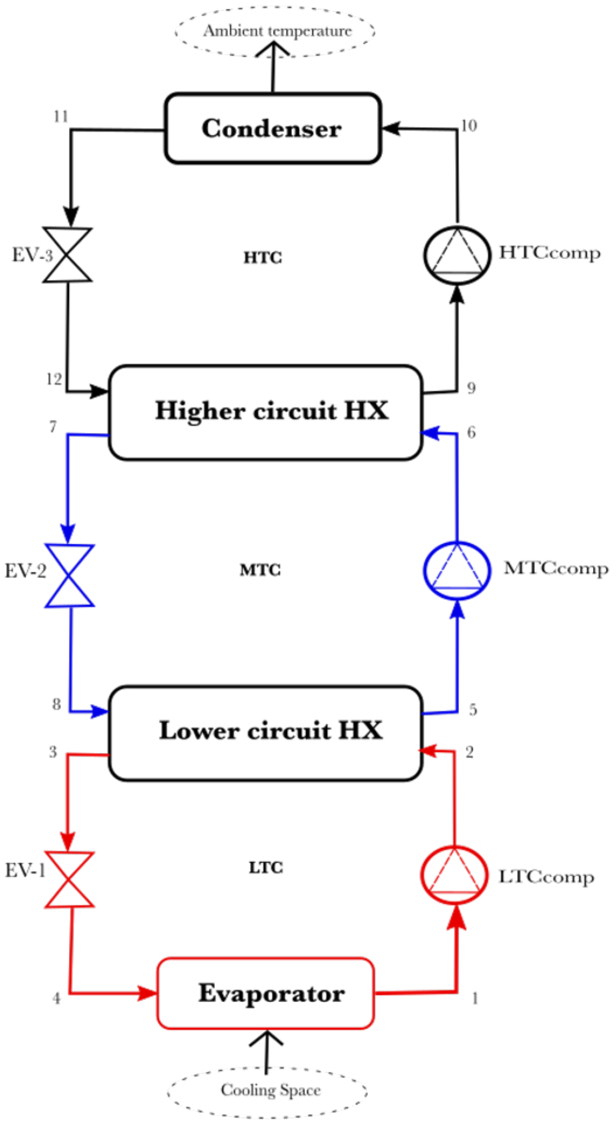


Figure 3.1: Schematic diagram of TCRS.

pressure of the HTC. From state 12 to 9, it starts to absorb heat again from the 2nd CHX and the cycle continues.

Overall, the heat absorbed by the evaporator of the LTC section is rejected to the surroundings by going through three stages of VCR system to reduce the total work of the compressors. By doing this, total power consumption of all three compressors is less than the power consumption of a single stage compressor to achieve the same cooling effect.

3.2 Energy Analysis

Energy analysis of a system is represented by showing a system's performance from the perspective of the 1st law of thermodynamics. This analysis is the 1st step for a researcher to understand a new system's performance and to confirm if this system is important enough to be explored further. If a new system gives better performance under a particular set of conditions compared to an existing one, one can replace that system with the new one from the 1st law perspective. The main concern of energy analysis for a refrigeration system is to find the overall COP of the system. COP of the refrigeration parameter's coefficient of performance shows the desired cooling effect of a system over the provided work input. In TCRS, four types of COP can be calculated.

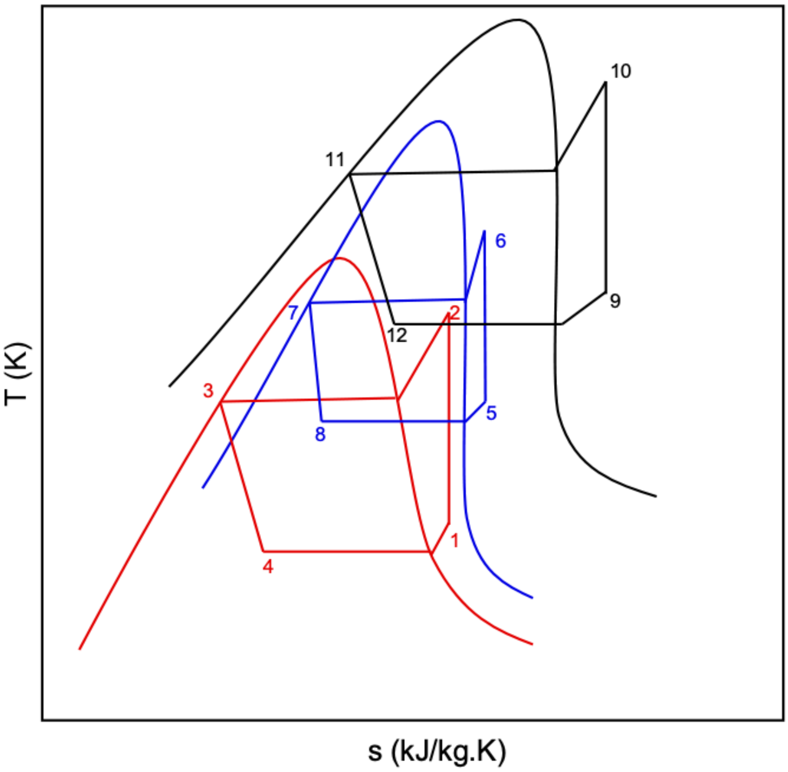


Figure 3.2: T-s diagram of TCRS.

At first, COP of the lower temperature circuit (LTC) should be calculated as it is directly connected with the cooling load, and then COP of the MTC and COP of the HTC is calculated. At the end of the energy analysis, overall COP is calculated, which gives the system's idea from the 1st law of thermodynamic analysis.

According to the basic principle of TCRS, a theoretical model is established and a set of assumptions are considered to simplify the analysis. Those assumptions are:
1) The superheating of refrigerant gas is effective heating, and there is no subcooling for liquid refrigerant.
2) The pressure drop and heat losses/gains of the refrigerant flowing inside the pipelines and system components are all negligible.
3) All system components are assumed to be in a steady-state and steady-flow process. The changes in kinetic and potential energy of the components are negligible.
4) Gas leakage is negligible in all the joints connecting the pipes to each component.
5) The compression and expansion processes are adiabatic.
6) The temperature difference ΔT in the cascade heat exchanger is 5°C.

The T-s diagram shown in Fig. 3.2 illustrates the thermodynamic processes of the cascade refrigeration system. The above assumptions are made to avoid the complexity in the analysis as the scope of this article is to provide a thermodynamic analysis using organic refrigerants. The CHX is considered as an ideal one to simplify the fact that heat transfer can be varied by the effectiveness of a heat exchanger. No losses are considered for the heat exchanger to avoid complexity as there are numerous types of design parameters to consider designing a heat exchanger.

mass flow rate of LTC:

$$m_L = \frac{Q_e}{h_1 - h_4} [\mathrm{kg\,s^{-1}}] \tag{3.1}$$

the compressor work of LTC is:

$$W_{compL} = m_L \times \left(\frac{h_{2s} - h_1}{\eta_s} \right) \; [\mathrm{kJ\,s^{-1}}] \tag{3.2}$$

In equation 2, η is is isentropic efficiency of the pump and it is taken as 0.8 for all pumps of LTC, MTC, HTC.

The heat load of 1st cascade heat exchanger is:

$$Q_{cas,1} = m_L \times (h_2 - h_3)\,[\mathrm{kJ\,s^{-1}}] \tag{3.3}$$

or,

$$Q_{cas,1} = m_M \times (h_5 - h_8)\,[\mathrm{kJ\,s^{-1}}] \tag{3.4}$$

COP of LTC is:

$$COP_L = \frac{Q_e}{W_{compL}} \tag{3.5}$$

mass flow rate of MTC is:

$$m_M = \frac{m_L \times (h_2 - h_3)}{(h_5 - h_8)}[\mathrm{kg\,s^{-1}}] \tag{3.6}$$

the compressor input of MTC is:

$$W_{compM} = m_M \times \left(\frac{h_{6s} - h_5}{\eta_s}\right)[\mathrm{kJ\,s^{-1}}] \tag{3.7}$$

the heat load of 2nd heat exchanger is:

$$Q_{cas,2} = m_M \times (h_6 - h_7)\ \ [\mathrm{kJ\,s^{-1}}] \tag{3.8}$$

or,

$$Q_{cas,2} = m_H \times (h_9 - h_{12})\ \ [\mathrm{kJ\,s^{-1}}] \tag{3.9}$$

COP of MTC is:

$$COP_M = \frac{Q_{cas,1}}{W_{compM}} \tag{3.10}$$

mass of HTC is:

$$m_H = \frac{m_M \times (h_6 - h_7)}{(h_9 - h_{12})}[\mathrm{kg\,s^{-1}}] \tag{3.11}$$

the compressor input work of HTC is:

$$W_{compH} = m_H \times \left(\frac{h_{10s} - h_9}{\eta_s}\right) [\mathrm{kJ\,s^{-1}}] \tag{3.12}$$

COP of HTC is:

$$COP_H = \frac{Q_{cas,2}}{W_{compH}} \tag{3.13}$$

Total compressor work is:

$$W_{comp} = W_{compL} + W_{compM} + W_{compH} [\mathrm{kJ\,s^{-1}}] \tag{3.14}$$

Overall COP of the system is:

$$COP = \frac{Q_e}{W_{comp}} \tag{3.15}$$

3.3 Exergy Analysis

Exergy analysis of a thermodynamics shows the available energy to produce work for a specific state. Exergy of a point can be stated as how much work can be done if that state point is moved to the dead state condition. Exergy analysis also shows that how much energy loss is occurring at a components. To achieve higher efficiency and better economical design, exergy analysis of a cycle is crucial.

Exergy analysis of cycle can be done either by considering the specific exergy across each component or the exergy destruction across each component. In this thesis, exergy destruction was considered to understand the loss of exergy occurring at the components.

Exergy destruction at LTC compressor:

$$X_{LTC,comp} = T_a \times m_L \times (s_2 - s_1) [\mathrm{kJ\,s^{-1}}] \tag{3.16}$$

Exergy destruction at LTC expansion valve:

$$X_{LTC,EV} = T_a \times m_L \times (s_4 - s_3)\,[\mathrm{kJ\,s^{-1}}] \tag{3.17}$$

Exergy destruction at evaporator:

$$X_{eva} = T_a \left[m_L\,(s_1 - s_4) - \frac{Q_e}{T_e + \Delta T} \right]\ [\mathrm{kJ\,s^{-1}}] \tag{3.18}$$

Exergy destruction at 1st cascade heat exchanger:

$$X_{Cas,1} = T_a \times [m_L \times (s_3 - s_2) + m_M \times (s_5 - s_8)]\ \ [\mathrm{kJ\,s^{-1}}] \tag{3.19}$$

Exergy destruction at MTC compressor:

$$X_{MTC,comp} = T_a \times m_M \times (s_6 - s_5)\,[\mathrm{kJ\,s^{-1}}] \tag{3.20}$$

Exergy destruction at MTC expansion valve:

$$X_{MTC,EV} = T_a \times m_M \times (s_8 - s_7)\,[\mathrm{kJ\,s^{-1}}] \tag{3.21}$$

Exergy destruction at 2nd cascade heat exchanger:

$$X_{Cas,2} = T_a \times [m_M \times (s_7 - s_6) + m_H \times (s_9 - s_{12})]\ \ [\mathrm{kJ\,s^{-1}}] \tag{3.22}$$

Exergy destruction at HTC compressor:

$$X_{HTC,comp} = T_a \times m_H \times (s_{10} - s_9)\,[\mathrm{kJ\,s^{-1}}] \tag{3.23}$$

Exergy destruction at HTC expansion valve:

$$X_{HTC,EV} = T_a \times m_H \times (s_{12} - s_{11})\ \ [\mathrm{kJ\,s^{-1}}] \tag{3.24}$$

Exergy destruction at condenser:

$$X_{cond} = T_a \times \left[m_H \times (s_{11} - s_{10}) + \frac{Q_K}{T_a} \right] [\mathrm{kJ\,s^{-1}}] \tag{3.25}$$

So, total exergy destruction at the cycle is:

$$X_{des} = \sum X_i \ [\mathrm{kJ\,s^{-1}}] \tag{3.26}$$

Exergy efficiency of the TCRS system can be defined as the ratio of minimum work input to the provided actual work input.

$$\eta_{exergy} = \frac{W_{comp} - X_{des}}{W_{comp}} \tag{3.27}$$

Also, second law efficiency can be calculated by the following equation:

$$\eta_{II} = \frac{COP}{COP_{Carnot}} \tag{3.28}$$

and Carnot efficiency of the TCRS system:

$$COP_{Carnot} = \frac{T_e}{T_k - T_e} \tag{3.29}$$

3.4 Refrigerants Selection

Choosing refrigerants is dependent on some variable like critical pressure, boiling temperature, triple point temperature. Those parameters are considered from a thermodynamic perspective to achieve higher efficiency. From an economical perspective, ODP and GWP should be lower for environmentally friendly design.

During earlier phase of refrigeration selection, all the available refrigerants from different literature were listed. Special emphasis was given on refrigerants used in double cascade refrigeration systems. 101 refrigerants were listed without imposing any restrictions of critical temperature and pressure. Secondary screening of refrigerants was conducted on the basis of ODP and GWP. From primary list, the refrigerants which had higher ODP and GWP were eliminated. For the purpose of current study, the refrigerants which fulfilled the critical pressure and temperature requirements were listed and later on separated on the basis of different pressure stage of triple cascade refrigeration system. Finally, only organic refrigerants were selected for the purpose of this study. Organic refrigerants were selected because they have zero ODP and low GWP (20). Also, they are proved to be very useful in organic Rankine cycle in different literature.

	Substances	**Boiling temp.**	**Critical temp.**	**critical pressure**
LTC	1butene	266.84 (K)	419.29 (K)	4.01 (MPa)
MTC	trans-2-butene	274.03 (K)	428.61 (K)	4.03 (MPa)
	n-Heptane	371.53 (K)	540.13 (K)	2.74 (MPa)
	cis-2-butene	276.87 (K)	435.75 (K)	4.23 (MPa)
	toluene	383.75 (K)	591.75 (K)	4.13 (MPa)
HTC	mXylene	412.15 (K)	616.89 (K)	3.5 (MPa)

Table 3.1: Physical data for hydrocarbon refrigerants 'selected' , [64]

In current study, organic refrigerants were used to find better COP than existing literatures as they have very low GWP and ODP which is environmentally friendly and economically justified. They were selected due to their following characteristics:

- They have moderate critical parameters
- Considerably high specific heat
- Moderate evaporating and condensing pressures
- Exceptional heat transfer characteristics
- Lack of side effects

- Lower market price

In this study, organic refrigerants were used. ODP and GWP of the organic refrigerants were taken from different literatures and GWP is 20 for all refrigerants and they have zero ODP. Refrigerants that are selected for LTC and HTC were considered after plotting pressure ratio, the compressor work and which combination gives the maximum COP output.

3.5 Design and Simulation

Designing a cycle from a thermodynamic perspective includes evaluating all the points of the working fluids. Two independent parameters are needed to determine all the necessary properties of each state to simulate a thermodynamic analysis. The simulation code for the TCRS was conducted in Python 3.0. Thermodynamic properties of the refrigerants were taken from CoolProp 6.4. CoolProp [5] can be easily integrated to Python using a function provided in the official website of CoolProp to make it easy to determine the thermodynamic properties of each state by calling the refrigerant's name.

The parameters considered to simulate the cycle is summarised below.

Parameters	**Values**	**Parameters**	**Values**
Qe (kW)	10	T_{lc} (°C)	-80 to -45
ΔT (°C)	5	T_{mc} (°C)	-40 to -0
T_a (°C)	25	Superheating at HTC (°C)	5
T_k (°C)	40	Superheating at MTC (°C)	12
T_e (°C)	-100	Superheating at HTC (°C)	12

Table 3.2: Basic parameter values of simulation model

Cooling load for the refrigeration was considered 10 kW, evaporation temperature was taken -100°C and cascade temperature difference was 5K. A flow chart is shown in Fig. 3.3 to show the steps of the simulation.

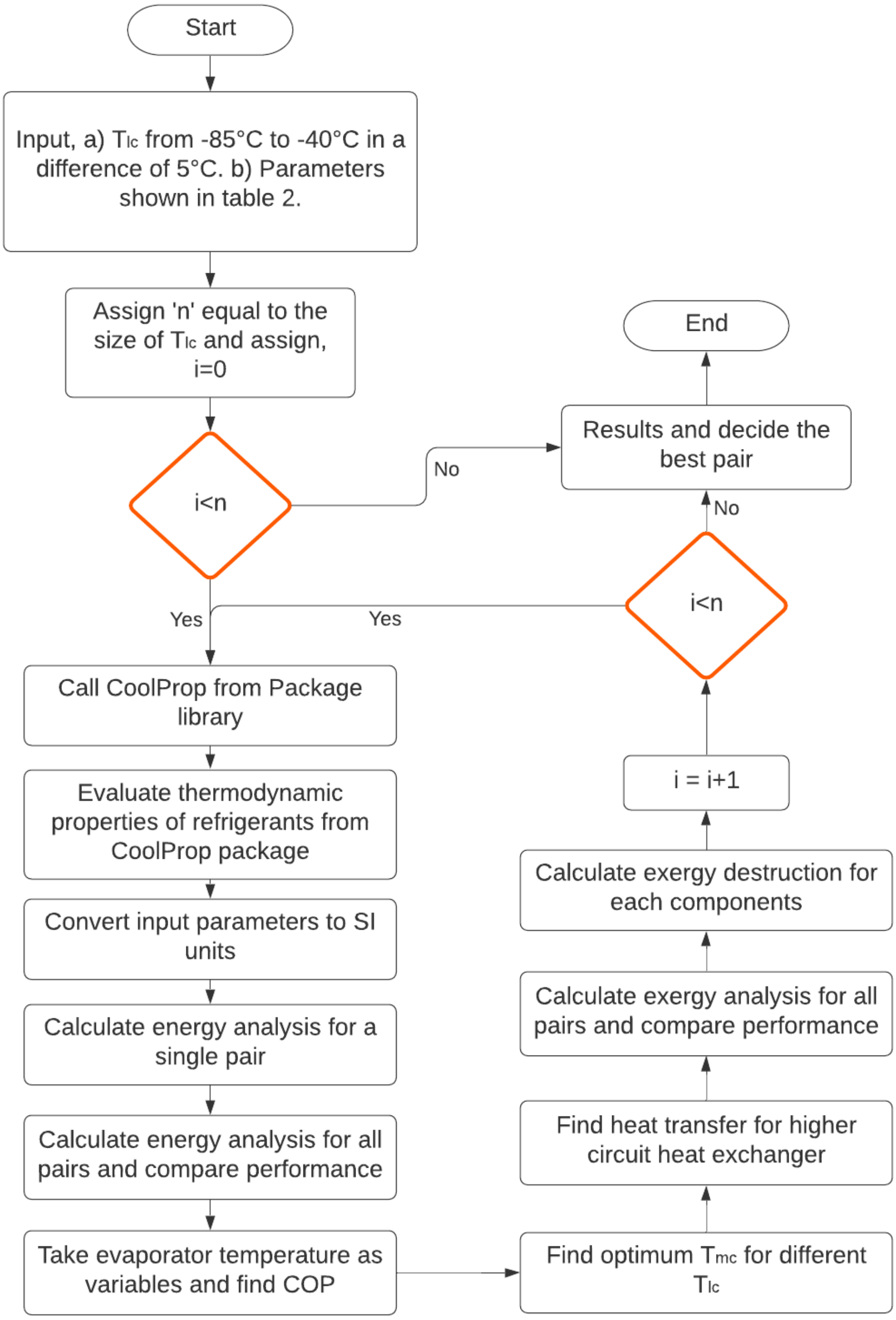


Figure 3.3: Flow chart of the simulation model.

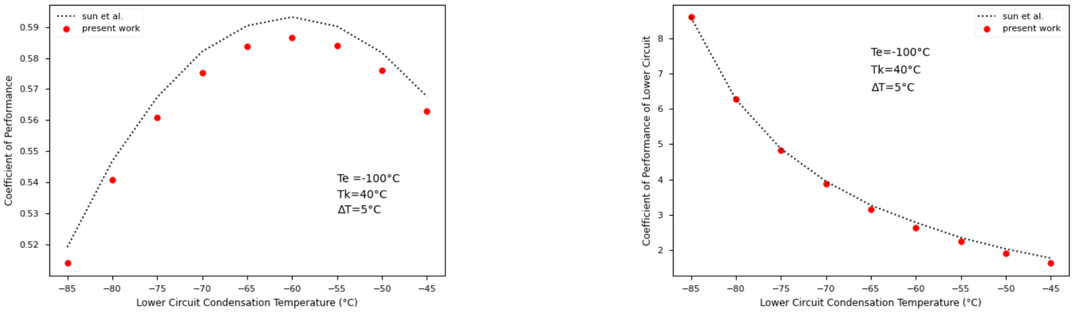
3.6 Model Validation

The simulation model was validated against the work done by Sun el al. [58]. In their study they showed that R1150/R170/R161 has better performance than other low GWP and low ODP refrigerants. Based on that information, the simulation code was validated by using the same refrigerant pair and it can be seen from fig.04 that the accuracy of the simulation model is acceptable with error less than 2 percent. One of the key reasons for the error is that, in the simulation model, COOLPROP 6.4 was used to find the thermodynamic properties of the refrigerant where REFPROP 8.0 was used in the Zhili Sun et al. [58]. For validation purposes, the simulation model was conducted by using the same parameters that are stated in the Zhili Sun et al. [58] and that is, evaporator temperature -100°C, cascade temperature difference 5°C, higher circuit condensation temperature 40°C and ambient temperature 25°C.

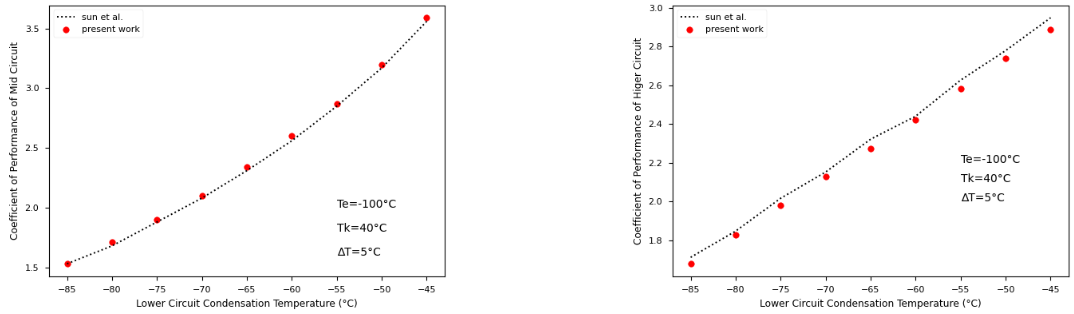
T_{lc} (°C)	Present work	Sun et al. [58]	Error (%)
-85	0.513995	0.5191	0.983443
-80	0.540667	0.5468	1.121678
-75	0.560891	0.5674	1.14709
-70	0.575336	0.5821	1.162024
-65	0.583783	0.5904	1.120795
-60	0.583783	0.5904	1.120795
-55	0.584008	0.5902	1.049208
-50	0.576010	0.5817	0.97815
-45	0.562902	0.5676	0.827781
-40	0.513995	0.5191	0.983443

Table 3.3: Error table.

From the given table above, slight difference between present work and work of Sun et al. [58] is noticeable. For validation, COP values of Sun et al. [58]. were extracted using an open-source online software "Web Plot Digitizer". The calibration process of the graph heavily depends on the accuracy of the input graph image, which makes it difficult to gain pin point accuracy while extracting values from the graph. Unless the author provides accurate data table, it is quite impossible to find the exact value. Although the simulation values found were within 2% of the web plot digitizer extracted values, which met the validation requirements.



(a) Overall COP validation (b) Lower circuit COP validation



(c) Mid circuit COP validation (d) Higher circuit COP validation

Figure 3.4: Validation of Present work with Sun et al.[58]

Chapter 4

Results and Discussion

Energy and exergy analysis has been conducted to evaluate the performance of hydrocarbon (HC) refrigerants on TCRS and to select the best refrigerant pair between the stated pairs. In a TCRS, most significant components are the cascade heat exchangers between LTC-MTC and MTC-HTC because they work as both evaporator and condenser for individual circuits. However, in this study, condensation temperature of LTC was taken as the prime variable to analyze the system to simplify the understanding. But, condensation temperature of MTC is dependent on the condensation temperature of LTC and finding the relation between them is the 1st step to calculate the COP of the system. After finding the relation, other parameters are analyzed to evaluate the performance of the TCRS such as Individual circuit COP, Overall COP, COP change with respect to evaporator temperature, Total Compressor Work with respect to T_{lc} and Total Compressor work with respect to T_e.

4.1 Energy Analysis

COP indicates the performance of a refrigeration system. It is the most important parameter to find out effectiveness of the system with respect to the input provided. However, evaluating COP of an TCRS system is dependent on multiple parameters in which T_{lc} and T_{mc} are the most important ones. In addition to that, T_{lc} and T_{mc} are also dependent on each other. For a certain range of T_{lc} optimum T_{mc} is different for each pair of refrigerants. Fig 4.1 shows optimum T_{mc} for a certain range of Tlc. Fig 4.1 represents that, for each stated pair of refrigerant optimum T_{mc} will be different and this optimum T_{mc} will give optimum COP of a TCRS. From Fig 4.1, it is clear that 1-butene/toluene/mXylene has the highest optimum Tmc temperature which eventually leads this pair to have the lowest COP among the stated pairs as the TCRS will work for a larger temperature range in the MTC. This phenomenon causes the MTC to have lower COP, leading to a lower overall COP for the pair.

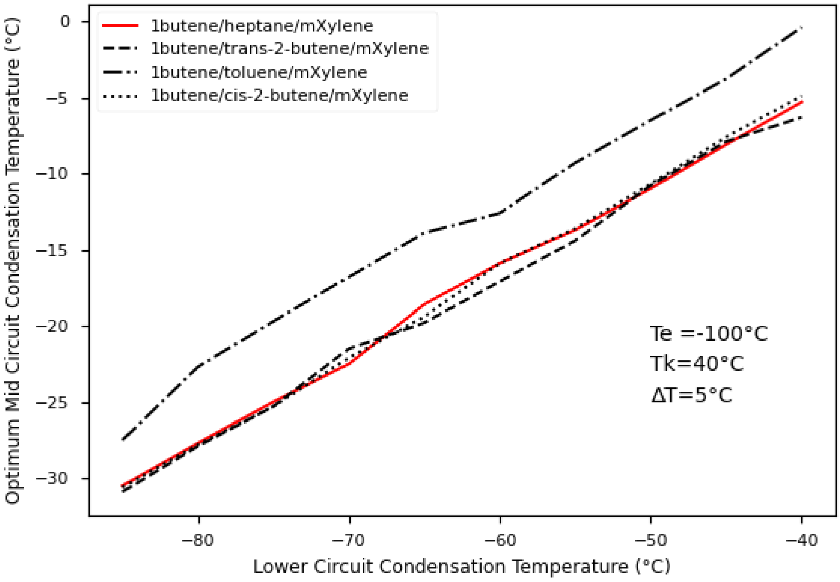


Figure 4.1: Relation between optimum T_{mc} and T_{lc}.

T_{lc} (°C)	-85	-80	-75	-70	-65	-60	-55	-50	-45	-40
t_{mc} (°C)	-30.5	-27.7	-25	-22.5	-18.6	-15.9	-13.7	-11	-8.1	-5.3

Table 4.1: Change of T_{mc} with T_{lc} at evaporation temperature of -100°C.

Table 4.1 shows the optimum T_{mc} of 1-butene/Heptane/mXylene pair for T_{lc} temperature. Based on this table, Fig 4.2 represents the behavior of the TCRS in terms of performance. From Fig 4.2, it is seen that, with the increase of T_{lc}, COP_L decreases but COP_M and COP_H increases. This is justified because increasing T_{lc} causes LTC to work for a higher temperature range from evaporation to condensation while MTC and HTC start to work for a smaller temperature range. This leads LTC to work in between higher-pressure ratio and MTC and HTC to work in smaller pressure ratio. Hence, increasing the lower circuit condensation temperature has opposite effects on LTC with respect to MTC and HTC. This phenomenon causes overall COP of a TCRS to increase with the increase of T_{lc} up to a certain point of T_{lc} temperature but starts to decrease after reaching maximum point (also shown in Fig 4.3).

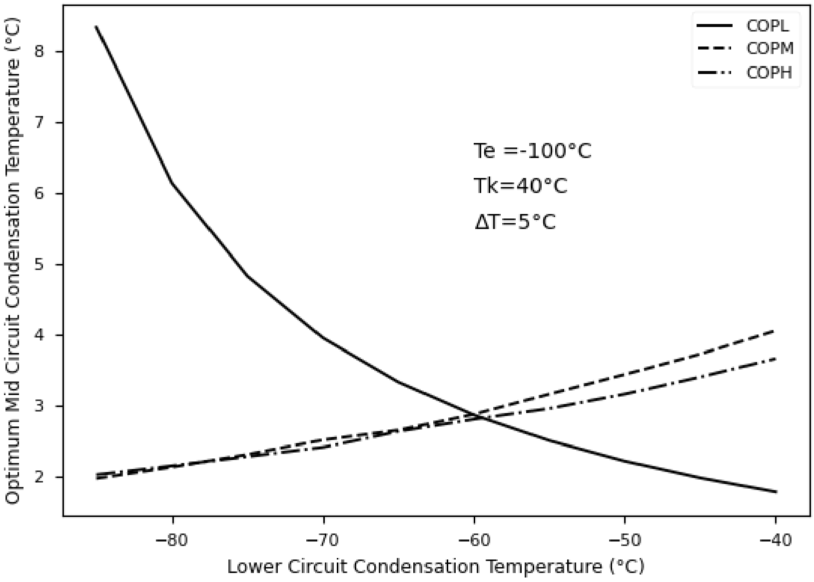


Figure 4.2: Effect lower circuit condensation temperature on circuit COP.

COP of the system is also dependent on the evaporation temperature (T_e). Fig 4.4 shows the relation between T_{lc} and COP, when Te is changed from -150°C to -100°C with a difference of 10°C. From the figure, it is clear that with increase of evaporation temperature, COP increases too. This is justified because the system starts to work on small temperature differences between T_e to T_k. The total compressor work gets lower as they start to work between lower pressure ratios according

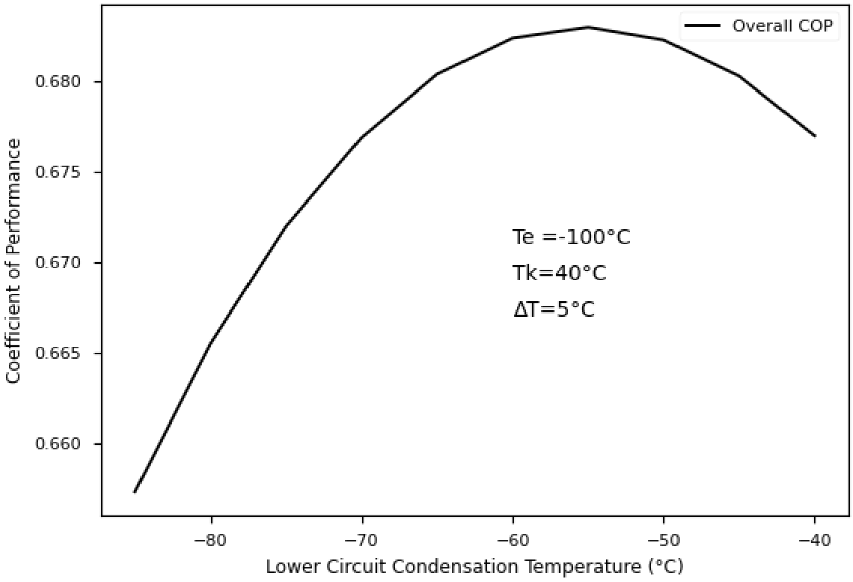


Figure 4.3: Effect of lower circuit condensation temperature on overall COP.

to the T-s diagram of the TCRS. Fig 4.4 was plotted for 1-butene/Heptane/mXylene refrigerant pair of the TCRS.

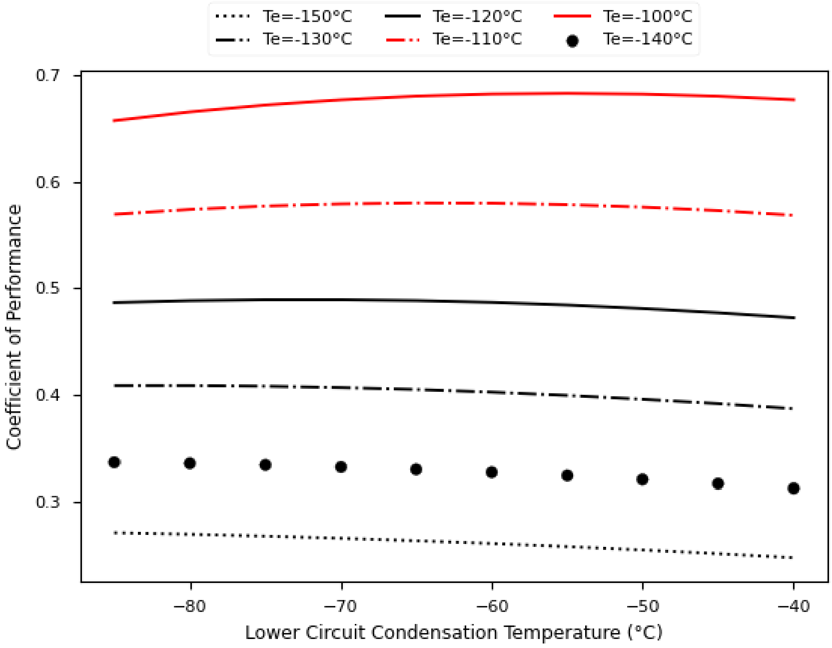


Figure 4.4: Effect of evaporation temperature and lower circuit condensation temperature on overall COP.

After running the simulation code for the refrigerant pairs stated in table 3.1, it is observed in Fig 4.5 that 1-butene/Heptane/m-Xylene gives a maximum COP of 0.683 at the lower circuit temperature of -55°C and 1-butene/toluene/m-Xylene gives a maximum COP of 0.6798 at -60°C of T_{lc}. Fig 4.5 shows that 1-butene/Heptane/m-Xylene gives the best overall COP out of other stated refrigerant pairs in table 3.1. For all the stated refrigerant pairs, it is seen that COP increases and then decreases with respect to the lower circuit condensation temperature. This lower circuit condensation temperature is an essential factor in achieving the highest COP. From Fig 4.5, it is also observed that all the pairs give approximately the same value at high T_{lc}. Finally, it is visible that 1- butene/Heptane/m-Xylene gives the best COP output for the TCRS as an HC refrigerant pair. The reason for better COP of 1- butene/Heptane/mXylene can be shown from Fig 4.6.

With the increase of lower circuit condensation temperature, total compressor work starts to decrease at first, and after reaching a particular T_{lc}, it again starts to rise (Fig 4.6). This particular behavior is that with T_{lc} increasing, the compressor

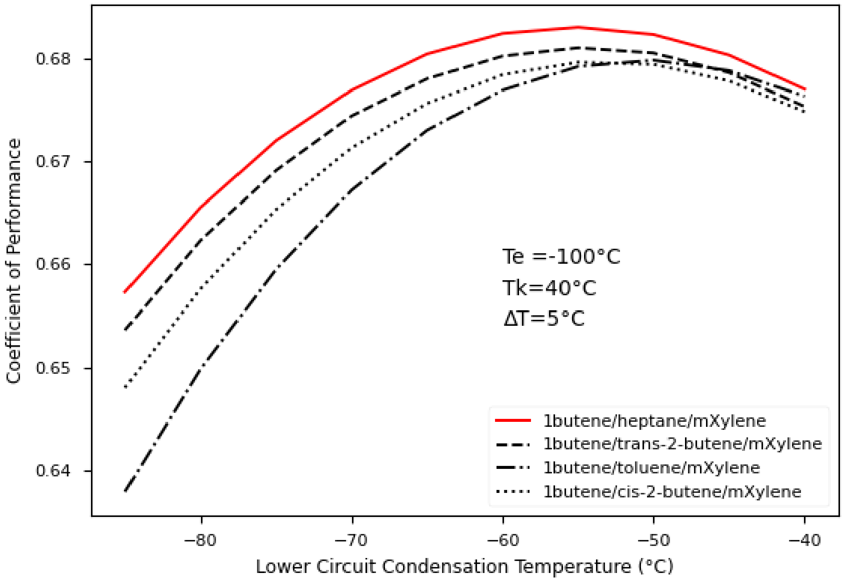


Figure 4.5: Effect of lower circuit condensation temperature on overall COP.

of the LTC starts to work between higher pressure and temperature ratio where, the compressor of MTC and HTC starts to work between smaller pressure and temperature ratio. So, the increase of T_{lc} also causes the opposite effect for compressor work of LTC with respect to MTC and HTC. Due to this, total compressor work starts to decrease at first, and then it rises again with the temperature increase. This behavior of total compressor work also exploits the curve of COP shown in Fig 4.5. Fig 4.6 shows that the 1- butene/Heptane/mXylene pair takes the lowest compressor work to give the desired cooling effect.

As stated before, heat transfer in CHX plays a crucial role in a cascade system. Both CHX of LTC and HTC control the performance of the triple cascade refrigeration system.Heat transfer of CHX in LTC is the same for all pairs as 1-butene is a common refrigerant for all pairs. Fig 4.7 represents the heat transfer of cascade heat exchangers between HTC and MTC. Fig 4.7 also represents the lowest and highest heat transfer of CHX between MTC and HTC. Fig 4.7 shows that again 1-butene/toluene/m-Xylene pair has the highest heat transfer rate in CHX, leading to the lowest COP and highest compressor work were seen in Fig 4.5 and Fig 4.6, re-

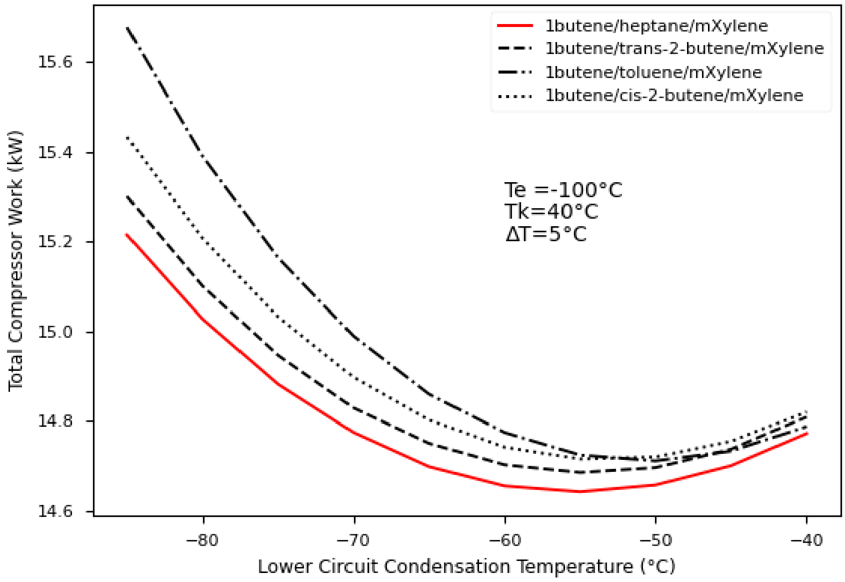


Figure 4.6: Effect of lower circuit condensation temperature on total compressor work.

spectively. Furthermore, 1-butene/Heptane/m-Xylene gives the lowest heat transfer rate in CHX, and due to this, this particular pair gives the highest COP and lowest compressor work.

4.2 Exergy Analysis

Quantity of energy in and out of the system can be measured using the first law of thermodynamics. This law serves as a vital tool for analyzing a system. However, from the first law perspective, the quality of energy cannot be understood. In analyzing the quality of extracted energy, the second law analysis proves to be more useful. Second law analysis analyzes the entropy generation and degradation of energy during different processes to be more specific. It also determines the lost opportunity of energy conversion. The term which refers to the lost opportunity to do work is called exergy destruction. Exergy analysis of individual components can indicate where most of the losses are encountered and find improvements. In

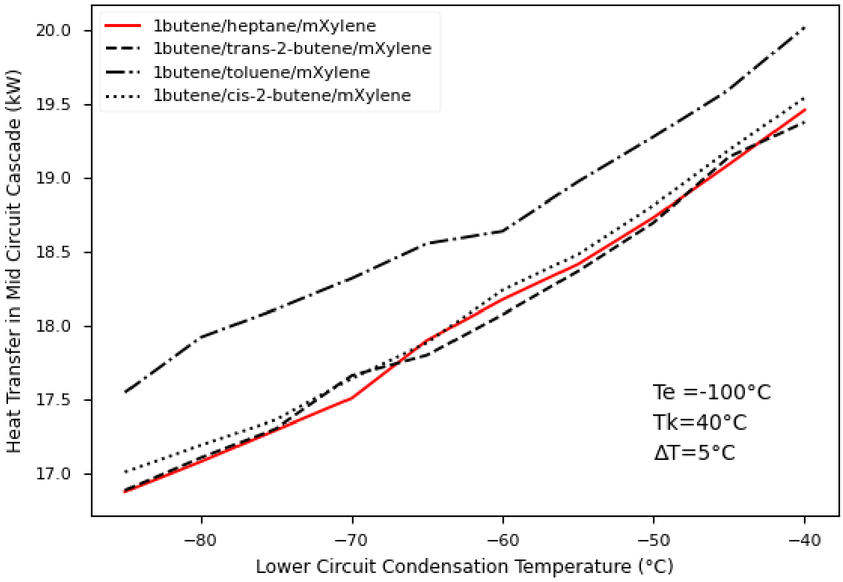


Figure 4.7: Effect of lower circuit condensation temperature on heat transfer of CHX in MTC.

this study, the second law analysis of the current system is conducted for different refrigerants' pairs. Exergy destruction of each individual component is analyzed to determine the component responsible for most of the losses associated with the system.

Fig 4.8 indicates the effect of condensation temperature of LTC on the second law efficiency. The second law efficiency increases with increasing Tlc and becomes maximum at a specific LTC condensation temperature. In this case, the evaporation temperature was fixed at -100°C. For evaporation temperature -100°C, each refrigerant pair shows an optimum Tlc value, which gives the highest second law efficiency. Overall circuit Carnot COP stays constant as the two extreme temperatures are fixed in this study.

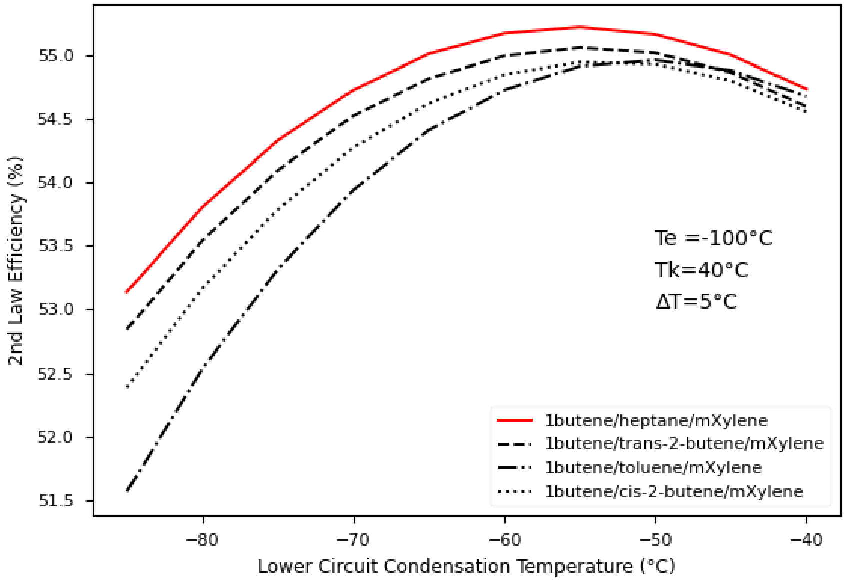


Figure 4.8: Variation of 2nd Law efficiency with condensation temperature of LTC.

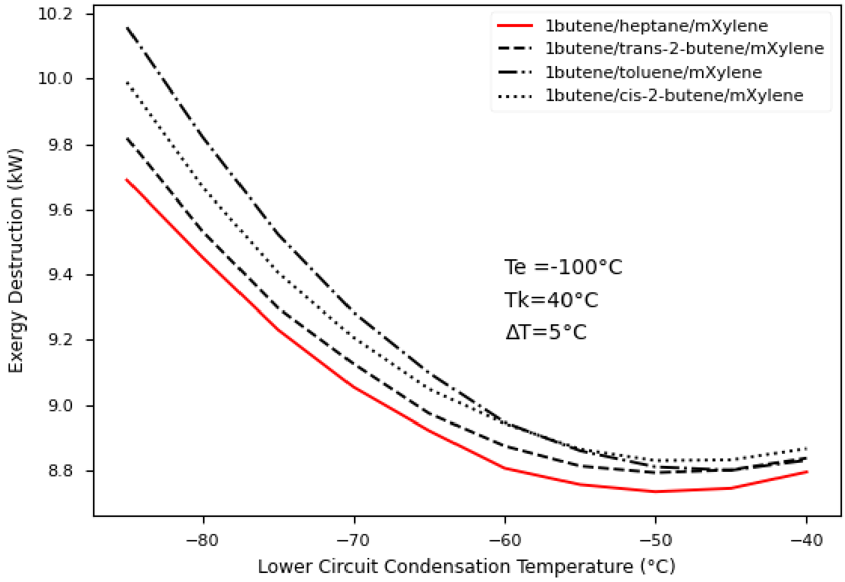


Figure 4.9: Variation of exergy destruction with condensation temperature of LTC.

The second law efficiency depends on net compressor work. 4.6 shows that with the increasing T_{lc}, the LTC compressor work will increase, and MTC compressor work will decrease simultaneously. As a result, the network will reduce, which will lead to increasing second law efficiency with increasing T_{lc}. After a particular value of T_{lc}, the compressor work of LTC becomes so large that it suppresses the advantage of reduced work of MTC. This phenomenon results in a reduction of second law efficiency with increasing T_{lc}. Fig 4.07 shows that 1-butene/n-Heptane/m-Xylene gives the highest value of second law efficiency, 55% at -55°C of T_{lc}. For T_{lc} -85°C to -55°C second law efficiency increases from 53.139% to 55.217%. Then up to -40°C, it reduces to 54.731%.

Similarly for pair 1-butene/trans-2-butene/m-Xylene, from -85°C to -55°C second law efficiency increases from 52.843% to maximum value of 55.055% and reduces to 54.594% after -55°C up to -40°C. Again, for pair 1-butene/cis-2-butene/m-Xylene, from -85°C to -55°C second law efficiency increases from 52.387% to maximum value of 54.942% and reduces to 54.553% after -55°C up to -40°C. Finall, for pair 1-butene/toluene/m-Xylene, from -85°C to -55°C second law efficiency increases from 51.570% to maximum value of 54.909% and reduces to 54.675% after -55°C up to -40°C.

Fig 4.9 confirms that the total exergy destruction reduces with the increase of T_{lc}. The reason behind the reduction of exergy destruction is that with the increase of T_{lc} the pressure ratio and expansion ratio of the MTC start to decrease. This phenomenon leads to the reduction of exergy destruction associated with compression, and expansion starts to reduce. Besides, the temperature difference between the condenser of MTC and the first cascade heat exchanger reduces. Due to all these, the overall exergy destruction reduces. The minimum value is achieved at -50°C for all the pairs except for the pair with toluene. Pair with toluene achieves lowest exergy destruction at -45°C. The lowest value of total exergy destruction is 8.7334 kW for n-Heptane in MTC, 8.7921 kW for trans-2-butane in MTC, and

8.8293 kW for cis-2-butane and 8.7995 kW toluene in MTC. After the minimum values are achieved, the exergy destruction starts to rise slightly. This incident happens because the LTC compressor and expansion work increases as the evaporation temperature are fixed, and this suppresses the advantage previously gained by decreased MTC exergy destruction.

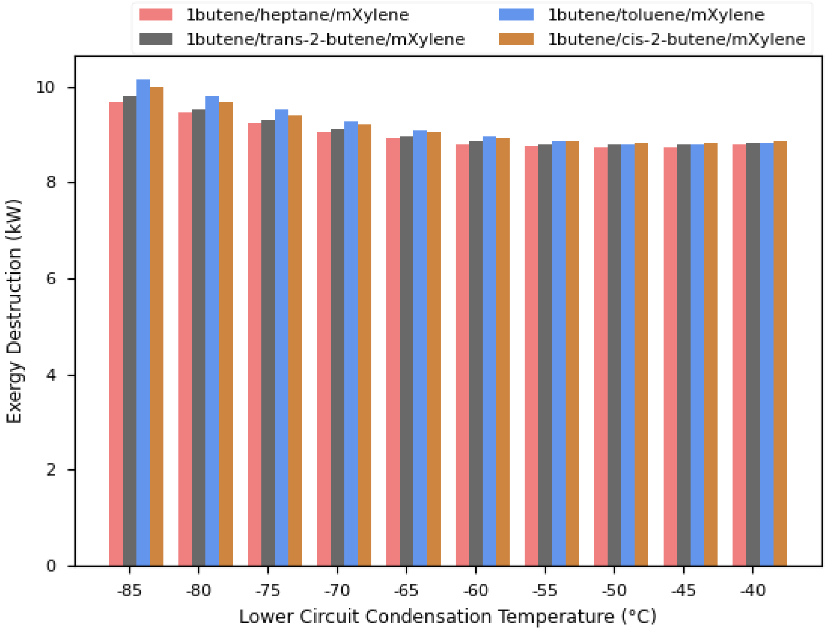


Figure 4.10: Variation of exergy destruction of the Pairs at different T_{lc}

Fig 4.10 shows the variation of total exergy destruction for different pairs using the bar chart. It is observed from the figure that, for any temperature T_{lc} the exergy destruction of 1-butene/n-Heptane/m-Xylene is lowest. For T_{lc} = -85°C to -55°C refrigerant pair 1-butene/toluene/m-Xylene gives the highest value of exergy destruction. After -55°C, exergy destruction becomes almost identical. It can be understood from Fig 4.10 that the best second law performance can be found for pair 1-butene/n-Heptane/m-Xylene and worst for pair 1-butene/toluene/m-Xylene.

Pie-charts of component-wise exergy destruction best and worst pair are shown in Fig 4.11a and 4.11b, respectively. The percentage of exergy destruction in different components are very similar for 1-butene/n-Heptane/m-Xylene and 1-butene/toluene/m-Xylene.

(a) 1-butene/n-Heptane/m-Xylene

(b) 1-butene/toluene/m-Xylene

(c) 1-butene/trans-2-butene/m-Xylene

(d) 1-butene/cis-2-butene/m-Xylene

Figure 4.11: Pi-charts of component-wise exergy destruction.

This result confirms that the exergy destruction percentage does not depend on the pair of refrigerants used in the system. Fig 4.11a shows that the HTC condenser encounters the most exergy destruction of close to 15% for both pairs. This phenomenon happens because a significant heat is rejected to the environment, which is lost permanently and cannot be used again. LTC cascade heat exchanger also shows significant exergy destruction for both pairs. 1.1906 kW (close to 14%) exergy is lost for 1-butene/n-Heptane/m-Xylene and 1.2023 kW (close to 14%) is lost for 1-butene/toluene/m-Xylene. For 1-butene/n-Heptane/m-Xylene exergy losses of LTC compressor, LTC Evaporator, MTC Compressor, HTC compressor is 0.8825 kW (close to 10%), 1.0967 kW (close to 13%), 0.8678 kW (close to 10%), 1.0329 kW (close to 12%). For 1-butene/toluene/m-Xylene exergy losses of LTC compressor, LTC Evaporator, MTC Compressor, HTC compressor is 0.8825 kW (close to 10%), 1.0967 kW (close to 12%), 0.8375 kW (close to 9%), 0.9564 kW (close to

10%). MTC cascade heat exchanger encounters low exergy loss close to 6% for 1-butene/n-Heptane/m-Xylene and higher loss close to 11% is encountered for 1-butene/toluene/m-Xylene. This phenomenon also confirms that pair of 1-butene/n-Heptane/m-Xylene is more suitable than 1-butene/toluene/m-Xylene.

Chapter 5

Conclusion

In this study, triple cascade refrigeration system was analyzed for low temperature application. Eco-friendly hydrocarbon (HC) refrigerants were used for the triple cascade refrigeration system (TCRS). In this study, TCRS was analyzed for evaporation temperature ranging from -100^{o}C to -150^{o}C. Different applications of TCRS for this kind of range can be suggested.

1. LNG liquefaction needs evaporation temperature ranging from around -150^{o}C [29].
2. 5. LPG liquefaction needs below -60^{o}C [58].
3. Steel alloy treatment needs around -100^{o}C [15].
4. Military and national defence -80^{o}C [58].
5. Different industries (e.g freeze-drying, cooling in pharmaceutical, chemical and petroleum industries) might need cooling space of around -100^{o}C [45].

For the analysis of TCRS, the simulation was conducted using four different pairs of hydrocarbon refrigerants. In LTC and HTC 1- butane and m-xylene were used. For intermediate system trans-2-butane, n-heptane, cis-2-butane, toluene, xenon was employed individually. To sum up, the four pairs are-

1. 1-butene/n-Heptane/m-Xylene

2. 1-butene/trans-2-butene/m-Xylene
3. 1-butene/cis-2-butene/m-Xylene
4. 1-butene/toluene/m-Xylene

The simulation results were based on the COP, LTC condenser temperature, fixed evaporation temperature (-100°C), heat transfer of MTC CHX, network input, second law efficiency, and exergy destruction. From the result of the simulation following conclusions were drawn:

- The refrigerant pair 1-butene/n-Heptane/m-Xylene had the best performance in the simulated TCRS for the T_{lc} value The highest COP of 0.683 at the lower circuit condensation temperature of -55°C was obtained for 1-butene/n-Heptane/m-Xylene. The pair 1-butene/toluene/m-Xylene gives the lowest COP of 0.6798 at -60°C of T_{lc}.

- The lowest heat transfer in the MTC cascade heat exchanger was obtained 18.25 kW by 1-butene/n-Heptane/m-Xylene, where the pair of 1-butene/toluene/m-Xylene gains the heat transfer of 18.9 kW at -55°C of T_{lc}.

- At -55°C is the net compressor work is lowest for 1-butene/n-Heptane/m-Xylene and highest for 1-butene/toluene/m-Xylene.

- For pair 1-butene/n-Heptane/m-Xylene, at -55°C of T_{lc}, the second law efficiency is maximum (55.217%). On the other hand, for pair 1-butene/toluene/m-Xylene, at the same T_{lc} temperature, the second law efficiency obtains the value of 54.909%

- At optimum temperature -55°C of T_{lc}, the total exergy destruction of 1-butene/n-Heptane/m-Xylene is 8.73 kW and 8.86 kW for 1-butene/toluene/m-Xylene.

- As the refrigerant pairs used in this research are eco-friendly and perform better than previous TCRS studies, for low-temperature applications, these pairs can be of great potential based on the performance analysis.

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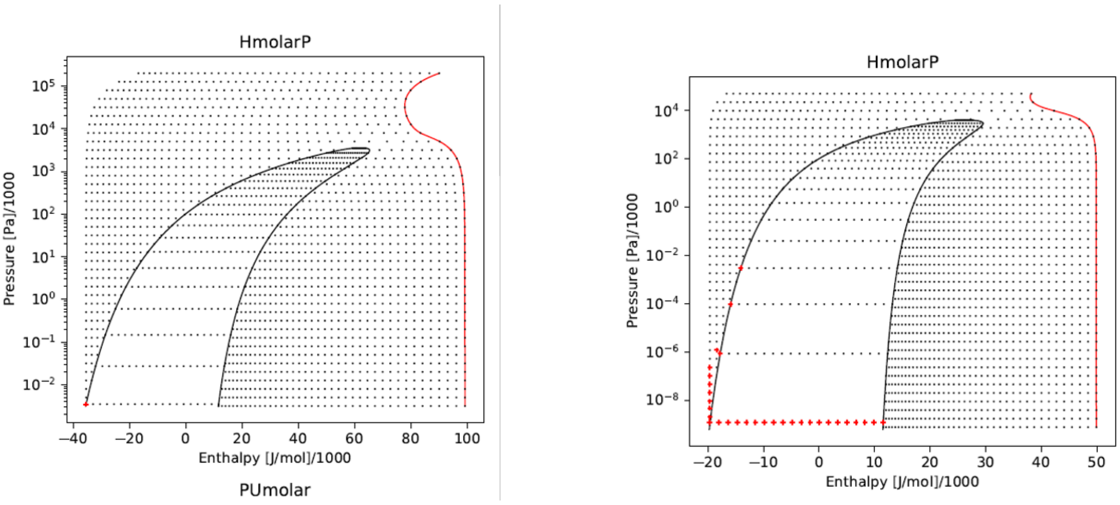
Chapter 6

Appendix



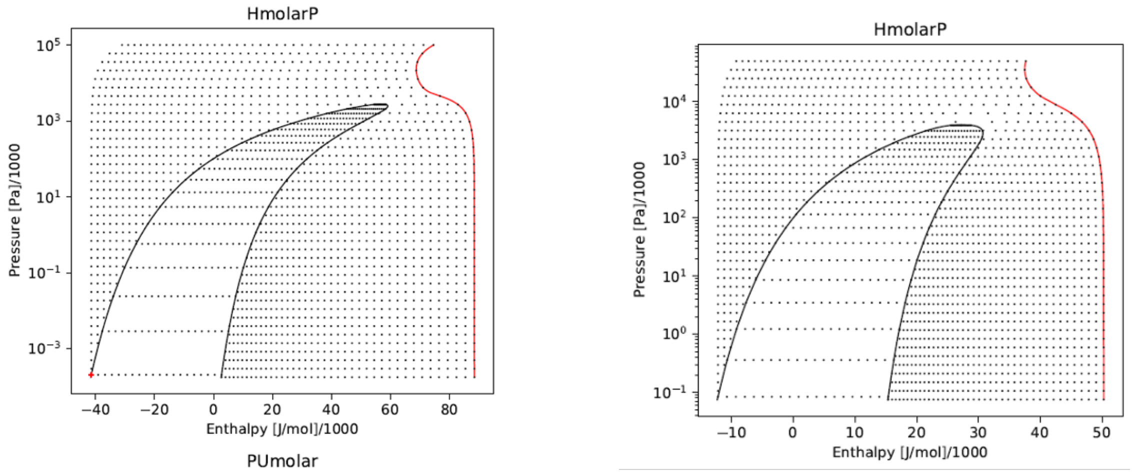
(a) P-h diagram of mXylene

(b) P-h diagram of 1butene

Figure 6.1: P-h diagrams of the refrigerants

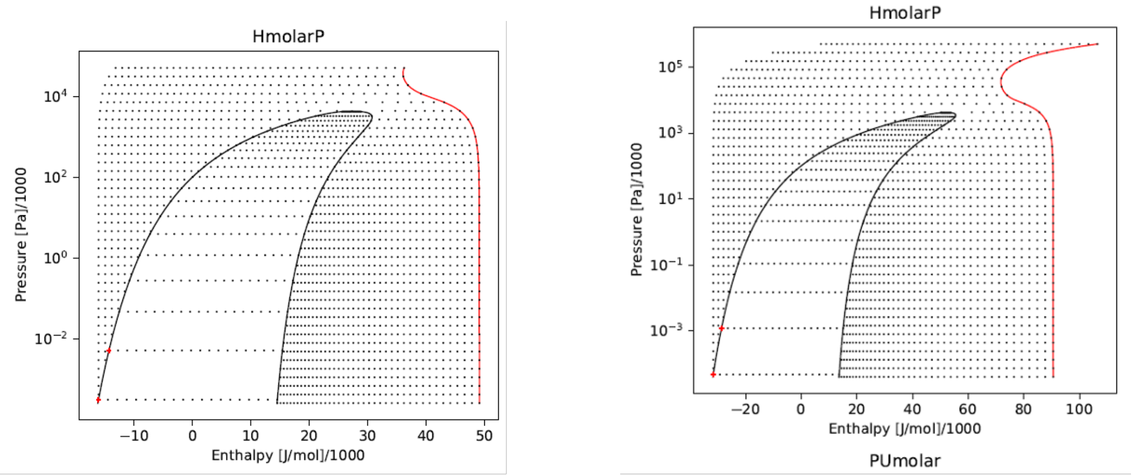
	m-Xylene	**1-butene**	**Heptane**
Molar Mass (kg/mol)	0.05610631	0.5191	0.100202
ASHRAE class	UNKNOWN	UNKNOWN	UNKNOWN
Formula	C_8H_10	C_4H_8	C_7H_16
Acentric Factor	0.326	0.1918	0.349

Table 6.1: Properties Table of Refrigerants.



(a) P-h diagram of Heptane

(b) P-h diagram of trans-2-butene



(c) P-h diagram of cis-2-butene

(d) P-h diagram of Toluene

Figure 6.2: P-h diagrams of the refrigerants (continue)

	Toluene	**cis-2-butene**	**trans-2-butene**
Molar Mass (kg/mol)	0.09213842	0.05610631	0.05610632
ASHRAE class	UNKNOWN	UNKNOWN	UNKNOWN
Formula	C_7H_8	C_4H_8	C_4H_8
Acentric Factor	0.2657	0.20236	0.21008

Table 6.2: Properties Table of Refrigerants (continue).

T_{lc} (°C)	-85	-80	-75	-70	-65	-60	-55	-50	-45	-40
Heptane	-30.5	-27.7	-25	-22.5	-18.6	-15.9	-13.7	-11	-8.1	-5.3
trans-2-butene	-30.9	-27.9	-25.8	-21.5	-19.8	-17.1	-14.4	-10.8	-7.9	-6.3
cis-2-butene	-30.6	-27.8	-25.3	-22.1	-19.4	-15.9	-13.6	-10.7	-7.6	-4.9
Toluene	-27.5	-22.7	-19.7	-16.8	-13.9	-12.6	-9.3	-6.5	-3.8	-0.4

Table 6.3: Optimum T_{mc} values.

Chapter 7

Nomenclature

Symbols	
T	Temperature
P	Pressure
h	Enthalpy
S	Entropy
m	Mass flow rate
Q	heat transfer
W	Work or Power input
X	Exergy
η	Efficiency
Δ	Difference
Acronyms	
TCRS	Triple Cascade Refrigeration System
LTC	Low Temperature Circuit
MTC	Mid Temperature Circuit
HTC	High Temperature Circuit
CRS	Cascade Refrigeration System
HX	Heat Exchanger
COP	Coefficient of Performance
VCR	Vapour Compression Refrigeration
ODP	Ozone Depletion Potential
GWP	Global Warming Potential

Table 7.1

Subscript	
1,2,3	State at different points
K	Condensation temperature of high temperature circuit
a	Environment
e	Evaporator
lc	Low temperature circuit condenser
mc	Mid temperature circuit condenser
L	Low Temperature Circuit
M	Mid Temperature Circuit
H	High Temperature Circuit
Evap	Evaporator
Cond	Condenser
Comp	Compressor
Cas	Cascade Heat Exchanger
s	Isentropic
dest	Destruction
carnot	Carnot Cycle

Table 7.2

Chapter 8

Codes

```
from CoolProp.CoolProp import PropsSI
import numpy as np
import matplotlib.pyplot as plt

    def sun2018(Tlc, Tmc,ref1,ref2,ref3):
Input Parameters

    Qe = 10 * 1000
Te = -100 + 273.16
Tk = 40 + 273.16
Tlc = Tlc + 273.17  Kelvin Conversion
Tmc = Tmc + 273.16  Kelvin Conversion

    delT=5 will be used later
super heating of htc mtc ltc are as 5,12,12.
Enthalpy and Entropy of each point is needed.

    LTC
```

Point-1

```
T1 = Te + 12
P1 = PropsSI(’P’, ’T’, Te, ’Q’, 1, ref1)
H1 = PropsSI(’H’, ’P’, P1, ’T’, T1, ref1)
S1 = PropsSI(’S’, ’P’, P1, ’T’, T1, ref1)
```

Point-3

```
T3 = Tlc
P3 = PropsSI(’P’, ’T’, T3, ’Q’, 0, ref1)
H3 = PropsSI(’H’, ’T’, T3, ’Q’, 0, ref1)
S3 = PropsSI(’S’, ’T’, T3, ’Q’, 0, ref1)
```

Point-4

```
H4 = H3
T4 = Te
P4 = P1
S4 = PropsSI(’S’, ’H’, H4, ’P’, P1, ref1)
```

Point-2

```
P2 = P3
S2s = S1
H2s = PropsSI(’H’, ’P’, P2, ’S’, S2s, ref1)
effcompL = 0.8
H2 = H1 + ((H2s - H1) / effcompL)
S2 = PropsSI(’S’, ’P’, P2, ’H’, H2, ref1)
T2 = PropsSI(’T’, ’P’, P2, ’H’, H2, ref1)
```

MTC

Point-5

Tme = Tlc - 5

T5 = Tlc - 5 + 12

P5 = PropsSI(’P’, ’T’, Tme, ’Q’, 1, ref2)

H5 = PropsSI(’H’, ’T’, T5, ’P’, P5, ref2)

S5 = PropsSI(’S’, ’T’, T5, ’P’, P5, ref2)

Point-7

T7 = Tmc

P7 = PropsSI(’P’, ’T’, T7, ’Q’, 0, ref2)

H7 = PropsSI(’H’, ’T’, T7, ’Q’, 0, ref2)

S7 = PropsSI(’S’, ’T’, T7, ’Q’, 0, ref2)

Point-8

H8 = H7

T8 = T3 - 5

P8 = P5

S8 = PropsSI(’S’, ’H’, H8, ’P’, P8, ref2)

Point-6

P6 = P7

S6s = S5

H6s = PropsSI(’H’, ’P’, P6, ’S’, S6s, ref2)

effcompM = 0.8

H6 = H5 + ((H6s - H5) / effcompM)

S6 = PropsSI(’S’, ’P’, P6, ’H’, H6, ref2)

```
T6 = PropsSI(’T’, ’P’, P6, ’H’, H6, ref2)

    HTC

    Point-9
The = Tmc - 5
T9 = The + 5
P9 = PropsSI(’P’, ’T’, The, ’Q’, 1, ref3)
H9 = PropsSI(’H’, ’P’, P9, ’T’, T9, ref3)
S9 = PropsSI(’S’, ’P’, P9, ’T’, T9, ref3)

    Point-11
T11 = Tk
P11 = PropsSI(’P’, ’T’, T11, ’Q’, 0, ref3)
H11 = PropsSI(’H’, ’T’, T11, ’Q’, 0, ref3)
S11 = PropsSI(’S’, ’T’, T11, ’Q’, 0, ref3)

    Point-12
H12 = H11
T12 = The
P12 = P9
S12 = PropsSI(’S’, ’H’, H12, ’P’, P12, ref3)

    Point-10
P10 = P11
S10s = S9
H10s = PropsSI(’H’, ’P’, P10, ’S’, S10s, ref3)
effcompH = 0.8
```

H10 = H9 + ((H10s - H9) / effcompH)

S10 = PropsSI(’S’, ’P’, P10, ’H’, H10, ref3)

T10 = PropsSI(’T’, ’P’, P10, ’H’, H10, ref3)

Flow Rates:

LTC

mL = Qe / (H1 - H4)

MTC

mM = (mL * (H2 - H3)) / (H5 - H8)

HTC

mH = (mM * (H6 - H7)) / (H9 - H12)

Compressor Work:

LTC

wLTC = mL * (H2 - H1)

MTC

wMTC = mM * (H6 - H5)

HTC

wHTC = mH * (H10 - H9)

Total Compressor Work

wCOMP = wLTC + wMTC + wHTC

```
    COP

    COP = Qe / wCOMP

    COPL = Qe / wLTC

    COPM = (H5 - H8) / (H6 - H5)

    COPH = (mM * (H6 - H7)) / (mH * (H10 - H9))

    COPcarnot = Te / (Tk - Te)

    Effsecond = COP / COPcarnot
return COP, COPL, COPM, COPH

    Tlc=np.array([-85, -80, -75, -70, -65, -60, -55, -50, -45, -40])
Tmc=np.array([-30.5, -27.7, -25, -22.5, -18.6, -15.9, -13.7, -11, -8.1, -5.3])
ref1=str(’1Butene’) LTC
ref2=str(’HEPTANE’) MTC
ref3=str(’mXylene’) HTC

    m=10
COP=[]
COPL=[]
COPM=[]
COPH=[]

    for i in range (m):
```

```
a,b,c,d=sun2018(Tlc[i],Tmc[i],ref1,ref2,ref3)
a=round(a,4)
b=round(b,4)
c=round(c,4)
d=round(d,4)
COP.append(a)
COPL.append(b)
COPM.append(c)
COPH.append(d)

   plt.figure(1)
plt.plot(Tlc, COP,label = ’COP’, marker=’o’,markersize=3 )

   plt.figure(2)
plt.plot(Tlc, COPL,label = ’COPL’, marker=’o’,markersize=3 )
plt.plot(Tlc, COPM,label = ’COPM’, marker=’o’,markersize=3 )
plt.plot(Tlc, COPH,label = ’COPH’, marker=’o’,markersize=3 )

   plt.title(’COP of organic refrigerants’)
plt.xlabel(’Lower condensation temperature,°C’)
plt.ylabel(’COP’)
plt.legend()
plt.show()

   print(’COP= ’, COP)
print(”)
print(’COPL= ’,COPL)
print(”)
```

```
print(’COPM= ’,COPM)
print(”)
print(’COPH= ’,COPH)
```