

# “Cascade Refrigeration Cycle”

Major project-II

*Submitted to Delhi Technological University in partial fulfilment of the requirement for the award of Degree of*

**Master of Technology**

In

**Thermal Engineering**

**SUBMITTED BY-**

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# **CERTIFICATE**

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This is to certify that report entitled “**Cascade Refrigeration Cycle**” by **PRAKASH KUMAR** in the requirement of the partial fulfilment for the award of Degree of **Master of Technology (M.Tech)** in **Thermal Engineering** at **Delhi Technological University**. This work was completed under my supervision and guidance. He has completed his work with utmost sincerity and diligence. The work embodied in this project has not been submitted for the award of any other degree to the best of my knowledge.

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## **DECLARATION**

I declare that the work presented in this thesis titled “Cascade Refrigeration Cycle”, submitted to Department of Mechanical Engineering, is an authentic record of my own work carried out under the supervision of asst. prof. Dr. Akhilesh Arora, Department of Mechanical Engineering, Delhi technological university, Delhi.

This report does not, to the best of my knowledge, contain part of my work which has been submitted for the award of any other degree either of this university or any other university without proper citation.

Date:

Place: DTU, Delhi

Signature of candidate

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## **ABSTRACT**

A thermodynamic analysis is done for cascade cycle which is a combination of vapour compression system with vapour absorption system. The objective of the work is to determine the effects of different parameters on the performance of the cycle. Equations used follow the first law of thermodynamics. Capacity of the cascade cycle is taken as 50 kw. Refrigerants used in the cycle are lithium bromide-water in the absorption cycle and R134a in the vapour compression cycle. EES has been used to find out the results through doing simulation in it. The result of the simulation is to study the relationship between different components of the cycle such as generator, condenser, heat exchanger evaporator on the cop of the cycle. The outcome can be used to enhance the performance and determine the parameters for better working of the cascade cycle.

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## NOMENCLATURE

<b>LiBr</b>	<b>Lithium bromide</b>
<b>H<sub>2</sub>O</b>	<b>Water</b>
<b>VARs</b>	<b>Vapour absorption refrigeration system</b>
<b>VCRS</b>	<b>Vapour compression refrigeration system</b>
<b>T<sub>c</sub></b>	<b>Condenser temperature</b>
<b>T<sub>g</sub></b>	<b>Generator temperature</b>
<b>T<sub>e</sub></b>	<b>Evaporator temperature</b>
<b>T<sub>a</sub></b>	<b>Absorber tempera</b>
<b>M</b>	<b>Mass of refrigerant</b>
<b>Q<sub>a</sub></b>	<b>Heat rejected in absorber(Kw</b>
<b>Q<sub>g</sub></b>	<b>Heat supplied in generator(kW)</b>
<b>Q<sub>e</sub></b>	<b>Refrigerating capacity(kW)</b>
<b>Q<sub>c</sub></b>	<b>Heat rejected in condenser(kW)</b>
<b>X<sub>w</sub></b>	<b>Mass fraction of lithium bromide in the weak solution</b>
<b>X<sub>s</sub></b>	<b>Mass fraction of lithium bromide in the strong solution</b>
<b>M<sub>w</sub></b>	<b>Mass flow rate of the weak solution (kg/s)</b>
<b>M<sub>s</sub></b>	<b>Mass flow rate of the strong solution (kg/s)</b>

## SYMBOLS

<b>Letter</b>	<b>Description</b>	<b>Unit</b>
<b>M</b>	<b>Mass of refrigerants</b>	<b>kg/s</b>
<b>h</b>	<b>Enthalpy</b>	<b>kJ/kg</b>
<b>P</b>	<b>Pressure</b>	<b>KPa</b>
<b>T</b>	<b>Temperature</b>	<b>K</b>
<b>Q</b>	<b>Heat Energy</b>	<b>kW</b>
<b>e</b>	<b>Effectiveness</b>	

# CHAPTER 1

## INTRODUCTION

In today's world more emphasis is given to use clean and efficient source of energy. So the technologies are developed to use less amount of energy. One such method is to use the combination of vapour compression system with vapour absorption systems. These system uses the advantages of both the refrigeration cycles. To make this system more energy efficient waste heat from power plants and renewable sources of energy is used [10]. Further we will see that how this technology is used and effects of this on saving the energy. In order to understand the cascade refrigeration cycle of vapour absorption and vapour compression we first need to understand both of these cycles individually.

### 1.1 ABSORPTION SYSTEM

The major input to the system is in the form of low grade heat. Absorption system has the advantage of the using clean source of energy for cooling purpose because of the use of negligible amount of electricity in the pump compared to vapour compression system. In a LiBr absorption cycle the refrigerant used is water and LiBr<sub>2</sub> which is salt so there isn't any adverse effect on environment. This can be used in food industry to preserve food. Use of natural refrigerant makes it environment friendly. Use of pump to elevate the refrigerant solution pressure instead of a compressor makes it free from unnecessary vibrations and noise.

#### 1.1.1 BASIC PRINCIPLE OF OPERATION

When a solute is dissolved in a solvent the boiling point of the solution is increased. In a basic absorption system two vessels are connected to each other. One vessel contains pure refrigerant or the solvent and the other vessel contain the solution that is the mixture of refrigerant and the adsorbent. Therefore, at equilibrium when the pressure is same, pure refrigerant temperature will be lower than that of LiBr<sub>2</sub> water solution. So if the solution is at surrounding temperature then the refrigerant will be

at lower temperature than the surrounding. This temperature difference between the refrigerant and surrounding causes the refrigeration effect. By manipulating pressure by supplying heat or rejecting heat we can transfer vapour from pure refrigerant side to solution side during refrigeration process and solution side to refrigerant side during regeneration process. But refrigeration happens only in one step and is not continuous. we have to apply high heat during the regeneration process. Heat is rejected during both refrigeration process and regeneration process. We need to develop a continuous refrigeration system. These is done by using two pairs of vessels. In addition to that a pump and expansion valve is used. Basic working of vapour absorption system is explained below.

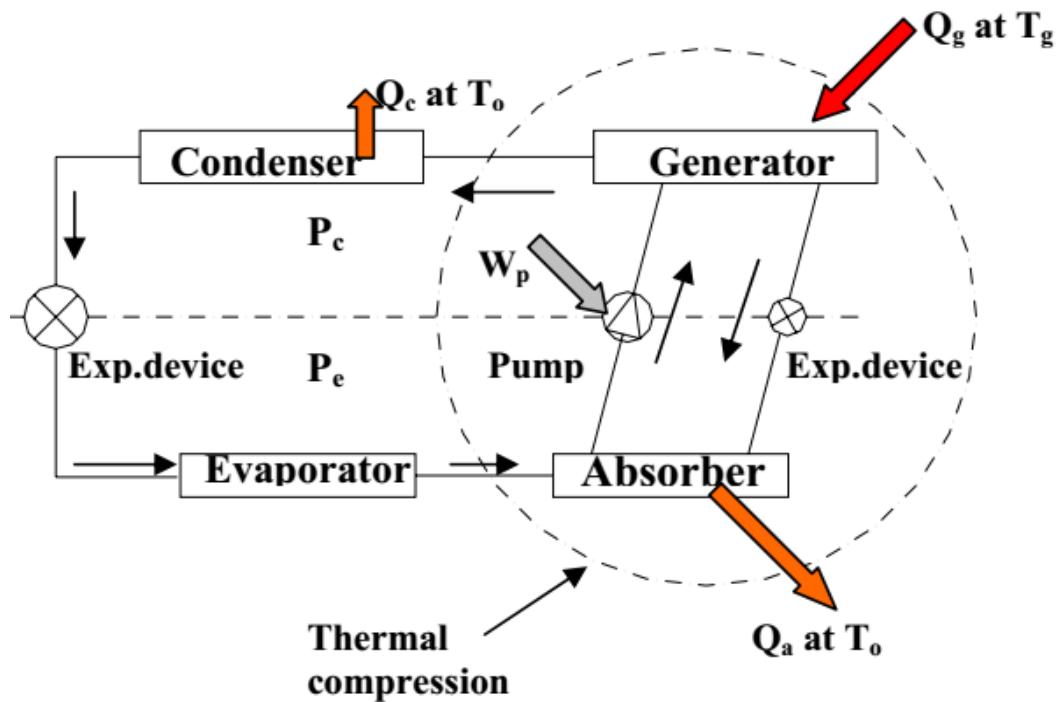


Figure 1.1 Vapour absorption system



### 1.1.3 WORKING

From the evaporator a low pressure and low temperature vapour refrigerant goes to the absorber. In the absorber the refrigerant comes in contact with the solution weak in refrigerant from the generator. The refrigerant vapour gets absorbed in the absorber. This process is exothermic and rejects heat  $Q_a$  at atmospheric temperature  $T_0$ . After mixing the solution becomes rich in refrigerant. Rich solution gets pumped into the generator. This compresses the solution. In the generator high heat is supplied. Rich solution gives the refrigerant in vapour form in generator. This refrigerant vapour goes into condenser. In the condenser the vapour converts into liquid. This liquid refrigerant expands in the expansion device. This low temperature and pressure liquid enters the evaporator. In the evaporator this liquid absorbs heat from the surrounding and refrigeration takes place.

### 1.1.4 REFRIGERANTS USED AND PROPERTIES

Most commercially and practically used refrigerant absorbent mixture pair are

- $\text{LiBr}_2$  pair but this has a disadvantage that since water is a refrigerant so it cannot be used for refrigeration processes below  $0^\circ\text{C}$ . Lithium bromide is salt and acts as an absorbent.
- Ammonia water pair. Ammonia is a refrigerant here and water acts as an absorbent.

Some of the desirable properties of the refrigerants are

Solubility of the refrigerant and the solution in the absorber should be high.  
Deviation in the Raoult's law should be negative.

1. Refrigerant and absorbent boiling point difference should be high so that in the generator only refrigerant should be converted into vapour.
2. It should exhibit low viscosity and thermal conductivity should be high.
3. Crystallisation and solidification should be avoided.
4. It should be (a) safe

(b)chemically stable

(c)non corrosive (d) easily available

### 1.1.5 COP OF THE VAPOUR ABSORPTION SYSTEM

The cop of the vapour absorption system is given by =

$$\frac{\text{Refrigerant effect}}{\text{energy supplied}}$$

### 1.2 VAPOUR COMPRESSSION SYSTEM

Most commonly used refrigeration system. 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 is a pure substance therefore the temperature remains constant during the vaporisation process. The basic vapour refrigeration cycle diagram is given below.

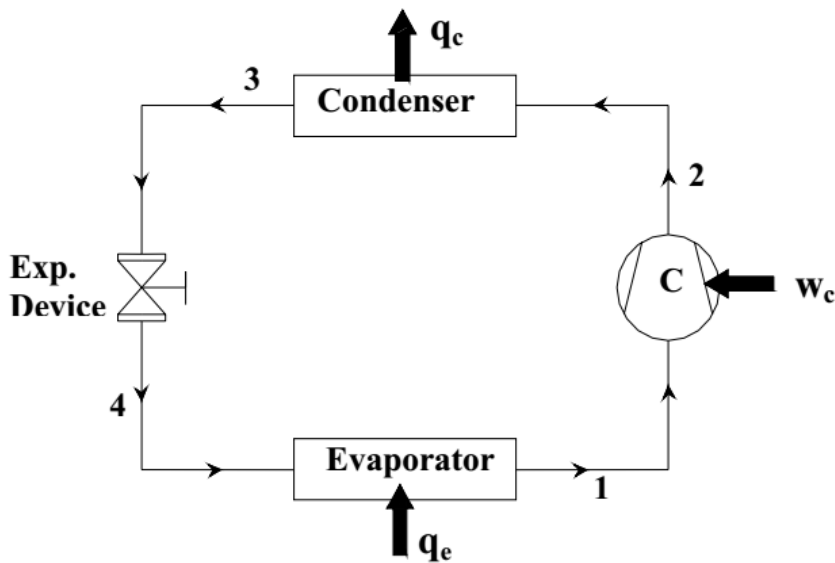


Figure 1.2 Vapour compression system

### **1.2.1 WORKING**

As shown in figure a mechanical input of work is given through a compressor. The compressor compresses the vapour into a high temperature and pressure gas. The compression is assumed to be isentropic. 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 high pressure and high temperature. In the condenser the vapour refrigerant condenses into liquid. It rejects the heat,  $Q_c$  into the surrounding. The liquid refrigerant then enters into the refrigerant expansion valve. In the expansion valve the liquid refrigerant is throttled and pressure is reduced to the pressure of the evaporator. In the evaporator again the phase changes. The liquid refrigerant takes the heat,  $Q_e$  from the surrounding and converts into vapour. This low temperature and pressure vapour then again enters into the compressor and the cycle continues.

### **1.2.2. COP OF THE VCRC**

Cop= refrigeration effect/work input =  $Q_e / W_c$

### 1.3 CASCADE REFRIGERATION CYCLE

A single stage vapour absorption and a single stage compression vapour compression system is used. The condenser of the vapour compression system rejects the heat to the evaporator of the VARS. In the vapour compression side of the cascade cycle we use R134a as a refrigerant. The refrigerants get compressed in the compressor and goes into the condenser. The condenser rejects the heat, this rejected heat gets absorbed by the evaporator of the VARS side of the cascade cycle. Hence in further calculation we will see that the values heat rejected and heat accepted is same. The refrigerant goes to the REV and then to the evaporator of the VCRS. The cycle continues in VCRS while in VARS the heat rejected by condenser<sub>1</sub> evaporates the refrigerant in VARS. The refrigerant used here is water. The LiBr and water forms a solution. This solution has different enthalpy at different concentration of LiBr<sub>2</sub> at respective temperature and pressure. The refrigerant gets into absorber. Here it mixes with LiBr solution and gets pumped into generator. The effectiveness of the cycle is increased by adding a heat exchanger to it. From the generator adding heat drives the refrigerant away from the LiBr solution. It goes to condenser and then to the REV. From there it goes to the evaporator and the cycle gets completed in the VARS. Different sets of refrigerants can be used like aqua ammonia with r134a, ammonia and r410a or LiBr water with r134a, ammonia and r410a [1]. For better understanding below a figure 1.3 is drawn of the cascade cycle.

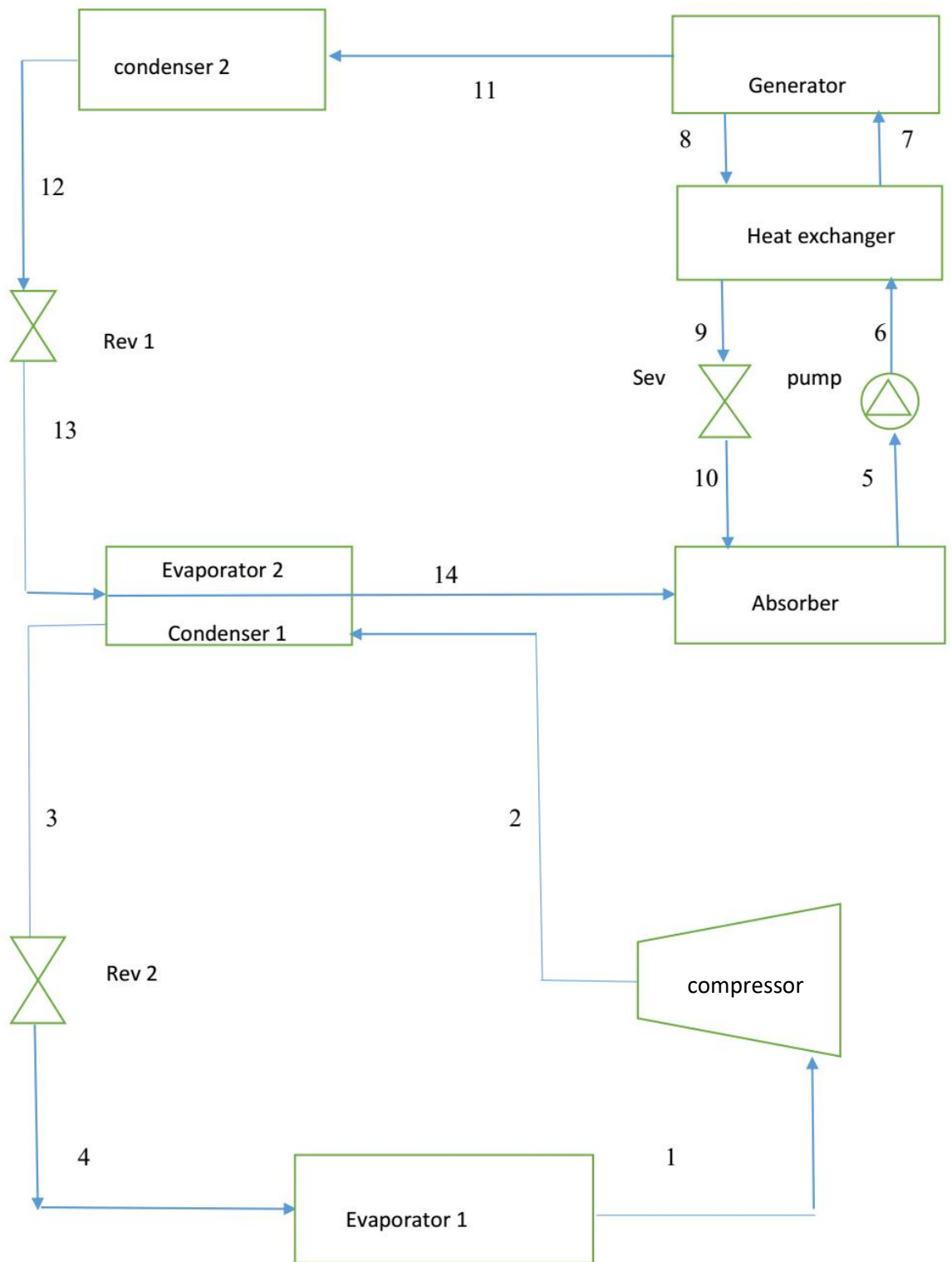


Figure 1.3 cascade refrigeration cycle

## **CHAPTER 2**

### **LITERATURE REVIEW**

Conservation of energy is a big concern in today's world. Every necessary and innovative steps are tried in order to lower the energy consumption and use it more efficiently. For a conventional refrigeration cycle like VCRS when compared to a cascade cycle containing both VCRS and VARS, a cascade cycle uses 48-51% less electricity. LiBr-H<sub>2</sub>O pair was used for the first time in cascade cycle and was compared with different set of refrigerants [1].

Performance analysis is done by simulation with different working fluid. Performance is enhanced when LiBr-H<sub>2</sub>O was used in absorption cycle and ammonia, R134a and carbon dioxide in compression cycle. The cop of the system was found to be higher than the VCRS using the same refrigerant in the refrigeration cycle [2]. Different type of waste heat recovery technologies available on board ships from the perspective of technical principle and application feasibility is discussed. Waste heat can be used in cascade cycle for better performance. Waste heat from turbine is used for absorption system. To compress the ammonia vapour from evaporator to condenser mid temperature waste heat was used [3-4].

Little and Garimella [5] assessed for waste heat recovery applications based on vapour compression or expansion. Waste heat at 60 °C and 120 °C is taken for two case small scale lower temperature and higher scale higher temperature respectively. Also case study is done for waste recovery data from vehicles, process plants to utilize primary source input.

Electricity consumption is reduced to 61 % for a capacity of 66.67 kw. A thermodynamic model was developed to achieve -40 °C for high heat flux application with lithium bromide water absorption system and carbon dioxide system coupled together. While comparing it with conventional vapour compression system the cascade system saves 31% electricity [6-7].

Traditional absorption refrigeration system can save energy by using the low-grade heat but its efficiency is low, especially when the temperature of the heat source is low. It was also reported that the COP of an absorption refrigeration cycle decreased rapidly with the decrease in evaporating temperature [8]

The vapour compression refrigeration system usually has higher efficiency, whereas it consumes electric energy. Therefore, absorption compression cascade refrigeration systems were recommended by many studies [9-10]. In 1993, Chinnappa et al. [9] investigated an absorption compression cascade refrigeration system used for air conditioning and found that with the assistance of absorption subsystem, the COP of compression subsystem increased from 2.55 to 5, while the power consumption of compression subsystem decreased from 4.35 kW to 2.2 kW. An energy and economic analysis of an absorption compression refrigeration system for supermarket applications was presented by Marimona et al. [10]. The investment cost of the cascade system was higher than that of a traditional refrigeration system. The shortest payback time was 4.6 years.

Kairouani et al. [11] studied the performance of compression absorption refrigeration (cascade) cycles and  $\text{NH}_3/\text{H}_2\text{O}$  fluid pair was used at the absorption section of the refrigeration cycle and three different working fluids (R717, R22, R134a) were used at the vapour compression section. He concluded that the coefficient of performance of the cycle is 37-54% higher than the vapour compression cycle using R717, R22 and R134a refrigerants for the same operating conditions.

Seara et al. [12] analysed a cascade refrigeration cycle with compression system at low temperature stage and absorption system at high temperature stage. He has also tries to find the possibility of powering it by the means of cogeneration system

The studies above shared the same principle and their cycles were similar. Chen et al. [13-14] proposed a novel absorption compression cascade refrigeration system based on a different principle. In Chen's system, the refrigerant in the compression subsystem was condensed by the ambient air or water first. Then, the condensed refrigerant was super cooled by an absorption subsystem. The

advantage of the consequence was that the high-temperature cooling capacity provided by the absorption subsystem (low-grade cooling capacity,  $Q_{lg}$ ) was upgraded to the low-temperature cooling capacity of the compression subsystem (high-grade cooling capacity,  $Q_{hg}$ ). It was called the principle of cooling capacity upgrade [15].

Chen et al. [16] then developed an absorption compression solar heat pump and refrigeration system based on the principle. Experimental results showed the COP of its compression subsystem increased from 5.52 to 6.32 at the evaporating temperature of 4.4 C, as the generating power of the absorption system increased from 0 to 2750 W.

Cimsit et al. [17] presented the thermos economic optimization of LiBr/H<sub>2</sub>O-R134a compression-absorption cascade refrigeration cycle. The analysis pointed out that the evaporator and solution heat exchanger should be designed carefully according to the exergy economic factor values. Xu et al. [18] developed a novel absorption-compression cascade refrigeration system, which can reach an evaporating temperature of 170 C. In the cascade system, the refrigerant of compression subsystem (CS) is subcooled by the refrigerant of low-grade heat driven absorption subsystem (AS) to reduce the electric power consumption. Theoretical and experimental investigations were carried out over the CS evaporating temperature ranging from 100 to 170 C.

Lee et al. [19] studied the optimal condensing temperature and maximum coefficient of performance for carbon dioxide-ammonia cascade refrigeration cycle by means of a thermodynamic analysis. Bhattacharyya et al. [20] presented a cascade cycle operating with propane in the low cycle and carbon dioxide in the trans critical cycle. They demonstrated that the carbon dioxide propane cascade system offers a wider operating range for simultaneous heating and cooling applications compared with that provided by the carbon dioxide ammonium cascade system. This cascade system can operate simultaneously with a refrigerating space temperature of 233 K and a heating output temperature of about 393 K.



Colorado et al. [21] compared the performance of NH<sub>3</sub>, CO<sub>2</sub>, butane and propane as a working fluids in the two-stage cascade compression system for simultaneous cooling and heating. According to the authors, R134a refrigerant at low temperature cycle and carbon dioxide at high temperature cycle configuration seems to be a better alternative for industrial applications in comparison with the other cycles evaluated.

On the other hand, double-stage absorption systems have been studied by several authors. Arun et al. [22] presented the performance analysis of a double-effect in series LiBr/H<sub>2</sub>O absorption system in which all the vapour generated at the high pressure generator is fully condensed at the low pressure generator.

Garousi et al. [23] carried out first and second law analyses for series, parallel and reverse parallel double-effect absorption refrigeration cycles and compared the results. The COP and exergetic efficiency for the parallel and reverse parallel systems were similar and higher than those of the series system. The authors also reported the crystallization of LiBr/h<sub>2</sub>o solution risk, it's most likely to take place in the weak solution entering the absorber caused by lower temperature and highest concentration. However, Gebreslassie et al. [24] presented a study based on second law of thermodynamics for the diverse double-effect cycles and found that the exergy destruction in the generators of the different configurations are similar for the system in series and reverse flows, but higher for the system in parallel.

Jain et al. [25] addressed the size and cost estimation of vapour compression absorption cascade system for water chilling application taking R410a and water-LiBr in compression absorption system respectively. The result showed that the cascading of compression absorption systems becomes attractive for lower rate of interest and increase life span and operational period.

Colorado and Velazquez [26] developed a thermodynamic model for a compression absorption system for refrigeration. Carbon dioxide, R134a and ammonia were used as working fluids in the low temperature cycle and LiBr/H<sub>2</sub>O was used in the high temperature cycle. Regarding to second laws and exergy

analysis, in the refrigeration field, there are analysis based on energy and exergy of a single and double effect absorption cycle [27-29]

Herold et al. [30] studied the analysis of hybrid compression absorption cycle using LiBr-H<sub>2</sub>O as the working fluid. In this analysis steam compressor was used for improving performance of absorption cycle and all of the cycle used water (steam) as refrigerant. Due to high cost and low isentropic efficiency of the steam compressor this cycle analysis resulted in poor economics.

Ayala et al. [31] found that the vapour compression absorption (combined) refrigeration cycle using NH<sub>3</sub>-NaSCN fluid pair has greater performance than the vapour compression or absorption refrigeration cycle. Separately also compression absorption (combined) refrigeration cycle has been studied; the main parameters of the cycle have been investigated and compared with the compression cycle by Hulten et al. [32].

Ahlby et al. [33] performed a comparison study between the vapour compression cycle and the compression absorption (combined) refrigeration cycle by using absorption section with NH<sub>3</sub>-H<sub>2</sub>O fluid pair and the vapour compression systems using R12. He concluded that the coefficient of performance for the former always results in a higher coefficient and the capacity of the NH<sub>3</sub>-H<sub>2</sub>O system is also considerably higher.

## **2.1 Research Gap**

Many of the studies have been conducted in the cascade refrigeration cycle. The thermodynamic calculation has been done without using a analyser and rectifier in the absorption system. In the vapour compression system also flash chamber has not been used. Actual cycle has not been used instead ideal cycles have been used.

## **Formulation Of Problem**

In this thesis first law of thermodynamics is used for calculation of the heat flowing in and out of the system components. Calculating the net heat interaction helped us to find the cop and factors affecting the performance of the system.

# CHAPTER 3

## THERMODYNAMIC ANALYSIS OF THE CASCADE VAPOUR COMPRESSION AND ABSORPTION CYCLE

### 3.1 THEORY

A simple vapour compression system is cascaded with vapour absorption system. The whole system is shown in figure. Different component is labelled. The heat transfer directions are shown. State points of the two different refrigerants used separately in vapour compression and vapour absorption systems respectively are show. Ln P-1/T diagram of vapour absorption system and p-h diagram of vapour compression system shows the state of the refrigerants.

Thermodynamic analysis is done for various components of the VARS and VCRS. The different components are two condensers (condenser 1 and condenser 2), two evaporators (evaporator 1 and evaporator 2), a compressor, three pressure reducing valves, heat exchanger, a pump and a generator.

Steady flow analysis in vapour absorption system is done by

$\dot{m}$ =mass flow of refrigerant

$m_{ss}$ = mass flow of strong solution

$m_{ws}$  = mass flow of weak solution

$f = \frac{m_{ss}}{\dot{m}}$ , where f is the circulation ratio

Analysis is carried out by mass and energy balance of the system. Vapour pressure of the lithium bromide system is calculated from the Raoult's law which states that "the partial vapour pressure of each component of ideal mixture of liquids is equal to the vapour pressure of the pure component multiplied by its mole fraction in the mixture." Thus from this law

Vapour pressure,  $P = P_w (1-x)$

Where  $P_w$  is the saturated pressure of water at temperature and  $x$  is the mole fraction. The enthalpy of the LiBr solution is mixing is exothermic so the heat is liberated. Properties of lithium bromide system composition

Mass fraction,  $\epsilon = \frac{m_l}{m_l + m_w}$ , where  $m_l$  is the mass of anhydrous LiBr and  $m_w$  is the mass of water.

Specific enthalpy of LiBr solution is given by :-

$H = \epsilon h_l + (1 - \epsilon)h_w + \Delta h_{mix}$ , here  $\Delta h_{mix}$  negative since the mixing of LiBr solution is exothermic.

Whereas enthalpy of pure water in liquid state is given by

$$H_w \cong 4.19(t - t_{ref})$$

Enthalpy of superheated vapour is

$$H_v \cong 2501 + 1.88(t - t_{ref})$$

Where  $t$  is the temperature we need to know and  $t_{ref}$  is the reference temperature, in this case it is  $0^\circ\text{C}$

A  $\ln-1/t$  diagram of a single effect absorption cycle is shown below. Different states of the diagram and state and quality of refrigerant solution is shown in table 3.1.

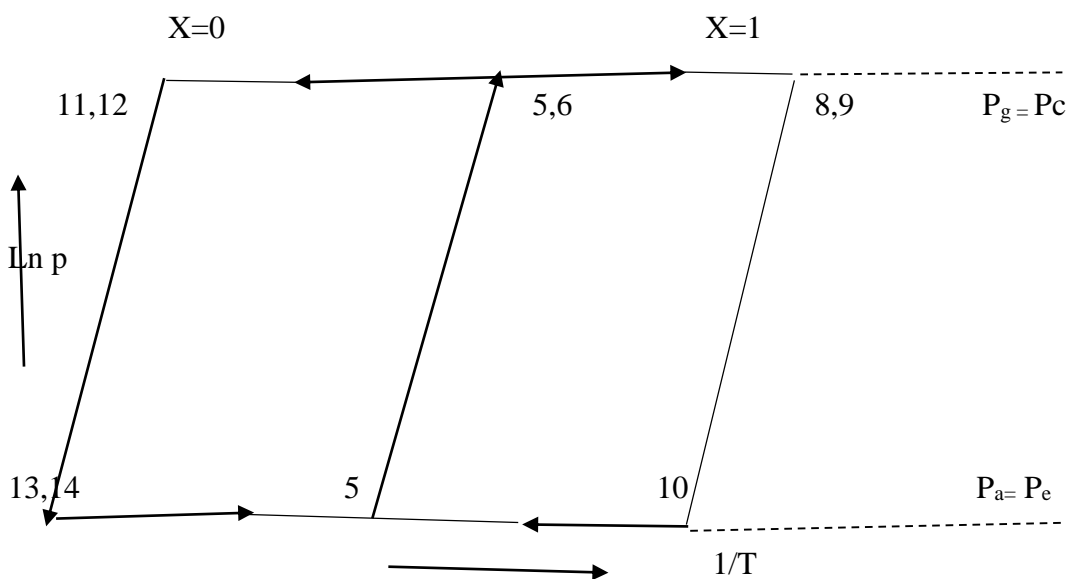


Figure 3.1  $\ln P$  vs  $1/T$  of VARS in cascade cycle

The saturation pressure versus saturation temperature relationship of pure substance can be expressed in form of

$$\ln p = a - b \frac{h_{fg}}{T}$$

The vapour compression cycle carries out four process in the refrigeration cycle. The compression is isentropic or kept close to isentropic process meaning the entropy does not change while compressing the gas in dry state. After the compression the it goes into condenser where it changes its phase from vapour to liquid. Here the enthalpy change occurs at constant pressure. After that it goes it goes into throttling process where the enthalpy remains constant but the pressure gets reduced. The process is also adiabatic. No transfer of heat is observed here. In the evaporator again the phase changes and liquid gets converted into vapour. The whole process is shown in the P-H diagram shown in fig 3.2. A table 3.2 is drawn to show the states of the refrigerant for better understanding.

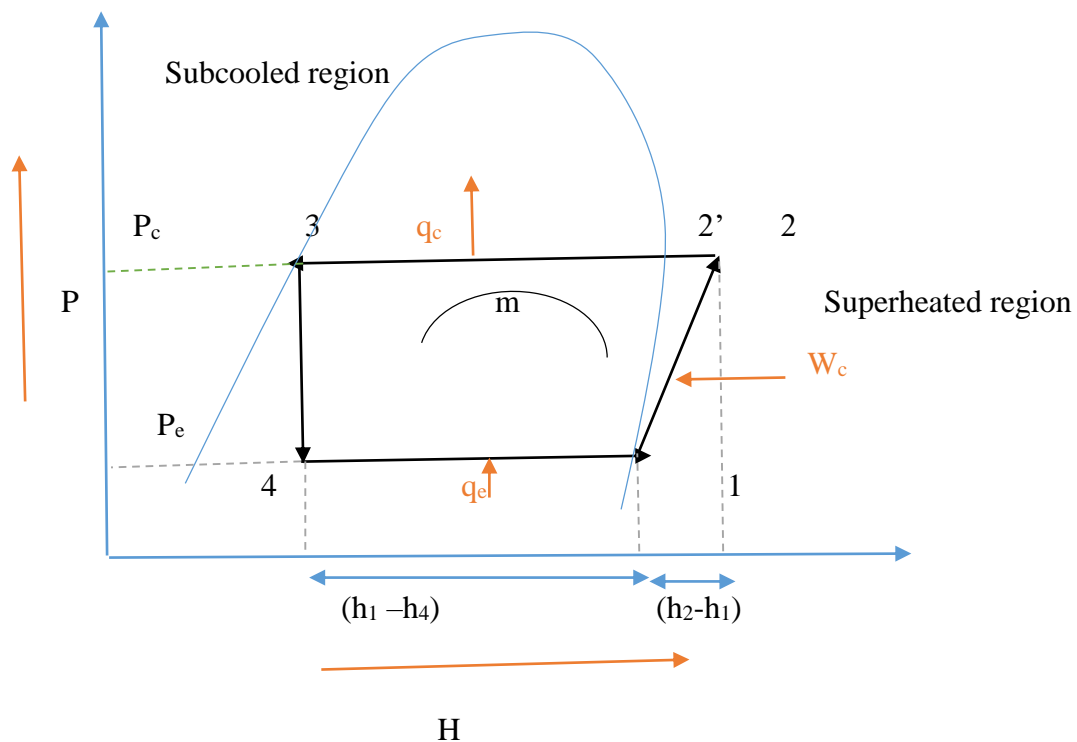


Figure 3.2 P-H diagram of VCRC in cascade cycle

From the above diagram we can see that

$$Q_c = Q_e + W_c = h_2 - h_3$$

$$\text{Cooling cop} = \frac{h_1 - h_4}{h_2 - h_3}$$

$$\text{Heating cop} = \frac{h_2 - h_3}{h_2 - h_1}$$

Circulation rate of refrigeration,  $m = \frac{\text{refrigeration capacity}}{\text{refrigeration effect per Unit mass}}$

In the isentropic process the saturation ratio and the specific heat of refrigerant vapour allows us to use the formula

$$S_1 = S_2 = S'_2 + C_p \ln \left( \frac{T_2}{T'_2} \right)$$

### TABLE OF THE REFRIGERANT STATE DURING VARIOUS PROCESS OF THE CASCADE CYCLE

Table 3.1 Refrigerant state for cascade cycle in absorption system

POINT	REFRIGERANT STATE
5	weak solution coming out from the evaporator
6	weak solution at generator pressure
7	preheated weak solution
8	strong solution
9	slightly cooled strong solution
10	strong solution at evaporator pressure
11	high pressure vapour refrigerant
12	high pressure refrigerant liquid
13	low pressure refrigerant liquid
14	low pressure low temperature saturated vapour

Table 3.2 for vapour compression section

POINT	REFRIGERANT STATE
1	Low pressure low temp saturated vapour refrigerant
2	high pressure high temperature vapour refrigerant
3	saturated liquid high pressure refrigerant
4	low pressure saturated liquid refrigerant

### Mass Flow Analysis

Net mass flowing into each component of the vapour compression and vapour absorption system must be zero in steady state flow. There is no chemical reaction between the salt lithium bromide which acts as an absorbent and pure water which is a refrigerant. There is also a conservation of mass in the vapour compression section of the cascade cycle side. The net mass of the refrigerant flowing is zero.

### Energy Analysis

The total energy analysis of the cascade system involves the mass conservation, laws of thermodynamics and species conservation. All of the above principles are applied using the following equations: -

### Mass Conservation

The formula used is

$$\sum m_i = \sum m_o$$

### Energy conservation

$$\sum Q - \sum W = \sum m_o h_o - \sum m_i h_i$$

The energy conservation equation is applied in the control volumes assumed in each component. The work and heat interaction is done in the control volume as per the laws of thermodynamics.

### 3. 2 ASSUMPTIONS

The following assumption are made in order to carry out the necessary thermodynamic analysis [1].

1. Steady state system is used.
2. At their respective temperature and pressure in the generator and absorber the water lithium bromide solution is assumed to be in equilibrium.
3. At condenser and evaporator refrigerant is assumed to be in saturated state.
4. In the absorber and generator, the strong and weak refrigerant solution is saturated.
5. Vapour is saturated at the entry of the compressor
6. Pressure losses are neglected. This includes the heat exchangers and the pipe line.
7. Pump work in the absorption section is neglected.
8. In the throttling valve, temperature of the LiBr – water solution is kept 7-8°C above the crystallisation temperature. This is done to avoid the crystallisation.

Thermal capacities are calculated using the mass and energy balance equation for the individual components of the cascade cycle.



### 3.3 COMPONENT ANALYSIS IN THE VAPOUR ABSORPTION SIDE OF THE CASCADE CYCLE

#### 3.3.1 CONDENSER

In the condenser the water vapour condenses into the liquid. Only pure refrigerant enters the condenser. Condensation is an exothermic process. The vapour comes in contact with the cooling water coil or through air and takes away the heat to condense into water. So the condenser has the pure water in liquid form at high pressure. The pressure at the condenser is the saturation pressure at the condenser temperature i.e.

$$P_c = P_{\text{sat}} \text{ at } T_c$$

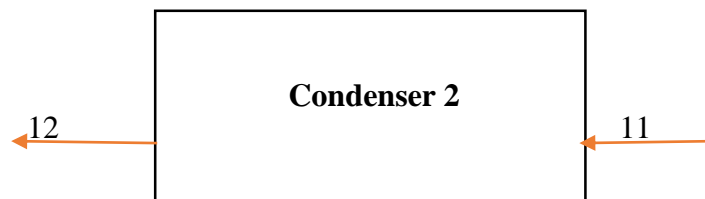


Figure 3.3 Schematic diagram of condenser<sub>2</sub>

Pure refrigerant vapour enters through point 11 and leaves as a liquid through point 12

Total mass balance

$$M_{11} = M_{12} = M$$

The total mass of the refrigerant is 0.02748 kg/s

Energy balance

Heat rejected by the condenser is equal to

$$Q_c = M (h_{11} - h_{12}) = 66.12 \text{ kJ/kg}$$

### 3.3.2 ABSORBER

At the exit of the evaporator we have a low pressure and low temperature refrigerant vapour. This refrigerant enters the absorber. Here it comes in contact with the strong solution of LiBr coming from generator through the heat exchanger. Since the strong LiBr solution has the affinity for water vapour, hence water vapour gets absorbed here. A weak solution of LiBr is formed here. This process of mixing is exothermic and hence heat is released. Heat released is  $Q_a$ . here circulation ratio is

$$f = \frac{\dot{m}_8}{\dot{m}_{11}} = \frac{X_7}{X_8 - X_7} \text{ where}$$

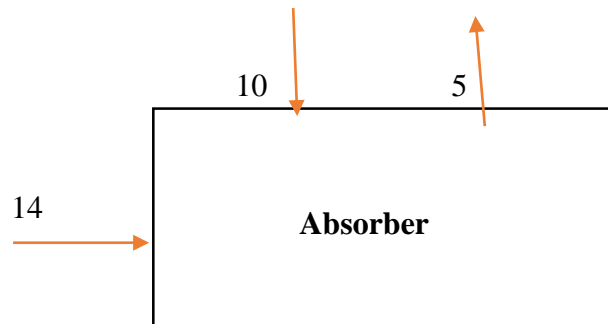


Figure 3.4 Schematic diagram of absorber

#### TOTAL MASS BALANCE

#### REFRIGERANT MASS BALANCE

$$M_{ws} = M_{ss} + M$$

$$M_{ss} = fM$$

$$M_{ws} = M(f+1)$$

Amount of refrigerant entering the system must leave the system.

$$M + (1 - \epsilon_{ss})M_{ss} = (1 - \epsilon_{ws})M_{ws} \quad \text{and}$$

$$f = \frac{\epsilon_{ws}}{\epsilon}$$

COP of the system depends upon the circulation ratio  $f$ ,

$$Q_a = M[(h_{14} - h_5) + f(h_{10} - h_5)]$$

$Q_a$  is desired to be low.

Mass balance

$$M_{ws} = M_{ss} + M$$

$$m_5 = m_{10} + m_{14}$$

#### ENERGY BALANCE

$$Q_a = m_{10}h_{10} + m_{14}h_{14} - m_5 h_5$$

### 3.3.3 GENERATOR

In the generator we have to supply heat. Heat can be supplied in a wide variety of way using solar energy or through waste heat in power plants etc. When the heat is supplied to weak solution in generator water vapour is generated. Here desorption takes place as the water gets separated from the water LiBr solution. The generated vapour goes in to the condenser. The left over solution is settled down in generator due to gravity. Since the generator is high pressure and high heat capacity device, so the difference between the absorber and generator makes it flow to the absorber through the pressure reducing throttling valve.

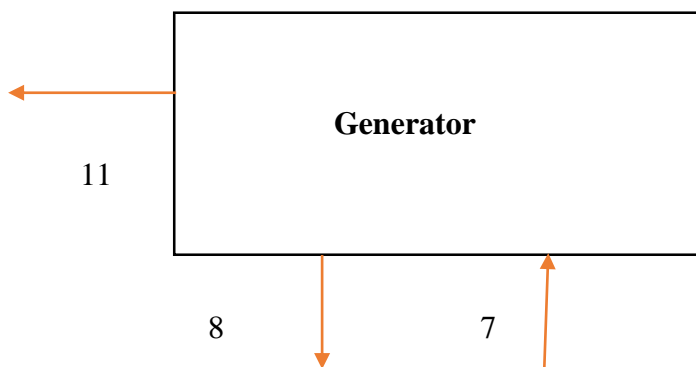


Figure 3.5 Schematic diagram of generator

Mass balance

$$M_1 = M_{11} + M_8$$

Material balance

Energy balance

$$Q_{gen} = m_{11} h_{11} + m_8 h_8 - m_7 h_7$$

The total energy can be also calculated from this formula

$$Q_{gen} = m[(h_{11} - h_7) - f(h_8 - h_7)]$$

### 3.3.4 EVAPORATOR

In the evaporator there is a low pressure water. The pressure is reduced before entering into the evaporator through a pressure reducing expansion valve. The liquid refrigerant absorbs the heat from the surrounding or the area to be cooled. The process is endothermic. So it takes the heat and the refrigerants get converted into low pressure vapour. This low pressure vapour then enters the absorber. In the evaporator refrigeration effect takes place by taking the heat  $Q_e$ . The liquid refrigerant enters through point 13 and leaves at point 14.



Figure 3.6 Schematic diagram of evaporator<sub>2</sub>

Mass balance

$$M_{13} = M_{14} = M$$

Energy balance

Heat absorbed by the evaporator

$$Q_e = m(h_{14} - h_{13})$$

### 3.3.5 REFRIGERANT EXPANSION VALVE

The high pressure water from the condenser enter into the refrigerant expansion valve and pressure is reduced to the evaporator pressure. The process is isenthalpic so there is no change in enthalpy across the refrigerant expansion valve.

Mass balance

$$M_{12} = M_{13} = M$$

Energy balance

$$h_{12} = h_{13}$$

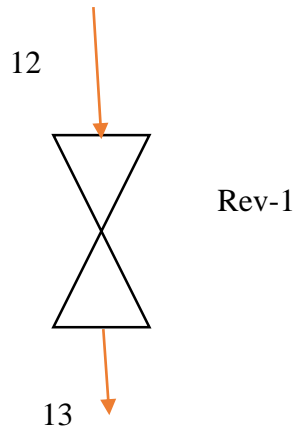


Figure 3.7 Schematic diagram of REV 2

### 3.3.6 SOLUTION EXPANSION VALVE

strong solution from the heat exchanger enters into the expansion valve at high pressure. This strong solution is reduced to a low pressure strong solution. This low pressure is equal to the absorber pressure. The process is isenthalpic; enthalpy remains constant throughout the solution expansion valve.

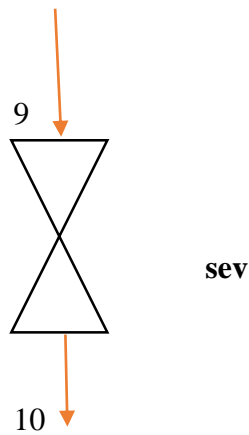


Figure 3.8 Schematic diagram of SEV

Mass balance

$$M9 = M10$$

Energy balance

$$h9 = h10$$

### 3.3.7 PUMP

work done by the pump is neglected here. So the change in energy is negligible. Only the pressure is increased from the low pressure weak solution from the absorber to the high pressure weak solution generator pressure.



Figure 3.9 Schematic diagram of pump

Mass balance

$$M5 = M6$$

Energy balance

$$h5 = h6$$

### 3.3.8 SOLUTION HEAT EXCHANGER

In the solution heat exchanger, the high temperature strong solution from the generator enters and it exchanges the heat from high pressure weak solution coming from the absorber and leaving the solution heat exchanger.

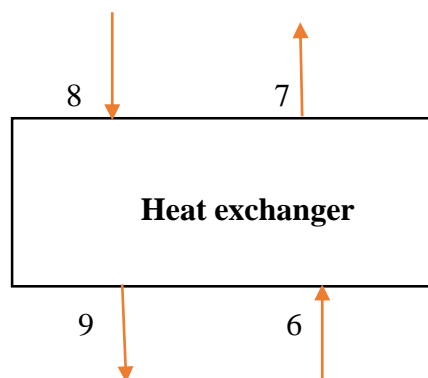


Figure 3.10 Schematic diagram of SHE

Effectiveness of the solution heat exchanger

$$\epsilon_{hx} = \frac{T_7 - T_6}{T_8 - T_6} = \frac{Q_{actual}}{Q_{maximum}}$$

From the above equation temperature of the weak solution entering the generator can be obtained since  $T_6$  is equivalent to  $T_5$  and  $T_8$  is equivalent to generator temperature i.e.  $T_e$ .

Here strong solution enters through point 8 and leaves through point 9. Weak solution enters through point 6 and leaves through point 7. Here weak solution means strong in refrigerant and weak in LiBr concentration.

### **3.4 COMPONENT ANALYSIS OF VAPOUR COMPRESSION REFRIGERATION CYCLE OF THE CASCADE CYCLE**

In this analysis the total mass balance and energy balance in the vapour compression system components is done. P-h diagram of the vapour compression system is shown in figure 3.2. The p-h diagram of the system shows the different state of the refrigerant used in this cycle.

#### **3.4.1 COMPRESSOR**

Compressor compresses the refrigerant from the low pressure refrigerant coming out of the evaporator to high pressure refrigerant before going into 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 0.80 and 0.90 respectively [1].

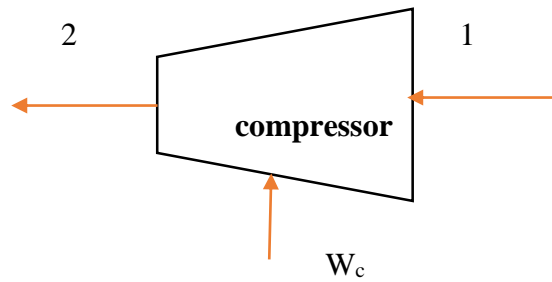


Figure 3.11 Schematic diagram of compressor

### Mass balance equation

Dry compression is used and there is no loss of refrigerants in compression.

$$M_1 = M_2 = M$$

### Energy Balance:

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

$$h_2 = h_1 + (h_{2s} - h_1) / \eta_{is}$$

so total work done by the compressor

$$\dot{W} = M_1(h_2 - h_1) / \eta_e$$

### 3.4.2 CONDENSER

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



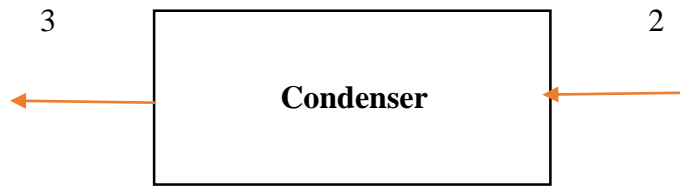


Figure 3.12 Schematic diagram of condenser<sub>1</sub>

**Total mass balance**

$$M_2 = M_3 = M$$

**Total energy balance**

$$Q_{\text{condenser}} = M_3 (h_2 - h_3)$$

**3.4.3 REFRIGERANT EXPANSION VALVE**

The high pressure refrigerant from the condenser is reduced to low pressure refrigerant by passing through a refrigerant expansion valve. Energy flow across this remains constant since it is isenthalpic.

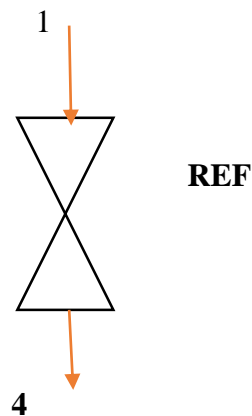


Figure 3.13 schematic diagram of REV 1

**Mass Balance**

$$M_3 = M_4 = M$$

### Total Energy Balance

$$h_3 = h_4$$

### 3.4.4 EVAPORATOR

the evaporator in the vapour compression system acts as a source since it accepts heat from the surrounding area to be cooled. The refrigerant in the liquid form takes away the heat from the area to be cooled and converts itself into the vapour form. The refrigerant vapour formed is at low pressure and low temperature before entering into the compressor.



Figure 3.14 Schematic diagram of evaporator<sub>1</sub>

### Mass Balance

$$M_4 = M_1 = M$$

### Energy Balance

$$Q_e = m_1 (h_1 - h_4)$$

Table 3.3 summary of mass and energy equations

Component	Mass equation	Energy equation
<b>Pump</b>	$M_6 = M_5$	$M_5 (h_6-h_5)$
<b>Heat exchanger</b>	$M_6 = M_7$ $M_8 = M_9$	$M_6(h_7-h_6) =$ $M_8(h_8-h_9)$
<b>Generator</b>	$M_7 = M_{11}+M_8$	$Q_g = M_{11}h_{11}+M_8h_8-$ $M_7h_7$
<b>Condenser 2</b>	$M_{11}=M_{12}$	$Q_c=M_{11} (h_{11}-h_{12})$
<b>Evaporator 2</b>	$M_{13} = M_{14}$	$Q_e = M_{13}(h_{14}-h_{13})$
<b>Absorber</b>	$M_5 = M_{10} + M_{14}$	$Q_a =M_{10}h_{10}+M_{14}h_{14}-$ $M_5h_5$
<b>Condenser 1</b>	$M_2 = M_3$	$Q_c = M(h_2-h_3)$
<b>Evaporator 1</b>	$M_4 = M_1$	$Q_e=M_1(h_1-h_4)$
<b>Compressor</b>	$M_1 = M_2$	$W_c=M_1(h_2-h_1)/\eta_e$
<b>Overall cycle</b>	$M_{sol} = M_{LiBr}+M_{water}$ $M_{sol}x_{sol} =$ $M_{LiBr}x_{LiBr}+M_{water}x_{water}$	$(Q_g +Q_{e1}+W_c+W_p)_{inlet}$ $= Q_a+Q_{c2}$ $COP_{abs} = Q_{e2}/Q_g$ $COP_{cycle}=Q_{e1}/(Q_g+W_c)$ $COP_{vap} =Q_{e1}/W_c$

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

State point	T(k)	P(kPa)	h	M
1	263	199.6	244.5	1.216
2	298	535	293.7	1.216
3	291	235	76.3	1.216
4	263	199.6	76.3	1.216
5	313	1.216	93.79	-
6	313	7.323	93.79	-
7	337	7.323	143.3	-
8	363	7.323	224.2	-
9	333	7.323	168.1	-
10	333	1.216	168.1	-
11	363	7.323	2573	0.02748
12	313	7.323	166.9	0.02748
13	283	1.216	166.9	0.02748
14	283	1.216	2519	0.02748

## CHAPTER 4

### RESULT AND DISCUSSION

The results are compared with the results obtained by cimsit and ozturk. The results are obtained by taking the following input parameters

For all the calculation in the vapour compression side of the cascade cycle temperatures are taken as following

Cooling load = 50 kW

Condenser temperature = 291 K

Temperature after dry compression = 298 K

Isentropic efficiency of the compressor = 0.8

Electric motor efficiency = 0.9

For all the calculations in the vapour absorption side of the cascade cycle following assumption is made

Condenser temperature = 313 K

Generator temperature = 363 K

Heat exchanger effectiveness = 0.6

Evaporator temperature = 283 K

Table 4.1 Comparison of The Present Work Done with The Numerical Value Given in Cimsit and Ozturk (2012)

Component	Cimsit and Ozturk (2012)	Present work	Difference in percentage
	Q(kW)	Q(kW)	
Condenser <sub>1</sub>	57.41	64.63	0.11
Generator	76.45	83.63	0.08
Absorber	72.76	82.13	0.11
Condenser <sub>2</sub>	61.06	66.13	0.07
Cop <sub>abs</sub>	0.750	0.7728	0.02
Cop <sub>cyclegen</sub>	0.590	0.5006	-0.17
Cop <sub>classical vc</sub>	2.9	3.076	0.05

Work done by the compressor = 16.25 kW

Mass rate of refrigerant flow in absorption section = 0.02748 kg/s

Mass rate of refrigerant flow in vapour compression cycle = 0.2973 kg/s

Mass flow of strong solution = 0.2359 kg/s

Mass flow rate of weak solution = 0.2084 kg/s

Mass fraction of lithium bromide in weak solution = 0.6216

Mass flow of lithium bromide in strong solution = 0.5492

Cop of the cascade cycle = 0.5006

## 4.1 Analysis with different evaporator temperature

### 4.1.1 Evaporator<sub>1</sub> temperature with cop

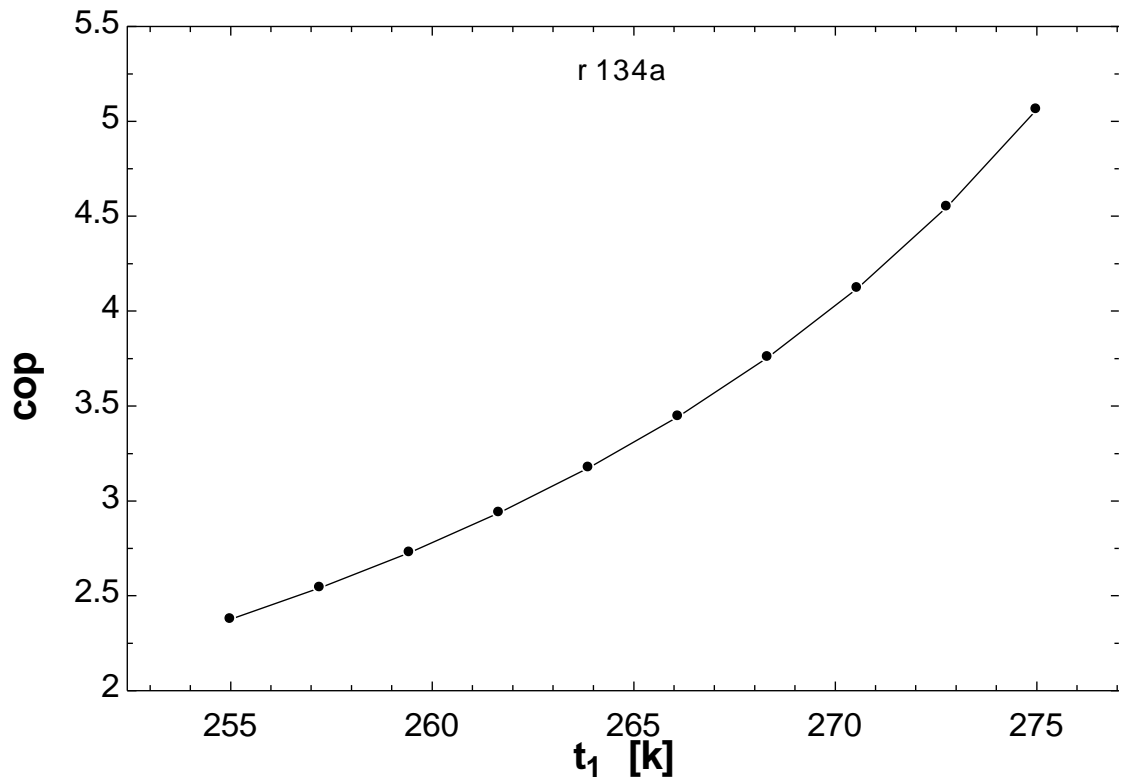


Figure 4.1 Variation of  $\text{cop}_{\text{vap comp}}$  with  $T_1$

From the above figure we can see that as the evaporator temperature increases the cop of the vapour compression side of the cascade system increases. the refrigerant R134a shows better performance as the temperature in the evaporator rises. The value of the evaporator temperature varies from 255 K to 275 K.

### 4.1.2 Effect of Condenser Heat with Increase in Evaporator Temperature

As it can be seen from below figure that with the increase in the evaporator temperature the value of the heat rejected by the condenser is decreased. The value of the evaporator temperature varies from 255 K to 275 K. as the condenser rejects less heat the heat supplied to the evaporator section of the absorption section is reduced.

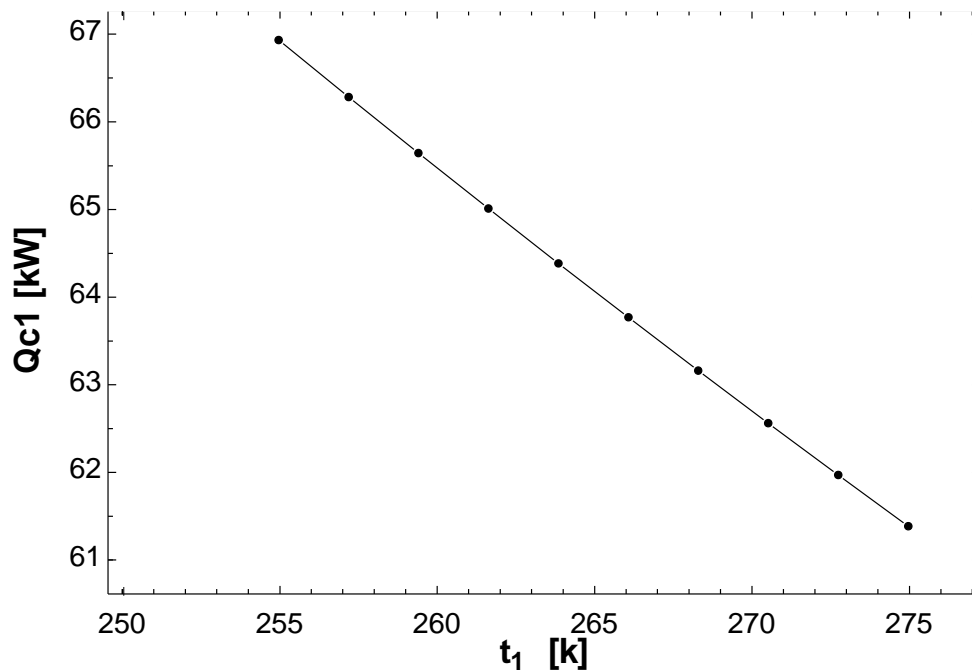


Figure 4.2 Variation of  $Q_{c1}$  with  $T_1$



### 4.1.3 Variation of Generator Heat with Evaporator Temperature

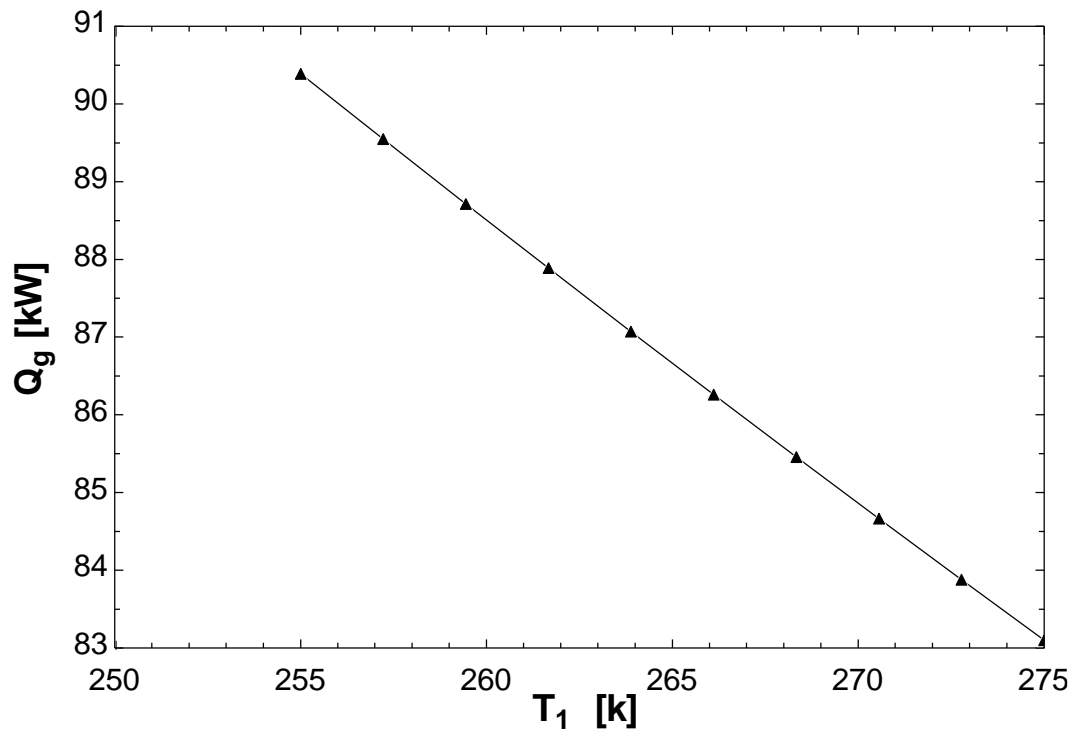


Figure 4.3 Variation of  $Q_g$  with  $T_1$

As the evaporator temperature of the vapour compression section of the cascade cycle is increased the generator heat in the absorption section of the cascade cycle is decreased.

## 4.2 Analysis for Different Condenser Temperature

### 4.2.1 Cop of cascade cycle with different condenser temperature

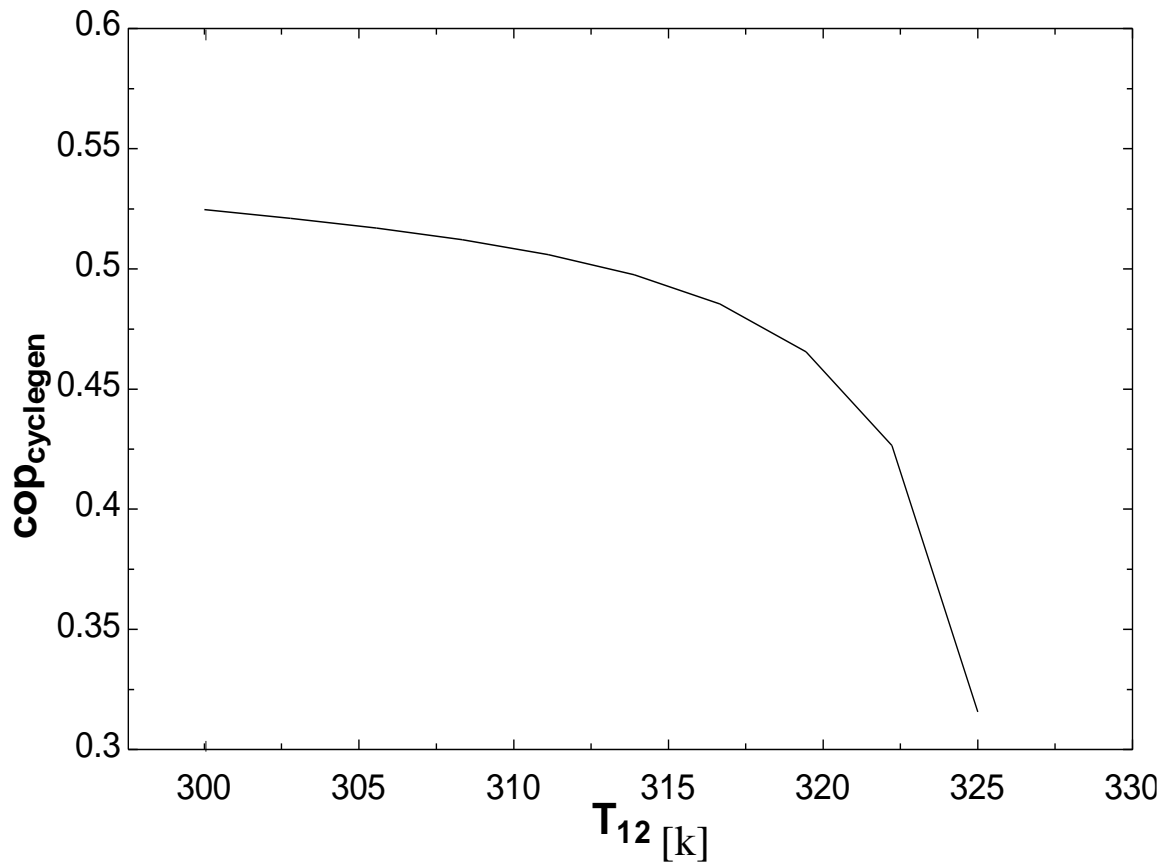


Figure 4.4 Variation of  $T_{12}$  with COP of cascade cycle

As it can be seen that with the increase in condenser temperature the coefficient of performance of the cascade cycle decreases. There is a strong dip in coefficient of performance, once the temperature exceeds 322 K.

## 4.2.2 Cop of The Absorption System with Different Condenser Temperature

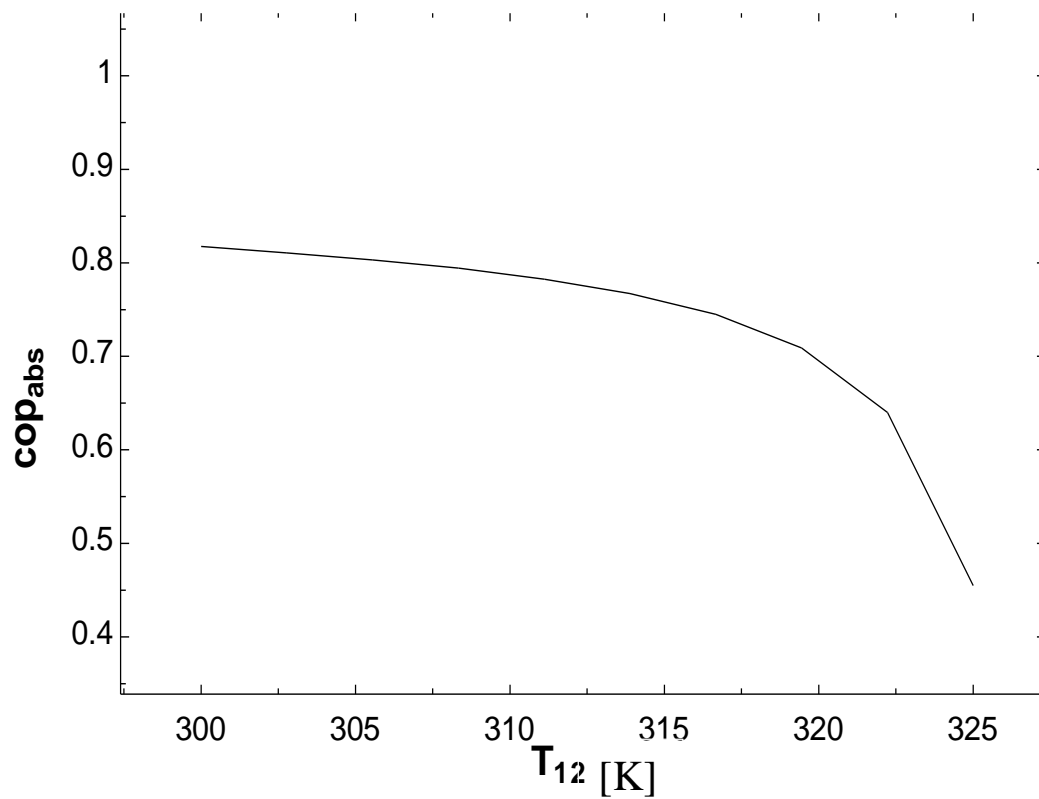


Figure 4.5 Variation of T<sub>12</sub> with COP of absorption side

With the decrease in the condenser temperature in the vapour absorption cycle of the system the cop increases. the performance gets better but after certain increase in temperature the cop decreases drastically as it can be seen in the plotted graph.

### 4.2.3 Generator Heat and Condenser Temperature

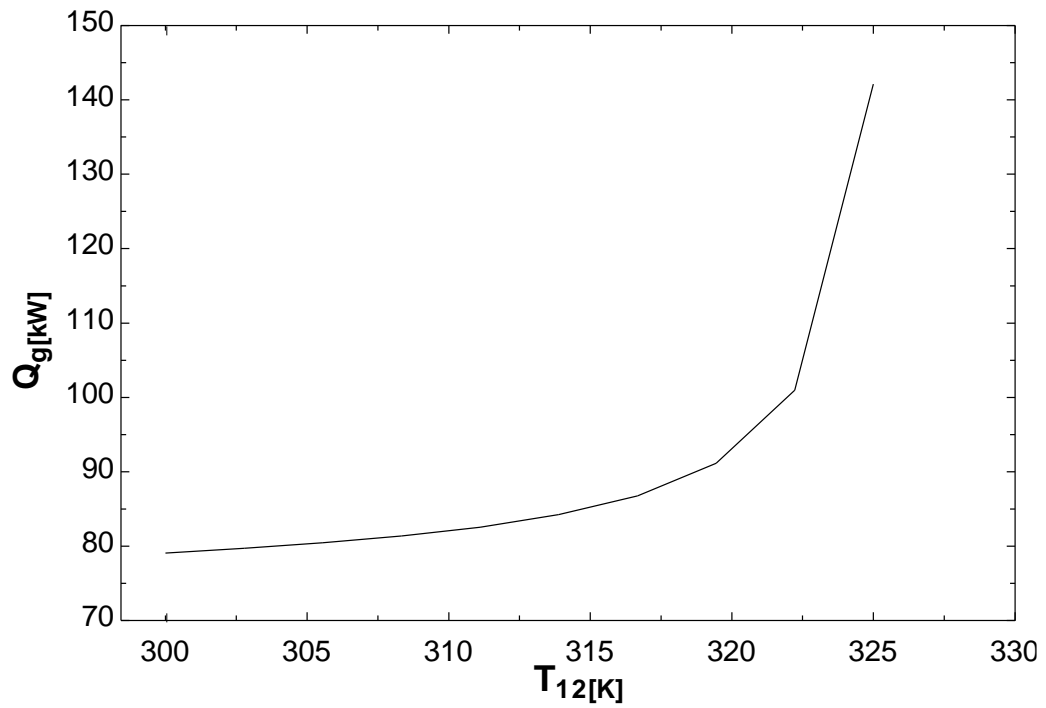


Figure 4.6 Variation of  $T_{12}$  with generator heat

With the increase in the condenser temperature of the absorption section of the cascade cycle the heat required in the generator is also increased.

## 4.3 ANALYSIS FOR DIFFERENT GENERATOR TEMPERATURE

### 4.3.1. Generator Heat and Generator Temperature

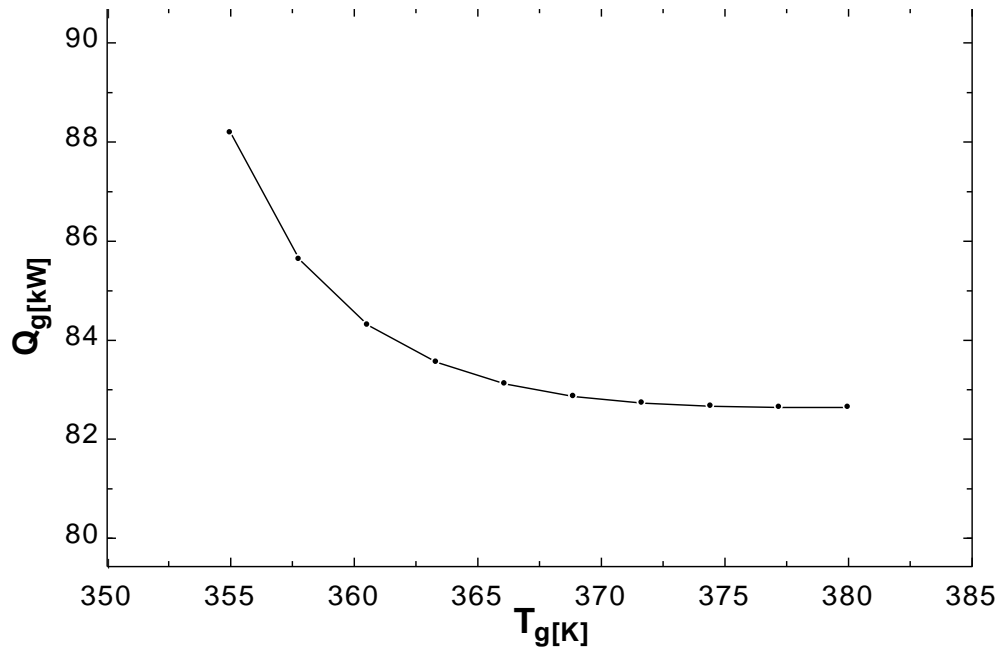


Figure 4.7 Variation of  $Q_g$  with  $T_g$

The above graph shows that with the increase in the generator temperature the heat required for the generator becomes less. Heat capacity for the generator is reduced.

### 4.3.2 Generator Temperatures and Cop of the Absorption Section

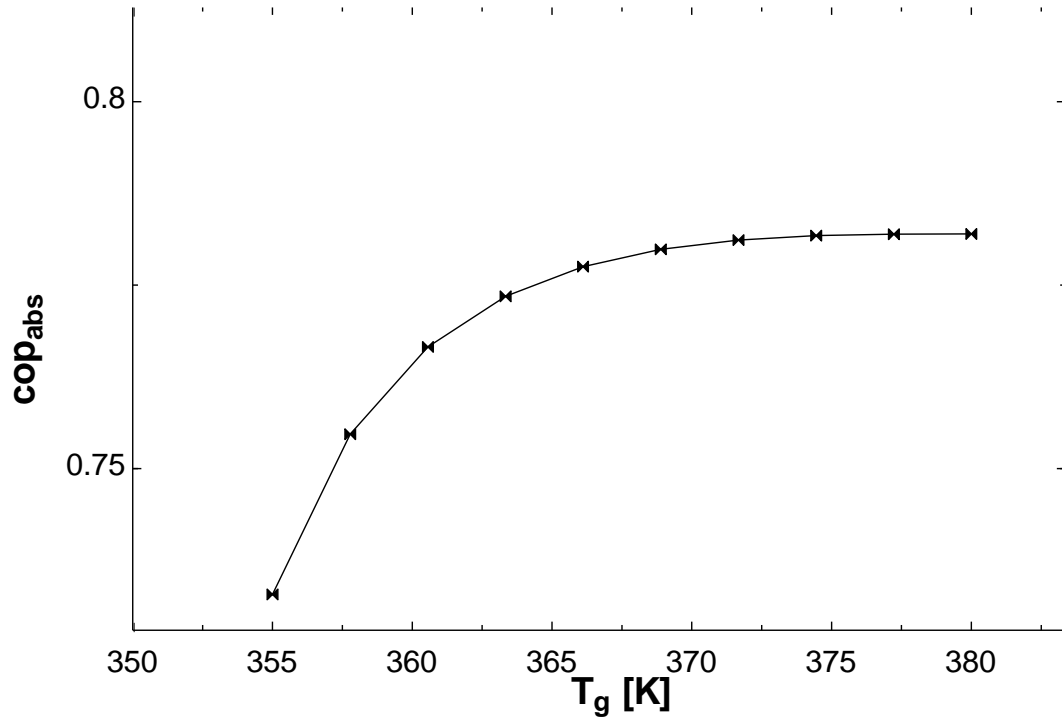


Figure 4.8 Variation of  $COP_{abs}$  with  $T_g$

With the increase in the generator temperature heat required by the generator becomes less, so the cop of the absorption cycle increases. The graph also shows that from 350 K the cop increase at continues to do show with the increase in the generator temperature.

### 4.3.3 Cop Cascade Cycle and Generator Temperature

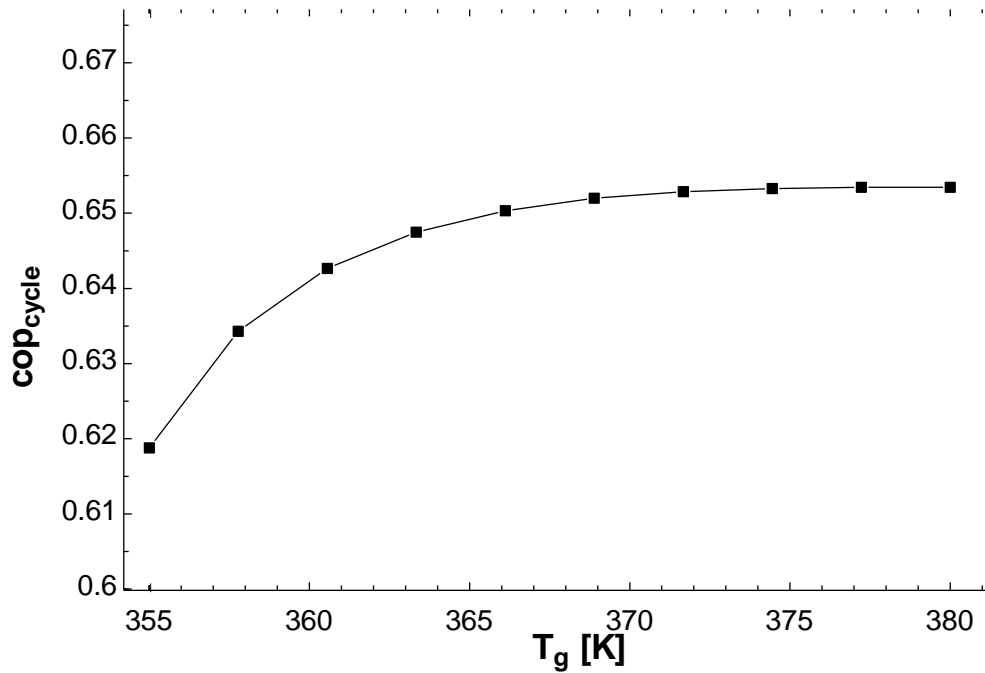


Figure 4.9 Variation of  $COP_{cycle}$  with  $T_g$

As we can see from the formula of cop of the cascade cycle if the work done by the compressor in the vapour compression side of the cascade cycle is fixed, the cop of the cascade cycle is inversely proportion to the generator heat. With the increase in the generator temperature heat required by the generator is decreased, hence we can see increase in the cop of the cycle.

## 4.4 Analysis for Different Temperature Differences of Cascade Heat Exchanger

### 4.4.1 $\delta T$ with heat in the generator

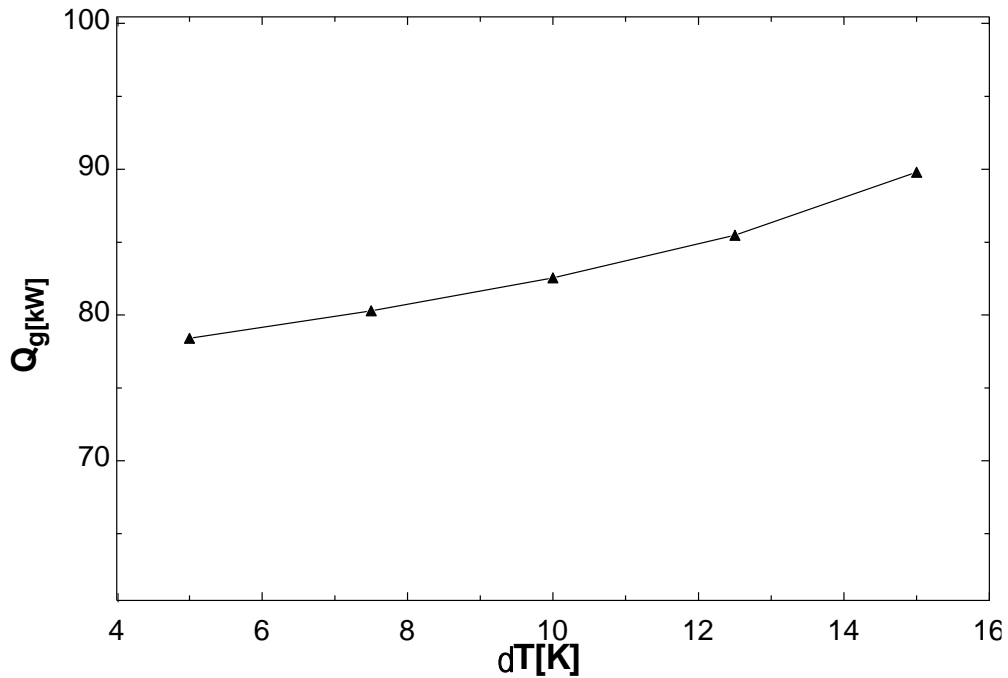


Figure 4.10 Variation of  $\delta T$  with generator heat

Operating conditions used is same as mentioned at the beginning of the chapter. The temperature difference of the heat exchanger is between condenser<sub>1</sub> and evaporator<sub>2</sub>. The different temperature is mentioned below

Condenser<sub>1</sub> temperature = 293 K

Evaporator<sub>2</sub> temperature = 278 K, 280.5 K, 283 K, 285.5 K, 288 K

As  $\delta T$  increases the heat required by the generator in the absorption section is also increased.



#### 4.4.2 $\delta T$ With Cop Of The Cascade Cycle

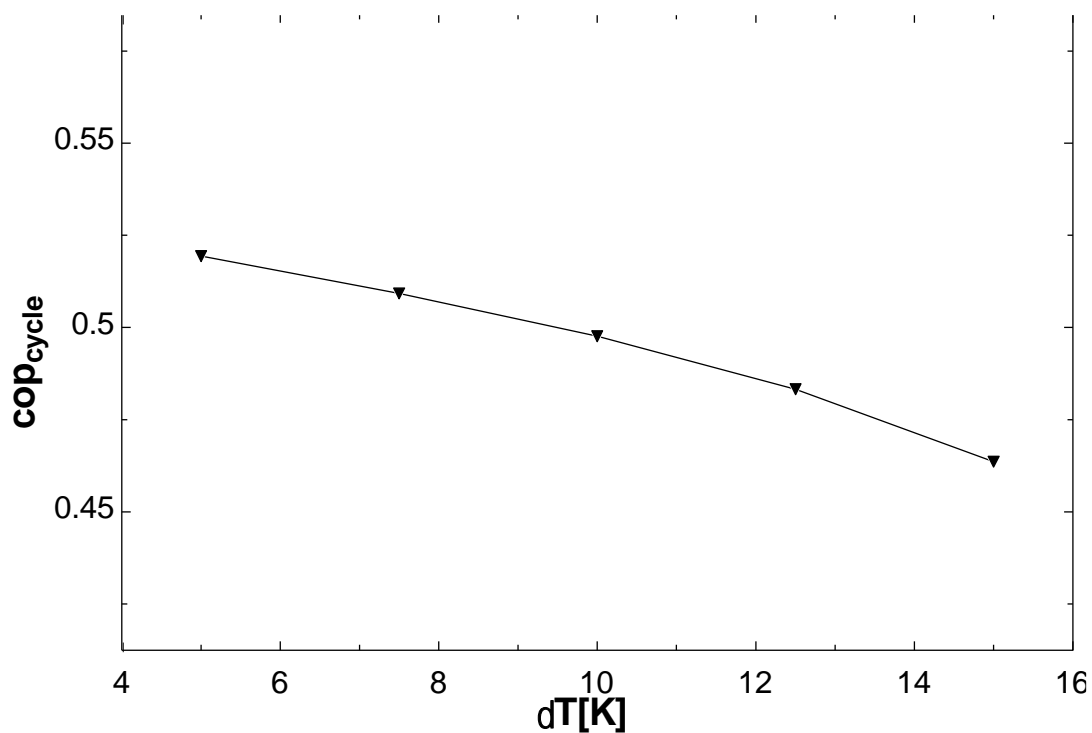


Figure 4.11 variation of  $\delta T$  with COP of the cascade cycle

The difference in the condenser temperature and the evaporator temperature is same as mentioned above. The parameters are same only the effect of  $\delta T$  on cop of the cascade cycle is studied here. With the increase in the temperature difference the cop of the overall cycle decreases. This happens due to increase in the generator temperature required due to increase in temperature difference.

## CHAPTER 5

### CONCLUSION

Extensive thermodynamic analysis using the first law of thermodynamics in an assumed control volume of each component of the cascade cycle is done. The evaporator temperature or the ambient temperature is taken as 263K. capacity of the refrigeration system is taken as 50 kW.

It has been found out that when using a conventional vapour compression system the work done in the compressor is about 32% more than the work done by the compressor in the cascade cycle. The operating conditions are taken similar i.e. 263K in the evaporator and 313 K in the condenser of both the cycles. The condenser in the cascade cycle is of the absorption section. So due to this less electricity is used. Cooling can be done more energy efficiently.

It has been also found out that the COP of the cascade cycle can be increased by increasing the evaporator temperature and the generator temperature but when the condenser temperature is increased then the performance decreases.

The heat exchanger is used to increase the efficiency by exchanging the heat through generator and absorber solution. The waste heat of generator is taken by the absorber solution going into the generator. Temperature difference in the cascade heat exchanger is when increased then the performance of the cycle is also decreased. Effects of different temperature in generator, condenser and evaporator in cascade cycle performance is plotted and studied.

#### **5.1 Scope for Future Work**

In this study I have taken only one a single refrigerant pair of R134a and LiBr-water solution. Further this cycle can be studied for different refrigeration pairs and their effect on the performance of the cycle. Different types of low grade heat can be used in the absorption section of the cycle without much wastage. Commercialisation of this cycle can be done for low electricity consumption.

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## Appendices

```
t_1=263 [k]
P_1=Pressure(R134a,T=T_1,x=x_1)
t_2=298 [k]
P_sat_3=P_sat(R134a,T=T_3)
t_3=291 [k]

t_3=t_4
eta_is=0.80;eta_e=0.90
q_evap=50

h_1=Enthalpy(R134a,T=T_1,x=x_1);x_1=1
s_1=Entropy(R134a,T=T_1,x=x_1)
s_1=s_2
h_2s=Enthalpy(R134a,T=T_2,s=s_2)
h_2=h_2s+(h_2s-h_1)/eta_is
h_4=Enthalpy(R134a,T=T_3,x=x_2);x_2=0
condenser_1=m_dot*(h_2-h_4)

q_evap=m_dot*(h_1-h_4)
w_comp=m_dot*(h_2-h_1)/eta_e
cop=q_evap/w_comp

t_g=363
t_a=313
t_12=313
t_14=283

q=condenser_1

P_12=P_sat(Steam,T=T_12)
P_14=P_sat(Steam,T=T_14)
x_weak=x_LiBrH2O(T_g,P_12)
x_strong=x_LiBrH2O(T_a,P_14)
h_8=h_LiBrH2O(T_g,x_weak)

p_a=P_LiBrH2O(T_a,x_strong)
p_g=P_LiBrH2O(T_g,x_weak)

epsilon=0.6
epsilon=(t_g-t_9)/(t_g-t_a)
h_9=h_LiBrH2O(T_9,x_weak)
h_5=h_LiBrH2O(T_a,x_strong)

h_11=Enthalpy(Water,T=T_12,x=1)
h_12=Enthalpy(Water,T=T_12,x=0)
h_12=h_13
h_14=Enthalpy(Water,T=T_14,x=1)
m_ref=q/(h_14-h_13)
m_strong*x_strong=m_weak*x_weak
m_ref+m_weak=m_strong
m_strong*(h_7-h_5)=m_weak*(h_8-h_9)
q_gen=(m_ref*h_11)+(m_weak*h_8)-(m_strong*h_7)

cop_abs=q/q_gen
cop_cyclegen=q_evap/(q_gen+w_comp)

q_condenser2=m_ref*(h_11-h_12)

q_absorber=(m_ref*h_14)+(h_9*m_weak)-(m_strong*h_5)
```

```

t_1=263 [k]
t_1=t_5
t_2=298 [k]
t_3=291 [k]
t_3=t_4
eta_is=0.80;eta_e=0.90
m_dot=0.2973 [kg/s]

h_1=Enthalpy(R134a,T=T_1,x=x_1);x_1=1
s_1=Entropy(R134a,T=T_1,x=x_1)
s_1=s_2
h_2s=Enthalpy(R134a,T=T_2,s=s_2)
h_2=h_2s+(h_2s-h_1)/eta_is
h_4=Enthalpy(R134a,T=T_3,x=x_2);x_2=0
condenser_1=m_dot*(h_2-h_4)

h_4=h_5

q_evap=m_dot*(h_1-h_4)
w_comp=m_dot*(h_2-h_1)/eta_e
cop=q_evap/w_comp

```

**Parametric Table: Table 2**

	$t_1$ [k]	condenser <sub>1</sub>
Run 1	255	66.92
Run 2	257.2	66.27
Run 3	259.4	65.63
Run 4	261.7	65
Run 5	263.9	64.37
Run 6	266.1	63.76
Run 7	268.3	63.15
Run 8	270.6	62.55
Run 9	272.8	61.96
Run 10	275	61.37



```

t_g=363
t_a=313
"t_12=313"
t_14=283
q=condenser_1
P_12=P_sat(Steam,T=T_12)
P_14=P_sat(Steam,T=T_14)
x_weak=x_LiBrH2O(T_g,P_12)
x_strong=x_LiBrH2O(T_a,P_14)
h_8=h_LiBrH2O(T_g,x_weak)

epsilon=0.6
epsilon=(t_g-t_9)/(t_g-t_a)
h_9=h_LiBrH2O(T_9,x_weak)
h_5=h_LiBrH2O(T_a,x_strong)

h_11=Enthalpy(Water,T=T_12,x=1)
h_12=Enthalpy(Water,T=T_12,x=0)
h_12=h_13
h_14=Enthalpy(Water,T=T_14,x=1)
m_ref=q/(h_14-h_13)
m_strong*x_strong=m_weak*x_weak
m_ref+m_weak=m_strong
m_strong*(h_7-h_5)=m_weak*(h_8-h_9)
q_gen=(m_ref*h_11)+(m_weak*h_8)-(m_strong*h_7)

t_1=263 [k]
t_1=t_5
t_2=298 [k]
t_3=291 [k]
t_3=t_4
eta_is=0.80;eta_e=0.90
m_dot=0.2973 [kg/s]

h_1=Enthalpy(R134a,T=T_1,x=x_1);x_1=1
s_1=Entropy(R134a,T=T_1,x=x_1)
s_1=s_2
h_2s=Enthalpy(R134a,T=T_2,s=s_2)
h_2=h_2s+(h_2s-h_1)/eta_is
h_4=Enthalpy(R134a,T=T_3,x=x_2);x_2=0
condenser_1=m_dot*(h_2-h_4)

```

**Parametric Table: Table 1**

	$T_{12}$	$q_{gen}$
Run 1	300	79.04
Run 2	302.8	79.7
Run 3	305.6	80.45
Run 4	308.3	81.37
Run 5	311.1	82.56
Run 6	313.9	84.22
Run 7	316.7	86.75
Run 8	319.4	91.16
Run 9	322.2	101
Run 10	325	142.1

```

"t_g=363"
t_a=313
t_12=313
t_14=283

P_12=P_sat(Steam,T=T_12)
P_14=P_sat(Steam,T=T_14)
x_weak=x_LiBrH2O(T_g,P_12)
x_strong=x_LiBrH2O(T_a,P_14)
h_8=h_LiBrH2O(T_g,x_weak)

epsilon=0.6
epsilon=(t_g-t_9)/(t_g-t_a)
h_9=h_LiBrH2O(T_9,x_weak)
h_5=h_LiBrH2O(T_a,x_strong)

h_11=Enthalpy(Water,T=T_12,x=1)
h_12=Enthalpy(Water,T=T_12,x=0)
h_12=h_13
h_14=Enthalpy(Water,T=T_14,x=1)
m_ref=q/(h_14-h_13)
m_strong*x_strong=m_weak*x_weak
m_ref+m_weak=m_strong
m_strong*(h_7-h_5)=m_weak*(h_8-h_9)
q_gen=(m_ref*h_11)+(m_weak*h_8)-(m_strong*h_7)

q=condenser_1

t_1=263 [k]
t_1=t_5
t_2=298 [k]
t_3=291 [k]
t_3=t_4
eta_is=0.80;eta_e=0.90
m_dot=0.2973 [kg/s]

h_1=Enthalpy(R134a,T=T_1,x=x_1);x_1=1
s_1=Entropy(R134a,T=T_1,x=x_1)
s_1=s_2
h_2s=Enthalpy(R134a,T=T_2,s=s_2)
h_2=h_2s+(h_2s-h_1)/eta_is
h_4=Enthalpy(R134a,T=T_3,x=x_2);x_2=0
condenser_1=m_dot*(h_2-h_4)

```

**Parametric Table: Table 1**

	$T_g$	$q_{gen}$
Run 1	355	88.18
Run 2	357.8	85.63
Run 3	360.6	84.3
Run 4	363.3	83.55
Run 5	366.1	83.11
Run 6	368.9	82.86
Run 7	371.7	82.73
Run 8	374.4	82.67
Run 9	377.2	82.64
Run 10	380	82.64

```

t_g=363
t_a=313
t_12=313
t_14=283
q=70.38
P_12=P_sat(Steam,T=T_12)
P_14=P_sat(Steam,T=T_14)
x_weak=x_LiBrH2O(T_g,P_12)
x_strong=x_LiBrH2O(T_a,P_14)
h_8=h_LiBrH2O(T_g,x_weak)

epsilon=0.6
epsilon=(t_g-t_9)/(t_g-t_a)
h_9=h_LiBrH2O(T_9,x_weak)
h_5=h_LiBrH2O(T_a,x_strong)

h_11=Enthalpy(Water,T=T_12,x=1)
h_12=Enthalpy(Water,T=T_12,x=0)
h_12=h_13
h_14=Enthalpy(Water,T=T_14,x=1)
m_ref=q/(h_14-h_13)
m_strong*x_strong=m_weak*x_weak
m_ref+m_weak=m_strong
m_strong*(h_7-h_5)=m_weak*(h_8-h_9)
q_gen=(m_ref*h_11)+(m_weak*h_8)-(m_strong*h_7)

```



```

t_g=363
t_a=313
t_12=313
"t_14=283"
q=condenser_1
P_12=P_sat(Steam,T=T_12)
P_14=P_sat(Steam,T=T_14)
x_weak=x_LiBrH2O(T_g,P_12)
x_strong=x_LiBrH2O(T_a,P_14)
h_8=h_LiBrH2O(T_g,x_weak)

epsilon=0.6
epsilon=(t_g-t_9)/(t_g-t_a)
h_9=h_LiBrH2O(T_9,x_weak)
h_5=h_LiBrH2O(T_a,x_strong)

h_11=Enthalpy(Water,T=T_12,x=1)
h_12=Enthalpy(Water,T=T_12,x=0)
h_12=h_13
h_14=Enthalpy(Water,T=T_14,x=1)
m_ref=q/(h_14-h_13)
m_strong*x_strong=m_weak*x_weak
m_ref+m_weak=m_strong
m_strong*(h_7-h_5)=m_weak*(h_8-h_9)
q_gen=(m_ref*h_11)+(m_weak*h_8)-(m_strong*h_7)

t_1=263 [k]
t_1=t_5
t_2=298 [k]
t_3=293 [k]
t_3=t_4
eta_is=0.80;eta_e=0.90
m_dot=0.2973 [kg/s]

h_1=Enthalpy(R134a,T=T_1,x=x_1);x_1=1
s_1=Entropy(R134a,T=T_1,x=x_1)
s_1=s_2
h_2s=Enthalpy(R134a,T=T_2,s=s_2)
h_2=h_2s+(h_2s-h_1)/eta_is
h_4=Enthalpy(R134a,T=T_3,x=x_2);x_2=0
condenser_1=m_dot*(h_2-h_4)
deltaT= t_3-t_14

```

**Parametric Table: Table 1**

	$\delta T$	$T_{14}$	$q_{gen}$
Run 1	15	278	89.79
Run 2	12.5	280.5	85.48
Run 3	10	283	82.54
Run 4	7.5	285.5	80.28
Run 5	5	288	78.4
Run 6			
Run 7			
Run 8			
Run 9			
Run 10			