Major Report

Dissertation on

"THERMODYNAMICS ANALYSIS OF VAPOUR ABSORPTION REFRIGERATION SYSTEM"

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

Master of Technology

In

Thermal Engineering

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2K14/THE/15

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ABSTRACT

A thermodynamic analysis is carried out to study the single effect and double effect series absorption refrigeration technology. The objective of this work is to design a lithium bromide–water (libr-H2O) absorption refrigerator having a capacity of 3.5 kW .Through the application of first law of the thermodynamics the calculations for the heat transfer area of each component is done for the single effect VARS. And the tube sizing and the number of tubes used in the absorber are calculated. The simulation has been made with the help of EES. The results of simulation are to study the effects of the various working parameters such as generator, evaporator, condenser temperatures; effectiveness of the solution heat exchanger on the COP .the outcome can be useful in the design and the performance enhancement of these absorption systems.

CERTIFICATE

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Date:………………

This is to verify that report entitled **" THERMODYNAMICS ANALYSIS OF THE VAPOUR ABSORPTION REFRIGERATION SYSTEM"** by **PRANAV GARG** is the requirement of the partial fulfillment of the award of Degree of **Master of Technology (M.Tech)** in **Thermal Engineering** at **Delhi Technological University** .

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CANDIDATE'S DECLARATION

I hereby declare that the work which being presented in the major thesis entitled "**THERMODYNAMIC ANALYSIS OF THE VAPOUR ABSORPTION REFRIGERATION SYSTEM**" in the partial fulfilment for the award of the degree of Master of Technology in "THERMAL ENGINEERING" submitted to Delhi Technological University (Formerly Delhi College of Engineering), is an authentic record of my own work carried out under the supervision of Dr. Raj Kumar Singh and Dr. Akhilesh Arora, Department of Mechanical Engineering, Delhi Technological University (Formerly Delhi College of Engineering). I have not submitted the matter of this dissertation for the award of any other Degree or Diploma or any other purpose what so ever. I confirm that I have read and understood 'Plagiarism policy of DTU'. I have not committed plagiarism when completing the attached piece of work, similarity found after checking is 1% which is below the permitted limit of 20%.

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<u>NOMENCLATURE</u>

LiBr	**lithium Bromide**
COP	**Coefficient of Performance**
VARS	**Vapour Absorption Refrigeration System**
Tc	**Condenser Temperature (°C)**
Tg	**Generator Temperature (°C)**
Ta	**Absorber Temperature (°C)**
Te	**Evaporator Temperature (°C)**
U	**Overall heat transfer coefficient (W/m2K)**
Xw	**mass fraction of lithium bromide in the weak solution**
Xs	**mass fraction of lithium bromide in the strong solution**
Mw	**mass flow rate of the weak solution (kg/s)**
Ms	**mass flow rate of the strong solution (kg/s)**
Qa	**heat rejected in absorber(kW)**
Qg	**heat supplied in generator(kW)**
Qe	**refrigerating capacity(kW)**
Qc	**heat rejected in condenser(kW)**
LMTD	**Log mean temperature difference((°C)**
HX	**heat exchanger**
Pr	**Prandtl number**
Re	**Reynold number**
Nu	**Nusselt number**
LPG	**Low Pressure Generator**
HPG	**High Pressure Generator**

SYMBOLS

Letters	Description	Unit
m	Mass flow rate	kg/s
h	Enthalpy	kJ/kg
P	Pressure	kPa
T	Temperature	°C
Q	Heat Energy	kW
e	Effectiveness	

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION TO ABSORPTION SYSTEM

Today's world is facing two mainly important ecological problems. They are the energy shortage and global warming. Scientists are operational on how to wipe out these problems. Most of the today's innovations are based on this fact. Lithium-Bromide and water driven absorption refrigeration cycle is a very strong example of this concept, which not only helps in diminishing the fossil fuel usage, hence the minimizes CO2 gas emission but also utilizes the low-grade heat from various sources [8]

The vapour absorption refrigeration cycle or the absorption refrigerator is a closed cycle that uses low grade heat to provide cooling or refrigeration. It is different from the usually used vapour compression refrigerator in the sense that it operates on chemical energy rather than electrical energy. The absorption refrigerator uses a chemical substance as a absorbent which absorbs the refrigerant in the absorber and the waste heat is being used to recover the refrigerant free absorbent and facilitate it to be reused. (Ammonia + water) and (Lithium-bromide + water) are the two commercially used working pairs for this kind of refrigerators with their operability restrictions[8]

These machines also reduce the concerns about lubricants which is mixed with the refrigerants. Absorption system has less efficiency compared to vapour compression system but by increasing the number of effects in absorption systems, the performance can be enhanced[23]. Single effect cycles cannot make efficient use of higher temperature heat source. an increment in the number of effects leads to higher efficiency in the system because it increases the number of times the heating power provided by the heat source that is used in the system.

1.2 LIBR + WATER BASED ABSORPTION REFRIGERATION SYSTEM

The use of LiBr and Water for absorption refrigeration system started about 1930s [8]. The great features of LiBr and water system is the non-volatility of absorbent i.e. LiBr. This eliminates the use of rectifier, which is used in Ammonia and water based absorption refrigeration system. a further advantage is the high Latent heat of vaporization of the refrigerant i.e. water. But the use of water as a refrigerant restricts the use in the low temperature applications. The COP of these kinds of refrigeration systems is higher than the ammonia and water based refrigeration system. The thermodynamic analysis of these systems involves finding main parameters like enthalpy, mass flow rate, coefficient of performance (COP), heat and mass transfer and crystallization in LiBr and Water system.

Refrigerant - absorbent systems should have some desirable properties for vapour absorption cycle. These are as follows-

1. The refrigerant should be more volatile than the absorbent, in other words, the boiling point of the refrigerant should be less than that of the absorbent.
2. There should be huge difference in their boiling points so that it becomes simple to separate the refrigerant from the absorbent in the generator. This ensures that the pure refrigerant flows through the refrigerant circuit (condenser, expansion valve and evaporator)
3. The refrigerant should show high solubility with solution in the absorber.
4. The absorbent should have a high affinity for the refrigerant; this will lessen the amount of refrigerant to be circulated.
5. Operating pressure should be low so that the pipe walls need not to be so strong
6. It should not go through crystallization otherwise, it will obstruct the pipes and flow rates will be changed
7. The mixture should be chemically stable, harmless and inexpensive [4].

1.3. PRINCIPLE OF OPERATION

The working fluid used in the absorption refrigeration system is a binary solution consisting of refrigerant and absorbent. Consider two evacuated vessels are linked to each other. The left vessel contain liquid refrigerant while the right vessel contain a binary solution of absorbent and refrigerant. The solution in the right vessel will absorb the refrigerant vapour from the left vessel, which will reduce the pressure. While the refrigerant vapours are being absorbed, the temperature of the residual refrigerant will decrease as a result of its vaporizations. So a refrigeration effect occurs inside the left vessel. And the solution inside the right vessel becomes more dilute because of the large content of refrigerant absorbed. This is called the "absorption process". Normally, the absorption process is an exothermic process, so, it must reject heat out to the surrounding in order to maintain its absorption potential. Whenever the solution cannot persist with the absorption process because of saturation of the refrigerant, the refrigerant must be removed from the diluted solution. Heat is in general the key for this separation process. It is applied in the right vessel so to dry the refrigerant from the solution. The refrigerant vapour gets condensed by releasing the heat to the surroundings. With these processes, the refrigeration effect can be obtained by using heat energy. However, the cooling effect cannot be produced endlessly as the process cannot be done simultaneously. Therefore, a absorption refrigeration cycle [23] is a combination of these two processes as shown in Fig.1.1. The separation process occurs at elevated pressure than the absorption process, so a circulation pump is required to circulate the solution. COP of an absorption refrigeration cycle is obtained from;

$$COP=\frac{\textit{Cooling capacity obtained at evaporator}}{\textit{heat input in the generator} + \textit{work input in the pump}}$$

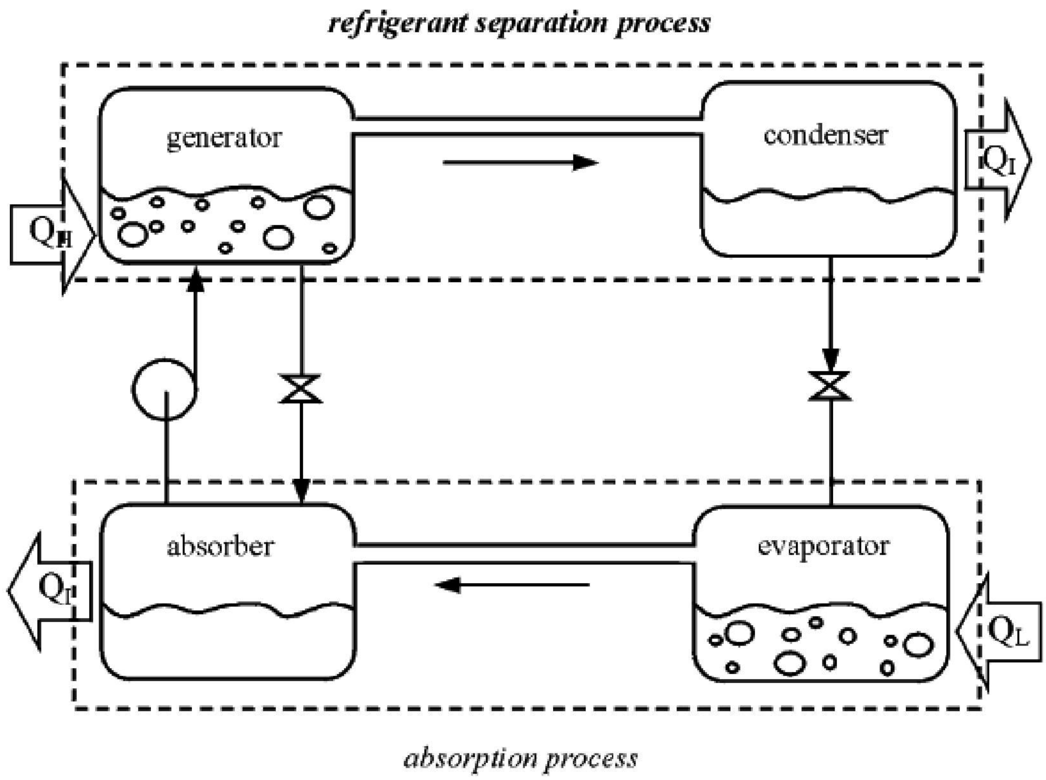

Fig. 1.1 Absorption refrigeration cycle.

CHAPTER 2

LITERATURE REVIEW

In recent years, special attention has been paid to absorption refrigeration systems (ARSs) because of their low cost and environmentally friendly operation. In comparison to conventional mechanical vapor compression refrigeration systems, ARSs require lower energy level, therefore renewable energy sources or heat wasted by industrial processes can be used to operate such systems.

For LiBr/H_2O systems, since water is used as the refrigerant, their application is limited by the freezing point of water and therefore they are usually used in the air-conditioning industry. In contrast, H_2O/NH_3 systems are used in food refrigeration or ice making, since ammonia is the refrigerant. These systems have been studied theoretically [13, 14] and experimentally [7, 8]. In order to improve the performance of absorption systems, new absorbent/refrigerant pairs have been developed. These pairs include LiBr/H_2O-NH_3, LiBr-$ZnBr_2$/CH_3OH and $LiNO_3$-KNO_3-$NaNO_3$/ H_2O , LiBr-$ZnBr_2$/CH_3OH and NH_3/$LiNO_3$. In this Study, LiBr/H_2O pair is used.

The performance of the double-effect absorption chiller has been investigated by many researchers in recent years. Most of the simulation programs in the literature consider operational variables, including temperature and flow rate of cooling water, chilled water and hot water on the cycle performance. However, studies of the effect of design parameters (for example, the solution circulation ratio, and the heat recovery ratio on the COP and the heat transfer area) are rare. Takada [9, 10] studied the effect of solution circulation ratio and the heat recovery ratio on the COP for a single-effect absorption chiller using LiBr/H_2O. The double-effect absorption system offers a considerable improvement in the performance over the single-effect system (1.2 over 0.7 of single-effect system) by effectively making use of the availability of high temperature heat source. The performance analysis of double-effect series using lithium bromide and water absorption system has been made recently by some researchers [1-3]. Most of the analysis in literature considers the influence of operational variables including temperatures and flow rates of cooling water, chilled water and hot water on the thermodynamic performance.

Yokozeki[15] has first done the modeling of vapour absorption refrigeration using equation of states. He has considered various refrigerant-absorbent pairs, mainly two conventionally available pairs i.e. LiBr+water and ammonia+water.

About 85% of the world's energy consumption is derived from the burning/combustion of fossil fuels as primary energy. In tropical countries, more than one third of the total electrical energy generated is consumed in commercial/residential buildings [16-18], and 70% of residential building energy consumption is used by air conditioning systems [18] .Many works were done to evaluate the performances of a solar cooling system working in water- lithium bromide pair, [19, 20]. In 2012 [20] Rosiek, evaluated the performance of a solar-assisted 70 kW single effect LiBr-Water chiller located in Spain and achieved a maximum COP of 0.6.

In 2013, Cascales[12], studied the global modeling of an absorption system working with LiBr/H2O assisted by solar energy A water based vapor absorption refrigeration system with four binary mixtures was examined in the study of Saravanan and Maiya [21]. The variations of various performance parameters were compared for the water based working fluid combinations. Kaynakli and Yamankaradeniz [22] investigated the effect of heat exchangers that are used to recover heat energy in the ARSs on the coefficient of performance (COP).

Srikhirin [23] presented a literature review on absorption refrigeration technology such as various types of ARSs, researches on working fluids and improvement of absorption processes. Kececiler [24] performed an experimental study on the thermodynamic analysis of a reversible ARS using a water– lithium bromide mixture. Joudi and Lafta [25] developed a steady state computer simulation model to predict the performance of an ARS using LiBr/H2O as a working pair.

During recent years, many studies have been conducted by various researchers in thermodynamic and exergy analysis of absorption refrigeration systems. Bejan performed a theoretical analysis of the systems based on entropy generation minimization (Bejan, 1996)[26]. Thermodynamic analysis of LiBr-water absorption system for cooling and heating applications based on first and second law analysis was carried out by Lee and Sherif [26].

Research Gap

As seen that many studies have been conducted by various researchers in thermodynamic analysis of absorption refrigeration systems. And the area calculations have also been done by considering the same value of the overall heat transfer coefficients in all components [1]. But here, the method used to calculate the area of the components using the LMTD and the calculations for the absorber area is done by using how the cold fluid passes inside the tubes and the weak solution flow across the tube bundle And evaluation of the overall heat transfer coefficient using the concepts of the conduction and convection is not done yet.

Formulation of Problem

In the thesis on the vapour absorption refrigeration system, the thermodynamic first law analysis of the single effect and double effect in series configuration using the refrigerant absorbent mixture of H_2O + LiBr is carried out. In which the calculations of the mass flow rate of the solution in the various components is made and the heat interaction with the surrounding and also the calculations of the COP and the solution circulation ratio is done And apart from that the component analysis such as the area calculation of the four main components is done. The design of the absorber is made in details by considering the flow of the coolant in the tubes and the weak solution flow downward across the tubes.

CHAPETER 3

THERMODYNAMIC ANALYSIS OF SINGLE EFFECT WATER/LITHIUM BROMIDE CHILLER

3.1 THEORY:

A single effect absorption system using water/lithium bromide as the working fluid is shown in Fig. 3.1. The main components of the cycle are labeled and the state points in connecting lines are assigned the state point numbers .The fig 3.1 shows the direction of the heat transfer in the four main components (generator, condenser, evaporator , absorber) by an arrows . Ln P – 1/T Diagram for single effect vapour absorption refrigeration system is shown in fig 3.2

Here, the high temperature heat supplied to the generator is used to evaporate refrigerant from the solution (goes out to the surroundings at the condenser) and is used to heat the solution from the absorber temperature (goes out to the surroundings at the absorber). Thus, an irreversibility occurs as heat added at high temperature at the generator is wasted at the absorber and the condenser. In order to reduce this irreversibility, a solution heat exchange is used as shown in Fig. 3.1. The heat exchanger allows the solution from the absorber to be preheated before it enters the generator by means of the heat from the hot solution leaving the generator. Hence the COP is improved as the heat input at the generator is reduced. Moreover, the size of absorber can be compacted as less heat is rejected.

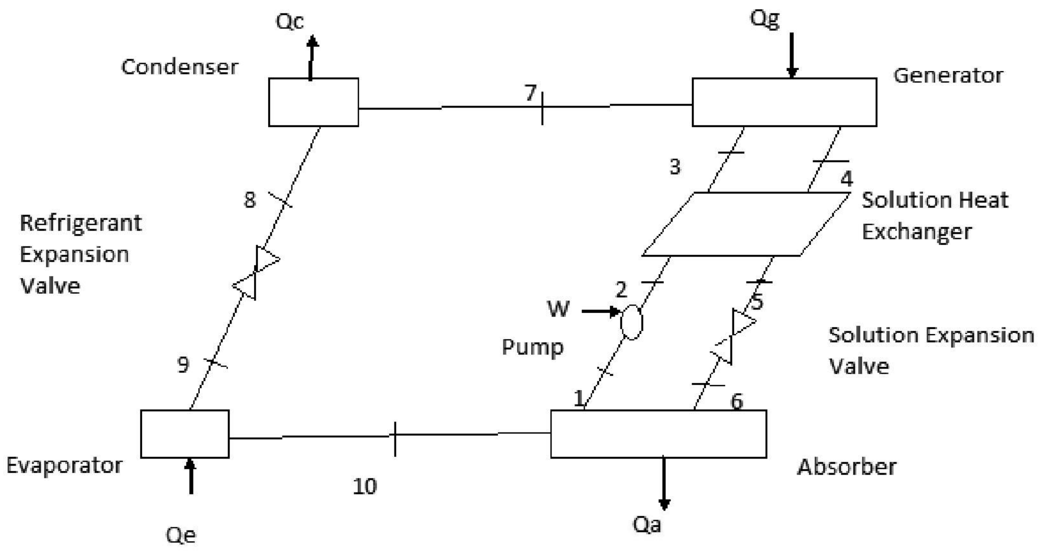

Fig.3.1 single effect VARS

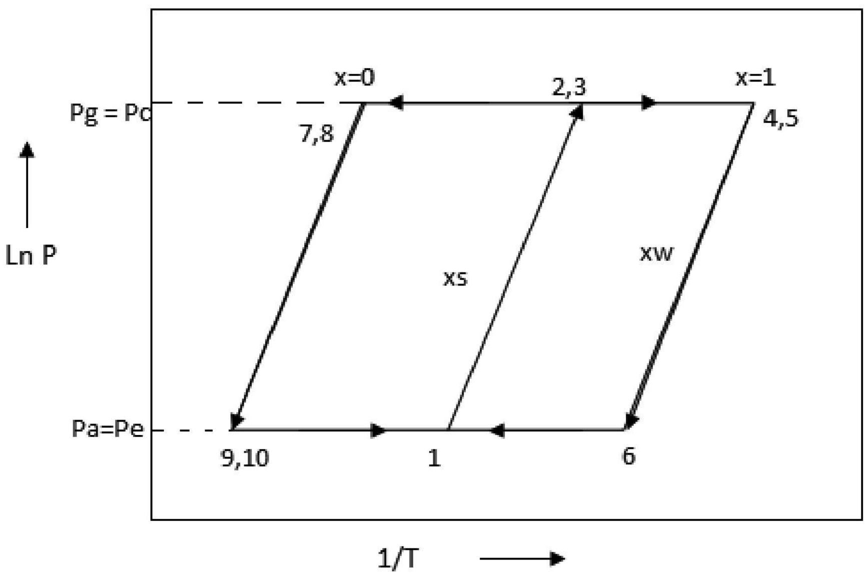

Fig. 3.2 Ln P – 1/T Diagram for single effect vapour absorption refrigeration system

The thermodynamics states of each of the points within the cycle must be clearly understood to well understand the cycle. A summary of these state points description is listed in the table 3.1 .

Table 3.1 Thermodynamic state point summary:

Point	State	Notes
1	Saturated liquid solution	vapour quality set to 0 as an assumption
2	subcooled liquid solution	state calculated from the pump model
3	subcooled liquid solution	state calculated from the solution heat exchanger model
4	Saturated liquid solution	vapour quality set to 0 as an assumption
5	subcooled liquid solution	state calculated from the solution heat exchanger model
6	vapour- liquid solution state	Vapour flashes as liquid passes through expansion valve
7	superheated water vapour	assumed to have zero salt content
8	saturated liquid water	vapour quality set to 0 as an assumption
9	vapour- liquid water state	vapour flashes as liquid passes through the expansion valve
10	saturated water vapour	vapour quality set to 1.0 as an assumption

Mass flow analysis

At the steady state the net mass flow into each one of the component must be zero. Since it is assumed that no chemical reaction occurs between the water and the lithium bromide, the net mass flow of each of the species into any component must be zero. There are two species (i.e. water and lithium bromide) i.e. there are only two independent mass balances.

Energy analysis

The energy analysis of absorption systems involves the application of principles of mass conservation, species conservation, and first law of thermodynamics. The general equations of these principles are specified below:

Mass conservation

$\sum m_i = \sum m_0$

Species conservation

$\sum m_i \, x_i = \sum m_o \, x_o$

Energy conservation

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

where, Q is the heat transfer rate between the control volume and environment and W is the work transfer rate.

The COP of the system is defined by the equation

$$COP = \frac{Qe}{Qg + Wp}$$

3.2 COMPONENTS ANALYSIS

The thermodynamic analysis is carried out using the following assumptions:-

1. Steady state and steady flow
2. No pressure drops due to the friction.
3. Pure refrigerant comes out from generator through the refrigerator circuit in form of vapours.
4. The pump work is isentropic .
5. The flow restrictors (valves) are adiabatic.

3.2.1 Condenser

A liquid state of refrigerant is must in order for the refrigeration process to occur. Hence, the vapour phase of the refrigerant from the generator is changed to a liquid state by the condenser. The condensation process of a high pressure refrigerant vapours is done by rejecting the vapour's latent heat to the surrounding .The heat is rejected to the coolant fluid which enters at point 15 and exits at point 16 .The sub-cooled liquid from the condenser then passes through an expansion valve which lowers the pressure level; a outcome of this process is that some low quantity liquid may flash into vapour. However, the refrigerant can still receive latent heat from the environment.

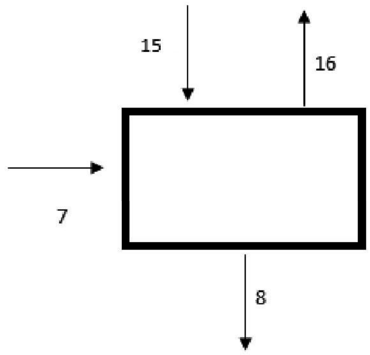

Fig. 3.3 Schematic diagram of Condensor

Energy balance:

Q_c=mr*h_7- mr*h_8

Heat transfer equation:

Qc=m_{15} (T_{16}-T_{15}) = Uc Ac $\Delta T_{lm,c}$

$$\Delta T_{lm,c} = \frac{(Tc-T15)-(Tc-T16)}{Ln((Tc-T15)/(Tc-T16))}$$

In the condenser, there is a desuperheating process and the latent heat transfer process. The value of overall heat transfer coefficient Uc is based on the latent heat process. It needs to be determined experimentally.[6]

U_C = 3200 (W/m^2 K)

Enthalpy (h) of the steam at Tc i.e. at 36°C =2566 kJ/kg

Heat transferred during the desuperheating process =mr*(h_7 –h) =0.1144kW

Latent heat =mr*(h-h_8) =3.58kW

Percentage sensible heat =3.19%

3.2.2 Evaporator

The temperature of evaporation process regulates the lower pressure level of the absorption system. A low pressure two phase refrigerant from the flow restrictor continues to evaporate due to the addition of latent heat from the refrigerating environment. The liquid to be cooled enters the evaporator at point 17 and leaves at point 18 .An entire evaporation process will convert the two phase refrigerant into vapours.

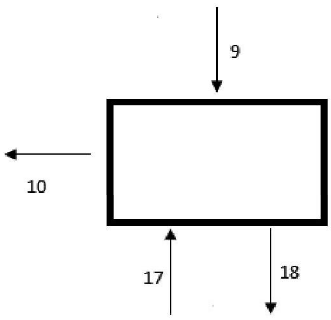

Fig.3.4 Schematic diagram of evaporator

Energy balance:

Q_e=mr*h_{10}- mr*h_9

Heat transfer equation:

$Qe = m_{17}\ Cp\ (T_{17} - T_{18}) == U_e\ A_e \Delta T_{lm,e}$

$$\Delta T_{lm,e} = \frac{(T7 - Te) - (T8 - Te)}{Ln((T7 - Te)/(T8 - Te)}$$

A search of the literature has shown that the preferred construction method is to allow the liquid to enter inside the tube. Water passing through the evaporator tubes supplies the required heat to vaporize the falling film of water around every tube. It is yet not possible to predict all of the characteristics of this process quantitatively because of greater number of variables on which the process depends and the complexity of the two phase flow patterns that occur during the vaporization. So, in this case value of Ue is determined experimentally.

Ue= 190(W/m^2 K) [6]

3.2.3 Generator

The generator operates under the high pressure which is controlled by the temperature of the arriving heat to the generator .The steam enters the generator at point 11 and leaves at point 12. This process generates refrigerant vapours and separate the refrigerant and the absorbent by the addition of the external heat by the heat source; i.e. the desorption of water out of the lithium bromide and water solution. The refrigerant vapours move to the condenser while the liquid absorbent (weak solution) is settled by gravity at the bottom of the generator; the pressure difference between generator and absorber then causes it to flow to an absorber through an pressure reducing valve.

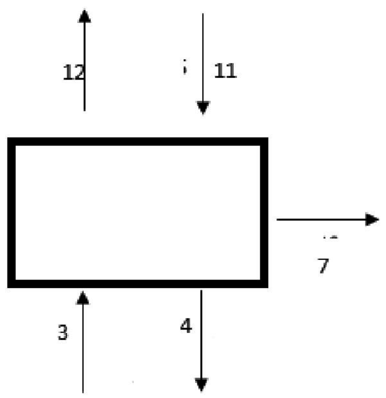

Fig. 3.5 Schematic diagram of generator

Mass balance :

mr+mw =ms

Material balance:

$ms*xs=mw*xw$

Energy balance:

$Q_g + ms*h_3 = mw*h_4 + mr*h_7$

Heat transfer equation:

$Qg = m_{11}\, Cp\, (t_{11} - t_{12}) = U_g A_g\, \Delta T_{lm,g}$

$$\Delta T_{lm,g} = \frac{(T11-Tg)-(T12-T3)}{Ln((T11-Tg)/(T12-T3))}$$

As in case of condenser, here also both the sensible and latent heat transfer occur. So based on the experimental results, the value of the overall heat transfer coefficient in generator is in range of 1600-2300 $W/m^2\, K$

$Ug = 2000\ (W/m^2\, K)$ [6]

3.2.4 Absorber

The absorber is a compartment where the weak absorbent solution and the refrigerant vapour are mixed together. It is provided with a heat rejection system, i.e. bundles of tubes as in the condenser, in which the coolant fluid flow, and operates at a low pressure level which corresponds to the evaporator temperature. The absorption process only occur if the absorber is at a sensible low temperature level, hence the heat elimination system needs to be attached. The mixing process of the absorbent and the refrigerant vapours generates latent heat of condensation and increases the solution temperature. The heat liberated is simultaneously taken away by the coolant fluid then the absorber temperature decreases and, together with the solution temperature, creates a well blended strong solution that is ready for the next cycle. A lower absorber temperature means high refrigerating capacity due to a higher refrigerant's flow rate from the evaporator.

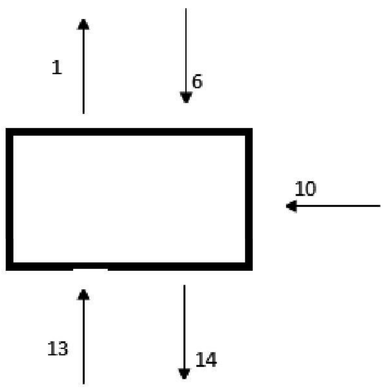

Fig. 3.6 Schematic diagram of absorber

Mass balance :

mr+mw=ms

Material balance:

ms*xs=mw*xw

Energy balance:

Q_a+ ms*h_1 = mw*h_6+ mr*h_{10}

Heat transfer equation:

<u>Steps</u>:

1.Calculate the properties of the coolant at the average temperature. i.e.

Dynamic viscosity , Specific heat , Kinematic viscosity , Prandtl number , density

2.By taking the square pitch arrangement of the tubes , calculate the total flow area required

3.Based on the inner diameter of the tube , calculate the number of tubes.

4.Calculate : Reynold number , Nusselt number , heat transfer coefficient on inner side .

5.Now take the average temperature of the absorbent solution and calculate the properties

Dynamic viscosity , Prandtl number , Nusselt number and heat tranfer coefficient on the outer side

6.Calculate the overall heat transfer coefficient using the inner and outer heat tranfer coefficients and the fouling resistance .

7.Using the equation:

$Qa = U\ a_a\ (LMTD)_a$

Calculate the outer surface area of the tube

8.Total length of the tube = outer surface area of the tube / Perimeter of the tube

9.Length of the tube per pass = total length of tube / number of tubes

<u>Solution Circulation Ratio (SCR):</u>

SCR is an essential design and optimization parameter. It is defined as the ratio of mass flow of the strong solution to the mass flow rate of the refrigerant, i.e.

$SCR = m_S / m_r$

3.2.5 <u>Solution Heat Exchanger</u>

A solution heat exchanger is a heat exchange unit with the function of pre-heating the strong solution before it enters the generator and removing heat from the weak absorbent solution. The heat exchange process within the solution heat exchanger decreases the amount of heat supplied from the heat source in the generator and also reduces the quantity of heat to be rejected to the heat sink (cooling water) in the absorber as well.

3.3 CRYSTALLISATION PROBLEM

Aqueous Lithium Bromide is a salt solution substance and the salt component will start to precipitate when the mass fraction of salt in the solution exceeds the maximum allowable of solution solubility. Since the temperature and the mass fraction of the solution affects the solution solubility more than the pressure, so these two components will affect the crystallization process extensively. Crystallization tends to occur at the outlet of the solution heat exchanger where temperatures of the solution are relatively low and mass fractions are high.

CHAPTER 4

THERMODYNAMIC ANALYSIS OF DOUBLE EFFECT WATER/ LITHIUM BROMIDE CYCLE

4.1 THEORY

Due to the relatively low COP related with the single effect cycles. It is difficult for the single effect systems to compete economically with conventional vapour compression systems except in the low temperature waste heat applications where the input energy is at no cost. Double effect system with the COP in the range of 1.0 to 1.2 is much more competitive.

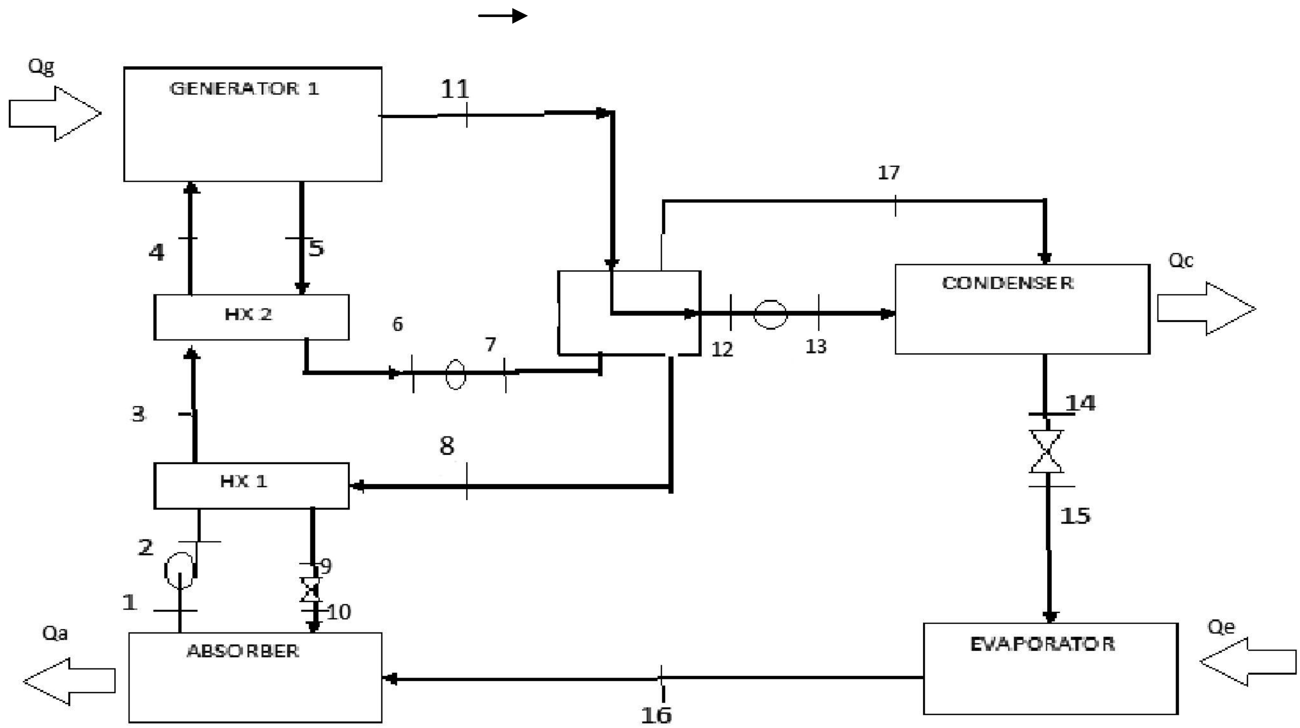

Fig. 4.1 Double effect (series) VARS

In the fig. 4.1 , the external heat transfer interactions are shown by the arrows .Heat is transferred into the cycle in both the high generator (generator 1) and the evaporator. Heat is rejected out from the cycle in the absorber and the condenser. The double effect cycle has two solution heat exchangers that have similar role in the solution circuits as described in the single effect. Fig. 4.2 shows the Ln P – 1/T diagram for double effect vapour absorption refrigeration system The new feature of the double effect (series) is the internal heat exchange between the condenser and the low generator. This internal heat exchange is achieved by using these two components into a single heat transfer device. One side of the exchanger is the high condenser and other side is the low generator (generator 2) .the low generator and the condenser operates at approximately the same conditions as the generator and the condenser of the single effect system. There are the pressure reducing valves between point 6 and point 7 , which reduces the pressure of the weak solution coming through the HX1 from high (generator 1) pressure to the condenser pressure. Similarly , between points 12 and 13 ,there is also a pressure reducing valve which reduces the pressure of the refrigerant in liquid form , from high (generator 1) pressure to the condenser pressure. The heat input in the double effect cycle occurs in the range of (95-145°C) which is much higher than that in the single effect cycle (55-95°C). The COP of the double effects series configuration is more than that of the single effect systems because in the double effect system the availability of the high temperature is used. And in this system the heat is added at the higher temperature. But dump it at about same temperature as that of the single effect and so it is capable to produce refrigerating effect at the same temperature. And other reason for the higher COP of the double effect system is the supplementary amount of refrigerant obtained in the second generator because the refrigerant vapour leaving the high temperature generator condenses and release the latent heat of the condensation which is used to produce the vapours of the refrigerant in the second generator (i.e. exterior heat is not added in the second effect low pressure generator).

Following assumptions are considered for the analysis:

1. The Conditions of the refrigerant (water) at the exit of evaporator and condenser are saturated.

2. The state of the Solution at the exits of absorber, LP generator and HP generator is at equilibrium condition

3.Loss of pressure because of the friction in the pipe lines and heat exchangers are negligible.

4. Heat transfer between the system and the surroundings, other than that agreed heat transfer at the generator, evaporator, condenser and the absorber, does not takes place.

In a double effect series flow configuration vapour absorption refrigeration system, all the refrigerant vapour produced at the high pressure (HP) generator is condensed in the low pressure (LP) generator. This is obtained in realistic systems both at design and at off-design situation by using an orifice, which allow only the condensate refrigerant and restrict the refrigerant vapours from leaving the condenser. The LP generator set itself to an equilibrium temperature to make possible the full condensation process. A information of the equilibrium temperature at LP generator, and other system parameter are necessary to design of the absorption system.

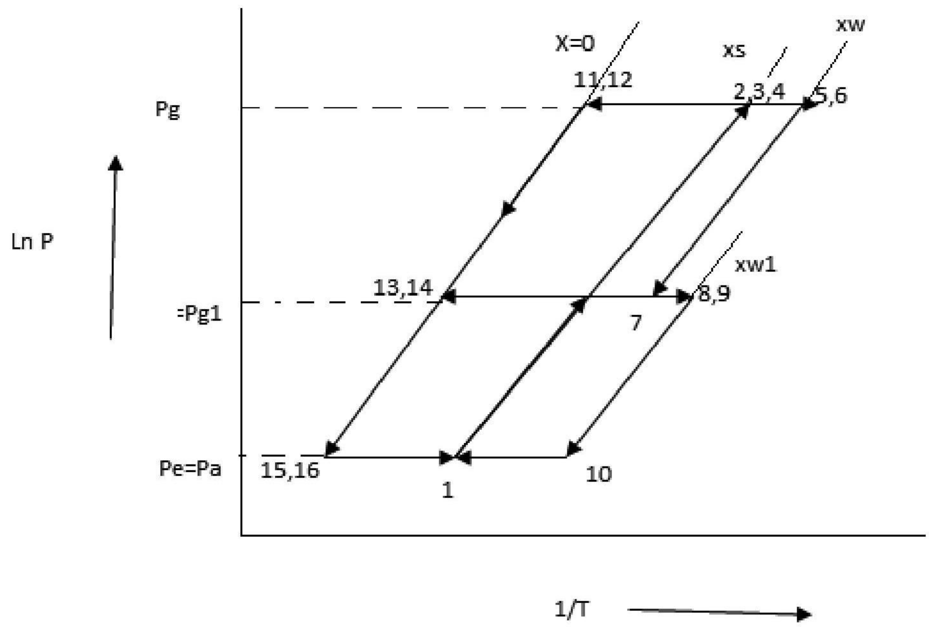

Fig. 4.2 Ln P – 1/T Diagram for double effect vapour absorption refrigeration system

Mass and energy analysis:

4.2 COMPONENT ANALYSIS:

4.2.1 Condenser:

Mass balance :

$mr = m_{17} + m_{11}$

Energy balance:

$Q_c + mr*h_{14} = m_{17}*h_{17} + m_{11}*h_{13}$

4.2.2. Evaporator:

Energy balance:

$Q_e + mr*h_{15} = mr*h_{16}$

4.2.3 Absorber:

Mass balance :

$mr + m_{10} = m_1$

Material balance:

$m_1 xs = m_{10}*xw_1$

Energy balance:

$Q_a + m_1*h_1 = m_{10}*h_{10} + mr*h_{16}$

4.2.4 High temperature Generator :

Mass balance :

$m_1 = m_5 + m_{11}$

Material balance:

$m_1 * xs = m_5 * xw$

Energy balance:

$Q_g + m_1 * h_4 = m_{11} * h_{11} + m_5 * h_5$

4.2.5 <u>Low temperature generator</u>:

Mass balance :

$m_5 = m_{17} + m_{10}$

Material balance:

$m_5 * xw = m_{10}\ xw_1$

Energy balance:

$m_{11} * h_{11} + m_5 * h_7 = m_{11} * h_{12} + m_{10} * h_8 + m_{17} * h_{17}$

4.2.6 <u>Heat Exchanger 1</u>

Energy balance:

$m_1 * h_2 + m_{10} * h_8 = m_1 * h_3 + m_{10} * h_9$

4.2.7 <u>Heat Exchanger 2</u>

Energy balance:

$m_1 * h_3 + m_5 * h_5 = m_1 * h_4 + m_5 * h_6$

4.3 Procedure For Modeling A Double Effect Refrigeration System

The operating parameters are: evaporator temperature Te, condenser temperature Tc, absorber temperature Ta, HPG temperature Tg, effectiveness of heat exchangers (e1,e2) and refrigeration load Qe

⇩

Strong solution concentration, pressure condenser Pc, evaporator pressure Pe, absorber pressure Pa (Pa=Pe) , and LPG pressure Pg1(Pg1=Pc) are calculated.

⇩

Assuming the temperature in the LPG, the concentration of the weak solution leaving LPG is calculated

⇩

Assume initial value of medium solution concentration (xw) and find the pressure in HPG: Pg = f(xw, Tg).

⇩

Find enthalpy and mass flow rate of all points in the system.

⇩

Verify the energy balance for LPG (energy balance in the LPG less or equal to 10^{-4} kW). If energy balance is not occurred to the desired accuracy, increase xw and repeat calculation with the new value of xw, till energy balance occurs across the LPG.

⇩

If energy balance is occurred, to the desired accuracy, calculate energy flow at the various components of the system,

CHAPTER 5

RESULTS AND DISCUSSION

5.1 SINGLE EFFECT VAPOUR ABSORPTION REFRIGERATION SYSTEM

Table5.1 Comparison of results of energy analysis of present work (single effect system) with numerical data given in Anand and Kumar (1987)

SINGLE EFFECT VARS SYSTEMS RESULTS Parameters: Tg=87.8°C , Te=7.2°C , Tc=Ta=37.8°C, effectiveness of solution heat exchanger=0.7 , mass flow rate of refrigerant(water)=1kg/s			
Component	Anand and Kumar (1987)	Present work	Difference
	Q(kW)	Q(kW)	(%)
Generator	3073.11	3085	0.386
absorber	2922.39	2943	0.705
condenser	2507.89	2498	-0.394
evaporator	2357.17	2355	-0.092
COP (dimensionless)	0.76703	0.7635	

5.1.1 Calculations:

Input Parameters:

T_c=36°C

T_e=3°C

T_g=80°C

T_a=35°C

e=0.7

RC =3.5kW

Result:

Circulation ratio =18.15

COP of the VARS cycle = .7176

Circulation ratio =18.15

Lmtd of generator =35.5

Lmtd of condenser =10.55

Lmtd of evaporator =6.166

Mass flow rate of the weak solution = .02548 kg/s

Mass flow rate of the strong solution =.02696 kg/s

Mass flow rate of the refrigerant =0.001486 kg/s

Mass fraction of the libr in strong solution =.5646

Mass fraction of the libr in weak solution =.5975

Heat rejected from condenser =3.704 kW

Heat rejected from absorber =4.674 kW

Heat supplied in the generator = 4.877 kW

5.1.2. Variation of COP with different parameters:

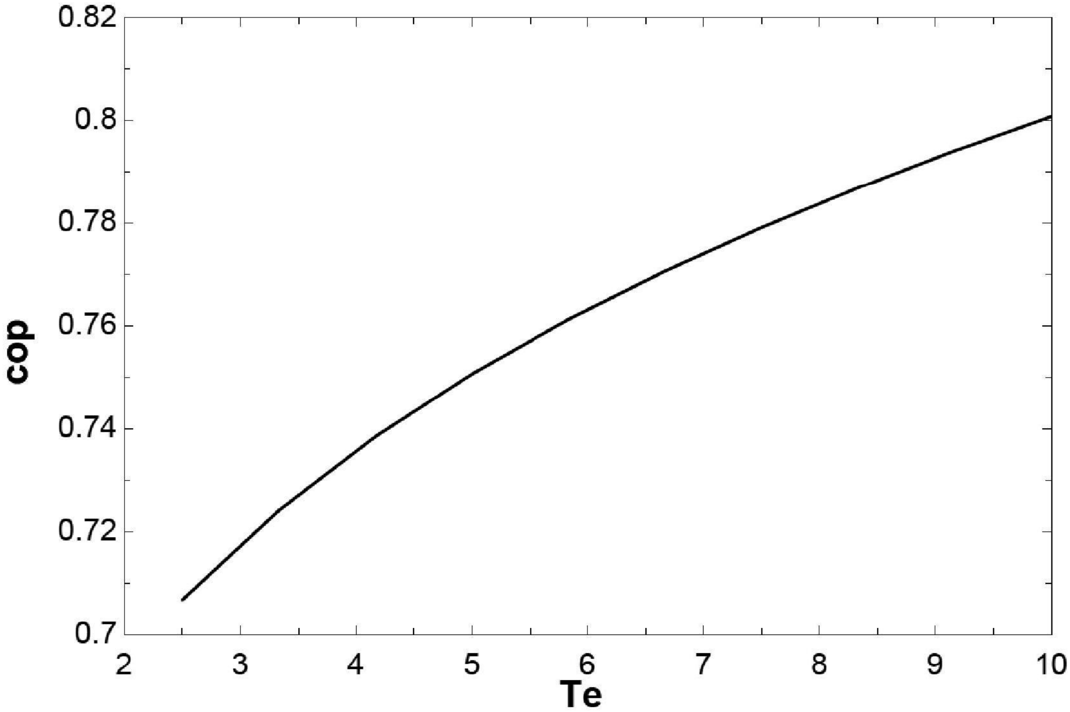

Fig 5.1 Variation of COP with evaporator temperature Te . (Tc=36°C , Tg=80°C , Ta=35°C , e=0.7)

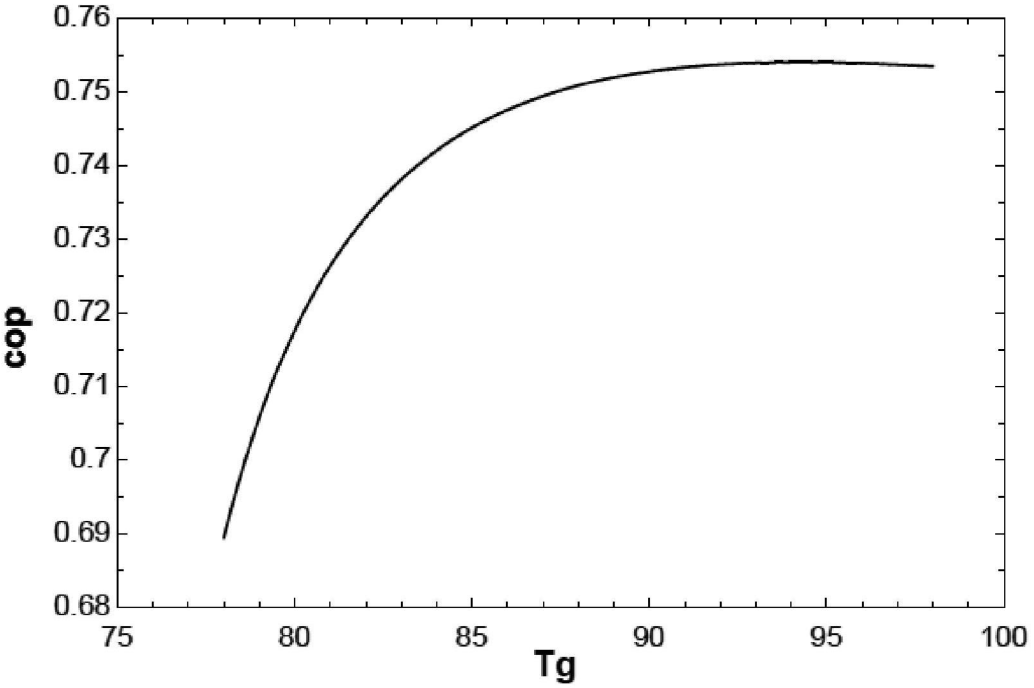

Fig 5.2 Variation of COP with generator temperature Tg (Tc=36°C , Te=3°C , Ta=35°C , e=0.7)

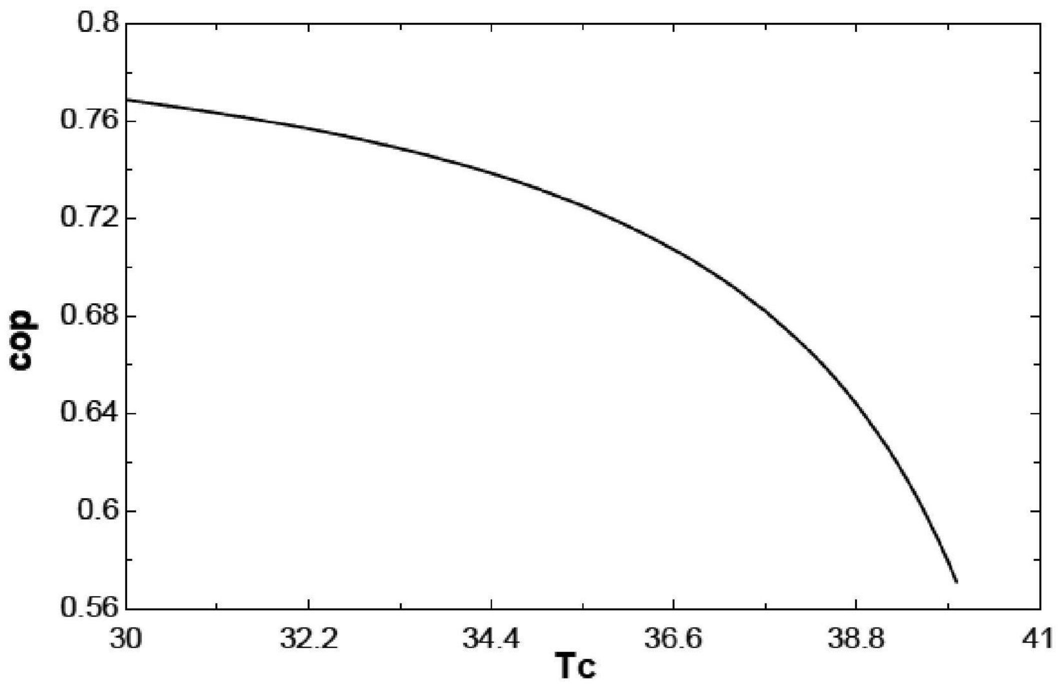

Fig 5.3 Variation of COP with condenser temperature Tc . (Te=3°C , Tg=80°C , Ta=35°C , e=0.7)

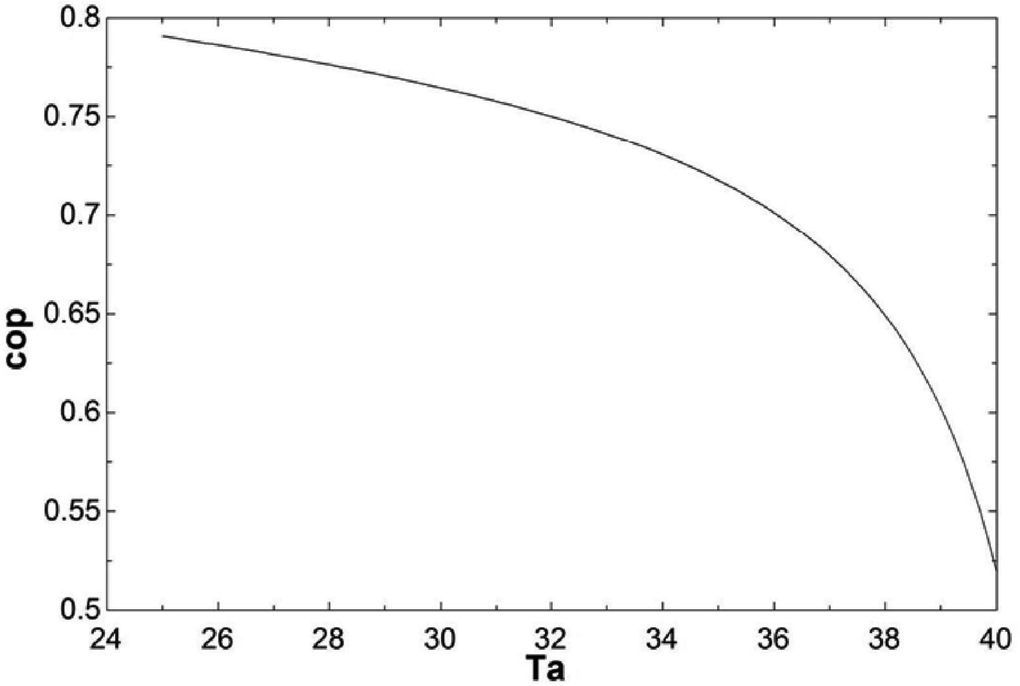

Fig 5.4 Variation of COP with absorber temperature Ta . (Tc=36°C , Tg=80°C , Te=3°C , e=0.7)

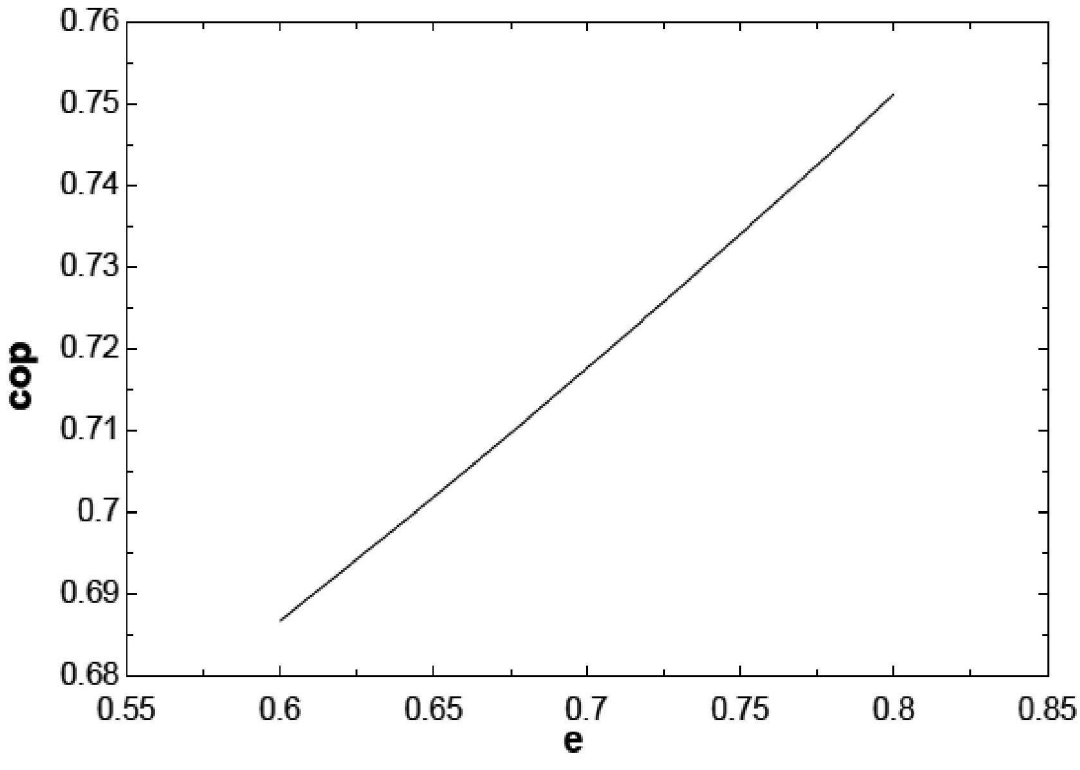

Fig 5.5 Variation of COP with effectiveness of the solution heat exchanger

Effect of variation in evaporator temperature

From the Fig. 5.1, it can be seen that COP of the system increase with increase in evaporator temperature .Here the evaporator temperature range from 2.5 to 10°C. In the variation of COP with the evaporator temperature. When the evaporator temperature increases, the quantity of heat to be extracted is reduced. The consequence is an increase in the coefficient of performance of the system .

Effect of variation in generator temperature

We can see from fig.5.2, the COP of the system increase with the increase in the generator temperature, it implies that the complete system performance goes enhanced when Tg is raised.

Effect of variation in condenser temperature

Fig. 5.3 shows that the COP of the system is decreasing with rising the condenser temperatures. In the condenser, the water vapor required to be cooled for superior condensation. The cooling can be achieved by using cooling towers or natural air cooling systems. For the water-lithium bromide system, the utilization of the water for cooling purpose is more efficient than natural air cooling system because of the main difficulty of crystallization.

Effect of variation in absorber temperature

Fig. 5.4 shows that the COP of the system decreases with increase in the absorber temperature, because the absorption of water by the lithium bromide is a chemical reaction that required to be cooled for better performance, therefore by reducing the absorber temperature will advance the absorption reaction and the COP of the system.

Effect of effectiveness of solution heat exchangers

Fig. 5.5 shows that the system coefficient of performance increases with the increase of the solution heat exchanger effectiveness. The solution heat exchanger helps to raise the strong solution (rich in refrigerant) temperature prior to entering into the generator, which will decrease the amount of input energy supplied in the generator. And also the weak solution gets cooler so it affinity for the refrigerant increases and the amount of the heat liberated in the absorber also decreases by increasing the effectiveness of the solution heat exchanger.so, this improve the COP of the system.

5.2 DOUBLE EFFECT (SERIES) VAPOUR ABSORPTION REFRIGERATION SYSTEM

Table 5.2 Comparison of results of energy analysis of present work (series flow double effect system) with numerical data given in Anand and Kumar (1987).

DOUBLE EFFECT VARS SYSTEMS RESULTS Parameters: Tg=140.6°C , Te=7.2°C , Tc=Ta=37.8°C, effectiveness of solution heat exchangers 1 and 2 =0.7 , mass flow rate of refrigerant(water)=1kg/s			
Component	Anand and Kumar (1987)	Present work	Difference
	Q(kW)	Q(kW)	(%)
Generator	1858.94	1865	0.325
Absorber	2922.39	2945.1	0.777
Condenser	1289.53	1281.2	-0.645
Evaporator	2357.17	2361.3	-0.175
COP (dimensionless)	1.268	1.266	

5.2.1. **Calculations**:

Input Parameters:

Tg=150

Tc=40

Te=10

Ta=35

RC=100kW

e=.7

Result:

COP of the VARS cycle = 1.334

Mass flow rate of the weak solution entering the absorber = .1835 kg/s

Mass flow rate of the strong solution =.207kg/s

Mass flow rate of the refrigerant =0.01906 kg/s

Mass fraction of the libr in strong solution =.5219

Mass fraction of the libr in weak solution entering the absorber =.6569

Heat rejected from condenser =53.67 kW

Heat rejected from absorber =121.4 kW

Heat supplied in the generator = 74.98kW

5.2.2. **Variation of COP with different parameters:**

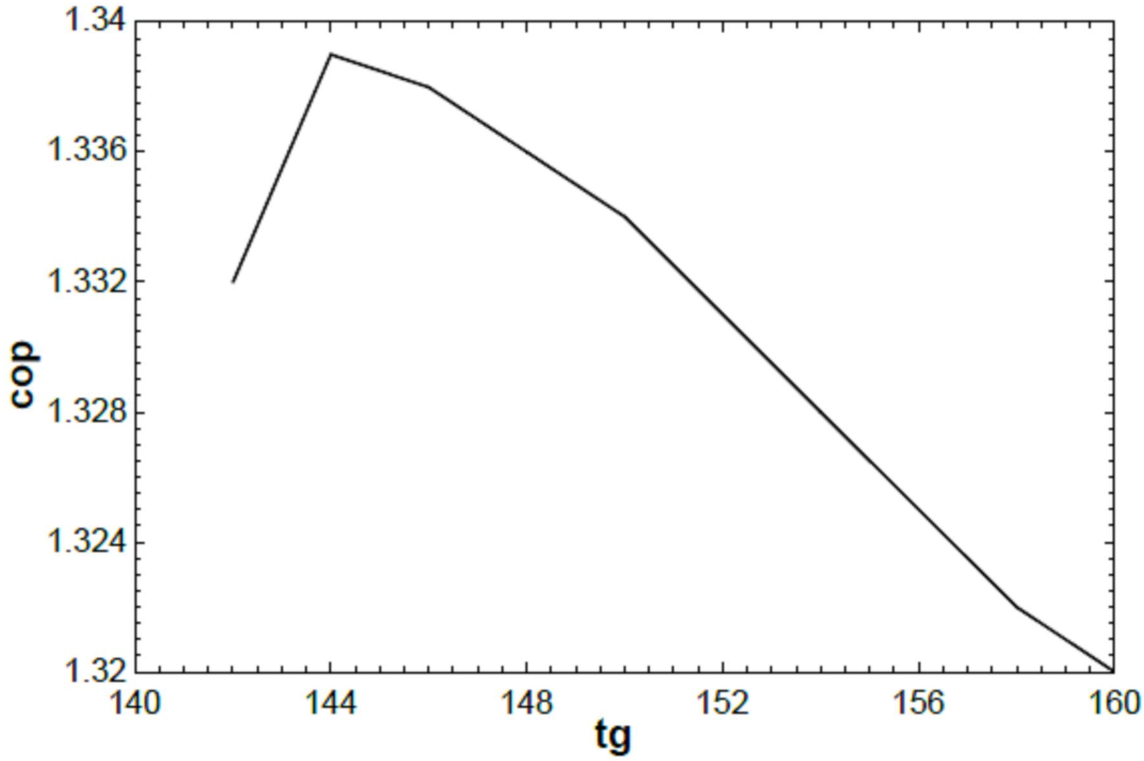

Fig.5.6 Variation of COP with the generator temperature (Tc=40°C ,Te=10°C , Ta=35°C)

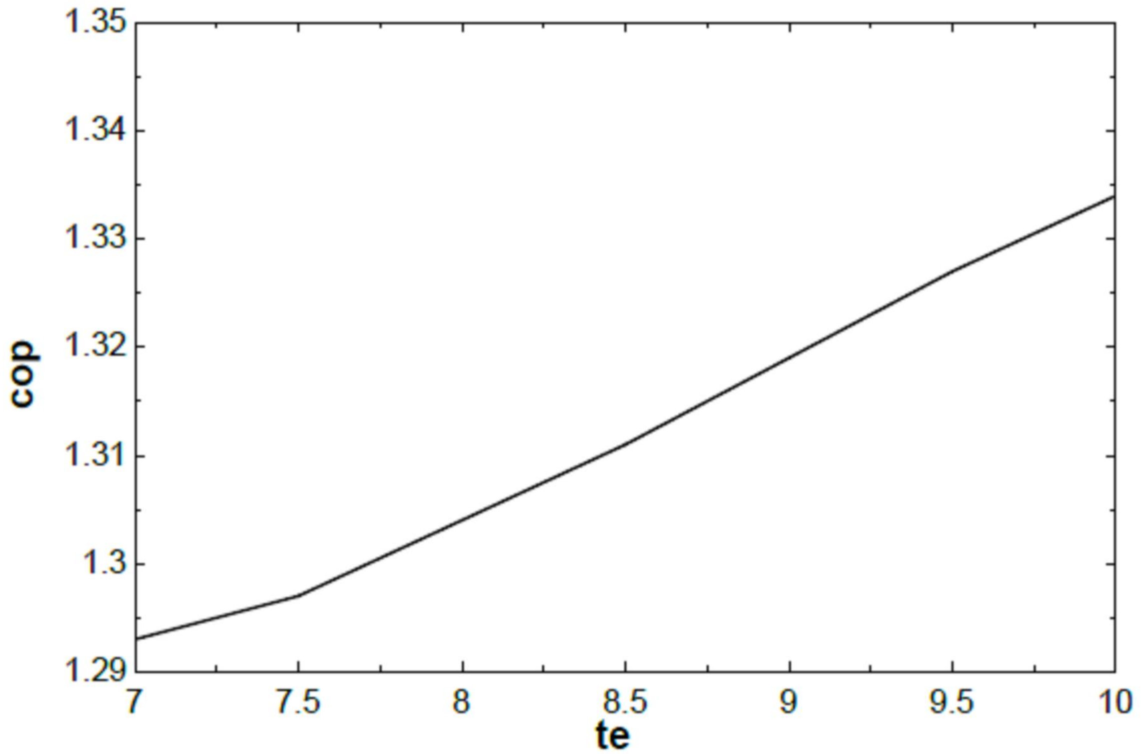

Fig.5.7 Variation of COP with the Evaporator temperature (Tc=40°C ,Tg=150°C , Ta=35°C)

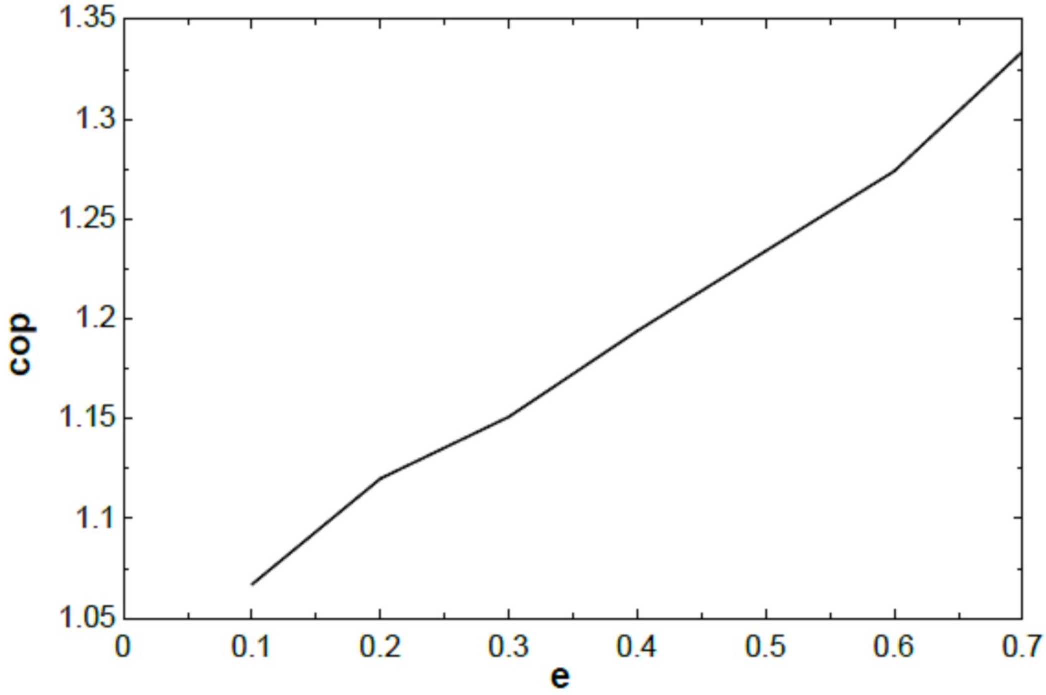

Fig.5.8 Variation of COP with the Effectiveness of the solution heat exchanger (Tc=40°C ,Tg=150°C , Te=10°C)

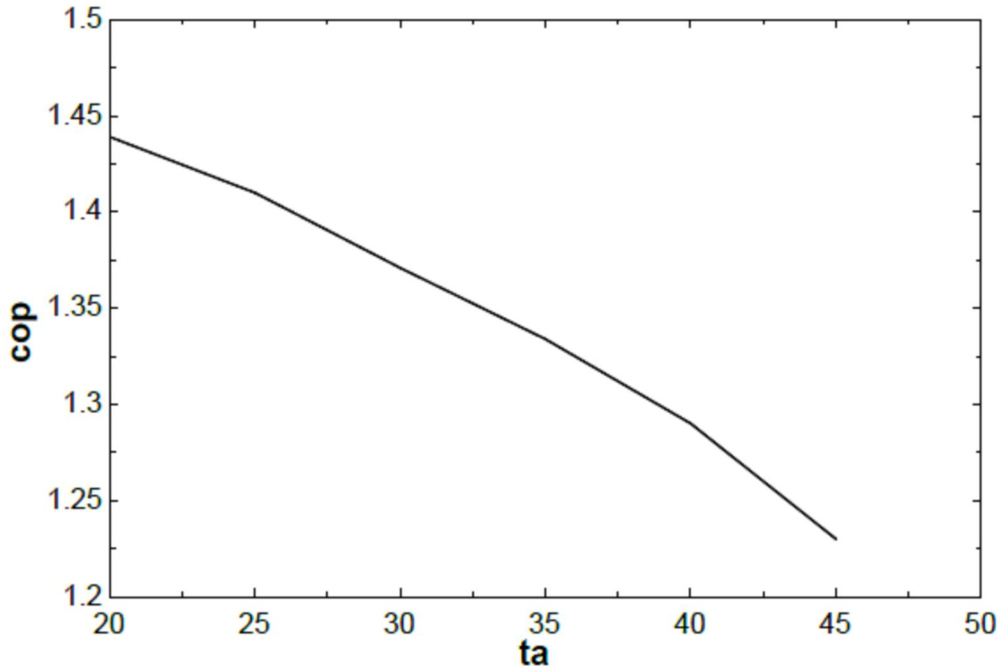

Fig.5.9 Variation of COP with the absorber temperature (Tc=40°C ,Tg=150°C , Te=10°C)

Effect of variation in generator temperature

Fig. 5.6 show the effect of generator temperature on the coefficient of performance (COP) of the system .It is observed that as the increase in generator temperature increases the COP initially ,then COP tends to become more or less constant rather than persist to increase, and with a additional raise in generator temperature even drops rather. The rise in the generator temperature consequences in an raise in temperatures of refrigerant vapour and the weak solution at the exit of the generator which are high than earlier . Thus, increase in the mean temperatures of both the absorber and condenser is observed and this causes the irreversibility get increased in the components. Thus, the constructive effect of raise in the COP by virtue of increase in the generator temperature is counterbalance by the decrease of the COP because of increase in the absorber and condenser temperatures.

Effect of variation in absorber temperature

The result of the variation in the absorber temperature on the COP of the system is shown in Fig.5.9. It is essential to study the effect of absorber temperature on solution circulation ratio (scr) in order to realize its significance. The Solution circulation ratio is defined by the term SCR = $(xw-xw_1)/xs$. With raise in absorber temperature (while keeping the evaporator temperature constant and absorber pressure constant) the concentration 'xs' decreases while concentration 'xw' remains invariable because generator pressure and generator temperature are maintained constant. So the numerator remains same but the denominator decreases and therefore solution circulation ratio increase. The raise in the absorber temperature enhances the solution circulation ratio, which increases the heat duty of the generator and work input in pump. The evaporator load i.e. the refrigerating capacity remains stable since the mass flow rate of refrigerant (H_2O) is assumed to be invariable. Thus the COP decreases with increase in the absorber temperature (Ta).

Effect of variation in evaporator temperature

Fig. 5.7 shows the effect of variation of the evaporator temperature on the COP. The COP of system increases with rise in evaporator temperature.

Effect of effectiveness of solution heat exchangers

Fig. 5.8 shows the effect of variation of the effectiveness of the solution heat exchanger. It is studied that the effectiveness of solution heat exchanger '2' i.e. e2 has a larger effect on COP of the system than the effectiveness of solution heat exchanger '1' i.e. e1. It is also observed that the temperature of the weak solution (rich in LiBr) at the exit of the SHX1 decreases with the raise in effectiveness of solution heat exchanger 1 and the probability of crystallization of lithium bromide salt here in solution get increases.

CHAPTER 6

CONCLUSIONS

Thermodynamic analysis of the single effect and double effect in series vapour absorption refrigeration system is described. We have considered the 3.5 kW capacity of single effect system. And the areas of the four main components are calculated.

In the design of the absorber, I consider that the coolant fluid flows through the tubes and the weak solution flow downward across the tubes. Hence by the calculations we obtain the result that there are 30 tubes of copper each having the length of 51.25 cm.

In the single effect system the effect of the various parameters are described such as the various temperatures and the solution heat exchanger and the concentration of the LiBr. The results obtained are, the COP increases with the increase in the evaporator temperature. As the effectiveness of the solution heat exchanger increases the COP also increases. With the increase in the condenser temperature the value of the COP decreases. As the concentration of the libr in the weak solution increases, the COP initially increases and then becomes constant. And the same result of the COP with the increase in the generator temperature. But with the increase in the concentration of the libr in the strong solution the COP initially decreases and then increases. Similarly various plots are made for the double effect in series vapour absorption refrigeration system with respect to temperatures, concentration, and effectiveness.

The COP of the double effect VARS comes to be greater than that of single effect VARS.

6.1 **SCOPE FOR FUTURE WORK**:

Here I have taken the shell and tube heat exchanger for the area calculations in the single effect VARS system. But in the future, the work can be done on if we take a plate heat exchanger.

The work can also be done on the distribution of the weak solution (entering the absorber) on the tubes bundle such as the nozzle spacing, arrangement of the nozzles and the spray pattern.

Appendix 1 Program for the Single Effect Vapour Absorption Refrigeration System

```
Tc=36
Te=3
Tg=80
Ta=35
q=3.5[kw]                                 " refrigerating capacity"
Pc=P_sat(Steam,T=Tc)                      "higher pressure"
Pe=P_sat(Steam,T=Te)                      "lower pressure"
xw=x_LiBrH2O(Tg,Pc)                       "weak sol."
xs=x_LiBrH2O(Ta,Pe)                       "strong sol."

h4=h_LiBrH2O(Tg,xw)
h1=h_LiBrH2O(Ta,xs)

r=rho_LiBrH2O(Ta,xs)                      " density of strong sol."
v=1/r                                     " specific volume"
wp=v*(Pc-Pe)                              "pump work"
h2=h1+wp

e=.7                                      " effectiveness of sol. HX"
e=(Tg-T5)/(Tg-Ta)                         "EFFECTIVENESS FORMULA"
h5=h_LiBrH2O(T5,xw)
ms*(h3-h2)=mw*(h4-h5)                     " energy balance in solution HX"
h3=h_LiBrH2O(T3,xs)
h7=Enthalpy(Water,T=Tg,x=1)
h8=Enthalpy(Water,T=Tc,x=0)
h9=h8                                     "THROTTLING"
h10=Enthalpy(Water,T=Te,x=1)
h5=h6
mr=q/(h10-h9)                             "mass flow rate of refrigerant"
mr+mw=ms                                  " mass balance in absorber"
ms*xs=mw*xw

qg=mw*h4+mr*h7-ms*h3                      " heat supplied in generator"
qc=mr*h7-mr*h8                            " heat rejected in condensor"
qa+ms*h1=mw*h6+mr*h10
cop=q/qg                                  " cop"
scr=ms/mr                                 " circulation ratio"

"condensor"
t15=25
m15=1
cp=4.183
p=100
qc=m15*cp*(t16- t15)
Uc = 3200
lmtdc =((Tc-t15)-(Tc-t16))/(ln((Tc-t15)/ (Tc-t16)))
qc=Uc*ac*lmtdc                            " calculate ac"

"evaporator"
t17=12
t18=7
q=m17*cp*(t17-t18)                        "calculate m17"
lmtde =((t17-Te)-(t18-Te))/(ln((t17-Te)/ (t18-Te)))
ue=190                                    " assume"
q=ue*ae*lmtde                             " calculate ae"

"generator"
t11=120
t12=95
lmtdg =((t11-Tg)-(t12-T3))/(ln((t11-Tg)/ (t12-T3)))
Ug=2000
qg=Ug*ag*lmtdg
```

SOLUTION

Unit Settings: SI C kPa kJ mass deg

ac = 0.0001097	ae = 0.002988	ag = 0.0000687
cop = 0.7176 [kw]	cp = 4.183	e = 0.7
h1 = 90.04 [kPa]	h10 = 2506 [kJ/kg]	h2 = 90.05
h3 = 147.4	h4 = 193.3	h5 = 132.6
h6 = 132.6	h7 = 2643 [kJ/kg]	h8 = 150.8 [kJ/kg]
h9 = 150.8 [kJ/kg]	lmtdc = 10.55	lmtde = 6.166
lmtdg = 35.5	m15 = 1	m17 = 0.1673
mr = 0.001486 [kw-kg/kJ]	ms = 0.02696 [kw-kg/kJ]	mw = 0.02548 [kw-kg/kJ]
p = 100	Pc = 5.945 [kPa]	Pe = 0.7581 [kPa]
q = 3.5 [kw]	qa = 4.674 [kw-kg-kPa/kJ]	qc = 3.704 [kw]
qg = 4.877	r = 1640	scr = 18.15
t11 = 120	t12 = 95	t15 = 25
t16 = 25.89	t17 = 12	t18 = 7
T3 = 63.66	T5 = 48.5	Ta = 35
Tc = 36	Te = 3	Tg = 80
Uc = 3200	ue = 190	Ug = 2000
v = 0.0006097	wp = 0.003163 [kPa]	xs = 0.5646
xw = 0.5975		

Appendix II Program for the Calculation of an Absorber

xs= 0.5646 (mass fraction of LiBr in the strong solution)

xw=0.5975 (mass fraction of LiBr in the weak solution)

mw=.02548 (mass flow rate of the weak solution)

T_{13}=18.14ºC (inlet temperature of the coolant fluid in absorber)

(keeping the mass flow rate of the coolant fluid and outlet temperature of the coolant fixed , then by the energy balance in the absorber gives the inlet temperature of the coolant =18.14°C)

T_{14}=25ºC (outlet temperature of the coolant fluid in absorber)

m13 =m=1 kg/s (Mass flow rate of the coolant fluid in the absorber)

T=(18.14+25) /2 (Average temperature of the coolant)

Assume di=.282″ = .705 cm (inner diameter of the tube)

do =.5″= 1.25 cm (outer diameter of the tube)

Pt=1.25do (assume square pitch)

Tube pitch is the shortest centre to centre distance between the adjacent tubes. The tubes are generally placed in square or triangular patterns (pitch) [6]

$Pr = \frac{\mu\ Cp}{K}$ (Prandtl number for the coolant fluid)

Assume v =0.9 m/s

Select the number of tubes per tube side pass to give optimum velocity of 0.9m/s-1.1 m/s for liquids [4]

$A = \frac{m}{\rho v}$ (Total flow area required)

$n_t = \frac{A}{0.785\, di^2} = 28.46 \sim 30$ (No. of tubes required)

If $n_t = 30$

$A = 1.171 * 10^{-3} m^2$

$Re = \frac{m\, di}{A\, \mu} = 17993$ (Reynolds Number)

$Nu = .0023\, Re^{.8} Pr^{.33}$ [4] (Nusselt Number)

To calculate the heat transfer coefficient on inner side (hi):

$Nu = \frac{hi\, di}{K}$

$de = 4\,(Pt^2 - .785 do^2)/3.14\, do$ (Equivalent dia. for shell side) [4]

Area of cross flow =As

To = (Ta + T6)/2 (average temperature of the absorbent)

$Re1 = \frac{m\, de}{As\, \mu}$ (Reynolds number of the absorbent)

Pr1 (prandtl number of the absorbent). And Nu1 (Nusselt number of the absorbent) are calculated at the average temperature of the absorbent and the mass fraction of the LiBr in the solution.

$Nu1 = 10^{(.75-.087XW)} Re1^{.8} Pr1^{1.1}$ [11]

S = .000377 m [5] (Average film thickness)

f = .00025 [4] (Fouling resistance)

To calculate the heat transfer coefficient on outer side (ho):

Nu = ho S / K

$$U = \frac{1}{(\frac{1}{hi} + 2R + \frac{1}{ho})}$$ (overall heat transfer coefficient)

$Qa = U\ a_a\ (LMTD)_a$

$(LMTD)_a$ = (T6-T13) -.65 (T6-Ta)-.5(T13-T14) [1]

=25.97

aa =.6058 m^2

$$L = \frac{aa}{3.14\ do}$$ (Total length of the tube)

$$L1 = \frac{L}{nt} = 51.45\ cm$$ (Length of tube / pass)

Appendix III Program for the Double Effect Vapour Absorption Refrigeration System

Input parameters

```
 tg=150
 tc=40
 te=10
 ta=35
 tg2=98.07
 qe=100[kw]
 e=.7
 pc=P_sat(Steam,T=tc)
 pe=P_sat(Steam,T=te)
pg=P_sat(Steam,T=tg)
xw=x_LiBrH2O(tg,pg)
xs=x_LiBrH2O(ta,pe)
xw1=x_LiBrH2O(tg2,pc)
h1=h_LiBrH2O(ta,xs)
r=rho_LiBrH2O(ta,xs)
v=1/r wp=v*(pg-pe)
h2=h1+wp
h5=h_LiBrH2O(tg,xw)
h11=Enthalpy(Water,T=tg,P=pg)
h14=Enthalpy(Water,T=tc,x=0)
h15=h14
h16=Enthalpy(Water,T=te,x=1)
qe=mr*(h16-h15)
```

```
h17=Enthalpy(Water,T=tg2,P=pc)
 h12=Enthalpy(Water,P=pg,x=0)
h13=h12
 h8=h_LiBrH2O(tg2,xw1)
e=(tg2-t9)/(tg2-ta)
 h9=h_LiBrH2O(t9,xw1)
h10=h9
mr+m10=m1
m1*xs=m10*xw1
m5*xw=m10*xw1
m5=m17+m10
m1=m5+m11
m1*(h3-h2)=m10*(h8-h9)
 qg2=m11*(h11-h12)
qg3=m17*h17+m10*h8-m5*h7
h3=h_LiBrH2O(t3,xs)
e=(tg-t6)/(tg-t3)
h6=h_LiBrH2O(t6,xw)
 h7=h6
 m1*(h4-h3)=m5*(h5-h6)
 qg+m1*h4=m5*h5+m11*h11
m11*h13+m17*h17=qc+mr*h14
mr*h16+m10*h10=qa+m1*h1
qadd=qg+qe
qrej=qc+qa
```

cop=qe/qg

scr=xw1/(xw1-xs)

SOLUTION:

h1 = 76.11[kJ/kg]	qe = 100 [kW]	t3 = 63.94°C
h2 = 76.17 [kJ/kg]	qa = 121.4 [kW]	t6 = 89.76°C
h3 = 137.8[kJ/kg]	qc = 53.67 [kW]	t9 = 53.92 °C
h4 = 246.6[kJ/kg]	qg = 74.98 [kW]	tg2 = 98.07 °C
h5 = 331.7 [kJ/kg]	qg2 = 55.51 [kW]	
h6 = 208.9[kJ/kg]	qg3 = 55.52 [kW]	pc = 7.381 [kPa]
h7 = 208.9 [kJ/kg]	qadd = 175 [kW]	pe = 1.228 [kPa]
h8 = 259.9 [kJ/kg]	qrej = 175 [kW]	pg = 94.54[kPa]
h9 = 182.3 [kJ/kg]		
h10 = 182.3[kJ/kg]	xs = 0.5219	mr = 0.04253
h11 = 2776 [kJ/kg]	xw = 0.5887	m1 = 0.207
h12 = 410.9[kJ/kg]	xw1 = 0.6569	m10 = 0.1645
h13 = 410.9 [kJ/kg]	v = 0.000639	m11 = 0.02347
h14 = 167.5 [kJ/kg]	wp = 0.05963	m17 = 0.01906
h15 = 167.5 [kJ/kg]	r = 1565	m5 = 0.1835
h16 = 2519 [kJ/kg]	cop = 1.334	
h17 = 2684[kJ/kg]	scr = 5.356	

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