“Comparative analysis of Thermodynamics performance of Cascade Refrigeration system for refrigerant Couple R23/R290 and R23/R600A”

Major project-II

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by

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CERTIFICATE

This is to certify that the thesis entitled **"Comparative analysis of thermodynamic performance of a cascade refrigeration system for refrigerant couple R23/R290 and R23/R600A**" submitted by **Saurabh Anand** (2K17/THE/14), during the session 2017-2019 for the award of M. Tech degree of Delhi Technological University, Delhi is absolutely based upon his work done under my supervision and guidance and that also it has not been directly copied from any source without giving its proper reference.

The assistance and help received during the course of investigation have been fully acknowledged. He is a good student and I wish him good luck in future.

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DECLARATION

I, **Saurabh Anand**, hereby certify that the work which is being presented in this thesis entitled **"Comparative analysis of Thermodynamics performance of Cascade Refrigeration system for refrigerant Couple R23/R290 and R23/R600A"** is submitted, in the partial fulfilment of the requirements for degree of Master of Technology at Delhi Technological University is an authentic record of my own work carried under the supervision of **Dr Rajesh Kumar,** MED DTU and **Dr Sunita Rattan,** AIAS Addl. Director, Amity University Noida, I have not submitted the matter embodied in this seminar for the award of any other degree also it has not been directly copied from any source without giving its proper reference.

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ABSTRACT

This study presents a comparative analysis of thermodynamic performance of cascade refrigeration systems (CRSs) for refrigerant couples R23/R290 and R23/R600A to discover whether R600A is a suitable substitute for R290. The discharge temperature, input power of the compressor, coefficient of performance (COP), exergy loss (X) and exergy efficiency (g) are chosen as the objective functions. The operating parameters considered in this thesis include condensing temperature, evaporating temperature in both high-temperature cycle i.e HTC and lower cycle i.e LTC and Temperature difference in the cascade heat exchanger. Under the same operation condition, the input power of R23/R600A CRS is lower than that of R23/R290 CRS, and COP opt is higher than that of R23/R290 CRS. The theoretical analysis indicates that R23/R600A is a more potential refrigerant couple than R23/R290 in Cascade Refrigeration System (CRS).

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NOMENCLATURE

VCRS	Vapour absorption refrigeration system
CRS	Cascade refrigeration system
COP	Coefficient of performance
LTC	Low Temperature Circuit
HTC	High Temperature Circuit
BP	Boiling Point
Tc	Condenser temperature
$\mathbf{T_e}$	Evaporator temperature
$\mathbf{Q_H}$	Heat rejected in condenser in upper ckt(kw)
$\mathbf{Q_L}$	Heat observed in evaporator in lower ckt(kw)
Qe	Refrigerating capacity(kW)
M	refrigerant mass flow rate
X	Exergy lose
$\boldsymbol{\eta}$	Exergy efficiency
Δ **T**	Temperature difference
$\mathbf{T_{casL}}$	Low temperature cycle condenser temperature
$\boldsymbol{\eta_{isen,L}}$	Compressor isentropic efficiency for LTC
$\boldsymbol{\eta_{isen,H}}$	isentropic efficiency of compressor for HTC
$\dot{w}_l$	Compressor power consumption for LTC
$\dot{w}_h$	Compressor power consumption for HTC
$\mathbf{w_{act}}$	Total work done or Actual work done
$\mathbf{Q_{CC}}$	The rate of heat transfer in the cascade condenser

Q_H	The rate of heat rejection by the air-cooled condenser
COP_L	COP of lower compression system
COP_H	COP of higher compression system
COP_{CC}	COP of the cascade system
η_{Exergy}	Exergetic efficiency or Second law efficiency

SYMBOLS

Letter	Description	Unit
M	Mass of refrigerants	kg/s
h	Enthalpy	kJ/kg
s	Entropy	kJ/kg
P	Pressure	KPa
T	Temperature	K
Q	Heat Energy	kW
η	Efficiency	

CHAPTER 1

INTRODUCTION

1.1 GENERAL

The role of Refrigeration and Air conditioning is very indispensable in our modern-day life for the cooling and heating purposes in various field of requirements. It has very wide range of application starting from stationary to mobile application that provides either comfort conditions or used as suitable conditions for food preservatives. The use of heat pump and refrigeration and air conditioning equipment has increased now a days in such a way that it affects the environment globally. It causes a problem like global warming, degradation of ozone layer etc just because of use of CFC refrigerants. So, this becomes a globally concern to the people about the efficient energy utilization so that environmental problem will be reduced for the better sustainable development. The international organization like Kyoto and Montreal protocol had also raised this issue globally for the less use of CFC refrigerants. The Kyoto protocol had setup aim for the reduction of global warming elements in the year 1997 and subsequently the heat pump industry has consequently been forced to look for substitutes of CFCs and HCFCs. In many applications hydrocarbons have been used but this has been limited by safety considerations Energy saving and climate change is the outcome of system design, which includes the choice of refrigeration cycle, the operating fluid (refrigerant), and the minimization of refrigerant quantity and leakage. It also relates to the installation, the service procedures, and the improvement of energy sector application so that the carbon dioxide emission will get reduced into the surrounding atmosphere. The CFC and HCF refrigerants remain as better choice for some application but we have to find the alternate refrigerants which produces less damages to the ozone layers. A lot of research is going on toward alternate refrigerant which are eco environmentally.

By seeing the shortage of energy, it becomes very indispensable for all of us to conserve it in all possible way and now a days it is becoming a slogan for the present decades to save the energy for the future development. Energy recovery from waste heat and/or to utilize it for useful applications to improve the system efficiency is growing concern in scientific community and hence, is in use for industrial installations now-days. So, this ongoing environmental crisis has compelled researchers and technician around the globe to take into account for the better steps to handle energy saving measures in multiple sectors**.** RAC systems have also a major contributor to the energy consumption.

We cannot avoid the use of energy power and thermal power in today fast life. Generally, this has been increased form a time when industrial revolution takes place all around the world. This Refrigeration and Air conditioning plays a big role in making the industrial revolution successful for the better growth and eliminated the problem of energy requirement throughout the world.

As we have seen now a days that energy conservation and utilization is becoming an important concern that ever one must think for the optimization of thermodynamic processes so that minimum consumption of energy will takes place. There are many parameters which affects the performances of refrigeration circuit So, in order to maximize the design, a proper and full study of exergy analysis i.e 2nd low analysis is required because first law of thermodynamics cannot reveal the irreversibility where destruction of energy will take place in the equipment or in a process. The 2nd low based exergic analysis gives the proper indication of point where engineer or scientist should give proper attention so that the performance of Refrigeration system will become better.

1.2 VAPOUR COMPRESSSION SYSTEM

VCRS is the frequently utilized refrigeration system. It is the most widely used method for air conditioning of building and automobiles. In a vapour cycle the refrigerant changes its phase at least once. The refrigeration happens due to phase change in evaporator. Since the refrigerant used in VC cycle is act as pure substance therefore the temp remains constant at the time of vaporisation process. The basic vapour refrigeration cycle diagram is given below.

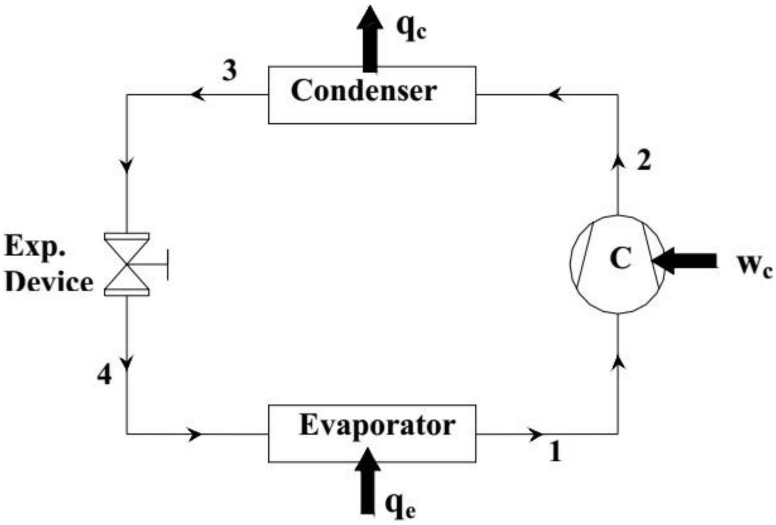


Figure 1.1 Vapour compression system

The basic cycle used in refrigeration is called VC cycle and it consists of four major equipment namely an evaporator, compressor, condenser and the expansion device as shown in above fig 1.1

1– 2: Compression (Reversible Adiabatic Compression) Compressor plays an important part in refrigeration system as it increases the temp and pressure of refrigerants up to the condenser pressure. The refrigerants condition is saturated vapor (Dryness is a unity) before they come into the compressor and after the compression, they become superheated vapor.

2– 3: Condensation (heat rejection at constant pressure). After compressing the refrigerants in compressor, refrigerants now enter the condenser in the superheated vapour form where heat rejection occurs at constant pressure and thus temperature of refrigerants will get reduced and it becomes saturated liquid where dryness fraction becomes zero.

3 – 4: Expansion (constant enthalpy process). After passing from condenser refrigerants now enter to expansion device where isenthalpic process will take place. The temperature and pressure of refrigerants will get reduced up to the desired set limit.

4 – 1: Evaporation (heat absorption at constant pressure). By taking the heat from surrounding space, the low-quality refrigerants evaporate up to the saturated vapour line where dryness fraction becomes unity. Phase change will take place inside the evaporator which increases the rate of heat transfer.

1.2.1 WORKING

As shown in above figure a mechanical input of work is given through a compressor. The compressor compresses the vapour into a high temperature and pressure gas. Practically it's not possible hence it kept close to isentropic process by compressing the vapour in various stages or by intercooling. After this the refrigerant enters into the condenser at elevated pressure and elevated temp**.** In the condenser the compressed gas condenses into liquid. It emits the heat, Q_c into the environment. The liquid ref then reaches the expansion device and the liquid refrigerant is throttled where pressure is lowered to the pressure of the evaporator. Now phase change will take place inside the evaporator which improves the rate of heat transfer. The liquid refrigerant takes the heat, Q_e from the surrounding and converts into vapour. This low temperature and pressure vapour then again enter into the compressor and the cycle continues.

1.2.2 COP OF VCRC

The efficiency of any ref. system is commonly indicate in terms of coefficient of performance i.e COP and it is defined as the ratio of ref effect (heat absorbed by refrigerants when passed through the evaporator) to power input which is needed to compress the refrigerants into the compressor .The COP is highly dependent on the working condition mainly on relative temp between source and sink. The

Cop= refrigeration effect/work input = Q_e / W_c

1.2 The Need of Alternative Refrigerant

For enhancing the efficiency of the System, a refrigerant is used in the power cycle in which phase change will take place from liquid to gas. Chlorofluorocarbon are used as a Refrigerants from the long time but now they are ruled out because of their ozone depletion effect. The common refrigerants like Sulphur dioxide, NH_3, CO_2 and nonhalogenated hydrocarbon like propane are used in different applications. The important refrigerant like R134a used in many domestic Refrigerator and in automobile air conditioning. Since these refrigerants have toxic and harmful effect on people health so we are searching for alternate refrigerants.

In the field of automotive air conditioning system many technologies have emerged to use HFC 134A as a refrigerant because it has the properties of stable non corrosive and non-toxic which has less impact on ozone layer depletion. So alternate refrigerant is necessary to replace the fully halogenated refrigerants which affect most to the ozone layer. So it becomes very indispensable for us to understand the issues that may happened while selecting the refrigerants. Here I have taken only those refrigerants which are mainly used in vapor compression refrigeration cycle. Natural refrigerants like the Ammonia, HCs and CO_2 could substitute for HFCs and HCFCs. The properties of different refrigerants used in this study are given below.

1.3.1 REFRIGERANTS USED AND THEIR PROPERTIES

The refrigerants used in this analysis will be taken as R23, R290 and R600A.The thermophysical properties of these refrigerants are given below:

Refrigerants	Chemical Name	Mol. weight kg/kmol	Normal BP(°C)	Critical temp (K)	Critical Pressure Mpa	ODP	GWP
R-23	CHF_3	70.02	-82.03	25.92	4.83	0	14800
R-290	C_3H_8	44.10	-42.2	96.7	4.25	0	3.3
R-600A	C_4H_{10}	58.12	-11.8	134.7	3.64	0	4

Table 1.1 Refrigerants used and their properties

1.3 CASCADE REFRIGERATION SYSTEM

In order to produce ultralow temperature, we generally prefer CRS over 2 stage refrigeration system. The cascade condensation unit used two refrigerant systems or cycles and stated as cycles A and B. The condenser of cycle B, called as the "high stage", is sometimes fan cooled or in some another cases a water supply will be used to cool but air cooling most commonly used. The Evaporator of cycle B called as the "high stage" is employed to cool the condenser of cycle A called as "low stage". The unit that comprises condenser of cycle A and evaporator of cycle B. is commonly stated as the "cascade condenser" but sometimes it also called as inter stage condenser. So Therefore, it act as a evaporator for upper cycle which is called as high temp circuit and act as condenser for lower circuit which is called as a low temp circuit. The difference of temperature between condenser temp of lower circuit and the evaporator temp of higher circuit is termed as temperature overlap and it is necessary for heat transfer. Cascade refrigeration system uses 2 distinct refrigerants in each cycle because it becomes uneconomical if we want high compression ratio for single stage refrigeration system. We cannot obtain evaporating and condensing temperature if compression ratio is not obtained. The high temperature cascade system utilizes a ref having low boiling temperature such as R-13 or R-13B1 because these ref. have exceptionally high pressure which require less capacity compressor which in turn reduces work input and therefore improves the COP of cascade refrigeration system.

Another set of refrigerants commonly used for liquefaction of gases in three stage cascade system is ammonia, ethylene and methane. The additional advantage of a cascade system over multi stage compression is that the lubricating oil from first compressor cannot wander to the next compressors.

Cascade system incorporates various individual refrigeration systems in which heat exchangers is used to attain low operated temp and sensible condensing pressure. For some implementations like in industries where requirement of low temperature is required then single stage refrigeration cycle becomes inefficient so cascade system are used to achieve high temp difference between the source & heat sink. This **s**ystem can be used for temperature range from -70^{o}C to -100^{o} C.

The p-h graph of CRS is given in Fig 1.2. In this cascaded system both Circuit called as lower circuit and upper circuit that uses different refrigerants and these two are thermally linked to a heat exchanger which act as condenser for lower circuit and evaporator for the upper circuit. The upper circuit operates with refrigerant which has greater BP and critical temp whereas lower circuit works with refrigerant which has lower BP. The properties of various refrigerants have mention in Table 1.1. Fig.1.2 demonstrate that the condenser rejects heat at ambient condition and useful refrigeration effect is produced form lower circuit by absorbing the cooling load from cooling area at the evaporating temp.

1.4.1 NEED OF CASCADE SYSTEM

A CRS consists of two individual cycle i.e lower cycle and upper cycle and both are operated independently to each other. A lower cycle maintains the low evaporating temp and produces the refrigeration impact whereas higher cycle operates at higher evaporating temp. For this first stage vapor compression refrigeration cycle will be practical but it becomes impractical for some industrial application where ultralow temperature with considerably large pressure differences required.In such cases there is solution to perform the refrigeration more than one stages that operate in series and these refrigeration cycle known as CRS .Therefore, cascade refrigeration system are used where we want temperature range of -70°C to -100°C.But for evaporating temperature below -70°C the application of three stage VRS system is limited because it is very too difficult to achieve refrigerant requirement freezing temperature.

Also, Kyoto protocol limit the use of CFCs and HCFs provides the alternate substitution for this refrigerant because these CFCs and HCFs have very bad effect on stratospheric ozone layer that protect the earth from Ultra Violet waves.

For many industrial and medical applications, very low temperatures are required. Thus, the temperatures of the order of -80^{o}C are required to freeze and store blood and for precipitation hardening of special alloy steels, temperatures as low as -90^{o}C are required.

To obtain such low temperature by conventional system as mentioned earlier becomes difficult because of extremely low evaporator Pressures. Thus, even with a High-pressure refrigerant like R-22, the evaporator pressure is 0.105 bar at -80^{o}C evaporator temperature. For R-12, the pressure is still lower. Operation at such low pressure becomes difficult both because of sealing problem as well as high displacement volume. Only R-13 can be safely used at such lower temperature as its saturation temperature pressure is 1.12bar at -80^{o}C.However its critical temperature (28.8^{o}C) is very low and direct or stage compression up to condensing pressures is ruled out. Therefore, it is no alternative except cascading which consists of using two different vapour compression plants operating with different refrigerants and coupled together so that the condensing of low temperature stage vapour is achieved by evaporation of high temperature stage liquid.

The maximum low temperature achieved by compound compression system is also governed by the temp of freezing point of the refrigerant used. This difficulty can be solved by using cascade system in which two or more refrigerants can be used separately, and combining the operating cycle of one with other.

1.4.2 Advantages of cascade refrigeration system

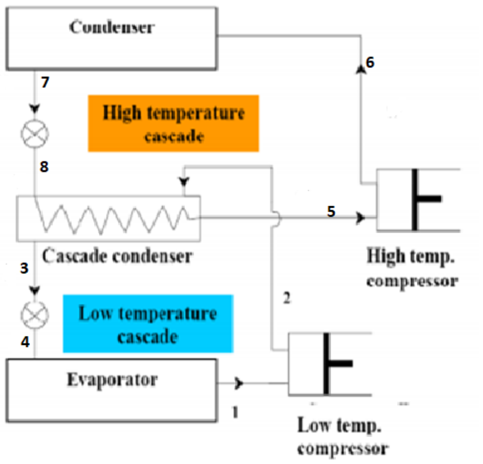
1. Energy is saved as a result of the system permits use of refrigerants that have appropriate temperature characteristics for every refrigerant which is used either for the higher temp cycle and also for the lower temp cycle.

2. It permits stable ultra-low-temperature operation.

3. It conjointly minimize the lubricant issues as the lubricant involved with each refrigerant has to take a temp less than 60°C, but in case of dual stage system, the lubricant operates over a range of temp of nearby 105°C.

4. The Performances of cascaded refrigeration system will be enhanced by decreasing temp difference for heat transfer within the condenser evaporator and cascade condenser, compared to bigger compressor.

5. By using a cascaded refrigeration system we can reduce the power consumption to about 9.5%.



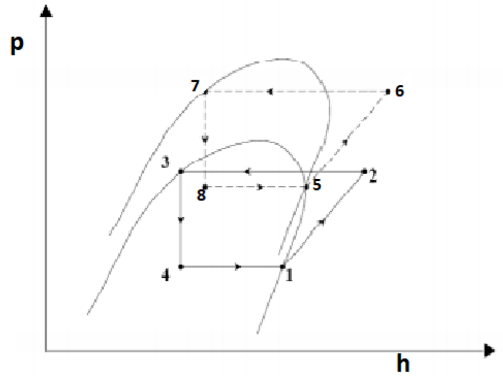


Fig. 1.2 Two stage cascade refrigeration system,[20]   Fig. 1.3: P-H chart

1.4.3 P-h Diagram of Ideal Refrigeration Cycle:

Process 1-Reversible Adiabatic compression in low temp Compressor.

Process 2-3 Heat Rejection at constant pressure in low temperature circuit

Process 3-4 Expansion Throttling Process**,** Isenthalpic Process

Process 4-1 Constant heat Addition Process

Process 5-6 Reversible adiabatic compression in high temperature compressor

Process 6-7 Heat Rejection at constant pressure in high temperature circuit

Process 7-8 Expansion Throttling Process, Isenthalpic Process

Process 8-1 Heat addition at constant pressure

1.5 ENERGY SITUATION

In Asian regions, an economic growth and increase in population in the last few years has boosted the use of air-conditioning for space cooling and increases the use of ref system. This has led to the increasing energy demand and also the demand of economical energy use for air -conditioning and ref systems specifically in hot season. In Asian nation the energy consumption patterns within the use of refrigeration and air conditioning are quick growing as per table 1.2.

Consumption of power by different Cooling devices (2006-2031)

Heating Cooling	Year	2006	2011	2016	2021	2026	2031
Fans	Operating GWh /Yr	22,724	34,100	49,310	67,521	84,441	100,185
Air -cooler	Operating GWh /Yr	8,091	13,373	21,186	31,828	43,626	55,975
Air - conditioning	Operating GWh /Yr	2,298	5,084	10,783	20,966	34,675	49,913
Total energy consumption	Total GWh/yr	2,308	5,099	10,806	21,005	34,737	50,000
% Increase	from previous yr	NIL	46	47	51	60	96

Table 1.2 Projected Consumption of power by different cooling devices

The following are key factors for contributing to this growing worldwide power consumption. Important variables are population growth, greater demand for space cooling, better comfort levels among peoples etc.

1.6 EXERGY

Exergy or highest available energy is the maximised theoretical work that can be acquired from a specified system as it goes into equilibrium with the ambient conditions (reaches the dead state).The dead state temp and pressure will be equal to room temp conditions i.e 25°C and 1.0135 kpa Thus, exergy is a property which quantifies the availability of a system, that is generally degraded due to the irreversibility associated with thermodynamic processes. However, the exergy (maximum theoretical work) not only will be degraded by irreversibility but can also be moved to some other systems. The exergy transfer from one system to its environment which term as exergy loss can be calculated by an equation given by Gouy-Stodola. By reducing the destruction of available energy, we can improve the utilization of energy and thus performance of system improves.

$$\frac{dE_{vc}}{dt} = \sum_j \left(1 - \frac{T_0}{T_j}\right)\dot{Q}_j - \left(\dot{W}_{vc} - p_0 \frac{dV_{vc}}{dt}\right) + \sum_e \dot{m}_e e_{fe} - \sum_s \dot{m}_s e_{fs} - \dot{E}_d$$

CHAPTER 2

LITERATURE REVIEW

2.1 REVIEW OF PAST STUDIES

Various scientists have made continuous attempts in relation to different types of cascade refrigeration systems and are wondering how they can enhance their efficiency and make them cost-effective. Some of the past studies are:

Gupta and Prasad [1983] He conducted a comparative assessment on distinct refrigerants to determine the best pair of refrigerants that provide optimum output by bringing and researching the impact of subcooling, overheating and cascading overlap temp on the performance of the cascade refrigeration system.[1]

Ratts and Brow[2000] performed a second low evaluation of an optimal cascaded refrigeration system for vapor compression using a single ref named R134A. He performed this to determine the optimum intermediate temp based on the method of minimizing entropy gen. [19]

Molenaar [2002] conducted an investigation into various alternative refrigerant couples with low ozone depletion potential i.e. low ODP at lower temp and assessed the efficiency and operating features of distinct refrigerants.[2]

A. Kilicarslan [2004] Different pair of refrigerants were analysed by Kilicareslan for energy and irreversibility in a cascade refrigeration system He analysed various performance parameter on COP, irreversibility, effectiveness, exergy efficiency, total compressor work input, exergy losses and so on. [3]

Bhattacharyya [2005] did a survey on the efficiency of cascaded refrigeration heat pump system that was based on the design which has incorporated both internal as well as external irreversibility. He also studied the optimal allocation of heat exchanger inventories in cascaded refrigeration circuit so that minimum cost and maximum performance of system will be obtained .He finds that the overall performance of cascaded refrigeration system will remain constant with the effectiveness of the low temperature cycle i.e propane cycle but performance increases when the efficiency of high temperature cycle is increased It is clear that COP of this CRS increases when refrigerated space temp increases and decreases when the temperature of outlet increases.[4]

Lee [2006] optimized condensing temperature for a two-stage cascaded carbon-dioxide and ammonia refrigeration system to enhance COP and to minimize exergy loss to a certain value. It was also found that ideal condensing temperature rises when the temperature of evaporation and condensation rises. During the inquiry he did not take the impacts of overheating and sub-cooling. The exergy efficiency was also not carried out during his research.[5]

Tung-wei chen [2006] analysed a CRS that uses carbon dioxide & ammonia as a refrigerant in lower cycle and upper cycle and the optimal condensing temp of the condenser cascade was determined, providing different design parameters like maximizing COP and minimizing system energy destruction. [6]

Hosoz and Ertunc [2006] used artificial neutral system to asses multiple performance variables like the compressor power, evaporating temp, and COP of a dual-stage cascaded ref system using R134A in both the circuit i.e lower circuit and upper circuit [18]

Di Nicola et al [2007] analysed a thermodynamic assessment of a cascaded refrigeration scheme using a R744 HC mix. He concluded that the performance coefficient of a cascaded scheme with R744 mix rises to an appropriate value but better performance shown by pure HC refrigerants but in a low stage cascade system Still considering the mixture is preferable because it has a reduced environmental impact.

Getu and Bansa[2008] analysed a thermodynamic energy and exergy assessment of a CRS by taking greater tempe side refrigerant R717 and low temperature side refrigerant R744. To achieve maximum COP, he carried out an assessment on different parameters such as subcooling, superheating, optimum evaporation temperature and mass flow rate. It was found that highest mass flow rate compare to R717 was required in the elevated temp cycle of the CRS as compare to R404. He also found that the highest COP was achieved for ethanol accompanied by R717 and the lowest for R404a under the similar circumstances. [8]

Bingming eta [2009] performed a analysis on CRS with carbon dioxide and ammonia and reported an experimental information. He also contrasted the outcome of the CO_2/NH_3 cascade system with the single stage ammonia system with or without economizer, concluding that the cascade system's COP is the best of all system.[9]

In **Kilicarslan and Hoso [2010]** analysed the CRS with the different refrigerants couple like R404a–R23, R717–R23, etc to know irreversibility associated with the CRS and also analysed the COP obtained from different refrigerant couple. He contrasted the system with the significant parameter such as condenser temp, evaporator temp, the temp difference between the saturation conditions of the lower and higher temp circuit in the heat exchanger and lastly concluded that R23/R717 is the best refrigerant pair for the vapor compression CRS including all that has taken during analysis.[10]

Dopazo [2011] investigated by analysing the CO_2/NH_3 CRS and concluded that two common **dual** stage NH3 ref systems are preferable at or below the evaporation temperature of-40 ° C. However, due to its high triple-point temperature, the lowest CO_2 refrigeration **temp** is limited above -55 ° C. [17]

Messineo [2012] The thermodynamic assessment of the CRS working at TE=-35°Cand TC=35°Cis revealed. Six distinct coolant types (R717, R290 and R600), three synthesized coolant (R404A, R410 A and R134a) were analysed in specific in the HTC. Carbon dioxide was solely regarded in the low-temperature cycle. Finally, the findings indicate that a CRS with natural refrigerants is an alternative option to schemes which use synthetic coolants in terms of energy, safety and the environment.

Wang [2013] has carried out numerical investigations to examine the ability of the auto-cascade refrigeration system, which is operated by binary refrigerants, some of them containing R23.[16]

Parekh and Tailor [2014] A energy and exergy analytical of a CRS, analysed by him with 3 distinct cooling pairs like R404A-R23, R13-R12, R290-R23 and thermodynamic assessment, has shown that the COP of the R290-R23 refrigerant couple is the highest among three ref couple that he has taken during his assessment. [11]

Dubey [**2014]** studied a CO2 and propylene (R744-R1270) cascade system while others authors modelled a CO2 and NH3 system. These studies have sought to determine the conditions of optimum efficiency of a cascade cooling system according to variables like evaporation and condensation temp. Overall, the results of these studies show efficiency values according to a certain combination of those parameters and finally he made a conclusion that (i) the COP of the CRS is enhanced by 9% using the economizer.[ii] It is hard to obtain the required evaporator temp TE when the temperature of the intermediate temperature is greater. The analysis findings indicate that the intermediate temperature must be lower to achieve optimal system efficiency. This is achieved by reducing the temp of the gas cooler outlet. (iii) The total rise in the system COP because of the more productivity of the internal heat exchanger is marginal Therefore it is not suggested to use an internal heat exchanger [12]

Manoj Dixit et al (2016) studied a CRS and find out the best refrigerants and appropriate operation parameters. He made a conclusion after analysis that compressor, cascade condenser and ref throttle valve are the main cause of exergy degradation and irreversibility. He has taken a mathematical and computational model of the ref system for the analysis of CRS.[13]

Umesh C. Rajmane [2017] studied a CRS by using refrigerant R404A in upper cycle and R23 in lower cycle. In his study, the operational parameters included evaporation, condensation, overheating and subcooling by carrying refrigerant (R23) at reduced temperature and (R404a) elevated temperature cycles [14]

R.S. Mishra [2017] investigated the energy and exergy assessments of three CRS using natural refrigerants which are eco-friendly and used for low temp applications**.** The outcome of different variables on the of first low thermal performance COP and also the effectiveness of the cascade system under second law will be analysed. He also optimized an exergy destruction ratio by using entropy generation principle. His paper demonstrates that R134a is significantly improved on the interim temperature circuit with R134a in the reduced temperature circuit in comparison to R134yf. it was ascertained that the most effective refrigerants combination is R1234ze-R134a-R404a which provides higher thermal performance than using R1234yf-R134a-R404a [15]

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2.2 PRESENT WORK

In this present study the comparison of thermodynamic assessments of CRS has been performed with the 2 refrigerant pairs particularly R23-R600A and R23-R290. In these systems, performance of 2 stage cascade compression system using above combine pair are studied and therefore the impact of condenser temperature & the evaporator temperature, difference in cascaded condenser and low temp cycle(condenser temp) have been performed to know the performance on COP and also total compressor work, exergy efficiency and total exergy loss has also been performed. thermodynamic and mathematical analysis have carried out by establishing computational model in Engineering Equation Solver.

2.3 OBJECTIVES OF STUDY

The primary objective of this work is to compare two different cascade system using two refrigerant pairs R23-R600A and R23-R290. The objectives of the present work can be listed as follows:

1. To study energy and exergy analysis of (CRSs) for ref couples R23-R600A and R23-R290.
2. To find out whether refrigerant R600A become an appropriate replacement for refrigerant R290 or not
3. To study the impact of operating variables on performance of CRSs for R23-R600A and R23-R290 refrigerant couple.

CHAPTER 3

SYSTEM MODELING

To aid in analysis of engineering problem it is necessary to realize the Physical model in a mathematical model. To do this, we first write state point equations of thermodynamic properties and then develop a polynomial for thermodynamic properties with the help of software or, directly taken from the reference.

Therefore, this chapter involves the description of physical model, mass, and the energy balance, assumptions, state point equations and their thermodynamic properties. To show the superiority of cascade system for low temperature application or to justify the utility of CRS for low temp cooling (below -40° c), it becomes necessary to analyse them separately. Thus, this chapter deals with the mathematical modelling of two sections.

3.1 CASCADE SYSTEM (TWO STAGE COMPRESSION SYSTEM USING R23-R290 REFRIGERANT)

This CRS consists of 2 single stage circuit separately which are connected by a cascaded heat exchanger. For lower circuit it used as a condenser where hear rejection takes place at constant pressure whereas for higher circuit it becomes evaporator. Refrigerant R23 will be used in the lower circuit due to its low boiling point and Refrigerants R293 will be used in upper stage to condense the R23 of lower temperature stage.

The ref R23 that flows inside the evaporator absorb the heat from the cooling area at Tf temp and pass through the lower-circuit compressor, which raise the temperature and pressure of ref and then condense into the cascade heat exchanger at the Tcond condensation temp.

In the upper circuit condenser in which refrigerant R290 flows rejected heat at condensing temp of QcondR290 to the condensing medium at To temp. The refrigerants R290 expanded in the expansion device of upper circuit at the evaporating temp of

Tevap R290 in cascaded heat exchanger and then compressed in upper circuit compressor and finally released into the condenser.

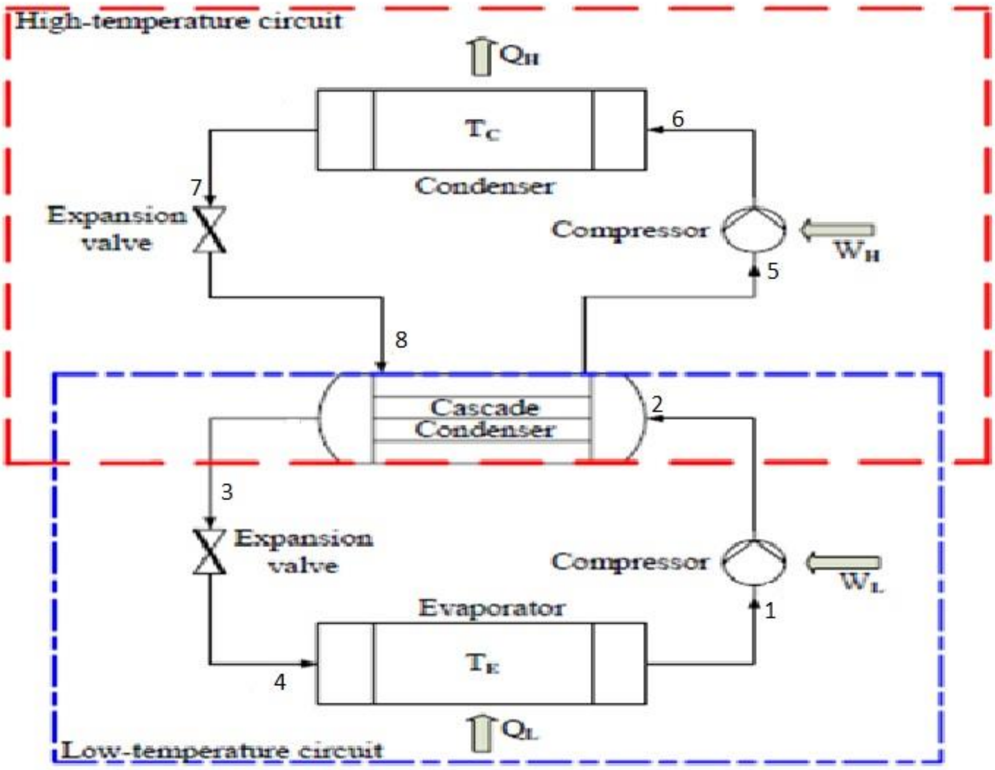

Fig:3.1 Schematic diagram of the R23-R290 CRS [21]

3.1.1 ASSUMPTIONS

1. All components used in CRS shall be presumed to function in a steady condition. The P.E and K.E changes are considered insignificant.
2. The low circuit compressor is assumed to be work as reversible adiabatic condition.
3. All throttling devices are isenthalpic.
4. Refrigerants which are coming out of cascaded heat exchanger, evaporator or the condenser are assumed to be saturated.
5. Pressure and thermal losses and gains in pipe or system components are considered as negligible.
6. T_0=25 ° C and P_0=1 atm are the dead state.

3.1.2 MASS BALANCE AND ENERGY BALANCE

The required equations for each element of each system are provided in Table 3.1. The following equation calculated the COP of CRS

$$COP = Q_L/W_{Total}$$

Table 3.1: Mass and Energy Balance for R-23/R-290 CRS

Components	Mass	Energy
R23-Compressor	$m_2=m_1$	$W_{comp}R23=m_1.(h_{2s}-h_1)$
R290-Compressor	$m_6=m_5$	$W_{comp}R90=m_5.(h_{6s}-h_5)$
R23- Exp. Device	$m_4=m_3$	$h_4=h_3$
R290- Exp. Device	$m_8=m_7$	$h_8=h_7$
Evaporator (R23)	$m_1=m_4$	$Q_{evap}R23=m_1(h_1-h_4)$
Condenser (R290)	$m_7=m_6$	Q_{cond} R290$= m_5(h_7-h_6)$
Cascade heat exchanger	$m_3=m_2,m_5=m_8$	$m_1.(h_3-h_2)=m_5.(h_5-h_8)$

3.1.3 STATE POINT EQUATIONS

1. H_1= HV (TE, PE)
2. H_{2S}= HV (TMC, PMC)
3. H_1= HL (TMC, PMC)
4. H_3= H_4 (isentropic expansion)
5. H_5=HG (TME, PME)
6. H_8=HF (TC, PC)
7. S_{6S}=SGL (TC, PC)
8. PC=PR (TC)
9. CPE=CPV (TME)
10. S_1=S_2=SG (TE, PE)
11. S_{2S}=SG (TMC, PMC)
12. PE=PP (TE)
13. PMC=PP (TMC)
14. H_{6S}=HG (TC, PC)
15. S_6=SGL (TME, PME)
16. PME=PR (TME)
17. CPC=CPV (TC)
18. T_2= (EXP ((S_2S_{2S})/CPV))*(TMC+273.16) - 273.16

3.1.4 STATE POINT PROPERTIES

Table 3.2 Temperature pressure enthalpy and mass of the refrigerants at different state points

State point	Temp(t) °C	Pressure(p) Kpa	Enthalpy(h) KJ/kg	Entropy(s) KJ/kgK
1	-80	113.6	324.6	1.753
2	-25.17	2178	444.6	1.855
3	-5	2178	191.4	0.9695
4	-80	113.6	191.4	1.064
5	-2	446.2	572.4	2.374
6	25	952.2	622.9	2.374
7	25	952.2	265.5	1.226
8	-2	446.2	265.5	1.242

3.1 CASCADE SYSTEM (TWO STAGE COMPRESSION SYSTEM USING R-23/R-600A REFRIGERANT)

The CRS consists of 2 single stage circuit separately which are connected by a cascaded heat exchanger. For lower circuit it is used as a condenser where hear rejection takes place at constant pressure whereas for higher circuit it becomes evaporator. Refrigerant R23 will be used in the lower circuit due to its low boiling point and Refrigerants R600a will be used in upper stage to condense the R23 of lower temperature stage.

The refrigerants R23, which flow inside the evaporator, absorb heat from the cooling region at Tf temp and then pass through the lower circuit compressor, which raises the tempe and pressure of the refrigerants and then condenses it in the cascaded heat exchanger at Tcond condensation temp and then sends it to where the evaporator is applied.

In the upper circuit condenser in which refrigerant R600A flows rejected heat at condensing temp of QcondR600A to the condensing medium at To temp. The refrigerants R600A expanded in the expansion device of upper circuit at the evaporating temp of Tevap R600A in cascaded heat exchanger and then compressed in upper circuit compressor and finally released into the condenser.

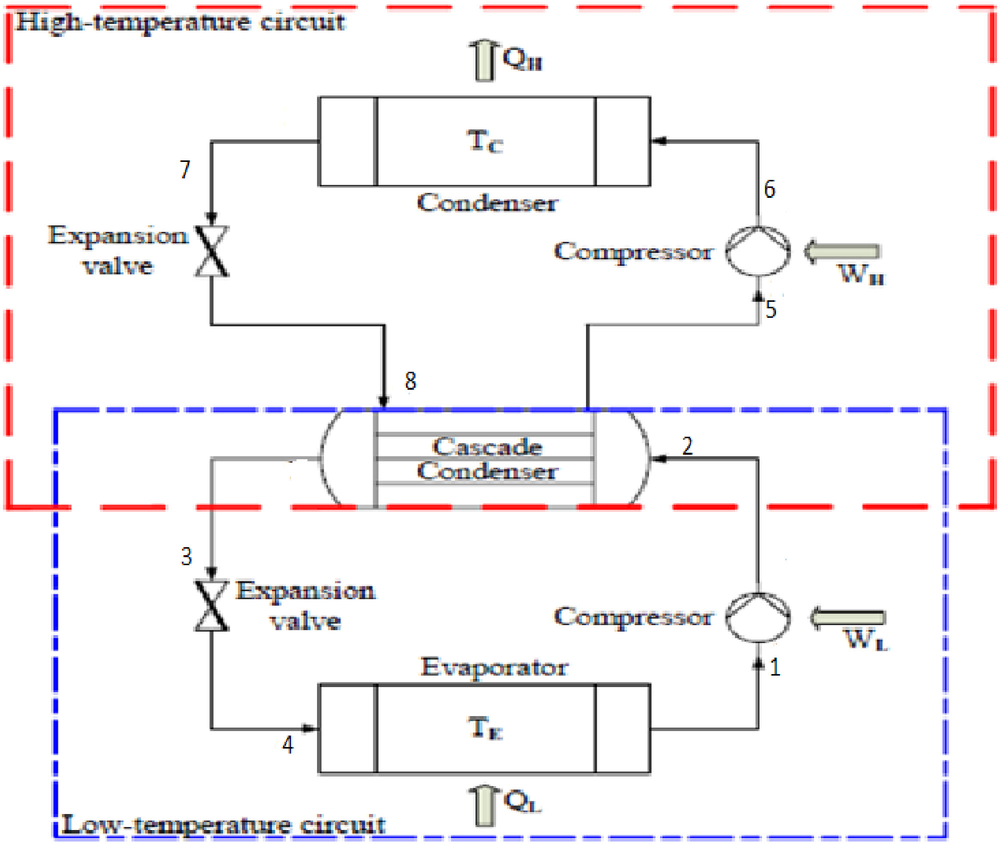

Fig. 3.2: Schematic diagram of the R23/R600A CRS [21]

3.1.1 ASSUMPTIONS

1. All components used in CRS shall be presumed to function in a steady condition. The P.E and K.E changes are considered insignificant
2. The low circuit compressor is assumed to be work as reversible adiabatic condition.
3. All throttling devices are isenthalpic.
4. Refrigerants which are coming out of cascaded heat exchanger, evaporator or the condenser are assumed to be saturated.
5. Pressure and thermal losses and gains in pipe or system components are considered as negligible.
6. T_0=25 ° C and P_0=1 atm are the dead state.

3.2.2 MASS BALANCE AND ENERGY BALANCE

The required equations for each element of each system are provided in Table 3.1. The following equation calculated the COP of CRS

$$COP= Q_L/W_{Total}$$

Table 3.3: Mass and Energy Balance for R-23/R-600A CRS

Components	Mass	Energy
R23-Compressor	$m_2=m_1$	$W_{comp}R23=m_1.(h_{2s}-h_1)$
R600A-Compressor	$m_6=m_5$	$W_{comp}R600A=m_5.(h_{6s}-h_5)$
R23- Exp. Device	$m_4=m_3$	$h_4=h_3$
R600A- Exp. Device	$m_8=m_7$	$h_8=h_7$
Evaporator (R23)	$m_1=m_4$	$Q_{evap}R23=m_1 (h_1-h_4)$
Condenser (R600A)	$m_7=m_6$	$Q_{cond}R600A = m_5. (h_7-h_6)$
Cascade heat exchanger	$m_3=m_2,m_5=m_8$	$m_1.(h_3-h_2)=m_5.(h_5-h_8)$

3.2.3 STATE POINT EQUATIONS

1. H_1= HV (TE, PE)
2. H_{2S}= HV (TMC, PMC)
3. H_1= HL (TMC, PMC)
4. H_3= H_4 (isentropic expansion)
5. H_5=HG (TME, PME)
6. H_8=HF (TC, PC)
7. S_{6S}=SGL (TC, PC)
8. PC=PR (TC)
9. CPE=CPV (TME)
10. S_1=S_2=SG (TE, PE)
11. S_{2S}=SG (TMC, PMC)
12. PE=PP (TE)
13. PMC=PP (TMC)
14. H_{6S}=HG (TC, PC)
15. S_6=SGL (TME, PME)
16. PME=PR (TME)
17. CPC=CPV (TC)
18. T_2= (EXP ((S_2S_{2S})/CPV))*(TMC+273.16)

3.2.4 STATE POINT PROPERTIES

Table 3.4 Temperature pressure enthalpy and mass of the refrigerants at different state points

State point	Temp(t) °C	Pressure(p) Kpa	Enthalpy(h) KJ/kg	Entropy(s) KJ/kgK
1	--80	113.6	324.6	1.753
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3	-5	2178	191.4	0.9695
4	-80	113.6	191.4	1.064
5	-2	446.2	572.4	2.374
6	25	952.2	622.9	2.374
7	25	952.2	265.5	1.226
8	-2	446.2	265.5	1.242

CHAPTER 4

RESEARCH METHODOLOGY

4.1 THERMODYNAMIC ANALYSIS

Thermodynamic assessment of the CRS R23-R290 and R23-R600A are conducted according to the following assumptions:

1.The Compression process which take place inside both the cycle called as LTC and HTC are consider as work at reversible adiabatic condition with the with an isentropic efficiency of 0.7.

2. It is presumed that the expansion mechanism in throttling is isenthalpic

3.Negligible heat interaction in the cascaded heat exchanger with the surrounding

4.The changes in P.E and K.E in components of CRS are consider as negligible.

5.The system is at steady state condition. All processes that take place inside various components of CRS are consider as steady flow processes.

6. In the cascade heat exchanger the temperature difference is -3°C.

The thermo-physical properties of R23-R290 and R23-R600A described in this work were calculating using a software tool known as Engineering equation solver (EES) This EES software has a significant feature in the high accuracy of thermodynamics and transport property. The cycle is modelled by applying mass balance and energy balance equation for every individual process of the cycle. The equations for the various parts of the CRS are given in the previous chapter.

4.2 DESIGN PARAMETER FOR TWO STAGE CASCADE SYSTEM FOR R23-R290 and R23-R600A

The calculation for the two, stage cascade system is initiated by assigning certain specified values for the evaporator and condenser temp. Subsequently, the saturation pressure, the liquid and vapour enthalpies, the entropies, specific heats are computed from EES. The evaporator is supposed to carry heat from the refrigerated area. For this, the evaporator temperature of low temp circuit is initiated by assuming T_E=-80°c and then varied as $T_E=T_E+5$ with 5^0C interval, the T_C= 25°c and then varied as $T_C=T_C+5$ with 5°C interval. Low temperature cycle condenser temperature (T_{casL}) = -5°c. Then the optimal condensing temp has been designed under different evaporating temp corresponding to the minimum energy required. The coolant mass flow rate is chosen as 1Kg/s via condenser (m1) and condenser (m2). Thus, the other parameter, W_{Total}, COP, η_{Exergy} and $X_{Total\ loss}$ are evaluated for each set of operating temperature.

4.3 MATHEMATICAL FORMULATION

Balance of Mass

$$\sum \dot{m}_{in} = \sum \dot{m}_{out}$$

Balance of Energy

$$\dot{Q} - \dot{W} = \sum(\dot{m}h)_{in} - \sum(\dot{m}h)_{out}$$

Exergy balance:

$$\dot{X}_{Lost} = \sum_{out}\left(1 - \frac{T_o}{T_i}\right)\dot{Q}_i - \dot{w} + \sum_{in}\dot{m}\varphi - \sum_{out}\dot{m}\,\varphi$$

Capacity of the evaporator:

$$\dot{Q}_E = \dot{m}_l\ (h_1 - h_4)$$

isentropic efficiency of compressor for low-temperature circuit:

$$\eta_{isen,L} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

isentropic efficiency of compressor for high-temperature circuit:

$$\eta_{isen,H} = \frac{h_{6s} - h_5}{h_6 - h_5}$$

Compressor power consumption at low temp cicuit is specified as:

$$\dot{w}_l = \dot{m}_l\ (h_2 - h_1)$$

power consumption of compressor for high-temp circuit is given as:

$$\dot{w}_h = \dot{m}_h\ (h_6 - h_5)$$

Total work done or Actual work done:

$$w_{act} = \dot{w}_h + \dot{w}_l$$

The heat transfer rate is calculated from the cascaded condenser by equation:

$$\dot{Q}_{CC} = \dot{m}_l\ (h_2 - h_3) = \dot{m}_h\ (h_5 - h_6)$$

The air-cooled condenser rejects heat as follows:

$$\dot{Q}_H = \dot{m}_h\ (h_6 - h_7)$$

COP compression system is given by

$$COP_L = \frac{QE_1}{WC_1} = \frac{(h_1 - h_4)}{(h_2 - h_1)}$$

$$COP_H = \frac{QE_2}{WC_2} = \frac{(h_5 - h_8)}{(h_6 - h_5)}$$

COP of the cascade system is given as

$$COP_{CAS} = \frac{QE}{WC_1 + WC_2}$$

$$COP_{CAS} = \frac{COP_L \times COP_H}{1 + COP_L + COP_H}$$

Exergy efficiency or Second law efficiency will be given by:

$$\eta_{Exergy} = \frac{\dot{w_h} + w_L - \dot{X}_{Total}}{\dot{w_h} + w_L}$$

Exergy destruction in the system components:

$\dot{X}_{LT\ Compressor} = \dot{m}_L T_o (s_2 - s_1)$

$\dot{X}_{HT\ Compressor} = \dot{m}_H T_o (s_6 - s_5)$

$\dot{X}_{Condenser} = \dot{m}_H T_o (s_7 - s_6) + \dot{m}_H T_o (h_6 - h_7)/T_C$

$\dot{X}_{LT\ Throttling\ device} = \dot{m}_L T_o (s_4 - s_3)$

$\dot{X}_{HT\ Throttling\ device} = \dot{m}_H T_o (s_8 - s_7)$

$\dot{X}_{Cascade\ condenser} = T_o [\dot{m}_L (s_3 - s_2) + \dot{m}_H (s_5 - s_8)]$

$\dot{X}_{Evaporator} = T_o [\dot{m}_L (s_1 - s_4) - \dot{m}_L (h_1 - h_4)/T_E]$

Total exergy destruction:

$\dot{X}_{Total} = \dot{X}_{LT\ Compressor} + \dot{X}_{HT\ Compressor} + \dot{X}_{Condenser} + \dot{X}_{LT\ Throttling\ device}$
$+$

$\dot{X}_{LT\ Throttling\ device} + \dot{X}_{Cascade\ condenser} + \dot{X}_{Evaporator}$

4.4 COMPONENT ANALYSIS OF VCRS OF CASCADE CYCLE

In this analysis the total mass balance and energy balance in the vapour compression system components is done. Figure 1.2 shows P-h diagram of the VCRS. The p-h diagram of the system shows the different state of the refrigerant used in this cycle.

4.4.1 COMPRESSOR

Compressor compresses a low-pressure ref from the evaporator into a high-pressure ref before entering the condenser. After the compression the refrigerant temperature and pressure is increased. To minimize the compressor work it is compressed very close to isentropic process. This can be done by providing intercooler between the stages of compressor. Here we are using a single stage compressor. Reciprocating type of compressor is used Isentropic and electric efficiencies are 0.70 and 0.90 respectively [1].

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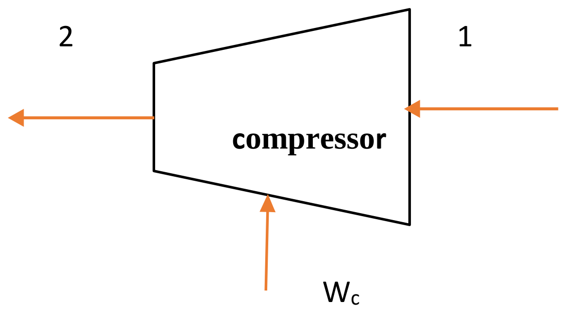

Figure: 4.1 Schematic diagram of Compressor

Mass Balance equation

Dry compression is isenthalpic, so considering the isentropic efficiency, enthalpy at the end of the compression is given by formula

$M_1 = M_2 = M$

Energy Balance

The compression is isenthalpic, so considering the isentropic efficiency, enthalpy at the end of the compression is given by formula

$h2 = h2 + (h2s\text{-}h1)/\acute{\eta}_{is}$

Total done by the compressor

$W = M_1(h_2\text{-}h_1)/\ \acute{\eta}_e$

4.4.2 CONDENSER

After the compression the refrigerants gets cooled in the condenser and reject its heat to the sink; in this case second evaporator attached to the condenser and gets converted into liquid refrigerants. Though the temperature is decreased but it is still at high temperature.

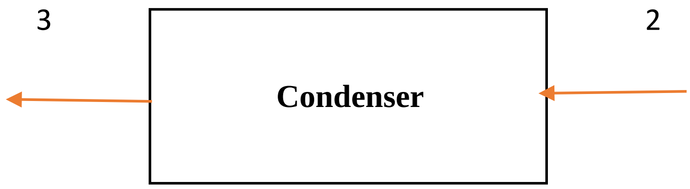

Figure 4.2 Schematic diagram of condenser

Total mass balance

$M_2 = M_3 = M$

Total energy balance

Q condenser = M3 (h2-h3)

4.4.3 REFRIGERANT EXPANSION DEVICE

The condenser's high pressure refrigerant is decreased to low pressure refrigerant by passing through a refrigerant expansion valve. The energy flow through this component remains constant since it is enthalpic.

Figure 4.3 REV 1 Schematic diagram

Mass Balance
$M_3 = M_4 = M$

Total energy balance

$h_3 = h_4$

4.4.4 EVAPORATOR

The evaporator functions as a source because it accepts heat from the surroundings and cools the coolants in liquid form, removes the heat from the cooling region and turns into a vapor form. The vapor of the ref is at low pressure before entering the compressor.

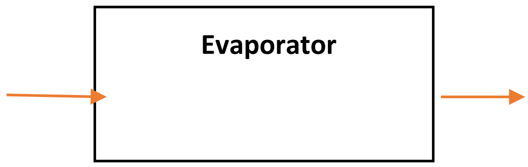

Fig 4.4 Schematic diag of evaporator

Mass Blalnce

$M_4 = M_1 = M$

Energy Balance

$Q_e = m_1 (h_1 - h_4)$

CHAPTER 5

RESULTS AND DISCUSSION

Thermodynamic model established in the Engineering Equation Solver Software and outcomes of the assessment were provided in the following parts in the current paper.

PERFORMANCE OF R23- R290 CASCADE SYSTEM

5.1 IMPACT OF EVAPORATOR TEMPERATURE

Table 5.1 depicts the simulation results obtained when the evaporator temp varied from -80°C to -35°C in the interval of 5°C keeping other parameters constant. Variation of COP, total compressor work, exergy efficiency and exergy loss of cascade system with change in temp of low temp evaporator of cascade system has been shown in the Fig. 5.1-5.4.

Table 5.1: Obtained result with varying evaporator temp

1..10	T_1 [C]	COP	COP_{HTC}	COP_{LTC}	η_{exergy}	W_{total} [KJ/s]	X_{total} [KW]
Run 1	-80	0.727	4.007	1.11	0.9168	183.2	15.25
Run 2	-75	0.7922	3.836	1.259	0.9095	170.5	15.43
Run 3	-70	0.862	3.679	1.432	0.9016	158.8	15.62
Run 4	-65	0.9369	3.533	1.636	0.8931	147.9	15.82
Run 5	-60	1.017	3.398	1.879	0.8838	137.8	16.01
Run 6	-55	1.103	3.272	2.173	0.8737	128.4	16.22
Run 7	-50	1.195	3.155	2.535	0.8627	119.6	16.42
Run 8	-45	1.294	3.044	2.991	0.8507	111.4	16.63
Run 9	-40	1.399	2.939	3.581	0.8375	103.7	16.85
Run 10	-35	1.512	2.84	4.372	0.8232	96.6	17.08

For a constant condensing temp, as the evaporator temp decreases, pressure ratio increases as shown in above table. As the evaporator temp increases, the refrigeration effect increases marginally and the required compressors work decrease significantly therefore the performance of the cascade system increases considerably. Compression

Work required in LTC decreases with increase in evaporator temp since pressure ratio is decreases. Hence combined work required also reduces.

5.1.1 Impact of evaporator temp on COP

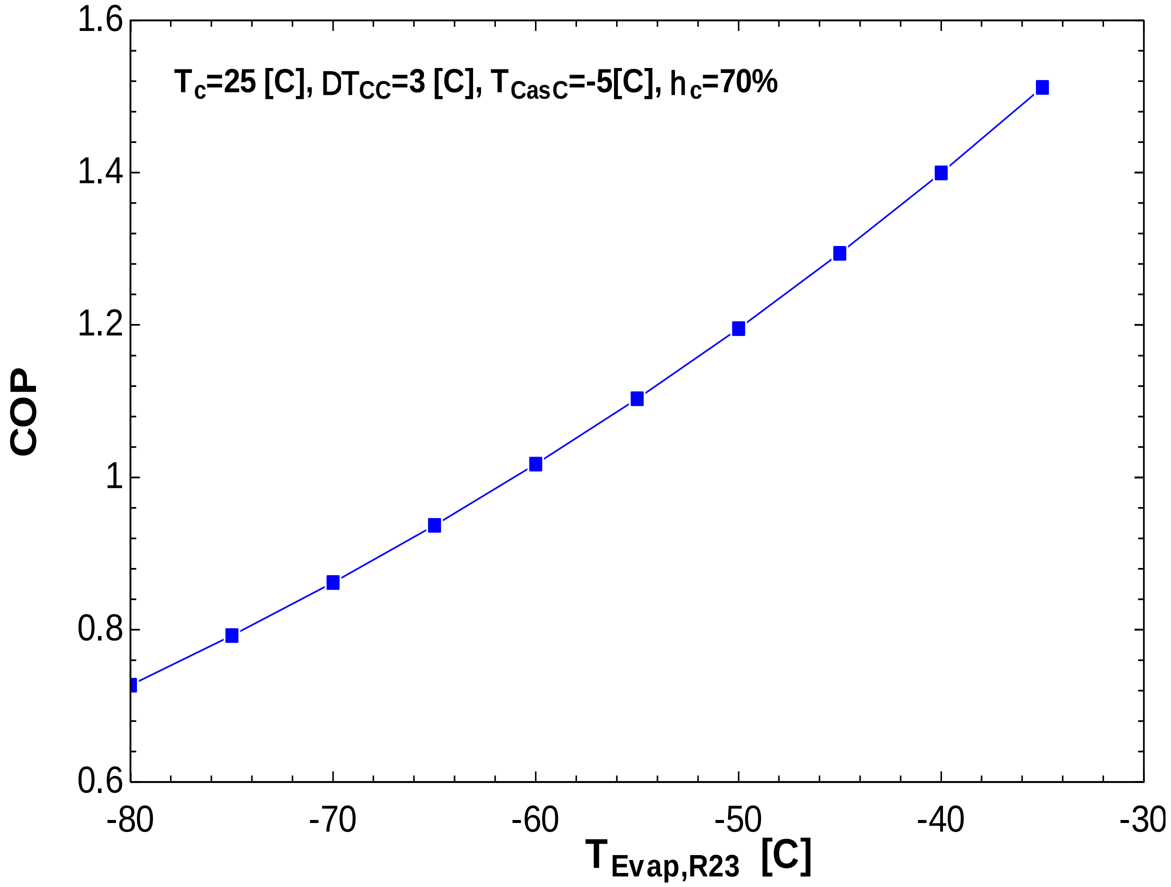

Fig. 5.1: Impact of evaporator temp on COP

This is obvious from the above graph that as the evaporating temperature T_E increases, the COP also increases. In the graph it is shown that when the ΔT= -3°C and the evaporating temp is -80°C and condensing temp is 25°C, the COP is around 0.727 and after that it is increasing with increase in evaporating temp at constant condensing temp of 25°C. But as the condensing temp T_C increases the COP decreases. After comparing the results with those of A.D Parekh [11], results are in a substantial agreement with those of A.D Parekh [11] and our results differ slightly from those of results obtained by A.D. Parekh [11] just because of assumption made in the study.

5.1.2 Impact of evaporator temp on exergetic efficiency

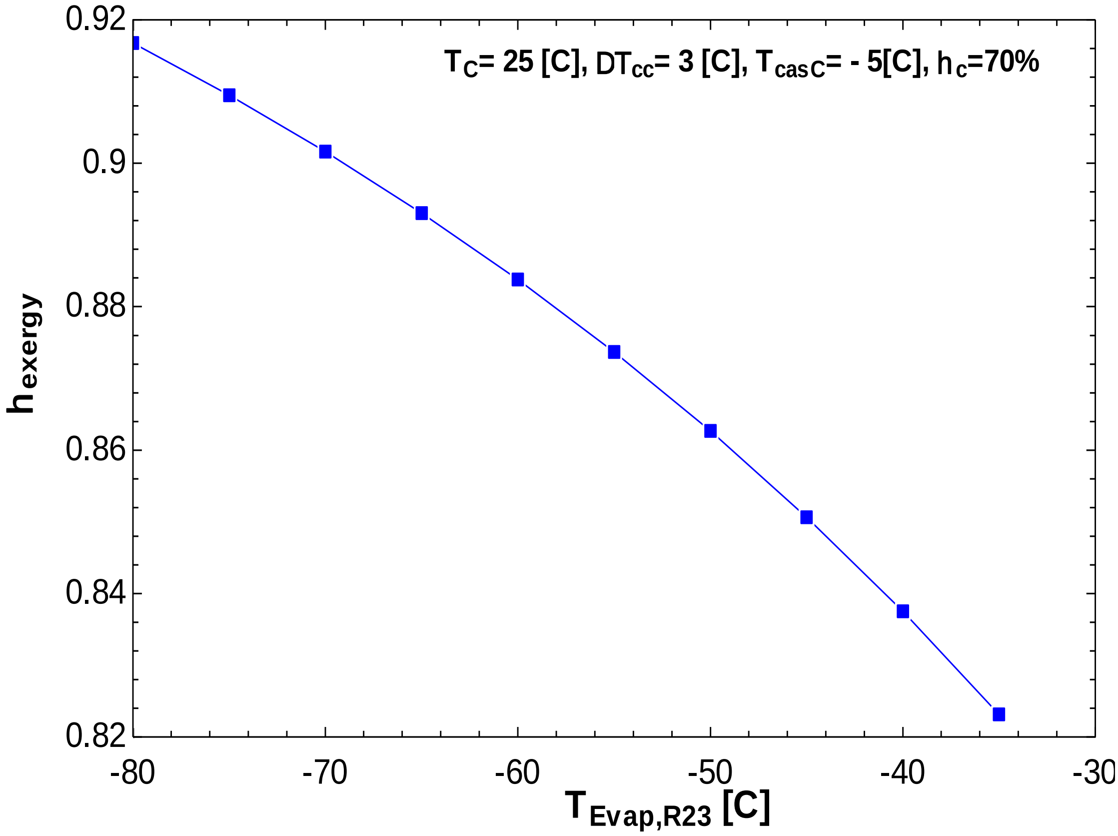

Fig. 5.2: Impact of evaporator temp on exergetic efficiency

This is obvious from the above graph that as the evaporating temp T_E increases, the exergetic efficiency decreases when the evaporating temp T_E =-80°C, T_C=25°C and ΔT = -3°C and the exergetic efficiency is 91.68 and after that it is decreasing with the increase in evaporating temp at constant condensing temp of 25°C. After comparing the results with those of A.D Parekh [11] results are in a substantial agreement with those of A.D Parekh [11] and our results differ slightly from those of results obtained by A. D. Parekh [11] just because of assumption made in the study.

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5.1.3 Impact of the evaporator temp on total compressor work

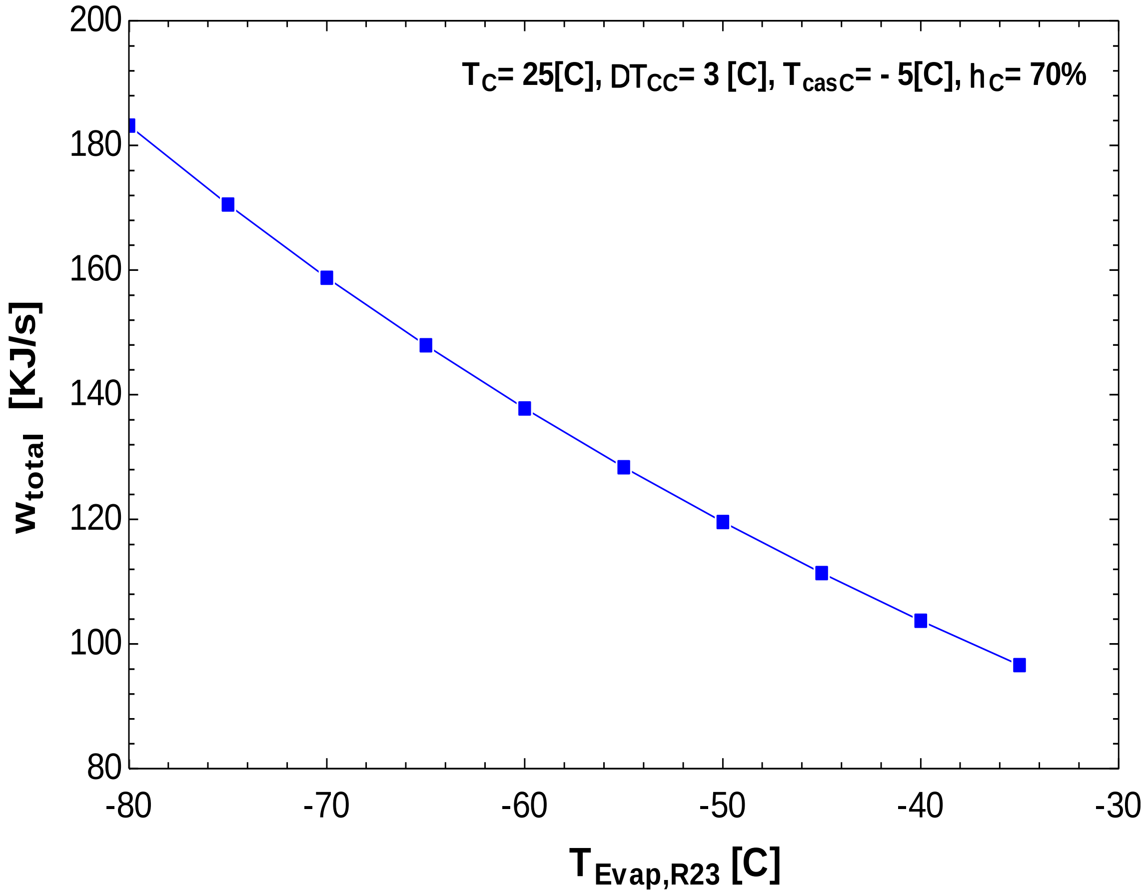

Fig. 5.3: Impact of the evaporator temp on total compressor work

It is obvious from the graph that as the evaporating temp T_E increases, the total compressor work decreases. Total compressor work obtained from EES calculation is 183.2 KJ/s when the evaporating temp T_E =-80°C, T_C=25°C and ΔT= -3°C and after that it is decreases with the increase in evaporating temp at constant ondensing temp of 25°C. After comparing the results with those of A.D Parekh [11], results are in substantial agreements with those of A.D Parekh [11]. Our results differ slightly from those of results obtained by A.D Parekh [11] just because of assumption made in the study.

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5.1.4 Impact of evaporator temp on the complete exergy loss

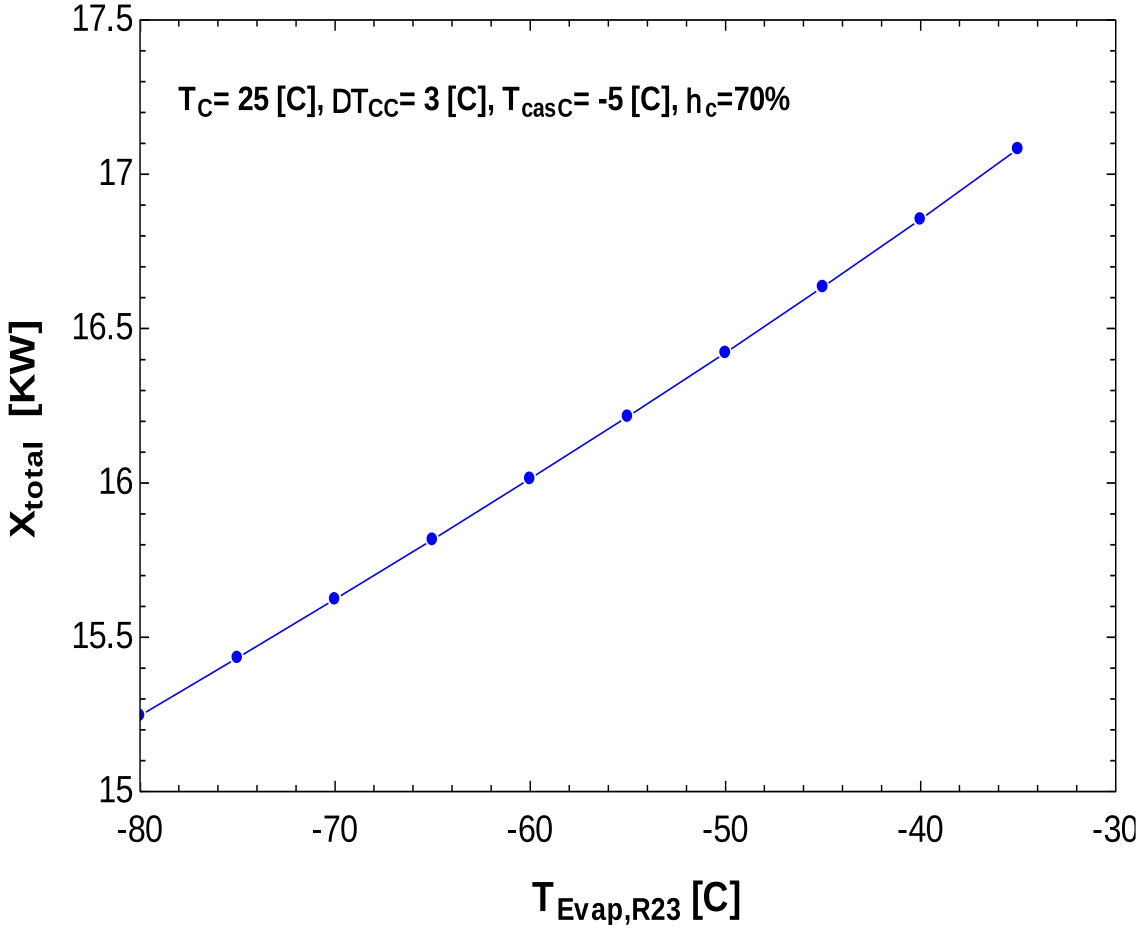

Fig. 5.4: Impact of evaporator temp on total exergy loss

This is obvious from the above graph that as the evaporating temp T_E increases, the total exergetic loss increases. When the evaporating temp T_E =-80°C, T_C=25°C and ΔT= -3°C, the total compressor work is 15.25 KW and after that it is increasing with the increase in evaporating temp at constant condensing temp of 25°C. After comparing the results with those of A.D Parekh [11], results are in substantial agreements with those of A.D Parekh [11]. Even though there is a slight difference between our findings and those of [11], because of assumption made in the study.

5.2 IMPACT OF CONDENSER TEMP

Fig. 5 shows the impact of condenser temp on the COP of cascade system, refrigeration effect and required work in compressors. The results are obtained at fixed -35°C evaporator temp and 15°C coupling temp. As the evaporator and coupling temp are fixed, the refrigeration effect will be constant for entire range of condenser temperature. However, required work in HTC increases due to increase in pressure ratio in HTC. Hence combined work required increases, therefore the COP of cascade system decreases.

Table 5.2: Obtained result with varying condenser temperature

Parametric Table

Table 1 | Table 2

1..10	T_6 [C]	COP	COP_{HTC}	COP_{LTC}	η_{exergy}	X_{total} [KW]	W_{total} [KJ/s]
Run 1	25	0.727	4.007	1.11	0.9168	15.25	183.2
Run 2	30	0.6948	3.532	1.11	0.9256	14.25	191.7
Run 3	35	0.6663	3.17	1.11	0.9337	13.25	199.9
Run 4	40	0.6409	2.883	1.11	0.9412	12.23	207.8
Run 5	45	0.6181	2.652	1.11	0.9481	11.19	215.5
Run 6	50	0.5976	2.462	1.11	0.9545	10.13	222.9
Run 7	55	0.5791	2.302	1.11	0.9606	9.058	230
Run 8	60	0.5623	2.167	1.11	0.9664	7.956	236.9
Run 9	65	0.547	2.05	1.11	0.972	6.824	243.5
Run 10	70	0.533	1.95	1.11	0.9774	5.654	249.9

The condenser temp is varied from 25°C to 70°C in the interval of 5°C by keeping other parameter constant. The effect on COP, exergy efficiency, compressor work and exergy loss is shown in fig 5.5,5.6,5.7, and 5.8.

5.2.1 Impact of condenser temp on COP

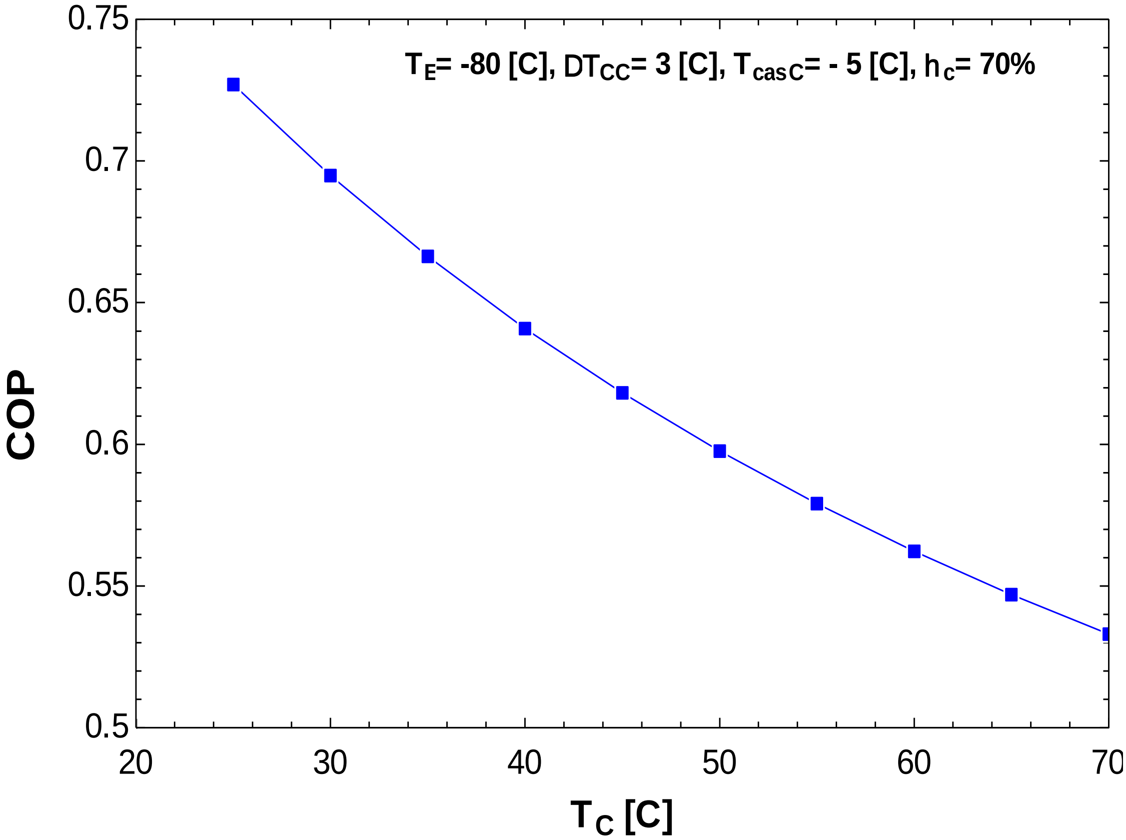

Fig. 5.5: Impact of condenser tem on COP

Fig 5.5 shows that when condenser temp varied from 20°Cto 70°C the COP of system decreases from 0.727 to 0.533. After comparing the results with those of A.D Parekh [11], results are in substantial agreements with those of A.D Parekh [11]. Our results differ slightly from those of results obtained by A.D. Parekh [11] just because of assumption that made in the study.

5.2.2 Impact of condenser temp on exergetic efficiency

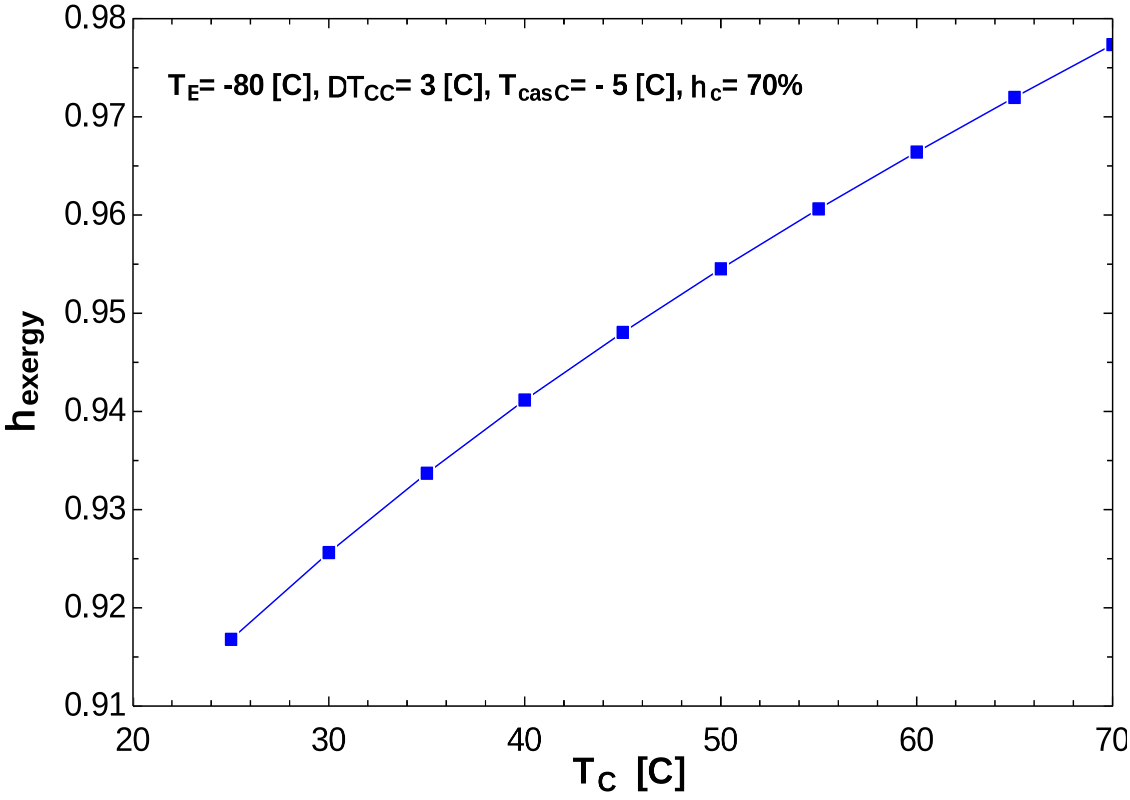

Fig. 5.6: Impact of condenser temp on exergetic efficiency

Fig 5.6 shows that when condenser temp increases which varied from 20°C to 70°C the energetic efficiency increases from 0.9168 to 0.9744.

5.2.3 Impact of condenser temp on total compressor work

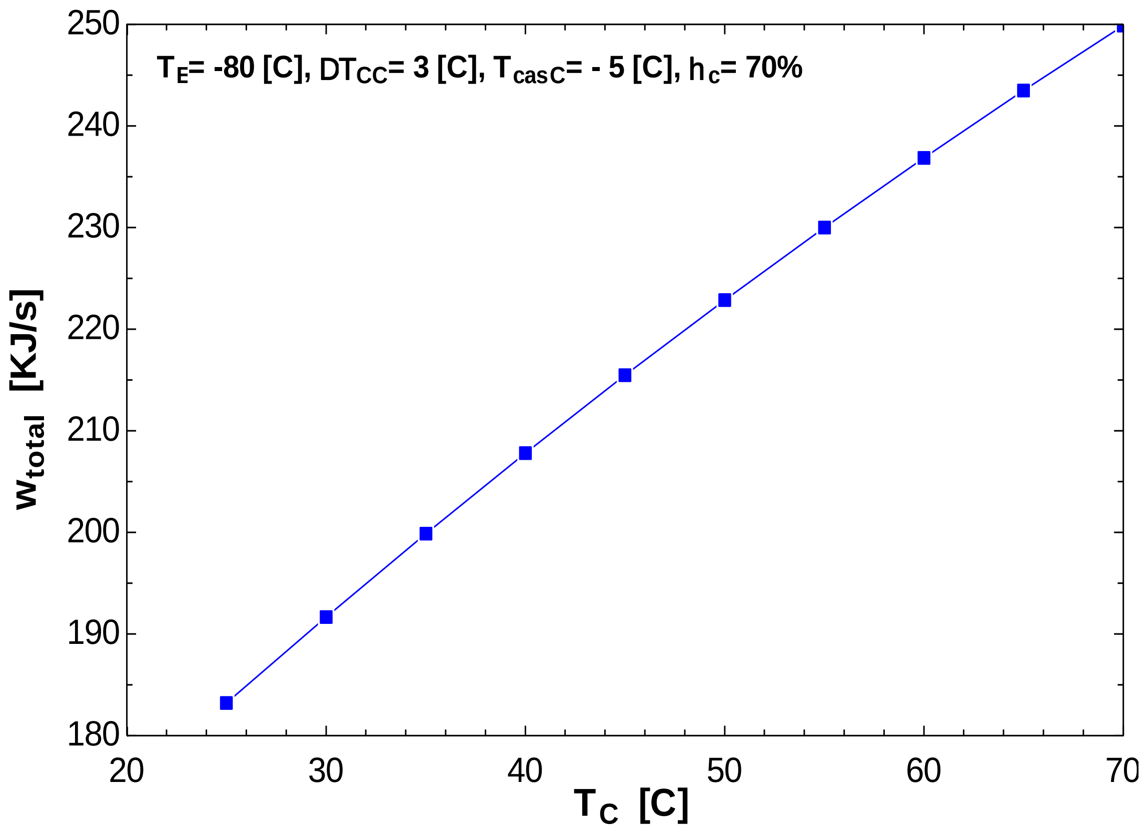

Fig. 5.7: Impact of condenser temp on total compressor work

Fig 5.7 shows the impact of condenser temp Tc on the work input to the compressor and it shows on that when condenser temp increases by varying the temp from 25°C to 70°C total work increases from 183.2 KJ/s to 249.9KJ/s. As the evaporator and coupling temperatures are fixed, the refrigeration effect will be constant for entire range of condenser temperature. However, required work in HTC increases due to increase in pressure ratio in HTC. Hence combined work required increases, therefore the COP of cascade system decreases.

5.2.4 Impact of condenser temp on total exergetic loss

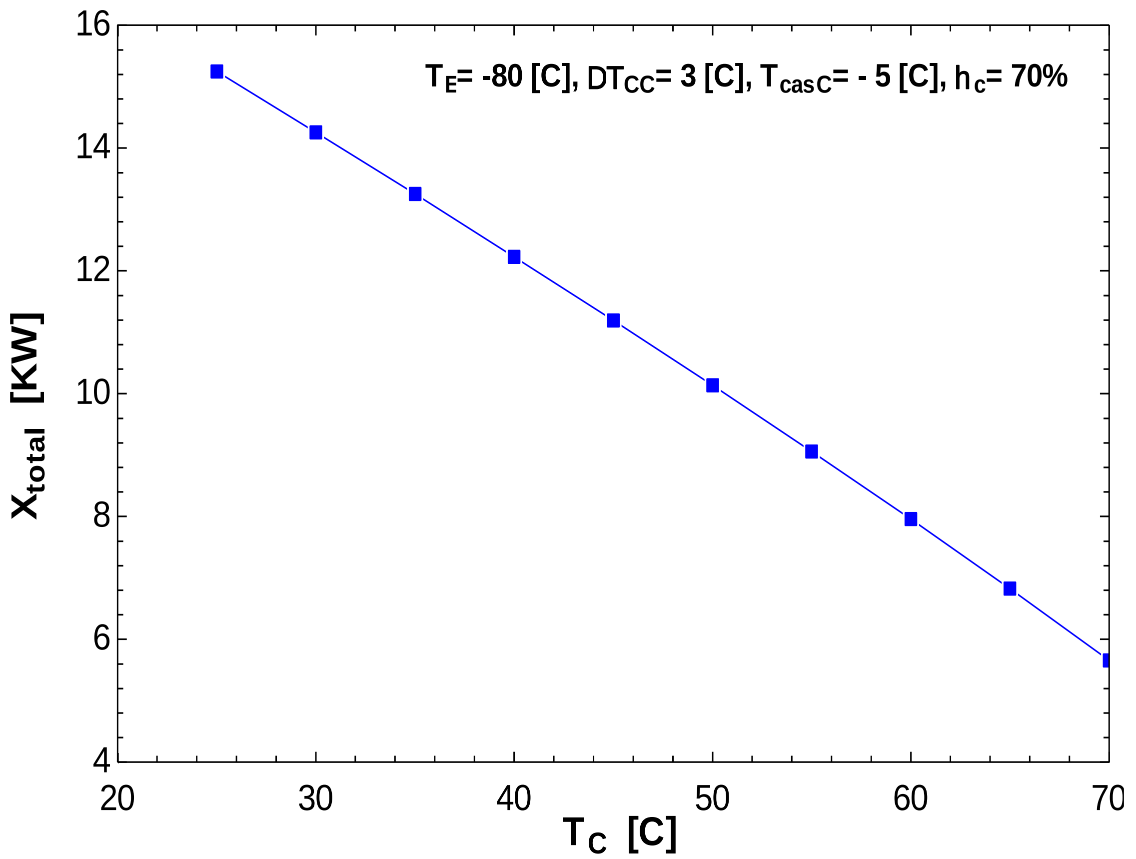

Fig. 5.8: Impact of condenser temp on total exergetic loss

Fig 5.8 depicts the impact of condenser temp Tc on the total exergy loss and it shows on that when condenser temperature increases by varying the temp from 25°C to 70°C total exergy loss decreases from 15.25 KW to 5.64 KW. After comparing the results with those of A.D Parekh [11], results are in substantial agreements with those of A.D Parekh [11]. Our results differ slightly from those of results obtained by A.D. Parekh [11] just because of assumption that I made in this study.

5.3 IMPACT OF L.T CYCLE CONDENSER TEMPERATURE (T_{CasC})

The low temp cycle condenser temp is varied from -35°C to -5°C in the interval of 5°C and the effect on performance parameters is analysed. The effect of low temp cycle condenser temp on COP, exergetic efficiency and refrigerant mass flow ratio is shown in Figs. 5.7-5.9 respectively.

Table 5.3: Obtained result with varying L.T cycle condenser temperature (T_{CasC})

Parametric Table

Table 1 | Table 2 | Table 3

1..10	COP	COP_{HTC}	COP_{LTC}	η_{exergy}	W_{total} [KJ/s]	$X_{cascade,conden}$ [KW]	$X_{condenser}$ [KW]	$X_{Evaporator}$ [KW]	$X_{HT,Expansion}$ [KW]	$X_{LT,compressorA}$ [KW]	$X_{LT,compressorB}$ [KW]	$X_{LT,Expansion}$ [KW]	X_{total} [KW]
Run 1	0.8353	1.842	2.359	0.9399	213.5	2.342	1.34	1.09	2.091	1.909	3.218	0.846	12.84
Run 2	0.8263	2.055	2.055	0.9369	207.4	2.962	1.316	1.054	1.753	2.037	2.92	1.039	13.08
Run 3	0.8138	2.305	1.804	0.9336	201.7	3.587	1.293	1.017	1.452	2.154	2.629	1.254	13.39
Run 4	0.7979	2.604	1.592	0.93	196.5	4.218	1.272	0.9787	1.185	2.263	2.346	1.491	13.75
Run 5	0.7783	2.968	1.41	0.926	191.7	4.855	1.252	0.9394	0.9512	2.364	2.071	1.751	14.18
Run 6	0.7548	3.422	1.251	0.9216	187.3	5.502	1.234	0.8988	0.7472	2.458	1.803	2.037	14.68
Run 7	0.727	4.007	1.11	0.9168	183.2	6.162	1.217	0.8564	0.5713	2.546	1.542	2.351	15.25
Run 8	0.6945	4.789	0.9819	0.9115	179.4	6.839	1.2	0.8121	0.4219	2.628	1.289	2.697	15.89
Run 9	0.6566	5.897	0.8641	0.9056	176	7.542	1.185	0.7651	0.2971	2.705	1.043	3.08	16.62
Run 10	0.6119	7.593	0.7532	0.899	172.8	8.284	1.171	0.7144	0.1955	2.778	0.8037	3.512	17.46

As it is obvious from above table it shows, as condenser temp in low temp cycle increases, COP decreases, the exergetic efficiency when the evaporating temp T_E = -80°C and (ΔT_{CC}) = -3°C,is 93.9% and the exergetic efficiency is decreasing with the condensing temp at constant evaporating temp of 25°C.

5.3.1 Impact of L.T cycle condenser temp on COP

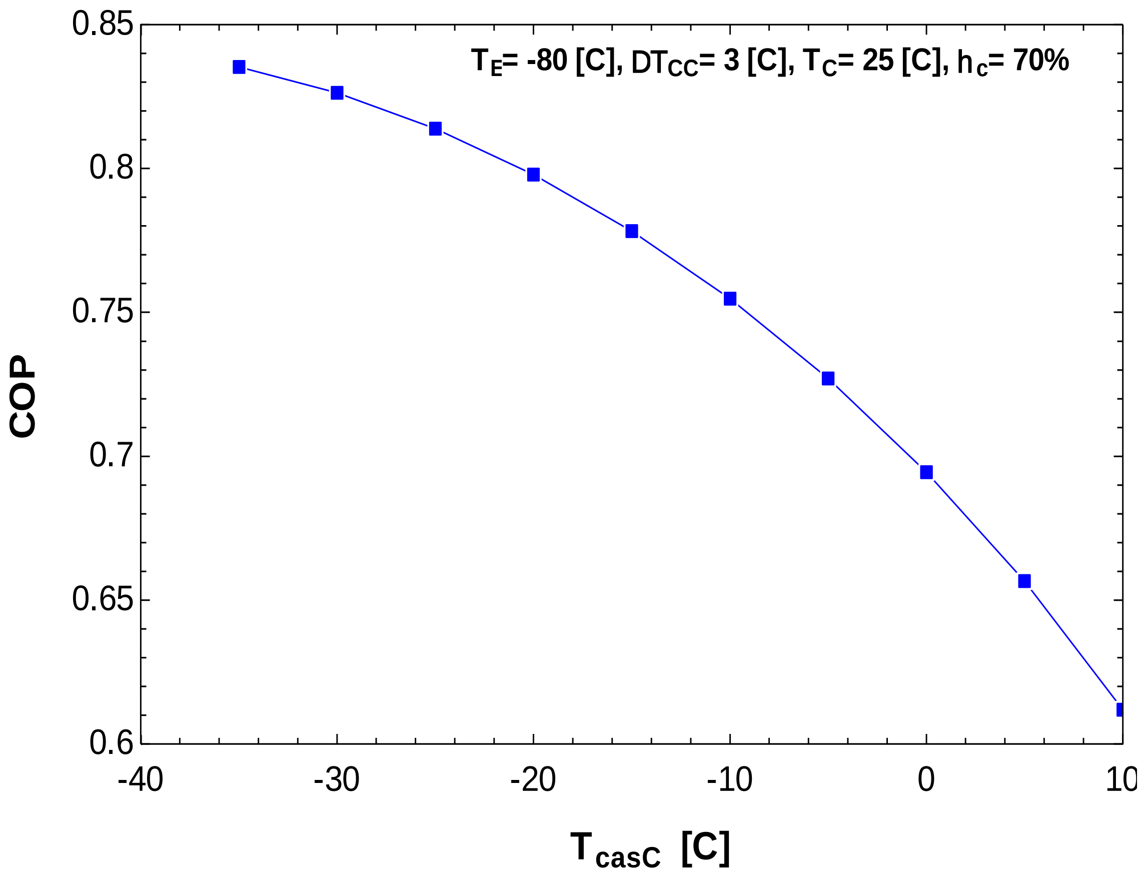

Fig. 5.9: Impact of L.T cycle condenser temp on COP

Fig 5.9 depicts the impact of L.T cycle condenser temp on COP and it shows that as condenser temp increases which varied from -35°C to -5°C COP of the system decreases from 0.83 to 0.61 After comparing the results with those of A.D Parekh [11], results are in substantial agreements with those of A.D Parekh [11]. Our results differ slightly from those of results obtained by A.D. Parekh [11] just because of assumption made in the study.

5.3.2 Impact of L.T cycle condenser temp on total compressor work

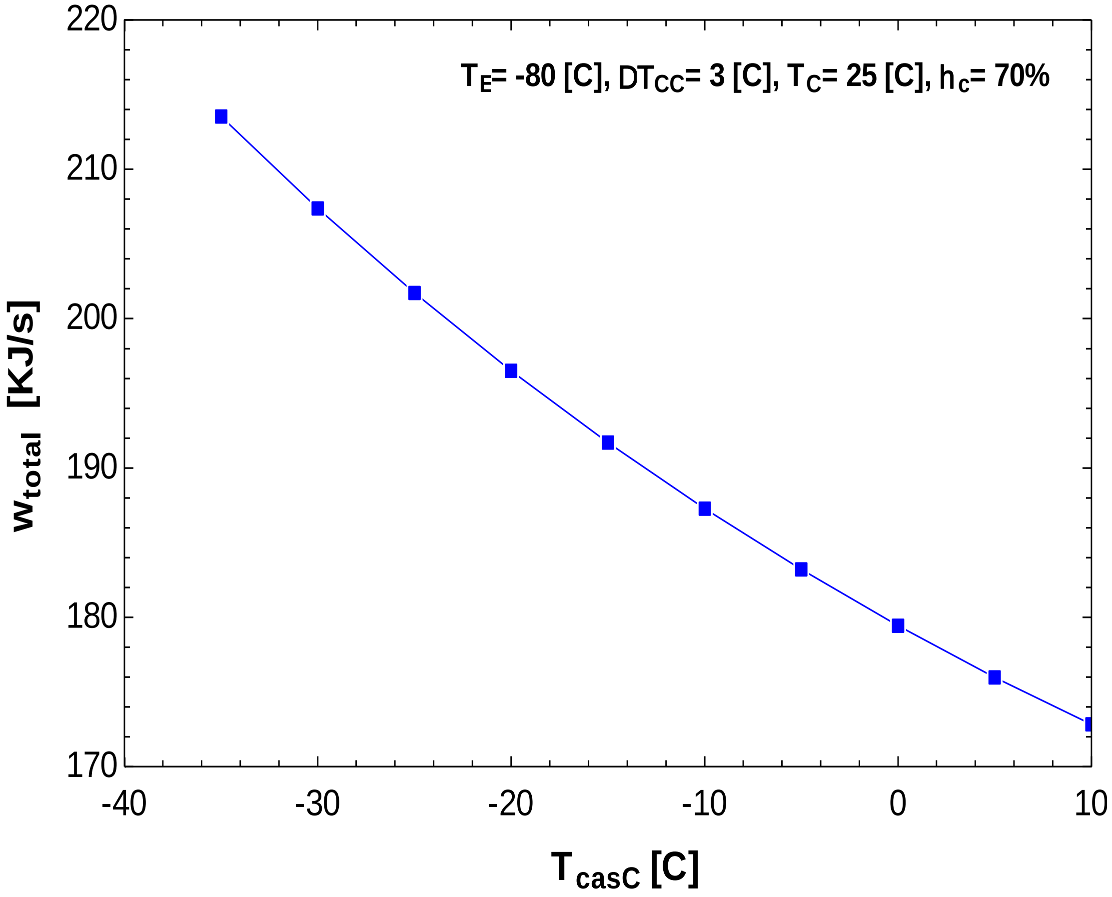

Fig. 5.10: Impact of L.T cycle condenser temp on total compressor work

Fig 5.10 depicts the impact of L.T condenser temp on the work input to the compressor and it shows on that when condenser temperature increases by varying the temp from -35°C to -5O°C total work decreases from 213.5 KJ/s to 172.8KJ/s.

5.3.3 Impact of L.T cycle condenser temp on exergetic efficiency

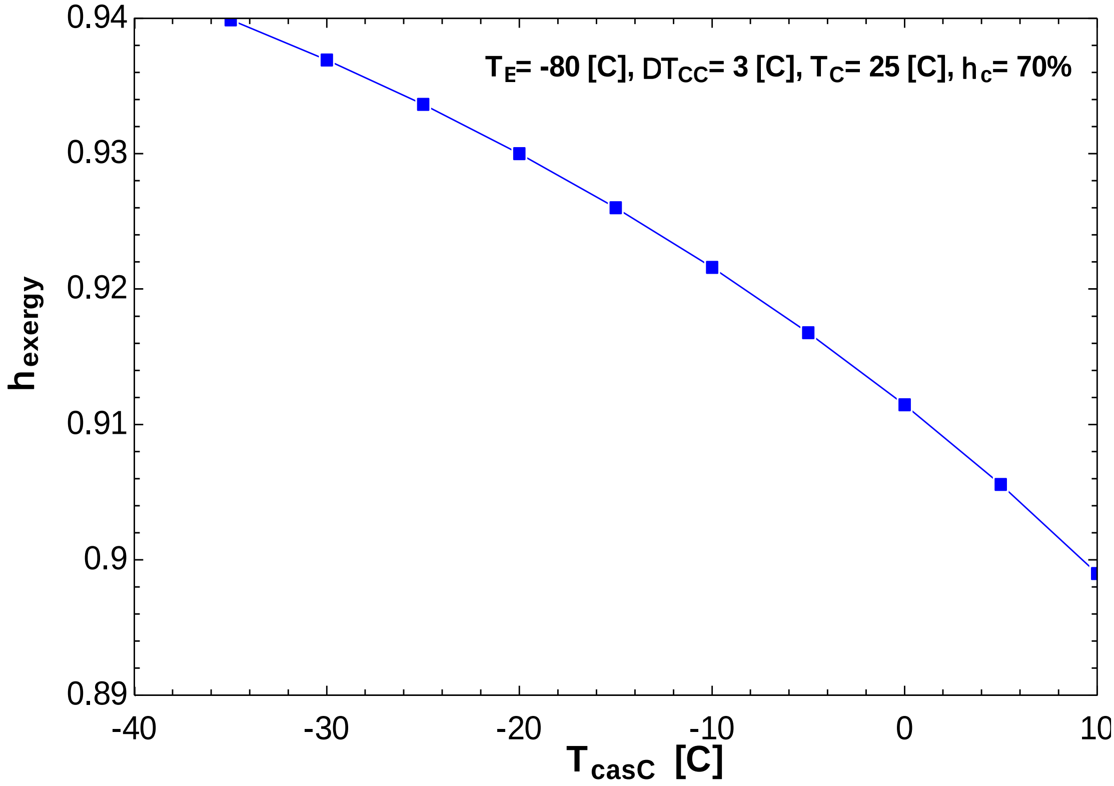

Fig. 5.11: Impact of L.T cycle condenser temp on exergetic efficiency

Fig 5.11 depicts the impact of low temp cycle condenser temp on the exergy efficiency and it shows that as L.T condenser increases, exergy efficiency decreases' from 0.9399 to 0.889 when temp varied from -35oc to -5oc. 61 After comparing the results with those of A.D Parekh [11], results are in substantial agreements with those of A.D Parekh [11]. Our results differ slightly from those of results obtained by A.D. Parekh [11] just because of assumption made in the study.

5.4 IMPACT OF TEMP DIFFERENCE (ΔT_{CC})

The temp difference in the cascade heat exchanger is varied from -3°C to 1.5°C in the interval of 1°C and other parameters are kept constant. The effect of temp difference in cascade condenser on COP, total compressor work and exergetic efficiency is shown in Figs. 5.10- 5.13 respectively.

Table 5.4: Obtained result with varying temperature difference (ΔT_{CC})

Parametric Table

Table 1 | Table 2 | Table 3 | Table 4

1..10	δT	COP	COP_{HTC}	COP_{LTC}	η_{exergy}	W_{total} [KJ/s]	X_{total} [KW]
Run 1	-3	0.727	4.007	1.11	0.9168	183.2	15.25
Run 2	-2.5	0.7313	4.077	1.11	0.9165	182.1	15.2
Run 3	-2	0.7357	4.149	1.11	0.9163	181	15.16
Run 4	-1.5	0.7401	4.224	1.11	0.916	180	15.12
Run 5	-1	0.7445	4.301	1.11	0.9157	178.9	15.07
Run 6	-0.5	0.749	4.38	1.11	0.9155	177.8	15.03
Run 7	0	0.7535	4.462	1.11	0.9152	176.7	14.99
Run 8	0.5	0.7581	4.547	1.11	0.9149	175.7	14.95
Run 9	1	0.7627	4.635	1.11	0.9146	174.6	14.91
Run 10	1.5	0.7673	4.726	1.11	0.9144	173.6	14.86

This is obvious from the graph that as the temp difference (ΔT_{CC}) increases, the COP increases when the evaporating temp T_E = -80°C and ΔT_{CC} =-3°C, the COP is increasing with the condensing temp at constant evaporating temp of 25°C and the exergetic efficiency is decreasing but total exergetic losses decreases.

5.4.1 Impact of temp difference (ΔT_{CC}) on COP

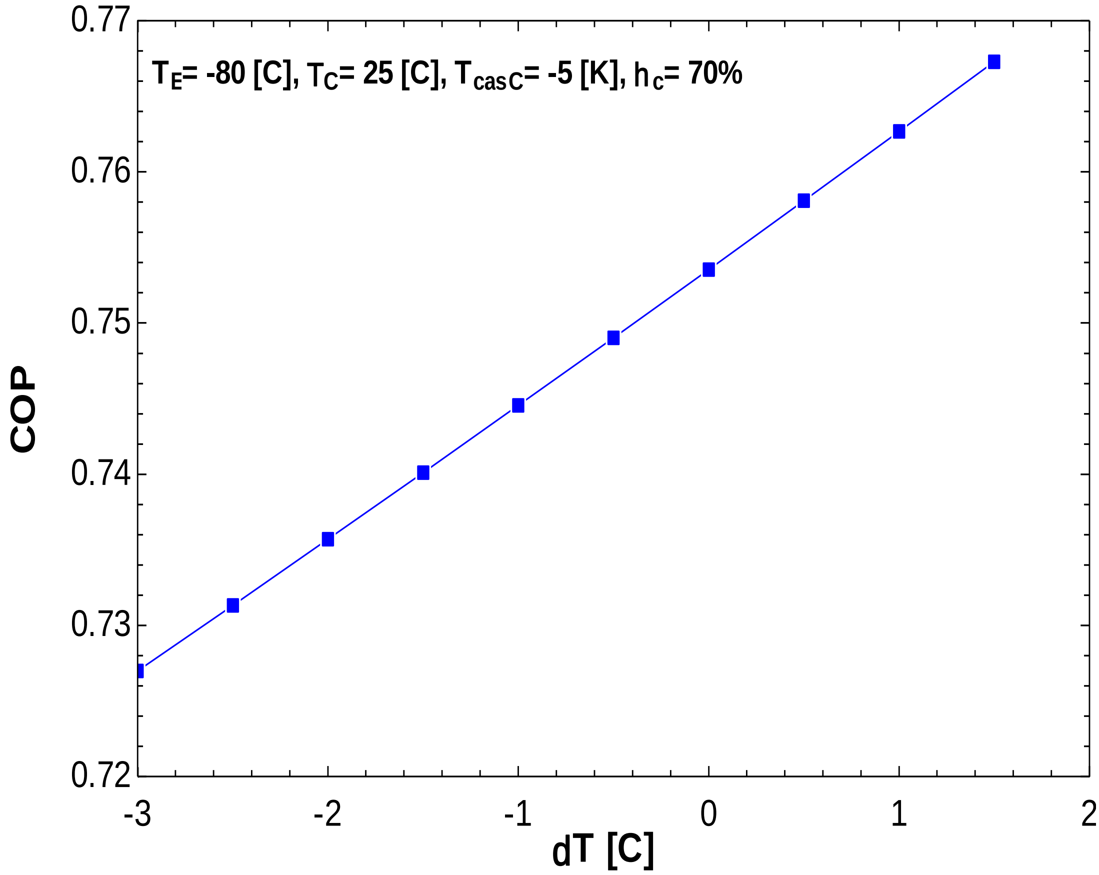

Fig. 5.12: Impact of temp difference (ΔT_{CC}) on COP

The effect of temp difference in cascade condenser on COP is shown in fig 5.12 when the temp differences in the cascaded condenser increases the COP of the system decreases. COP varies from 0.727 to 0.7673 when temp varied from -3 to 1.5oc in the interval of 1°C.

5.4.2 Impact of temp difference (ΔT_{CC}) on total compressor work

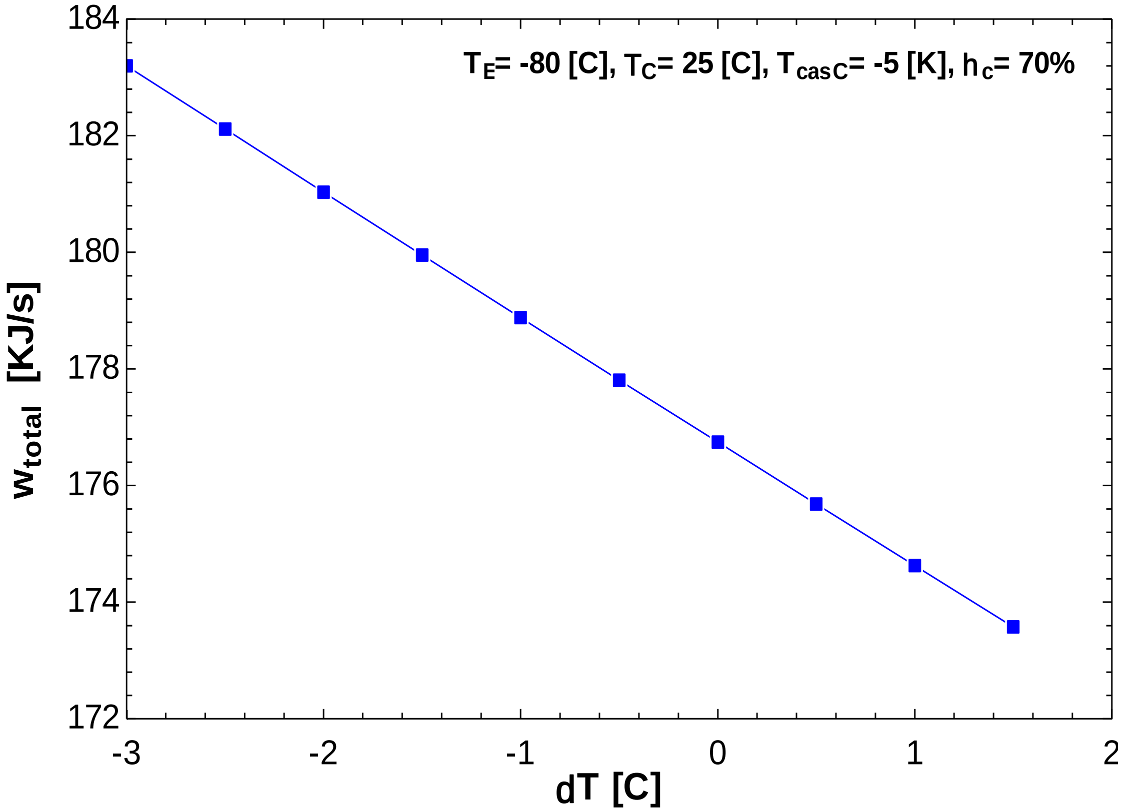

Fig. 5.13: Impact of temp difference (ΔT_{CC}) on total compressor work

The effect of temp difference in cascade condenser on total work input is shown in fig 5.13. It depicts that when the temp differences in the cascaded condenser increases the work input of the system decreases. Total work input to the compressor varies from 183.2 KW to173.6 KW when temp varied from -3 to 1.5oc in the interval of 1°C.

5.4.3 Impact of temp difference (ΔT_{CC}) on exergetic efficiency

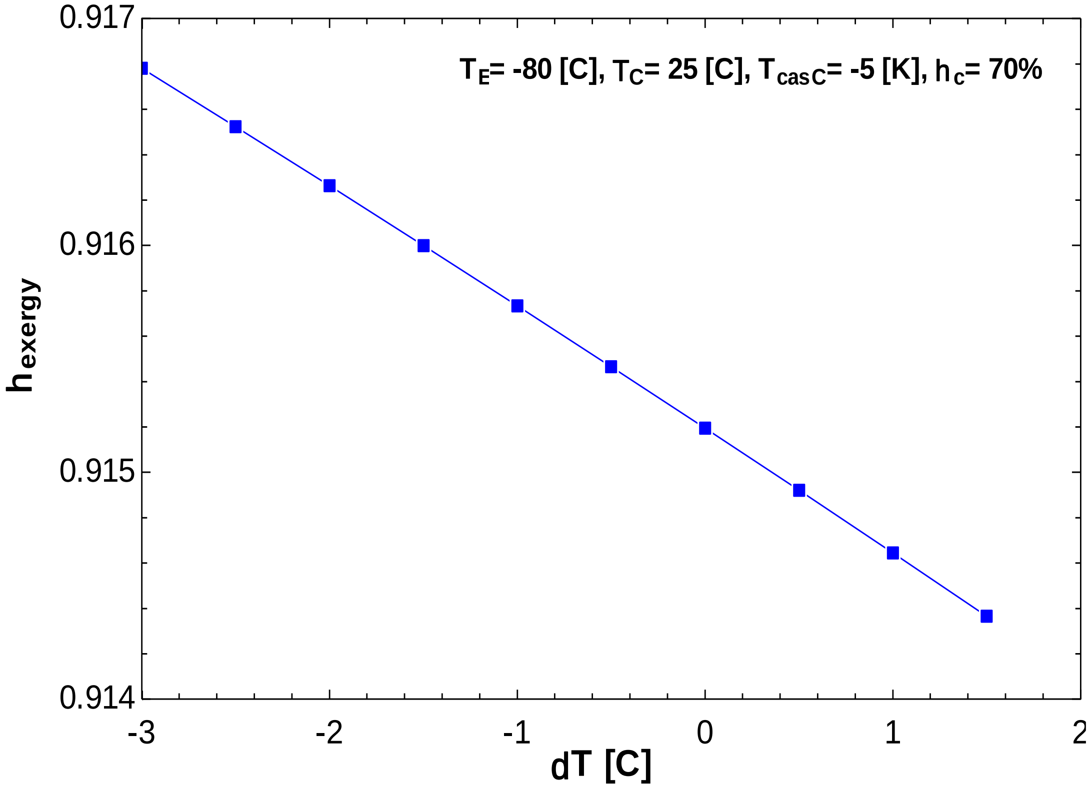

Fig. 5.14: Impact of temp difference (ΔT_{CC}) on exergy efficiency

The effect of temp difference in cascade condenser on exergy efficiency is shown in fig 5.14 When the temp difference in cascade increases, the exergy efficiency decreases and it varies from 0.9168 to 0.9144 when temp varies from -3°C to -1.5°C.

5.4.4 Impact of temp difference (ΔT_{CC}) on total exergetic loss

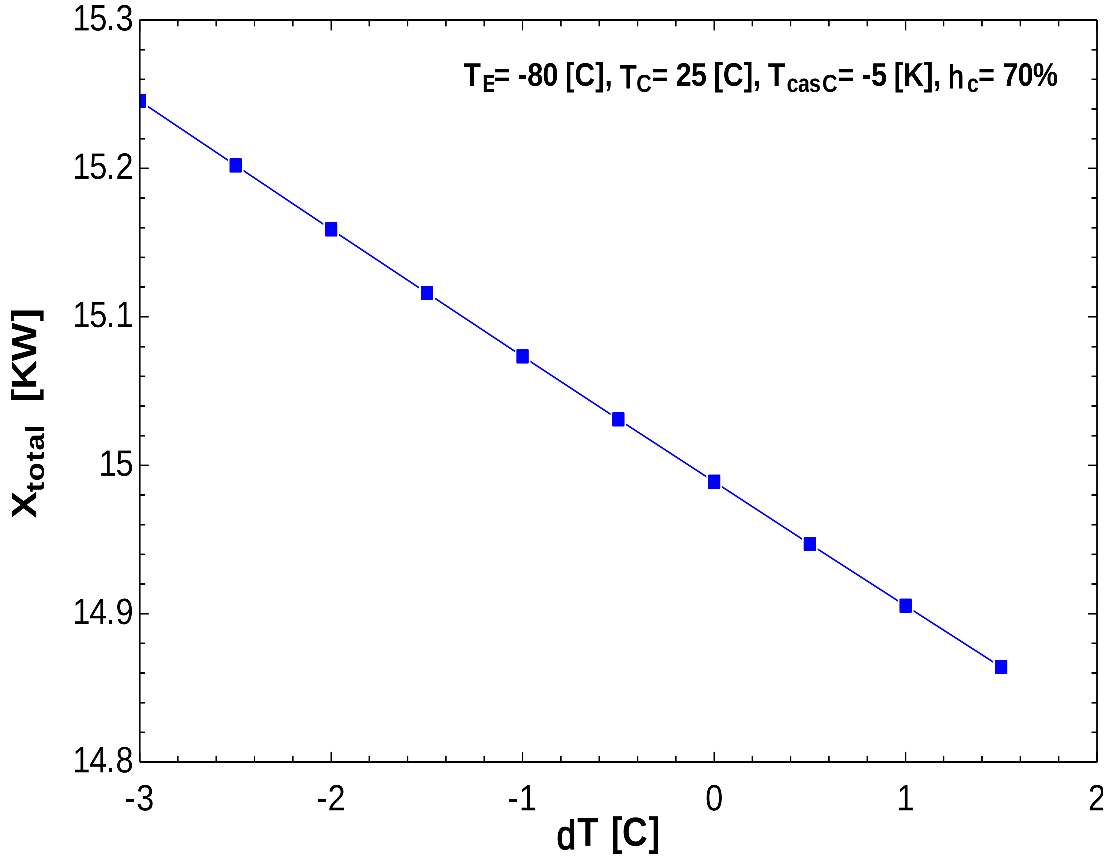

Fig. 5.15: Impact of temp difference (ΔT_{CC}) on total exergetic loss

It depicts from the graph that as the temp difference (ΔT_{CC}) increases, the total exergetic loss is decreases when the evaporating temperature T_E = -80°C and ΔT_{CC} = -3°C.

5.5 IMPACT OF EVAPORATOR TEMPERATURE

The effect on COP, exergy efficiency and total exergy loss, when evaporator temp varied from -80°C to -60°C in the interval of 5°C keeping other parameters constant is shown in Figs. 3 (a), (b), and (c) respectively. For a constant condenser temp, the pressure ratio increases when the evaporator temp decreases as shown in table 5.5

Table 5.5: Obtained result with varying evaporator temperature

1..10	T_1 [C]	COP	η_{exergy}	X_{total} [KW]	W_{total} [KJ/s]
Run 1	-80	0.7491	0.9273	12.92	177.8
Run 2	-75	0.8181	0.9206	13.11	165.1
Run 3	-70	0.8924	0.9133	13.3	153.4
Run 4	-65	0.9724	0.9053	13.49	142.5
Run 5	-60	1.059	0.8966	13.69	132.4
Run 6	-55	1.152	0.887	13.89	123
Run 7	-50	1.252	0.8765	14.1	114.2
Run 8	-45	1.36	0.865	14.31	106
Run 9	-40	1.476	0.8522	14.53	98.34
Run 10	-35	1.601	0.8382	14.76	91.19

Table 5.5 shows that at constant condenser temp the ref effect increases slightly when the evaporator temp increases the required compressors work decrease significantly therefore the erformance of the CRS increases considerably. Compression Work required in LTC decreases with increase in the evaporator temp since pressure ratio is decreases. Hence combined work required also reduces.

5.5.1 Impact evaporator temp on COP

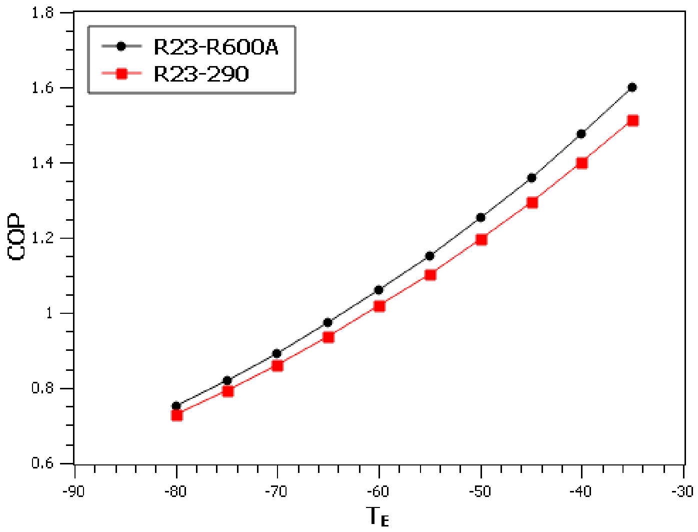

Fig. 5.16: Impact of evaporator temp on COP

Fig 5.16 shows the impact of evaporator temp on the COP of the system and it shows that when evaporator temp increases by varying the temp from -90°C to -30°C COP increases from 0.7491 to 1.601.COP increases for both the refrigerant pair R23-R600A and R290-R23 respectively. But among two pair R23-R600A shows maximum change in COP followed by R290-R23.

5.5.2 Impact of evaporator temp on total compressor work

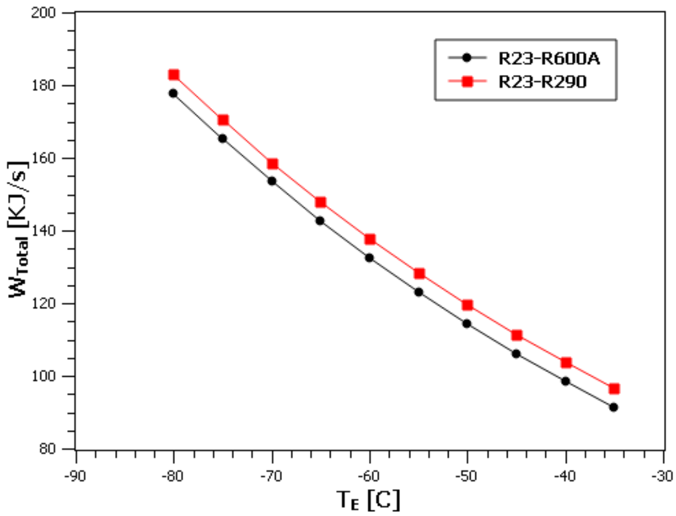

Fig. 5.17: Impact of evaporator temp on total compressor work

Fig 5.17 shows the impact of evaporator temp on the total work input to compressor of the system and it shows that when evaporator temp increases by varying the temp from -90°C to -30°C total work input decreases' total compressor work decreases for R23-R600A followed by R290R23 respectively. But Among two pair R23-R600A shows minimum change in total compressor work followed by R290-R23.

5.5.3 Impact of evaporator temp on exergetic efficiency

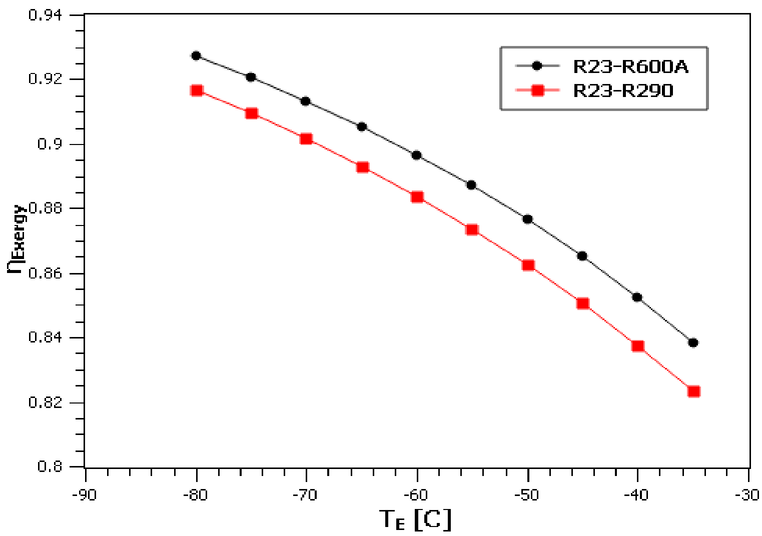

Fig 5.18 Impact of evaporator temp on exergetic efficiency

Fig 5.18 shows the impact of evaporator temp on the second low or exergy efficiency of the system and it shows that when evaporator temp increases by varying the temp from -90°C to -30°C exergy efficiency decreases. But among two pair R23-R600A shows maximum change in exergy efficiency followed by R290-R23.

5.5.4 Impact of evaporator temp on total exergetic loss

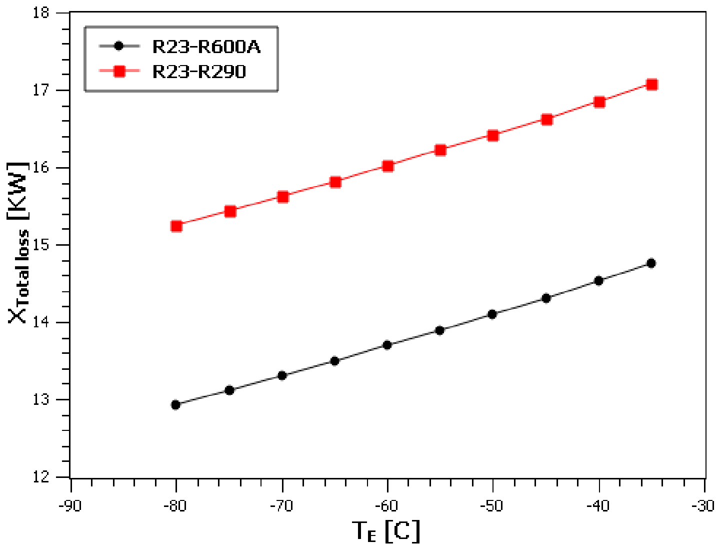

Fig. 5.19: Impact of evaporator temp on total exergetic loss

Fig 5.16 shows the impact of evaporator temp on the complete exergy loss of the system and it shows that when evaporator temperature increases by varying the temp from -90°C to -30°C the exergy losses decreases. But among these two pair R23-R600A shows maximum change in exergetic efficiency followed by R290-R23.

5.6 IMPACT OF CONDENSER TEMPERATURE

The condenser temp is varied from 25°C to 45°C in the interval of 5°C and while another Variable are kept constant. The impact on COP, exergy efficiency and complete exergy loss is shown in Figs. 5.20,5.21,5.23 and 5.24 respectively.

Table 5.6: Obtained result with varying condenser temperature

1..10	1 T_6 [C]	2 COP	3 η_{exergy}	4 W_{total} [KJ/s]	5 X_{total} [KW]
Run 1	25	0.7491	0.9273	177.8	12.92
Run 2	30	0.7182	0.9353	185.4	11.99
Run 3	35	0.6909	0.9426	192.8	11.07
Run 4	40	0.6664	0.9493	199.8	10.14
Run 5	45	0.6445	0.9554	206.6	9.212
Run 6	50	0.6248	0.9611	213.2	8.288
Run 7	55	0.607	0.9664	219.4	7.367
Run 8	60	0.5908	0.9714	225.4	6.447
Run 9	65	0.5761	0.9761	231.2	5.529
Run 10	70	0.5627	0.9805	236.7	4.611

5.6.1 Impact of condenser temp on COP

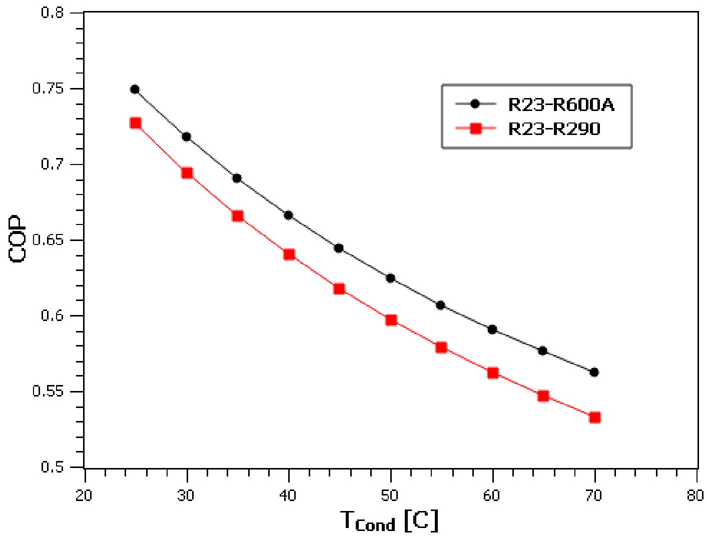

Fig. 5.20: Impact of condenser temp on COP

Fig 5.20 shows the impact of condenser temp on the COP of the system and it shows that when condenser temp increases by varying the temp from 20°C to 80°C the COP decreases. But among two pair R23-R600A shows maximum change in COP followed by R290-R23

5.6.2 Impact of condenser temp on total compressor work

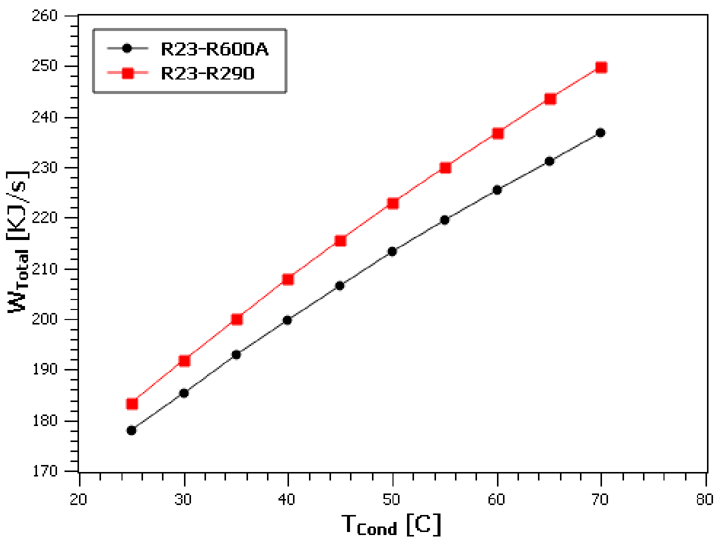

Fig. 5.21: Impact of condenser temp on total compressor work

Fig 5.21 shows the impact of condenser temp on the total work input of the system and it shows that when condenser temp increases by varying the temp from 20°C to 80°C the total work input to the compressor decreases. But among two pair R23-R600A shows minimum change in total compressor work followed by R290-R23.

5.6.3 Impact of condenser temp on exergetic efficiency

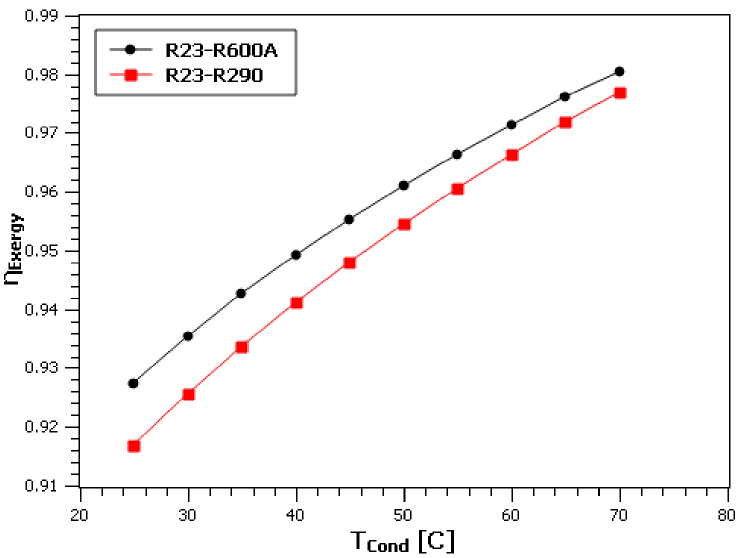

Fig. 5.22: Impact of condenser temp on exergetic efficiency

Fig 5.22 depicts the impact of condenser temp on the exergy efficiency of the system and it shows that when condenser temp increases by varying the temp from 20°C to 80°C the exergy efficiency increases. But among two pair R23-R600A shows maximum change in exergetic efficiency followed by R290-R23.

5.6.4 Impact of condenser temp on total exergetic loss

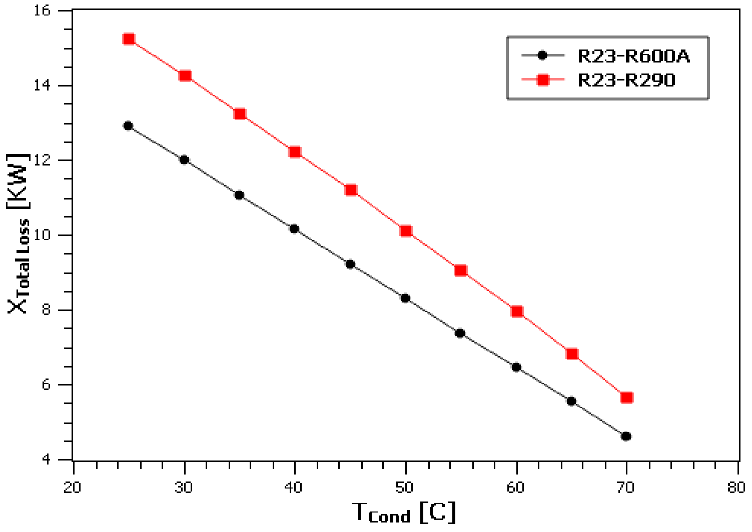

Fig. 5.23: Impact of condenser temp on total exergetic loss

Fig 5.23 shows the impact of condenser temp on the total exergy loss of the system and it shows that when condenser temp increases by varying the temp from 20°C to 80°C the exergy loss decreases. But among two pair R23-R600A shows minimum change in total exergetic loss followed by R290-R23.

5.7 IMPACT OF L.T CYCLE CONDENSER TEMP (T_{CASL})

Table 5.7: Obtained result with varying L.T cycle condenser temp

Parametric Table

Table 6 | Table 2 | Table 3

1..10	1 T_3 [C]	2 COP	3 W_{total} [KJ/s]	4 X_{total} [KW]	5 η_{exergy}
Run 1	-35	0.8911	200.2	9.605	0.952
Run 2	-30	0.8762	195.6	9.994	0.9489
Run 3	-25	0.8579	191.4	10.45	0.9454
Run 4	-20	0.8361	187.5	10.96	0.9415
Run 5	-15	0.8109	184	11.55	0.9372
Run 6	-10	0.782	180.8	12.2	0.9325
Run 7	-5	0.7491	177.8	12.92	0.9273
Run 8	0	0.7118	175.1	13.73	0.9216
Run 9	5	0.6695	172.6	14.63	0.9152
Run 10	10	0.621	170.3	15.64	0.9082

As it is obvious from above table it shows, as condenser temperature in low temperature cycle increases COP decreases, and also decreases the total exergetic loss the exergetic efficiency decreases and when the evaporating temperature increases. And at T_E=-80°C and ΔT_{CC}=-3°C its exergy efficiency is 89.11% .

5.7.1 Impact of L.T cycle condenser temp on COP

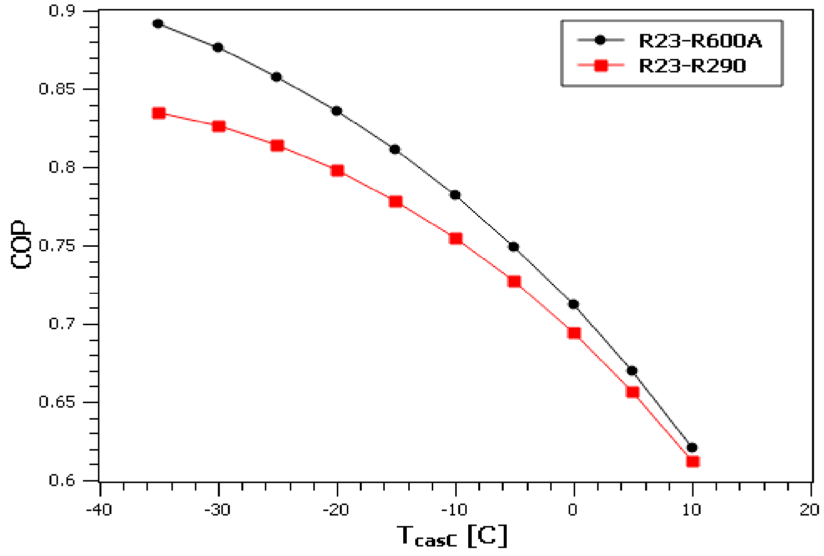

Fig. 5.24: Impact of L.T cycle condenser temp on COP

Fig 5.24 shows the impact of low temp condenser cycle on the COP of the system and it shows that when condenser temp increases by varying the temp from -40°C to 20°C the COP decreases. Among two pair R23-R600A shows minimum change in COP loss followed by R290-R23.

5.7.2 Impact of L.T cycle condenser temp on total compressor work

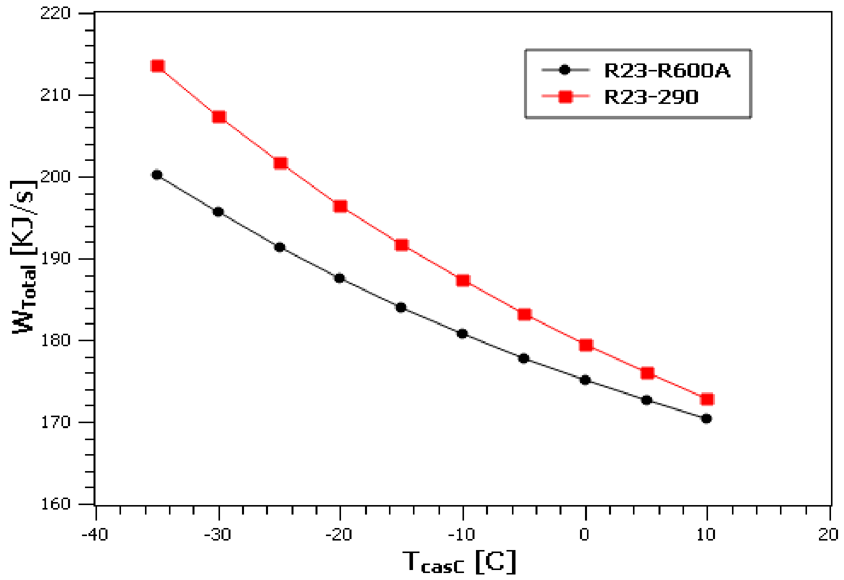

Fig. 5.25: Impact of L.T cycle condenser temp on total compressor work Fig

Fig 5.25 depicts the effect of condenser temp on the total work input of the system and it depicts that when condenser temp increases the total work input to the compressor decreases. But among two pair R23-R600A shows minimum change in total compressor work followed by R290-R23.

5.7.3 Impact of L.T cycle condenser temp on exergetic efficiency

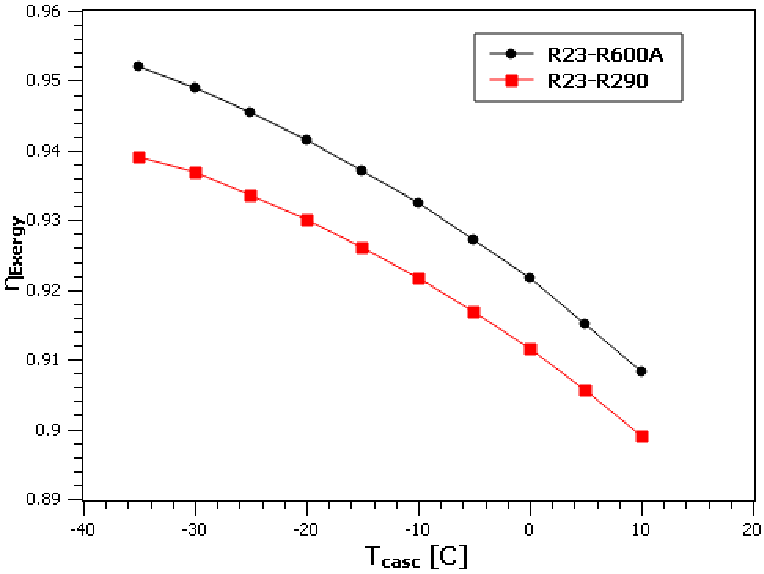

Fig. 5.26: Impact of L.T cycle condenser temp on exergetic efficiency

Fig 5.26 depicts the impact of low temp condenser temp on the exergetic efficiency of the system and it depicts that when condenser temp increases the total exergy efficiency decreases. Among two pair R23-R600A shows minimum change in exergetic efficiency followed by R290-R23.

5.7.4 Impact of L.T cycle condenser temp on total exergetic loss

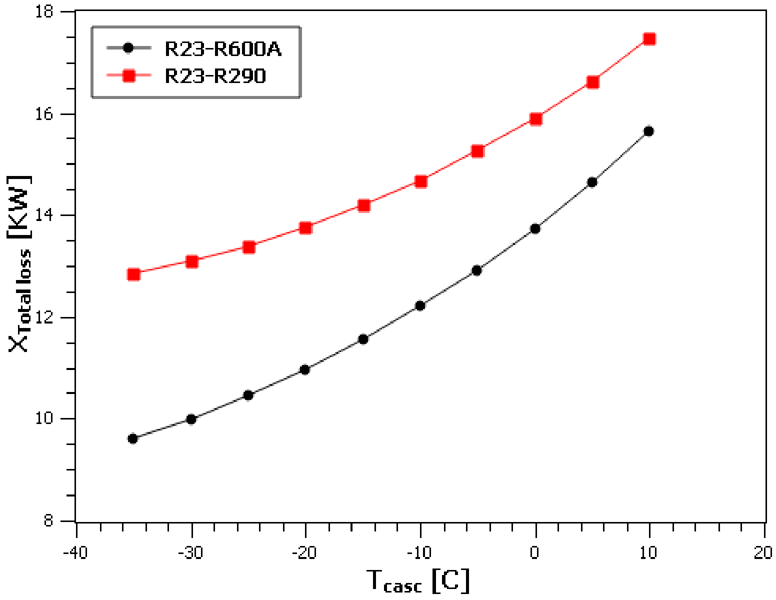

Fig. 5.27: Impact of L.T cycle condenser temp on total exergy loss

Fig 5.27 shows the impact of low temp cycle condenser on the complete exergy loss of the system and it shows that when L.T cycle condenser temp increases the complete exergy loss also increases. Among two pair R23-R600A shows minimum change in exergy loss followed by R290-R23.

5.8 IMPACT OF TEMPERATURE DIFFERENCE (ΔT_{CC})

The temp difference in cascade heat exchanger is varied from 2°C to 6°C in the interval of 1°C while other variables are kept constant. The effect of temp difference in cascade condenser on COP, exergy efficiency, total compressor work and total exergetic loss is shown in Figs. 5.25- 5.28 respectively.

Table 5.8: Obtained result with varying temperature difference (ΔT_{CC})

1..10	δT	COP	η_{exergy}	W_{total} [KJ/s]	X_{total} [KW]
Run 1	-3	0.7491	0.9273	177.8	12.92
Run 2	-2.5	0.7532	0.9271	176.8	12.9
Run 3	-2	0.7574	0.9268	175.8	12.87
Run 4	-1.5	0.7616	0.9266	174.9	12.84
Run 5	-1	0.7659	0.9263	173.9	12.82
Run 6	-0.5	0.7701	0.926	172.9	12.79
Run 7	0	0.7744	0.9258	172	12.76
Run 8	0.5	0.7788	0.9255	171	12.74
Run 9	1	0.7831	0.9252	170.1	12.71
Run 10	1.5	0.7875	0.925	169.1	12.69

This is obvious from above table that as the temp difference (ΔT_{CC}) increases, the COP increases when the evaporating temp T_E = -80°C and ΔT_{CC}= -3°C, the COP is increasing with the condensing temp at constant evaporating temp of 25°C and the second low efficiency is decreasing marginally but total exergetic losses decreases.

5.8.1 Impact of temp difference (ΔT_{CC}) on COP

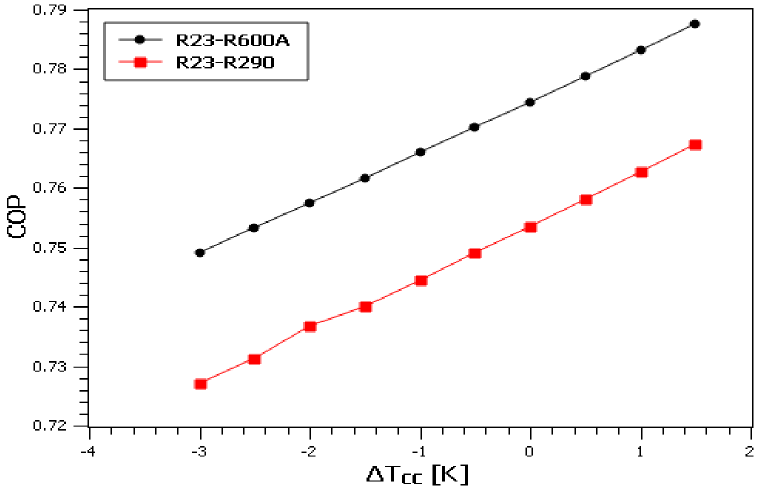

Fig. 5.28: Impact of temp difference (ΔT_{CC}) on COP

Fig 5.21 depicts the impact of temp difference on the COP the system and it shows that when temp difference increases, the COP of the system also increases. Out of two refrigerant pairs R23-R600A responds maximum for change in temp difference in cascade condenser.

5.8.2 Impact of temp difference (ΔT_{CC}) on total compressor work

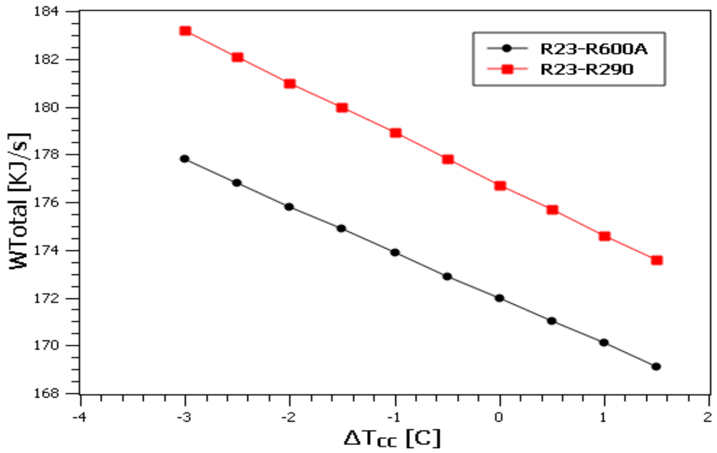

Fig. 5.29: Impact of temp difference (ΔT_{CC}) on total compressor work

Fig 5.21 depicts the impact of temp difference in the cascade condenser on the total input compressor work the system and it shows that when temp difference increases the compressor work of the system decreases. Out of two refrigerant pairs R23-R600A responds maximum for change in temp difference in cascade condenser.

5.8.3 Impact of temp difference (ΔT_{CC}) on exergetic efficiency

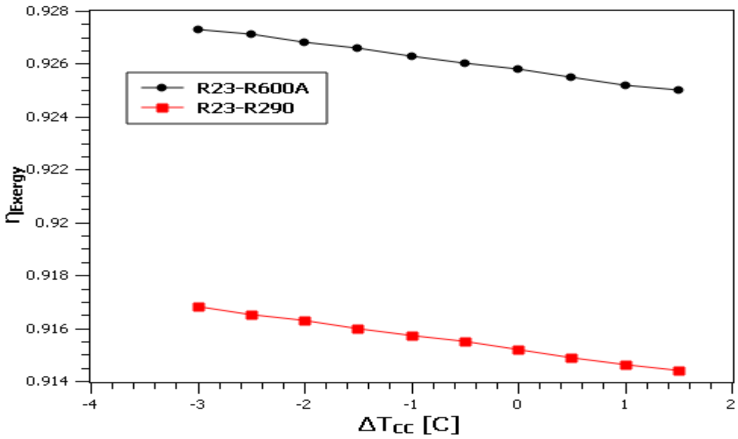

Fig. 5.30: Impact of temperature difference (ΔT_{CC}) on exergetic efficiency

Fig 5.30 depicts the impact of temp difference in Cascade condenser on the total exergetic efficiency the system and it shows that when temp difference increases the exergetic efficiency of system decreases. Out of two refrigerant pairs R600A-R23 responds maximum for change in temp difference in condenser.

5.8.4 Impact of temp difference (ΔT_{CC}) on total exergetic loss

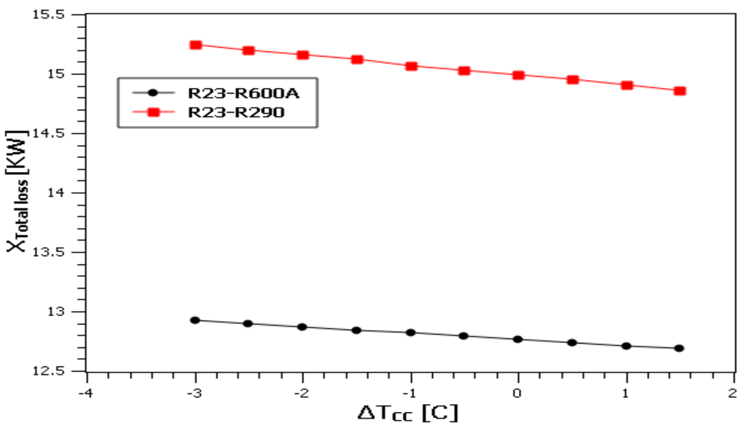

Fig. 5.31: Impact of temp difference (ΔT_{CC}) on total exergetic loss

Fig 5.31 depicts the impact of temp difference in the cascade condenser on the total exergy loss of the system and it shows that when temp difference increases the total exergy loss of system decreases. Out of two refrigerant pairs R600A-R23 responds maximum for change in temp difference in cascade condenser. Out of two refrigerant pairs R290-R23 responds maximum for change in temp difference in cascade condenser.

CHAPTER 6

CONCLUSION AND FUTURE SCOPE

6.1 CONCLUSION

Current work has included the development of a computational model in EES to study the impact of multiple operative variables on the performance parameters of cascade ref rigeration systems. This research draws the following findings.

1. Thermodynamics analysis shows that R23-R600A has a better performance than R23-R293.
2. It has been found out that when evaporator temperature varied from -80°Cto -60°C the COP of the system increased by 39.88% for R23-R290 and 41.36 % for R23-R600A.
3. The findings shows that when condenser temp change from 25°C to 45°C COP decreased by 14.97% for R23-Rr290 and 13.96% for R23-R600A.
4. It is observed that when temp difference in cascade condenser varied from -3°C to 1.5°C the COP of system increased by 5.54 % for R23-R290 and 5.126 % for R23-R600A.
5. When lower temperature cycle condenser temperature varied from -35°C to -5°C COP of the system decreased by 12.96 % for R23-R290 and 15.93 % for R23-R600A.
6. When evaporator temperature varied from -80°C to -60°C the exergy efficiency decreased by 3.59 % for R23-R290 and 3.3 % FOR R23-R600A.
7. When condenser temperature varied from 25°C to 45°C the exergy efficiency increased by 3.414 % for R23-R290 and 3.03% for R23-R600A.
8. When temp difference in cascade condenser varied from -3°C to 1.5°C the exergy efficiency decreased by 0.26 % for R23-R290 and 0.215 % for R23-R600A.
9. When lower temperature cycle condenser temperature varied from -35°C to -5°C exergy efficiency decreased by 5.32 %for R23-R290 and 4.6 % for R23-R600A.

10. when evaporator temperature varied from -80°C to -60°C the total power input to the compressor of the system decreased by 24.78% for R23-R290 and 25.53 % for R23-R600A.
11. when condenser temperature varied from 25°C to 45°C the total work input increased by 17.63 % for R23-Rr290 and 1% for R23-R600A.
12. that when temp difference in cascade condenser varied from -3°C to 1.5°C the total work input of compressor of system decreased by 5.24 % for R23-R290 and 4.89 % for R23-R600A.
13. When lower temperature cycle condenser temperature varied from -35°C to -5°C total work input to compressor of the system decreased by 14.19 % for R23-R290 and 11.18 % for R23-R600A.
14. when evaporator temperature varied from -80°C to -60°C the complete exergetic loss of the system increased by 4.98% for R23-R290 and 5.95 % for R23-R600A.
15. When condenser temperature varied from 25°C to 45°C total exergy loss decreased by 26.62% for R23-Rr290 and 28.69% for R23-R600A.
16. when temp difference in cascade condenser varied from -3°C to 1.5°C the complete exergy loss of system decreased by 1.78 % for R23-R290 and 2.55 % for R23-R600A.
17. When lower temperature cycle condenser temperature varied from -35°C to -5°C total exergy loss of the system increased by 35.98 % for R23-R290 and 34.51 % for R23-R600A.

5.2 FUTURE SCOPE

In this report, two refrigerant pairs, R290-R23 and R600A-R23, were used to perform thermodynamic assessment of the cascade refrigeration system. Effect of different parameters of operations like temp of the evaporator, temp of the condenser, difference of temp in cascades condenser and low cycle condenser tem have been performed on the performance variables like COP, exergy efficiency, total input compressor, total exergy losses etc. This work can be expanded, future trends and the direction of studies can be taken into account as follows:

1.Develpoment of computer software code to determined different types of refrigerant properties.

2. Hunt for the alternative refrigerants without hampering the COP can be added to this work

3. Exergy analysis can be done to each components of the system.

4. More refrigerants and mixture of refrigerants can be used as the working substance.

5. Actual cycle assessment can be performed by taking into account the volumetric efficiency of the compressor and the pressure loss in the system.

Further this cycle can be studied for different refrigeration pairs and their effect on performance of the cycle.

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APPENDEX

EES CODE FOR R23-R290

```
"input"
T_1=-80
T_3=-5
T_6=25
deltaT=-3
T_o=25
T_F=30
x_1=1
Eta_c=0.7
m_dot_1=1
m_dot_2=1

"LTC"

"Evaporator outletA"
P_1=Pressure(R23,T=T_1,x=x_1)
h_1=Enthalpy(R23,T=T_1,x=x_1)
s_1=Entropy(R23,T=T_1,h=h_1)

Q_L=m_dot_1*(h_1-h_4)

"CompressorA outlet"
P_2=P_3
x_3=0
s_1=s_2s
s_2=Entropy(R23,P=P_2,h=h_2)
P_3=Pressure(R23,T=T_3,x=x_3)
h_2s=Enthalpy(R23,P=P_2,s=s_1)
h_2=(h_2s-h_1)/Eta_c+h_1
w_1=m_dot_1*(h_2-h_1)

"Exergy loss in compressorA"
X_LT_compressorA=m_dot_1*T_o*(s_2-s_1)

"Cascade condenser LTC"
T_5=T_3+deltaT
h_3=Enthalpy(R23,T=T_3,x=x_3)
s_3=Entropy(R23,T=T_3,x=x_3)
Q_M=m_dot_1*(h_2-h_3)

"Exergy loss in Cascade condenser LTC"
X_cascade_condenser=T_o*m_dot_1*(s_3-s_2)+T_o*m_dot_2*(s_5-s_8)

"Expansion device LTC"
P_4=P_1
h_4=h_3
T_4=T_1
s_4=Entropy(R23,P=P_1,h=h_4)

"Exergy loss in expansion device LTC"
X_LT_Expansion=m_dot_1*T_o*(s_4-s_3)
```

```
"HTC"

"Evaporator outletB"
x_5=1
P_5=Pressure(R290,T=T_5,x=x_5)
h_5=Enthalpy(R290,T=T_5,x=x_5)
s_5=Entropy(R290,T=T_5,x=x_5)

"Exergy loss in evaporator"
X_Evaporator=m_dot_1*(s_1-s_4)+m_dot_1*(T_F-T_O)/T_F

"CompressorB outlet"
x_7=0
P_6=P_7
P_7=Pressure(R290,T=T_6,x=x_7)
h_6s=Enthalpy(R290,P=P_6,s=s_5)
h_6=(h_6s-h_5)/Eta_c+h_5
w_2=m_dot_1*(h_6-h_5)

"Exergy loss in compressorB"
s_6=Entropy(R290,P=P_6,h=h_6)
X_LT_compressorB=m_dot_1*T_o*(s_6-s_5)

"Condenser Outlet"
T_7=T_6
h_7=Enthalpy(R290,T=T_7,x=x_7)
s_7=Entropy(R290,T=T_7,x=x_7)

"Exergy loss in condenser"
X_condenser=m_dot_2*(s_6-s_7)

"Expansion device HTC"
P_5=P_8
T_8=T_5
h_8=h_7
s_8=Entropy(R290,P=P_5,h=h_8)

"Exergy loss in Expansion device HTC"
X_HT_Expansion=m_dot_2*T_o*(s_8-s_7)


"Total work"
w_total=w_1+w_2

COP_LTC=Q_L/w_1

COP_HTC=Q_M/w_2

"COP of system"
COP=(COP_LTC*COP_HTC)/(1+COP_LTC+COP_HTC)

"Total exergy destruction"
X_total=X_LT_compressorA+X_cascade_condenser+X_LT_Expansion+X_Evaporator+X_LT_c
ompressorB+X_condenser+X_HT_Expansion

"Exergy efficiency"
Eta_exergy=(w_1+w_2-X_total)/(w_1+w_2)
```

EES CODE FOR R23-R600A

```
"input"
T_1=-80
T_3=-5
T_6=25
deltaT=-3
T_o=25
T_F=30
x_1=1
Eta_c=0.7
m_dot_1=1
m_dot_2=1

"LTC"

"Evaporator outletA"
P_1=Pressure(R23,T=T_1,x=x_1)
h_1=Enthalpy(R23,T=T_1,x=x_1)
s_1=Entropy(R23,T=T_1,h=h_1)

Q_L=m_dot_1*(h_1-h_4)

"CompressorA outlet"
P_2=P_3
x_3=0
s_1=s_2s
s_2=Entropy(R23,P=P_2,h=h_2)
P_3=Pressure(R23,T=T_3,x=x_3)
h_2s=Enthalpy(R23,P=P_2,s=s_1)
h_2=(h_2s-h_1)/Eta_c+h_1
w_1=m_dot_1*(h_2-h_1)

"Exergy loss in compressorA"
X_LT_compressorA=m_dot_1*T_o*(s_2-s_1)

"Cascade condenser LTC"
T_5=T_3+deltaT
h_3=Enthalpy(R23,T=T_3,x=x_3)
s_3=Entropy(R23,T=T_3,x=x_3)
Q_M=m_dot_1*(h_2-h_3)

"Exergy loss in Cascade condenser LTC"
X_cascade_condenser=T_o*m_dot_1*(s_3-s_2)+T_o*m_dot_2*(s_5-s_8)

"Expansion device LTC"
P_4=P_1
h_4=h_3
T_4=T_1
s_4=Entropy(R23,P=P_1,h=h_4)

"Exergy loss in expansion device LTC"
X_LT_Expansion=m_dot_1*T_o*(s_4-s_3)
```

```
"HTC"

"Evaporator outletB"
x_5=1
P_5=Pressure(R600A,T=T_5,x=x_5)
h_5=Enthalpy(R600A,T=T_5,x=x_5)
s_5=Entropy(R600A,T=T_5,x=x_5)

"Exergy loss in evaporator"
X_Evaporator=m_dot_1*(s_1-s_4)+m_dot_1*(T_F-T_O)/T_F

"CompressorB outlet"
x_7=0
P_6=P_7
P_7=Pressure(R600A,T=T_6,x=x_7)
h_6s=Enthalpy(R600A,P=P_6,s=s_5)
h_6=(h_6s-h_5)/Eta_c+h_5
w_2=m_dot_1*(h_6-h_5)

"Exergy loss in compressorB"
s_6=Entropy(R600A,P=P_6,h=h_6)
X_LT_compressorB=m_dot_1*T_o*(s_6-s_5)

"Condenser Outlet"
T_7=T_6
h_7=Enthalpy(R600A,T=T_7,x=x_7)
s_7=Entropy(R600A,T=T_7,x=x_7)

"Exergy loss in condenser"
X_condenser=m_dot_2*(s_6-s_7)

"Expansion device HTC"
P_5=P_8
T_8=T_5
h_8=h_7
s_8=Entropy(R600A,P=P_5,h=h_8)

"Exergy loss in Expansion device HTC"
X_HT_Expansion=m_dot_2*T_o*(s_8-s_7)


"Total work"
w_total=w_1+w_2

COP_LTC=Q_L/w_1

COP_HTC=Q_M/w_2

"COP of system"
COP=(COP_LTC*COP_HTC)/(1+COP_LTC+COP_HTC)

"Total exergy destruction"
X_total=X_LT_compressorA+X_cascade_condenser+X_LT_Expansion+X_Evaporator+X_LT_c
ompressorB+X_condenser+X_HT_Expansion

"Exergy efficiency"
Eta_exergy=(w_1+w_2-X_total)/(w_1+w_2)
```

Comparative analysis of Thermodynamics performance of Cascade Refrigeration system for refrigerant Couple R23/R290 and R23/R600A

by Saurabh Anand

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