

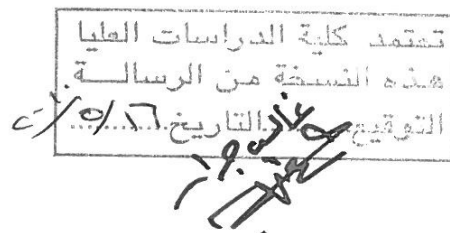
MODELING AND PERFORMANCE STUDY OF EINSTEIN REFRIGERATION CYCLE

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This Thesis was Submitted in Partial Fulfillment of the Requirements
for the
Master's Degree of Science in Mechanical Engineering

Faculty of Graduate Studies
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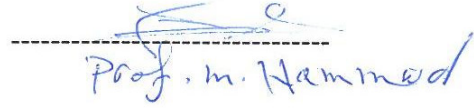
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This Thesis/Dissertation (Modeling and performance study of Einstein refrigeration cycle) was Successfully Defended and Approved on

Examination Committee

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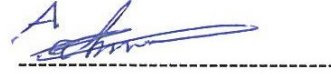
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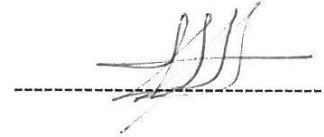
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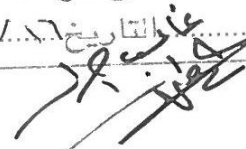

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تعتمد كلية الدراسات العليا
هذه النسخة من الرسالة
التوقيع..... التاريخ 2013/10/10


Dedication

To My Father, Mother

And Brothers

ACKNOWLEDGMENT

I would like to state my thanks to my supervisor Prof. Dr. Mahmoud Hammad for his continuous support and guidance. I had gained a lot of experience through working with him in my master study.

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LIST OF SYMBOLS

Nomenclatures

L	Length (m)	P	Pressure (bar)
A	Area (m ²)	\dot{m}	Mass flow rate (kg/s)
COP	Coefficient of Performance	\dot{Q}	Heat transfer rate (kW)
d	Diameter (m)	R	Universal gas constant (kJ/mol- K)
f	Fugacity (bar)	Re	Rynold's number
g	Gravity (m/s ²)	S	Entropy (kJ/K)
r	Radial location (m)	s	Specific entropy (kJ/kg-K)
H	Enthalpy (kJ)	T	Temperature
h	Height (m)	V	Volume (m ³), or velocity (m/s)
h	Specific enthalpy(kJ/kg)	v'	Volume flow rate (m ³ /s)
Δ	Property of mixing (-)	l	liquid
c _p	Constant pressure specific heat (kJ/kg- k)	sat	Saturated
n	Number of moles (-)	t	Total
x	Liquid mole fraction (-)	v	Vapor
y	Vapor mole fraction (-)	#	State point
ig	Ideal gas	a	Ammonia
is	Ideal solution	b	Butane
w	Water	c	Condenser
e	Ethanol	e	Evaporator
p	Propane	g	Generator
bp	Bubble pump	i	Species i

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MODELING AND PERFORMANCE STUDY OF EINSTEIN REFRIGERATION CYCLE

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ABSTRACT

Einstein refrigeration cycle utilizes butane as refrigerant fluid, ammonia as pressure equalizing fluid, and water as absorbing fluid. The partial pressure of the butane is reduced by ammonia vapor and increased when the ammonia vapor is absorbed by liquid water, thus allowing the butane to evaporate at a low temperature and condense at a higher temperature.

This study calculated the performance of the above working fluids with alternative working fluids, ethanol-water-propane. All fluid mixture properties were modeled with ideal gas, ideal solution, and Vapor Liquid Equilibrium VLE equations.

This study shows that the system pressure should be the saturation pressure of the refrigerant fluid at the temperature of the condenser/absorber.

In this study the first case was using ammonia-water-butane the COP values were around 0.176, while the optimum value was 0.5. The second case using ethanol-water-propane the COP values were around 0.15, while the optimum value was 0.43.

CHAPTER ONE: INTRODUCTION AND LITERATURE REVIEW

1.1 Introduction:

The International Institute of Refrigeration estimated that fifteen percent of all electricity produced worldwide is used for refrigeration and air-conditioning purposes, LUCAS (1998), therefore, nowadays the refrigeration sector is one of the major energy consumer processes in the industry.

This high use of electrical energy makes significant contribution to global warming effect (GWE), directly through the leakage of refrigerant gases (CFC's) and indirectly by large electrical energy used for such systems. Moreover in areas where grid electricity is not available such as villages in rural areas, mountains, deserts, and islands, there is a need for an alternative refrigeration system.

The main two advantages of absorption cycles are no use of CFC's, and no use of electricity . Vapor compression refrigeration cycle systems dominated the cooling industry since the 1930's years. However these systems are characterized by using expensive electrical energy as well as contributing to the environmental global warming effect, (GWE). This study considers the Einstein refrigeration cycle as a single-pressure absorption refrigerator, similar in design to the gas absorption refrigerator. This refrigeration cycle uses pressure-equalizing fluid, refrigerant, and water as absorbing fluid. The Einstein refrigerator is portable, made of inexpensive, nonmoving parts, operates silently, very reliable, and takes advantage of the fact that liquids boil at lower temperatures when the air pressure around them is lower. Dannen, (1997).

If they introduce a new vapor above the (refrigerant) in the evaporator, the liquid boiling temperature decreases and, as it boils off, it takes energy from the surroundings to do so, McCulloch(1990). That's what makes it cold. No moving parts is a real benefit because it can carry on going without maintenance. This could have real applications in rural areas.

Einstein refrigeration cycle was suggested as an alternative cycle that uses a bubble pump which saves energy and absorption refrigeration cycle which consumes thermal energy rather than electricity as power input and not (CFC's) refrigerants.

Improvement of this cycle through performance methods is of high interest for all researchers in this field.

This study aims to develop a mathematical model for Einstein refrigeration cycle and to generate a computer simulation for this mathematical model for the modeled thermodynamic properties.

Through this simulation, the performance for the cycle will be studied and the parameters of the cycle such as the COP, capacity and heating power will be the parameters of the study. They will be treated as dependent variables, while evaporation temperature, condensing temperature, generation temperature and absorption temperature will be the independent variables.

The present work used the working fluid that consists of ethanol-propane-water. In addition, ammonia-butane-water had been also used for the purpose of comparing the obtained results of the present work with that of previous work. This will give confidence about the accuracy of the results of the ethanol-propane-water working fluid.

1.2 LITERATURE REVIEW

Following are examples of literature found published in the same subject of this work. Patek, and Klomfar, 1995, presented a set of five equations describing vapor-liquid equilibrium properties of the ammonia-water mixture. They were intended for use in the design of absorption processes. The equations were constructed by fitting critically assessed experimental data using simple functional forms. They covered the region within which absorption cycles commonly used to operate most often. The enthalpy of the gas phase was calculated in the ideal mixture approximation. The results are presented in the form of an enthalpy-concentration diagram.

Schaefer, 2000, examined the configuration of the Einstein cycle, as heat pump, for heating purposes (for domestic water heaters). Changes were made to increase the coefficient of performance. These changes were primarily implemented on the generator side. The bubble pump performance was enhanced through selection of optimum operating parameters. An external heat exchanger was added between the generator and the condenser/absorber to improve heat recovery. The partial internal heat exchanger of the generator was expanded to a full internal heat exchanger in

order to minimize entropy generation. The Einstein cycle has been modeled using two separate property models:

- 1) A corresponding states/ideal solution property model, and
- 2) A Patel-Teja/Panagiotopoulos and Reid property model.

The first model was used to predict which parameters would increase the COP and the second model was used for more accurate prediction of the behavior of the cycle. The study investigated the COP of the cycle as heat pump over the ranges of temperatures of; evaporation temperature used 22°C to 33°C and the condensing temperature ranged from 43°C to 52°C and used that for domestic water heating purposes.

Qudah, 2002, improved the Einstein cycle by using solar energy. He studied the performance of the cycle; the working fluid used in the cycle was n-butane, ammonia and water. This study resulted in finding model for Einstein using solar energy. The area of the collectors was evaluated due to refrigeration capacity was 76 Kw, condensing pressure, P_c was 4.5 bar and generator temperature $T_{\text{generation}}$ was 350 K, heat quantity needed Q_{needed} was equal 390.3 kW. Area of the collector was 661.2 m². The COP of the cycle ranged from 0.14 to 0.18.

Qenawy, et. al. 2004, investigated the economics of solar powered Einstein refrigeration system for the application of ice production for the fisheries and food industries. Authors explained that an icemaker like this could be used to refrigerate vaccines, meat, dairy products, or vegetables. Solar powered Einstein refrigeration cycle needs solar energy to produce the required cooling effect. Although solar radiant energy is free, the equipment required to convert it to useful cooling is not. The objective of Qenawy's economic study is to determine the economic feasibility of solar powered Einstein refrigeration system. The calculated equivalent annual cost of the system (EAC_{system}) is found to be \$225.174/m² of the solar collector area. The system ice production cost is found to be about 50% more than other conventional refrigeration systems, and the use of the system in rural areas is seen to be essential.

Qenawy, et. al. 2004, powered the Einstein refrigeration cycle by solar energy; to combine the advantages of using solar energy and single pressure absorption refrigeration cycle. A computer program was written to model the thermodynamic properties and the performance of the solar powered refrigeration cycle using the engineering equation solver (EES) software package. As a result of this study, the system pressure is selected according to the condenser coolant temperature. The results showed also that the system pressure had a defined working range. For

ammonia-butane mixture, which is the working fluid of the investigated cycle, the pressure ranged from 4.2 to 9 bars. The cycle had a COP that ranged from 0.1207 down to 0.1025 for that range of pressure. The performance of the cycle was evaluated throughout summer and winter days. Effect of various design and operating parameters on the COP was also investigated. It was found that as the cycle pressure increased the COP increased. Increasing both generator and evaporator temperatures caused the COP to increase. But increasing the condenser temperature decreased the COP.

Mejbri, 2006, modeled the thermodynamic properties of the water– ammonia–butane system; the working fluids mixture used in the Einstein cycle, with the Patel–Teja cubic equation of state was performed. Numerical simulation was used to investigate the feasibility limits of this refrigeration cycle. Two modified configurations of the cycle were considered. A conflict between the evaporator and the condenser/absorber operating conditions was noted. The condenser/absorber operation needed a higher system pressure, which limited the evaporation temperature. On the other hand, the condensation of ammonia and the presence of a small quantity of water in the evaporator limited also the evaporator temperature. In the case of a water-cooled machine, with a condenser/absorber temperature of 30°C and 8°C respectively, the cycle COP reached 0.19 which is considered to be low.

This study used the information in literature to produce a mathematical model and a computer simulation for Einstein refrigeration cycle which is meant to be used as a refrigeration unit. The intended range for evaporation temperature was (-5°C up to 15°C). Performance will be presented graphically. The graphical presentation will be used to find the optimum conditions of the cycle, objective function will be considered to be the COP, or the capacity.

This study is different from other studies by the followings:

- Temperature ranges: In the previous studies, the range of the generator temperature was limited by the heat source used, e.g. Qudah, used the flat plate collectors, and the limit of temperature of his generator was about 70°C in such region, Qenawy, used evacuated tube system, which had higher temperature range which was about 95 °C, and in Sheaefer study, boiler stack exhaust gases supported with electrical heater were used as the generator heat source. In this study the limits of temperature of the generator was selected to be (72 °C, 100 °C). The mathematical model will be carried out only with limit on evaporation range (-5°C to 15°C) to have the maximum COP possible.
- Pressure ranges: In this study, pressure range was selected depending on the condenser/absorber temperature range which has been determined through performing simulation as a part of the performance, and the pressure rang is selected to be 1.8 bar for Ethanol water propane, and 4 bar for ammonia–water–butane while Qenawy studied the cycle in the pressure Range 4.2 - 9 bar ,sheaefer studied the cycle at 4.2 bar , and Qudah studied the cycle at 4.5 bar.
- Mathematical model: In the previous studies, the mathematical models were used to describe the cycle, some of them used the first law of thermodynamics to describe the energy balance, and others used both first and second laws for the balance study. This Study used a software package to optimize the COP for heating purpose.
- The mathematical model and the performance study will be combined to optimize the cycle.

CHAPTER TWO: MATHEMATICAL MODEL FOR THE SYSTEM

2.1 Introduction

In this chapter, the cycle was divided into five separate control volumes: the evaporator, the pre-cooler, the condenser/absorber, the generator, and the internal generator. Next, mass conservation and energy equations were written for each control volume. Each control volume was assumed to operate under steady state conditions with no fluid friction.

2.2 CYCLE DESCRIPTION:

The cycle consists of six components: evaporator, combined condenser/absorber, generator, bubble pump, external heat exchanger and pre-cooler.

Three working fluids are used in the cycle: butane is the refrigerant, ammonia is the pressure equalizer and water is the absorbent (carrier). In the evaporator the partial pressure of butane decreases allowing the butane to evaporate at low temperature. The ammonia vapor is absorbed by liquid water in the condenser the partial pressure of the butane increases to the system pressure allowing it to condense at a higher temperature.

A second working triple combined fluid had been used (ethylene- propane- water) to check for the cycle efficiency (COP), since it is our objective to find the performance for the cycle. The cycle shown in figure (2-1) is modification of Einstein's original configuration.

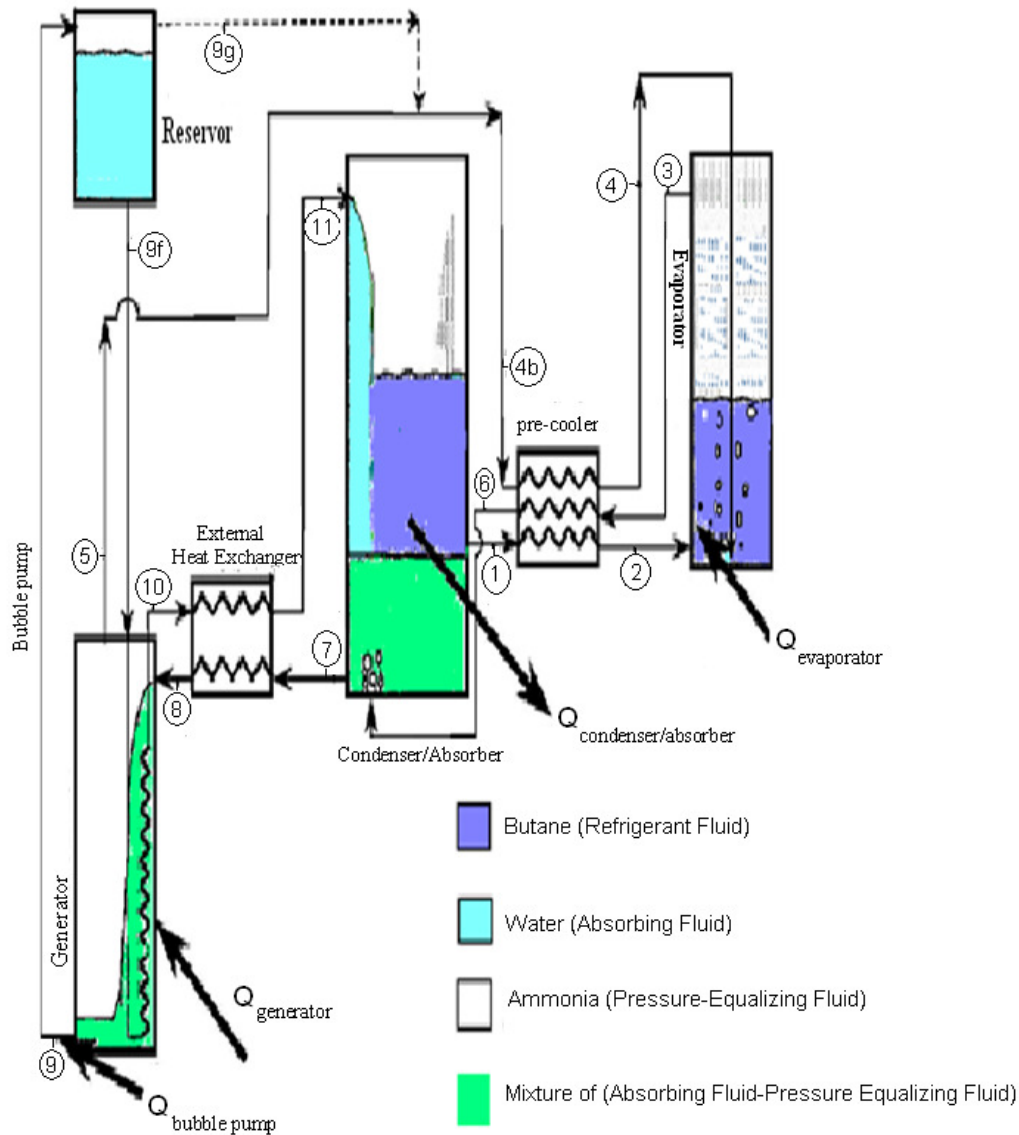


Figure 2-1: Einstein refrigeration cycle: by Sheaefer, (2000)

This cycle consists of two main loops, the right loop is mainly the ammonia and butane cycle, and the left loop is mainly the ammonia water cycle, the right loop is our goal because it has the evaporation condensation processes, required to absorb heat from the environment to be cooled, and to reject heat to outside surrounding.

The detailed processes depend on the physicochemical properties of butane, ammonia mixture, that is:

At the bottom of the evaporator ammonia vapor enters the butane liquid from point (4), so this causes the butane partial pressure to be reduced and this yields to

evaporate at low temperature, the butane vaporization energy is absorbed from the space to be cooled so it is considered to be the cooling load. Vapor ammonia butane mixture leaves the evaporator from point (3) and enters the condenser/absorber at point (6), the mixture faces a fall of water (or some time a spray fall of water), water fall desolves the ammonia only because its physio chemical properties (molecule polarity), so the partial pressure of the butane increases allowing the vapor to condense at high temperature.

The left loop is mainly to separate the water ammonia mixture and to drive the flow. The ammonia water mixture leaves the condenser/absorber point (7), and enters the generator at (8), the heat addition at the bottom of the generator raises the temperature of the mixture, and this yields to reduce the ability of the water to desolve ammonia due to phase equilibrium diagram (liquid vapor- vapor equilibrium states), so as temperature increases, the ammonia concentration in the mixture decrease, thus vapor ammonia leaves the generator at point (5), flow to point (4) at the bottom of the evaporator and the loop goes on.

The water (ammonia water lean mixture) enter the bubble pump, and due to heat addition some water evaporates and due to high rate of bubbles that travel upward, it carries the water upward, leading the cycle flow, then this water enters the condenser/absorber at point (11) as water shower and the cycle continues.

Two heat exchanger were added to the cycle to enhance the performance. That is to have pre-heating or pre-cooling for the substances that will have mixing with other substances and phases so as to minimize irreversibility due to mixing of two different fluids at different temperatures and may have different phases.

The first heat exchanger is the pre-cooler, the vapor ammonia enters the pre-cooler at high temperature point (4b-4), so it is cooled to some low temperature to reduce difference between ammonia temperature at evaporator bottom and butane liquid, this cooling requires semicooled counter flow substance and this is the ammonia butane vapor mixture comes from evaporator point (3-6), and further more the liquid butane leaves the condenser/absorber is cooled before it enters the evaporator to enhance the performance of the cycle.

The second is external heat exchanger, this one raises the temperature of ammonia water mixture coming from the condenser/absorber to reduce the required generator energy, and reduces the temperature of water (ammonia water lean mixture) leaving

the generator at point (10), enters the condenser/absorber at point (11) to lower the condenser/absorber temperature.

2.3 The Evaporator:

In the evaporator, the saturated liquid refrigerant, and the saturated vapor equalizing pressure enters at state point (2) and (4) respectively, both arrive from the pre-cooler in nearly pure form. The saturated vapor equalizing pressure bubbled into the liquid refrigerant, due to this process the partial pressure of the refrigerant is reduced, causing it to evaporate.

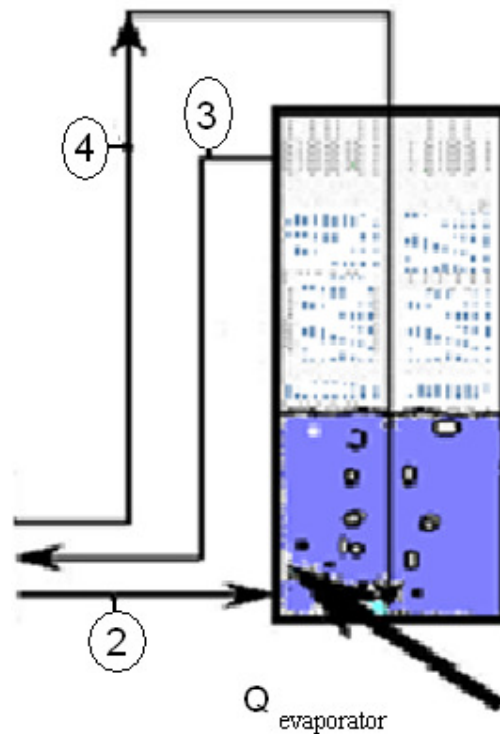


Figure 2-2: The Evaporator

The mass conservation is:

$$\text{(For equalizing pressure fluid)} \quad y_{pe,3} \dot{m}_3 = x_{pe,2} \dot{m}_2 + \dot{m}_4 \quad (2-1)$$

$$\text{(For refrigerant fluid)} \quad y_{ref,3} \dot{m}_3 = x_{ref,2} \dot{m}_2 \quad (2-2)$$

Where y_i represents the vapor mass fraction for species i fluid, while x_i represents the liquid mass fraction for species i fluid.

The conservation of energy for the evaporator is:

$$\dot{Q}_{evap} = \dot{m}_3 h_3 - \dot{m}_2 h_2 - \dot{m}_4 h_4 \quad (2-3)$$

The entropy generation for the evaporator is:

$$\dot{S}_{\text{evap}} = \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 - \frac{\dot{Q}_{\text{evap}}}{T_{\text{evap}}} \quad (2-4)$$

2.4 The pre-cooler:

In the pre-cooler where the stream leaving the evaporator is carrying some cold is heated due to the two streams entering the evaporator. The pre-cooler is assumed to operate under steady state conditions with no fluid friction and is insulated so that the only heat transfer occurs between entering streams and exiting streams.

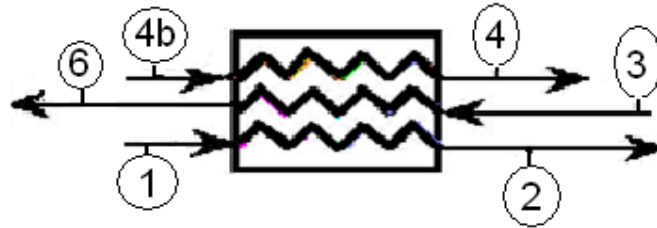


Figure 2-3: The Pre-Cooler

The conservation energy is:

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 + \dot{m}_{4b} h_{4b} = \dot{m}_2 h_2 + \dot{m}_4 h_4 + \dot{m}_6 h_6 \quad (2-5)$$

The entropy generated in the pre-cooler due to the heat transfer across a finite temperature difference is:

$$\dot{S}_{\text{pre-cooler}} = \dot{m}_2 s_2 + \dot{m}_4 s_4 + \dot{m}_6 s_6 - \dot{m}_1 s_1 - \dot{m}_3 s_3 - \dot{m}_5 s_5 - \dot{m}_{9g} s_{9g} \quad (2-6)$$

In this study pinch points are used. In a perfect, counter current flow, two streams heat exchanger, the entering stream and exiting stream on one side would approach the same temperature (the case when a heat exchanger has infinite area). The streams have temperature difference which in this study is defined as pinch point. Since the pre-cooler transfers heat between three streams it requires two pinch points. The first is defined between the vapor mixture entering at point 3 and the liquid leaving at state point 2. The second is defined between the vapor mixture leaving at state point 6 and the liquid entering at state 4b.

$$T_2 = T_3 + \text{pinch}_{2,3} \quad (2-7)$$

$$T_5 = T_6 + \text{pinch}_{5,6} \quad (2-8)$$

$$\dot{m}_1 = \dot{m}_2 \quad (2-9)$$

2.5 The condenser absorber:

From the name, the condenser absorber is combined into one component. Separation of the mixture vapor refrigerant and equalizing pressure fluid occurs in this unit.

The leaving vapor mixture from pre-cooler at state point 6 enters the condenser absorber, while falling film of a liquid absorbing fluid in weak pressure equalizing fluid enters the condenser absorber at point state 11, and then on the pressure equalizing fluid is absorbed by the absorbing fluid in a process that generates heat.

The refrigerant fluid returns to the system pressure, as a result it can condense at its saturation temperature also this temperature is higher than the temperature of the evaporator.

A rich liquid mixture of absorbing fluid and equalizing pressure fluid (water-ammonia) passing through state point 7 in to generator. The refrigerant liquid flows out at point 1.

Mention that the mixture of equalizing pressure and absorbing liquid fluid is immiscible with refrigerant liquid fluid, also it is denser.

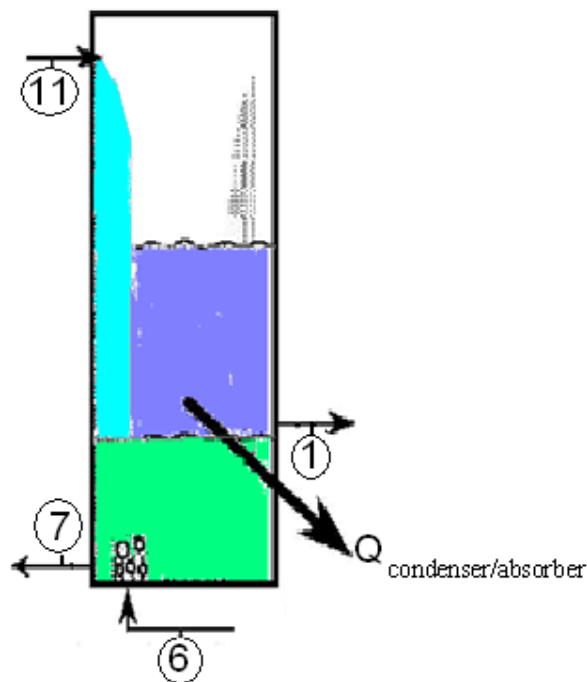


Figure 2-4: The Condenser/Absorber

The conservation of mass for condenser/absorber is:

$$\dot{m}_1 + \dot{m}_7 = \dot{m}_{11} + \dot{m}_6 \quad (2-10)$$

$$x_{i,1} \dot{m}_1 + x_{i,7} \dot{m}_7 = x_{i,11} \dot{m}_{11} + y_{i,6} \dot{m}_6 \quad (2-11)$$

The conservation of energy for condenser/absorber is:

$$\dot{Q}_{\text{cond/abs}} = \dot{m}_1 h_1 - \dot{m}_7 h_7 - \dot{m}_6 h_6 - \dot{m}_{11} h_{11} \quad (2-12)$$

The entropy generation for the condenser/absorber is:

$$\dot{S}_{\text{cond/abs}} = \dot{m}_1 s_1 + \dot{m}_7 s_7 - \dot{m}_{11} s_{11} - \frac{\dot{Q}_{\text{cond/abs}}}{T_{\text{cond/abs}}} \quad (2-13)$$

2.6 The generator:

The main job for the generator unit is to generate the vapor equalizing pressure. The process of generation of vapor pressure equalizing fluid produced here from the rich mixture of equalizing pressure and absorbing fluid arriving from condenser absorber, which then flows to the evaporator, also the liquid absorbing fluid which containing less equalizing pressure fluid where it flows into the bubble pump and returns to the condenser/absorber.

The generated vapor pressure equalizing fluid and the remained pressure equalizing in the reservoir are combined which then flow to the evaporator.

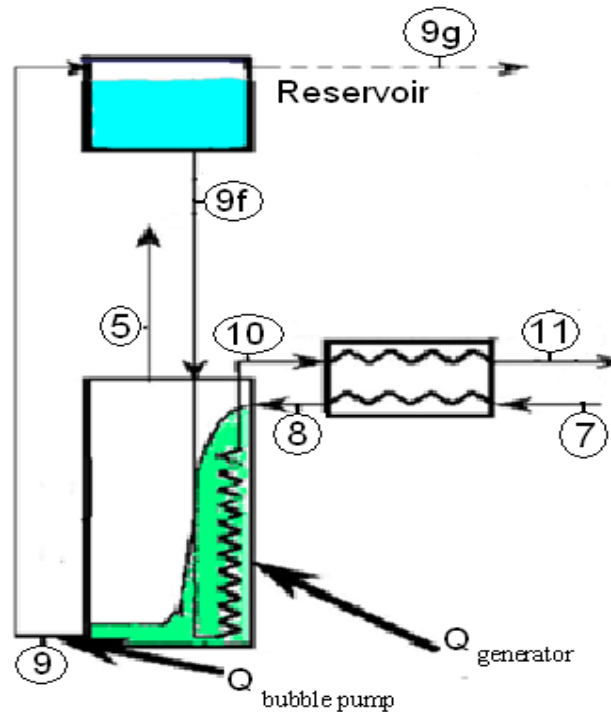


Figure 2-5: The Generator

Conserving mass equation for the generator control volum is:

$$\dot{m}_7 = \dot{m}_5 + \dot{m}_{9g} + \dot{m}_{11} \quad (2-14)$$

For equalizing pressure fluid:

$$x_{pe,7} \dot{m}_7 = y_{pe,5} \dot{m}_5 + y_{pe,9g} \dot{m}_{9g} + x_{pe,11} \dot{m}_{11} \quad (2-15)$$

No mixing occurs in the internal and external solution heat exchangers, so the mass flow rate remained constant in each stream:

$$\dot{m}_7 = \dot{m}_8 \quad (2-16)$$

$$\dot{m}_{9f} = \dot{m}_{10} = \dot{m}_{11} \quad (2-17)$$

Mass conservation equation is needed in addition to the mixture conservation of mass:

$$\dot{m}_7 = \dot{m}_5 + \dot{m}_{9g} + \dot{m}_{11} \quad (2-18)$$

For equalizing pressure fluid:

$$x_{pe,7} \dot{m}_7 = y_{pe,5} \dot{m}_5 + y_{pe,9g} \dot{m}_{9g} + x_{pe,11} \dot{m}_{11} \quad (2-19)$$

To account for the heat transferred in the internal and external solution heat exchangers, additional mass, species, and energy equations are required:

$$\dot{m}_8 = \dot{m}_5 + \dot{m}_9 \quad (2-20)$$

$$x_{pe,8} \dot{m}_8 = y_{pe,5} \dot{m}_5 + x_{pe,9} \dot{m}_9 \quad (2-21)$$

$$\dot{m}_7 h_7 + \dot{m}_{11} h_{11} = \dot{m}_8 h_8 + \dot{m}_{10} h_{10} \quad (2-22)$$

$$\dot{Q}_{generator} + \dot{Q}_{bubble\ pump} = \dot{m}_5 h_5 - \dot{m}_{9g} h_{9g} + \dot{m}_{10} h_{10} - \dot{m}_8 h_8 \quad (2-23)$$

The entropy generated by the generator is:

$$\dot{S}_{gen} = \dot{m}_5 h_5 + \dot{m}_{9g} h_{9g} + \dot{m}_{11} h_{11} - \dot{m}_7 h_7 - \frac{\dot{Q}_{generator} + \dot{Q}_{bubble\ pump}}{T_{gen}} \quad (2-24)$$

The pinch point is:

$$T_{11} = T_7 + pinch_{7,11} \quad (2-25)$$

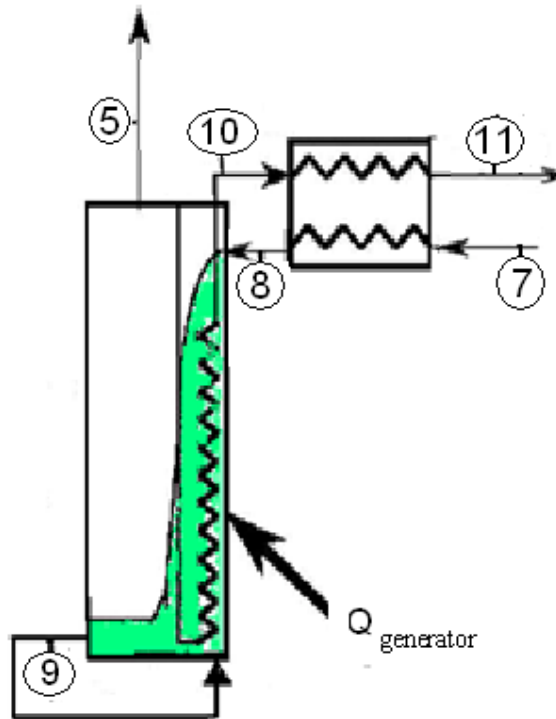


Figure 2-6: Internal and External Solution Heat Exchangers

2.7 Bubble pump:

A bubble pump is heated tube (length L and diameter d), that lifts working fluid from a lower reservoir to a higher reservoir, Figure 2-7. All studies show that:

- The liquid in a lower reservoir initially fills the tube to the same level.
- Heat is applied at the bottom of the tube at a rate sufficient.
- The resulting vapor bubbles rise in the tube carrying the liquid above them to the higher reservoir.
- The bulk density of the fluid in the tube is reduced relative to the liquid in the generator, creating an overall buoyancy lift.
- There are five flow regimes for two phase up flow in a fixed diameter vertical pipe: bubbly, slug, churn, wispy-annular, and annular flow. As the heat that is added to the pipe increases, the two phase flow moves sequentially through these flow patterns. The first stage is bubbly flow, where the vapor exists as discrete bubbles with in the liquid phase.
- In the second stage, the bubbles expand until they are bullet-shaped and nearly span the diameter of the tube.

- These bubbles are separated by (slugs) of liquid, which may contain smaller gas bubbles.
- The next stage is churn flow, where the large gas bubbles begin to break down in oscillatory fashion, so that the center of the tube is alternately filled with the liquid and the vapor phases. The fourth stage, wispy-annular flow, can be difficult to differentiate from annular flow. In the wispy-annular flow, the liquid phase is present both as a film on the wall of pipe and as long filaments of agglomerated droplets dispersed throughout the center core of the flow. The final stage is annular flow, which is characterized by a liquid film surrounding a vapor core (colier,1981)

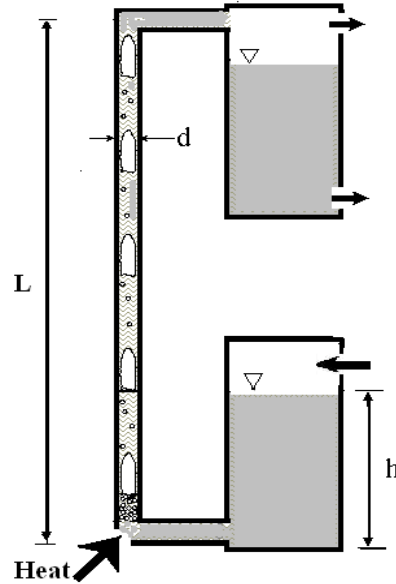


Figure 2- 7: Bubble Pump

Delano, 1998 has shown that a bubble pump operates most efficiently in the slug flow regime (Chisholm, 1983).

$$m_{liq} = 1.22 \times 10^{-2} \cdot Q_{bp} \quad (2-26)$$

$$m_{liq} = 28.6 \cdot m_{vap} \quad (2-27)$$

These two equations describe the bubble pump's performance within the cycle, also improved of 7.9% over Delano's bubble pump efficiency. Sheafer, (1999).

2.8 Second law analysis:

The first and second laws of thermodynamics can be combined to provide a relationship for the ideal COP of a three temperature reservoir heat pump in terms of only the constant temperature reservoir temperatures. Due to irreversibilities such as fluid mixing and heat transfer across a finite temperature difference, the COP_{rev} is degraded to the actual COP.

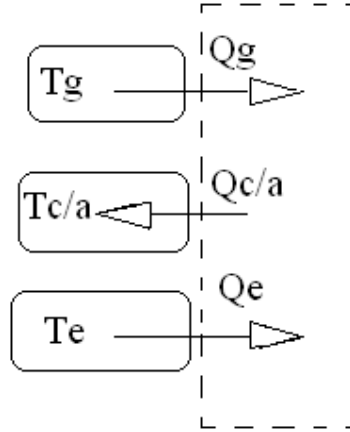


Figure 2-8: Second Law Analysis Diagram

Cooling coefficient of performance is defined as:

$$COP_c = \frac{Q_L}{Q_H} \quad (2-28)$$

The first and second laws of thermodynamics are:

$$Q_{ca} = Q_{evap} + Q_{gen} \quad (2-29)$$

$$\Sigma \dot{S}_i = \frac{\dot{Q}_{evap} + \dot{Q}_{gen}}{T_{ca}} - \frac{\dot{Q}_{evap}}{T_{evap}} - \frac{\dot{Q}_{gen}}{T_{gen}} \quad (2-30)$$

The reversible, or ideal, heating coefficient of performance for a system operating at three temperatures is:

$$COP_{c,rev} = \left(\frac{T_{evap}}{T_{gen}} \right) \left(\frac{T_{gen} - T_{ca}}{T_{ca} - T_{evap}} \right) \quad (2-31)$$

Where:

$$COP_c = COP_c^{rev} - \frac{T_{ca} \times T_{evap}}{T_{ca} - T_{evap}} \frac{\dot{\Sigma}S_i}{\dot{Q}_{gen}} \quad (2-32)$$

Now, the thermodynamic models are ready to be entered into the commercially available software Engineering Equation Solver (EES, Klein, 1997).

CHAPTER THREE: MATHEMATICAL MODEL FOR THERMAL PROPERTIES

3.1 Introduction

To model the Einstein cycles thermodynamically, the working fluids properties: Temperature (T), Pressure (P), Enthalpy (h), Entropy (s), Volume (v), Quality (x and y for mixtures) and that of their mixtures are needed at each state point in the cycle. For pure substances experimental property data exist for all of the states, while such data is not available for their mixtures. Property equations are used to find these mixtures properties before thermodynamically modeling and analyzing the Einstein cycle.

3.2 Property Models

The thermal properties of the working fluids have to be modeled. Reliable data exist on the pure substances; ammonia, butane, ethanol, propane, and water, however, properties of mixtures of these substances need to be worked out. In the Einstein cycle, properties of the mixtures of ammonia and butane as well as ammonia and water are necessary.

Ammonia-water data is readily available. Ammonia-butane, ethanol-water, and ethanol-propane data are quite rare. A literature search did produce one paper with P-T-x data for the ammonia-butane mixture; Unfortunately, the range of pressures was much too high for direct use, Kay and Fisch, (1958). Ideal gas mixtures were assumed for the vapor phases and ideal solutions for the liquids phase.

3.2.1 The Ideal Gas Mixtures

A total thermodynamic property (nU, nH, nC_p, or nS) of an ideal gas mixture is the sum of the total properties of the individual species, each evaluated at the mixture temperature but at its own partial pressure.

For example, the ideal gas (^{ig}) enthalpy of a system comprised of several different chemical species is:

$$H^{ig} = \sum y_i H_i^{ig} \quad (3-1)$$

In other words, the enthalpy change of mixing is zero. Other properties that are independent of pressure can be written similarly, Elliott, (1998).

While the enthalpy change of mixing is zero for the ideal gas mixture, the entropy change of mixing must be some positive quantity since mixing is irreversible. Recall for an ideal gas:

$$S^{ig} = c_p \ln T - R \ln P \quad (3-2)$$

Since the entropy change of an ideal gas at constant temperature depends on pressure, the change of entropy of a species i at a constant temperature T is:

$$dS_i^{ig} = -R d \ln P \quad (3-3)$$

Integrating from the partial pressure p_i to the total pressure P yields:

$$S_i^{ig}(T, P) - S_i^{ig}(T, P_i) = -R \ln \frac{P}{P_i} = -R \ln \frac{P}{y_i P} = R \ln y_i \quad (3-4)$$

So, according to Gibb's theorem, the entropy of an ideal gas mixture at a fixed temperature would be:

$$S^{ig} = \sum y_i S_i^{ig} - R \sum y_i \ln y_i \quad (3-5)$$

Rearranging the above equation as

$$\Delta S_{\text{mixing}}^{ig} = S^{ig} - \sum y_i S_i^{ig} = R \sum y_i \ln \frac{1}{y_i} \quad (3-6)$$

yields the entropy change of mixing for ideal gases. Since $1/y_i > 1$, this change is always positive and in agreement with the second law.

3.2.2 The Ideal Solution

Ideal solution models require pure substance properties data. The only information necessary on the mixture is its composition. Mathematically, the ideal solution model is very simple due to the following assumption:

The formation of an ideal solution (^{id}) results in no change in molecular energies or volumes.

Mathematical interpretation of the above assumption results in the following equations.

$$V^{id} = \sum x_i V_i \quad (3-7)$$

$$H^{id} = \sum x_i H_i \quad (3-8)$$

Similar to the ideal gas mixture, the entropy change of mixing for the ideal solution is:

$$\Delta S_{\text{mixing}}^{ig} = R \sum x_i \ln \frac{1}{x_i} \quad (3-9)$$

And can be shown via methods of statistical thermodynamics. The total entropy of an ideal solution is then:

$$S^{ig} = \sum x_i S_i - R \sum x_i \ln x_i \quad (3-10)$$

3.2.3 Vapor-Liquid Equilibrium and Modeling

In the Einstein cycle, the ammonia-butane and ammonia-water systems appear as a vapor mixture and a liquid mixture together in vapor-liquid equilibrium (VLE). In order to couple the ideal gas and ideal solution models, a criterion for VLE is necessary. This criterion is the equivalency of the fugacity for a particular species in the vapor and the liquid.

$$x_i f_i^l = y_i f_i^v \quad (3-11)$$

For the liquid phase, two simplifying assumptions are made:

1. Assume the fugacity of component i at T and P of the system equals the fugacity of saturated i (liquid or vapor) at the same T , and its corresponding saturation pressure, P_i^{sat} :

$$f_i^l = f_i^{sat} \quad (3-12)$$

2. Next, assuming that pure saturated vapor i at T and P_i^{sat} behaves as an ideal gas the fugacity of saturated i is P_i^{sat} , and equation (3-12) becomes

$$f_i^l = P_i^{sat} \quad (3-13)$$

The only assumption for the vapor phase is that each pure gas component behaves as an ideal gas at T and P , so that

$$f_i^v = P \quad (3-14)$$

Substituting equations (3-13) and (3-14) into equation (3-11) yields Raoult's Law:

$$x_i P_i^{sat} = y_i P \quad (3-15)$$

For a two component system, equation 3-15 together with:

$$\sum x_i = 1 \quad (3-16)$$

$$\sum y_i = 1 \quad (3-17)$$

forms a set of four equations in four unknowns. At a given T and P , this set of equations can be solved to determine the VLE compositions in each phase (P_i^{sat} depends only on T). mixture flow chart diagram shown, blow explain how equations can be solved to determine the VLE compositions in each state point.

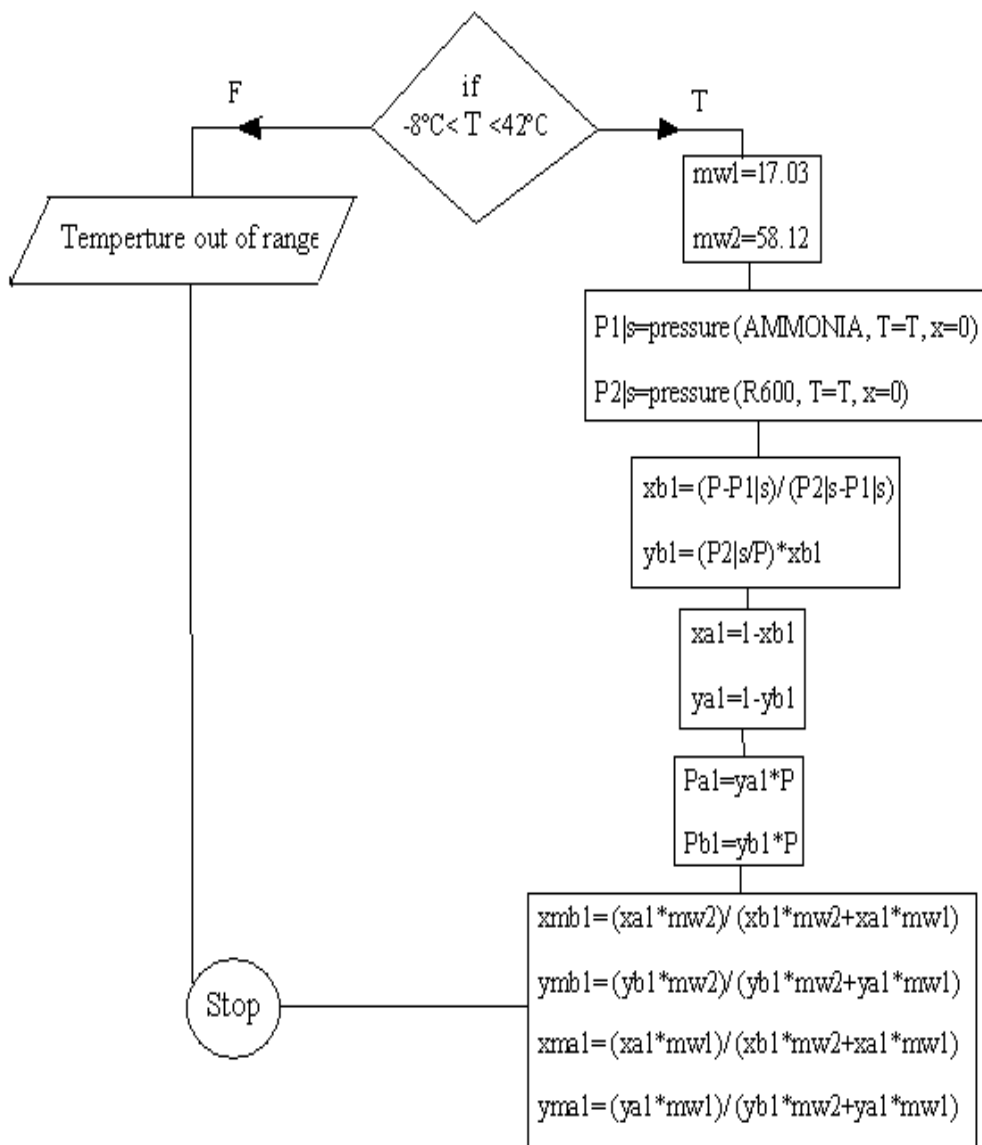


Figure 3-1: Mixture flow chart diagram

Two compositions are required to make vapor-liquid equilibrium VLE calculations. For example, Figure 3-2, shows that when a subcooled 50/50 liquid mixture of ammonia-butane was heated from -8°C (point 1), the first form of gas will be above -3°C (point 2) and the mixture will be in vapor-liquid equilibrium. More heating, the overall composition of both the vapor and the mixture will remain 50/50, but the compositions of the liquid and vapor phases will vary (point 3). Finally, as the last drop of liquid is evaporated, the 50/50 mixture is now a saturated gas (point 4). More heating will superheat the gas (point 5).

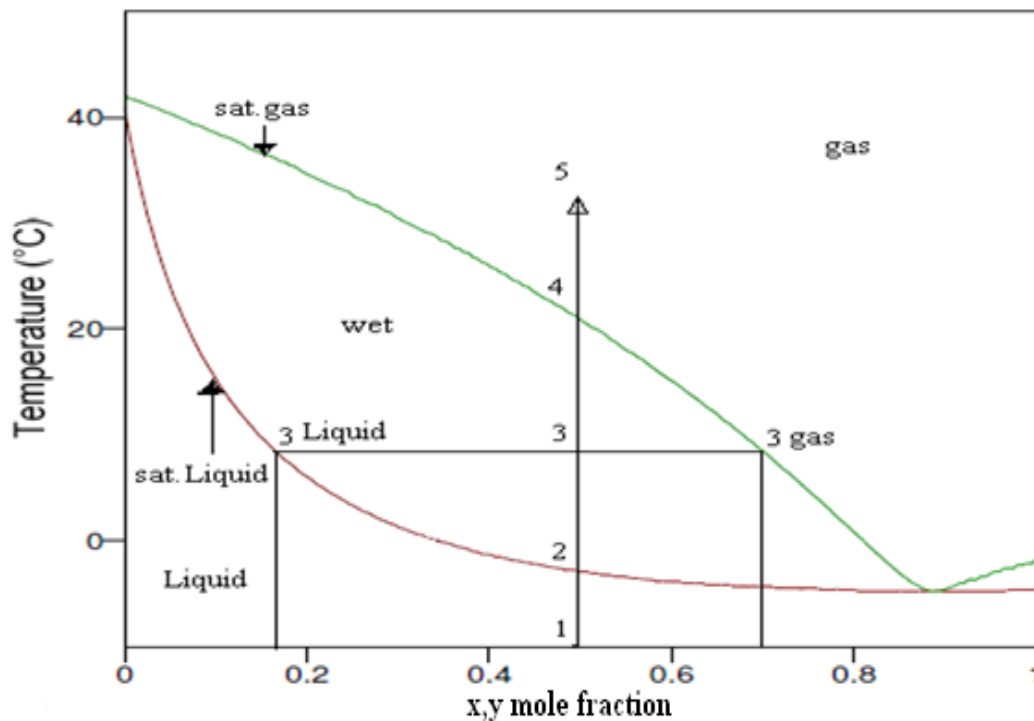


Figure 3-2: Temperature-Composition Diagram for Ammonia-Butane at 4 bar

3.3 Computer Model

After completing the property and thermodynamic models, they were entered into the commercially available software Engineering Equation Solver (EES, Klein, 1997). EES is quite useful, it has property models of many pure substances including ammonia, butane, ethanol, propane, and water are easily accessible in EES, and it's able to handle large systems of equations including the transcendental type found in this model, also the constraints can be easily changed, it's also capable to solve the system of equations while varying a chosen parameter, finally it's capable to plot results.

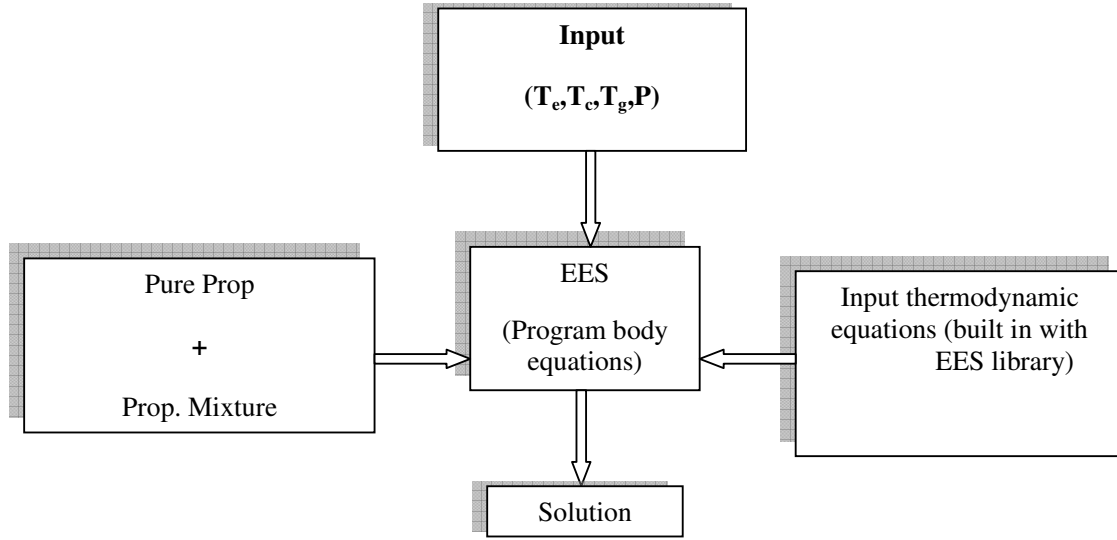


Figure 3-3: Computer Model Flow Chart Diagram

CHAPTER FOUR: SIMULATION RESULTS AND COMPARISON WITH THEORY

4.1 Introduction:

Einstein refrigeration cycle depends mainly on the system pressure, which is the pressure of condenser. The cases study for Einstein refrigeration cycles were chosen to be ammonia-water-butane at pressure of 4bar. While the second case was ethanol-water-propane at pressure of 1.8bar.

For the two cases the temperature of condenser/absorber was suitable to be rejected to ambient air. The results for the two cases show that at fixed system pressure, the characteristics of the ammonia-butane and ethanol-propane mixtures constrain the evaporator and condenser/absorber temperatures to minimum and maximum respectively. This referred to as the maximum lift condition, where lift is the difference between the condenser/absorber and evaporator temperatures.

The pinch points for each heat exchanger were assumed to be zero. Finally, all mass flow rates were normalized to the mass flow rate into the bubble pump, since this flow rate can be controlled by the heat input. With ammonia-water-butane, ethanol-water-propane fluids and evaporator, condenser/absorber, and generator temperatures, zero pinch points, and the flow mass flow of the bubble pump, the refrigeration cycles is fully specified.

4.2 Selection of Operating Parameters:

This Einstein refrigeration cycle was put into work with two different mixtures. Mixtures chosen for these studies were: first, ammonia-water-butane at: system pressure of 4bar; for condenser/absorber temperature of 42°C, evaporator temperature of 0°C, and generator temperature of 72°C. Second mixture was ethanol-water-propane. Two setting of temperature for the condenser/absorber were chosen, and they were 42°C, and 45°C. The VLE behavior of the ammonia-butane mixture and ethanol-propane are shown in figure 4-1 and 4-2, respectively.

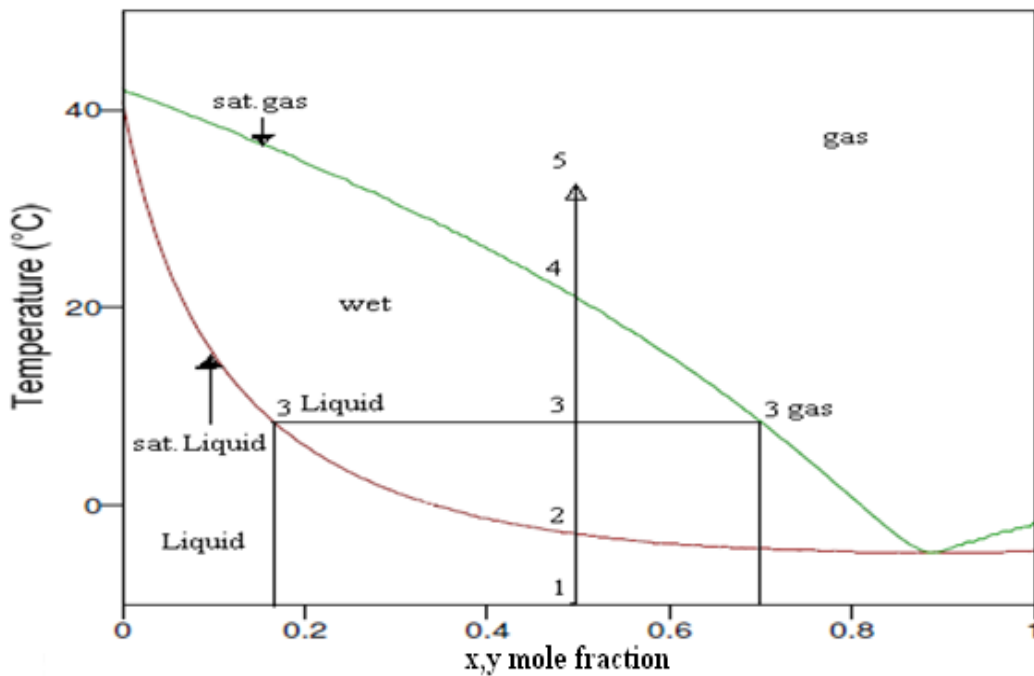


Figure 4-1: T-x,y Diagram for Ammonia-Butane at P = 4 bar

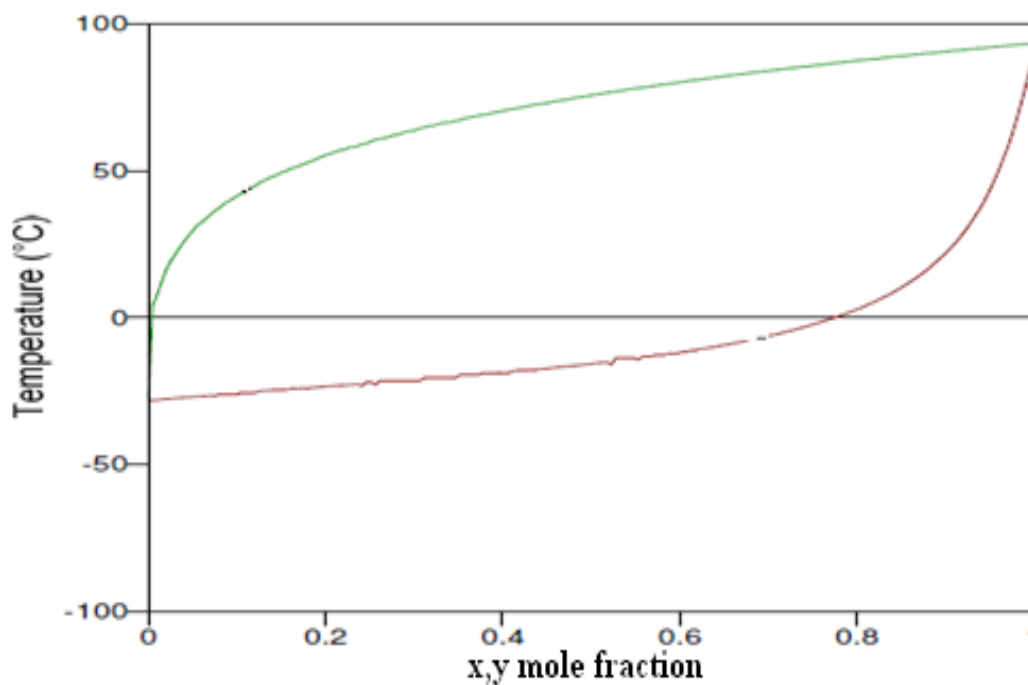


Figure 4-2: T-x,y Diagram for Ethanol-Propane at P = 1.8 bar

As seen in figure 4-1 and 4-2, pure refrigerants at a pressure of 4 bar for ammonia-water-butane, and 1.8bar for ethanol-water-propane condenses at 42°C, and 45°C. The addition of equalizing pressure allows the mixture to boil as low as -8°C for ammonia-water-butane, and -5°C for ethanol-water-propane , respectively, therefore, the system pressure was chosen to be 4bar for ammonia-water-butane, and 1.8bar for ethanol-water-butane. Therefore, at this pressures -8°C, and -5°C are used for the evaporator temperature.

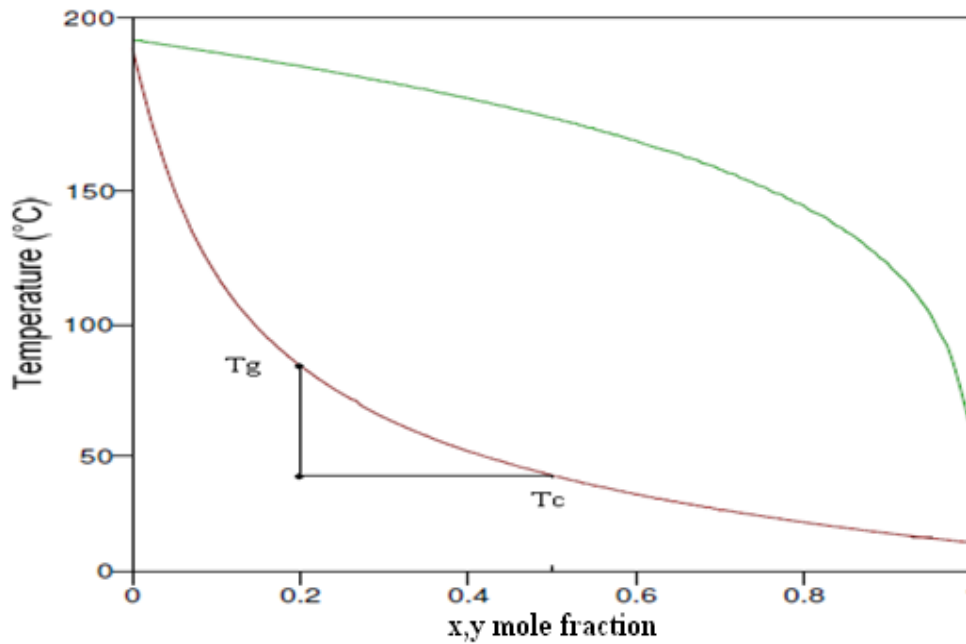


Figure 4-3: T-x,y Diagram for Ammonia-Water at P = 4 bar

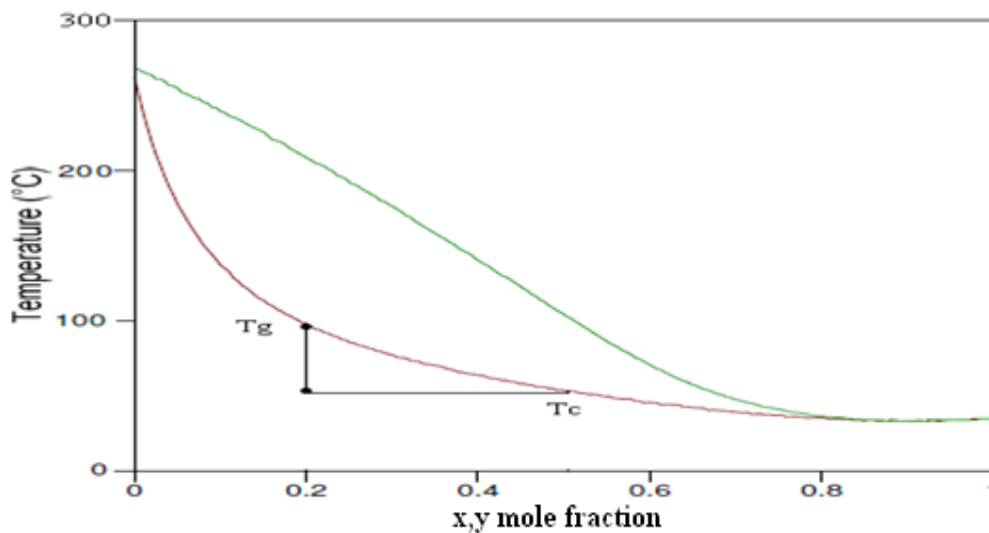


Figure 4-4: T-x,y Diagram for Ethanol-Water at P = 1.8 bar

To determine a generator temperature, the behavior of the ammonia water mixture and ethanol-water mixture at the system pressure, as shown in figure 4-3 and 4-4, is necessary. To generate equalizing pressure vapor, the nearly 50/50 mixtures flowing from the condenser/absorber at 42⁰C and 45⁰C respectively, are heated, driving off mixtures vapor. Heating the mixtures to 72⁰C and 100⁰C reduces the mass concentration of pressure equalizer in the liquid to under 0.2 and doesn't generate too much water vapor. Therefore, the generator temperature will be 72⁰C for ammonia-water-butane and 100⁰C for ethanol-water-propane.

Figure 4-1 and 4-2, shows that at a fixed system pressure, the characteristics of the ammonia-butane mixture, and ethanol-propane mixture constrain the evaporator and condenser/absorber temperatures to a minimum and a maximum respectively. In this study, these temperatures are often used for calculations. When these temperatures are used for a calculation, it will henceforth be referred to as the "maximum lift" condition where lift is the difference between the condenser/absorber and evaporator temperatures.

4.2.1 Effect of the system pressure:

The maximum lift is plotted in figures 4-5 and 4-6, for Ammonia-Water-Butane, and Ethanol-Water-Propane, respectively. As seen in the figures as the system pressure increases the lift also increases. This is due to the increase in the difference between the saturation temperatures of the pure refrigerant liquid and the three phase flash temperature of the mixture with increase in pressure in Fig4-7 and Fig4-8, the pressure is varied for fixed lift condition when the condenser absorber temperature is set to 42⁰C , and generator temperature of 72⁰C , and evaporator temperature of 0⁰C for Ammonia-Water-Butane, and 45⁰C and generator temperature of 100⁰C , and evaporator temperature of 0⁰C for Ethanol-Water-Propane. Both figure shows that when the pressure was increasing the COP was decreasing.

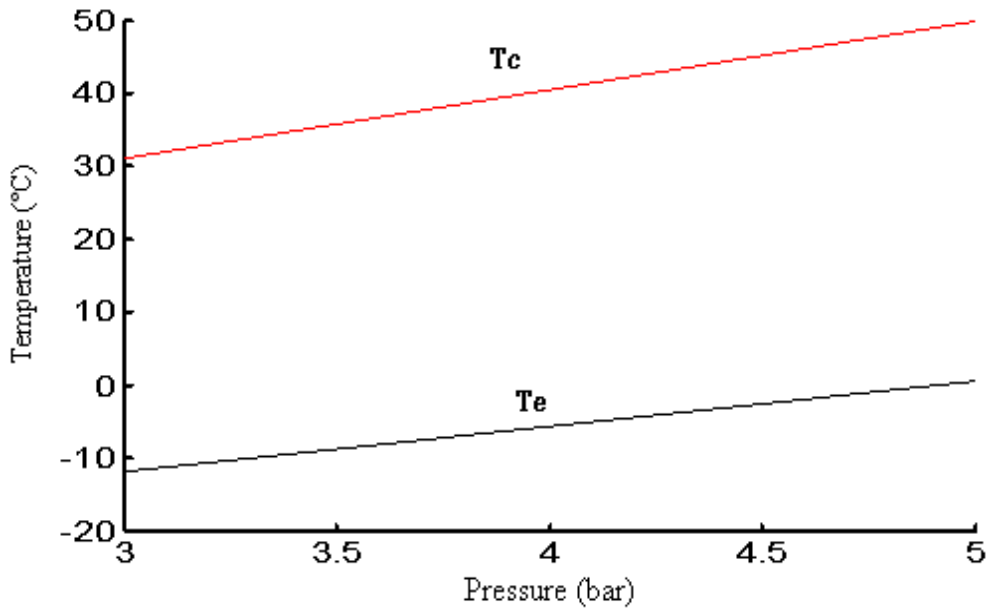


Figure 4-5: Maximum Lift Temperature vs. System Pressure for Ammonia-Butane

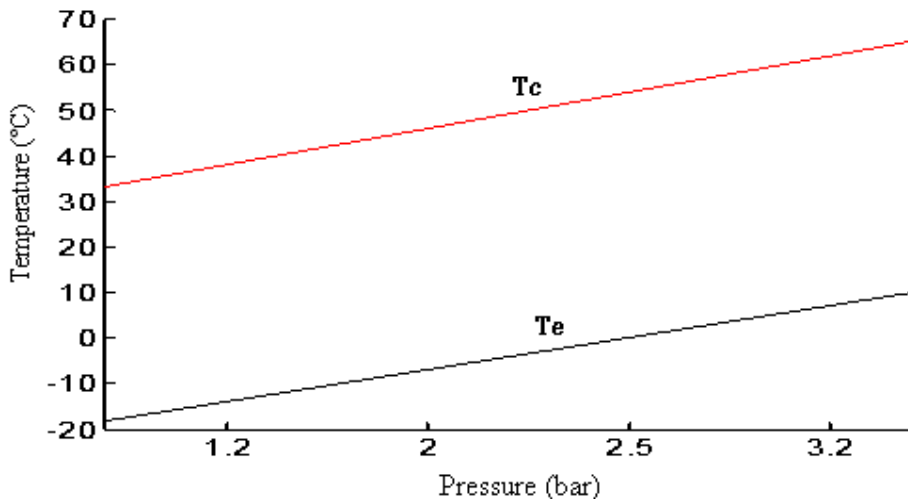


Figure 4-6: Maximum Lift Temperature vs. System Pressure for Ethanol-Propane

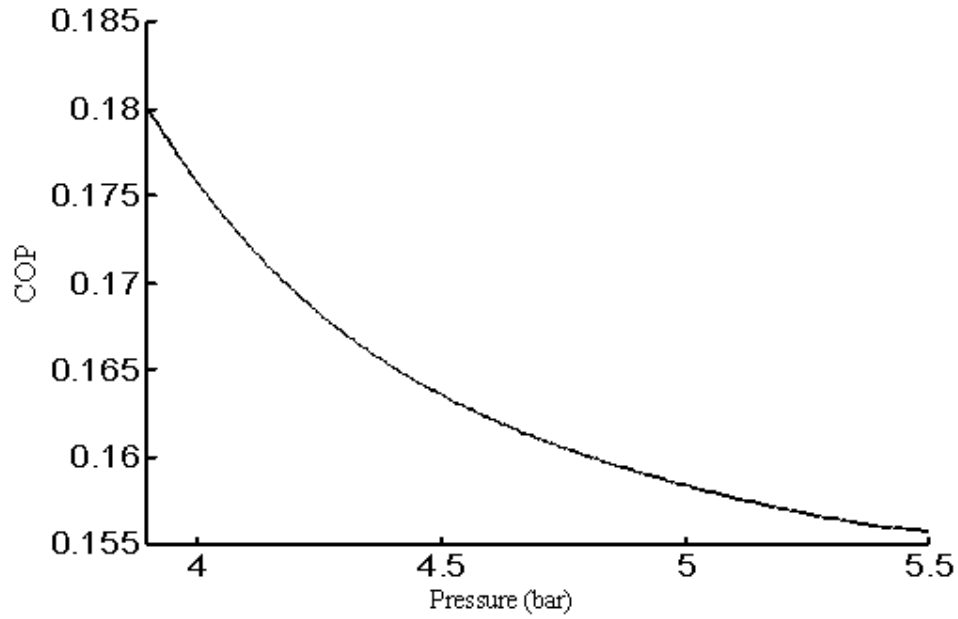


Figure 4-7: COP vs. System Pressure for Ammonia-Water-Butane

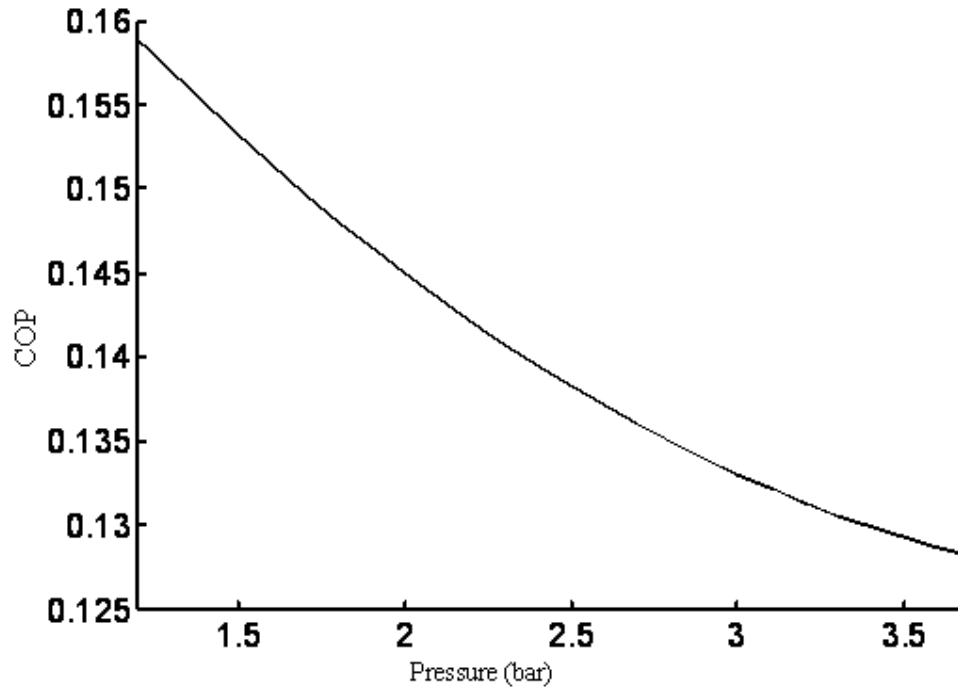


Figure 4-8: COP vs. System Pressure for Ethanol-Water-Propane

4.2.2 Effect of Condensing Temperature:

In Figures 4-9 and 4-10, the COP decreases as the condenser/absorber temperature increases and reach 0.176 and 0.15 at 42°C, and 45°C, respectively for the Ammonia-Water-Butane, and Ethanol-Water-Propane refrigeration cycles. The temperature for the condenser/absorber was chosen to be 42°C in the Ammonia-Water-Butane and 45°C for Ethanol-Water-Propane. The COP was found to be slightly better, in the Ammonia-Water-Butane refrigeration cycle, than the Ethanol-Water-Propane. This is due to difference in the maximum lift temperature. Figure 4-9 and 4-10, are plotted under the following condition for Ammonia-Water-Butane, system pressure of 4 bar, generator temperature of 72°C, Evaporator temperature of 0°C. For Ethanol-Water-Propane, system pressure of 1.8 bar, generator temperature of 100°C, Evaporator temperature of 0°C.

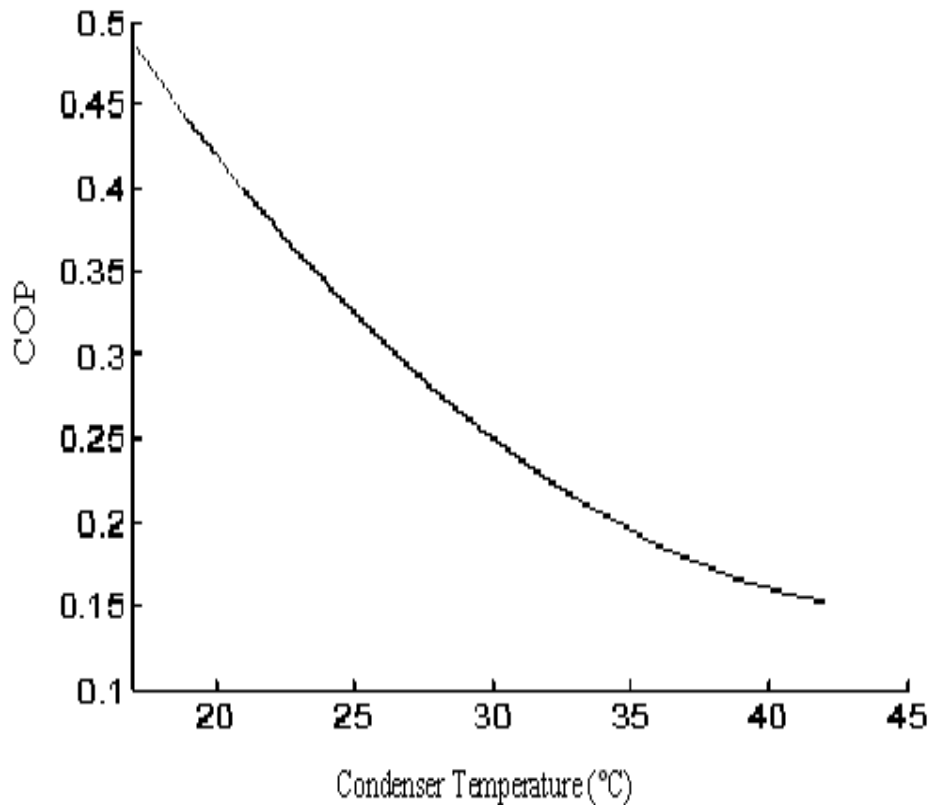


Figure 4-9: COP vs. Condenser Temperature for Ammonia-Water-Butane

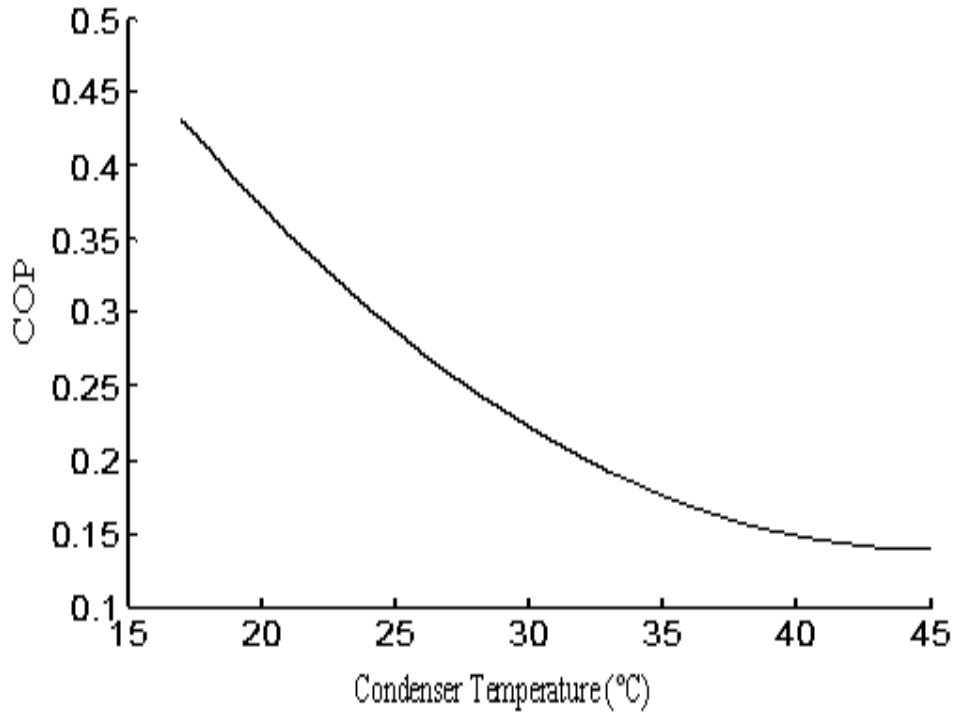


Figure 4-10: COP vs. Condenser Temperature for Ethanol-Water-Propane

4.2.3 Effect of the Evaporator Temperature:

In Figures 4-11 and 4-12; evaporator temperature varied in rang from -5°C to 15°C for both refrigeration cycles. For Ammonia-Water-Butane refrigeration cycle, the generation temperature of 72°C , system pressure of 4 bar and condenser/absorber of 42°C , and for Ethanol-Water-Propane refrigeration cycle , the generation temperature of 100°C , system pressure of 1,8bar and condenser/absorber of 45°C . When the evaporator temperature increasing the COP also increased. When evaporating temperature was 0°C , for both cycle, the COP are 0.176 and 0.15, respectively, for Ammonia-Water-Butane, and Ethanol-Water-Propane. In the Ammonia-Water-Butane, system pressure of 4bar, 42°C is the maximum condenser/absorber temperature; however, 0°C is well above the minimum evaporator temperature. In the Ethanol-Water-Propane, system pressure of 1.8bar, 45°C is the maximum condenser/absorber temperature; however, 0°C is well above the minimum evaporator temperature.

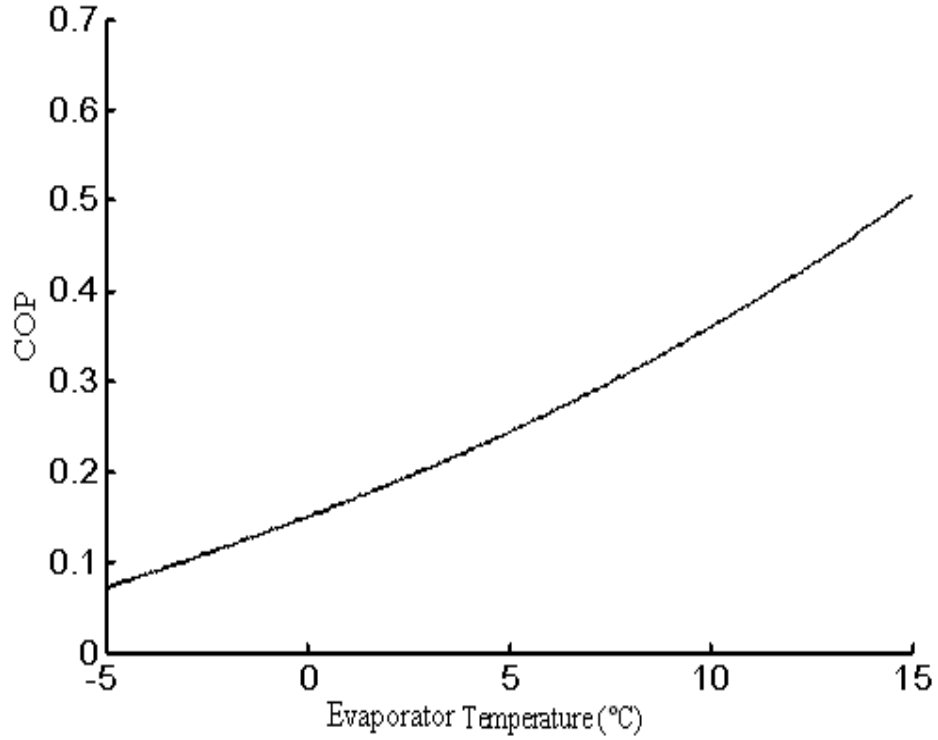


Figure 4-11: COP vs. Evaporator Temperature for Ammonia-Water-Butane

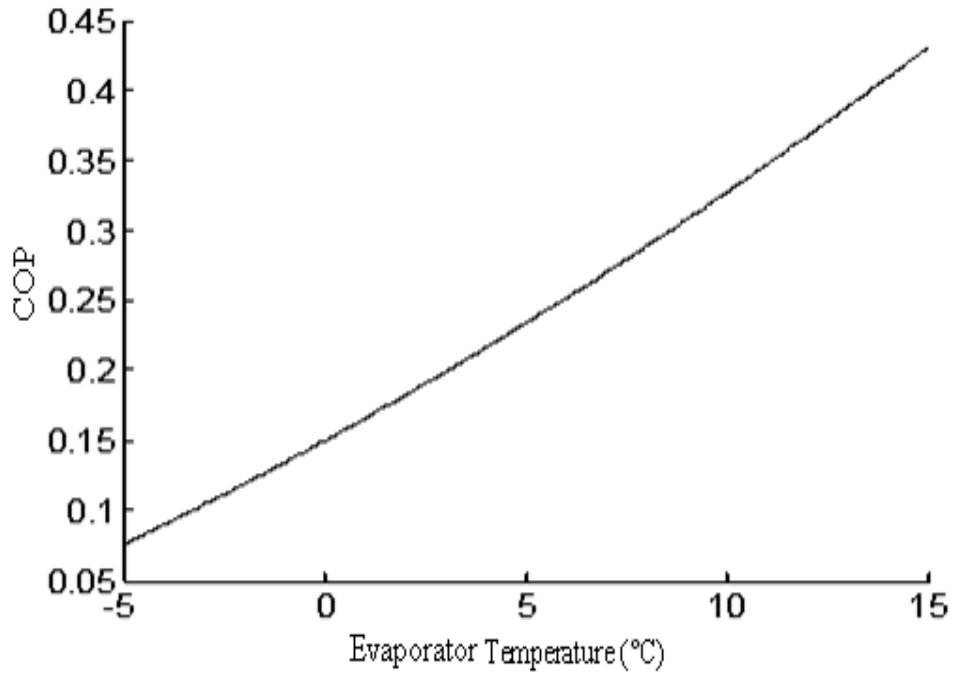


Figure 4-12: COP vs. Evaporator Temperature for Ethanol-Water-Propane

4.2.4 Effect of the Generator Temperature:

in the Figures 4-13 and 4-14, the generator temperature is varied while the other temperature remain fixed, at Ammonia-Water-Butane, system pressure of 4 bar, condenser/absorber temperature of 42°C, and evaporator temperature of 0°C, Figure 4-13 shows that the maximum COP 0.176 at generator temperature around 72°C. While at Ethanol-Water-Propane, system pressure of 1.8 bar, condenser/absorber temperature of 45°C, and evaporator temperature of 0°C. Figure 4-14 shows that the maximum COP 0.15 at generator temperature around 100°C. Heat required for bubble pump is found less in Ammonia-Water-Butane refrigeration cycle than the heat required for Ethanol-Water-Propane refrigeration cycle by about, because the liquid arriving must be heated through a high temperature difference.

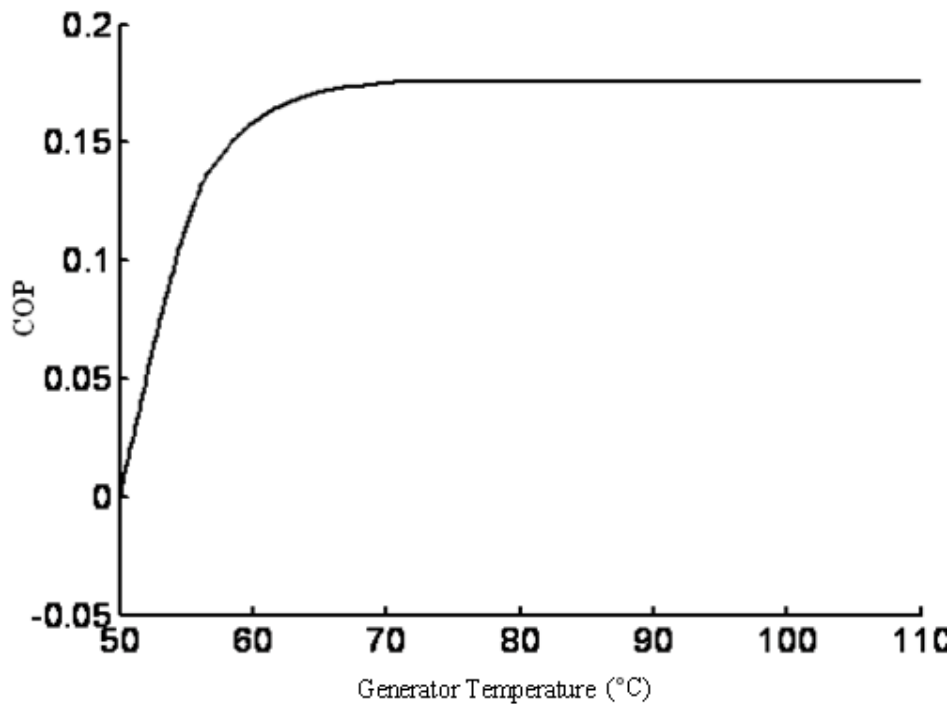


Figure 4-13: COP vs. Generator Temperature for Ammonia-Water-Butane

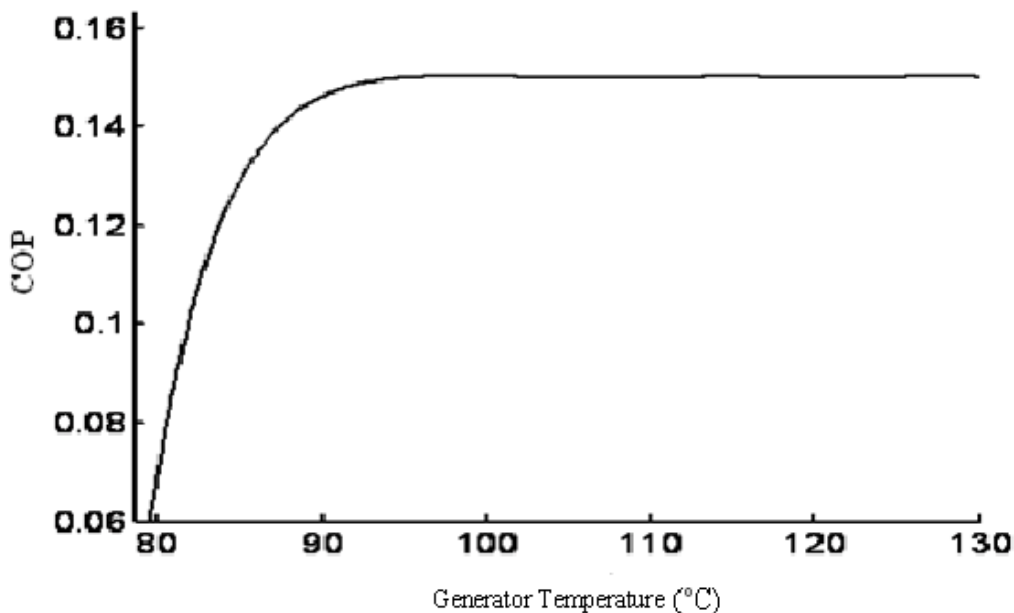


Figure 4-14: COP vs. Generator Temperature for Ethanol-Water-Propane

4.2.5 Effect of Pinch Point:

Figures 4-15, 4-16, 4-17, 4-18, 4-19, and 4-20, shows the effect of varying each of pinch points when other parameters are constant.

- In the evaporator pre-cooler, the largest thermal mass flow rate is the equalizing pressure-refrigerant vapor stream, which cools both the incoming liquid refrigerant and vapor equalizing pressure vapor. When pinch (2-3) is increased the temperature of the exiting equalizing pressure vapor is decreased, and the COP remains the same.
- Increasing pinch (6-4b) reduces the COP as expected since reducing the exiting temperature of this stream reduces the cooling available for the other two streams refer to Figuer (3-3).
- The performance of the external heat exchanger affects on the lift between condenser/absorber and evaporator temperatures. Increasing the pinch between streams 7 and 11 increasing the lift temperature, this increasing in the lift temperature reduces the COP.

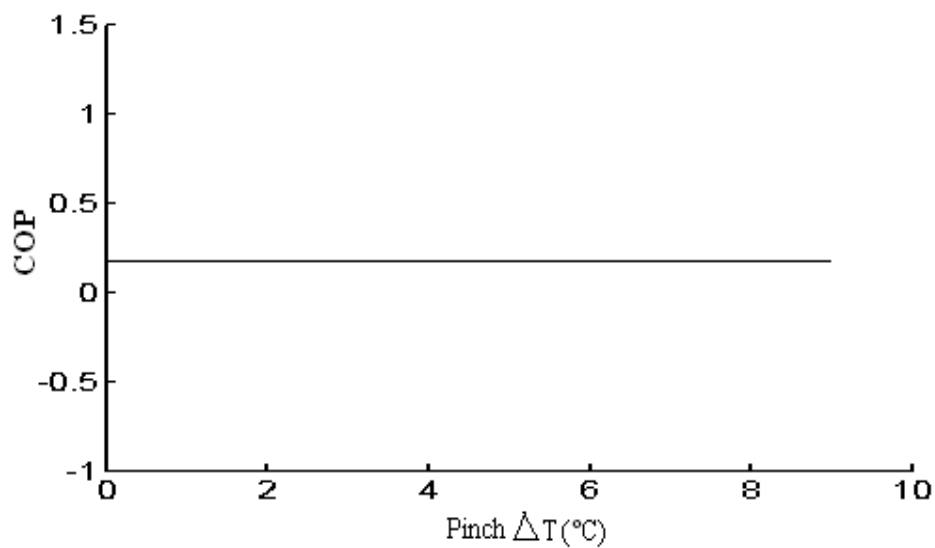


Figure 4-15: COP vs. Pinch ΔT value 2-3 for Ammonia-Water-Butane

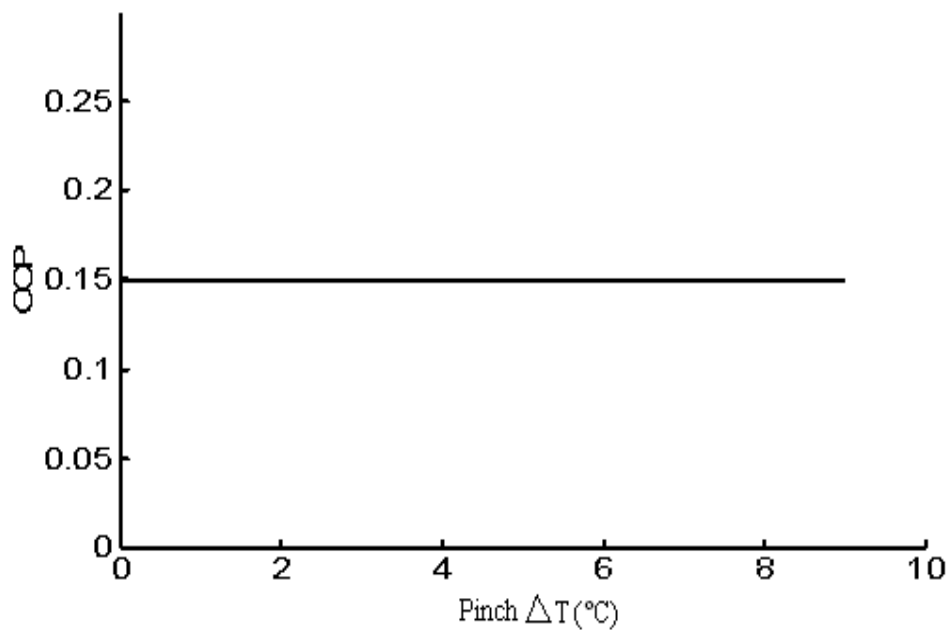


Figure 4-16: COP vs. Pinch ΔT value 2-3 for Ethanol-Water-Propane

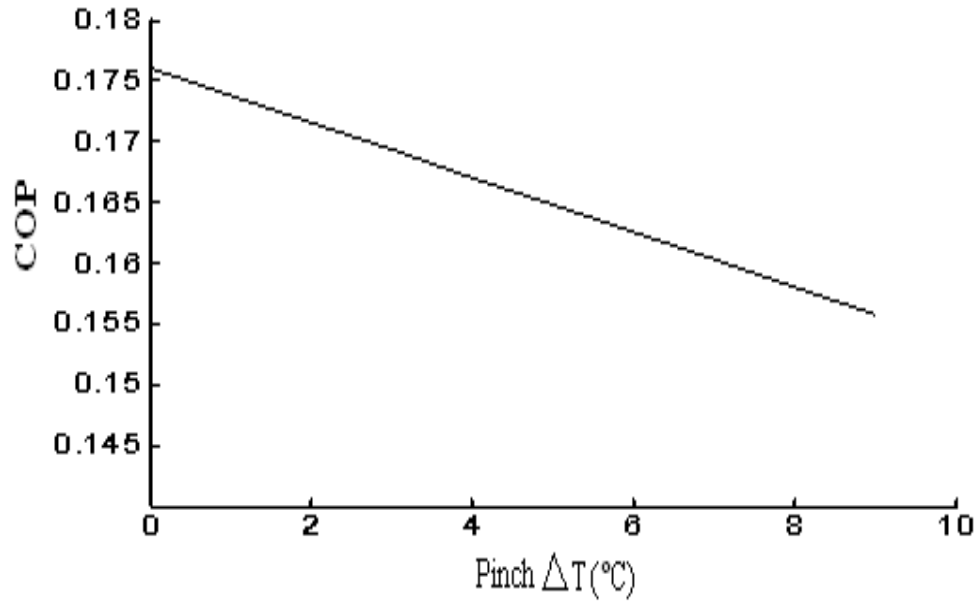


Figure 4-17: COP vs. Pinch ΔT value 6-4b for Ammonia-Water-Butane

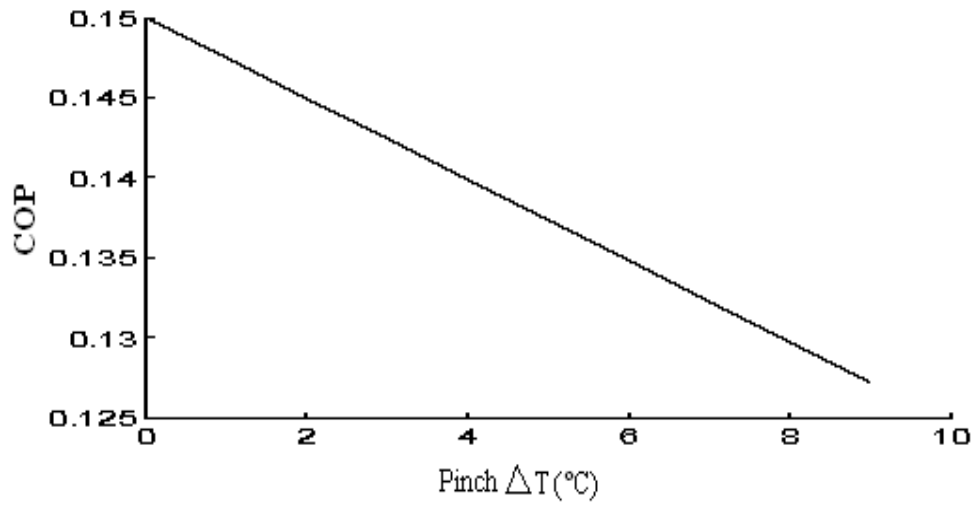


Figure 4-18: COP vs. Pinch ΔT value 6-4b for Ethanol-Water-Propane

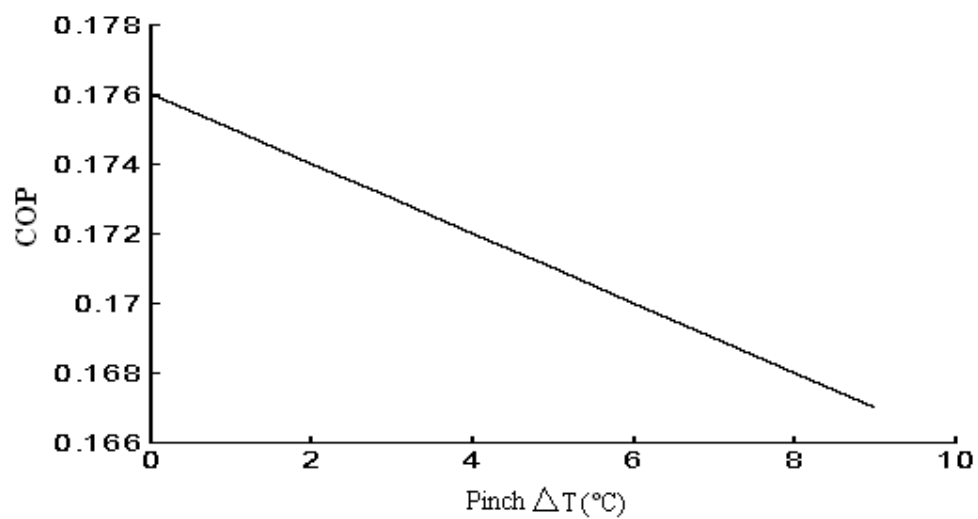


Figure 4-19: COP vs. Pinch ΔT value 7-11 for Ammonia-Water-Butane

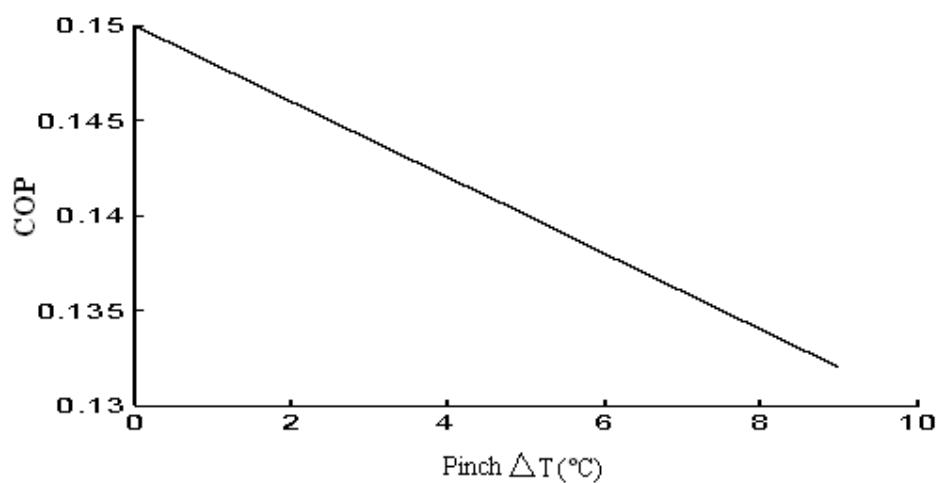


Figure 4-20: COP vs. Pinch ΔT value 7-11 for Ethanol-Water-Propane

CHAPTER FIVE: CONCLUSION AND RECOMMENDATION

5.1 Introduction:

This study investigated Einstein refrigeration cycle using triple working fluid ethanol-water-propane. A simulation program is modeled to study the cycle for refrigeration purpose, and compare results with ammonia-water-butane cycle.

5.2 Conclusions

This study analyzed the performance of a single pressure, direct thermally fired refrigeration cycle patented by Albert Einstein in 1927. The cycle uses ammonia-water-butane, and ethanol-water-propane as working fluids which were modeled using ideal gas and ideal solution assumptions and Raoult's law for the prediction of vapor-liquid equilibrium. Through the use of thermodynamic analysis, the study determined the operating parameters and their effect on the system performance. In this study, the second law of thermodynamics was used to obtain the ideal COP of the system. The major conclusions from the results of this work are as follows.

1. In this study the COP of the ammonia-water-butane, and ethanol-water-propane refrigeration cycles are obtained. Current ammonia-water-butane, and ethanol-water-propane cycles have calculated COP's around 0.176 and 0.15, respectively. The three temperatures (evaporator temperature, condenser/absorber temperature, and generator temperature) for ammonia-water-butane, and ethanol-water-propane refrigeration cycles are chosen for this analysis due to maximum lift temperature, and T-x-y diagram for the generator temperature. The model used many idealizing assumptions.
2. Another important conclusion of this study is the dependence of the temperature lift from the evaporator to the condenser on the system pressure. Vapor liquid equilibrium plotted for the ammonia-butane, ammonia-water, ethanol-propane, and ethanol-water mixtures. System's condensing and evaporating temperatures are constrained at the system pressure for any refrigeration cycle. This fixes the minimum evaporator temperature and the maximum condenser temperature for a particular pressure.

3. However temperature, it was shown that while increasing the generator heat transfer rate the bubble pump heat also increases, the system's performance was independent of the evaporator temperature, condenser/absorber temperature generator temperature. In this study, we can supply heat from solar energy or any industrial waste heat). This study showed the cycle to operate with generator temperatures of 72⁰C for ammonia-water-butane, and 100⁰C for ethanol-water-propane and in this condition to give highest COP in the operating domain.

4. Referring to Figers 4-12, 4-12, the optimum COP found in this study was: for the first case, ammonia-water-butane, COP of 0.5, when the evaporator temperature of 15⁰C, condenser/absorber of 42⁰C, and generator temperature of 72⁰C. For the second case, ethanol-water-propane, COP of 0.43, when the evaporator temperature of 15⁰C, condenser/absorber of 45⁰C, and generator temperature of 100⁰C.

5. The last conclusion comes from comparing the result of ethanol-water-propane with the ammonia-water-butane working fluid. The lift temperature was found higher in ethanol- propane mixture than ammonia-butane mixture at a system pressure used for any refrigeration cycle. Due to these differences, a lower COP obtained in ethanol-water-propane refrigeration cycle; however, the system pressure was lower in the ethanol-water-propane refrigeration cycle than ammonia-water-butane refrigeration cycle.

5.3 Recommendations

The recommendations for this thesis are:

- 1- There are many equations of state which account for non-ideal effects and could be applied to these mixtures.
- 2- An experimental prototype must be examined and compared with theoretical result.
- 3- The heat exchangers in this study were modeled using a zero pinch idealization. In reality the heat exchangers would have some non-zero pinch, and the model should accurately reflect this.
- 4- Ethanol-water-propane, working fluid, used for cooling purpose in this study as a refrigeration cycle, while another study can use this working fluid for heating purpose.
- 5- Other working fluids can be investigated. For example:
 - Water-Ammonia-Propne.
 - Water-Ammonia-Pentane.
 - Water-Methanol-Propylene.
 - Water- Methanol -Butane.
 - Water-Ethanol-Isobutane.
 - Water- Ethanol - Propylene.

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APPENDIX A

COMPUTER PROGRAM CODE

EES Program

This program is to calculate coefficient of performance for Einstein refrigeration cycle.

PROCEDURE makkitew(T,P:xa4,xw4,xma4,xmw4,ya4,yw4,yma4,ymw4,Pa4,Pw4)

mw1=46.07

mw2=18.015

P1s=pressure (Ethanol, T=T, x=0)

P2s=pressure (WATER, T=T, x=0)

xw4= (P-P1s)/ (P2s-P1s)

yw4= (P2s/P)*xw4

xa4=1-xw4

ya4=1-yw4

Pa4=ya4*P

Pw4=yw4*P

xmw4= (xw4*mw2)/ (xw4*mw2+xa4*mw1)

ymw4= (yw4*mw2)/ (yw4*mw2+ya4*mw1)

xma4= (xa4*mw1)/ (xw4*mw2+xa4*mw1)

yma4= (ya4*mw1)/ (yw4*mw2+ya4*mw1)

END

PROCEDURE makkitep(T,P:xa1,xb1,xma1,xmb1,ya1,yb1,yma1,ymb1,Pa1,Pb1)

mw1=46.07

mw2=44.1

P1ls=pressure (Ethanol, T=T, x=0)

P2ls=pressure (Propane, T=T, x=0)

xb1= (P-P1ls)/ (P2ls-P1ls)

yb1= (P2ls/P)*xb1

xa1=1-xb1

ya1=1-yb1

Pa1=ya1*P

Pb1=yb1*P

xmb1= (xa1*mw2)/ (xb1*mw2+xa1*mw1)

ymb1= (yb1*mw2)/ (yb1*mw2+ya1*mw1)

xma1= (xa1*mw1)/ (xb1*mw2+xa1*mw1)

yma1= (ya1*mw1)/ (yb1*mw2+ya1*mw1)

END

PROCEDURE makkitaw(T,P:xa4,xw4,xma4,xmw4,ya4,yw4,yma4,ymw4,Pa4,Pw4)

mw1=17.03

mw2=18.015

P1ls=pressure (AMMONIA, T=T, x=0)

P2ls=pressure (WATER, T=T, x=0)

xw4= (P-P1ls)/ (P2ls-P1ls)

yw4= (P2ls/P)*xw4

xa4=1-xw4

ya4=1-yw4

Pa4=ya4*P

Pw4=yw4*P

xmw4= (xw4*mw2)/ (xw4*mw2+xa4*mw1)

ymw4= (yw4*mw2)/ (yw4*mw2+ya4*mw1)

xma4= (xa4*mw1)/ (xw4*mw2+xa4*mw1)

yma4= (ya4*mw1)/ (yw4*mw2+ya4*mw1)

END

PROCEDURE makkitab(T,P:xa1,xb1,xma1,xmb1,ya1,yb1,yma1,ymb1,Pa1,Pb1)

mw1=17.03

mw2=58.12

P1ls=pressure (AMMONIA, T=T, x=0)

P2ls=pressure (R600, T=T, x=0)

xb1= (P-P1ls)/ (P2ls-P1ls)

yb1= (P2ls/P)*xb1

xa1=1-xb1

ya1=1-yb1

Pa1=ya1*P

Pb1=yb1*P

xmb1= (xa1*mw2)/ (xb1*mw2+xa1*mw1)

ymb1= (yb1*mw2)/ (yb1*mw2+ya1*mw1)

xma1= (xa1*mw1)/ (xb1*mw2+xa1*mw1)

yma1= (ya1*mw1)/ (yb1*mw2+ya1*mw1)

END


```

$INCLUDE C:\Documents and Settings\Makki\Desktop\aaaaaaaaaaa\ahmad\makkitab.lib
$INCLUDE C:\Documents and Settings\Makki\Desktop\aaaaaaaaaaa\ahmad\makkitaw.lib
{Einstein Refrigeration Cycle}
R=8.314 {kJ/kmolK}
mwb=58.124 {kg/kmol}
mwa=17.03 {kg/kmol}
mww=18.015 {kg/kmol}
Rb=R/mwb {kJ/kgK}
Ra=R/mwa {kJ/kgK}
Rw=R/mww {kJ/kgK}
{adjustable parameters}
adj=0.1
T1=315
T3=275
T9=350
Tp=T1+adj
P=pressure (R600, T=Tp, x=0)
{Limiting Temperatures}
T1a=temperature (AMMONIA, P=P, x=0)
T1b=temperature (R600, P=P, x=0)
{State 1: Ammonia-Butane sat. liquid leaving condenser}
CALL makkitab(T1a,P:xa1,xb1,xma1,xmb1,ya1,yb1,yma1,ymb1,Pa1,Pb1)
h1al=enthalpy (AMMONIA, T=T1, x=0)
h1bl=enthalpy (R600, T=T1, x=0)
u1al=intenergy (AMMONIA, T=T1, x=0)
u1bl=intenergy (R600, T=T1, x=0)
v1al=volume (AMMONIA, T=T1, x=0)
v1bl=volume (R600, T=T1, x=0)
s1al=entropy (AMMONIA, T=T1, x=0)
s1bl=entropy (R600, T=T1, x=0)
s1mixl=-xma1*Ra*ln(xa1)-xmb1*Rb*ln(xb1)
u1=xma1*u1al+xmb1*u1bl
v1=xma1*v1al+xmb1*v1bl
h1=u1+P*v1
s1=xma1*s1al+xmb1*s1bl+s1mixl
{State 2: butane subcooled liquid leaving pre-cooler}
T2=T3+pinch23
Pinch23=0
xma1=xma2
xmb1=xmb2

```

$s1_{mixl}=s2_{mixl}$
 $u2a= \text{intenergy (AMMONIA, T=T2, x=0)}$
 $u2b= \text{intenergy (R600, T=T2, x=0)}$
 $v2a= \text{volume (AMMONIA, T=T2, x=0)}$
 $v2b= \text{volume (R600, T=T2, x=0)}$
 $s2a= \text{entropy (AMMONIA, T=T2, x=0)}$
 $s2b= \text{entropy (R600, T=T2, x=0)}$
 $v2=x_{ma2} * v2a + x_{mb2} * v2b$
 $u2=x_{ma2} * u2a + x_{mb2} * u2b$
 $h2=u2 + P * v2$
 $s2=x_{ma2} * s2a + x_{mb2} * s2b + s2_{mixl}$
 {State 3: ammonia-butane sat. vapor}
 CALL makkitab(T3,P:xa3,xb3,xma3,xmb3,ya3,yb3,yma3,ymb3,Pa3,Pb3)
 $h3a= \text{enthalpy (AMMONIA, T=T3, P=Pa3)}$
 $h3b= \text{enthalpy (R600, T=T3, P=Pb3)}$
 $u3a= \text{intenergy (AMMONIA, T=T3, P=Pa3)}$
 $u3b= \text{intenergy (R600, T=T3, P=Pb3)}$
 $s3a= \text{entropy (AMMONIA, T=T3, P=Pa3)}$
 $s3b= \text{entropy (R600, T=T3, P=Pb3)}$
 $s3_{mixv} = -y_{ma3} * R_a * \ln(ya3) - y_{mb3} * R_b * \ln(yb3)$
 $s3_{mixl} = -x_{ma3} * R_a * \ln(xa3) - x_{mb3} * R_b * \ln(xb3)$
 $h3 = y_{ma3} * h3a + y_{mb3} * h3b$
 $u3 = y_{ma3} * u3a + y_{mb3} * u3b$
 $s3 = y_{ma3} * s3a + y_{mb3} * s3b$
 {State 5: Superheated Ammonia entering pre-cooler}
 $T5=T1$
 $h5= \text{enthalpy (AMMONIA, T=T5, P=P)}$
 $s5= \text{entropy (AMMONIA, T=T5, P=P)}$
 {State 6: Superheated Ammonia-butane leaving pre-cooler}
 $T6=T5\text{-pinch64b}$
 $\text{Pinch64b}=0$
 CALL makkitab(T6,P:xa6,xb6,xma6,xmb6,ya6,yb6,yma6,ymb6,Pa6,Pb6)
 $h6a= \text{enthalpy (AMMONIA, T=T6, P=Pa6)}$
 $h6b= \text{enthalpy (R600, T=T6, P=Pb6)}$
 $u6a= \text{intenergy (AMMONIA, T=T6, P=Pa6)}$
 $u6b= \text{intenergy (R600, T=T6, P=Pb6)}$
 $s6a= \text{entropy (AMMONIA, T=T6, P=Pa6)}$
 $s6b= \text{entropy (R600, T=T6, P=Pb6)}$
 $s6_{mixv} = -y_{ma6} * R_a * \ln(ya6) - y_{mb6} * R_b * \ln(yb6)$
 $h6 = y_{ma6} * h6a + y_{mb6} * h6b$

$u6 = y_{ma6} * u_{6av} + y_{mb6} * u_{6bv}$
 $s6 = y_{ma6} * s_{6av} + y_{mb6} * s_{6bv}$
 {State 7: Ammonia-Water liquid leaving condenser}
 $T7 = T1$
 CALL makkitab(T7,P:xa7,xw7,xma7,xmw7,ya7,yw7,yma7,ymw7,Pa7,Pb7)
 $u7al = \text{intenergy (AMMONIA, T=T7, x=0)}$
 $u7bl = \text{intenergy (water, T=T7, x=0)}$
 $v7al = \text{volume (AMMONIA, T=T7, x=0)}$
 $v7bl = \text{volume (water, T=T7, x=0)}$
 $s7al = \text{entropy (AMMONIA, T=T7, x=0)}$
 $s7bl = \text{entropy (water, T=T7, x=0)}$
 $s7mixl = -x_{ma7} * R_a * \ln(x_{a7}) - x_{mb7} * R_w * \ln(x_{w7})$
 $u7 = x_{ma7} * u_{7al} + x_{mw7} * u_{7wl}$
 $v7 = x_{ma7} * v_{7al} + x_{mw7} * v_{7wl}$
 $h7l = u4 + P * v4$
 $s7l = x_{ma7} * s_{7al} + x_{mw7} * s_{7wl} + s_{7mixl}$
 {The Evaporator}
 {mass balances}
 $m3 = m2 + m4$ {overall}
 $m3 * y_{mb3} = m2 * x_{mb2}$ {butane}
 $m3 * y_{ma3} = m2 * x_{ma2} + m4$ {ammonia}
 {Lower}
 {Heat transfer & entropy generation}
 $Q_e = m3 * h3 - m2 * h2 - m4 * h4v$
 $S_e = m3 * s3 - m2 * s2 - m4 * s4 - (Q_e / T3)$
 {The pre-cooler/rectifier}
 $m1 * h1 + m3 * h3 + m4b * h4b = m2 * h2 + m4 * h4 + m6 * h6$
 {Cond/abs overall}
 $m1 + m7 = m11 + m6$ {overall}
 $m1 * x_{mb1} = m6 * y_{mb6}$ {butane}
 $m1 * x_{ma1} + m7 * x_{ma7} = m6 * y_{ma6} + m11 * x_{ma11}$ {ammonia}
 $m7 * x_{mw7} = m11 * x_{mw11}$ {water}
 {Heat transfer & entropy generation}
 $Q_c = m1 * h1 - m7 * h7 - m6 * h6 - m11 * h11$
 $S_c = m1 * s1 + m7 * s7 - m11 * s11 - (Q_c / T_c)$
 {State 9: Ammonia-water Weak liquid solution}
 CALL makkitab(T9,P:xa9,xw9,xma9,xmw9,ya9,yw9,yma9,ymw9,Pa9,Pw9)
 $U9al = \text{intenergy (AMMONIA, T=T11, x=0)}$
 $u9wl = \text{intenergy (water, T=T9, x=0)}$
 $v9al = \text{volume (AMMONIA, T=T9, x=0)}$

$v9wl = \text{volume (water, } T=T11, x=0)$
 $s9al = \text{entropy (AMMONIA, } T=T9, x=0)$
 $s9wl = \text{entropy (water, } T=T9, x=0)$
 $s9mixl = -xma9 * Ra * \ln(xa9) - xmw9 * Rw * \ln(xw9)$
 $u9 = xma9 * u9al + xmw9 * u9wl$
 $v9 = xma9 * v9al + xmw9 * v9wl$
 $h9 = u9 + P * v9$
 $s9 = xma9 * s9al + xmw9 * s9wl + s9mixl$
 {State 11: Ammonia-water sub-cooled liquid leaving generator external heat exchanger}
 $T11 = T7 + \text{pinch711}$
 $\text{Pinch711} = 0$
 $\text{CALL makkitaw}(T11, P: xa11, xw11, xma11, xmw11, ya11, yw11, yma11, ymw11, Pa11, Pw11)$
 $u11al = \text{intenergy (AMMONIA, } T=T11, x=0)$
 $u11wl = \text{intenergy (water, } T=T11, x=0)$
 $v11al = \text{volume (AMMONIA, } T=T11, x=0)$
 $v11wl = \text{volume (water, } T=T11, x=0)$
 $s11al = \text{entropy (AMMONIA, } T=T11, x=0)$
 $s11wl = \text{entropy (water, } T=T11, x=0)$
 $s11mixl = -xma11 * Ra * \ln(xa11) - xmw11 * Rw * \ln(xw11)$
 $u11 = xma11 * u11al + xmw11 * u11wl$
 $v11 = xma11 * v11al + xmw11 * v11wl$
 $h11 = u11 + P * v11$
 $s11 = xma11 * s11al + xmw11 * s11wl + s11mixl$
 {Generator}
 {mass conservation}
 $m7 = m8$
 $m10 = m11$
 $m7 = m5 + m11 + m9. \quad \{\text{overall}\}$
 $xma7 * m7 = m5 * yma5 + m9g * yma9g + xma11 * m11 \quad \{\text{ammonia}\}$
 $xmw11 * m11 = xmw7 * m7 \quad \{\text{water}\}$
 $m8 = m5 + m9$
 $xma8 * m8 = m5 * yma5 + xma10 * m10 \quad \{\text{ammonia}\}$
 $m7 * h7 + m11 * h11 = m8 * h8 + m10 * h10$
 {Energy & Entropy}
 $Tg = T9$
 $Qg + Qbp = m5 * h5 + m10 * h10 - m9 * h9 - m8 * h8$
 $Sg = m5 * h5 + m9 * h9 + m11 * h11 - m7 * h7 - (Qg + Qbp) / Tg$
 $mbp = 1$
 $m9v = mbp / 28.6$
 $m9 = mbp - m9v$

$$m11=m9$$

$$COP=Qe/Qg$$

$$Tc=T1$$

$$Tg=T9$$

$$Te=T3$$

$$Qg=m10*h*10+m5*h5-m9v*h9v-m8*h8$$

$$Qch=Qc+Qe+Qg$$

{2nd Law }

$$COP\ ideal= ((Tg-Tc)/ (Tg))/ ((Tc-Te)/Te)$$

$$Sgen=-(Qg/Tg)-(Qe/Te)-(Qc/Tc)$$

$$COP\ act=COP\ ideal-((Te*Tc)/ (Tc-Te))*Sgen/Qg$$

APPENDIX B

SECOND LAW ANALYSIS

The coefficient of performance (COP_c) is directly related to the cycle's entropy generation. This can be demonstrated by combining a first and second law analysis.

Cooling coefficient of performance is defined as:

$$COP_c = \frac{Q_L}{Q_H} \quad (B-1)$$

The first and second laws of thermodynamics are:

$$Q_{ca} = Q_{evap} + Q_{gen} \quad (B-2)$$

$$\Sigma \dot{S}_i = \frac{\dot{Q}_{ca}}{T_{ca}} - \frac{\dot{Q}_{evap}}{T_{evap}} - \frac{\dot{Q}_{gen}}{T_{gen}} \quad (B-3)$$

$$\Sigma \dot{S}_i = \frac{\dot{Q}_{evap} + \dot{Q}_{gen}}{T_{ca}} - \frac{\dot{Q}_{evap}}{T_{evap}} - \frac{\dot{Q}_{gen}}{T_{gen}} \quad (B-4)$$

$$T_{ca} \Sigma \dot{S}_i = \dot{Q}_{evap} + \dot{Q}_{gen} - \dot{Q}_{evap} \frac{T_{ca}}{T_{evap}} - \dot{Q}_{gen} \frac{T_{ca}}{T_{gen}} \quad (B-5)$$

$$T_{ca} \Sigma \dot{S}_i = \dot{Q}_{evap} \left(1 - \frac{T_{ca}}{T_{evap}}\right) + \dot{Q}_{gen} \left(1 - \frac{T_{ca}}{T_{gen}}\right) \quad (B-6)$$

$$T_{ca} \Sigma \dot{S}_i = \dot{Q}_{evap} \left(\frac{T_{evap} - T_{ca}}{T_{evap}}\right) + \dot{Q}_{gen} \left(\frac{T_{gen} - T_{ca}}{T_{gen}}\right) \quad (B-7)$$

Dividing equation (B-7) by equation (B-2)

$$T_{ca} \frac{\Sigma}{\dot{Q}_{gen}} = COP_c \left(\frac{T_{evap} - T_{ca}}{T_{evap}}\right) + \left(\frac{T_{gen} - T_{ca}}{T_{gen}}\right) \quad (B-8)$$

The reversible, or ideal, heating coefficient of performance for a system operating at three temperatures is:

$$COP_{c,rev} = \left(\frac{T_H - T_M}{T_H}\right) \left(\frac{T_L}{T_M - T_L}\right) \quad (B-9)$$

$$COP_{c,rev} = \left(\frac{T_{gen} - T_{ca}}{T_{gen}}\right) \left(\frac{T_{evap}}{T_{ca} - T_{evap}}\right) \quad (B-10)$$

$$COP_{c,rev} = \left(\frac{T_{evap}}{T_{gen}}\right) \left(\frac{T_{gen} - T_{ca}}{T_{ca} - T_{evap}}\right) \quad (B-11)$$

$$T_{ca} \frac{\dot{\Sigma}S_i}{\dot{Q}_{gen}} = COP_c \left(\frac{T_{evap} - T_{ca}}{T_{evap}} \right) + COP_c^{rev} \left(\frac{T_{ca} - T_{evap}}{T_{evap}} \right) \quad (B-12)$$

$$T_{ca} \frac{\dot{\Sigma}S_i}{\dot{Q}_{gen}} = -COP_c \left(\frac{T_{ca} - T_{evap}}{T_{evap}} \right) + COP_c^{rev} \left(\frac{T_{ca} - T_{evap}}{T_{evap}} \right) \quad (B-13)$$

$$T_{ca} \frac{\dot{\Sigma}S_i}{\dot{Q}_{gen}} = (-COP_c + COP_c^{rev}) \left(\frac{T_{ca} - T_{evap}}{T_{evap}} \right) \quad (B-14)$$

$$\frac{T_{ca} \times T_{evap}}{T_{ca} - T_{evap}} \frac{\dot{\Sigma}S_i}{\dot{Q}_{gen}} = (-COP_c + COP_c^{rev}) \left(\frac{T_{ca} - T_{evap}}{T_{evap}} \right) \quad (B-15)$$

$$\frac{T_{ca} \times T_{evap}}{T_{ca} - T_{evap}} \frac{\dot{\Sigma}S_i}{\dot{Q}_{gen}} = (COP_c^{rev} - COP_c) \quad (B-16)$$

$$COP_c = COP_c^{rev} - \frac{T_{ca} \times T_{evap}}{T_{ca} - T_{evap}} \frac{\dot{\Sigma}S_i}{\dot{Q}_{gen}} \quad (B-17)$$

نمذجة و دراسة أداء لدورة انشأتين التبريدية

إعداد

أحمد مكي محمد أحمد الشيخ

المشرف

الأستاذ الدكتور محمود حماد

ملخص

دورة آينشتاين للتبريد تستخدم البيوتان كسائل لتبريد ، والأمونيا كمعادل للضغط ، والماء كمادة ماصة . الضغط الجزئي لغاز البيوتان يخفض بواسطة بخار الأمونيا ويزداد عندما يتم امتصاص بخار الأمونيا من الماء السائل ، مما يسمح لغاز البيوتان للتبخر عند درجة حرارة منخفضة وتتكثف في درجة حرارة عليا .
 تحسب هذه الدراسة أداء السوائل أعلاه مع سوائل بديلة ، الإيثانول- الماء- البروبان. جميع خصائص الخليط شكلت مع الغاز المثالي والمحلول المثالي ومعادلات التوازن للبخار والسائل. هذه الدراسة تبين أن الضغط يجب أن يكون نظام التشبع للسائل التبريد عند درجة حرارة المكثف / الامتصاص.
 في هذه الدراسة الحالة الأولى كانت تستخدم الأمونيا والماء وغاز البيوتان معامل أداء حول 0.176 ، بينما القيمة المثلى 0.5. الحالة الثانية استخدام الايثانول والماء والبروبان معامل أداء حول 0.15 ، بينما القيمة المثلى 0.43.