THERMODYNAMIC & ENVIRONMENTAL ANALYSIS OF NOVEL MULTI TARGET TEMPERATURE CASCADE REFRIGERATION SYSTEM

A DISSERTATION

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OF

MASTER OF TECHNOLOGY
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Submitted by:

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I, VAIBHAV SHARMA, Roll No. 2K20/THE/22, student of M. Tech Thermal Engineering, hereby declare that the project Dissertation titled *"*THERMODYNAMIC AND ENVIRONMENTAL ANALYSIS OF NOVEL MULTI TARGET TEMPERATURE CASCADE REFRIGERATION SYSTEM*"* which is submitted by me to Department of Mechanical engineering, Delhi Technological University, Delhi in partial fulfillment of the requirement for the award of the degree of Masters of Technology, is original and is not copied from any source proper citation. This work has not previously formed the basis for the award of any Degree, Diploma Associateship, Fellowship or other similar title or recognition.

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CERTIFICATE

I hereby certify that the Project Dissertation titled “THERMODYNAMIC AND ENVIRONMENTAL ANALYSIS OF NOVEL MULTI TARGET TEMPERATURE CASCADE REFRIGERATION SYSTEM”, which is submitted by Vaibhav Sharma 2K20/THE/22, Mechanical Department, Delhi Technological University, Delhi, in partial fulfilment of the requirement for the award of degree of Masters of Technology, is a record of the project work carried out by the student under my supervision. To the best of my knowledge this work has not been submitted in part of full for any Degree or Diploma to this University or elsewhere.

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ABSTRACT

A novel multi target temperature cascade refrigeration system has been proposed in this study. This system is a type of multi evaporator refrigeration system. A thermodynamic model of the system has been developed using engineering equation solver for theoretical investigation. The results of system's performance have been compared with another configuration of multi target temperature cascade refrigeration system proposed by Zhu et al. [10] which was proved to be better than two configurations of combined cascade refrigeration systems. This novel system has three circuits, namely high, medium and low temperature circuits, and the high temperature circuit is connected to medium and low separately by cascade heat exchangers. For performance enhancement, this system utilizes two throttles in high temperature circuit, to maintain optimum temperature in cascade heat exchangers, a much efficient scroll compressor, and a subcooler. First law and second law analysis have been performed for parametric analysis of the system. This novel system functions really well as on average, the system's COP is almost 10% higher and can even rise to nearly 21% in some operating conditions. This system utilizes low global warming potential (GWP) refrigerants like ammonia, carbon dioxide, and R1234ze(z), therefore, for environmental assessment of the system, the total equivalent warming impact (TEWI) of the system has been calculated.

Keywords: Multi Target Temperature cascade system (MTTC), Low temperature circuit (LTC), Medium Temperature circuit (MTC), High Temperature Circuit (HTC), Throttle, subcooler, scroll compressor, exergy, Total Equivalent Warming Impact (TEWI).

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CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF ABBREVIATIONS

MTTC:	Multi target temperature cascade system
CRS:	Cascade refrigeration system
LTC:	Low temperature circuit
MTC:	Medium temperature circuit
HTC:	High temperature circuit
GWP:	Global warming potential
ODP:	Ozone depletion potential
T_{CHX}:	Temperature in cascade heat exchanger
η_{exergy}:	Exergy efficiency
$\eta_{\mathrm{isentropic}_{NH3}}$:	Isentropic efficiency of compressor working with ammonia
$\eta_{\mathrm{isentropic}_{\mathrm{CO2}}}$:	Isentropic efficiency of compressor working with carbon dioxide
sbc:	Refrigerant fraction through subcooler
TEWI:	Total equivalent warming impact
L_{annual}:	Refrigerant leakage per year (Kg)
n:	Operational life of system (Year)
$\alpha_{recovery}$:	Refrigerant recycling/recovery factor
E_{annual}:	Energy consumption per year (kWh)
β =	Indirect emission factor or CO_2 emission factor

CHAPTER 1

INTRODUCTION

Refrigeration and air conditioning plays a crucial part in almost all the sectors, from domestic applications to industrial and commercial operations. The main aim of any refrigeration or air conditioning system is to remove the heat from a space and most of the such systems are based on vapor compression refrigeration system. Traditional systems based on single stage vapor compression cycles have been used for many years to provide refrigeration in commercial, industrial and other large sectors. However, as the requirement of the cooling speed and extremely low temperature needs are continually uprising, customary single staged cycle-based refrigeration system have become out of space on account of high-pressure ratio across condenser and evaporator, which drastically increases compressor work, and it ultimately causes the system's COP to decline, and therefore, gives poor results in terms of economy when applied to the circumstances with enormous temperature ranges. All things considered; multi-stage cycles arise as the occasions require.

There are a few alternative options to achieve this demand of lower refrigeration temperature in terms of configurations for multi stage cycles. Among various multi-stage cycles, compression cycle with two stage and cascade system are accepted to be appropriate option to meet the requirements. A cascade refrigeration cycle is a multi-stage thermodynamic cycle. In a cascade refrigeration cycle, at least two vapor-compression cycles, sometimes referred to as low temperature circuit (LTC) and high temperature circuit (HTC), which are connected via a heat exchanger, which supports different refrigerants are utilized. The evaporator-condenser temperatures of sub-circuit are successively lower with some cross-over to cover the absolute temperature drop needed for the refrigeration. The main heating load is taken up by the low temperature circuit's evaporator and it transfers it to cascade heat exchanger, connecting the two circuits, from where refrigerant in evaporator of high temperature circuits picks up the heat and heat is eventually thrown out though the condenser. Since cascade cycle-based system has multiple different sub-circuits which can be accused of various types of refrigerants, the general performance of such system can be improved by accomplishing the ideal conditions in each sub-circuit, which enables cascade cycles to accomplish lower refrigeration temperatures [1]. The determination of refrigerant plays a significant role in such systems. Normally Low and high boiling point refrigerants are suggested for low and high temperature loop respectively in these systems. Aside from thermodynamic properties, physical and chemical as well as most important nowadays, its effect on environment is also considered while planning of low and high temperature circuits in such system.

As mentioned by Kumar [2] in his work, when compared to a single refrigeration cycle-based system, a cascade system can be enabled to produce a significantly lower evaporation temperature, and still maintains lower compression proportion as there are at least two circuits to that maintain the overall cycle, and thus in most cases, supports higher volumetric efficiency. Along with the increasing demand of low temperature for various applications, recently there has been incline in the requirement of the systems that can produce multi evaporation temperature. Such load conditions can exist in various commercial, industrial application areas like supermarkets, where fishes (and other meat-based products) are stored at around -35℃ [3], whereas other dairy products require temperature of around -10℃, to keep them chilled. Various medicational institutes need to store variety of samples at different low temperature, and also for vaccine preservation extremely low temperature is needed. Current commercial refrigeration requires refrigeration system for food freezing and to maintain freshness of food items in retail stores and supermarkets, and roughly half of the supermarket energy utilization is related with the refrigeration demand, in this way, commercial refrigeration has become one of the most significant areas as far as energy utilization and ozone harming substance emissions to environment [4]. For applications based on requirement of most supermarket, they often need more than one evaporator temperature, for example chilling temperature, mostly to keep dairy-based products fresh, along with some other products, and freezing temperature, to keep up with the best quality of food that needs to be freeze.

Literature overview of the cascade cycle refrigeration system (CRS) was presented by Pan et al. [1]. It has been featured that such system can accomplish even -170˚C evaporating temperature and thus allows conventional cascade system to be used in wide application areas. In this paper, several exploration choices like different configuration of CRS, reviews of different refrigerants, and enhancement or developments on the system were talked about. Besides, the impact of various factors on CRS performance, the financial analysis, along with the applications were also characterized, conclusion by ends and also made ideas in regards to for future studies. Eini et al. [5] carried out an optimization of cascade refrigeration system on objective function employing multiple objectives, which took under consideration exergy, economy, and the environmental perspectives other than evaluation of the natural safety level. The outcomes showed no critical contrasts between CO_2/NH_2 and CO_2/C_3H_8 regarding financial and exergy efficiency objectives. Realizing the importance of low global warming (GWP) refrigerant, Soni et al. [6] presented a theoretical study on use of low GWP refrigerants in cascade refrigeration system. It included Ammonia/Carbon Dioxide, R1234yf with CO2, and R1234ze with CO_2. Along with lowest GWP combination, ammonia in HTC and carbon dioxide in LTC proved to be the best choice as it showed highest COP, with least compressor work. The NH_3/CO_2 combination in cascade refrigeration system was proved to be most efficient in achieving temperature of -30℃ to -50℃ by Messineo [7].

In the field of multi evaporator refrigeration system, three new multi evaporator cascade refrigeration systems (CRS) were proposed by Saini et al [8], and results were evaluated for seafood processing related application with high ambient for countries like India. Each CRS had four evaporators, and main difference between these systems were the number of employed compressors in it, evaporator load distribution and use of refrigerants with pumped circulation. This work suggested that using natural refrigerants in CRS not only is environmentally friendly, but also has great implementation in seafood processing industry. Few researchers, did multi-dimensional study of multi evaporator refrigeration system, Arpagaus et al. [9] worked on energy, exergy, and economic analysis of such systems.

Then, Zhu et al. [10] featured in his work that Cascade cycle refrigeration systems have been perceived as the forthcoming innovation to meet the necessity of ultra-low refrigeration and the associated energy requirement reduction. To satisfy the interest of the multi refrigeration temperatures from a single system and defeat the dominant issue in the commercial and industrial area of applications, a new multi-target-temperature cascade (MTTC) refrigeration system has been presented, which utilizes natural refrigerants, ammonia in the high temperature circuit od the system and carbon dioxide and R1270 in the low temperature and medium temperature circuits respectively. The ideal thermodynamic performance of the planned multi target temperature cascade refrigeration cycle was evaluated and also provided economic analysis of the system, and then compared with existing systems cable of producing multiple evaporation temperature by a single cycle. Results showed that, MTTC system’s COP is around 25% and 10% better than COP of CSC1 system and CSC2 system; the exergy efficiency of MTTC system was roughly 19% and 5% superior than that of other being compared. Therefore, in view of the such praiseworthy thermodynamic, and exergy-economy performance of this system, MTTC can be utilized to have a decent practical application prospect in various domains.

As referenced before in regards to the power utilization by refrigeration and cooling systems, according to Lombard et al. [11] refrigeration systems consume enormous percentage of energy in keeping up with needs of appropriate temperature for occupant's comforts and to maintain desired conditions in the area cooling applications, which made up half of building energy utilization. This features the need of making such systems more effective and accordingly lessening the power utilization, and for this, engineers now thrive to develop more efficient and green technology-based system. In the same direction, with the aim to make cascade system more efficient, Chi et al [12] introduced an auxiliary refrigeration loop in the HTC of cascade system, this subcooled the refrigerant in LTC. This small addition in system configuration reflected increase in COP, and exergy as well. Subcooling can significantly improve the system's performance, supporting this statement is the work by She et al. [13], as in ordinary vapor compression-based system, the power recovered from expansion was used to run the compressor of another supplementary cycle, which served the sole purpose of subcooling the refrigerant in primary vapor compression cycle. This supplementary cycle did, theoretically increased the COP of the system.

Another crucial part of this work is utilization of Scroll compressor and therefore it was explored in the sense of utilization in cascade based multi target temperature system. To study the efficiency of a scroll compressor, Tojo et al. [14] compared a scroll compressor with a reciprocating compressor working under similar conditions, and the results made it clear that not just the isentropic efficiency even the volumetric efficiency of scroll compressor is comparatively higher and the pressure ratio toll on volumetric and isentropic efficiency is significantly less for scroll compressor whereas reciprocating compressor is greatly affected by pressure ratio. On average, it can be accepted to assume that scroll compressor isentropic efficiency is nearly 10% greater than reciprocating compressor. Scroll compressors have the ability to handle multi stage pressure ratio as supported by various studies and experimental data. One study was conducted by Song et al. [15] and they concluded that multi-port vapor injected scroll compressor has higher efficiency as compared to a baseline model.

Another important study related to scroll compressor, where Tanveer and Bradshaw [16] conducted a performance assessment with scroll compressor having refrigerating capacity of 1 to 100 tons, with low GWP refrigerants. An approved model working with standard AHRI conditions was utilized for the assessment of the refrigerants. Seven refrigerants were taken into the consideration for this work, including low-pressure refrigerants, R1234ze(E) and R1234yf, and there were four medium-pressure refrigerants, namely R410A, R32, R454B, and R452B, along with one high-pressure refrigerant, carbon dioxide. From this study, it was found volumetric and isentropic efficiency showed similar trend, where refrigerants were able to reach more than 90% volumetric efficiency mark, and little over 70% isentropic efficiency over the load range.

As mentioned earlier, refrigerant plays a crucial role in multi evaporator cascade system, as to get the best performance, each circuit should work optimally and for that refrigerant must be selected carefully considering their thermo-physical properties, and now a days, refrigerants must have low GWP, ODP as well. To capture impact of refrigeration systems on global warming, TEWI: total equivalent warming impact is used. TEWI is a measure of global warming impact of the system in terms of total associated greenhouse gases emissions during the system's operation and when the operating fluids are disposed at the end of life [17]. It consists of total warming impact due to direct and indirect emissions, direct emissions are related to refrigerant released during operational life of system, as well as the refrigerant lost while the disposal at end of life, and on other hand, indirect emissions are those which are produced due to CO_2 emissions from fossil fuels that are used to produce electricity that powers the refrigeration system.

After considering all the aspects, developments, configurations, and requirements from a multi evaporator refrigeration system, it appears that MTTC system by Zhu et al. [10] does have the potential to meet refrigeration requirements, particularly in commercial and industrial sector, especially supermarkets and hospitals. Thermodynamic analysis has also given results in favour of multi target temperature cascade system. But still, limited research has been undertaken in the field for development of such system that can produce multi target temperature, and only one new design with substantial progress has been proposed lately. Moreover, these systems have not been worked with other promising refrigerants which might increase the efficiency of such systems. Along with this, environmental aspects of operation of such systems still need to be explored.

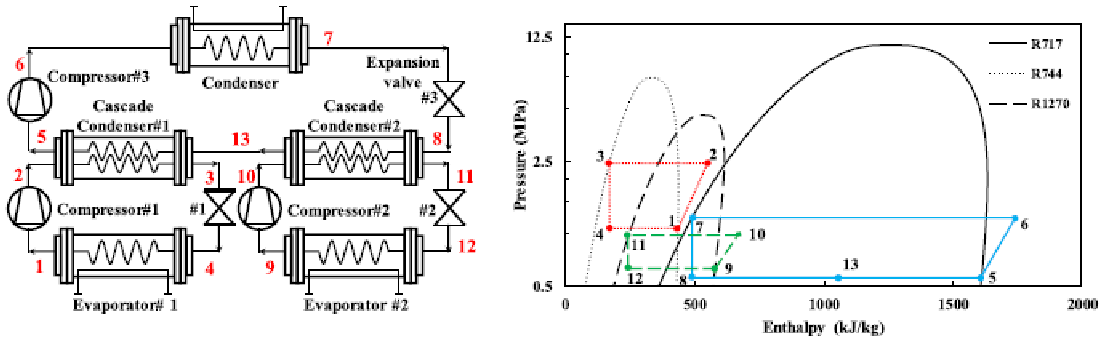

Figure 1.1: MTTC by ZHU et al. [10]

This work has utilized the MTTC system [10] for comparison and, developments and upgrades used in the novel system design or configuration has been made to increase the performance, and to minimize not just the power requirements, but also the environmental impact. For reducing the energy consumption, the novel system utilizes more efficient scroll compressor in HTC, which can also handle multiple pressure ratio, then as from the positive results by Chi et al. [12], and She et al. [13] regarding subcooling, a subcooler has also been used in HTC to subcool the refrigerant that picks up the load from LTC/HTC cascade heat exchanger. Furthermore, instead of using a single throttle in HTC, the new system uses two separate throttles in order to achieve optimum temperature for heat exchanger in each LT/HT, MT/LT cascade circuit, along with these, for environmental-emission, and performance related benefits, a new refrigerant, R 1234ze(Z) has been used in medium temperature circuit, which has extremely low GWP, of less than 6 [18]

CHAPTER 2

SYSTEM DESCRIPTION AND MODELING

2.1. SYSTEM'S DESCRIPTION

From the literature review, it can be said that there are various configurations by which a single refrigeration system can produce multi temperature. A new multi target temperature cascade system configuration has been developed and presented in this work and its performance comparison over MTTC developed by Zhu et al. [10] has been presented. This novel system has three main circuits, high temperature circuit (HTC) which has ammonia as refrigerant, medium temperature circuit (MTC) which employs R1234ze(Z) refrigerant for producing temperature in the range of -30℃ to -10℃, and a low temperature circuit using CO_2 as the refrigerant which is capable of producing extremely low temperature up to -50℃. To maintain Optimum temperature in Cascade heat exchanger and to support higher efficiency in the system, the system employs two throttles in HTC, so that refrigerant in higher temperature circuit can be throttled to two different temperatures as per the requirements for LT and MT circuit evaporator temperature. As the refrigerant in higher temperature circuit is throttled to two different pressures, so to compress the refrigerants back to condenser pressure, a scroll compressor is being used which not only has the ability to handle such pressure streams, but also is more efficient than reciprocating counterpart in the process. From the previous work available, scroll compressor is nearly 10% more efficient in terms of power consumption [14]. Although for the development of this new system, and for clear comparison with previous MTTC system, the efficiency of scroll compressor has been treated same as reciprocating compressor as used in previous MTTC. A part of refrigerant in HT circuit that has been throttled for MT circuit cascade heat exchanger has been used in SUBCOOLER to subcool the refrigerant for LT circuit's cascade heat exchanger, this reduces the quality of refrigerant, and hence increases the refrigerating capacity of ammonia refrigerant for LT's cascade heat exchanger or in other way around it reduces the compressor work by decreasing the refrigerant's mass flow rate in the circuit. As the subcooler is there in HTC, therefore the fraction of refrigerant passing through subcooler has been denoted by "sbc". The refrigerant used in MTC, is-1,3,3,3-tetrafluoropropene (R-1234ze(Z)) which is a fluorinated propane isomer as indicated in various research and development of the refrigerant. The biggest advantage and as also the most important requirement nowadays for any refrigerant, this refrigerant has zero ozone depletion potential and global warming potential figure is about of less than 6 [18].

The new proposed system has the biggest advantage of providing optimum temperature in the cascade heat exchangers connecting the LTC with HTC and MTC with HTC. In MTTC system developed by Zhu et al. [10], there was only a single throttle due to which the cascade heat exchanger was maintained at a single temperature. While the evaporator temperature in LTC and MTC varied, cascade was maintained at a single fixed temperature in MTTC system, this lowered the COP. The optimum temperature is decided by the relation this is provided by Kin et al. [25].

$$T_{CHX} = \sqrt{Tc * T_{eva}} \tag{2.1}$$

Here T_{CHX} denotes temperature of cascade heat exchanger in kelvin, Tc refers to high stage condenser temperature in kelvin, and T_{eva} is low stage evaporator temperature in kelvin.

The next biggest edge that the new system has is the subcooler installed in HTC. A part of refrigerant in HT circuit that is throttled for MT circuit's cascade heat exchanger and is used in SUBCOOLER to subcool the refrigerant for LT circuit's cascade heat exchanger, this reduces the quality of refrigerant, and hence increases the refrigerating capacity of HTC refrigerant for LT's cascade heat exchanger. The refrigerant that subcools the refrigerant for cascade heat exchanger of LTC is throttled to the same temperature and pressure as the refrigerant that goes into cascade heat exchanger of MTC, this allows the two refrigerant streams to be mixed together afterwards before entering into the scroll compressor. The refrigerant that gets subcooled is able to pick more heat from cascade heat exchanger that connects HTC and LTC as subcooling it allows throttling it to extremely low quality. The amount of subcooling is decided by mathematically equating the two-refrigerant stream in subcooler in such manner that after throttling subcooled refrigerant stream has quality nearly zero.

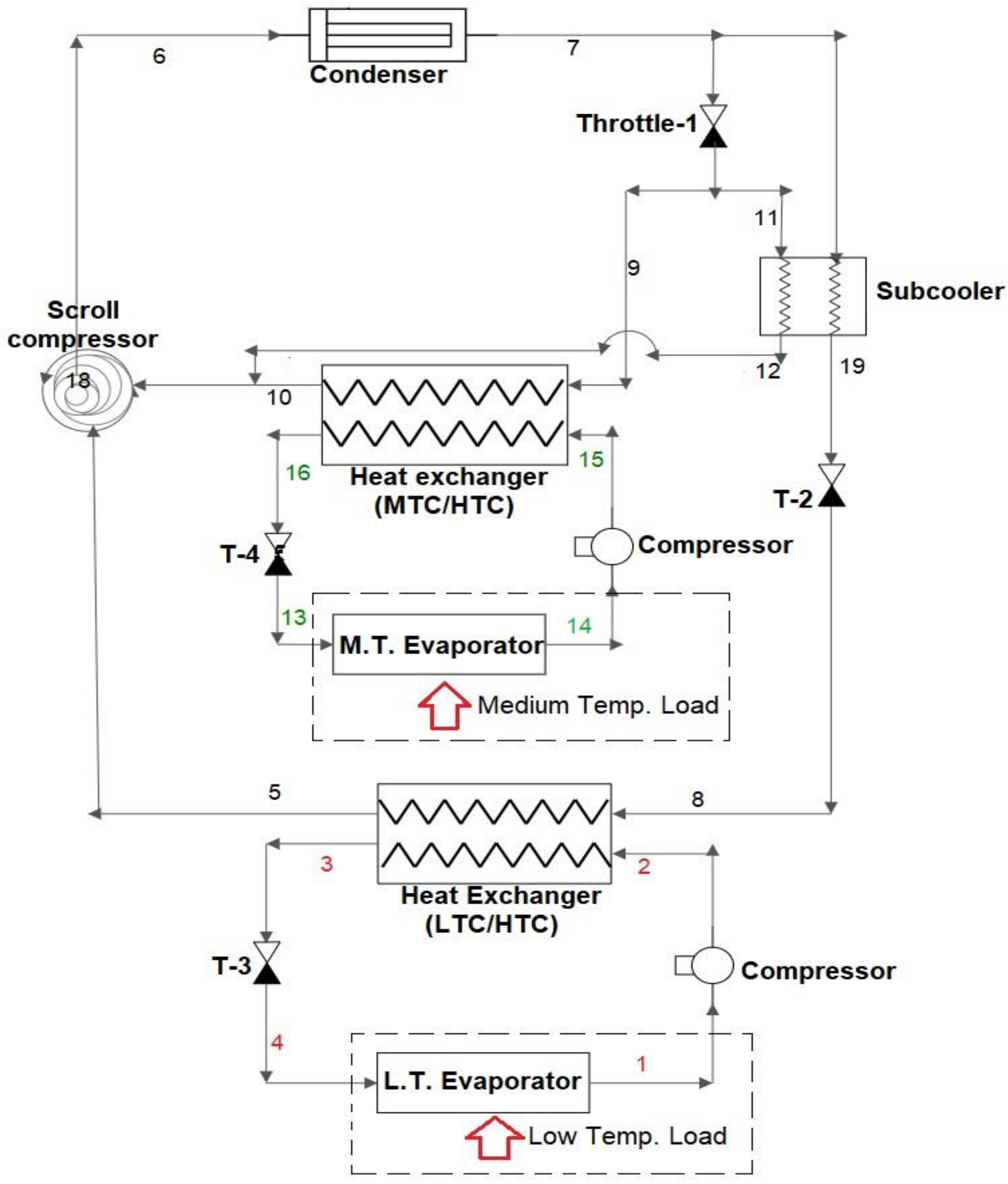

Figure 2.1: Novel multi target temperature cascade system

The LT evaporator is maintained in range -50℃ to -40℃, whereas MT evaporator -30℃ to -10℃. Whereas each of the heat exchangers connecting cascade circuits are maintained at optimum temperature and this purpose is served by the throttle 1 and throttle 2, which throttles the ammonia refrigerant in HTC to optimum temperature corresponding to different temperature in MTC and LTC evaporators. With the help of some previous work, the isentropic efficiency of compressors working with Ammonia and Carbon dioxide can be written as [19-21]

$$\eta_{\text{isentropic}_{NH3}} = -0.00097PR^2_{NH3} - 0.01026PR_{NH3} + 0.83955$$

$$\eta_{\text{isentropic}_{CO2}} = 0.00476PR^2_{CO2} - 0.09238PR_{CO2} + 0.89810$$

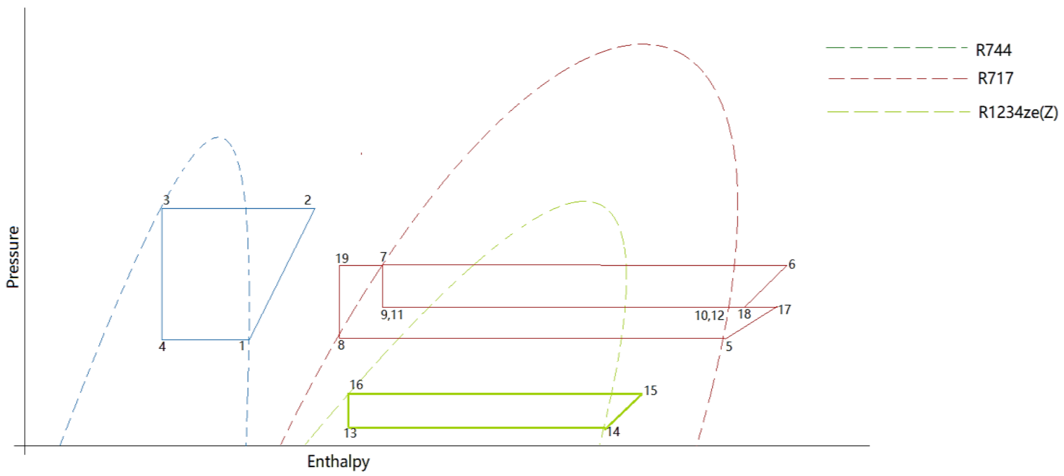

Figure 2.2: p - h diagram

Here PR stands for pressure ratio. The combined motor and mechanical efficiency of compressor working with ammonia and CO_2 is taken as 0.93 [22]. Therefore, total efficiency of compressor is $\eta_{\text{isentropic}}$*0.93. On the other hand, due to limited research with R1234ze(Z), the data and values associated with isentropic efficiency of compressor is not available and the isentropic efficiency for compressor has been assumed as 0.75. A term called load proportion ratio has been defined by Zhu et al. [10], denoted by “x”, which basically is ratio of LTC refrigerating load to total refrigerating load. In this project, when the load proportion ratio changes, the load across LTC evaporator remains same, and the load in MTC evaporator varies.

2.2. THERMODYNAMIC MODELING

Energy and exergy analysis define the capabilities of a system in term of theoretical performance and for that thermodynamic evaluation of the novel MTTC is carried out using first law of thermodynamic, along with second law, and associated equations, where mass, energy balance, exergy balance equations [23] are put together for the analysis and the input data has been provided in system description. In this analysis, the few assumptions have been made while performing the thermodynamic assessment.

- Kinetic and potential energies changes in the system are zero.
- Steady state condition of operation has been assumed in system.
- Pressure drop across the pipeline in the new system has been considered zero.
- Heat transfer or losses from any heat exchanging equipment have been neglected.
- There is no change in enthalpy while throttling, i.e., isenthalpic process.
- The isentropic efficiency of compressor working R1234ze(Z) is assumed to be 0.75.

The mass balance equation for the new system based on cascade refrigeration can be written as follows for each Component of the system; for HTC, MTC and for LTC:

$$\Sigma \dot{m}_{in} = \Sigma \dot{m}_{out} \tag{2.2}$$

In the similar sense, for every component and for each sub circuit, the general energy balance equation is evaluated as follows:

$$\Sigma \dot{m}_{in} h_{in} + \Sigma Q_{in} + \Sigma \dot{W}_{in} = \Sigma \dot{m}_{out} h_{out} + \Sigma Q_{out} + \Sigma \dot{W}_{out} \tag{2.3}$$

Here, “in" and "out" refers to the values or parameters at the entry and exit points of each component under consideration, while W represents work, h stands for enthalpy, and Q represents heat transfer rate, which are general notation in most cases. For the exergy analysis, following thermodynamic equations have been used:

$$\dot{E}x_{\mathrm{mass,in}} + \dot{E}x_{heat,in} + \dot{E}x_{work,in} = \dot{E}x_{mass,out} + \dot{E}x_{heat,out} + \dot{E}x_{work,out} + \dot{E}x_{Destroyed} \tag{2.4}$$

Here, $\dot{E}x_{\mathrm{mass}}, \dot{E}x_{heat}, \dot{E}x_{work}, and\ \dot{E}x_{Destroyed}$ stands for Exergy due to mass transfer, heat transfer, work transfer and exergy destruction or known as irreversibility, respectively.

A similar multi temperature refrigeration which was developed by Lontsi et al. [24] and as the system had two different evaporators at two different temperatures, the exergy efficiency was defined as follows:

$$\eta_{exergy} = \frac{\left|(1 - \frac{T_O}{T_{ev1}})\right| * Q_{E_{LTC}} + \left|\left(1 - \frac{T_O}{T_{ev2}}\right)\right| * Q_{E_{MTC}}}{W_{Net_{Compressor}}} \tag{2.5}$$

Here, T_O means the reference or atmospheric temperature that is used for exergy analysis, T_{ev1}, T_{ev2} stands for LTC and MTC evaporator temperature respectively, $Q_{E_{LTC}}, Q_{E_{MTC}}$ means the evaporator load in LTC and MTC respectively, and net compressor work, i.e., summation of LTC, MTC, and HTC compressor has been indicated by $W_{Net_{Compressor}}$.

$$\begin{aligned} Ex_{Dest_{NET}} = {} & Ex_{Dest_{EVA1}} + Ex_{Dest_{EVA2}} + Ex_{Dest_{COMP1}} + Ex_{Dest_{COMP2}} + Ex_{Dest_{THR1}} \\ & + Ex_{Dest_{THR2}} + Ex_{Dest_{Cond}} + Ex_{Dest_{HX1}} + Ex_{Dest_{HX2}} + Ex_{Dest_{THR3}} \\ & + Ex_{Dest_{THR4}} + Ex_{Dest_{SC}} + Ex_{Dest_{SB}} \end{aligned} \tag{2.6}$$

Table 2.1: Thermodynamic equations used for components

Component	Thermodynamic equation used
LTC Evaporator	$Q_{E_{LTC}} = \dot{m}_{ltc}(h_1 - h_4)$
MTC Evaporator	$Q_{E_{MTC}} = \dot{m}_{mtc}(h_{14} - h_{13})$
LTC Compressor	$h_2 = \frac{h_{\acute{2}} - h_1}{\eta_{\text{isentropic}_{CO2}}}; s_1 = s_2; W_{Comp_{LTC}} = \frac{\dot{m}_{ltc}(\square_2 - \square_1)}{\eta_{mec\square} * \eta_{elec}}$
MTC Compressor	$s_{14} = s_{1\acute{5}}; W_{comp_{MTC}} = \frac{\dot{m}_{mtc}(h_{1\acute{5}} - h_{14})}{0.75}$
HTC Scroll Compressor	**For first stage (to MTC/HTC cascade heat exchanger pressure)** $h_{17} = \frac{h_{1\acute{7}} - h_5}{\eta_{\text{isentropic}_{NH3}}}$; $s_5 = s_{1\acute{7}}$; $W_{Stage1} = \frac{\dot{m}_{htc1}(h_{17} - h_{15})}{\eta_{mec\square} * \eta_{elec}}$ **For final stage (to condenser pressure)** $sbc * \dot{m}_{mtc2} * h_{10} + (1 - sbc) * \dot{m}_{mtc2} * h_{12} + \dot{m}_{mtc1} * h_{17} = \dot{m}_{mtc} * h_{18}$ $s_{18} = s_{\acute{6}}$; $h_6 = \frac{(h_{\acute{6}} - h_{18})}{\eta_{\text{isentropic}_{NH3}}}$ $W_{stage2} = \frac{\dot{m}_{htc}(h_6 - h_{18})}{\eta_{mec\square} * \eta_{elec}}$ $W_{net} = W_{Stage1} + W_{Stage2}$
Subcooler	$\dot{m}_{htc1}(h_7 - h_{19}) = sbc * \dot{m}_{htc2} * (h_{11} - h_{22})$
MTC cascade heat exchanger	$\dot{m}_{htc2} * (1 - sbc) * (h_{10} - h_9) = \dot{m}_{htc} * (h_{15} - h_{16})$
LTC cascade heat exchanger	$\dot{m}_{htc1} * (h_5 - h_8) = \dot{m}_{ltc} * (h_2 - h_3)$
HTC Condenser	$Q_{Cond_{HTC}} = \dot{m}_{htc} * (h_6 - h_7)$

Table 1.2: Exergy Destruction Rate Equation

Component	Exergy Destruction Rate equation
LTC evaporator	$Ex_{Dest_{EVA1}} = \dot{m}_{ltc}(Ex_4 - Ex_1) + \mathrm{Q}_{\mathrm{E}_{LTC}}\left(1 - \frac{\mathrm{To}}{\mathrm{Teva}_{LTC}}\right)$
MTC evaporator	$Ex_{Dest_{EVA2}} = \dot{m}_{mtc}(Ex_{13} - Ex_{14}) + \mathrm{Q}_{\mathrm{E}_{MTC}}\left(1 - \frac{\mathrm{To}}{\mathrm{Teva}_{MTC}}\right)$
LTC compressor	$Ex_{Dest_{COMP_1}} = \dot{m}_{ltc}(Ex_1 - Ex_2) + \dot{W}_{Comp_{LTC}}$
MTC compressor	$Ex_{Dest_{COMP_2}} = \dot{m}_{mtc}(Ex_{14} - Ex_{15}) + \dot{W}_{Comp_{MTC}}$
LTC throttle	$Ex_{Dest_{THR_1}} = \dot{m}_{ltc}(Ex_3 - Ex_4)$
MTC throttle	$Ex_{Dest_{THR_2}} = \dot{m}_{mtc}(Ex_{16} - Ex_{13})$
HTC condenser	$Ex_{Dest_{Cond}} = \dot{m}_{htc}(Ex_6 - Ex_7) + \mathrm{Q}_{\mathrm{Cond}_{HTC}}\left(1 - \frac{To}{Tc}\right)$
LTC cascade heat exchanger	$Ex_{Dest_{HX_1}} = \dot{m}_{ltc}(Ex_2 - Ex_3) + \dot{m}_{htc}(Ex_8 - Ex_5)$
MTC cascade heat exchanger	$Ex_{Dest_{HX_2}} = \dot{m}_{mtc}(Ex_{15} - Ex_{16}) + \dot{m}_{htc}(Ex_9 - Ex_{10})$
HTC Throttle 1	$Ex_{Dest_{THR_3}} = \dot{m}_{htc_1}(Ex_{19} - Ex_8)$
HTC throttle 2	$Ex_{Dest_{THR4}} = \dot{m}_{htc_2}(Ex_7) - \dot{m}_{htc_2} * sbc * Ex_{11} - \dot{m}_{htc_2} * (1 - sbc) * Ex_9$
HTC scroll compressor	$Ex_{Dest_{sc1}} = \dot{m}_{htc_1}(Ex_5 - Ex_{17}) + \dot{W}_{Stage1}$ $Ex_{Dest_{sc2}} = \dot{m}_{htc}(Ex_{18} - Ex_6) + \dot{W}_{Stage2}$ $Ex_{Dest_{SC}} = Ex_{Dest_{sc1}} + Ex_{Dest_{sc2}}$
HTC subcooler	$Ex_{Dest_{SB}} = \dot{m}_{htc_1}(Ex_7 - Ex_{19}) + \dot{m}_{htc_2} * sbc * (Ex_{11} - Ex_{12})$

Exergy efficiency can also be calculated with the help of exergy destruction and total work requirement, the relation is as follows:

$$\eta_{exergy} = 1 - \frac{Ex_{Dest_{NET}}}{W_{Net_{Compressor}}} \tag{2.7}$$

The term $\dot{m}_{ltc}; \dot{m}_{mtc}; \dot{m}_{htc1}; \dot{m}_{htc2}; sbc * \dot{m}_{htc2}$ denotes mass flow rate of refrigerant in LTC, MTC, HTC which connects LTC to HTC, total refrigerant mass flow rate in HTC which connects MTC to HTC, fraction of refrigerant in HTC that passes through subcooler respectively. The forward slash " ´ " over few states points indicates isentropic compression state points. To is the reference/ambient temperature for exergy analysis. Teva_{LTC}; Teva_{MTC}; Tc indicates evaporator temperature in LTC and MTC, and condenser temperature respectively.

2.3. SYSTEM'S ENVIRONMENTAL ASSESSMENET: TEWI

Previously in this work TEWI (Total Equivalent Warming Impact) has been defined as a measure of global warming impact of the system in terms of total associated greenhouse gases emissions during the system's operation and when the operating fluids (refrigerant in this case) are disposed at the end of operational life [17]. It consists of total warming impact due to direct and indirect emissions, direct emissions are related to refrigerant released or leaked during operation, service and maintenance of the system, that can cause global warming, as well as the refrigerant lost while the disposal at end of life of the system, and on other hand, indirect emissions comprises of those which are produced due to CO_2 emissions from fossil fuel that is used to produce electricity or energy that powers the refrigeration system during its operational life.

$$TEWI = Direct\ Emissions + Indirect\ Emissions \tag{2.8}$$

$$TEWI = \left((GWP * m * L_{annual} * n) + GWP * m * \left(1 - \alpha_{recovery}\right)\right) + (E_{annual} * \beta * n) \tag{2.9}$$

Here,

GWP: Global Warming Potential of refrigerant relative to CO_2, (GWP CO_2 = 1)

m: Refrigerant charge (kg)

L_{annual}: Leakage per year (Kg)

n: Operational life of system (Year)

$\alpha_{recovery}$: refrigerant recycling/recovery factor (0 to 1)

E_{annual}: Energy consumption per year (kWh) (= Net compressor work (kW) * operation hours)

β = Indirect emission factor or CO_2 emission factor (kg of CO_2 emitted per kWh of electricity supplied)

Table 2.3: TEWI specifications

<table>
<tr><td>GWP NH_3 = 0</td><td colspan="2">GWP CO_2 = 1</td><td colspan="2">GWP R1243ze(Z) = 6</td></tr>
<tr><td>$L_{annual} rate: 12.5\% p.a.$ [17]</td><td colspan="2">$n = 15$</td><td colspan="2">$\alpha_{recovery} = 70\%$ [17]</td></tr>
<tr><td>E_{annual}= 198166.8 kWh</td><td colspan="3">$\dot{m}_{NH3}$=0.207kg, $\dot{m}_{CO2}$=0.384kg, $\dot{m}_{R1234ze(Z)}$=0.492kg</td><td>$\beta = 0.93$[26]</td></tr>
</table>

These values have been calculated for LTC evaporation temperature of -50̊ C and MTC evaporation temperature of -20̊ C, with 50% load distribution (or x= 0.5), and subcooler operating with 14% of HTC refrigerant. m, mass of the refrigerant or refrigerant charge has been calculated with an assumption of system charging time of 4 seconds. Operation hours has been assumed to 150 hrs. per week as some hours for maintenance and service activities needs to be considered. System's operational life has been assumed to be 15 years. In the case with system's refrigerant charge is greater than 100kg, then $\alpha_{recovery}$ can be considered to be around 90% to max 95% according to the set guidelines [17]. Similarly, annual leakage of refrigerant, L_{annual} has been set for centralized system, under typical annual leak rate category, with maintained system as sub-category [17]. Mittal et al. [26] has calculated carbon dioxide emission factor for Indian power plants, and as majority of Indian power plants operates with coal as fuel, so value of carbon dioxide emission factor is on higher side, with the value of β ranging between 0.92 to 0.94, and average value of 0.93 has been taken for calculation.

CHAPTER 3

RESULTS & DISCUSSION

3.1. COP comparison of the MTTC by Zhu et al. [10] & Novel system

3.1.1. With equal refrigeration load in MTC and LTC evaporator (x = 0.5)

For the comparison, the temperature of evaporator in LTC has been fixed at -45℃ (228.15K), while the evaporation temperature in MTC has been varying over the range -30℃ to -10℃ (243.15-263.15 K). From the fig. 3.1, it can be seen that new MTTC system is superior to old MTTC in performance over the entire temperature range, and as the MTC evaporation temperature increases, this difference increases. Around 13.6% of refrigerant (Ammonia) in HTC passes through subcooler, this helps in substantially reducing the compressor work and hence COP increases.

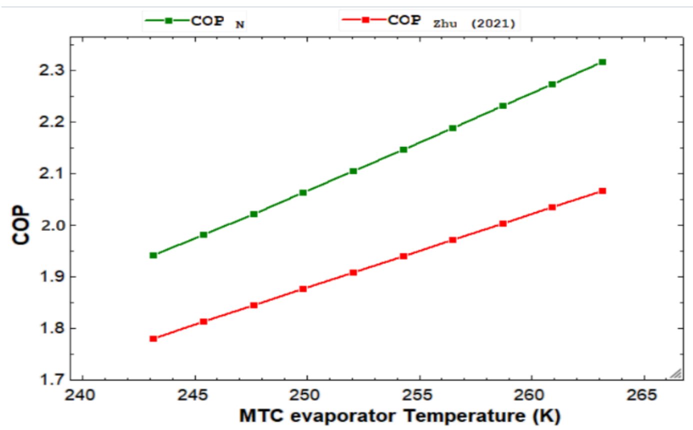

Figure 3.1: COP comparison at x=0.5

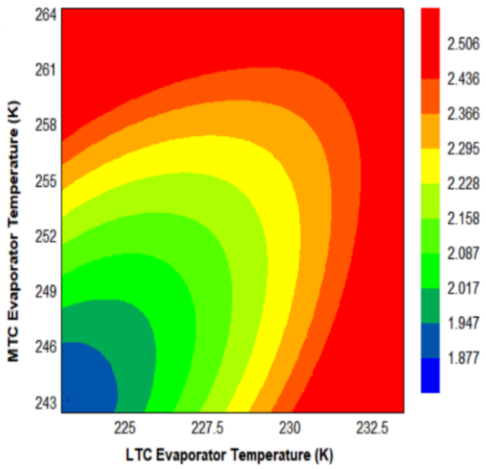

Figure 3.2: COP variation of new system

The improvement in terms of percentage is from 9.01% at 243.15K to 12.17% at 263.15K. Fig. 3.2, represents the COP of the new system with variation in LTC and MTC evaporator temperature at 50% load proportion ratio.

3.1.2. With 90% refrigeration load in MTC evaporator

Fig. 3.3, Similar to last comparison, the LTC evaporator temperature was fixed at -45℃, while the evaporation temperature in MTC evaporator varies from -30℃ to -10˚C (243 to 263K). In this refrigeration load distribution, the overall COPs for both the system is on higher side as more load is being handled by MTC's evaporator. Here again, the New System performs better over the entire temperature variation and in terms of digits, it performs 10.14% better at 243.15K and shows improvement of 14.8% at 263.15K. Here as LTC has to take up only 10% of the refrigeration load, so the subcooler needs only 1.7% (approx.) of the total refrigerant in HTC.

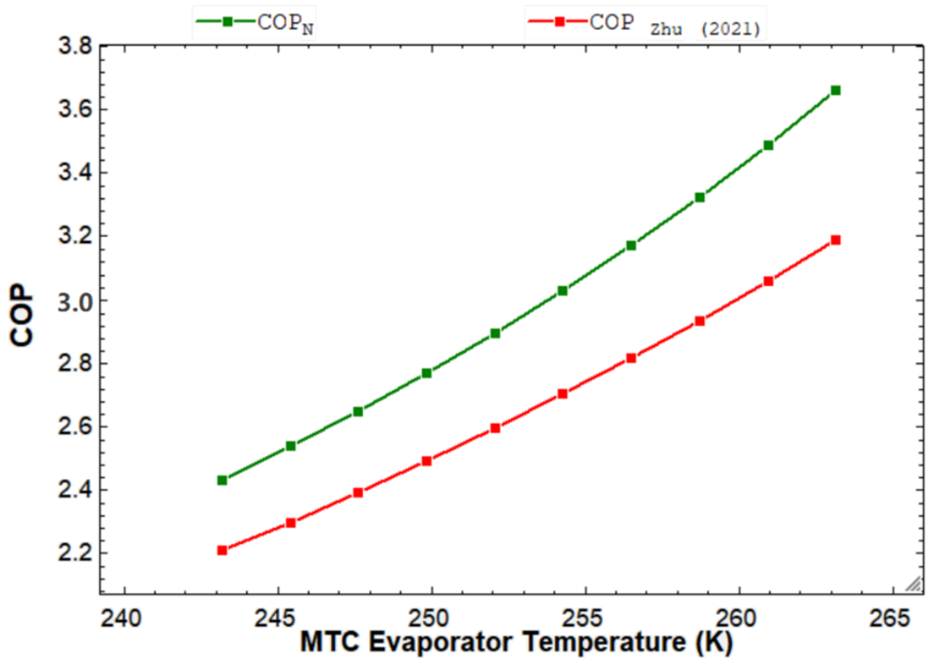

Figure 3.3: COP comparison at x=0.1

3.1.3. With 90% refrigeration load handled by LTC

In this type of load distribution where majority of load being handled is at very low temperature, the role of subcooler becomes much more important, because after getting subcooled, the ammonia in the cascade heat exchange is able to pick up more heat (per kg of refrigerant), which reduces its mass flow rate, leading to reduced compressor work and therefore improves the performance. Along with the subcooler, the separate throttles from heat exchangers, enables to maintain optimum temperature in cascade heat exchanger and hence reduce the overall compressor work. As apparent from fig. 3.4, the new system performs nearly 8% better than previous MTTC at 243.15K and this percentage improvement reaches nearly 21% at 263.15K, almost 58% of refrigerant in HTC passes through subcooler as MTC has just 10% refrigeration load requirement.

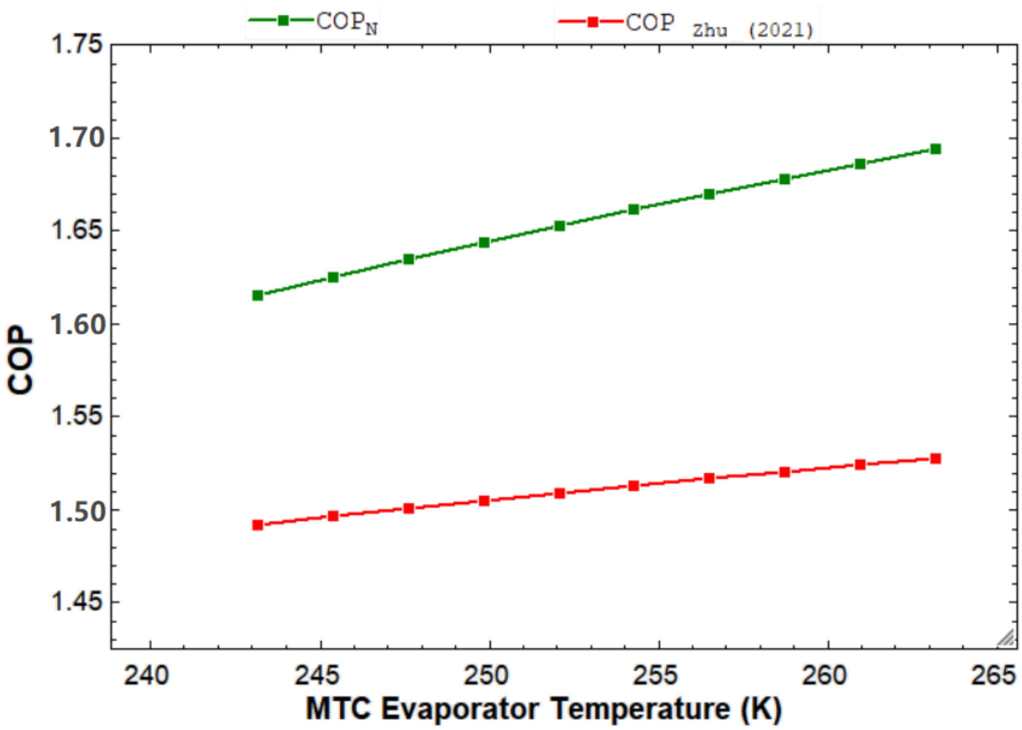

Figure 3.4: COP comparison at x=0.9

3.2. COP comparison for -50℃ evaporation temperature in LTC

With the evaporation temperature set at -50℃ and MTC's evaporator temperature varying from -30℃ to -10˚C and load proportion ratio at 0.5 (equal load distribution). The new system becomes very efficient in terms of COP in comparison with MTTC by Zhu et al. [10]. The system is almost 13% superior in performance at 243.15K and reaches up to 16.31% when MTC evaporator is at 263.15K, and the subcooler was working with approx. 14.5% of ammonia in HTC. This represents the benefits of providing subcooler, and an extra throttle for proving optimum cascade heat exchanger temperature at such low temperature requirements. The fig. 3.5 and table 3.1 has been provided to support the statement.

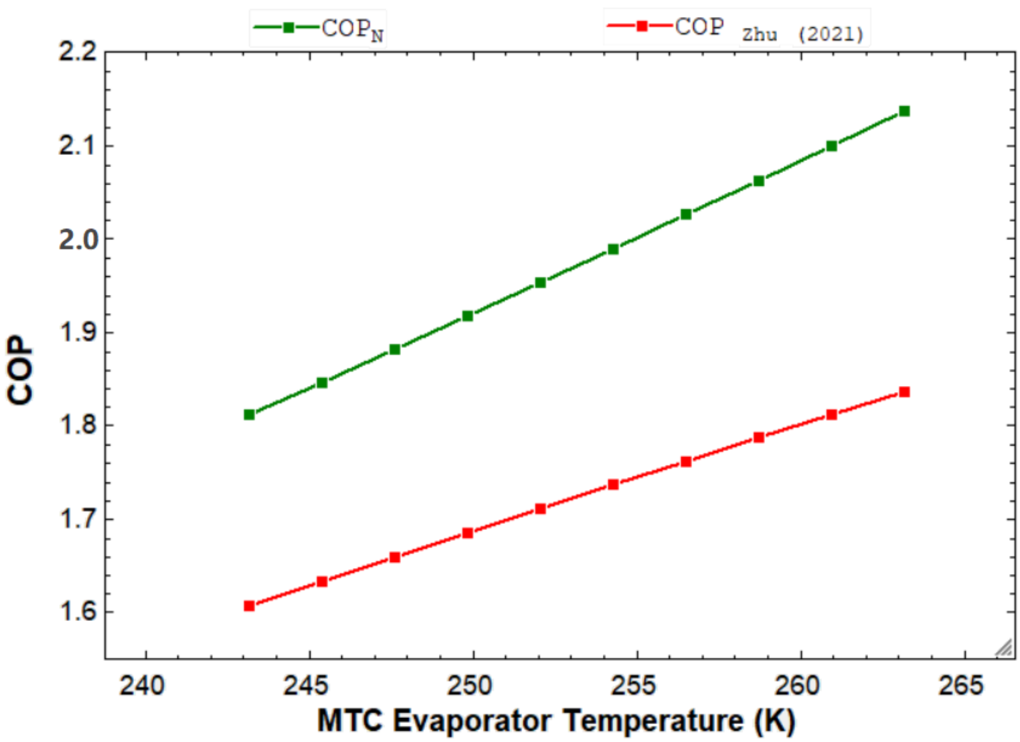

Figure 3.5: COP at -50℃ LTC evaporation temperature

Table 3.1: COP comparison at -50℃ LTC temperature and varying MTC evaporation temperature

S.no	MTC Evaporator Temperature (K)	COP Zhu et al. (2021)	COP Novel system	Percentage Improvement
1	243.15	1.607	1.811	12.68
2	245.15	1.631	1.843	13.00
3	247.15	1.654	1.875	13.33
4	249.15	1.678	1.907	13.67
5	251.15	1.701	1.939	14.02
6	253.15	1.724	1.971	14.38
7	255.15	1.746	2.004	14.74
8	257.15	1.769	2.037	15.12
9	259.15	1.792	2.069	15.51
10	261.15	1.814	2.103	15.90
11	263.15	1.836	2.136	16.31

For another similar comparison, in table 3.2, the evaporation temperature in MTC was fixed at -20˚C, with equal refrigeration load in each circuit, the LTC evaporator temperature was varied from -50℃ to -40℃. Here the similar trend can be observed, and the New upgraded System performs far better than MTTC, almost 14% higher COP at such low temperature, this can significantly reduce the compressor power input and also will be environmentally beneficial. The reason for better performance at very low evaporator temperature is use of separate throttles in HTC which maintains optimum temperature in cascade heat exchanger, whereas, in the old configuration MTTC the cascade temperature was fixed to perform efficiently at -40℃, that's why the percentage difference in improvement gets close, but still due to subcooler, there is performance difference of 7.6%. Another observation from these two comparisons, it can be said that the upgraded system performs very well when there is substantial difference between LTC and MTC's evaporator temperature, because this difference enables the subcooler to work more efficiently and hence increases the COP of the new system.

Table 3.2: COP comparison with MTC-20℃ and varying LTC evaporation temperature

S.no	LTC Evaporator Temperature (K)	COP Zhu et al. (2021)	COP New System	Percentage Improvement
1	223.150	1.724	1.965	13.99
2	224.150	1.764	1.996	13.15
3	225.150	1.804	2.027	12.36
4	226.150	1.844	2.058	11.62
5	227.150	1.884	2.090	10.92
6	228.150	1.924	2.122	10.28
7	229.150	1.964	2.154	9.68
8	230.150	2.003	2.186	9.12
9	231.150	2.043	2.218	8.60
10	232.150	2.082	2.251	8.11
11	233.150	2.121	2.284	7.66

From the above two comparisons, it can be said that the new system performs extremely well when there is substantial difference between LTC and MTC's evaporator temperature, because this difference enables the subcooler to work more efficiently and hence increases the COP of the new system.

3.3. Variation of Load Proportion Ratio (x)

Load proportion ratio, x is varied from 0.1 to 0.9, i.e., from 10 percent refrigeration load handled by LTC to 90 percent refrigeration load being handled by LTC, represented in fig. 3.6. For this the LTC and MTC evaporation temperature were fixed at -45℃ and -20℃. The new system performed nearly 12 % better, and at x=0.1 the new system was around 10% more efficient than the MTTC by Zhu et al. [10].

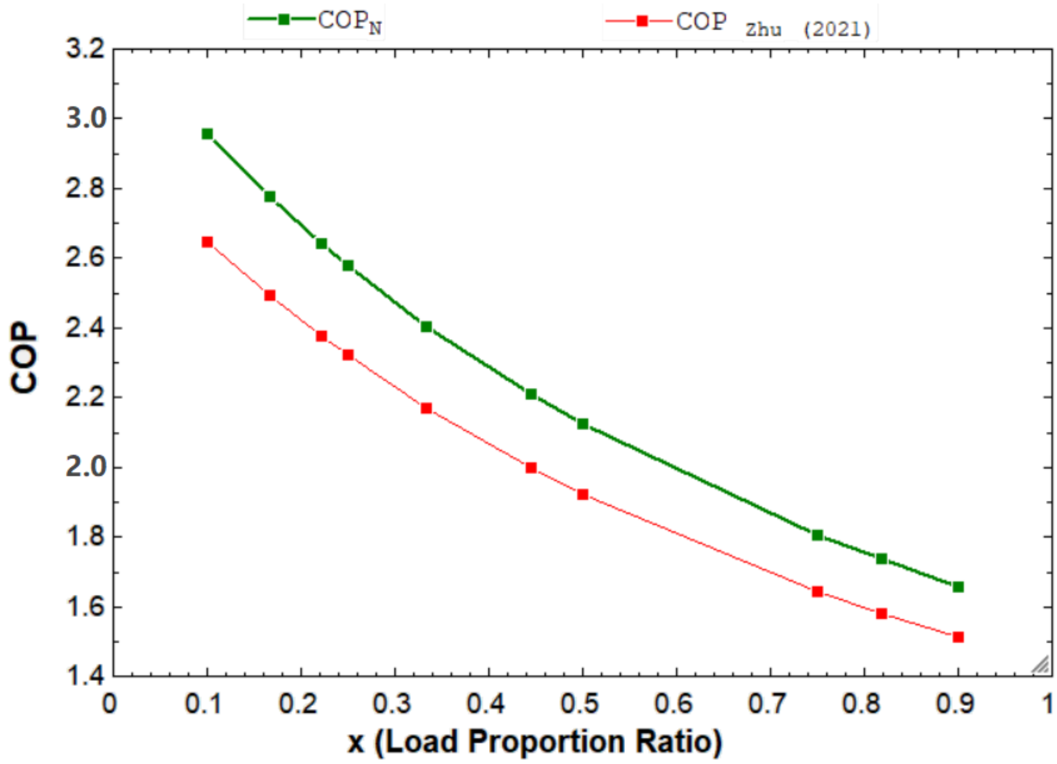

Figure 3.6: COP variation with Load proportion ratio

3.4. Effect of subcooler and two separate throttles in HTC

The biggest edge this new system has over the previous version of MTTC is the use of subcooler and two throttles. The subcooler reduces the quality of ammonia refrigerant entering the HTC/LTC cascade heat exchanger, and this reduced quality then significantly contributes to reduction in compressor work of HTC, while the throttle maintains optimum temperature in cascade heat exchanger and hence increases the COP. Moreover, it was presumed that using a subcooler might increase the ammonia refrigerant mass flow rate in HTC, but in reality, from the results it is clear that there is not much difference is Ammonia’s mass flow rate, but on other hand it has been reduced due to the use of two separate throttles in HTC for each cascade heat exchanger, as ammonia is throttled to optimum temperature for each circuit separately, not like the way in old MTTC, where there was just one throttle in HTC and therefore ammonia was throttled to a fixed temperature irrespective of evaporation temperatures in LTC and MTC. Table no.

3.3, 3.4 and 3.5, have been provided to support the statement, where MTC evaporator is at -20̊ C, with x=0.1, and subcooler operating at 12.045% of HTC's refrigerant. Table 3.3 and 3.4 here provides us the information regarding comparison of LTC compressor work and total compressor work of the system. Ammonia's mass flow rate in the MTTC by Zhu et al. [10] and novel system has been presented in table 3.5.

Table 3.3: LTC compressor work comparison

S.no	LTC Evaporator Temp (Kelvin)	SCX (in %)	COP Zhu et al. (2021)	COP New System	Improvement (%)	LTC Compressor Work (KW) Zhu et al. (2021)	LTC compressor work (kW) Novel System
1	223.150	13.5	1.724	1.967	14.10	11.30	9.76
2	224.150	13.5	1.764	1.998	13.25	10.77	9.52
3	225.150	13.5	1.804	2.029	12.46	10.26	9.30
4	226.150	13.5	1.844	2.060	11.71	9.77	9.08
5	227.150	13.5	1.884	2.092	11.02	9.31	8.87
6	228.150	13.5	1.924	2.123	10.37	8.86	8.66

Table 3.4: Net compressor work comparison

S.no	LTC Evaporator Temp (Kelvin)	SCX (in %)	COP Zhu et al. (2021)	COP New system	Improvement (%)	NET Compressor Work (KW) Zhu et al. (2021)	NET compressor work (kW) Novel system
1	223.150	13.5	1.724	1.967	14.10	29.01	25.43
2	224.150	13.5	1.764	1.998	13.25	28.35	25.03
3	225.150	13.5	1.804	2.029	12.46	27.72	24.65
4	226.150	13.5	1.844	2.060	11.71	27.11	24.27
5	227.150	13.5	1.884	2.092	11.02	26.54	23.90
6	228.150	13.5	1.924	2.123	10.37	25.99	23.55

Table 3.5: NH_3 mass flow rate comparison in HTC

S.no	LTC Evaporator Temp (Kelvin)	SCX (in %)	COP Zhu et al. (2021)	COP New System	Improvement (%)	NH_3 mass flow rate (kg/s) Zhu et al. (2021)	NH_3 mass flow rate (kg/s) Novel system
1	223.150	13.5	1.724	1.967	14.10	0.0519	0.0558
2	224.150	13.5	1.764	1.998	13.25	0.0518	0.0554
3	225.150	13.5	1.804	2.029	12.46	0.0517	0.0549
4	226.150	13.5	1.844	2.060	11.71	0.0516	0.0545
5	227.150	13.5	1.884	2.092	11.02	0.0515	0.0541
6	228.150	13.5	1.924	2.123	10.37	0.0515	0.0538

3.5. EXERGY ANALYSIS

The exergy efficiency or second law efficiency of refrigeration system can be defined as the ratio of thermal exergy of the load or heat picked up by the evaporator from cooling space to the compressor work done. The novel system's exergy efficiency has been calculated at various LTC and MTC evaporator temperature with different load proportion ratio, "x". The ambient temperature was assumed to be at 298.15K and 1 atm pressure.

3.5.1. With equal refrigeration load in MTC and LTC evaporator (x=0.5)

Exergy efficiency of the new system with the LTC evaporator at -45℃ and MTC evaporator temperature varying from -30℃ to -10℃, subcooler operating with 16% of HTC refrigerant, and condenser temperature at 40℃ with equal load in both circuit or x=1 has been provided in table 4. system operating at Table 3.6. With increase in MTC evaporator temperature, exergy destruction rate decreases as evaporator temperature reaches near to ambient temperature, but at the same time useful exergy also decreases, therefore exergy efficiency also decreases. At MTC temperature of -30℃, and with LTC evaporator of -45℃, exergy destruction rate is 18.48, and total compressor power is 31.82, and therefore using the equation 5, we get exergy efficiency of 0.419 or 41.9%.

Table 3.6: Exergy efficiency with x=0.5

MTC Evaporator Temperature (K)	COP	Exergy efficiency	Exergy Destruction rate (kW)
243.15	1.571	0.419	18.48
245.15	1.598	0.418	18.19
247.15	1.625	0.417	17.91
249.15	1.652	0.416	17.65
251.15	1.679	0.415	17.40
253.15	1.706	0.413	17.15
255.15	1.733	0.412	16.92
257.15	1.760	0.410	16.69
259.15	1.787	0.409	16.48
261.15	1.814	0.407	16.27
263.15	1.842	0.405	16.07

3.5.2. With 90% refrigeration load in MTC evaporator

The exergy efficiency of the new system with 90% load being handled by MTC evaporator which varies from -30℃ to -10℃, LTC fixed at -45℃, condenser temperature at 40℃, and subcooler operating with 2% of HTC's refrigerant is presented in table 3.7. Similar to the COP, the exergy is also on higher side as compared to case with x=1. The exergy destruction rate is higher because of increased overall refrigeration load. Like the previous case, as the MTC temperature increase, exergy destruction rate reduces along with exergy efficiency, reason being the declining difference in the evaporator and ambient temperature.

Table 3.7: Exergy efficiency at x=0.1

MTC Evaporator Temperature (K)	COP	Exergy efficiency	Exergy Destruction rate (kW)
243.15	1.973	0.462	68.14
245.15	2.042	0.460	66.10
247.15	2.113	0.457	64.15
249.15	2.188	0.454	62.28
251.15	2.266	0.451	60.48
253.15	2.348	0.448	58.75
255.15	2.433	0.444	57.08
257.15	2.522	0.439	55.47
259.15	2.616	0.435	53.92
261.15	2.715	0.429	52.42
263.15	2.818	0.424	50.97

3.5.3. With 90% refrigeration load being handled by LTC

Table 3.8 presents the exergy efficiency of the system, with LTC fixed at -45℃, MTC evaporator varying from -30℃ to -10℃, condenser temperature at 40℃ and subcooler operating with 63% refrigerant. Here, with the increase in temperature the exergy efficiency increases, the reason can be explained from the equation 4, as the LTC temperature is fixed, exergy of heat picked up by LTC which contributes 90%, remains constant as LTC evaporator temperature is fixed, whereas the net compressor work reduces due to increase in temperature of MTC and hence exergy efficiency increases as the MTC temperature increases.

Table 3.8: Exergy efficiency with x=0.9

MTC Evaporator Temperature (K)	COP	Exergy efficiency	Exergy Destruction rate (kW)
243.15	1.305	0.390	12.97
245.15	1.313	0.391	12.88
247.15	1.320	0.392	12.79
249.15	1.326	0.392	12.71
251.15	1.333	0.393	12.63
253.15	1.339	0.394	12.56
255.15	1.345	0.394	12.49
257.15	1.351	0.395	12.42
259.15	1.357	0.395	12.36
261.15	1.362	0.395	12.30
263.15	1.367	0.396	12.24

3.6. ENVIRONMENTAL ASPECT: Total Equivalent Warming Impact (TEWI)

The TEWI stipulates the global warming impact of the system in terms of total associated greenhouse gases emissions during the system's operation and when the operating fluids (refrigerant in this case) are disposed at the end of operational life. TEWI is calculated in equivalent CO_2 or e-CO_2. The table no. 3.9 below depicts the TEWI for the upgraded system under different LTC and MTC evaporator temperature, and varying load proportion ratio ("x"). For the TEWI calculations, the LTC load has been assumed to be of 25kWand MTC load changes with "x". In these results, "x" varies from 0.1 to 0.9, LTC evaporator temperature ranges from -50℃ to -30℃, and MTC evaporator varies from -30℃ to -10℃. From the results, it can be seen that lower is the evaporator temperature, higher is the TEWI as refrigerant charge in system as well as compressor work are also high. Increase of 10℃ in evaporator temperature of LTC reduces TEWI by almost 10%, and for MTC evaporator temperature it reduces by roughly 13%, as GWP of R1234ze(Z) is more than CO_2, so the TEWI reduction is also little more. Considering the fact that, system produces e-CO_2 in range of thousands of metric tons, so even 10% reduction of TEWI can greatly benefit the environment and make a significant difference.

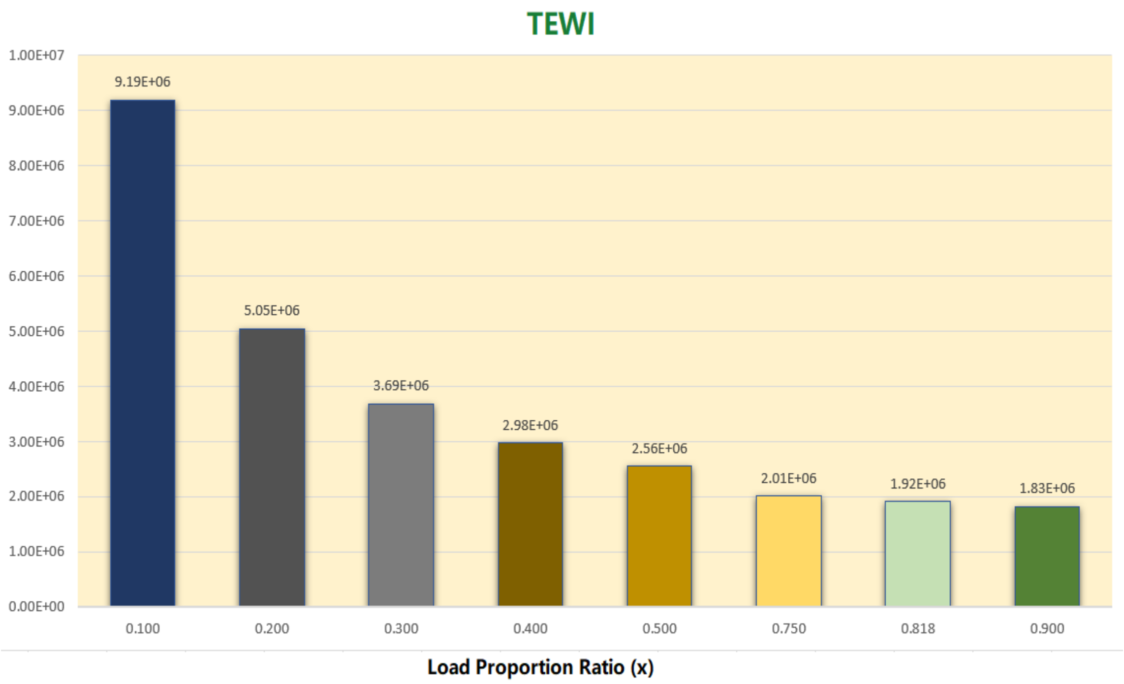

Figure 3.7: TEWI (LTC at 223.15K MTC at 253.15K, Tc = 301.15K)

Table 3.9: TEWI

LTC evap temp(K) T_{ev1}	MTC evap temp(K) T_{ev2}	Load Proportion Ratio (x)	CO_2 Charge (kg)	NH_3 charge (kg)	R1234ze(Z) Charge (kg)	Net compressor work (kW)	TEWI (e-CO_2)
228.150	243.150	0.5	0.392	0.208	0.492	25.762	2.803E+06
228.150	243.150	0.1	0.392	1.011	4.432	102.853	1.119E+07
228.150	243.150	0.9	0.392	0.119	0.056	17.198	1.871E+06
223.150	243.150	0.5	0.384	0.210	0.492	27.626	3.006E+06
223.150	253.150	0.5	0.384	0.207	0.492	25.406	2.764E+06
223.150	263.150	0.5	0.384	0.205	0.496	23.403	2.547E+06
238.150	253.150	0.5	0.412	0.202	0.492	20.420	2.222E+06
238.150	263.150	0.5	0.412	0.200	0.496	18.487	2.012E+06

CHAPTER 4

CONCLUSION AND SCOPE FOR FUTURE WORK

4.1. CONCLUSION

The MTTC system developed by Zhu et al [10] was in itself a new configuration for multi evaporator refrigeration system and it outperformed combined low/high temperature circuit cascade refrigeration systems. The performance of novel MTTC system proposed in this study, has surpassed the MTTC by Zhu et al. [10] in all operating and load conditions. The use of two throttles in high temperature circuit, along with a much efficient scroll compressor and a subcooler proved to be extremely beneficial. It can be noted that more is the difference in evaporation temperature of LTC and MTC, the better is the performance of subcooler and hence the system's performance increases significantly as compared to previous version of MTTC system. The performance in terms of COP increased by 7% on the lower side, and reached up nearly to 21 % in some conditions. Exergy efficiency and exergy destruction rate of the system were also presented. Total equivalent warming impact, that captures all the carbon dioxide emissions during the system's operational life was calculated for various temperature and load conditions in this work and from that it can be said that, higher is the load proportion ratio, i.e., more load on low temperature evaporator, lower will be the value of TEWI.

4.2. Scope for Future work

- Exergo-economic analysis of the novel multi target temperature cascades system can be performed.
- Refrigerant in medium temperature circuit can be varied for investigating other temperature and load requirement for optimum results in different scenario.
- Another circuit can be added in parallel with LTC and MTC, to cater the load at different target temperature.
- TEWI with different operational conditions can be calculated to study the detailed environmental analysis.
- More inclusive environmental assessment of the system's impact can be studied by calculating Life Cycle Climate Performance (LCCP).

APPENDIX

A.1. EES PROGRAM FOR Novel MTTC System

```
To=298.15 "Dead state conditions"
Po=101.325
CC=7.1"refrigeration in TR, nearly 25kW"

"Te1=(233.15)" "Evaporator Temp oF LTC"
Tc1=(0+273.15) "Cond temp of LTC"
 "Te2=(263.15)"  "Temp of evaporator 2 HTC"
"Tc2=(313.15)" "Temp of Cond2 HTC"

Tcc= (-13+273.15) "CHX temp of MTTC by ZHU"
Thx1= ((Te1)*(Tc2))^0.5
Thx2=((Te2)*(Tc2))^0.5
Thx3= 273.15
"PP=1" “FACTOR used in place of load proportion ratio”

“MTTC SYSTEM BY ZHU ET AL. [10] CODE”

"S2 refers to MTTC system by Zhu et al. [10]"

"CO2 LTC "

"Low temp circuit"
"hx_S2 denotes specific enthalpy at state point x in the LTC of system"
 "Sx_S2 denotes specific entropy at state point x in the LTC of  system"
```

```
"properties at state point 1" "After LTC Evaporator"
"T1=Te1"
h1_S2=enthalpy(CarbonDioxide,T=T1,x=1)
"Pe1=Pressure(CarbonDioxide,T=T1,x=1)"
s1_S2=entropy(CarbonDioxide,T=T1,x=1)

"properties at state point 2" "After LTC Compressor"
s2_S2=s1_S2
h2_S2=enthalpy(CarbonDioxide,s=s2_S2,P=Pc1_S2)
Pc1_S2=p_sat(CarbonDioxide,T=Tcc+5)
 h2_act_S2=((h2_S2-h1_S2)/zz_s2)+h1_S2 "Issentropic Efficiency "
s2_act_S2=entropy(CarbonDioxide,P=Pc1_S2,h=h2_act_S2)
 zz_s2=0.00476*(Pc1_S2/Pe1)*(Pc1_S2/Pe1)-0.0923*(Pc1_S2/Pe1)+0.89810 “isentropic
compressor efficiency”

 "properties at state point 3" "After LTC Condenser"
"Pc1=P_sat(CarbonDioxide,T=Tc1)"

h3_S2=enthalpy(CarbonDioxide,T=Tcc+5,x=0)
s3_S2=entropy(CarbonDioxide,T=Tcc+5,x=0)

"properties at state point 4" "After LTC Throttle"
h3_S2=h4_S2
x4_S2=quality(CarbonDioxide,T=T1,h=h4_S2)
s4_S2=entropy(CarbonDioxide,T=T1,x=x4_S2)

"Calculations"
qe1_S2=h1_S2-h4_S2"specific refrigeration effect of LTC"
WC1_S2= ((h2_act_S2)-h1_S2)/0.93 "specific compressor work of LTC"
qc1_S2=h2_S2-h3_S2 "heat discipated in condenser "

Qe_dot_S2=25" Refigeration effect in KW"
Qe_dot_S2=qe1_S2*MrLTC_dot_S2
```

```
Wc1_dot_S2= WC1_S2*MrLTC_dot_S2 "compressor work"
Qc1_dot_S2=MrLTC_dot_S2*qc1_S2 "Cascade heat exchanger "


"HTC  with AMMONIA"

"hx_AMMONIA_S2 denotes specific enthalpy at state point x in the HTC of system"
 "Sx_AMMONIA_S2 denotes specific entropy at state point x in the HTC of  system"

"Properties of state point7" "After HTC Condenser"
(Pc2_AMMONIA_S2)=p_sat(Ammonia,T=Tc2)
h7_AMMONIA_S2=enthalpy(Ammonia,T=Tc2,x=0)
s7_AMMONIA_S2=entropy(Ammonia,T=Tc2,x=0)

"Properties of state point8" "After HTC Throttle"
(h7_AMMONIA_S2)=(h8_AMMONIA_S2)
(x8_AMMONIA_S2)=quality(Ammonia,T=Tcc,h=(h8_AMMONIA_S2))
s8_AMMONIA_S2=entropy(Ammonia,T=Tcc,x=(x8_AMMONIA_S2))

MrHTC_dot_AMMONIA_S2*(h5_AMMONIA_S2-h8_AMMONIA_S2) =
MrLTC_dot_S2*(h2_act_S2-h3_S2) + MrMT_dot_propylene_S2*(h10_act_propylene_S2-
h11_propylene_S2) “cascade heat exchanger”

"Properties of state point5" "After HTC Evaporator"
h5_AMMONIA_S2=enthalpy(Ammonia,T=Tcc,x=1)
s5_AMMONIA_S2=entropy(Ammonia,T=Tcc,x=1)
Pe2_AMMONIA_S2=pressure(Ammonia,T=Tcc,x=1)

"Properties of state point6" "After HTC Compressor"
(s5_AMMONIA_S2)=(s6_AMMONIA_S2)
h6_AMMONIA_S2=enthalpy(Ammonia,P=(Pc2_AMMONIA_S2),s=(s6_AMMONIA_S2))
h6_act_AMMONIA_S2= (((h6_AMMONIA_S2)-(h5_AMMONIA_S2))/YY_s2) +
h5_AMMONIA_S2 "Issentropic Efficiency "
```

s6_act_AMMONIA_S2=**entropy**(Ammonia,***h***=(h6_act_AMMONIA_S2),P=Pc2_AMMONIA_S2)

YY_s2=-0.00097*(Pc2_AMMONIA_S2/Pe2_AMMONIA_S2)*(Pc2_AMMONIA_S2/Pe2_AMMONIA_S2)-0.01026*(Pc2_AMMONIA_S2/Pe2_AMMONIA_S2)+0.83955 "Issentropic efficiency of compressor"

qee_AMMONIA_S2= (h5_AMMONIA_S2)-(h8_AMMONIA_S2) "enthalpy drop in evaporator at HTC"
wc2_AMMONIA_S2=(h6_AMMONIA_S2)-(h5_AMMONIA_S2)" specific compressor work oh HTC"
WC2act_AMMONIA_S2= (((h6_act_AMMONIA_S2)-(h5_AMMONIA_S2))/0.93) "actual specific compressor work of HTC"

Qe1_dot_AMMONIA_S2=MrHTC_dot_AMMONIA_S2*qee_AMMONIA_S2 "CASCADE Heat Exchanger"
Qc2_dot_AMMONIA_S2= (MrHTC_dot_AMMONIA_S2)*(((h6_act_AMMONIA_S2)-(h7_AMMONIA_S2))) " condenser "
Wc2_dot_AMMONIA_S2=MrHTC_dot_AMMONIA_S2*wc2_AMMONIA_S2 "100% Issentropic compressor"
Wc2_act_dot_AMMONIA_S2= WC2act_AMMONIA_S2*MrHTC_dot_AMMONIA_S2 "real efficiency"

"MT with propylene"

"hx_propylene_S2 denotes specific enthalpy at state point x in the MTC of system"
"Sx_propylene_S2 denotes specific entropy at state point x in the MTC of system"

soo_propylene_S2=**entropy**(***Propylene***,T=To,P=Po)
hoo_propylene_S2=**enthalpy**(***Propylene***,T=To,P=Po)

"Properties of state point 11" "After MT propylene Condenser"

(Pc2_propylene_S2)=**p_sat**(***Propylene***,T=Tcc+5)

(Pe2_propylene_S2)=**p_sat**(***Propylene***,T=Te2)

pr_s2=Pc2_propylene_S2/Pe2_propylene_S2

H11_propylene_S2=**enthalpy**(***Propylene***,x=0,P=(Pc2_propylene_S2))

S11_propylene_S2=**entropy**(***Propylene***,x=0,P=(Pc2_propylene_S2))

"Properties of state point 12" "After MT propylene Throttle"

(h11_propylene_S2)=(h12_propylene_S2)

(x12_propylene_S2)=**quality**(***Propylene***,T=Te2,h=(h12_propylene_S2))

S12_propylene_S2=**entropy**(***Propylene***,T=Te2,x=(x12_propylene_S2))

"MrHTC_dot_AMMONIA_S2*(h5_propylene_S2-h8_propylene_S2) = MrLTC_dot_CO2*(h2-h3) + MrMT_dot_propylene_S2*(h10-h11)"

"Properties of state point 9" "After MT propylene Evaporator"

H9_propylene_S2=**enthalpy**(***Propylene***,T=Te2,x=1)

S9_propylene_S2=**entropy**(***Propylene***,T=Te2,x=1)

Pe_propylene_S2=**pressure**(***Propylene***,T=Te2,x=1)

"Properties of state point6" "After MT propylene Compressor"

(s9_propylene_S2)=(s10_propylene_S2)

H10_propylene_S2=**enthalpy**(***Propylene***,P=(Pc2_propylene_S2),***s***=(s10_propylene_S2))

H10_act_propylene_S2= (((h10_propylene_S2)-(h9_propylene_S2))/0.75) + h9_propylene_S2 "Issentropic Efficiency =0.75"

ww_S2=0.8014- 0.04842*(pr_s2)

S10_act_propylene_S2=**entropy**(Propylene,***h***=(h10_act_propylene_S2),P=Pc2_propylene_S2)

qee_propylene_S2= (h9_propylene_S2)-(h12_propylene_S2) "enthalpy drop in evaporator at MT propylene"

wc2_propylene_S2=(h10_propylene_S2)-(h9_propylene_S2)" specific compressor work of MT propylene"

"WC2act_propylene_S2= ((h10_act_propylene_S2)-(h9_propylene_S2))/0.93" "actual specific compressor work of MT propylene"

WC2act_propylene_S2=((h10_propylene_S2)-(h9_propylene_S2))/0.75 "FOR SAKE OF EQUIVALENT RESULT WITH R1234ZE"

Qe1_dot_propylene_S2=PP*25 "specific refrigeration effect MT"
Qe1_dot_propylene_S2=MrMT_dot_propylene_S2*qee_propylene_S2
Qc2_dot_propylene_S2= (MrMT_dot_propylene_S2)*((h10_act_propylene_S2)-(h11_propylene_S2)) "condeneser part of MT CHX"
Wc2_dot_propylene_S2=MrMT_dot_propylene_S2*wc2_propylene_S2 "100% Issentropic compressor"
Wc2_act_dot_propylene_S2= WC2act_propylene_S2*MrMT_dot_propylene_S2 "Issentropic Efficiency =0.75"

Wnet_dot_act_propylene_S2CO2= Wc1_dot_S2+Wc2_act_dot_propylene_S2+ Wc2_act_dot_AMMONIA_S2 "NET COMPRESSOR WORK"
COP_MTTC= (25+(PP*25))/Wnet_dot_act_propylene_S2CO2

EXERGY_efficiency_S2= ((25*(((To)/(Te1))-1))+(PP*25*(((To)/(Te2))-1)))/Wnet_dot_act_propylene_S2CO2

“**NOVEL SYSTEM CODE**”
"0<SCX<1" "SCX denotes fraction of HTC refrigerant in Subcooler"
"scx=0.11"
x= 1/(1+PP) "LOAD PROPORTION RATIO as defined by Zhu et al. [10]"

"CO2 LTC "
"Low temp circuit"
"hx_SC denotes specific enthalpy at state point x in the LTC of system"
"Sx_SC denotes specific entropy at state point x in the LTC of system"
"properties at state point 1" "After LTC Evaporator"

```
"T1=Te1"
h1_SC=enthalpy(CarbonDioxide,T=T1,x=1)
"Pe1=Pressure(CarbonDioxide,T=T1,x=1)"
s1_SC=entropy(CarbonDioxide,T=T1,x=1)

"properties at state point 2" "After LTC Compressor"
s2_SC=s1_SC
h2_SC=enthalpy(CarbonDioxide,s=s2_SC,P=Pc1_SC)
Pc1_SC=p_sat(CarbonDioxide,T=Thx1+2.5)
 h2_act_SC=((h2_SC-h1_SC)/zz_SC)+h1_SC "Issentropic Efficiency "
s2_act_SC=entropy(CarbonDioxide,P=Pc1_SC,h=h2_act_SC)

zz_SC=0.00476*(Pc1_SC/Pe1)*(Pc1_SC/Pe1)-0.0923*(Pc1_SC/Pe1)+0.89810 "isentropic
efficiency of compressor with CO2"

"properties at state point 3" "After LTC Condenser"
"Pc1=P_sat(CarbonDioxide,T=Tc1)"

h3_SC=enthalpy(CarbonDioxide,T=Thx1+2.5,x=0)
s3_SC=entropy(CarbonDioxide,T=Thx1+2.5,x=0)

"properties at state point 4" "After LTC Throttle"
h3_SC=h4_SC
x4_SC=quality(CarbonDioxide,T=T1,h=h4_SC)
s4_SC=entropy(CarbonDioxide,T=T1,x=x4_SC)

"Calculations"
qe1_SC=h1_SC-h4_SC"specific refrigeration effect of LTC"
 WC1_SC= ((h2_act_SC)-h1_SC)/0.93 "specific compressor work of LTC"
qc1_SC=h2_SC-h3_SC "heat discipated in condenser "
Qe_dot_SC=25 " Refigeration effect in KW"
Qe_dot_SC=qe1_SC*MrLTC_dot_SC "refrigertion effect in kW"
Wc1_dot_SC= WC1_SC*MrLTC_dot_SC "Compressor work"
```

Qc1_dot_SC=MrLTC_dot_SC*qc1_SC "heat exchanged through CHX1"

"HTC with AMMONIA"

"hx_AMMONIA_SC denotes specific enthalpy at state point x in the HTC of system"
"Sx_AMMONIA_SC denotes specific entropy at state point x in the HTC of system"
"Properties of state point7" "After HTC Condenser"
(Pc2_AMMONIA_SC)=**p_sat**(***Ammonia***,T=Tc2)
h7_AMMONIA_SC=**enthalpy**(***Ammonia***,T=Tc2,x=0)
s7_AMMONIA_SC=**entropy**(***Ammonia***,T=Tc2,x=0)

"After 7, it gets split into 77, 9 and 99"
MrHTC_dot_AMMONIA_SC=MrHTC1_dot_AMMONIA_SC+(SCX*MrHTC2_dot_AMMONIA_SC)+((1-SCX)*MrHTC2_dot_AMMONIA_SC)
MrHTC1_dot_AMMONIA_SC*(h7_AMMONIA_SC-h77)=
MrHTC2_dot_AMMONIA_SC*SCX*(h100-h99) "subcooling"

"here state point 12,77,99,100,22,32 referes to 10,19,11,12,17, and 18 respectively in the Fig. 2.1"
h100= **enthalpy**(***Ammonia***, ***x***=1, P=Pe2_2AMMONIA_SC)
h99=h7_AMMONIA_SC

s100=**entropy**(***Ammonia***, ***x***=1, ***h***=h100)
s77=**entropy**(***Ammonia***, ***h***=h77, P=Pc2_Ammonia_SC)

(x99)=**quality**(***Ammonia***,***T***=Thx2-2.5,***h***=(h99))
s99=**entropy**(***Ammonia***,***T***=Thx2-2.5,***x***=(x99))
(h77)=(h8_AMMONIA_SC)
(x8_AMMONIA_SC)=**quality**(***Ammonia***,T=Thx1-2.5,***h***=(h8_AMMONIA_SC))
s8_AMMONIA_SC=**entropy**(***Ammonia***,T=Thx1-2.5,***x***=(x8_AMMONIA_SC))

"Properties of state point5" "After HTC Evaporator"

h5_AMMONIA_SC=**enthalpy**(***Ammonia***,T=Thx1-2.5,***x***=1)
s5_AMMONIA_SC=**entropy**(***Ammonia***,T=Thx1-2.5,***x***=1)
Pe2_AMMONIA_SC=**pressure**(***Ammonia***,***T***=Thx1-2.5,***x***=1)

"Properties of state point6" "After HTC Compressor"
(s5_AMMONIA_SC)=(s22_AMMONIA_SC)
h22_AMMONIA_SC=**enthalpy**(***Ammonia***,***P***=(Pe2_2AMMONIA_SC),***s***=(s22_AMMONIA_SC))
h22_act_AMMONIA_SC= (((h22_AMMONIA_SC)-(h5_AMMONIA_SC))/YY_SC) + h5_AMMONIA_SC "Issentropic Efficiency =0.7"
s22_act_AMMONIA_SC=**entropy**(Ammonia,***h***=(h22_act_AMMONIA_SC),P=Pe2_2AMMONIA_SC)

YY_SC=-0.00097*(Pe2_2AMMONIA_SC/Pe2_AMMONIA_SC)*(Pe2_2AMMONIA_SC/Pe2_AMMONIA_SC)-0.01026*(Pe2_2AMMONIA_SC/Pe2_AMMONIA_SC)+0.83955 " compressor isentropic efficiency"

qee_AMMONIA_SC= (h5_AMMONIA_SC)-(h8_AMMONIA_SC) "enthalpy drop in evaporator at HTC"
wc2_AMMONIA_SC=(h22_AMMONIA_SC)-(h5_AMMONIA_SC)" specific compressor work oh HTC"
WC2act_AMMONIA_SC= (((h22_act_AMMONIA_SC)-(h5_AMMONIA_SC))/0.93) "actual specific compressor work of HTC"

Qe1_dot_AMMONIA_SC=MrHTC1_dot_AMMONIA_SC*qee_AMMONIA_SC "Refrigeration effect"

Qe1_dot_AMMONIA_SC=Qc1_dot_SC " CASCADE HEAT EXCHANGER 1"

Qc2_dot_AMMONIA_SC= (MrHTC_dot_AMMONIA_SC)*(((h6_act_AMMONIA_SC)-(h7_AMMONIA_SC))) " Condenser heat exchange"

Wc2_1_dot_AMMONIA_SC=MrHTC1_dot_AMMONIA_SC*wc2_AMMONIA_SC "100% Issentropic scroll compressor"
Wc2_1_act_dot_AMMONIA_SC= WC2act_AMMONIA_SC*MrHTC1_dot_AMMONIA_SC "real efficiency"

"FOR MT/HT"
"Properties of state point7" "After HTC Condenser"
"(Pc2_AMMONIA_SC)=p_sat(Ammonia,T=Tc2)"
"h7_AMMONIA_SC=enthalpy(Ammonia,T=Tc2,x=0)"
"s7_AMMONIA_SC=entropy(Ammonia,T=Tc2,x=0)"

"After 7, it gets split into 8 and 9"
"MrHTC_dot_AMMONIA_SC=MrHTC1_dot_AMMONIA_SC+MrHTC2_dot_AMMONIA_SC"

"Properties of state point9" "After second HTC Throttle"
(h7_AMMONIA_SC)=(h9_AMMONIA_SC)
(x9_AMMONIA_SC)=**quality**(***Ammonia***,T=Thx2-2.5,***h***=(h9_AMMONIA_SC))
s9_AMMONIA_SC=**entropy**(***Ammonia***,T=Thx2-2.5,***x***=(x9_AMMONIA_SC))

"Properties of state point 12" "After Hx2 Evaporator"
h12_AMMONIA_SC=**enthalpy**(***Ammonia***,***T***=Thx2-2.5,***x***=1)
s12_AMMONIA_SC=**entropy**(***Ammonia***,T=Thx2-2.5,***x***=1)
Pe2_2AMMONIA_SC=**pressure**(Ammonia,***T***=Thx2-2.5,***x***=1)

MrHTC1_dot_AMMONIA_SC*h22_act_AMMONIA_SC+ (1-SCX)*MrHTC2_dot_AMMONIA_SC*h12_AMMONIA_SC+MrHTC2_dot_AMMONIA_SC*SCX* h100=MrHTC_dot_AMMONIA_SC*h32_AMMONIA_SC "SCROLL COMPRESSOR MIXING"

"Properties of state point6" "After HTC Compressor"
s32_AMMONIA_SC=**entropy**(***Ammonia***, h=h32_AMMONIA_SC, p= Pe2_2AMMONIA_SC)

(s32_AMMONIA_SC)=(s6_AMMONIA_SC)

h6_AMMONIA_SC=enthalpy(Ammonia,P=(Pc2_AMMONIA_SC),s=(s6_AMMONIA_SC)) "100% isentropic efficiency"
h6_act_AMMONIA_SC= (((h6_AMMONIA_SC)-(h32_AMMONIA_SC))/YY_S2_2sc) + h32_AMMONIA_SC "real efficiency"
s6_act_AMMONIA_SC=**entropy**(Ammonia,***h***=(h6_act_AMMONIA_SC),P=Pc2_AMMONIA_SC)

YY_S2_2sc=-0.00097*(Pc2_AMMONIA_SC/Pe2_2AMMONIA_SC)*(Pc2_AMMONIA_SC/Pe2_2AMMONIA_SC)-0.01026*(Pc2_AMMONIA_SC/Pe2_2AMMONIA_SC)+0.83955 "Compressor isentropic efficicncy"

qee_2AMMONIA_SC= (h12_AMMONIA_SC)-(h9_AMMONIA_SC) "enthalpy drop in evaporator at HTC"
wc2_2AMMONIA_SC=(h6_AMMONIA_SC)-(h32_AMMONIA_SC)" specific compressor work oh HTC"
WC2act_2AMMONIA_SC= (((h6_act_AMMONIA_SC)-(h32_AMMONIA_SC))/0.93) "actual specific compressor work of HTC"

Qe1_dot_2AMMONIA_SC=(1-(SCX))*MrHTC2_dot_AMMONIA_SC*qee_2AMMONIA_SC
"Qc2_dot_AMMONIA_SC= (MrHTC_dot_AMMONIA_SC)*(((h6_act_AMMONIA_SC)-(h7_AMMONIA_SC)))"
Wc2_2_dot_2AMMONIA_SC=MrHTC_dot_AMMONIA_SC*wc2_2AMMONIA_SC "Issentropic compressor work"
Wc2_2_act_dot_2AMMONIA_SC= WC2act_2AMMONIA_SC*MrHTC_dot_AMMONIA_SC "compressor work with real efficiency"

"MT with R1234ze(Z)"
soo_R1234ze_SC=**entropy**(***R1234ze(Z)***,T=To,P=Po)
hoo_R1234ze_SC=**enthalpy**(***R1234ze(Z)***,T=To,P=Po)

"hx_R1234ze_SC denotes specific enthalpy at state point x in the MTC of system"
"Sx_R1234ze_SC denotes specific entropy at state point x in the MTC of system"

"Properties of state point 11" "After MT R1234ze(Z) Condenser"
(Pc2_R1234ze_SC)=**p_sat**(***R1234ze(Z)***,T=Thx2+2.5)
(Pe2_R1234ze_SC)=**p_sat**(***R1234ze(Z)***,T=Te2)
pr_SC=Pc2_R1234ze_SC/Pe2_R1234ze_SC

H16_R1234ze_SC=**enthalpy**(***R1234ze(Z)***,x=0,P=(Pc2_R1234ze_SC))
S16_R1234ze_SC=**entropy**(***R1234ze(Z)***,x=0,P=(Pc2_R1234ze_SC))

"Properties of state point 12" "After MT R1234ze(Z) Throttle"
(h16_R1234ze_SC)=(h13_R1234ze_SC)
(x13_R1234ze_SC)=**quality**(***R1234ze(Z)***,T=Te2,h=(h13_R1234ze_SC))
S13_R1234ze_SC=**entropy**(***R1234ze(Z)***,T=Te2,x=(x13_R1234ze_SC))

"MrHTC_dot_AMMONIA_SC*(h5_R1234ze_SC-h8_R1234ze_SC) =
MrLTC_dot_CO2*(h2-h3) + MrMT_dot_R1234ze_SC*(h10-h11)"

"Properties of state point 9" "After MT R1234ze(Z) Evaporator"
H14_R1234ze_SC=**enthalpy**(***R1234ze(Z)***,T=Te2,x=1)
S14_R1234ze_SC=**entropy**(***R1234ze(Z)***,T=Te2,x=1)
Pe_R1234ze_SC=**pressure**(***R1234ze(Z)***,T=Te2,x=1)

"Properties of state point6" "After MT R1234ze(Z) Compressor"
(s14_R1234ze_SC)=(s15_R1234ze_SC)
H15_R1234ze_SC=**enthalpy**(***R1234ze(Z)***,P=(Pc2_R1234ze_SC),***s***=(s15_R1234ze_SC))
H15_act_R1234ze_SC= (((h15_R1234ze_SC)-(h14_R1234ze_SC))/0.75) +
h14_R1234ze_SC "Issentropic Efficiency =0.75"
ww_SC=0.8014- 0.04842*(pr_SC)

S15_act_R1234ze_SC=**entropy**(R1234ze(Z),***h***=(h15_act_R1234ze_SC),P=Pc2_R1234ze_SC)

qee_R1234ze_SC= (h14_R1234ze_SC)-(h13_R1234ze_SC) "enthalpy drop in evaporator at MT R1234ze(Z)"
wc2_R1234ze_SC=(h15_R1234ze_SC)-(h14_R1234ze_SC)" specific compressor work oh MT R1234ze(Z)"
WC2act_R1234ze_SC= ((h15_act_R1234ze_SC)-(h14_R1234ze_SC)) "actual specific compressor work of MT R1234ze(Z) With 0.75 issentropic efficicncy"

Qe1_dot_R1234ze_SC=PP*25 "refrigeration effect MT"
Qe1_dot_R1234ze_SC=MrMT_dot_R1234ze_SC*qee_R1234ze_SC
Qc2_dot_R1234ze_SC= (MrMT_dot_R1234ze_SC)*((h15_R1234ze_SC)-(h16_R1234ze_SC)) "CASCADE HEAT EXCHANGER 2 "

Qe1_dot_2AMMONIA_SC= Qc2_dot_R1234ze_SC " HEAT EXCHANGER"
Wc2_dot_R1234ze_SC=MrMT_dot_R1234ze_SC*wc2_R1234ze_SC "Issentropic compressor"
Wc2_act_dot_R1234ze_SC= WC2act_R1234ze_SC*MrMT_dot_R1234ze_SC "Issentropic Efficiency =0.7"

Wnet_dot_act_R1234ze_SC_CO2= Wc1_dot_SC+Wc2_act_dot_R1234ze_SC+ Wc2_1_act_dot_AMMONIA_SC+Wc2_2_act_dot_2AMMONIA_SC “net compressor work”
COP_NEW_SYSTEM= (25+(PP*25))/Wnet_dot_act_R1234ze_SC_CO2

EXERGY_efficiency_SC= ((25*(((To)/(Te1))-1))+(PP*25*(((To)/(Te2))-1)))/Wnet_dot_act_R1234ze_SC_CO2

IMP_SC_ACT=((COP_NEW_SYSTEM-COP_MTTC)/COP_MTTC)*100

"EXERGY BALANCE"

Ex1_SC=(h1_SC-ho)-((To)*(s1_SC-so))

Ex4_SC=(h4_SC-ho)-((To)*(s4_SC-so))
Ex2_SC= ((h2_act_SC)-ho) - ((To)*((s2_act_SC)-so))
Ex3_SC=(h3_SC-ho)-((To)*(s3_SC-so))

Ex5_AMMONIA_SC=((h5_AMMONIA_SC)-hoo_AMMONIA_S1)-((To)*(s5_AMMONIA_SC-soo_AMMONIA_S1))
Ex22_AMMONIA_SC= ((h22_act_AMMONIA_SC)-hoo_AMMONIA_S1) -((To)*((s22_act_AMMONIA_SC)-soo_AMMONIA_S1))
Ex32_AMMONIA_SC=(h32_AMMONIA_SC-hoo_AMMONIA_S1)-((To)*(s32_AMMONIA_SC-soo_AMMONIA_S1))
Ex12_AMMONIA_SC=(h12_AMMONIA_SC-hoo_AMMONIA_S1)-((To)*(s12_AMMONIA_SC-soo_AMMONIA_S1))
Ex100_AMMONIA_SC=(h100-hoo_AMMONIA_S1)-((To)*(s100-soo_AMMONIA_S1))
Ex6_AMMONIA_SC= ((h6_act_AMMONIA_SC)-hoo_AMMONIA_S1) -((To)*((s6_act_AMMONIA_SC)-soo_AMMONIA_S1))
Ex9_AMMONIA_SC=(h9_AMMONIA_SC-hoo_AMMONIA_S1)-((To)*(s9_AMMONIA_SC-soo_AMMONIA_S1))
Ex99_AMMONIA_SC=(h99-hoo_AMMONIA_S1)-((To)*(s99-soo_AMMONIA_S1))
Ex77_AMMONIA_SC=(h77-hoo_AMMONIA_S1)-((To)*(s77-soo_AMMONIA_S1))
Ex8_AMMONIA_SC=(h8_AMMONIA_SC-hoo_AMMONIA_S1)-((To)*(s8_AMMONIA_SC-soo_AMMONIA_S1))
Ex7_AMMONIA_SC=(h7_AMMONIA_SC-hoo_AMMONIA_S1)-((To)*(s7_AMMONIA_SC-soo_AMMONIA_S1))

Ex13_R1234ze_SC=((h13_R1234ze_SC)-hoo_R1234ze_SC)-((To)*(s13_R1234ze_SC-soo_R1234ze_SC))
Ex14_R1234ze_SC=(h14_R1234ze_SC-hoo_R1234ze_SC)-((To)*(s14_R1234ze_SC-soo_R1234ze_SC))
Ex15_R1234ze_SC= ((h15_act_R1234ze_SC)-hoo_R1234ze_SC) -((To)*((s15_act_R1234ze_SC)-soo_R1234ze_SC))
Ex16_R1234ze_SC=(h16_R1234ze_SC-hoo_R1234ze_SC)-((To)*(s16_R1234ze_SC-soo_R1234ze_SC))

"EXERGY EQUATIONS"

"Exergy entering - exergy leaving - exergy destroyed = Total exergy change of system"

MrLTC_dot_SC*Ex1_SC - MrLTC_dot_SC*Ex4_SC= + (25*(1-(To/(Te1)))) - EX_DX_LTC_EV1 "Exergy balance of LTC EVAPORATOR"
MrLTC_dot_SC*Ex1_SC+(Wc1_dot_SC) = MrLTC_dot_SC*Ex2_SC+EX_DX_LTC_COMP "Exergy balance of LTC compressor"
MrLTC_dot_SC*Ex3_SC=MrLTC_dot_SC*Ex4_SC + EX_DX_LTC_THROT3 "Exergy balance of LTC throttle"

MrMT_dot_R1234ze_SC*Ex13_R1234ze_SC + ((PP*25)*(1-(To/(Te2)))) = MrMT_dot_R1234ze_SC*Ex14_R1234ze_SC + EX_DX_MTC_EV2 "Exergy balance of MTC Evaporat"
MrMT_dot_R1234ze_SC*Ex14_R1234ze_SC + (Wc2_act_dot_R1234ze_SC) = MrMT_dot_R1234ze_SC*Ex15_R1234ze_SC+EX_DX_MTC_COMP "Exergy balance of MTC compressor"
MrMT_dot_R1234ze_SC*Ex16_R1234ze_SC=MrMT_dot_R1234ze_SC*Ex13_R1234ze_SC + EX_DX_MTC_THROT4 "Exergy balance of MTC throttle"

"exergy balance of HTC condenser" MrHTC_dot_AMMONIA_SC*Ex6_AMMONIA_SC = Ex7_AMMONIA_SC*MrHTC_dot_AMMONIA_SC+ EX_DX_HTC_COND
"Exergy balance of Scroll Compressor stage 1"
MrHTC1_dot_AMMONIA_SC*Ex5_AMMONIA_SC+(Wc2_1_act_dot_AMMONIA_SC)= MrHTC1_dot_AMMONIA_SC*Ex22_AMMONIA_SC + EX_DX_HTC_SC1
"Exergy balance of Scroll Compressor stage 2"
MrHTC_dot_AMMONIA_SC*Ex32_AMMONIA_SC+(Wc2_2_act_dot_2AMMONIA_SC) = MrHTC_dot_AMMONIA_SC*Ex6_AMMONIA_SC + EX_DX_HTC_SC2

"NET EXERGY DESTRUCTION IN SCROLL COMPRESSOR" EX_DX_HTC_SC= EX_DX_HTC_SC2+EX_DX_HTC_SC1

"Exergy balance of HX1" MrLTC_dot_SC*Ex2_SC+ MrHTC1_dot_AMMONIA_SC*Ex8_AMMONIA_SC= MrLTC_dot_SC*Ex3_SC+MrHTC1_dot_AMMONIA_SC*Ex5_AMMONIA_SC+ EX_DX_HX1

"Exergy balance of HX2" MrMT_dot_R1234ze_SC*Ex15_R1234ze_SC+MrHTC2_dot_AMMONIA_SC*(1-(SCX))*Ex9_AMMONIA_SC = MrMT_dot_R1234ze_SC*Ex16_R1234ze_SC+ MrHTC2_dot_AMMONIA_SC*(1-(SCX))*Ex12_AMMONIA_SC+ Ex_DX_HX2

"Exergy balance of HTC throttle 1" MrHTC1_dot_AMMONIA_SC*Ex77_AMMONIA_SC = MrHTC1_dot_AMMONIA_SC*Ex8_AMMONIA_SC + EX_DX_HTC_THROT1

"Exergy balance of HTC throttle 2" MrHTC2_dot_AMMONIA_SC*Ex7_AMMONIA_SC= MrHTC2_dot_AMMONIA_SC*SCX*Ex99_AMMONIA_SC+ MrHTC2_dot_AMMONIA_SC*(1-(SCX))*Ex9_AMMONIA_SC+ EX_DC_HTC_THROT2

"Exeergy balance of Subcooler" MrHTC1_dot_AMMONIA_SC*Ex7_AMMONIA_SC + MrHTC2_dot_AMMONIA_SC*SCX*Ex99_AMMONIA_SC= MrHTC1_dot_AMMONIA_SC*Ex77_AMMONIA_SC+ MrHTC2_dot_AMMONIA_SC*SCX*Ex100_AMMONIA_SC + EX_DC_HTC_SUBCOOLER

Ex_Net_Dest =EX_DX_MTC_EV2+EX_DX_MTC_COMP+EX_DX_MTC_THROT4+EX_DX_HTC_COND+EX_DX_HTC_SC+EX_DX_HX1+Ex_DX_HX2+EX_DX_HTC_THROT1+EX_DC_HTC_THROT2+EX_DC_HTC_SUBCOOLER+ EX_DX_LTC_THROT3+EX_DX_LTC_COMP+EX_DX_LTC_EV1 “NET EXERGY DESTRUCTION RATE”

"TEWI CALCULATIONS"

Tewi_direct = (1*(MrLTC_dot_SC*4)*0.125*15) + (1*(MrLTC_dot_SC*4)*0.3) + (6*(MrMT_dot_R1234ze_SC*4)*0.125*15) + (6*(MrMT_dot_R1234ze_SC*4)*0.3)

Tewi_indirect = (Wnet_dot_act_R1234ze_SC_CO2*150*52)*0.93*15

TEWI= Tewi_direct + Tewi_indirect

OUTPUT

COP_MTTC=1.78

COP_NEW_SYSTEM=1.96

EXERGY_efficiency_SC=0.40

Ex_Net_Dest=15.12 [kW]

Wnet_dot_act_propylene_S2_OCO2=27.25 “Net compressor work in MTTC by Zhu et al. [10] ”

Wnet_dot_act_R1234ze_SC_CO2=25.47 “Net compressor work in Novel System”

IMP_SC_ACT=10.05 “Improvement in percentage for COP”

MrHTC_dot_AMMONIA_S2=0.0537 “Ammonia flow rate in MTTC by Zhu”

MrHTC_dot_AMMONIA_SC=0.0544 “Ammonia flow rate in novel system”

MrLTC_dot_S2=0.098 “Carbon Dioxide flow rate in MTTC by Zhu”

MrLTC_dot_SC=0.106 “Carbon Dioxide flow rate in novel MTTC”

MrMT_dot_propylene_S2=0.064 “Propylene flow rate in MTTC by Zhu”

MrMT_dot_R1234ze_SC=0.128 “R1234ze(Z) flow rate in Novel System”

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