A DISSERTATION ON

Performance Analysis of a Two Stage Vapour Compression Refrigeration System Utilizing the Waste Heat of the Intercooler for Water Heating

Master of Technology

Renewable Energy Technology

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(2K13/RET/06)

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Candidate's Declaration

I hereby declare that the project entitled **"Performance Analysis of a Two Stage Vapour Compression Refrigeration System Utilizing The Waste Heat of the Intercooler for Water Heating"** being submitted by me is an authentic work carried out under the supervision of **Mr. Naushad A. Ansari, Assistant Professor , Dr. Akhilesh Arora, Assistant Professor & Prof. R.S Mishra,**Department Of Mechanical, Production, Automobile & Industrial Engineering, Delhi Technological University.

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CERTIFICATE

It is certified that Mohit Gupta, Roll No. 2K13/RET/06, student of M.Tech. Mechanical Engineering(Renewable Energy Technology), Delhi Technological University, has submitted the dissertation titled "**Performance Analysis of a Two Stage Vapour Compression Refrigeration System Utilizing The Waste Heat of the Intercooler for Water Heating**" in my guidance towards the partial fulfillment of the requirements for the award of the degree of Master of Technology.

I wish him success in all his endeavors.

Naushad A. Ansari	**Prof. R.S Mishra**	**Dr. Akhilesh Arora**
(Assistant Professor)	(Professor)	(Assistant Professor)

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Generally, individuals set aims, but more often than not, their conquest are by the efforts of not just one but many determined people. This complete project could be accomplished because of contribution of a number of people. I take it as a privilege to appreciate and acknowledge the efforts of all those who have, directly or indirectly, helped me achieving my aim. First of all I express my sincere thanks to my head of the department **Prof. R.S. Mishra** who constantly motivated and supported me throughout the academic period.

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LIST OF SYMBOLS/ABBREVIATIONS

GWP	Global Warming Potential
ODP	Ozone Depletion Potential
COP	Coefficient of Performance
MMVCS	Modified Multistage Vapour Compression System
EDR	Exergy destruction ratio
HP	High Pressure
LP	Low Pressure
$\dot{Q}$	Rate of heat transfer (kW)
$\dot{W}$	Work rate (kW)
$\dot{ED}$	Exergy destruction rate (kW)
$\dot{X}$	Exergy rate (kW)
$\dot{X}^Q$	Rate of thermal exergy flow rate (kW)
$\dot{m}_r$	Mass flow rate of refrigerant (kg/s)
T	Temperature (K)
s	Specific entropy (kJ/kg-K)
h	Specific enthalpy (kJ/kg)
V	Velocity of fluid (m/s)

Greek symbols

η	Efficiency
ε	Effectiveness
x	Specific exergy
δ	Efficiency defect

Subscripts

E	Evaporator
comp	Compressor
c	Condenser
t	Throttle valve
j	j^{th} component of the system
r	Region to be cooled or refrigerant
i	Inlet to the control region
e	Outlet to the control region
R	Refrigerator
O	Ambient state
rev	Reversible
sub	Sub-cooling
su	Superheat
vcr	Vapour compression refrigeration system

ABSTRACT

There is a growing concern on the availability and utilization of energy. Researchers and scientists are focusing on alternative sources of energy and maximum consumption of available energy which also incorporates recovery and use of waste energy. This work focuses on the aspect of use of waste energy in a multi-stage vapour compression system. There is a limitation to the evaporation temperatures in case of a single stage vapour compression refrigeration system. To overcome this limitation along with decreased throttle losses and higher system performance the need of multi-stage vapour compression system arises. In the present work a thermodynamic model for the multi-stage system having an intercooler with waste heat recovery and a flash chamber is developed. A computer program has been made in Engineering Equation Solver to evaluate the performance of a modified multi-stage vapour compression system (MMVCS) .The first and second law analysis is performed with R-717(Ammonia) as a refrigerant with the input parameters of evaporator temperature as -40^0 C and condenser temperature as 40^0 C . The first law efficiency (COP) that can be achieved is approx. 3.24 at a condenser temperature 303 K and evaporator temperature 233 K with effectiveness of intercooler as 0.95. It is also observed that evaporator is the most efficient component in the system as efficiency defect comes out to be least in evaporator whereas throttling valve (corresponding to the higher pressure compressor side) is the worst component followed by the throttling vale of LP side, condenser, compressor, water intercooler and flash chamber. Incorporation of heat recovery through the intercooler proved to be beneficial as the COP of the system is improved by 4-5 % with heat recovery of upto 20 kJ/s with very high tons of refrigeration chosen. The analysis can represent a real system where water can be heated through the heat extracted from the intercooler and it can be used for household purposes and other applications with a reasonable accuracy and is useful for future research work of the whole system.

CHAPTER 1

INTRODUCTION

In the vapour compression cycle, vapour is compressed to a super heated fluid, then cooled and condensed at constant pressure. The refrigerant is then irreversibility throttled to a lower pressure, producing a mixture of liquid and vapour. Finally liquid is evaporated at constant pressure. Deviation from the Carnot for the ideal vapour compression cycle are the irreversible expansion of the liquid refrigerant above its condensing temperature, in addition to sub cooling in the condenser, superheating in the evaporator, friction losses in refrigerant tubing less than 100% efficient compressor, and finite temperature differences across evaporator and condenser surfaces further lower the actual coefficient of performance (COP).

The ratio of the ideal vapour compression COP to Carnot COP depends on the refrigerant, as well as source and sink temperature, but in the range of 70-80%.[1]

In a single stage VCRS basically there are two pressure sides corresponding to evaporator pressure(low) and condenser pressure(high). It is observed that these systems are not suitable with too much difference between the evaporator and condenser temperature. But there are applications in which very high temperature difference is there.. Some of the prominent examples of very low evaporator temperature are frozen food industry(-40^{0}C) and chemical industries(-150^{0}C). Owing the very low evaporator temperatures the difference between condenser and evaporator temperature becomes very high. Similarly in case of heat pumps(used in drying and process heating) the condenser temperature is very high , thereby again increasing the condenser and evaporator temperature difference. With the increase in this temperature difference the single stage system becomes very less efficient and almost impractical. [2]. Therefore multistage systems comes into play for meeting the high temperature differences. The high temperature difference between condenser and evaporator is generally termed as temperature lift.

Multi-stage systems are useful in providing refrigerant at different temperature applications besides the high temperature lift requirements. A typical example is a dairy plant where requirement of refrigerant at different temperature is there(-30^{o}C when ice cream is made and approximately at 2^{o}C for chilling milk). Hence incorporation of two evaporators is beneficial to suffice the low temperature(-30^{o}) and high temperature(2^{0}C) requirements.[7]

Major studies have been carried out for increasing the performance of multi-stage VCRS using thermodynamic analysis but because of irreversibility destruction and losses in components of a system which leads to difficulties in finding the results. [9]

The second law thermodynamic analysis of multi-stage VCRS is commonly used because it gives the idea for enhancement in coefficient of performance due to incorporation of multi-stage compression system. This analysis also provides space for improvement in the existing system. [16]

The performance criteria of a refrigeration system is the coefficient of performance(COP) also known as first law efficiency which is the mathematical ratio of refrigeration effect(heat taken up by the evaporator) to the net input work. The COP of a multi-stage VCRS can be increased either by increasing the numerator(refrigeration effect) or decreasing the denominator(work input) [18].

Most of the industrial and domestic applications use refrigerants that have high GWP and ODP. So the refrigerants that have to be used must be eco-friendly which will not affect the environment in the bad manner [19].

CHAPTER 2

LITERATURE REVIEW

Zubair and Yakub et al. [6] presented on thermodynamic analysis of VCRS is investigated by first and second laws. Thermodynamic Second law analysis is carried out for both multi-stage(two stage) and sub-cooling(mechanical) cycle with refrigerants HFC-134a and CFC-22. The analysis of every component revealed the overall irreversibility losses. According to their findings the interstage pressure(optimum) observed in industry is very close to the geometric mean of condenser and evaporator temperatures for a two-stage system as well as for mechanical sub-cooling cycle. In addition the performance analysis results were compared with the results shon in experiments for the above mentioned refrigerants.

Nikolaidis et al. [7] observed the performance of two-stage compression refrigeration having flash-chamber and water-intercooler using refrigerants R22 has been demonstrated by the thermodynamic analysis. The condenser temperature was varied from 298-308 K whereas the evaporator saturation temperature was varied ffrom 238-228 K. Irreversibility rates were determined. According to him with increasing difference between evaporator and condenser temperature the higher is the irreversibility. Also he inferred that reducing the irreversibility rates in condenser and evaporator contributes towards decreasing the overall irreversibility rate of the whole plant.

Rahman et al. [8] presented the performance of the recently developed integrated space condition system. A conventional split type air conditioner is modified to reclaim the superheated portion of the heat lea ving the compressor to be utilized for the space conditioning purposes. The experimental investigation revealed that his device can cool down the room air temperature to the desired level. by using this type of energy recovery device compressor efficiency can be improved. The end result was expected to be faster cooling and prolonged compressor life.

Cabello and Torrella et al. [9] performed the tests on a single-stage VCRS and used the refrigerants R22, R134a & R407C. The effect of evaporating pressure, the superheating degree at the compressor inlet and the condensing pressure was considered. Hence it was inferred that mass flow rate of the refrigerant is dependent on suction conditions.

Apera and Renno et al. [10] worked on the performance of VCRS using refrigerants R22 and R417a. The plant was used for a cold store applied for preservation of food.. They worked on regulated on/off cycles of the compressors which was operated at the nominal frequency of 50 Hz that was imposed by the classical thermostatic control. The problem or replacement of HCFC was partially solved. In his work, R417a was substituted in place of R22 in the experimental conditions.

Ouadha and En-nacer et al. [11] performed an exergy analysis of a two-stage VCRS and calculated the various exergetic losses. The evaporator temperature was kept constant at -30^{0}C and condensor temperatures were chosen as 30, 40 ,50 and 60^{0}C. The eco-friendly substitutes of R22 chosen were R290 and R717 as working fluids . They inferred that the optimum interstage pressure for a two-stage VCRS is in close proximity to the saturation pressure that corresponds to the arithmetical mean of the condensation and evaporation temperatures.

Arora and Kaushik[12] performed a detailed exergy analysis of a VCRS . A computational thermodynamic model was proposed for calculating the coefficient of performance, exergetic efficiency, exergy destruction and efficiency defects for R404A, R502 and R507A. The range of temperatures chosen for evaporator and condenser temperature were -50^0 C to 0^{0}C and 40^{0}C to 55^0 C respectively. The results clearly showed that R507A is a better substitute to R502 than R404A. The efficiency defect so calculated was found to be highest in the condenser and lowest in liquid vapour heat exchanger for the refrigerants chosen.

Torrella and Llopis et al. [13] performed a second law analysis which was based on experimental data obtained from a two-stage VCRS. Liquid water intercooler and flash chamber were used. The evaporator temperature was varied from -36^0 C to -20^0 C and for condensor temperature range was from 30^0C to 47^0C with R-404a. The results were compared with the previous studies on energy analysis. A new criterion of equivalence between single stage and multi-stage VCRS was established.

Santiago and Jose et al. [14] observed that in case of a two stage VCRS, when the difference between heat source and heat sink is high, higher pressure ratios are required and hence large swept volume for giving the required cooling capacity and refrigeration effect is needed. He worked on a two stage cycle with external sub cooling. COP dependent on intermediate pressure was found out for the standard low temperature conditions operating with R-404a and R-717. Results showed that optimum intermediate pressure is proximate to the arithmetic mean when R-404a is used but there is a significant difference when ammonia is used.

Mishra R.S et al. [15] proposed replacement of R22 with eco-friendly refrigerants. He used refrigerants like R134a, R410a, M20 & R407C. A thermodynamic model was developed to study the cycle. It was inferred that R407C proved to be a potential HFC refrigerant which can replace R22 amongst all the chosen refrigerants with minimum investment and no efforts. It proved to be a non –ozone depleting refrigerant giving high system efficiency.

Aggarwal et al. [16] showed the problems faced in working of different refrigerants due to their environmental impact (R11, R12). Toxicity (NH_3), flammability (HC) and high pressure (CO_2 , which makes them hazardous and harmful in comparison to other working fluids. They observed the performance of different eco-friendly refrigerants and their mixtures in different proportions. They found the effect of working parameters like working temperature & working pressure which affect the coefficient of performance (COP) of vapour compression refrigeration system. Hence they inferred that refrigerants will less ODP and GWP should be substituted with the currently used refrigerants.

Gupta et al.[17] performed a detailed exergy analysis for vapour compression refrigeration cycle using R404A, R407C and R410A. The equations of exergetic efficiency and exergy destruction were developed for the different components in the cycle. They developed an expression for coefficient of performance (COP)of refrigeration cycle and observed effects of evaporator, condenser temperature , degree of sub-cooling and effectiveness of liquid-vapoour heat exchanger on COP.

Mishra R.S.[19] described a thermodynamic modeling of a vapour compression refrigeration system using R134a in primary circuit and Al_2O_3-water based nanofluids in the secondary circuit. In this model certain input variables like geometric characteristics, size of the nanoparticles and the compressor speed were used to calculate the output of the system. This system can be used to design various components like evaporator, condenser and throttling valve for vapour compression refrigeration systems (VCS). Simulation showed that for the same geometric characteristics of the system performance increases from 17% to 20% by application of nanofluid as a secondary fluid in VCS.

CHAPTER 3

VAPOUR COMPRESSION REFRIGERATION SYSTEM

3.1 Introduction

Vapour compression refrigeration systems are most popular and widely used amongst all types of refrigeration systems available. Vapour compression systems are the systems, where the refrigerant suffers phase change and refrigeration effect is produced when the refrigerant fluid evaporates at evaporator temperatures or low temperatures. Energy input to the vapour compression system is the mechanical power needed to run the compressor. Due to this reason, these systems can also be termed as mechanical refrigeration systems. Vapour compression systems comes in all wattages as per the end use requirements with a variety of refrigerants being fed in suiting the requirements. Reverse Rankine cycle forms the basis of actual vapour compression refrigeration cycle.

A Carnot refrigeration cycle which is a completely reversible cycle is used as a base model for a refrigeration system working between two constant temperature limits. It can be used as a standard for other cycles also.

The Coefficient of Performance of a Carnot refrigeration cycle is basically a mathematical function of evaporator and condenser temperatures and has no dependency on the refrigerant used. The Carnot COP is the maximum COP that is possible in a refrigeration cycle. It can be deduced from the Carnot's theorems, that for the similar heat source and sink temperatures, there is no irreversible cycle possible which can have COP higher than Carnot COP.

It can inferred from the above discussion that the Carnot's refrigeration cycle needs to be modified. It is only possible to have a dry compression using a single compression if there is isobaric heat rejection rather than isothermal. On the same lines isenthalpic process of expanison is to be adopted instead of an isentropic one.

A vapour compression refrigeration system can be described as a system working on the principle of reverse heat engine. A compressor, an evaporator, an expansion device and a condenser forms the system as shown in Fig. 3.1.

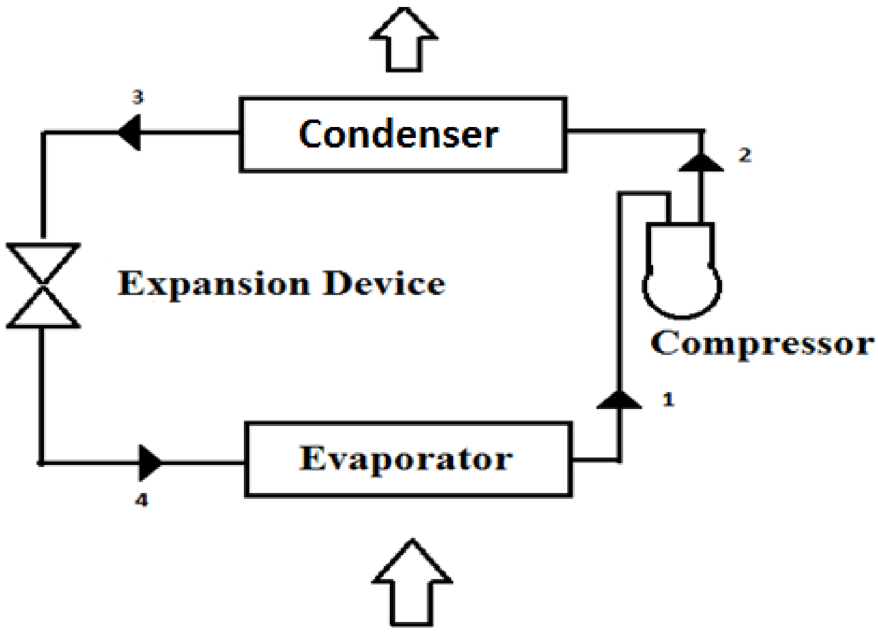

Fig 3.1: Schematic diagram of simple vapour compression refrigeration system

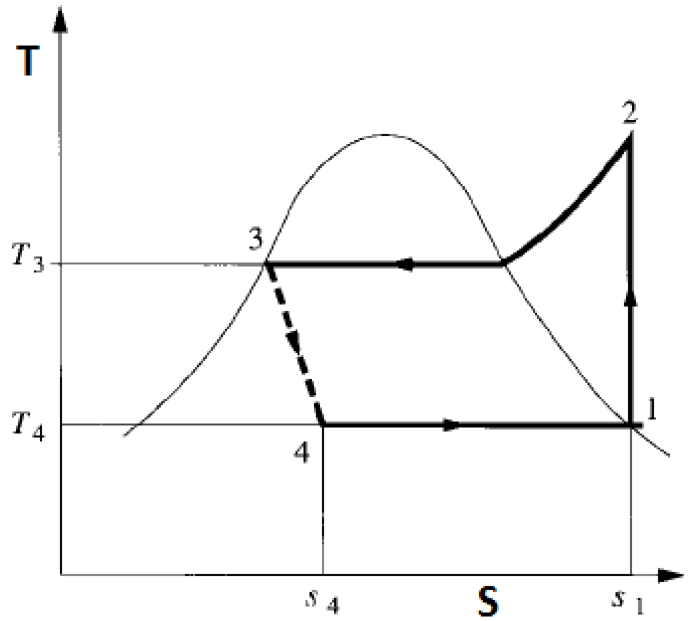

Fig 3.2: Simple vapour compression cycle presented on T-S diagram

The various Processes involved are shown in the figure 1 are as follows:

- 1-2 a compression process which is adiabatic so as to raise refrigerant vapour pressure to the condenser pressure.
- 2-3 a heat rejection process at a constant pressure involving de-superheating and causing the refrigerant to get condensed.
- 3-4 A throttling process which is an isenthalpic one bringing the refrigerant pressure from the condenser to evaporator pressure.
- 4-1 A heat absorption process at constant pressure resulting in the evaporation of the refrigerant.

3.2 Impracticality of the Carnot refrigeration system:

I. Wet compression occurs during the process indicated in the fig 3.1 as 1-2 which ideally has to be an isentropic one.

II. It is very difficult to have wet compression in reciprocating compressors. The severity of the problem rises with high speed compressors (reciprocating) where it can get damaged due to the presence of liquid droplets. Although some type of compressors are able to process and handle the liquid droplets as well but still it is always preferred to have a dry compression rather than wet compression.

III. The other impracticality with the above system extraction of work out of the turbine is not possible economically while the liquid refrigerant expands isentropically especially while using smaller capacities of the system. It can be explained as follows. Since, the specific work output i.e per kilogram of refrigerant extracted out of the turbine is given because the specific volume of liquid is too low when compared to the specific volume of a gas, hence the work obtained from the turbine will be smaller in case of liquid.

IV. Hence multi-stage compression is suggested to achieve dry compression by incorporating two compressors an isothermal and an isentropic.

3.3 Applications of compression Systems:

I. For application that employ lot of thermal energy for their various processes, a large chunk of heat is generally discarded to the surrounding as a waste. This waste heat can be utilized for further process heating of the fluid/ water.

II. For facilities that have a simultaneous requirement for heat and power (cogeneration system), absorption chillers can make use of the thermal energy to produce chilled water.

III. Used as domestic refrigeration in appliances used for maintenance of food in dwelling units.

IV. Used as commercial refrigeration in holding and displaying frozen and fresh food in retail outlets.

V. Used as food processing and cold storage in equipment to preserve process and store food from its source to the wholesale distribution point

CHAPTER 4

TWO STAGE VAPOUR COMPRESSION SYSTEM

4.1 Introduction

We have seen that single stage systems are suitable if the difference between the temperatures of condenser and evaporator is not too high. This difference is normally known as temperature lift. In many cases the requirement is such that either the condenser temperature is large or the evaporator temperature is too low thereby increasing the temperature lift. That means there are many applications where for a given condensing temperature that require very low evaporated temperatures this will increase the temperature lift. . Industries using frozen food have evaporator temperature around -40^{o}C whereas in chemical industries it can be upto -150^{o}C desired for gases to liquefy. The condensing temperature can be really high if the system is used as a heat pump. The system becomes less efficient with the increasing temperature lift.

4.2 Importance of two stage compression system over single stag compression system:

A single stage system has an evaporator side which is basically low pressure side and a condenser side corresponding to high pressure side. The performance of these single stage systems is satisfactory if the temperature lift is not too high as discussed earlier. Therefore a two-stage system is used to make sure the performance of the system doesn’t gets changed or deteriorated. Two stage systems can be multi- compressor system or multi- evaporator system as per the requirement of the system involved. In our case we have a two-compressor system .An LP compressor which compresses the refrigerant coming out of the evaporator and an HP compressor which compresses the refrigerant to further higher pressure which is the condenser pressure. Hence the problem of temperature lift is eliminated in a two-stage system.

4.3 Drawbacks with evaporator temperature increase:

1. There is a certain increase in the throttling losses

ii. Considerable superheat losses show increase

iii. Increase in the discharge temperature of the compressor observed

iv. An increase in the specific volume of the compressor is there

In the light of the above discussion it can be said that there is an increase in the refrigeration effect with increase in work of compression as indicated in P-h diagram (figure-4.2). A decrease in the volumetric refrigeration effect is seen as the specific volume increases with evaporator temperature fall. Almost similar effects will be observed with rising condenser temperatures. Not it becomes quite clear with the above discussion that single stage system becomes inefficient with high temperature lift. Hence multi-stage systems are recommendable.

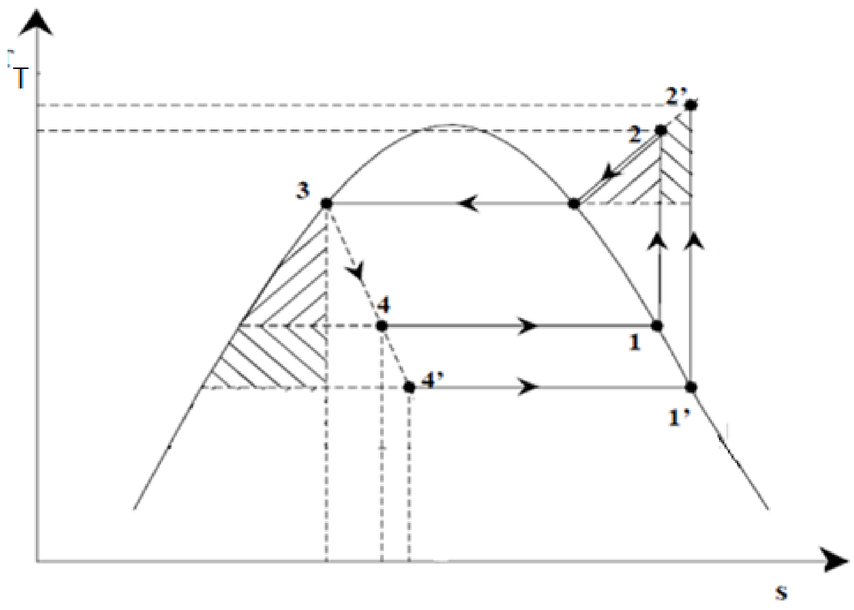

Figure 4.1 : Effect of evaporator temperature on cycle performance (T-s diagram)

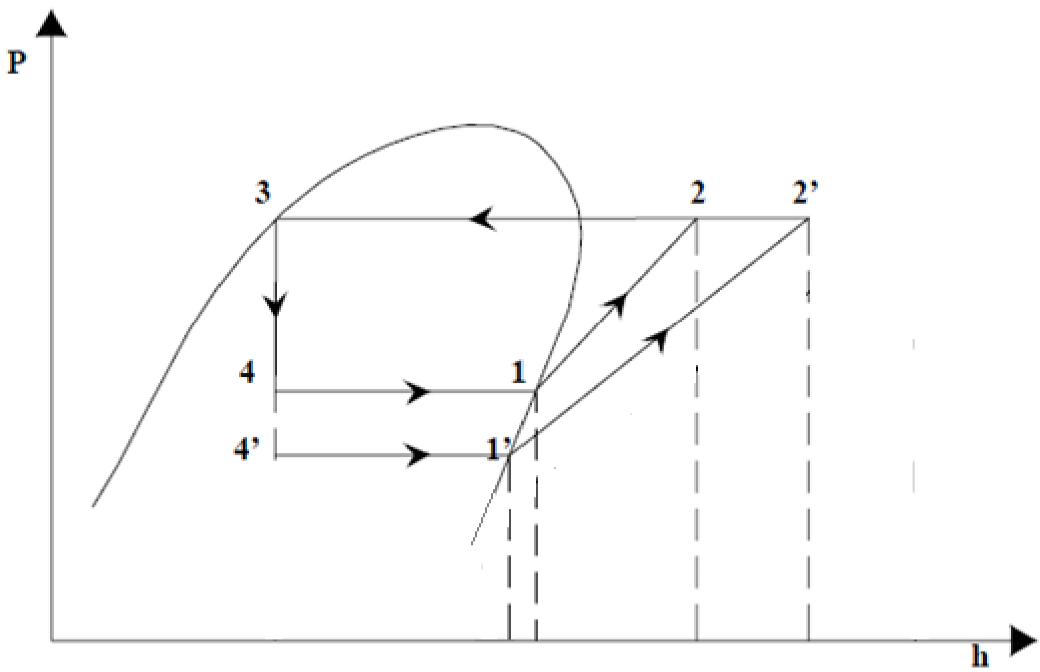

Figure 4.2: Effect of evaporator temperature on cycle performance (P-h diagram)

4.4 Differences between single stage and two stage Vapour compression cycles:

I. High volumetric efficiency as compared to single stage compression system.

II. Less pressure ratio, requiring less displacement volume per ton of refrigeration in comparison to that of single stage vapour compression system.

III. Higher coefficient of performance (COP) achieved than single stage compression system.

IV. Higher refrigerating efficiency as compared to that of single stage compression system.

V. Lesser adiabatic discharge occurs at particular temperature.

VI. There is lesser tendency of fuming and carbonization of lubricating oil operation problem as such in single stage compression system.

VII. Multistage having intercooling decreases the work requirement and multistage expansion with flash chamber increases the refrigeration effect generated.

4.5 Characteristics of multistage vapour compression systems:

The following characteristics of a multi-stage system are discussed:-

i. An LP compressor and a HP compressor.

ii. Series connection of compressors can be there.

iii. Two different refrigeration systems connected together.

iv. The outlet pressure of the LP compressor which should be equal to the inlet pressure of the HP compressor is basically the interstage pressure.

v. The compression ratio corresponding to every stage in a multistage system is less than that seen in a single stage, so high compressor efficiency is seen.

vi. Refrigerant with lower enthalpy (liquid refrigerant) entering the evaporator increases the refrigeration effect.

vii. De-superheating of the discharge vapour/gas from the LP compressor can be done at an inter-stage pressure. This leads to a comparatively lower discharge temperature of the refrigerant from the HP compressor than a single-stage compressor system with same pressure differences between the two temperature limits of condenser and evaporator.

viii. When there is a need of the part-load operation, multistage system(two-three compressors) ensures greater flexibility to adjust the varying refrigeration load. Obviously multi-stage system suffers from a higher cost as compared to the single-stages system.

4.6 Interstage Pressure:

For a two-stage compound system, interstage pressures P_i (kpa absolute), can be calculated as:

$$P_i = \sqrt{P_c * P_e * (\frac{t_c}{t_e})} \qquad (1)$$

Where P_c= condensing pressure (kPa absolute)

P_e= evaporating pressure (kPa. absolute)

t_c= condensing temperature(K)

t_e= evaporator temperature(K)

CHAPTER 5

TWO-STAGE VAPOUR COMPRESSION SYSTEM HAVING FLASH CHAMBER AND WATER INTERCOOLER

5.1 How two-stage compression system works:

It consists of a low pressure (LP) and high pressure (HP) compressor, a water intercooler, two expansion valves, evaporator and condenser. The vapour leaving the low pressure (LP) compressor is intercooled in the water intercooler. After leaving the water intercooler, the vapour enters the flash intercooler where it is cooled by direct contact with the cold liquid refrigerant. The flash intercooler is a pressure vessel in which a constant level of liquid refrigerant is maintained by a float type expansion valve. The pressure in this vessel is intermediate pressure. The cooling of low pressure vapour is done by evaporation of liquid refrigerant in the flash chamber. Compression system employs heat and a refrigerant to produce refrigeration effect. In its simplest design of multi-stage vapour compression system having flash chamber compression system consists of basic components as shown in given figure 4 are as follows:

1. Flash chamber
2. Condenser
3. Evaporator
4. High pressure (HP) compressor
5. Low pressure (LP) compressor
6. Expansion valve
7. Refrigerant

5.2 Flow Processes:

A two stage system with flash chamber and a water intercooler is to be described. The vapour in the superheated state from the heat exchanger which is cooled by water gets bubbled in the flash tank. In this whole process the vapour becomes completely in the saturated state. Although in practical applications complete saturation and de-superheating may not be possible. The incorporation of water cooling and flash tank inter-cooling definitely reduces the amount of vapour generated in the flash tank. Mass and energy balance can be performed to estimate the performance of the system. It is also assumed the potential and kinetic energy changes of refrigerant across each component in the refrigeration system are negligible and flash tank is well insulated.

Inter-cooling of refrigerant vapour using water-cooled heat exchangers is possible in ammonia systems due to high discharge temperature of ammonia. However, this is generally not possible in systems using refrigerants such as R-12 or R-134a due to their low discharge temperatures. In these systems instead of passing the refrigerant vapour from the low-stage compressor through the flash tank, vapour from the flash tank is mixed with the vapour coming from the low-stage compressor. As a result, the inlet condition to the high-stage compressor will be slightly superheated. A two-stage compression system with flash tank for flash gas removal is shown in fig. 4.1.

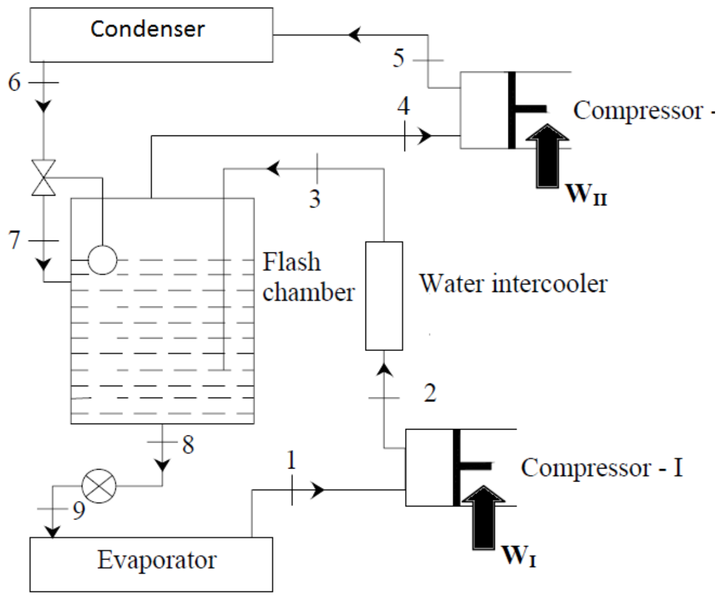

Figure 5.1: Schematic diagram of a two-stage compression system with flash chamber and water intercooler

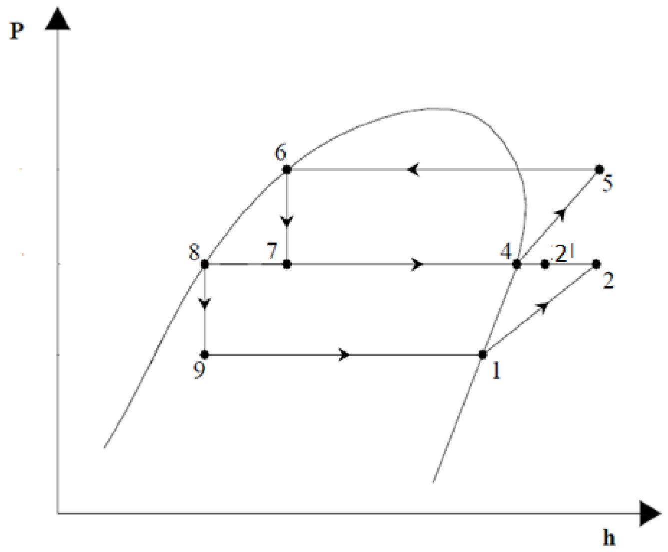

Figure 5.2: Two stage VCRS with flash chamber and water intercooler on P-h diagram

The above system offers several advantages:

I. Quality of refrigerant entering the evaporator reduces thus giving rise to higher refrigerating effect, lower pressure drop and better heat transfer in the evaporator

II. Throttling losses are reduced as vapour generated during throttling

III. Volumetric efficiency of compressors will be high due to reduced pressure ratios

IV. Compressor discharge temperature is reduced considerably

However, one disadvantage of the above system is that since refrigerant liquid in the flash tank is saturated, there is a possibility of liquid flashing ahead of the expansion valve due to pressure drop or heat transfer in the pipelines connecting the flash tank to the expansion device. Sometimes this problem is tackled by using a system with a liquid sub cooler. As shown in above figure in a liquid sub cooler the refrigerant liquid from the condenser is sub cooled by exchanging heat with the refrigerant liquid in the flash tank. As a result, a small amount of refrigerant vapour is generated in the flash tank, which needs to be compressed in the high-stage compressor. Compared to the earlier system, the temperature of refrigerant liquid from the sub cooler will be higher than the saturated refrigerant temperature in the flash tank due to indirect contact heat transfer. However, since the refrigerant at the inlet to the expansion valve is at high pressure and is sub cooled, there is less chance of flashing of liquid ahead of expansion valve.

5.3 Function of Flash Chamber and Intercooler:

Flash chamber

In multi-stage systems or compound systems, flash chambers are used to sub-cool refrigerant to the saturation state which corresponds to the inter-stage pressure by vaporizing a part of the liquid refrigerant in use .In a two-stage refrigeration system with a flash chamber, the part of the liquid refrigerant which is flashed into vapour entering into to the higher pressure suction inlet, so there is less refrigerant which is compressed in the lower stage compressor. in addition, the residual liquid refrigerant is cooled to the saturated temperature matching to the inter-stage pressure, which is too lower in comparison to the sub-cooled liquid temperature in a single-stage system. The enhancement in refrigeration effect and the decrease in compression work input lead to a higher Coefficient of performance than in a single-stage system.

Intercooler

Intercooler is a heat exchanger which brings down the temperature of the refrigerant compressed from a first stage compressor which was initially at higher temperature. An intercooler particularly in case of ammonia based systems is generally incorporate to lessen the temperature thereby ensuring the system works close to saturation vapour lines. The details on the functioning are discussed more in the section 6.7.

CHAPTER 6

THERMODYNAMIC MODELING OF MODIFIED TWO STAGE VAPOUR COMPRESSION SYSTEM

6.1 Introduction

The present study focuses on the waste heat recovery from a two-stage VCR system from intercooler. The purpose of incorporating heat recovery is to further decrease the temperature of refrigerant(ammonia) coming out of the intercooler exit thereby reducing the net work done on the system and hence increasing the overall coefficient of performance of the system.

A computational model was developed for carrying out the analysis of the system using Engineering Equation Solver software. The various properties of the selected refrigerant in the state points are taken from the EES. To analyze the possible realistic performance, a detailed thermodynamic analysis of modified two stage vapour compression refrigeration (VCR) system has been carried out by ignoring the kinetic energy and potential energy change.

6.2 System Description

The present work is on this modified multistage vapour compression refrigeration (MMVCS) system, where heat recovery is done from the intercooler. The removal of heat from the intercooler is carried out by water. It should be noted that. In the thermodynamic cycle, the saturated vapour of the refrigerant coming from the evaporator is drawn by low stage compressor and compressed adiabatically at the expense of input work W_{comp} (process 1-2). The compressed refrigerant is passed through a water intercooler where the temperature is reduced down to the state 3'. Water as a heat recovery medium is passed through this intercooler and it acts as an heat exchanger. The sensible heat removed through this way is $Q_{wic.}$ The compressed refrigerant vapour after passing through the intercooler is directed into the flash chamber (process 3'-4) where saturated liquid and saturated vapour refrigerant at intermediate pressure is separated out, and then it passes through high stage compressor (process 4-5) adiabatically. Then it passes through condenser where it condenses (process 5-6) at constant pressure as a result of the removal of heat of condenser Q_{cond} . The liquid refrigerant coming out from the condenser undergoes adiabatic expansion through high stage expansion valve, accompanied by a drop in pressure at constant enthalpy (process 6-7), after isenthalpic process it passes through again to flash chamber at intermediate pressure (process 7-8). The liquid refrigerant coming from the flash chamber undergoes adiabatic expansion through the low stage expansion valve, accompanied by a drop in pressure at constant enthalpy (process 8-9). This assumption of isenthalpic expansion, rather than isentropic expansion is good from the point of view of practical convenience and economic constraints. However, this may be a cause of energy loss during throttling. The liquid refrigerant under reduced pressure evaporates in the evaporator, thereby absorbing heat Q_{evap} from the space to be cooled. The low pressure vapour from the evaporator is compressed by the compressor and the cycle is repeated.

In order to recover heat from the intercooler, an external fluid (water) is used to remove heat .. A counter-flow arrangement is used . The maximum heat recovery is possible by increasing the tonnage of the system.

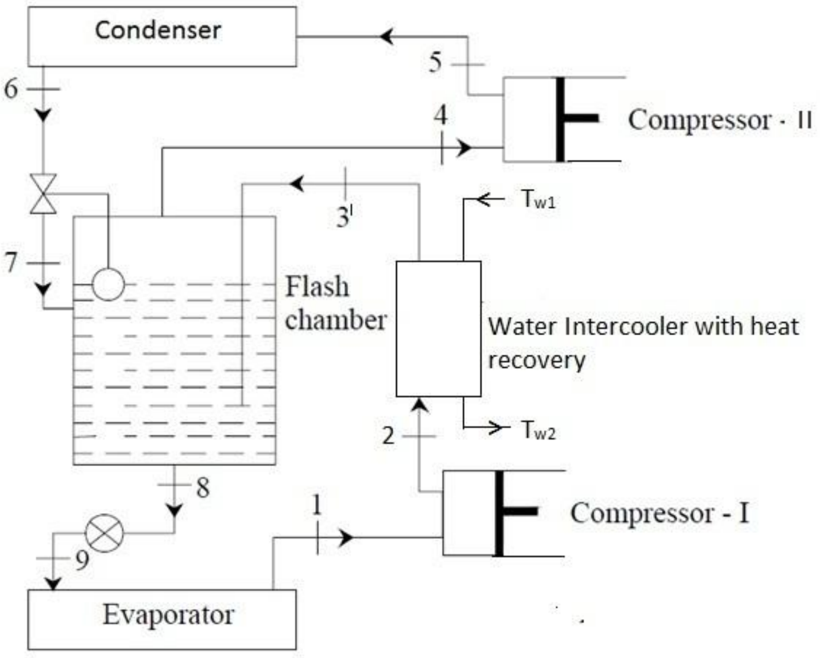

Figure 6.1: Two stage VCRS with waste heat recovery from intercooler using water as heat carrying media.

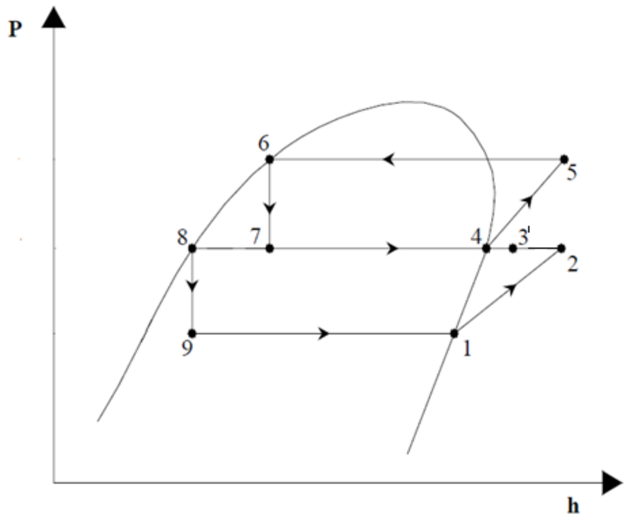

Figure 6.2: Two stage VCRS with waste heat recovery on p-h diagram

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

- *Process* 1-2: Adiabatic compression of the vapour refrigerant at low stage compressor.
- *Process* 2-3’: Heat transfer process in the intercooler.
- *Process 3’-4*: Heat transfer process at an intermediate pressure in the flash chamber
- *Process* 4-5: Adiabatic compression of the vapour refrigerant at high stage compressor.
- *Process* 5-6: Constant pressure heat rejection process in Condenser.
- *Process* 6-7: Isenthalpic expansion process in high stage expansion valve.
- *Process* 7-8: Heat transfer process at intermediate pressure in Flash chamber.
- *Process* 8-9: Isenthalpic expansion process in low stage expansion valve.
- *Process* 9-1: Constant pressure heat absorption process (phase change from liquid to vapour).

6.3 Advantages of the modified VCR system

- The waste heat rejected by the vapour compression system is recovered through the intercooler is utilized in food processing, drying, space heating etc. Heat rejected by the condenser can be recovered to heat the working fluid.
- Recovery of waste heat further contributes in conservation of energy as well as it would be helpful in reduction of global warming.
- Since waste heat is being utilized therefore there will be increase in overall capacity of the system and increase COP.

6.4 Thermodynamic modeling

6.4.1 Energy analysis

The first law of thermodynamics (energy analysis) is related to energy and work losses, while the second law of thermodynamics (exergy analysis) takes entropy into account via irreversibility .

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

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

$$\dot{Q}_i + \dot{W}_i + \sum \dot{m}_i \left(h_i + \frac{v_i^2}{2} + gz_i \right) = \dot{Q}_e + W_e + \sum \dot{m}_e \left(h_e + \frac{\dot{v}_e^2}{2} + gz_e \right) \quad (2)$$

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

- **Evaporator**

  Heat extracted in the evaporator:

$$\dot{Q}_{9-1} = \dot{Q}_e = \dot{m}_{LP} \left(h_1 - h_9 \right) \quad (3)$$

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

$$\dot{m}_{LP} = \mathrm{TR} \left[\frac{3.5167}{h_1 - h_9} \right] \quad (4)$$

Low Stage Compressor

Ideal work input to compressor:

$$\dot{W}_{LP} = W_1 = \dot{m}_{LP} \left(h_2 - h_1 \right) \quad (5)$$

- **Flash Chamber**

  energy balance of the flash tank gives[FIG]:

$$\dot{m}_{HP,1} * h_4 + \dot{m}_{LP} * h_8 = \dot{m}_{HP,1} * h_7 + \dot{m}_{LP} * h_{3,1} \quad (6)$$

Isentropic efficiency of compressor*:* Ratio of ideal (isentropic) work required to the actual work required to do given task. Isentropic efficiency of the higher pressure compressor is considered to be unity. For the lower pressure compressor

$$\eta_{comp,lp} = \frac{W_{isentropic}}{W_{actual}} = \left(\frac{h_3 - h_1}{h_2 - h_1} \right) \quad (7)$$

Please note that state 3' is the state corresponding to the heat recovered intercooler outlet.

$$W_{LP} = \dot{m}_{LP} (h_2 - h_1) \quad (8)$$

$$W_{HP} = \dot{m}_{HP} (h_5 - h_4) \quad (9)$$

$$W_{net} = W_{LP} + W_{HP} \quad (10)$$

$$COP = Q_e / W_{net} \quad (11)$$

$$W_{HP,1} = \dot{m}_{HP,1} (h_5 - h_4) \quad (12)$$

$$W_{net,1} = W_{LP} + W_{HP,1} \quad (13)$$

$$COP_{new} = Q_e / W_{net,1} \quad (14)$$

The actual enthalpy h_4 can be evaluated from the compressor isentropic efficiency, which is defined as the ratio of the isentropic compression work to the actual compression work.

- **Condenser**

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

$$Q_C = Q_e + W_{net} \tag{15}$$

- **Expansion device**

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

  $h_6 = h_7$ (for high stage expansion valve)

  $h_8 = h_9$ (for low stage expansion valve)

Heat removed through the intercooler is given by:

$$Q_{wic} = \dot{m}_w * C_w * \left(T_{w2} - T_{w1}\right) \tag{16}$$

The effectiveness ε_{wic} of the water intercooler with heat recovery can be written as

$$T_3 = T_{3,1} + \varepsilon_{wic} * (T_3 - T_{w1}) \tag{17}$$

6.4.2 EXERGY ANALYSIS

The second law of thermodynamics deduce the exergy, which is very important tool for determing both the quantity and quality of energy consumed. It can be defined as the greatest amount of work accessible when the stream of matter is taken from its initial state to the dead state by the processes in which the stream may interact only with the environment. The exergy balance is quite similar to what is an energy balance but has the basic difference that, while the energy balance is basically a statement of a law of conservation of energy, the exergy may be taken as a statement of law of degradation of energy.

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

Exergy analysis is useful for improving the efficiency of energy-resource use, since it quantifies the locations, types and magnitudes of losses [4].

Exergy balance employed in a steady-state process for a given control region is expressed as

$$\dot{X}_i + \dot{X}_j^Q = \dot{X}_e + \dot{W}_j + \dot{ED}_j \tag{17}$$

$$\dot{X}_i = \sum_{IN} \dot{m}\, x \tag{18}$$

$$\dot{X}_e = \sum_{OUT} \dot{m}\, x \tag{19}$$

$$\dot{X}_j^Q = \sum \left[\dot{Q}_j \frac{T - T_o}{T} \right] \tag{20}$$

$$x = (h - T_o s) - (h_o - T_o s_o) \tag{21}$$

where the first term on left hand and right hand side represent physical exergy (neglecting kinetic, potential and chemical exergy component) of stream of matter entering and leaving the control region respectively. The second term on left hand side and right hand side is thermal exergy flow, which gives exergy transfer rate corresponding to the heat transfer rate $\dot{Q}$ when the temperature at the control surface where heat transfer is occurring is T and exergy associated with work transfer to and from the control region. $\dot{ED}$ represents rate of exergy destruction [5].

6.4.3. Exergy Destruction(ED)

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

- ***Evaporator***

$$\dot{ED}_E = \dot{X}_9 + \dot{Q}_E\left(1 - \frac{T_o}{T_e}\right) - \dot{X}_1 = \dot{m}_{LP}[(h_9 - h_1) - T_o(s_9 - s_1)] + \dot{Q}_E\left(1 - \frac{T_o}{T_e}\right) \quad (22)$$

- ***Compressor-1***

$$\dot{ED}_{comp\ 1} = \dot{X}_1 + \dot{W}_{comp\ 1} - \dot{X}_3 = \dot{m}_{LP}[T_o(s_1 - s_2)] \quad (23)$$

- ***Condenser***

$$\dot{ED}_c = \dot{X}_5 - \dot{Q}_C\left(1 - \frac{T_o}{T_C}\right) - \dot{X}_6 = \dot{m}_{HP,1}[(h_5 - h_6) - T_o(s_5 - s_6)] - \dot{Q}_C\left(1 - \frac{T_o}{T_C}\right) \quad (25)$$

- ***Water Intercooler with heat recovery***

$$\dot{ED}_{wic} = \left(\dot{X}_3 + \dot{X}_{w1} - \dot{X}_{31} - \dot{X}_{w2}\right) = \dot{m}_{LP}\left[(h_3 - h_{31}) - T_o[(s_3 - s_{31})]\right] + \dot{m}_w\left[(h_{w1} - h_{w2}) - T_o\left(s_{w1} - s_{w2}\right)\right] \quad (26)$$

- ***Flash Chamber***

$$\dot{ED}_{flc\,hm} = \left(\dot{X}_{3,1} + \dot{X}_7 - \dot{X}_8 - \dot{X}_4\right) = \dot{m}_{LP}\left[\left(h_{3,1} - h_8\right) - T_o\left[\left(s_{3,1} - s_8\right)\right]\right] + \dot{m}_{HP,1}\left[\left(h_7 - h_4\right) - T_o\left(s_7 - s_4\right)\right] \quad (27)$$

- ***Throttle Valve-1***

$$\dot{ED}_{TV1} = \dot{X}_6 - \dot{X}_7 = \dot{m}_{hp,1}[(h_6 - h_7) - T_o(s_6 - s_7)] \quad (28)$$

- ***Throttle Valve-2***

$$\dot{ED}_{TV2} = \dot{X}_8 - \dot{X}_9 = m_{LP}[(h_8 - h_9) - T_o(s_8 - s_9)] \quad (29)$$

- ***Compressor-2***

$$\dot{ED}_{comp\,2} = \dot{X}_4 + \dot{W}_{comp\,2} - \dot{X}_5 = \dot{m}_{HP,1}[T_o(s_4 - s_5)] \quad (30)$$

6.4.4 Total Exergy Destruction

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

$$\dot{ED}_{Total} = \dot{ED}_E + \dot{ED}_{comp\,1} + \dot{ED}_{comp\,2} + \dot{ED}_{wic} + \dot{ED}_c + \dot{ED}_{TV1} + \dot{ED}_{TV2} + \dot{ED}_{flc\,hm} \quad (31)$$

6.4.5 Exergetic Efficiency

$$\eta_{\text{exergetic}} = \frac{\text{minimum exergy required to do given task}}{\text{actual exergy consumed}} \quad (32)$$

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

$$\dot{X}_E^Q = \dot{Q}_E \left| \left(1 - \frac{T_o}{T_r}\right) \right| \tag{33}$$

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

Hence, exergetic efficiency is given by

$$\eta_{exergetic} = \frac{\dot{Q}_E \left| \left(1 - \frac{T_o}{T_r}\right) \right|}{\dot{W}_{comp}} = \frac{COP_{vcr}}{COP_{rev}} \tag{34}$$

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

6.4.6 Exergy Destruction Ratio (EDR)

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

$$EDR = \frac{E\dot{D}_{total}}{\dot{X}_E^Q} \tag{34}$$

EDR in terms of exergetic efficiency can be written as

$$EDR = \frac{1}{\eta_{exergetic}} - 1 \tag{35}$$

6.4.7 Efficiency Defect $(\boldsymbol{\delta_j})$

That fraction of the input which is lost through irreversibilities in the different components is called efficiency defect. It gives a direct casual relationship between component irreversibilities and their effect on the efficiency of the plant []. It is the ratio between rate of exergy destruction in j-th component to the actual exergy consumed (i.e. actual compressor work) and is given by (2

$$\delta_j = \frac{\dot{ED}_j}{\dot{W}_{comp}} \tag{36}$$

- **Evaporator**

$$\delta_e = \frac{\dot{ED}_E}{\dot{W}_{comp}} \tag{37}$$

- **Compressor-1**

$$\delta_{comp} = \frac{\dot{ED}_{comp\ 1}}{\dot{W}_{comp}} \tag{38}$$

- **Condenser**

$$\delta_c = \frac{\dot{ED}_c}{\dot{W}_{comp}} \tag{39}$$

- **Water Intercooler with heat recovery**

$$\delta_{AHE} = \frac{\dot{ED}_{wic}}{\dot{W}_{comp}} \tag{40}$$

- **Throttle Valve-1**

$$\delta_{t1} = \frac{\dot{E}D_{TV\,1}}{\dot{W}_{comp}} \tag{41}$$

- **Throttle Valve-2**

$$\delta_{t2} = \frac{\dot{E}D_{TV\,2}}{\dot{W}_{comp}} \tag{42}$$

6.5 Simulation study

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

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

6.6 Current applications of refrigeration

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

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

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

6.7 Need For An Intercooler, Benefit of Heat Recovery & Selection of Ammonia as a Refrigerant

It is recommended to cool the lower pressure compressed vapour in a flash intercooler so that the compression process may take place as close as possible to the saturation vapour lines. In case of Ammonia where isentropic lines diverge to a considerable extent and the temperature of the compressed vapour in the first stage becomes much higher than the ambient temperature as compared to R-12 and R-22 [3]. Hence an intercooler can lower down the temperature of the compressed vapour in first stage. It will result in the reduced evaporation of liquid in the flash chamber resulting in lesser mass flow though the HP side. Further recovering the heat from intercooler will help in lowering down the temperature of the vapour coming out of the intercooler. Ultimately, the work done on the HP(2^{nd} stage) compressor will be reduced per unit mass flow through the LP side. Hence heat recovery will be an added advantage to the above system which in turn can be used to heat the water or any other fluid for various applications apart from improving the overall COP of the system. The following table indicates the values of the temperature obtained using EES after first stage compression and after passing through an intercooler with heat recovery for R717 (Ammonia), R-12 and R-22 at Evaporator temperature of -40^0C (233 K) , condenser temperature of 40^0C(313 K), ambient temperature =298 K.

Refrigerant	T_{comp1}(K)	$T_{wic,hr}$(K)
Ammonia	345	294.3
R-12	283.8	287.9
R-22	297.4	287.7

Table 6.1- Showing LP compressor outlet and Water Intercooler outlet temperatures for Ammonia, R-12 & R-22

The results obtained in the above table clearly establish the need of an intercooler for ammonia as the temperature obtained after the first stage compression (T_{comp1}= 345K) is quite high in comparison to ambient temperature(298 K) whereas it would be useless to use an intercooler in case of R-12 and R-22 due to insignificant changes in the results obtained after passing through intercooler with heat recovery.

CHAPTER 7

RESULTS AND DISCUSSIONS

7.1 Results

In order to study the thermodynamic model of a multi-stage vapour compression refrigeration system, a computer program has been developed in Engineering Equation Solver, using certain set of input of parameters.

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

The input data assumed for the calculation of results shown in table 1

1. Refrigerant: Ammonia
2. Effectiveness of the wic (ε_{wic}) : 0.85
3. Condenser temperature(T_c) : $40℃ - 52℃$.
4. Evaporator temperature(T_E) : -40℃ $(fixed)$
5. Isentropic efficiency of compressor: $(\eta_{comp,lp})$ = 0.8
6. Ambient state temperature (T_o) : 298K
7. Water inlet temperature flowing through the condenser (T_{w1}):290K
8. Ambient atmospheric pressure (P_{atm}):101.325Kpa
9. Tonns of refrigeration(TR) :10
10. Specific heat of cooling fluid water (C_{w}) :4.18 Kj/Kg-K
11. It is assumed that pressure drop in evaporator; condenser and liquid auxiliary heat exchanger is negligible.
12. Mass flow rate of cooling fluid (water) is varied from 0.18 to 0.85, while discussing its effect on the system performance.

7.1.1 Effect of condenser temperature at fixed evaporator temperature

T_C [K]	T_e [K]	COP	COP_{new}	$Q_{intercooler}$	ED_{total}	$\eta_{exergetic}$	EDR
313	223	2.037	2.1	2.261	2.518	0.6852	0.2129
314	223	2.01	2.074	2.322	2.565	0.676	0.2169
315	223	1.983	2.048	2.384	2.613	0.6671	0.2209
316	223	1.957	2.023	2.446	2.662	0.6583	0.225
317	223	1.932	1.998	2.508	2.71	0.6497	0.2292
318	223	1.907	1.974	2.57	2.76	0.6412	0.2333
319	223	1.882	1.951	2.632	2.809	0.633	0.2375
320	223	1.858	1.927	2.695	2.86	0.6249	0.2418
321	223	1.834	1.904	2.757	2.91	0.6169	0.2461
322	223	1.811	1.882	2.819	2.962	0.6092	0.2504
323	223	1.788	1.86	2.882	3.013	0.6015	0.2548
324	223	1.766	1.839	2.944	3.066	0.594	0.2592
325	223	1.744	1.817	3.007	3.118	0.5866	0.2637
326	223	1.723	1.797	3.07	3.172	0.5794	0.2682
327	223	1.702	1.776	3.133	3.225	0.5723	0.2727

Table 7.1. Showing variation of COP, COP_{new} ,$Q_{intercooler}$, ED_{total} , $\eta_{exergetic}$ & EDR with condenser temperature T_c at fixed evaporator temperature T_e= 223K

T_C [K]	T_e[K]	COP	COP_{new}	$Q_{intercooler}$	ED_{total}	$\eta_{exergetic}$	EDR
313	233	2.464	2.531	2.015	1.815	0.6874	0.185
314	233	2.428	2.496	2.078	1.854	0.6773	0.189
315	233	2.393	2.462	2.141	1.893	0.6675	0.193
316	233	2.358	2.429	2.205	1.933	0.6579	0.197
317	233	2.325	2.397	2.268	1.973	0.6485	0.2011
318	233	2.292	2.365	2.332	2.014	0.6393	0.2053
319	233	2.26	2.334	2.395	2.055	0.6304	0.2095
320	233	2.228	2.303	2.459	2.096	0.6216	0.2137
321	233	2.197	2.274	2.523	2.138	0.613	0.218
322	233	2.167	2.244	2.587	2.181	0.6046	0.2223
323	233	2.138	2.216	2.652	2.223	0.5964	0.2266
324	233	2.109	2.188	2.716	2.267	0.5884	0.231
325	233	2.081	2.161	2.781	2.31	0.5805	0.2355
326	233	2.053	2.134	2.845	2.354	0.5728	0.24
327	233	2.026	2.108	2.91	2.399	0.5653	0.2445

Table 7.2 Showing variation of COP, COP_{new} ,$Q_{intercooler}$, ED_{total} , $\eta_{exergetic}$ & EDR with condenser temperature T_c at fixed evaporator temperature T_e= 233K

T_C [K]	T_e[K]	COP	COP_{new}	$Q_{intercooler}$	ED_{total}	$\eta_{exergetic}$	EDR
313	243	3.012	3.087	1.849	1.279	0.6816	0.1607
314	243	2.963	3.039	1.914	1.311	0.6705	0.1647
315	243	2.915	2.993	1.98	1.343	0.6597	0.1687
316	243	2.868	2.948	2.045	1.375	0.6492	0.1728
317	243	2.823	2.904	2.111	1.408	0.6389	0.1769
318	243	2.779	2.861	2.177	1.441	0.6289	0.1811
319	243	2.736	2.819	2.243	1.475	0.6192	0.1853
320	243	2.694	2.778	2.31	1.509	0.6097	0.1895
321	243	2.653	2.739	2.376	1.543	0.6005	0.1938
322	243	2.613	2.7	2.443	1.577	0.5914	0.1982
323	243	2.574	2.662	2.51	1.612	0.5826	0.2025
324	243	2.536	2.625	2.578	1.648	0.574	0.207
325	243	2.499	2.59	2.645	1.683	0.5656	0.2115
326	243	2.463	2.554	2.713	1.719	0.5574	0.216
327	243	2.427	2.52	2.781	1.756	0.5494	0.2206

Table 7.3. Showing variation of COP, COP_{new} ,$Q_{intercooler}$, ED_{total} , $\eta_{exergetic}$ & EDR with condenser temperature T_c at fixed evaporator temperature T_e= 243K

7.1.2 Effect of ambient temperature

Tables 4-9 shows the effect of varying ambient temperature 298K -326K on the various parameters corresponding to a particular evaporator and condenser temperature.

T_0 [K]	EDR	$\eta_{exergetic}$
298	0.2129	0.6852
300	0.2094	0.7035
302	0.2061	0.7217
304	0.203	0.74
306	0.2	0.7583
308	0.1972	0.7765
310	0.1945	0.7948
312	0.1919	0.8131
314	0.1895	0.8314
316	0.1871	0.8496
318	0.1848	0.8679
320	0.1827	0.8862
322	0.1806	0.9045
324	0.1786	0.9227
326	0.1767	0.941

Table 7.4. Showing variation of $\eta_{exergetic}$ & EDR with ambient temperature T_0 at T_c=313K & T_e= 223K

T_0 [K]	EDR	$\eta_{exergetic}$
298	0.185	0.6874
300	0.1814	0.7085
302	0.178	0.7297
304	0.1748	0.7508
306	0.1717	0.772
308	0.1689	0.7931
310	0.1661	0.8143
312	0.1636	0.8354
314	0.1611	0.8565
316	0.1588	0.8777
318	0.1565	0.8988
320	0.1544	0.92
322	0.1524	0.9411
324	0.1505	0.9623
326	0.1486	0.9834

Table 7.5 Showing variation of $\eta_{exergetic}$ & EDR with ambient temperature T_0 at T_c=313K & T_e= 233K

T_0 [K]	EDR	$\eta_{exergetic}$
298	0.1607	0.6816
300	0.1569	0.7064
302	0.1534	0.7312
304	0.1501	0.756
306	0.147	0.7808
308	0.1442	0.8056
310	0.1414	0.8304
312	0.1389	0.8552
314	0.1365	0.8799
316	0.1342	0.9047
318	0.132	0.9295
320	0.13	0.9543
322	0.128	0.9791
324	0.1262	1.004
326	0.1244	1.029

Table 7.6. Showing variation of $\eta_{exergetic}$ & EDR with ambient temperature T_0 at T_c=313K & T_e= 243K

T_0 [K]	EDR	$\eta_{exergetic}$
298	0.1746	0.789
300	0.1715	0.81
302	0.1686	0.8311
304	0.1659	0.8521
306	0.1632	0.8731
308	0.1607	0.8942
310	0.1583	0.9152
312	0.156	0.9363
314	0.1538	0.9573
316	0.1518	0.9783
318	0.1498	0.9994
320	0.1478	1.02
322	0.146	1.041
324	0.1442	1.062
326	0.1425	1.084

Table 7.7 Showing variation of $\eta_{exergetic}$ & EDR with ambient temperature T_0 at T_c=303K & T_e= 223K

T_0 [K]	EDR	$\eta_{exergetic}$
298	0.1472	0.8034
300	0.144	0.8281
302	0.1411	0.8529
304	0.1383	0.8776
306	0.1357	0.9023
308	0.1332	0.927
310	0.1308	0.9517
312	0.1286	0.9765
314	0.1264	1.001
316	0.1244	1.026
318	0.1225	1.051
320	0.1206	1.075
322	0.1189	1.1
324	0.1172	1.125
326	0.1156	1.15

Table 7.8 Showing variation of $\eta_{exergetic}$ & EDR with ambient temperature T_0 at T_c=303K & T_e= 233K

T_0 [K]	EDR	$\eta_{exergetic}$
298	0.123	0.8128
300	0.1198	0.8424
302	0.1169	0.8719
304	0.1141	0.9015
306	0.1115	0.931
308	0.109	0.9606
310	0.1067	0.9902
312	0.1046	1.02
314	0.1025	1.049
316	0.1006	1.079
318	0.09877	1.108
320	0.09704	1.138
322	0.0954	1.167
324	0.09383	1.197
326	0.09235	1.227

Table 7.9 Showing variation of $\eta_{exergetic}$ & EDR with ambient temperature T_0 at T_c=303K & T_e= 243K

7.1.3 Effect of variation in effectiveness of water intercooler

COP_{new}	ε_{wic}	Q_{wic}	T_{w2}
2.895	0.25	0.4056	291
2.899	0.3	0.4871	291.2
2.902	0.35	0.5686	291.4
2.905	0.4	0.6502	291.6
2.908	0.45	0.732	291.8
2.911	0.5	0.8139	291.9
2.914	0.55	0.8959	292.1
2.918	0.6	0.9781	292.3
2.921	0.65	1.06	292.5
2.924	0.7	1.143	292.7
2.927	0.75	1.226	292.9
2.931	0.8	1.308	293.1
2.934	0.85	1.391	293.3
2.937	0.9	1.475	293.5
2.94	0.95	1.558	293.7

Table 7.10 Showing variation of COP_{new} , T_{w2} & Q_{wic} with ε_{wic} (effectiveness) at T_c=303K & T_e= 233K

COP_{new}	ε_{wic}	Q_{wic}	T_{w2}
2.483	0.25	0.5851	291.4
2.487	0.3	0.7027	291.7
2.491	0.35	0.8205	292
2.495	0.4	0.9386	292.2
2.499	0.45	1.057	292.5
2.503	0.5	1.175	292.8
2.507	0.55	1.294	293.1
2.511	0.6	1.414	293.4
2.515	0.65	1.533	293.7
2.519	0.7	1.653	294
2.523	0.75	1.773	293.2
2.527	0.8	1.894	294.5
2.531	0.85	2.015	294.8
2.536	0.9	2.137	293.1
2.54	0.95	2.259	294.4

Table 7.11 Showing variation of COP_{new} , T_{w2} & Q_{wic} with ε_{wic} (effectiveness) at T_c=313K & T_e= 233K

COP_{new}	ε_{wic}	Q_{wic}	T_{w2}
2.599	0.25	0.7201	291.7
2.604	0.3	0.8654	292.1
2.609	0.35	1.011	292.4
2.614	0.4	1.158	292.8
2.619	0.45	1.305	293.1
2.624	0.5	1.453	293.5
2.63	0.55	1.601	293.8
2.635	0.6	1.75	293.2
2.64	0.65	1.9	294.5
2.646	0.7	2.051	294.9
2.651	0.75	2.203	294.3
2.657	0.8	2.356	295.6
2.662	0.85	2.51	296
2.668	0.9	2.666	296.4
2.674	0.95	2.822	296.8

Table 7.12 Showing variation of COP_{new} , T_{w2} & Q_{wic} with ε_{wic} (effectiveness) at T_c=323K & T_e= 243K

7.1.4 Effect of variation in inlet water temperature

Q_{wic}	T_{w1}	T_{w2}
1.963	280	284.7
1.877	281.5	286
1.791	283	286.3
1.705	284.5	287.6
1.62	286	287.9
1.534	286.4	291.2
1.448	289	292.5
1.363	290.5	293.8
1.278	292	293.1
1.193	293.5	296.4
1.108	295	297.7
1.023	296.5	297.9
0.9387	298	300.2
0.8543	297.5	301.5
0.7701	301	302.8

Table 7.13 Showing variation of Q_{wic} , T_{w2} with T_{w1} (inlet water temperature) at T_c=303K & T_e= 223K

Q_{wic}	T_{w1}	T_{w2}
2.823	280	286.8
2.739	281.5	286.1
2.655	283	287.4
2.57	284.5	290.6
2.486	286	291.9
2.401	286.4	293.2
2.317	289	294.5
2.232	290.5	295.8
2.148	292	295.1
2.064	293.5	297.4
1.98	295	297.7
1.896	296.5	301
1.813	298	302.3
1.729	297.5	303.6
1.646	301	304.9

Table 7.14 Showing variation of Q_{wic} , T_{w2} with T_{w1} (inlet water temperature) at T_c=313K & T_e= 223K

Q_{wic}	T_{w1}	T_{w2}
2.478	280	285.9
2.383	281.5	285.2
2.288	283	287.5
2.194	284.5	287.7
2.099	286	291
2.005	286.4	292.3
1.911	289	293.6
1.818	290.5	294.8
1.725	292	294.1
1.632	293.5	297.4
1.539	295	297.7
1.447	296.5	300
1.356	298	301.2
1.264	297.5	302.5
1.173	301	303.8

Table 7.15 Showing variation of Q_{wic} , T_{w2} with T_{w1} (inlet water temperature) at T_c=313K & T_e= 243K

Q_{wic}	T_{w1}	T_{w2}
0.7091	0.45	0.05179
0.7075	0.475	0.1926
0.7059	0.5	0.3332
0.7043	0.525	0.4738
0.7028	0.55	0.6143
0.7012	0.575	0.7546
0.6996	0.6	0.8949
0.6981	0.625	1.035
0.6965	0.65	1.175
0.695	0.675	1.315
0.6934	0.7	1.455
0.6919	0.725	1.595
0.6904	0.75	1.735
0.6889	0.775	1.875
0.6874	0.8	2.015

Table 7.16 Showing variation of Q_{wic} , T_{w2} with T_{w1} (inlet water temperature) at T_c=313K & T_e= 243K

7.1.5 Effect of variation in isentropic efficiency of the compressor(LP)

$\eta_{exergetic}$	$\eta_{comp,lp}$	Q_{wic}
0.7091	0.45	0.05179
0.7075	0.475	0.1926
0.7059	0.5	0.3332
0.7043	0.525	0.4738
0.7028	0.55	0.6143
0.7012	0.575	0.7546
0.6996	0.6	0.8949
0.6981	0.625	1.035
0.6965	0.65	1.175
0.695	0.675	1.315
0.6934	0.7	1.455
0.6919	0.725	1.595
0.6904	0.75	1.735
0.6889	0.775	1.875
0.6874	0.8	2.015

Table 7.17 Showing variation of $\eta_{exergetic}$ & Q_{wic} with $\eta_{comp,lp}$(isentropic efficiency) at T_c=313K & T_e= 233K

$\eta_{exergetic}$	$\eta_{comp,lp}$	Q_{wic}
0.6996	0.45	0.1963
0.6983	0.475	0.3149
0.697	0.5	0.4334
0.6957	0.525	0.5517
0.6944	0.55	0.67
0.6931	0.575	0.7882
0.6918	0.6	0.9063
0.6905	0.625	1.024
0.6892	0.65	1.142
0.688	0.675	1.26
0.6867	0.7	1.378
0.6854	0.725	1.496
0.6842	0.75	1.614
0.6829	0.775	1.731
0.6816	0.8	1.849

Table 7.18 Showing variation of $\eta_{exergetic}$ & Q_{wic} with $\eta_{comp,lp}$(isentropic efficiency) at T_c=313K & T_e= 243K

$\eta_{exergetic}$	$\eta_{comp,lp}$	Q_{wic}
0.6268	0.45	0.3307
0.6249	0.475	0.5134
0.623	0.5	0.696
0.6212	0.525	0.8785
0.6193	0.55	1.061
0.6175	0.575	1.243
0.6157	0.6	1.425
0.6139	0.625	1.607
0.6121	0.65	1.79
0.6103	0.675	1.972
0.6085	0.7	2.154
0.6067	0.725	2.336
0.605	0.75	2.518
0.6032	0.775	2.7
0.6015	0.8	2.882

Table 7.19 Showing variation of $\eta_{exergetic}$ & Q_{wic} with $\eta_{comp,lp}$(isentropic efficiency) at T_c=323K & T_e= 223K

7.1.6 Effect of tons of refirigeration

TR	m_{LP}(kg/s)	$m_{HP,1}$(kg/s)	T_{w2}(K)	Q_{wic}(kJ/s)
10	0.0288	0.03599	294.8	2.015
20	0.0576	0.07198	297.6	4.03
30	0.0864	0.108	304.5	6.046
40	0.1152	0.144	307.3	8.061
50	0.144	0.1799	313.1	10.08
60	0.1728	0.2159	317.9	12.09
70	0.2016	0.2519	323.8	13.11
80	0.2304	0.2879	327.6	14.12
90	0.2592	0.3239	333.4	16.14
100	0.288	0.3599	336.2	20.15

Table 7.20. Showing variation of Q_{wic} ,m_{lp},$m_{HP,1}$ & T_{w2} vs TR(tons of refrigeration) at T_c=313K & T_e= 233K, m_w= 0.1 kg/s

7.2 RESULT PLOTS

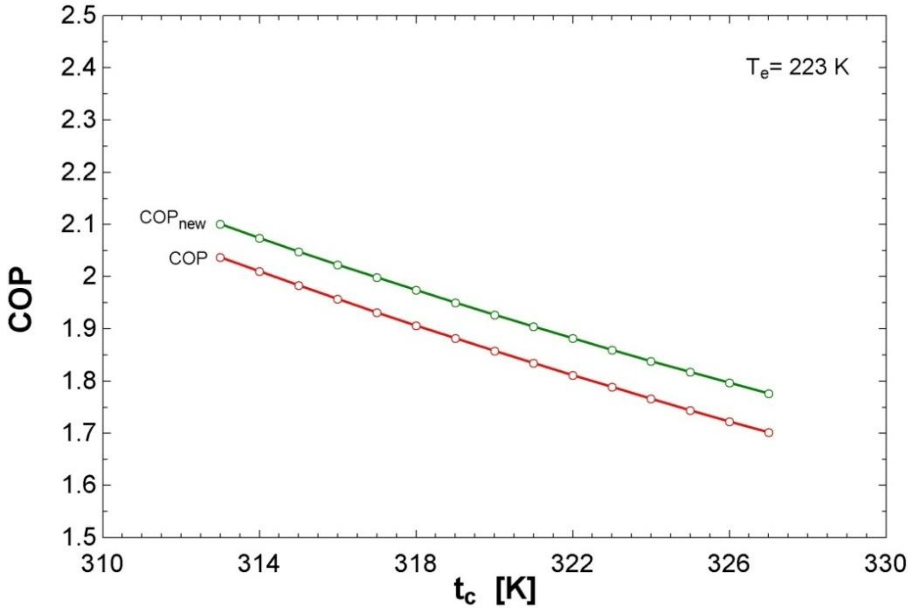

Fig 7.2.1 Showing variation of COPs with condenser temperature(t_c) at T_e=223 K

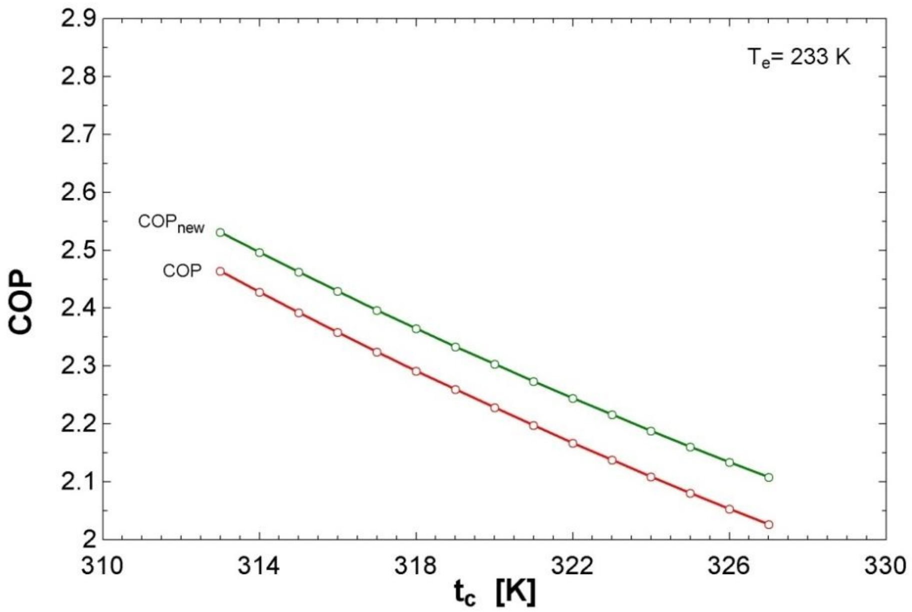

Fig 7.2.2. Showing variation of COPs with condenser temperature(t_c) at T_e=233 K

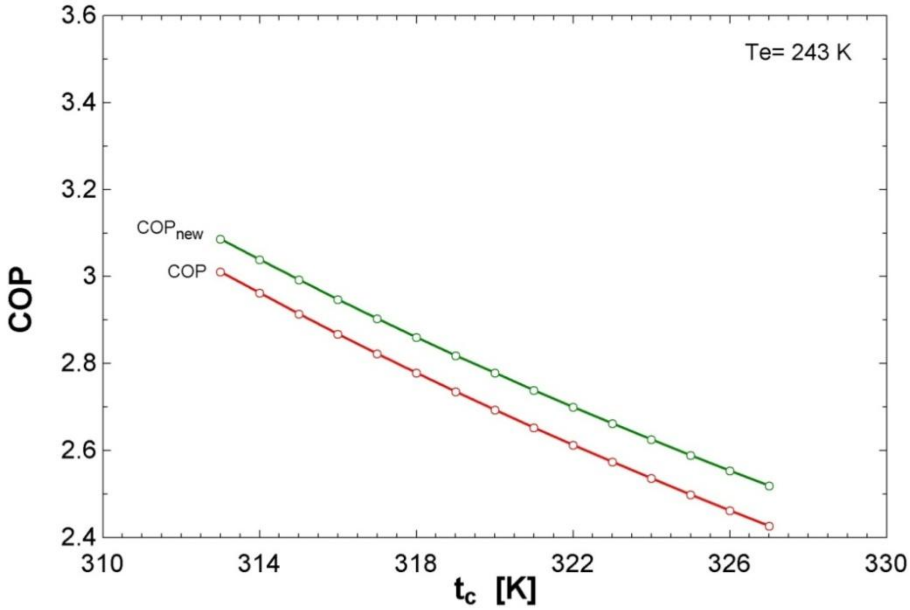

Fig 7.2.3 Showing variation of COPs with condenser temperature(t_c) at T_e=243 K

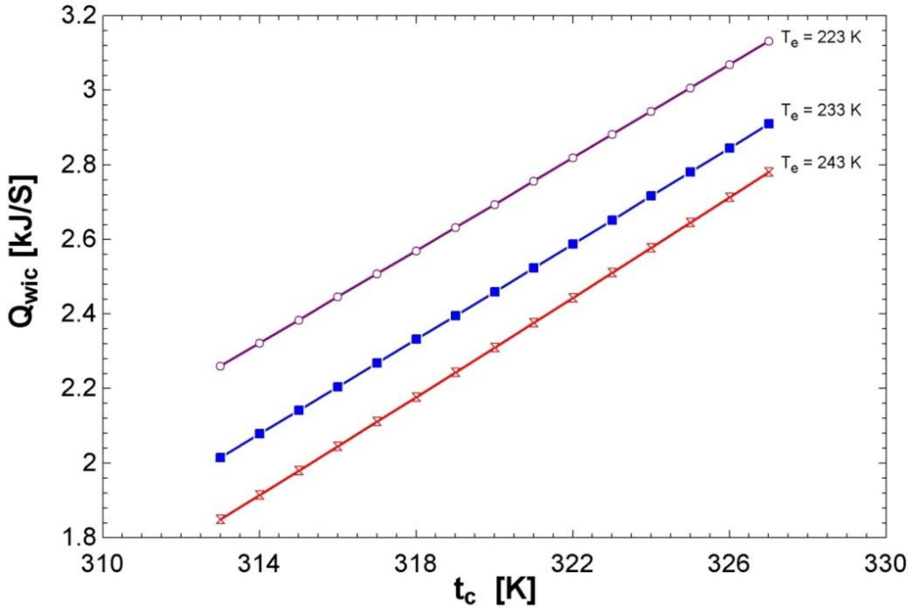

Fig 7.2.4. Showing variation of Q_{wic} with condenser temperature(t_c) at different evaporator temperatures.

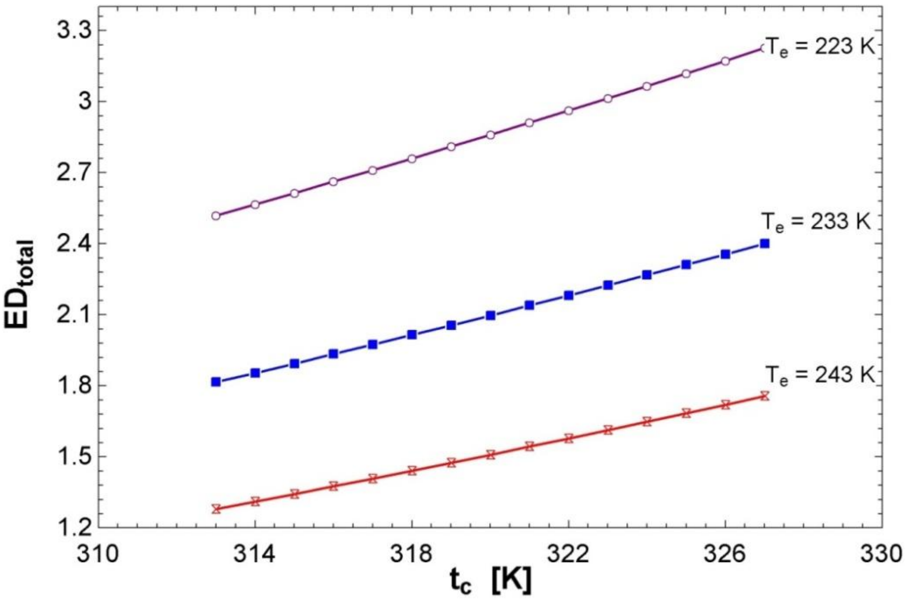

Fig 7.2.5 Showing variation of ED_{total} with condenser temperature(t_c) at different evaporator temperatures.

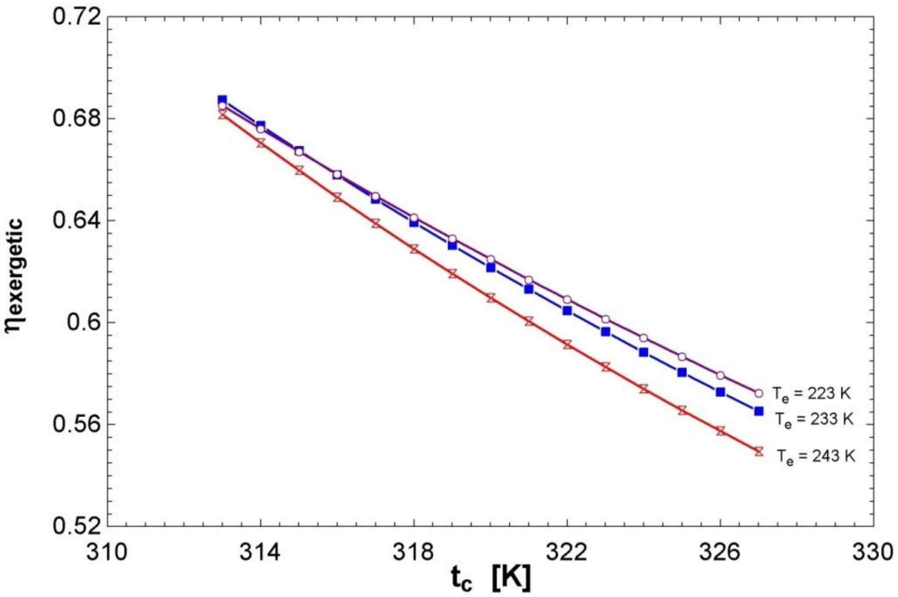

Fig 7.2.6 Showing variation of $\eta_{exergetic}$ with condenser temperature(t_c) at different evaporator temperatures.

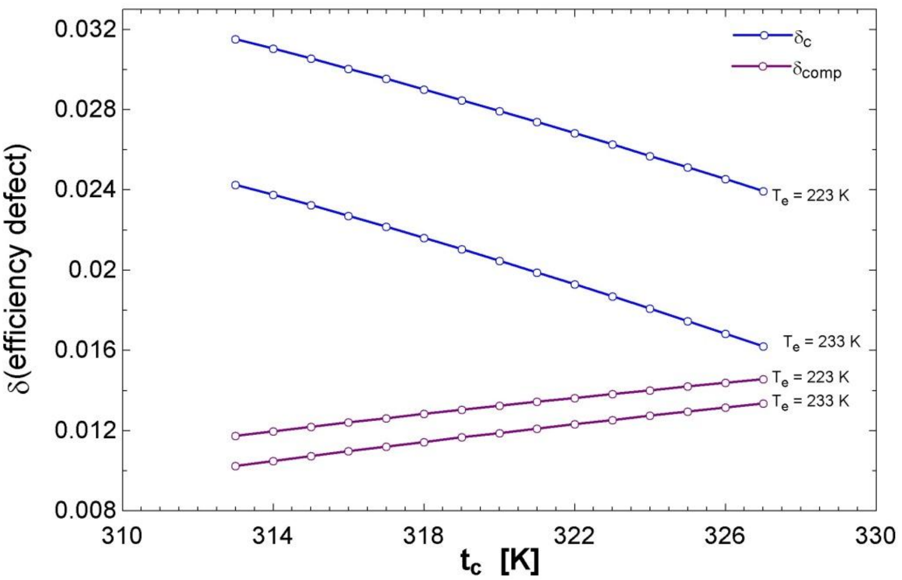

Fig 7.2.7. Showing variation of δ(efficiency defect) with condenser temperature(t_c) at different evaporator temperatures.

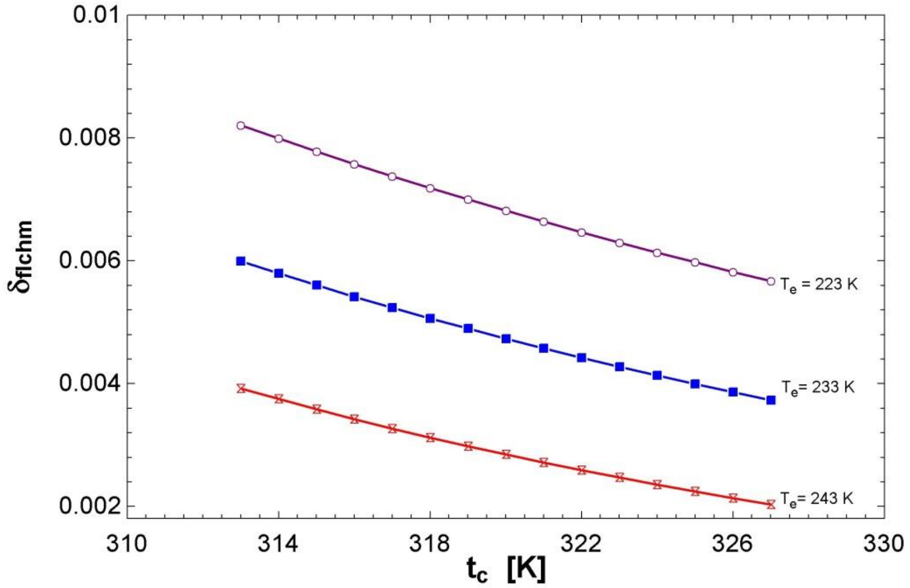

Fig 7.2.8 Showing variation of δ_{flchm} with condenser temperature(t_c) at different evaporator temperatures.

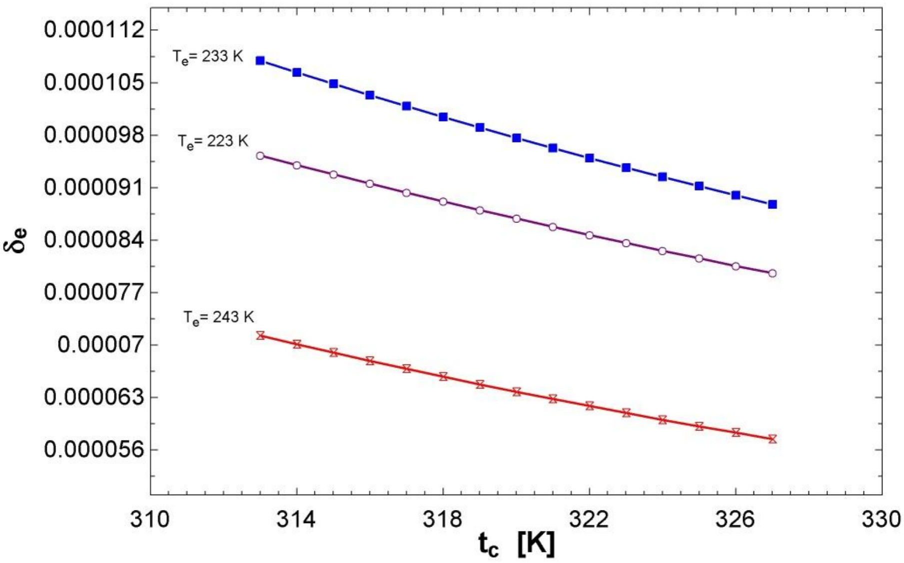

Fig 7.2.9 Showing variation of δ_e with condenser temperature(t_c) at different evaporator temperatures.

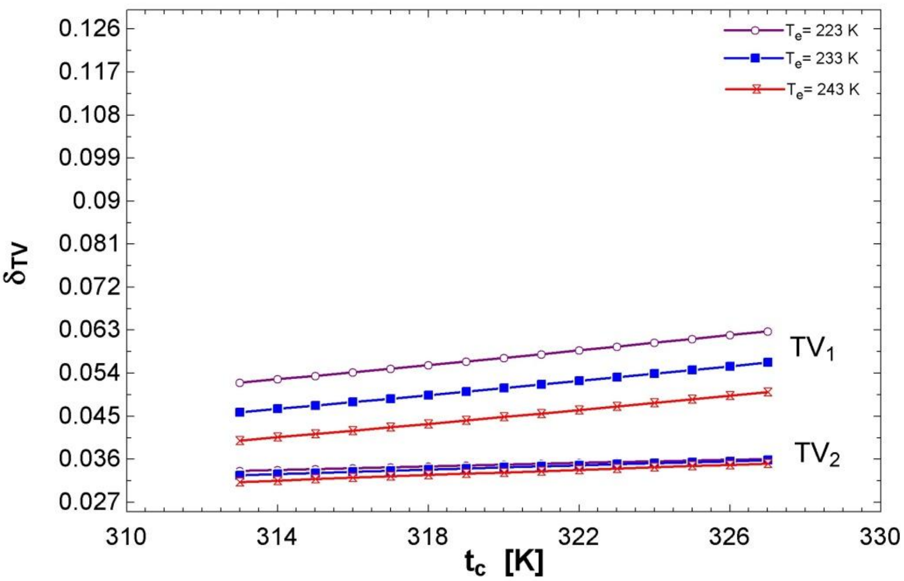

Fig 7.2.10 Showing variation of δ_{TV} with condenser temperature(t_c) at different evaporator temperatures.

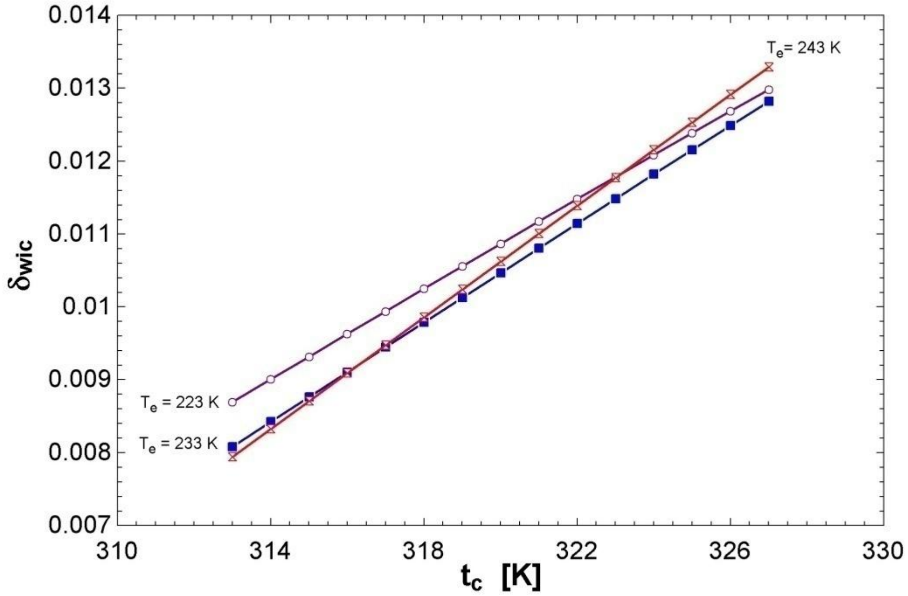

Fig 7.2.11 Showing variation of δ_{wic} with condenser temperature(t_c) at different evaporator temperatures.

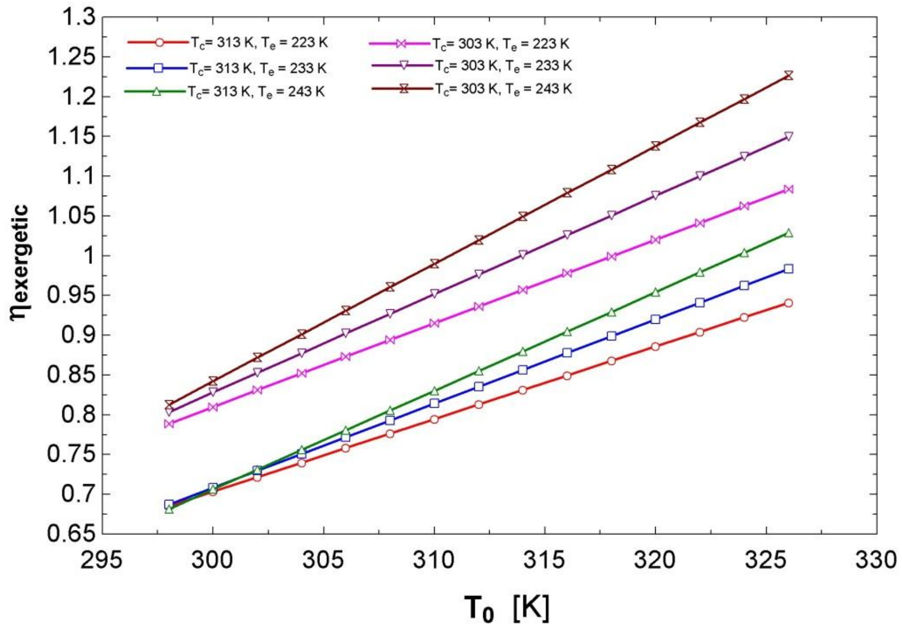

Fig 7.2.12 Showing variation of $\eta_{exergetic}$ with ambient temperature(T_0) at different evaporator & condenser temperatures.

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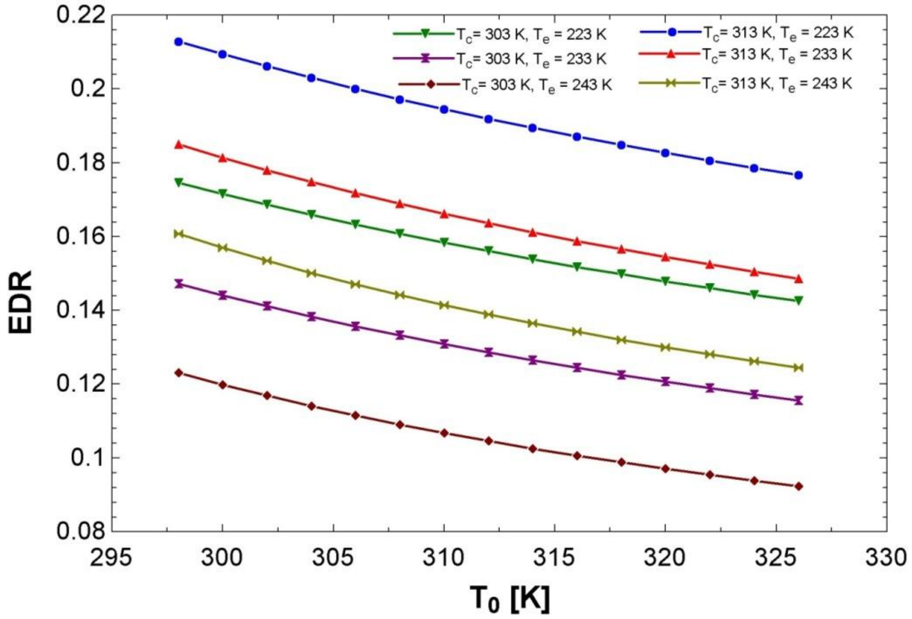

Fig 7.2.13 Showing variation of EDR with ambient temperature(T_0) at different evaporator and condenser temperatures.

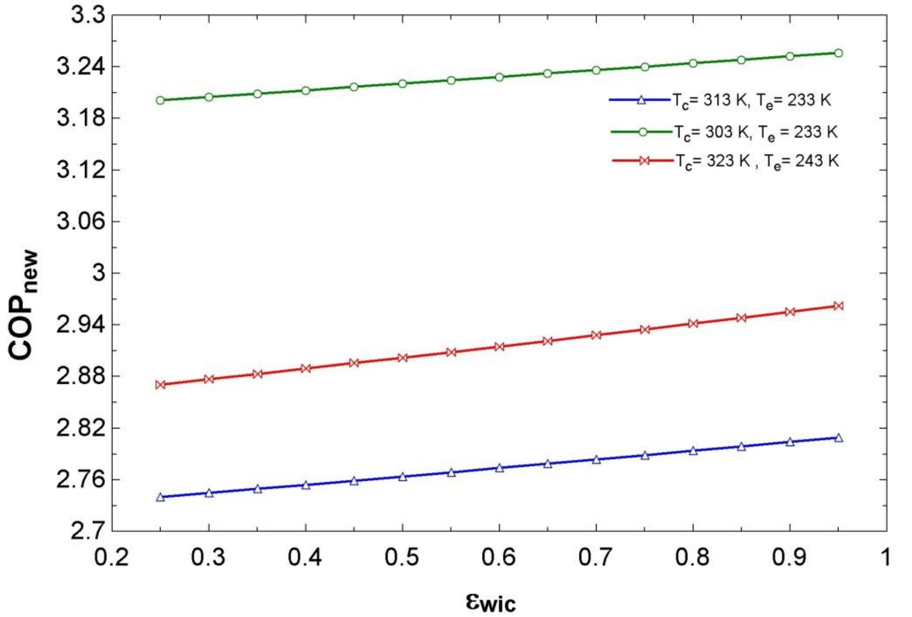

Fig 7.2.14 Showing variation of COP_{new} with ε_{wic} at different evaporator and condenser temperatures.

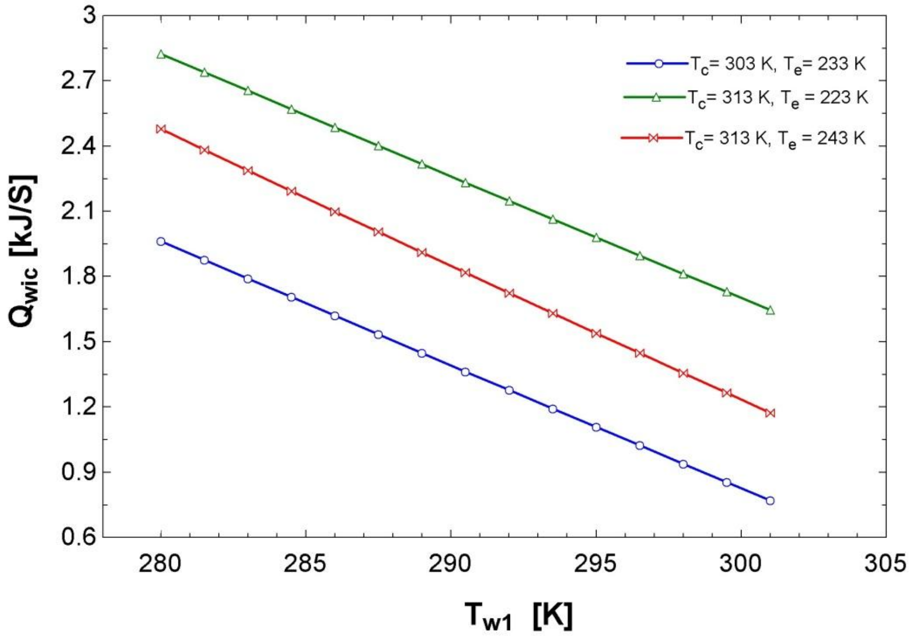

Fig 7.2.15 Showing variation of Q_{wic} with inlet water temperature(T_{w1}) at different evaporator & condenser temperatures.

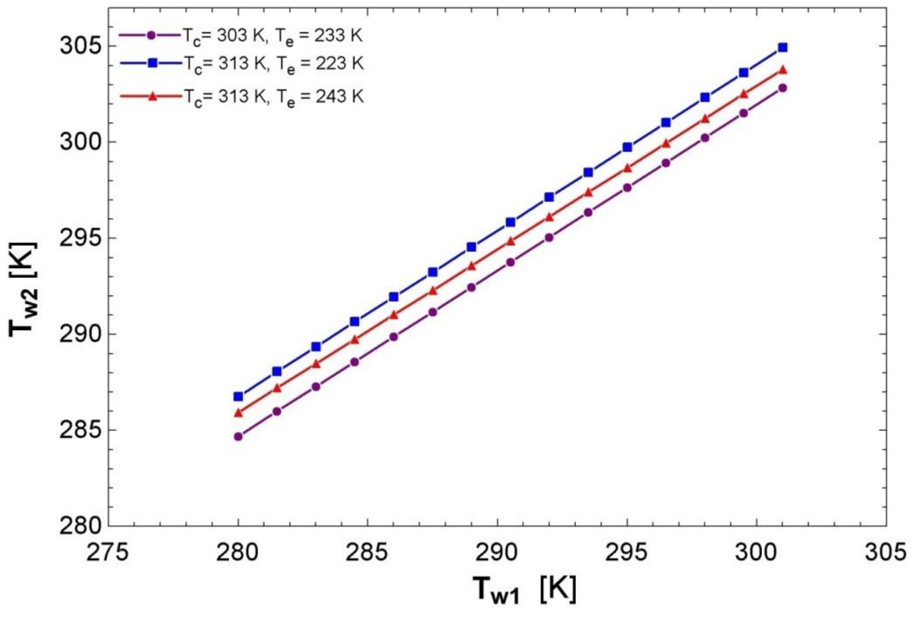

Fig 7.2.16 Showing variation of T_{w2} with inlet water temperature(T_{w1}) at different evaporator & condenser temperatures.

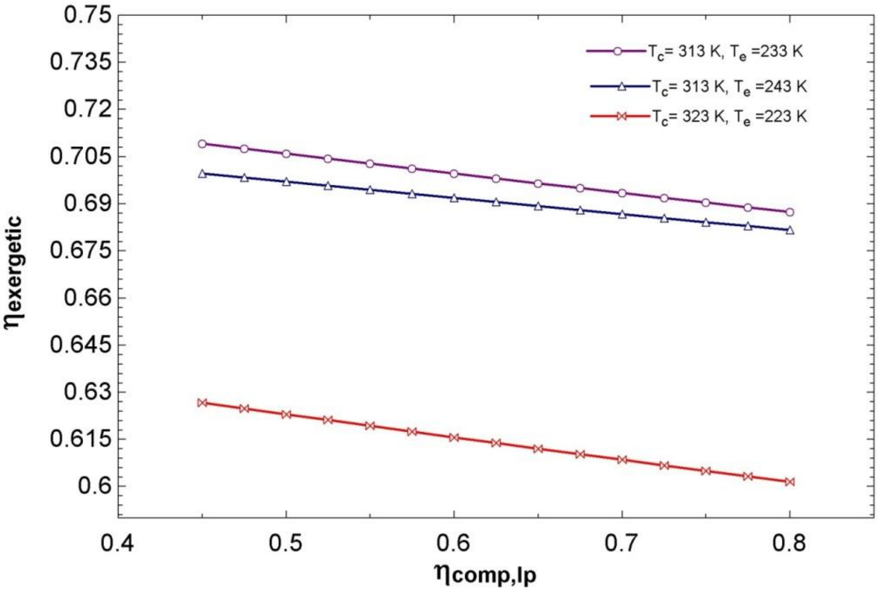

Fig 7.2.17 Showing variation of $\eta_{exergetic}$ with isentropic efficiency of compressor($\eta_{comp,lp}$) at different evaporator & condenser temperatures.

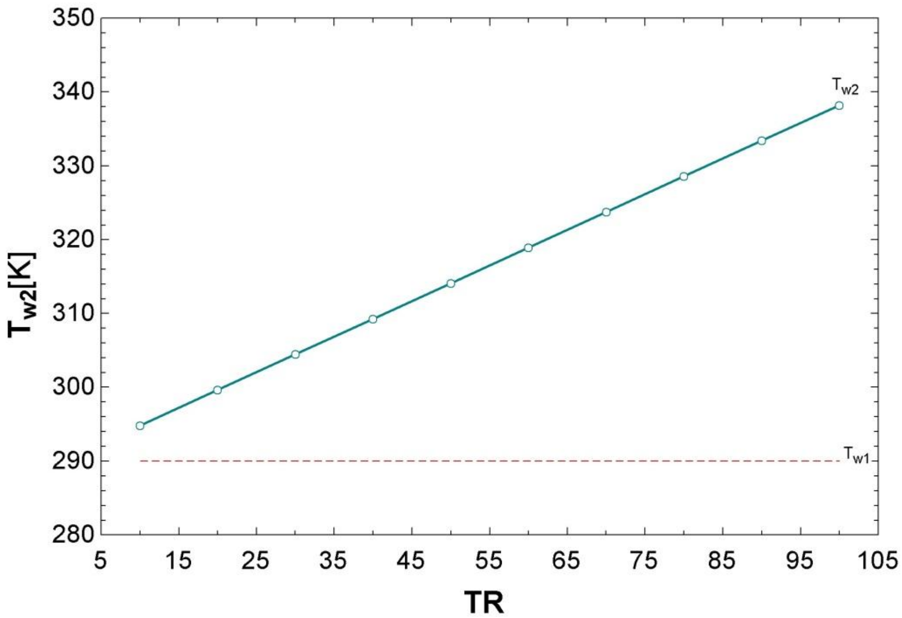

Fig 7.2.18 Showing variation of T_{w2} with tons of refrigeration(TR) at T_C=313 K, T_E=233K,m_w=0.1 kg/s and T_{w1}=290 K

7.3 DISCUSSION

Figure 7.2.1-7.2.3 shows the variation of coefficients of performance before and after incorporating the heat recovery media through the intercooler. The condenser temperature is varied from 313 K to 327 K whereas the evaporator temperature is varied from 233 K to 243 K . It can be clearly inferred that COP decreases with increase in condenser temperature at a fixed evaporator temperature. Also, as the evaporator temperature increases COP increases and it is maximum(3.087) for evaporator temperature,T_e = 243 K and condenser temperature, T_c= 313 K for the selected range of values. It can also be seen that with the incorporation of heat recovery due to decrease in the outlet temperature of intercooler, the mass flow rate on HP side decreases leading to the decrease in W_{HP} and hence increasing the overall COP_{new}. Therefore, COP_{new} is greater than COP for the selected range of condenser and evaporator temperatures in case of Ammonia.

Fig 7.2.4 shows that with increase in condenser temperature. Q_{wic} (heat recovery through water intercooler) increases which is attributed to the fact that since the temperature of the superheated vapour entering the water intercooler increases there is an increase in the outlet water temperature (T_{w2}) . Hence the term (T_{W2}- T_{w1}) in equation for Q_{wic} = m_w*c_w*(T_{W2}- T_{w1}) increases resulting in increase in Q_{wic}. An opposite effect is observed when evaporator temperature decreases .

Fig 7.2.5 shows that as the condenser temperature increases the total exergy destruction increases at a fixed evaporator temperature .Also with the decrease in the evaporator temperature from 243K to 223K the overall variation of ED_{total} shifts to a higher value or in other words it increases with decreases in evaporator temperature. Hence with the increase in the difference between condenser and evaporator temperature the overall exergy destruction (ED_{total}) increase or we can say that irreversibility increases with the temperature lift , hence this result validates (4).

Fig 7.2.6 shows the variation of exergetic efficiency with condenser temperature at fixed evaporator temperatures of 223K, 233K & 243K. As the condenser temperature increases the work done in compression increases hence the term $\dot{W}_{comp}$ increases and exergetic efficiency decreases . It is noticed that with the increase in evaporator temperature there is a decrease in exergetic efficiency which can be accredited to two parameters. First parameter is the thermal exergy flow in the evaporator i.e. $\dot{Q}_e \left|\left(1-\frac{T_o}{T_r}\right)\right|$, with increase in evaporator temperature, refrigerating effect $\dot{Q}_e$ increases, whereas the term $\left|\left(1-\frac{T_o}{T_r}\right)\right|$ decreases since Tr approaches To and the second parameter is compressor work, which decreases with the increase in the evaporator temperature. The effect of $\dot{Q}_e$ and $\dot{W}_{comp}$ is to increase the exergetic efficiency as opposite to the reducing effect of $\left|\left(1-\frac{T_o}{T_r}\right)\right|$. The collective effect of these two parameters is to augment the exergetic efficiency till it reaches the highest point and the evaporator temperature equivalent to this efficiency is optimum evaporator temperature, ahead of which the combined effect is to reduce the exergetic efficiency. Exergetic efficiency has a maximum value of 0.6852 at T_c=313 K,T_e = 223 K.

Fig 7.2.7 -7.2.11 shows the variation of efficiency defect in the various components . Efficiency defect shows the fraction of the input which is lost through irreversibilities in different components of the given system. Thus it helps in finding out the worst component in a given system. Evaporator shows the least efficiency defect . Hence it is the most efficient component in the system whereas throttling valve (corresponding to the HP side) is the worst component as it shows the maximum efficiency defect followed by the throttling vale of LP side, condenser, compressor, water intercooler and flash chamber.

Fig 7.2.12 indicates the variation of exergetic efficiency with ambient state temperature (T_0). Since the term $\left|\left(1-\frac{T_o}{T_r}\right)\right|$increases with increasing T_0 ,therefore exergetic efficiency increases at a given evaporator and condenser temperature. With the increase in evaporator temperature , refrigeration effect $\dot{Q}_e$ also increases ultimately increasing the exergetic efficiency . The variation of exergetic efficiency is shown at two different condenser temperatures 313 K and 303 K whereas evaporator temperature is varied from 223 K to 243 K. Since with increase in condenser temperature work done on compressor $\dot{W}_{comp}$ also increases leading to a decrease in exergetic efficiency which is clearly indicated as the shift in the curve from T_C = 303 K to T_C = 313 k. An opposite trend can be hence observed in fig 7.2.13 where EDR reduces with increasing ambient temperature T_0.

Fig 7.2.14 shows the variation of coefficient of performance of the system with heat recovery (COP_{new}) with effectiveness of water intercooler(ε_{wic}). Since with the increase in ε_{wic} the outlet temperature of the water intercooler increases and as explained earlier is results in increase in the COP_{new} for a given evaporator and condenser temperature. With the increase in condenser temperature from 303 K to 313 K the COP_{new} decreases which can be clearly seen in the plot. It is attributed to the fact that the work of compression increases with increasing condenser temperature leading to the decrease in coefficient of performance . Also with increase in evaporator temperature , refrigerating effect $\dot{Q}_e$ also increases leading to increase in COP_{new} but due to two counteracting effects of increasing condenser temperature (T_c=323 K) in this case and increasing evaporator temperature, the increase in refrigeranting effect is more hence the values of COP_{new} lies in between the values corresponding to T_C = 303 K & 313 K respectively. The maximum value of COP_{new} obtained is approx. 3.24 at T_C = 303 K , T_e = 303 K & ε_{wic} =0.95.

From Fig 7.2.15 and 7.2.16 it can be inferred that with the increasing inlet water temperature to the intercooler (T_{w1}) outlet temperature of the water (T_{w2}) also increases but the difference ($T_{w2} – T_{w1}$) decreases with the increase in T_{w1}. Hence the heat recovery through the intercooler Q_{wic} decreases . Q_{wic} is maximum at T_{w1} = 280 K , T_c= 313 K, T_e = 223 K with a value of 2.83 kJ/s.

Fig 7.2.17 indicates the variation between exergetic efficiency ,$\eta_{exergetic}$ and isentropic efficiency of LP compressor , $\eta_{comp,lp}$. It can be seen that as $\eta_{comp,lp}$ increases $\eta_{exergetic}$ decreases at a given evaporator and condenser temperature. It can be explained as with the increase in $\eta_{comp,lp}$ mass flow rate through HP side is increased leading to more compressor work and less exergetic efficiency.

Fig 7.2.18 shows the variation of intercooler outlet water temperature T_{w2} with tons of refrigeration TR. It can be seen that T_{w2} increases with increase in TR from 10 TR to 100 TR. Table 7.20 shows that at TR =100 heat recovery of 20.15 Kj/s is possible with T_{w2}= 337.2 K.

CHAPTER 8

CONCLUSION

Present study is related to a modified two-stage vapour compression with ammonia as a refrigerant. An extensive energy and exergy analysis has been carried out. Results are carried out by varying evaporator temperature and condenser temperature of a multi-stage (two stage) VCRS with waste heat recovery. Following conclusions can be drawn from the analysis :-

- The COP of the system is improved by 4-5 % with incorporation of heat recovery through intercooler.
- It is seen that COP decreases with increase in condenser temperature at a fixed evaporator temperature. Also, as the evaporator temperature increases COP increases and it is maximum(3.087) for evaporator temperature,T_e = 243 K and condenser temperature, T_c= 313 K.
- It is observed that with the increase in condenser temperature. Q_{wic} (heat recovery through water intercooler) increases.
- With the increase in condenser temperature the total exergy destruction increases at a fixed evaporator temperature.
- Exergetic efficiency decreases with the increase in condenser temperature. Exergetic efficiency has a maximum value of 0.6852 at T_c=313 K,T_e = 223 K.
- Evaporator is the most efficient component in the system whereas throttling valve (corresponding to the HP side) is the worst component followed by the throttling vale of LP side, condenser, compressor, water intercooler and flash chamber.
- With the increase in ε_{wic} there is an increase in the COP_{new} for a given evaporator and condenser temperature. The maximum value of COP_{new} obtained is approx. 3.24 at T_C = 303 K , T_e = 303 K & ε_{wic} =0.95.
- Q_{wic} decreases with the increase in T_{w1}. Qwic is maximum at T_{w1} = 280 K , T_c= 313 K, T_e = 223 K with a value of 2.83 kJ/s.
- With the increase in $\eta_{comp,lp}$, $\eta_{exergetic}$ decreases at a given evaporator and condenser temperature.
- T_{w2} increases with increase in TR from 10 TR to 100 TR. At TR =100 heat recovery of 20.15 Kj/s is achieved with T_{w2}= 337.2 K. This particular result is significant

from the point of view of the application of hot water coming from the intercooler. At lower tonnage this water is useful for household works in winter season and in cold countries. At high tonnage this water is useful for process industries also as difference of the outlet and inlet temperature (T_{w2}- T_{w1}) obtained is approximately 50^{0}C at 100TR.

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