Experimental Investigation on an Intermittent Ammonia Absorption Refrigeration System

by

Dipayan Mondal

A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in Engineering in the department of Mechanical Engineering

Khulna University of Engineering & Technology
Khulna 9203, Bangladesh
September 2014

Declaration

This is to certify that the project work entitled "*Experimental Investigation on an Intermittent Ammonia Absorption Refrigeration System*" has been carried out by *Dipayan Mondal* in the Department of Mechanical Engineering, Khulna University of Engineering & Technology, Khulna, Bangladesh. The above project work has not been submitted anywhere for the award of any degree or diploma.

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Approval

This is to certify that the project work submitted by *Dipayan Mondal* entitled " *Experimental Investigation on an Intermittent Ammonia Absorption Refrigeration System*" has been approved by the board of examiners for the partial fulfillment of the requirements for the degree of *Master of Science in Engineering* in the Department of Mechanical Engineering, Khulna University of Engineering & Technology, Khulna, Bangladesh in September 2014.

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ABSTRACT

This project work describes the experimental investigation on an intermittent ammonia absorption refrigeration system which consists of a generator/ absorber, a condenser, an evaporator and an expansion valve. The system used 2.5 kg of ammonia as refrigerant and 1.5 kg of calcium chloride as absorbent initially in generator and this NH_3-$CaCl_2$ refrigerant-absorbent pair is widely used to intermittent refrigeration system. There are two modes where in desorption mode, an oil bath arrangement heat source provides heat to the generator and the generator drives of the vapor form refrigerant around the system through a condenser to evaporator. The evaporator pipes were placed outside to the evaporator container which was insulated and 3 kg of water was filled to be the load for cooling into the evaporator container. During the absorption mode, the liquid refrigerant absorbed heat from the surrounding water and performed cooling. The average refrigeration temperature was found to be 13.25^0C with a minimum value of 11.5^0C for a period of time up to 2.5 hours absorption. The coefficient of performance was obtained on the average of 0.154 with a maximum value of 0.192. This coefficient of performance was higher than the coefficients of performance found by Vanek [1], Katejanekarn [2] and Moreno-Quintanar [3] on their intermittent solar absorption system and also lower than the COP found by Tangka [4] developed intermittent solar absorption system.

The generator temperature was found to be 91.40^0C corresponding to the oil bath temperature up to 110^0C during 2.5 hours desorption and the pressure was to be found 10.5 bar. On an average the system could provide the heat gained by the generator of 1492.96 kJ and the refrigeration effect of 223.65 kJ.

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Nomenclature

P	Saturation pressure, bar
H	Enthalpy, kJ/ kg
m	Mass of water (load), kg
m_a	Mass of ammonia, kg
Q_e	Cooling capacity or refrigeration effect, kJ
Q_g	Heat input to the generator, kJ
Q	Heat transfer, kJ
Q_t	Total heat generated by the oil bath, kJ
Q_{loss}	Heat loss from the oil bath, kJ
$T\alpha$	Ambient or room Temperature, ^{0}C
T_w	Wall Temperature, ^{0}C
ΔT	Temperature difference, ^{0}C
ΔT_w	Temperature reduction of water, ^{0}C
ΔT_{ln}	Logarithmic mean temperature difference,
ΔT_S	Temperature reduction of water due to system heat gained, ^{0}C
ΔT_{ln}	Logarithmic mean temperature difference (^{0}C)
L_s	Latent heat of ammonia corresponding evaporator pressure, kJ/kg-C
L_w	Latent heat of fusion of water, kJ/kg-C
D	Diameter of tube material, m
A	Total heat transfer surface area, m^2
L	Length of heat exchanger, m
k	Thermal conductivity of the material, W/m .^{0}C
U	Overall heat transfer coefficient, W/m^2.^{0}C
h	Convective heat transfer coefficient, W/m^2.^{0}C
c_p	Specific heat of fluid, kJ/kg-C
$C_{p,w}$	Specific heat of water, kJ/kg-C
x	Thickness, m

ρ	Density of fluid, kg/m^3
COP	Coefficients of performance
Re	Reynolds number
Pr	Prandtl number
N_u	Nusselt number

Subscripts

h	hot fluid
c	cold fluid
I	inside
o	outside
s	solution
n	section no.=1,2…..n
ref	refrigeration
sys	system

CHAPTER I

Introduction

1.1 Overview

Vapor absorption refrigeration system gained renewed interest due to environmental problems of commonly used refrigerants in vapor compression refrigeration system. Being a heat operated system it is gained major focus in solar energy based refrigeration system. Various types of vapor absorption system are used in solar energy based refrigeration system due to energy crises is a serious handicap for the socio-economic development of the rural and urban population.

In absorption refrigeration system the vapor is drawn from the evaporator by absorption into liquid having high affinity for the refrigerant. The refrigerant is expelled from the solution by application of heat and its temperature is also increased. This refrigerant in the form of vapor passes to the condenser where heat is rejected and the refrigerant get liquefied. This liquid again flows to the evaporator at reduced pressure. The intermittent refrigeration system actually is not a true refrigeration system because the mass flow rate is not constant throughout the system. The mass flow rate increases when the temperature of the generator increases and the rate of flow of vapor ammonia also increase. The simplest design of this intermittent system consists of three major parts such as a generator/ an absorber for desorption/absorption the salt-ammonia mixture, a condenser, and an evaporator. Ammonia flows back and forth between the generator and evaporator [1]. Cold storage of food products and medicines, especially immunization vaccines are important application of solar refrigeration. Many agricultural products like fruits, vegetables, meat, fish etc. can be maintained in fresh conditions for significantly longer period of time if they are stored at low temperature [1].

The intermittent absorption cooling system works in half a cycle at a time as schematically shown in Fig.1.1. During the step-1 in desorption mode, the container A acts as the generator producing high pressure and high temperature refrigerant vapor by heat from a source. The refrigerant vapor separating from the absorbent sent through the condenser, turned into liquid phase and stored inside the container B.

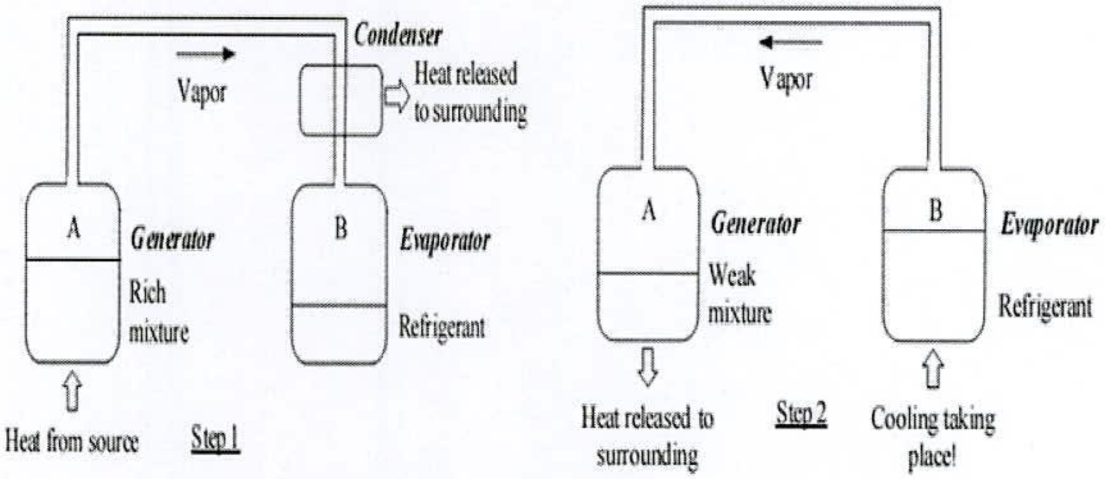


Figure1.1: Intermittent absorption cooling system

Again step-2 in absorption mode, the container B acts as the evaporator absorbing heat from the surroundings. If it is surrounded by water, the cold water or ice can be produced. The refrigerant will be evaporated into the vapor and flow back into the container A acting as the absorber where the concentrated absorbent awaits and both working substances are mixed [2].

Katejanekarn [2] developed an intermittent solar absorption cooling system having the COP of the system is about 0.029. Moreno-Quintanar [3] developed a solar intermittent refrigeration system for ice production and found the coefficients of performance is as high as 0.098. Tangka [4] used the NH_3-H_2O refrigerant pair in the intermittent vapor absorption system and measured the average lowest refrigeration temperature and the coefficient of performance (COP) of the system was 4°C and 0.487 respectively. Srinivasa Rao [5] investigated a two-stage vapor absorption refrigeration system employing the refrigerant absorbent combinations of NH_3-H_2O and NH_3-$LiNO_3$ and measured the COP of the system is about 0.69. Aphornratana and Eames [6] investigated single effect water lithium bromide system using energy analysis approach. It was shown that the irreversibility in generator was highest followed by absorber and evaporator. Bell et al [7] developed a LiBr-H_2O experimental absorption cooling system driven by heat generated by solar energy and concluded that the COP of the system depends on generator temperature and there is optimum value of generator temperature at which COP is maximum.

Performance of intermittent vapor absorption system depends on generator's temperature and heat addition. However, the generator's temperature and heat addition depends on incident solar radiation intensity which varies with time of the day and weather conditions. So investigations are required to evaluate the performance of the intermittent vapor absorption with constant temperature heat addition in generator. Also amount of heat addition in generator is a key parameter to design solar collector for intermittent vapor absorption system.

So, experimental investigations are required to evaluate the performance of intermittent vapor absorption system by varying the generator's heat addition and temperature.

1.2 Objectives

The objectives of this project work are:

1. To design and construct of an Intermittent Ammonia Absorption Refrigeration System
2. To investigate the refrigeration effect in evaporator
3. To investigate the effect of generator temperature on heat addition
4. To investigate the coefficient of performance (COP) of the system

1.3 Outline of the project

This project is organized in several chapters as follows:

Chapter II Describes the literature review and background of the research work.

Chapter III Presents the design and construction of the intermittent vapor absorption system.

Chapter IV Explain the experimental result obtained from this investigation

Chapter V Shows the conclusions of this project work.

CHAPTER II

Literature Review

In this chapter the most usable vapor absorption refrigeration system and relevant previous work on the various intermittent vapor absorption refrigeration systems are reviewed. The review is divided into three sections: 1) Basic of vapor absorption system; 2) Various types of refrigeration system; 3) Experimental analysis of different vapor absorption system.

2.1 Basic of Vapor Absorption System

2.1.1 History of Refrigeration System

In Europe, America and Iran a number of icehouses were built to store ice. Materials like sawdust or wood shavings were used as insulating materials in these icehouses. Later on, cork was used as insulating material. Literature reveals that ice has always been available to aristocracy who could afford it. In India, the Mogul emperors were very fond of ice during the harsh summer in Delhi and Agra, and it appears that the ice used to be made by nocturnal cooling. In 1805, an American inventor, Oliver Evans, designed the first refrigeration machine that used vapor instead of liquid. Evans never constructed his machine, but one similar to it was built by an American physician, John Gorrie.

In 1806, Frederic Tudor (who was later called as the "ice king") began the trade in ice by cutting it from the Hudson River and ponds of Massachusetts and exporting it to various countries including India. In India Tudor's ice was cheaper than the locally manufactured ice by nocturnal cooling. The ice trade in North America was a flourishing business. Ice was transported to southern states of America in train compartments insulated by 0.3m of cork insulation. Trading in ice was also popular in several other countries like Great Britain, Russia, Canada, Norway and France. In these countries ice was either transported from colder regions or was harvested in winter and stored in icehouses for use in summer. The ice trade reached its peak in 1872 when America alone exported 225000 tons of ice to various countries as far as China and Australia. However, with the advent of artificial refrigeration the ice trade gradually declined.

In 1842, the American physician John Gorrie, to cool sickrooms in a Florida hospital, designed and built an air-cooling apparatus for treating yellow-fever patients. His basic principle--that of compressing a gas, cooling it by sending it through radiating coils, and then expanding it to lower the temperature further--is the one most often used in refrigerators today. Giving up his medical practice to engage in time-consuming experimentation with ice making, he was granted the first U.S. patent for mechanical refrigeration in 1851.

Commercial production began in 1923 by the newly formed company AB Arctic, which was bought by Electrolux in 1925. In the 60s the absorption refrigeration saw a renaissance due to the substantial demand for refrigerators for caravans. AB Electrolux established a subsidiary in the U.S, named Domestic Sales Corporation. The company marketed refrigerators for caravans under the Domestic brand. In 1926 Albert Einstein and his former student LeóSzilárd proposed an alternative design known as refrigerator.

The CrosleyIcyball was first patented in 1927 by David Forbes Keith and then manufactured by Powel Crosley Jr., who purchased the rights. It has since been out of production although thousands of units were produced the 1930's. The Icyball is intermittent heat absorption refrigerator using a water/ammonia mixture as the absorbent and refrigerant pair. At room temperature water and ammonia combine into a single solution. TheIcyball consists of a hot and cold side, with the hot side being a ball of steel that holds the water/ammonia mixture. Heat is than added to the hot side boiling out the ammonia from the water and then condensing inside the cold ball that is in a water bath. After the hot side is heated for around 90 minutes most of the ammonia is condensed into a liquid and the cold ball is placed inside an insulated chest. After the hot ball cools down, the ammonia in the cold ball will start to evaporate and recombine with the water that is still in the hot ball causing the pressure to drop in the cold ball which allows for a refrigeration effect. These units were capable of cooling a 4cu ft. insulated chest for approximately 24 hours and operated at around 250 Psi.

In 1973, Prof. James Lovelock reported finding trace amounts of refrigerant gases in the atmosphere. In 1974, Sherwood Rowland and Mario Molina predicted that chlorofluorocarbon refrigerant gases would reach the high stratosphere and there damage the protective mantle of the oxygen allotrope, ozone. In 1985 the "ozone hole" over the Antarctic had been discovered and by 1990 Rowland and Molina's prediction was proved correct.

In 1996 the S.T.E.V.E.N Foundations (Solar Technology and Energy for Vital Economic Needs) Solar Ammonia Absorption Icemaker, developed their design also using the intermittent absorption cycle but uses calcium chloride salt as the absorber and pure ammonia as the refrigerant. Utilizing calcium chloride as the absorber instead of water allows for some practical advantages, primarily no water is evaporated when heated which produces a non-diluted ammonia solution and allowing for a stronger absorption process [8].

In 2001 Electrolux sold most of its Leisure Products line to the venture-capital company EQT which created Domestic as stand-alone company. At the 2007 TED Conference, Adam Grosser presented his research of a new, very small, "intermittent absorption" vaccine refrigeration unit for use in third world countries. The refrigerator is a small unit placed over a campfire that can later be used to cool 3 gallons of water to just above freezing for 24 hours in a 30 degree Celsius environment

2.1.2 Principle of Operation

As the name implies, absorption refrigeration systems involve the absorption of a refrigerant by transport medium or absorbent. The most widely used absorption refrigeration system is the ammonia-water system, where ammonia serves the refrigerant and water as the transport medium or absorber. Other useful system is water-lithium bromide system, where serves the refrigerant and lithium bromide as the absorber. In vapor absorption system is alike of vapor compression refrigeration system except that the compressor has been replaced by a complex absorption mechanism consisting of an absorber, a pump, a generator, an expansion valve, and a rectifier.

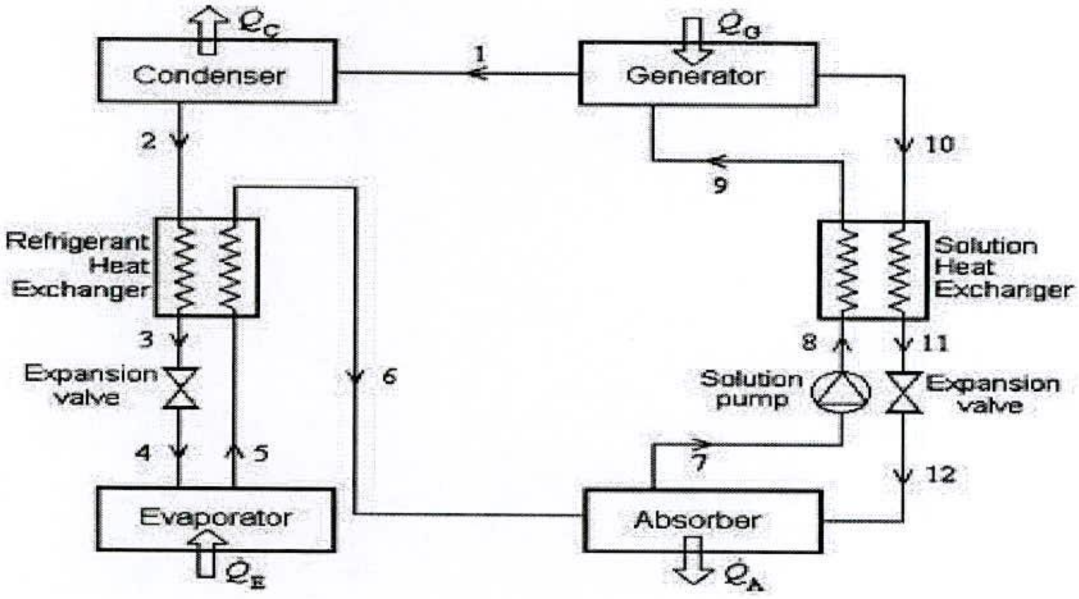


Figure 2.1: Basic principle of vapor absorption system

The vapor refrigerant when come from generator is cooled and condensed in the condenser by rejecting heat to the surroundings, is throttled to the evaporator pressure, and absorbs heat from the refrigerated space as it flows through the evaporator. The refrigerant vapor leaves the evaporator and enters the absorber, where it dissolves and reacts with absorber to form a aqueous solution. This is an exothermic reaction; thus heat is released during this process. The amount of refrigerant absorbed into the absorber is inversely proportional to the temperature. After absorption the rich in refrigerant mixture pumped to the generator. Here heat is transferred to the solution from a source to vaporize the refrigerant and then it continues to run in condenser and the cycle completes.

Again the working fluid in an absorption refrigeration system is a binary solution consisting of refrigerant and absorbent. In Fig. 2.2(a), two evacuated vessels are connected to each other. The left vessel contains liquid refrigerant while the right vessel contains a binary solution of absorbent/refrigerant. The solution in the right vessel will absorb refrigerant vapor from the left vessel causing pressure to reduce. While the refrigerant vapor is being absorbed, the temperature of the remaining refrigerant will reduce as a result of its vaporization. This causes a refrigeration effect to occur inside the left vessel. At the same time, solution inside the right vessel becomes more dilute because of the higher content of refrigerant absorbed. This is called the "absorption process". Normally, the absorption process is an exothermic process; therefore, it must reject heat out to the surrounding in order to maintain its absorption capability.

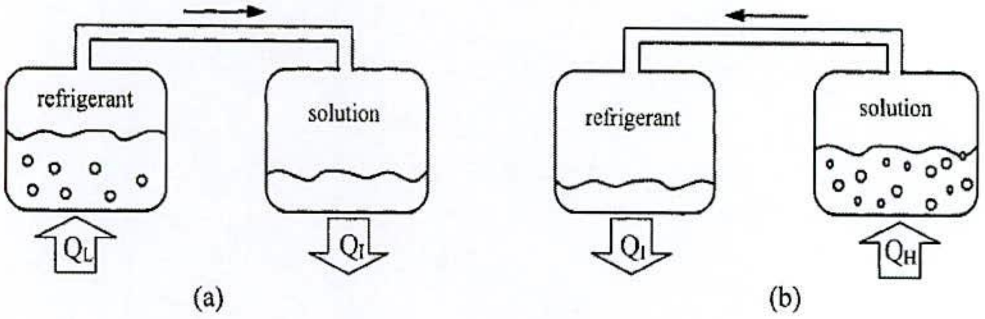


Fig 2.2 (a) Absorption process in right vessel causing cooling effect in the other; (b) Refrigerant separation process occurs in the right vessel as a result of additional heat from outside heat source.

Whenever the solution cannot continue with the absorption process because of saturation of the refrigerant, the refrigerant must be separated out from the diluted solution. Heat is normally the key for this separation process. It is applied to the right vessel in order to dry the refrigerant from the solution as shown in Figure 2.2(b). The refrigerant vapor will be condensed by transferring heat to the surroundings. With these processes, the refrigeration effect can be produced by using heat energy. However, the cooling effect cannot be produced continuously as the process cannot be done simultaneously. Therefore, an absorption refrigeration cycle is a combination of these two processes as shown in Figure 2.2. Coefficient of Performance of an absorption refrigeration system is obtained from;

$$COP = \frac{Cooling\ capacity\ obtained\ at\ evaporator}{Rate\ of\ heat\ input\ to\ the\ generator}$$

2.1.3 Working Fluid for Absorption Refrigeration Systems

Performance of absorption refrigeration systems is critically dependent on the chemical and thermodynamic properties of the working fluid. A fundamental requirement of absorbent/refrigerant combination is that, in liquid phase, they must have a margin of miscibility within the operating temperature range of the cycle. The mixture should also be chemically stable, non-toxic, and non-explosive. In addition to these requirements, the following are desirable.

The elevation of boiling (the difference in boiling point between the pure refrigerant and the mixture at the same pressure) should be as large as possible. Refrigerant should have high heat of vaporization and high concentration within the absorbent in order to maintain low circulation rate between the generator and the absorber per unit of cooling capacity.

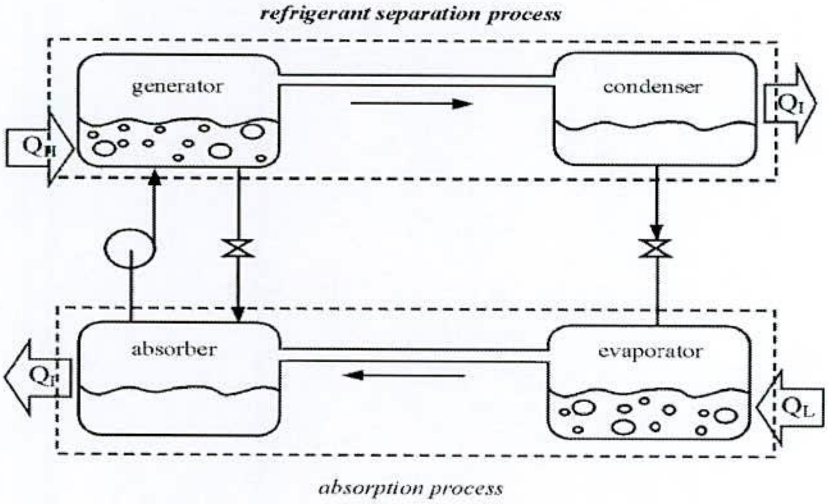


Figure 2.3: A continuous absorption refrigeration cycle composes of two processes

Transport properties that influence heat and mass transfer, e.g., viscosity, thermal conductivity, and diffusion coefficient should be favorable. Both refrigerant and absorbent should be non-corrosive, environmental friendly, and low-cost.

Many working fluids are suggested in literature. A survey of absorption fluids provided by Marcriss [9] suggests that, there are some 40 refrigerant compounds and 200 absorbent compounds available. However, the most common working fluids are Water/NH3 and LiBr/water.Since the invention of an absorption refrigeration system, water NH3 has been widely used for both cooling and heating purposes. Both NH3 (refrigerant) and water (absorbent) are highly stable for a wide range of operating temperature and pressure. NH3 has a high latent heat of vaporization, which is necessary for efficient performance of the system. It can be used for low temperature applications, as the freezing point of NH3 is 77°C. Since both NH3 and water are volatility, the cycle requires a rectifier to strip away water that normally evaporates with NH3. Without a rectifier, the water would accumulate in the evaporator and offset the system performance.

The use of LiBr/water for absorption refrigeration systems began around 1930. Two outstanding features of LiBr/water are non-volatility absorbent of LiBr (the need of a rectifier is eliminated) and extremely high heat of vaporization of water (refrigerant). However, using water as a refrigerant limits the low temperature application to that above 0°C. As water is the refrigerant, the system must be operated under vacuum conditions. At high concentrations, the solution is prone to crystallization. It is also corrosive to some metal and expensive. Thermodynamic properties of LiBr/water can be obtained from. Some additive may be added to LiBr/water as a corrosion inhibitor [10] or to improve heat-mass transfer performance.Although LiBr/water and water/NH3 have been widely used for many years and their properties are well known, much extensive research has been carried out to investigate new working fluids. Fluorocarbon refrigerant-based working fluids have been studied. R22 and R21 have been widely suggested because of their favorable solubility with number of organic solvents. The two solvents, which have stood out are Dimethyl Ether of Tetraethylene

Glycol (DMETEG) and Dimethyl Formamide (DMF). Research on these kinds of working fluids may be obtained from the literature [11, 12].A binary mixture using inorganic salt absorbent such as LiBr/water or NaOH/water may be the most successful working for an absorption refrigeration system. However, at high concentration such as at high temperature, the solution is prone to crystallization. It was found that the addition of a second salt as in a ternary mixture such as LiBr+ZnBr2/water can improve the solubility of the solution. Various ternary mixtures have been tested for using with an absorption system.

There are other disadvantages such as its high pressure, toxicity, and corrosive action to copper and copper alloy. However, water/NH3 is environmental friendly and low cost [13]. The relative comparison between various working pairs of absorbents and refrigerants [14] are given below:-

2.1.3.1 Water and Ammonia

☐ Most common working pair
☐ Water absorbent
☐ Ammonia refrigerant
☐ Continuous or intermittent absorption process
☐ Vaporized H20 reduces refrigeration
☐ High side 10-13 Bar @ 90 -100°C
☐ Ideal refrigerant regeneration below boiling point of water

2.1.3.2 Lithium Bromide and Water

☐ Environmentally safer than R12, R21, R134
☐ Lithium Bromide absorbent (like salt extreme hygroscopic) 26
☐ Water Refrigerant
☐ Absorption/adsorption process
☐ Continuousor intermittent process

☐ Refrigeration process requires a plumb/leveled working system

☐ Maximum high side temperature @ 552°C (melting point of Lithium Bromide)

2.1.3.3 Carbon and Methanol

☐ Intermittent adsorption process

☐ Carbon absorbent

☐ Methanolrefrigerant

☐ High side requires a vacuum condition prior to refrigeration regeneration

☐ High side 2-5 bar @ 90-110°C

2.1.3.4 Calcium Chloride and Ammonia

☐ Intermittent absorption process

☐ Calcium Chloride absorbent

☐ Ammonia refrigerant

☐ No vaporization of H_20 reducing refrigeration phase

☐ Does not require vacuum or level environment required

☐ Subject to heat crystallization

☐ Corrosive to aluminum brass and copper

☐ Target low side operation 20-30 °C

☐ Target high side operation 90-140°C

☐ Target operating pressure 8-14 Bar

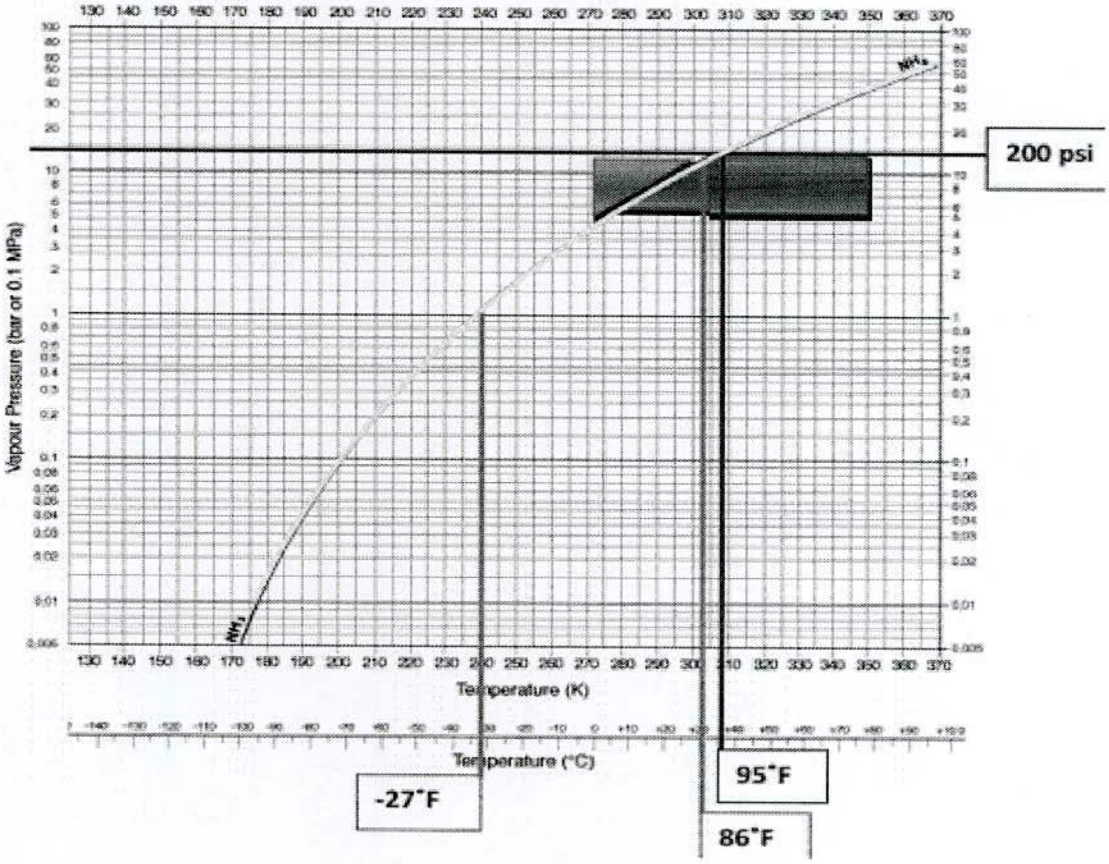


Figure 2.4: Temperature vs. pressure graph of ammonia

In this project ammonia (NH3) & Calcium Chloride ($CaCl_2$) are chosen as a refrigerant and absorbent because the difference between the vaporization temperatures is large. So it not essential to use any type of rectifier and analyzer in the system and the system is not bulky. The range of pressure is relatively low compare to other pairs and the system can work with a wide variety of temperature.

2.1.3.4.1 Ammonia (NH_3)

Molecular Weight: 17.03 g/mol

2.1.3.4.2 Solid phase

☐ Melting point:-78 °C

☐ Latent heat of fusion (1,013 bar, at triple point): 331.37 kJ/kg

2.1.3.4.3 Critical point

☐ Critical temperature: 132.4 °C

☐ Critical pressure: 112.8 bar

2.1.3.4.4 Miscellaneous

☐ Solubility in water (1.013 bar / 0 °C (32 °F)): 862 vol/vol

☐ Auto ignition temperature: 630 °C11

2.1.3.4.5 Liquid phase

☐ Liquid density (1.013 bar at boiling point): 682 kg/m3

☐ Liquid/gas equivalent (1.013 bar and 15 °C (59 °F)): 947 vol/vol

☐ Boiling point (1.013 bar):-33.5 °C

☐ Latent heat of vaporization (1.013 bar at boiling point): 1371.2 kJ/kg

☐ Vapor pressure (at 21 °C or 70 °F): 8.88 bar

2.1.3.4.6 Gaseous phase

☐ Gas density (1.013 bar at boiling point): 0.86 kg/m3 (1.013 bar and 15 °C (59 °F)): 0.73 kg/m3

☐ Compressibility Factor (Z) (1.013 bar and 15 °C (59 °F)): 0.9929

☐ Specific gravity (air = 1) (1.013 bar and 21 °C (70 °F): 0.597

☐ Specific volume (1.013 barand 21 °C (70 °F)): 1.411 m3/kg

☐ Heat capacity at constant pressure (Cp) 1.013 bar & 15°C (59°F):0.037 kJ/mol.K

☐ Heat capacity at constant volume (Cv) (1.013 bar & 15°C (59°F): 0.028 kJ/mol.K

☐ Ratio of specific heats (Gamma: Cp/Cv) (1.013 bar and 15°C(59°F): 1.309623

☐ Viscosity (1.013 bar and 0 °C (32 °F): 0.000098 Poise

☐ Thermal conductivity (1.013 bar and 0 °C (32 °F): 22.19 mW/(m.K)

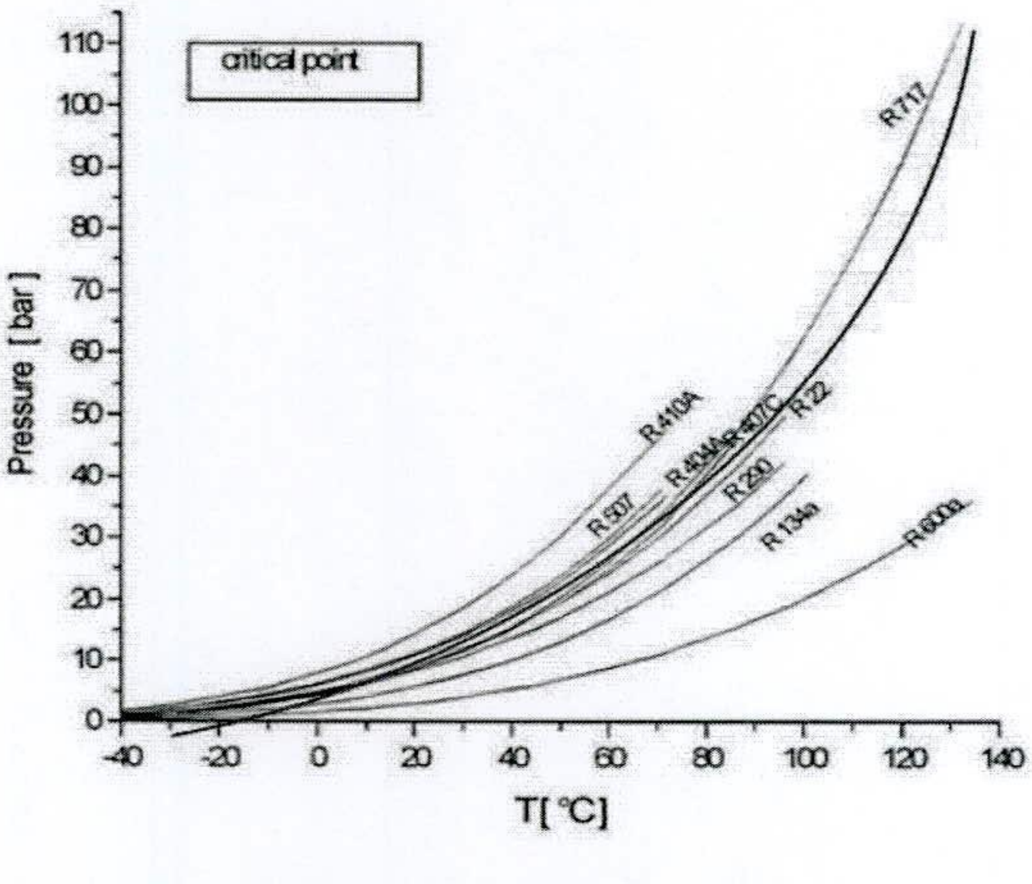


Figure 2.5: Temperature vs. pressure graph of common Refrigerants

2.1.3.5 Material Compatibility

Air Liquide Corp has assembled data on the compatibility of gases with materials to assist in evaluating which products can use be used for the gas system. Although the information has been compiled from what Air Liquide believes are reliable sources (International Standards: Compatibility of cylinder and valve materials with gas content; Part 1: ISO 11114-1 (Jul 1998), Part 2: ISO 11114-2 (Mar 2001)), NH$_3$it must be used with extreme caution.

The collected data mainly concern high pressure applications at ambient temperature and the safety aspect of material compatibility rather than the quality aspect [14].

Metals

☐ Aluminum	Satisfactory
☐ Brass	Non recommended
☐ Copper	Non recommended
☐ Ferritic Steels (e.g. Carbon steels)	Satisfactory
☐ Stainless Steel	Satisfactory

Plastics

☐ Polytetrafluoroethylene (PTFE)	Satisfactory
☐ Polychlorotrifluoroethylene (PCTFE)	Satisfactory
☐ Vinylidenepolyfluoride (PVDF)(KYNAR)	Non recommended, notable acceleration
☐ Polyamide (PA) (NYLON)	Satisfactory
☐ Polypropylene (PP)	Satisfactory
☐ ElastomersButyl (isobutene- isoprene)	Satisfactory
☐ Nitrile rubber (NBR)	Acceptable but significant loss of mass by extraction or chemical reaction.
☐ Chlorofluorocarbons (FKM) (VITON)	Non recommended, significant loss of mass by extraction or chemical reaction.
☐ Silicon (Q)	Non recommended, significant loss of mass by extraction or chemical reaction.
☐ Ethylene- Propylene (EPDM)	Satisfactory

Calcium Chloride ($CaCl_2$)

$CaCl_2$is a salt of calcium and chlorine which behaves as a typical ionic halide and is white solid at room temperature. Common applications include brine for refrigeration plants, ice and dust control on roads, and desiccation. Because of its hygroscopic nature, anhydrous calcium chloride must be kept in tightly-sealed air-tight containers.

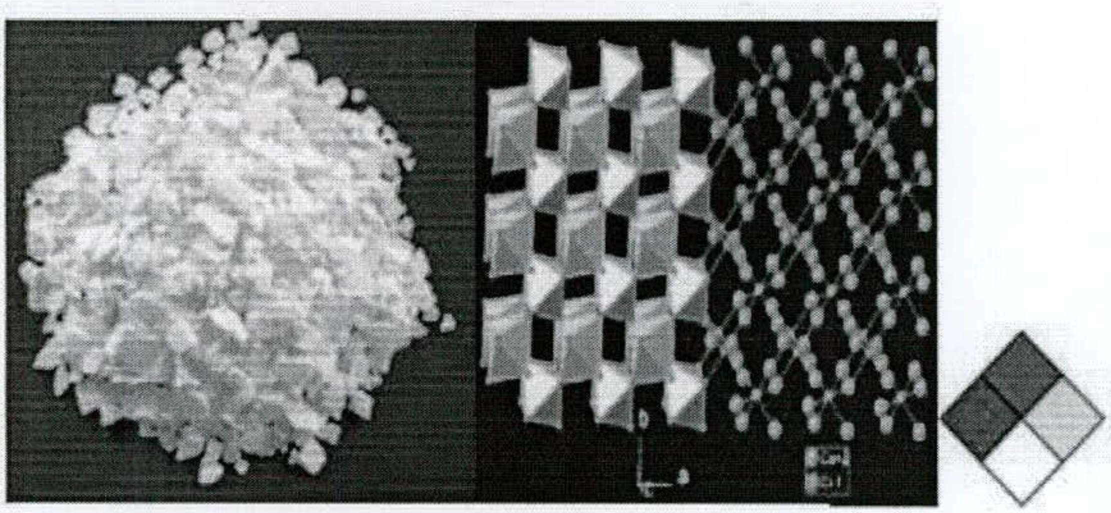


Figure 2.6: Calcium chloride in solid form (left), calcium chloride crystal structure (right)

Molar mass:	110.98 g/mol (anhydrous) 128.999 g/mol (monohydrate) 147.014 g/mol (dihydrate) 183.045 g/mol (tetrahydrate) 219.08 g/mol (hexahydrate)
Density:	2.15 g/cm^3(anhydrous) 1.835 g/cm^3(dihydrate) 1.83 g/cm^3(tetrahydrate) 1.71 g/cm^3(hexahydrate)

Melting point: 772 °C (anhydrous)
260 °C (monohydrate)
176 °C (dihydrate)
45.5 °C (tetrahydrate)
30 °C (hexahydrate)

Boiling point: 1935 °C (anhydrous)

2.1.4 Improving of Absorption Process

An absorber is the most critical component of any absorption refrigeration system. Experimental study shows that the solution circulation ratio (solution circulation rate per unit of refrigerant generated) is found 2 to 5 times greater than the theoretical value. This is due to a non-equilibrium state of solution in the absorber. For given temperature and pressure in the absorber, the solution absorbs fewer refrigerants than that of the theoretical value. Many researchers have been conducted in order to understand and to improve an absorption process between the vapor refrigerant and the solution.

The most common type of absorber used for LiBr/water system is absorption of vapor refrigerant into a falling film of solution over cooled horizontal tubes. In this type of absorber, during the absorption process, heat is simultaneously removed from the liquid film. Hence, the absorption rate is increased. However, this design requires a high recirculation rate in order to achieve a good performance. Another notable approach devised by Rotex is absorption of vapor refrigerant into liquid film on cooled rotating discs. For a given surface area, absorption rate on rotating discs is much greater than that on a convention design. Thus, size of an absorber used based on this design is much smaller than a convention falling film design. Absorption process within a rotating drum was also studied [15]. For water/NH3, literature on absorber designs is also provided.

2.1.5 Various Designs of Absorption Refrigeration Cycles

2.1.5.1 Single Effect Absorption System

A single-effect absorption refrigeration system is the simplest and most commonly used design. There are two design configuations depending on the working fluids used. Figure 2.7 shows a single-effect system using non-volatility absorbent such as LiBr/water.

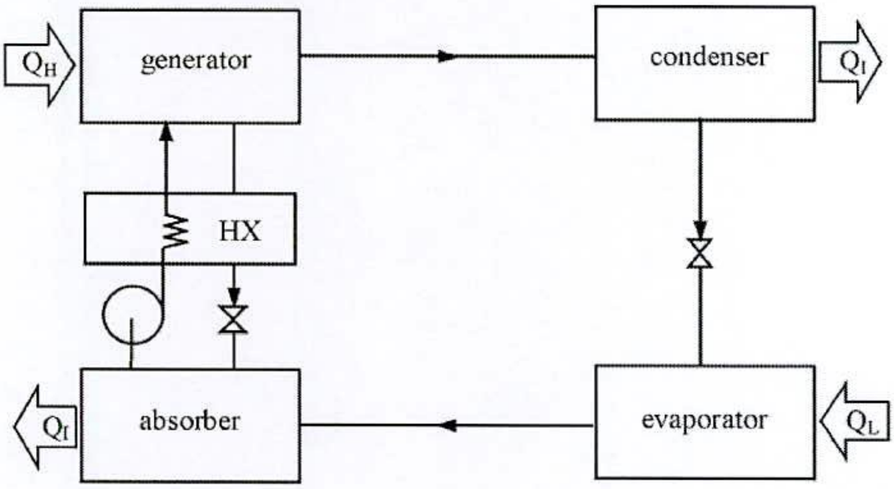


Figure 2.7: A single-effect LiBr/water absorption refrigeration system with a solution heat exchanger that helps decrease heat input at the generator

High temperature heat supplied to the generator is used to evaporate refrigerant out from the solution (rejected out to the surroundings of the condenser) and is used to heat the solution from the absorber temperature (rejected out to the surroundings of the absorber). Thus, irreversibility is caused as high temperature heat at the generator is wasted out at the absorber and the condenser. In order to reduce this irreversibility, a solution heat exchanger is introduced as shown in Figure2.7. The heat exchanger allows the solution from the absorber to be preheated before entering the generator by using the heat from the hot solution leaving the generator. Therefore, the COP is improved as the heat input at the generator is reduced. Moreover, the size of the absorber can be reduced as less heat is rejected. Experimental studies shows that COP can be increased up to 60% when a solution heat exchanger is used [16].

When volatile absorbent such as water/ NH_3 is used, the system requires an extra component called "a rectifier", which will purify the refrigerant before entering the condenser. As the absorbent used (water) is highly volatile, it will be evaporated together with ammonia (refrigerant). Without the rectifier, this water will be condensed and accumulated inside the evaporator, causing the performance to drop.

Even if the most common working fluids used are $CaCl_2/NH_3$, LiBr/water, and water/NH_3,various researchers have studied performance of a single-effect absorption system using other kinds of working fluids such as $SrCl_2/NH_3$, $LiNO_3/NH_3$, $NaSCN/NH_3$, LiBr+$ZnBr_2$/CH3OH, LiCl/water, Glycerol/water [17,18].

2.1.5.2 Absorption Heat Transformer

Any absorption refrigeration cycle exchanges heat with three external reservoirs; low, intermediate, and high temperature levels. When an absorption system is operated as a refrigerator or a heat pump, the driving heat is supplied from the high temperature reservoir. Refrigeration effect is produced at a low temperature level and rejects heat out at an intermediate temperature level. The difference between them is the duty. For a refrigerator, the useful heat transfer is at a low temperature. For the heat pump, the useful heat transfer is at an intermediate temperature. Normally, the surrounding is used as a low temperature reservoir for a heat pump or as an intermediate temperature reservoir for the refrigerator.

Another type of absorption cycle is known as "an absorption heat transformer" or "a reverse absorption heat pump". This system uses heat from an intermediate temperature reservoir as the driving heat (normally from industrial waste heat). The system rejects heat out at a low temperature level (normally to the surroundings).

The useful output is obtained at the highest temperature level. The use of an absorption heat transformer allows any waste heat to be upgraded to a higher temperature level without any other heat input except some work required circulating the working fluid.

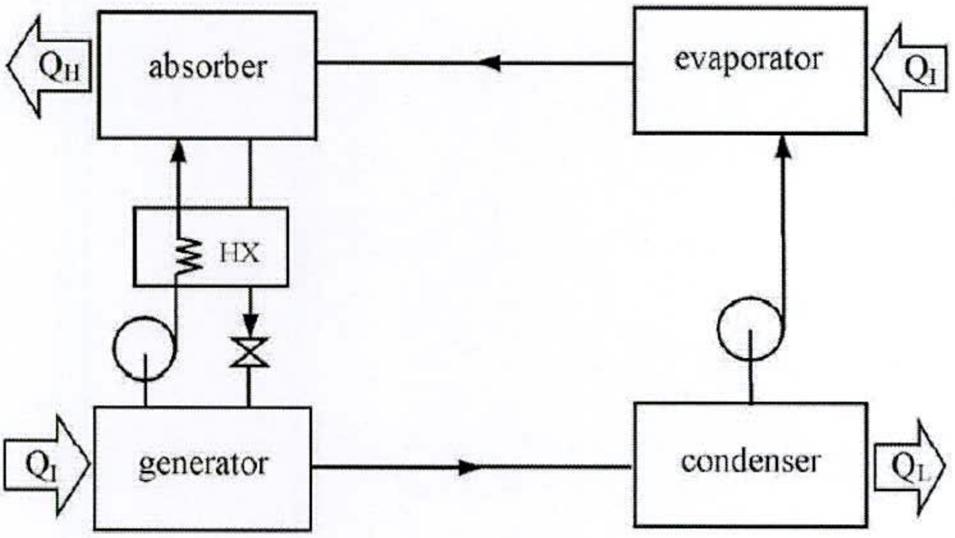


Figure 2.8: Absorption heat transformer absorbs waste heat at the generator. Liquid refrigerant is pumped to the evaporator to absorb waste heat. High temperature useful heat from the absorber is heat of absorption

Figure 2.8 shows a schematic diagram of an absorption heat transformer. This cycle has similar components as a single-effect absorption cycle. The difference is that an expansion device installed between the condenser and the evaporator is substituted by a pump. Waste heat at a relatively low temperature is supplied to the generator for refrigerant separation in the usual manner. Liquid refrigerant from the condenser is then pumped to the evaporator with elevated pressure. In the evaporator, it is vaporized by using the same low temperature waste heat used to drive the generator (absorption heat transformers are usually operated so that the generator and evaporator temperatures are equal). The vapor refrigerant is then absorbed into solution in the absorber which rejects the useful heat out at a high temperature level. Low-grade heat can be upgraded by using a heat transformer e.g. solar energy [19], industrial waste heat. Performance of an absorption heat transformer with various working fluids has been studied; LiBr/water, LiBr+ZnBr2/CH3OH, DMETEG/R21, DMF/R21 [20, 21].

2.1.5.3 Multi-Effect Absorption Refrigeration Cycle

The main objective of a higher effect cycle is to increase system performance when high temperature heat source is available. By the term "multi-effect", the cycle has to be configured in a way that heat rejected from a high-temperature stage is used as heat input in a low-temperature stage for generation of additional cooling effect in the low-temperature stage. Double-effect absorption refrigeration cycle was introduced during 1956 and 1958 [22]. Figure 2.9 shows a system using LiBr/water. High temperature heat from an external source supplies to the first-effect generator. The vapor refrigerant generated is condensed at high pressure in the second-effect generator. The heat rejected is used to produce addition refrigerant vapor from the solution coming from the first-effect generator. This system configuration is considered as a series-flow-double-effect absorption system.

A double-effect absorption system is considered as a combination of two single effect absorption systems whose COP value is COP single. For one unit of heat input from the external source, cooling effect produced from the refrigerant generated from the first-effect generator is 1×COPsingle. For any single-effect absorption system, it may be assumed that the heat rejected from the condenser is approximately equal to the cooling capacity obtained. Thus the heat supply to the second generator is 1×COP single. The cooling effect produced from the second-effect generator is (1×COPsingle) ×COPsingle. Therefore, the COP of this double-effect absorption system is COPdouble=COPsingle+(COPsingle)2. According to this analysis, a double effect absorption system has a COP of 0.96 when the corresponding single-effect system has a COP of 0.6. Theoretical studies of a double-effect absorption system have been provided for various working fluids [23, 24].

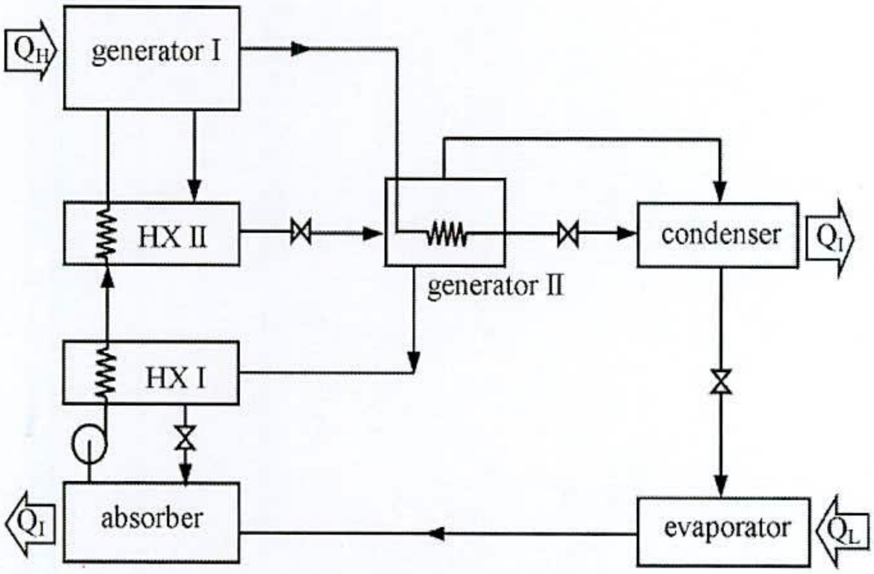


Figure 2.9: A double-effect water/LiBr absorption cycle. Heat released from the condensation of refrigerant vapor is used as heat input in generator II. This cycle is operated with 3 pressure levels i.e. high, moderate and low pressure

If LiBr/water is replaced with water/ NH_3, maximum pressure in the first-effect generator will be extremely high. Figure 2.10 shows a double-effect absorption system using water/NH_3. In contrast to the system for LiBr/water, this system can be considered as a combination of two separated single-effect cycles. The evaporator and the condensers of both cycles are integrated together as a single unit as shown. Thus, there are only two pressures levels in this system and the maximum pressure can be limited to an acceptable level. Heat from external source supplies to generator II only as water is an absorbent, there is no problem of crystallization in the absorber. Hence, absorber II can be operated at high temperature and rejects heat to the generator I. This system configuration is considered as a parallel-flow-double-effect absorption system.

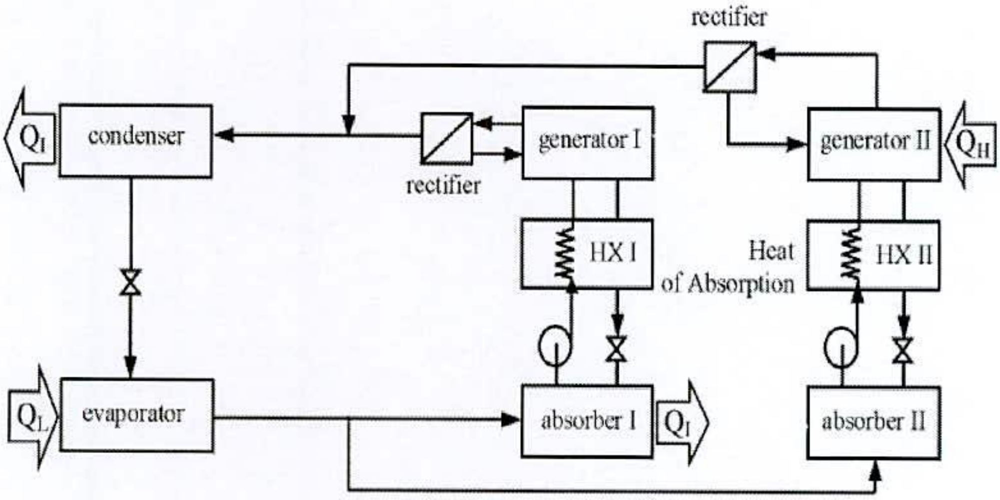


Figure 2.10: A double-effect absorption cycle operates with two pressure levels. Heat of absorption from absorber II is supplied to the Absorber I for the refrigerant separation process

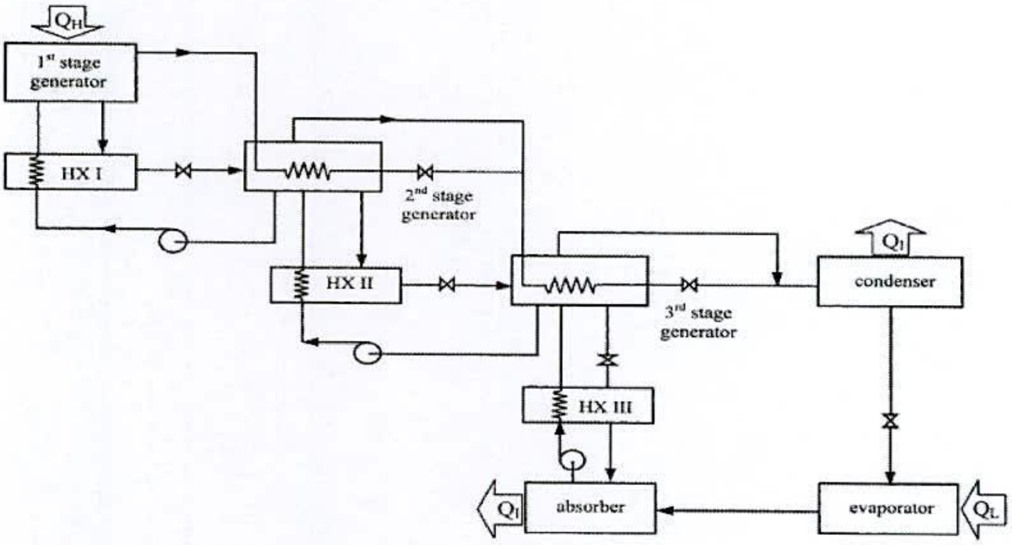


Figure 2.11: A triple-effect absorption cycle operates at 4 pressure levels. Heat of condensation from the higher pressure stage is used for refrigerant separation in the lower-pressure stage

Several types of multi-effect absorption cycle has been analyzed such as the triple-effect absorption cycle (Figure 2.11) and the quadruple-effect absorption cycle. However, an improvement of COP is not directly linked to the increment of number of effect. It must be noted that, when the number of effects increase, COP of each effect will not be as high as that for a single-effect system. Moreover, the higher number of effect leads to more system complexity. Therefore, the double-effect cycle is the one that is available commercially [25].

2.1.5.4 Absorption Refrigeration Cycle with GAX

GAX stands for generator/absorber heat exchanger or sometimes is called DAHX which stands for desorber/absorber heat exchanger. Higher performance can be achieved with a single-effect absorption system. Referring to the parallel flow double effect absorption system mentioned earlier, the system consists of two single-effect cycles working in a parallel manner. The concept of GAX is to simplify these two stages double effect absorption cycle but still produce the same performance. The ideal of GAX was introduced in 1911 by Altenkirch and Tenckhoff [26]. The simplified configuration is shown schematically in Figure 2.12.

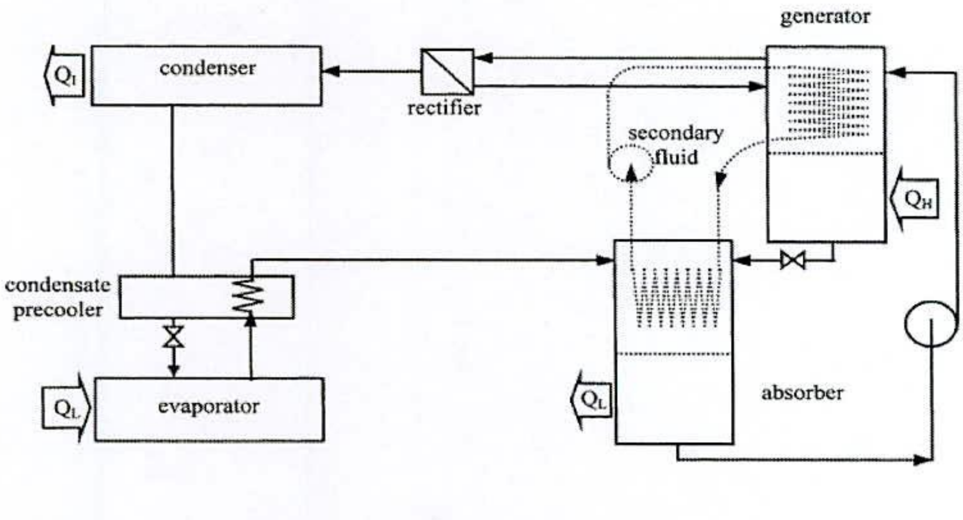


Figure 2.12: The dotted loop shows secondary fluid used for transferring heat from high the temperature section in the absorber to low temperature section in the generator

An absorber and a generator may be considered as a counter-flow-heat exchanger as shown in Figure 2.12. At the absorber, weak-refrigerant solution from the generator and vapor refrigerant from the evaporator enter at the top section. Heat produced during the absorption process must be rejected out in order to maintain ability to absorb the refrigerant vapor. At the top section, heat is rejected out at a high temperature. In the lower section, the solution further absorbs the vapor refrigerant while cooling down by rejecting heat to the surrounding. At the generator, rich-refrigerant solution from the absorber enters at the top section. In this section, the refrigerant is dried out from the solution as it is heated by using the heat rejected from the top section of the absorber. At the lower section of the generator, the solution is further dried as it is heated by the external source. Referring to Figure 8, there is an additional secondary-fluid, which used for transferring heat between the absorber and the generator. Therefore, a single-effect absorption system can provide as high COP as that for the two-stage-double-effect absorption system by using GAX. This system has been studied [27].

2.1.5.5 Absorption Refrigeration Cycle with an Absorber Heat Recovery

It is already mentioned earlier that the use of a solution heat exchanger improves the system COP. Rich-refrigerant solution from the absorber can be preheated before entering the generator by transferring heat from hot solution coming from the generator. By introducing an absorber-heat-recovery, temperature of the rich-refrigerant solution can be further increased.

Similar to the GAX system, the absorber is divided into two sections. Heat is rejected out at a different temperature. The lower temperature section rejects heat out to the surroundings as usual. However, the higher temperature section is used to preheat rich-refrigerant solution as shown in Figure 2.13. Therefore, the heat input to the generator is reduced causing the COP to increase. This system was studied theoretically by using various working fluids; water/NH3 and LiNO3/NH3 [28, 29]. The cycle with an absorber-heat-recovery was found to have 10% improvement in COP. However, the machine based on this absorber design has not yet been built.

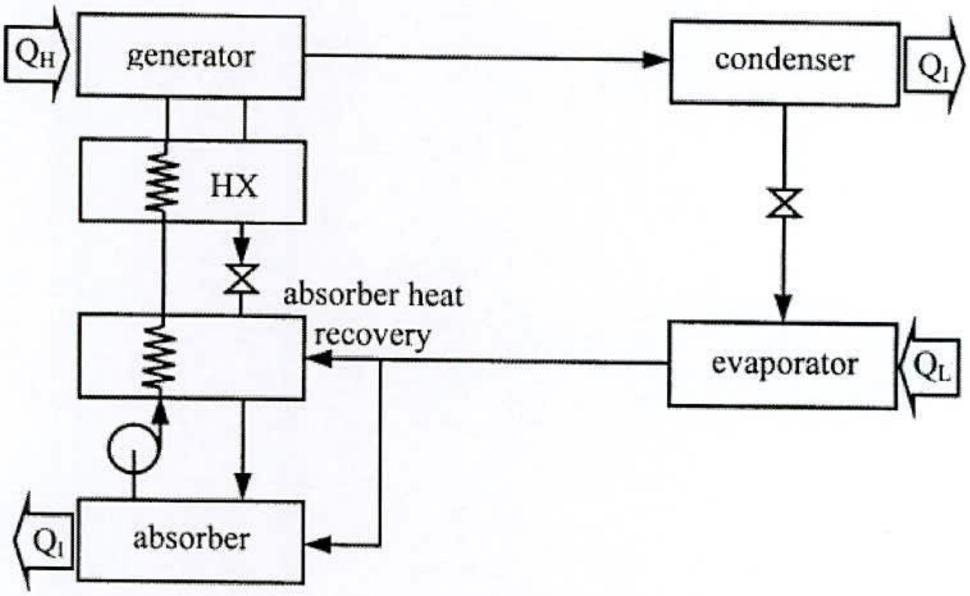


Figure 2.13: The cycle with absorber heat recovery uses heat of absorption to preheat the outgoing stream from the absorber to the generator

2.1.5.6 Half-Effect Absorption Refrigeration Cycle

It must be noted that, any absorption refrigeration system can be operated only when the solution in the absorber is richer in refrigerant than that in the generator. When the temperature increases or the pressure reduces, the fraction of refrigerant contained in the solution is reduced, and vice versa. When the generator temperature is dropped, the solution circulation rate will be increased causing the COP to drop. If it is too low, the system can be no longer operated.

A half-effect absorption cycle is a combination of two single-effect cycles but working at different pressure levels. Letting heat source temperature be lower than the minimum temperature is necessary for a single-effect cycle working at the same pressure level.

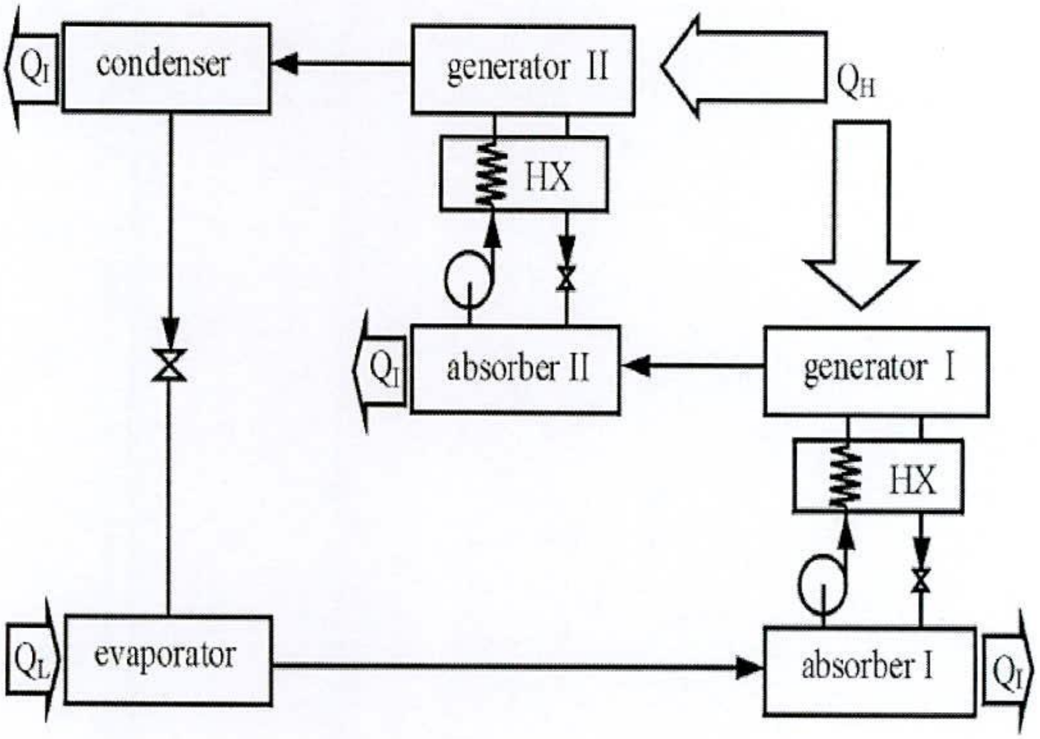


Figure 2.14: A half-effect absorption cycle

The half-effect absorption system was introduced for an application with a relatively low-temperature heat source [30]. Figure 2.14 shows a schematic diagram of a half-effect absorption refrigeration cycle. The system configuration is exactly the same as the double-effect absorption system using water/NH3 except the heat flow directions are different. Referring to Figure 2.14, high temperature heat from an external source transfers to both generators and absorbers reject heat out to the surroundings. Absorber II and generator I am operated at an intermediate pressure level. Therefore, the circulation rate between generator I and absorber I and between generator II and absorber II can be maintained at acceptable levels. It must be noted that COP of the half-effect absorption system is relatively low as it rejects more heat than a single-effect absorption cycle around 50% [31]. However, it can be operated with the relatively low temperature heat source.

2.1.5.7 Combined Vapor Absorption-Compression Cycle

This system is usually known as an absorption-compression system. A schematic diagram of a typical absorption/compression cycle is shown in Fig 2.15 (a). It can be seen that, a condenser and an evaporator of a conventional vapor-compression system are replaced with a vapor absorber and a vapor generator. Forgiven surrounding temperature and refrigerating temperature, the pressure differential across the compressor is much lower than a conventional vapor-compression system. Thus, the COP is expected to be better than a conventional vapor-compression system. Altenkirch did the first investigation in 1950 and proposed a potential for energy-saving [31]. The cycle can be configured as a heat pump cycle. Machielsen developed a heat pump cycle as shown in Fig 2.15(b).

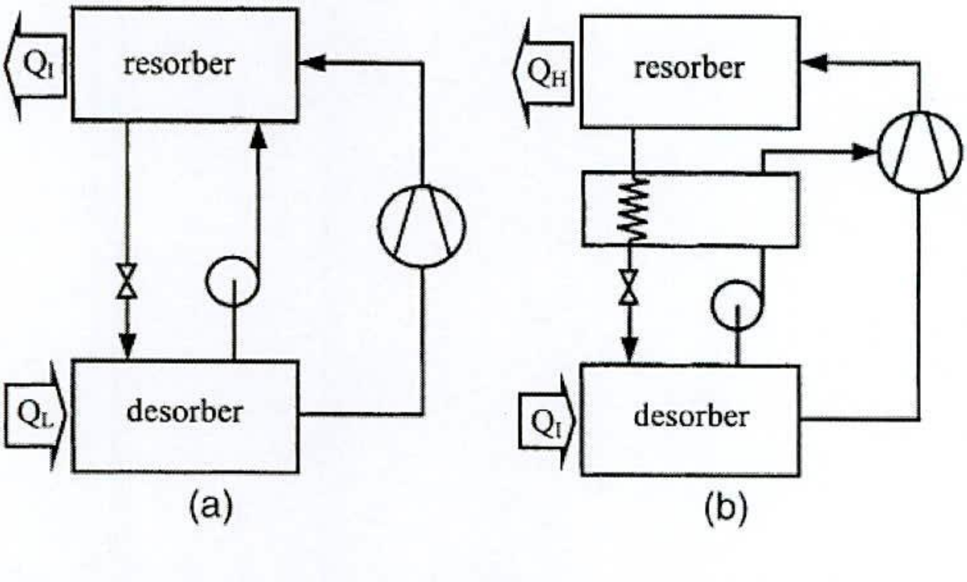


Figure 2.15: Combined vapor absorption/compression heat pump

An interesting configuration is a double-effect vapor absorption/compression cycle as shown in Fig 2.16. The rejected first stage absorber heat is supplied to the generator of the second-stage. The transfer of heat is done internally which overcomes the large temperature difference at the moderate pressure ratio. This concept has been shown successfully in several studies [32].

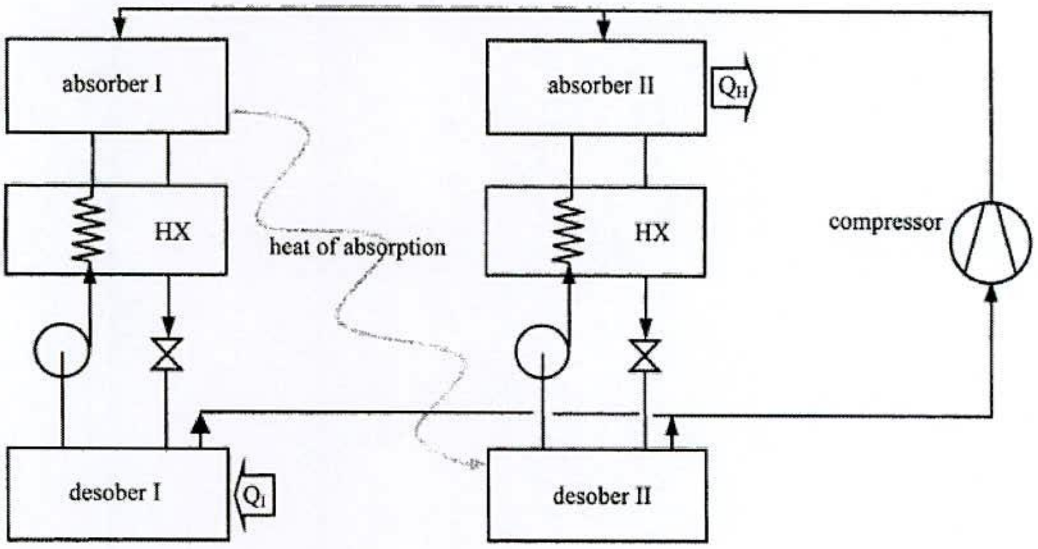


Figure 2.16: A double effect absorption-compression cycle is configured as a heat pump. Heat of absorption in the first stage will be supplied to the second stage for refrigerant separation

Another configuration of the vapor absorption/compression cycle, proposed by Cacciola [33] et al. is shown schematically in Figure 2.17 and employs two combinations of working fluids, water/NH3 and KHO/water. This is a compromise of the water/NH3 cycle and KHO/water cycle. The highest system pressure is reduced and the rectifier of water/NH3 system is abstained. This cycle can be operated with an ambient temperature lower than 0°C without freezing or crystallization problems. The first experimental results of an absorption/compression cycle with direct desorber/absorber heat exchanger was presented by Groll and Radermacher [34].

This is a modified plant from a two stage-solution circuit proposed by Rane and Radermacher and Rane et al. This technology is the basis for the study of GAX cycle in these days. Various designs of combined vapor absorption/compression cycle have been introduced. They can produce attractively high COP. However, they are complex and the driving energy is in the form of mechanical work. Thus, they cannot be considered as a heat-operated system.

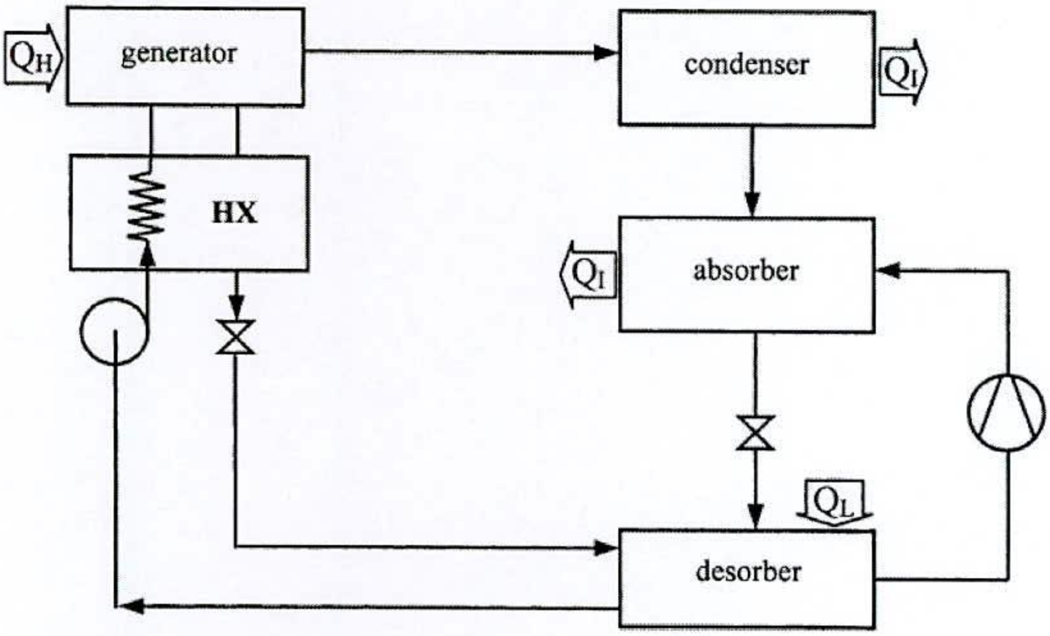


Figure 2.17: A combined cycle proposed by Caccoila et al. employing two combinations of working fluids i.e.NH_3/H_2O and H_2O/KHO. The rectifier is absent and also the highest pressure is decreased

2.1.5.8 Dual-Cycle Absorption Refrigeration

The concept of a dual-cycle absorption system is similar to a parallel-double-effect absorption system. However, this system consists of two completely separated cycles using different kinds of working fluid. Hanna [35] et al. invented a dual-cycle absorption refrigeration and heat pump as shown in Figure 2.18. This system consists of two single effect absorption cycles using water/NH_3 and LiBr/water.

The NH_3 system is driven by heat obtained from an external heat source. The heat reject from its absorber and condenser is used as a driving heat for the LiBr/water system. The LiBr/water system rejects heat out to the surrounding at the condenser and the absorber as usual. The cooling effect can be obtained from both evaporators.

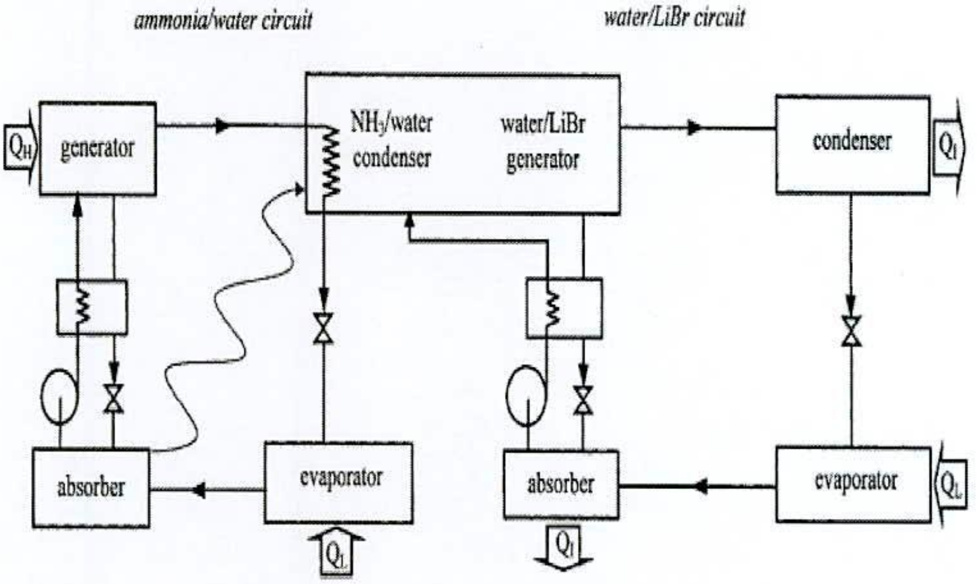


Figure 2.18: Solar driven dual cycle absorption employs two different working fluids i.e. NH3/water and water/LiBr. Heat of absorption and condensation from NH3/water cycle are supplied to the generator of water/LiBr cycle

2.1.5.9 Self-Circulation Absorption System Using LiBr/Water

Even if the prime energy for an absorption refrigeration system is in the form of heat, some electricity still required to drive a circulation pump. There are some absorption refrigeration systems that do not require any circulation pump. In such a system, working fluid is circulated naturally by a thermo syphon effect known as a bubble pump.

Yazaki Inc. of Japan introduced a self-circulate absorption refrigeration system based on a single-effect system using LiBr/water. Using water as a refrigerant, differential pressure between the condenser and the evaporator is very low and can be maintained by using the principle of hydrostatic-head. The solution from the absorber can be circulated to the generator by a bubble pump. Heat input to the generator is used for both circulation of working fluid and evaporation of refrigerant.

The weak-refrigerant solution returns gravitationally back to absorber. A schematic diagram of this system is shown in Figure 2.19. With the effect of the bubble pump, the solution is boiled and pumped at the same time. Smith and Khahra [36] carried out a study of performance of CH-900-B Yazaki absorption water chiller operated using propane gas. Eriksson and Jernqvist, developed a 10 kW self-circulation absorption heat transformer using NaOH/water. Due to the high temperature and pressure differential between the condenser and the evaporator, the absorber and evaporator are located at 7 and 10 m below the condenser and generator, respectively. The lowest and highest point of this machine is 14 m. which is equivalent to a pressure difference of 1 bar inside the system.

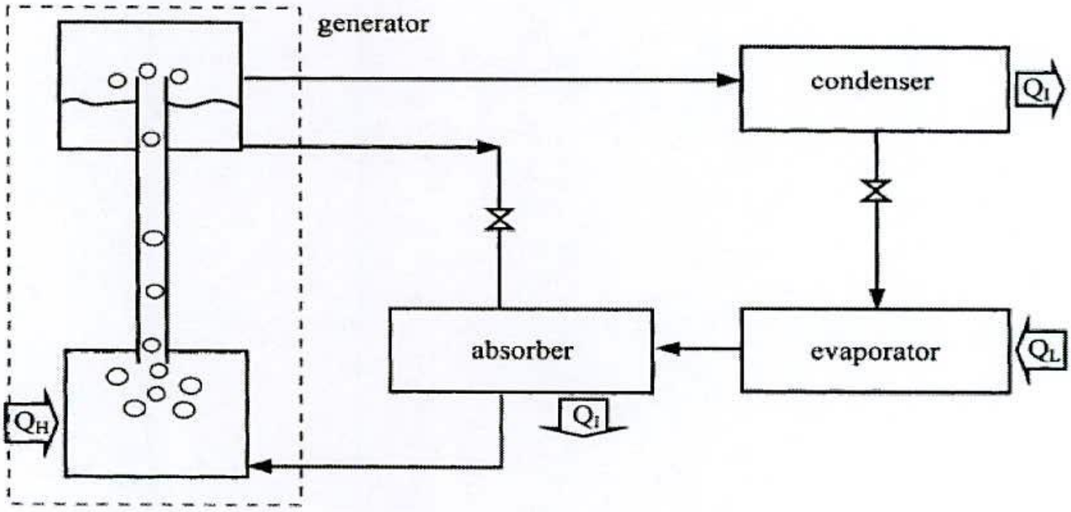


Figure 2.19: Bubbles pump in a generator module

2.1.6 Different Types of Heat Exchanger

A variety of heat exchangers are used in various industrial purposes. The objective of this section is to describe most of these heat exchangers in some detail using classification schemes. Starting with a definition, heat exchangers are classified according to heat transfer processes, number of fluids, and degree of surface compactness, construction features, flow arrangements, and heat transfer mechanisms. With a detailed classification in each category, the terminology associated with a variety of these exchangers is introduced and practical applications are outlined. Short mentions of various types of heat exchangers are given below.

2.1.6.1 Direct-Contact Heat Exchangers

In a direct-contact exchanger, two fluid streams come into direct contact, exchange heat, and are then separated. Common applications of a direct-contact exchanger involve mass transfer in addition to heat transfer, such as in evaporative cooling and rectification; applications involving only sensible heat transfer are rare. The enthalpy of phase change in such an exchanger generally represents a significant portion of the total energy transfer. The phase change generally enhances the heat transfer rate. Compared to indirect contact recuperates and regenerators, in direct-contact heat exchangers, very high heat transfer rates are achievable, the exchanger construction is relatively inexpensive and the fouling problem is generally nonexistent, due to the absence of a heat transfer surface (wall) between the two fluids.

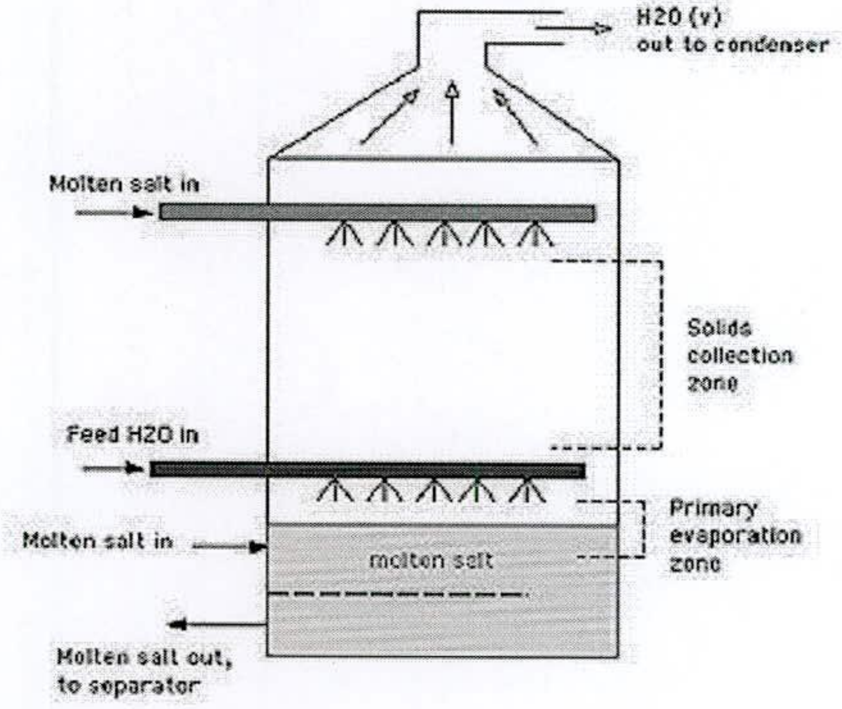


Figure 2.20: Direct-Contact Heat Exchangers

2.1.6.2 Storage Type Exchangers

In storage type exchanger, both fluids flow alternatively through the same flow passages, and hence heat transfer is intermittent. The heat transfer is generally cellular in structure and is referred to as a matrix or it is a permeable solid material, referred to as a packed bed. When hot gas flows over the heat transfer surface, the thermal energy from the hot gas is stored in the matrix wall, and thus the hot gas is being cooled during the matrix heating period. As cold gas flows through the same passages later, the matrix wall gives up thermal energy, which is absorbed by the cold fluid. Thus, heat is not transferred continuously through the wall as in a direct-transfer type exchanger, but the corresponding thermal energy is alternately stored and released by the matrix wall. This storage type heat exchanger is also referred to as a regenerative heat exchanger, or simply as a regenerator energy, which is absorbed by the cold fluid.

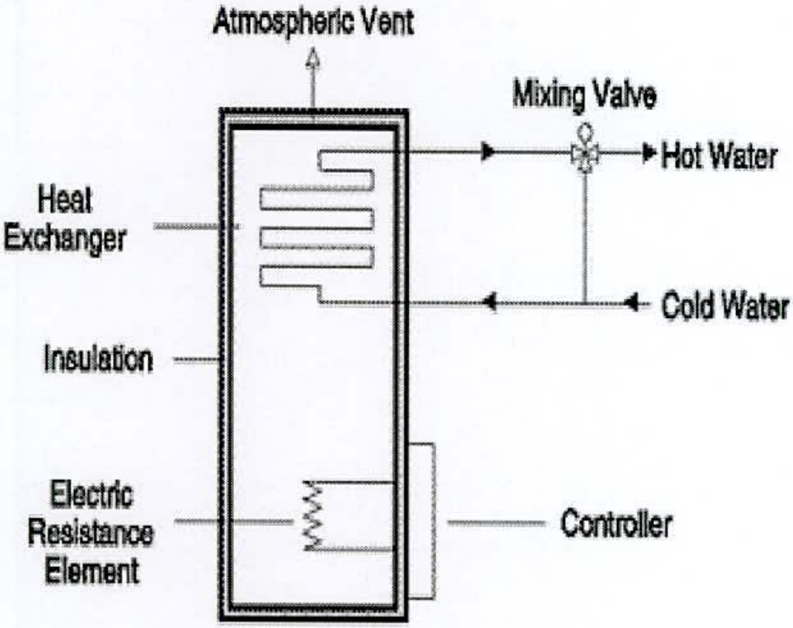


Figure 2.21: Storage Type Exchangers

Thus, heat is not transferred continuously through the wall as in a direct-transfer type exchanger, but the corresponding thermal energy is alternately stored and released by the matrix wall. This storage type heat exchanger is also referred to as a regenerative heat exchanger, or simply as a regenerator

2.1.6.3 Fluidized Heat Exchangers-Bed

In a fluidized-bed heat exchanger, one side ofa two-fluid exchanger is immersed in a bed of finely divided solid material, such as a tube bundle immersed in a bed of sand or coal particles, as shown in Figure 2.22 If the upward fluid velocity on the bed side is low, the solid particles will remain fixed in position in the bed and the fluid will flow through the interstices of the bed.

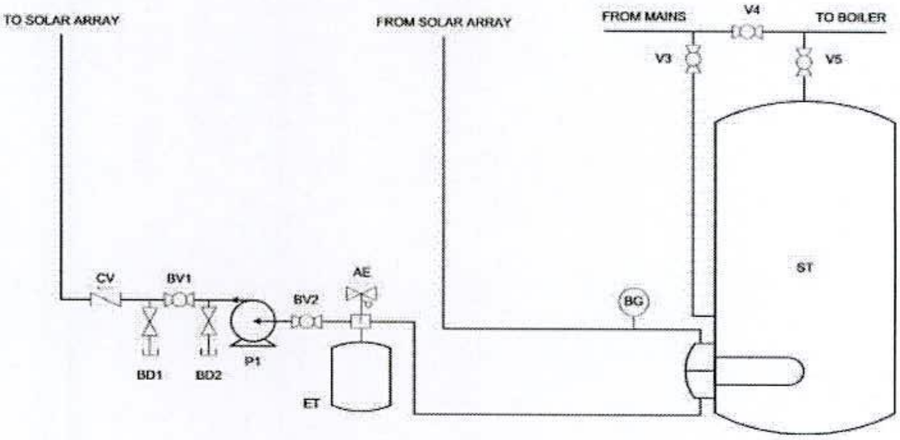


Figure 2.22: Fluidized Heat Exchangers –Bed

If the upward fluid velocity is high, the solid particles will be carried away with the fluid. At a ‘‘proper’’ value of the fluid velocity, the upward drag force is slightly higher than the weight of the bed particles. As a result, the solid particles will float with an increase in bed volume, and the bed behaves as a liquid. This characteristic of the bed is referred to as a fluidized condition. Under this condition, the fluid pressure drop through the bed remains almost constant, independent of the flow rate, and a strong mixing of the solid particles occurs. This results in a uniform temperature for the total bed (gas and particles) with an apparent thermal conductivity of the solid particles as infinity. Very high heat transfer coefficients are achieved on the fluidized side compared to particle-free ordilute-phase particle gas flows.

2.1.6.4 Liquid–Vapor Exchangers

In this type, typically steam is partially or fully condensed using cooling water, or water is heated with waste steam through direct contact in the exchanger. Non condensable and residual steam and hot water are the outlet streams. Common examples are desuperheaters and open feed water heaters in power plants.

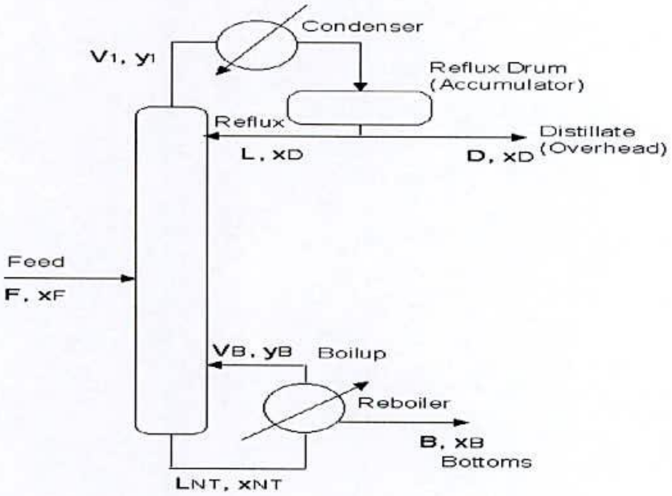


Figure 2.23: Liquid–Vapor Exchangers

2.1.6.5 Tubular Heat Exchangers

These exchangers are generally built of circular tubes; although elliptical, rectangular, flat twisted tubes have also been used in some applications. There is considerable flexibility in the design because the core geometry can be varied easily by changing the tube diameter, length, and arrangement.

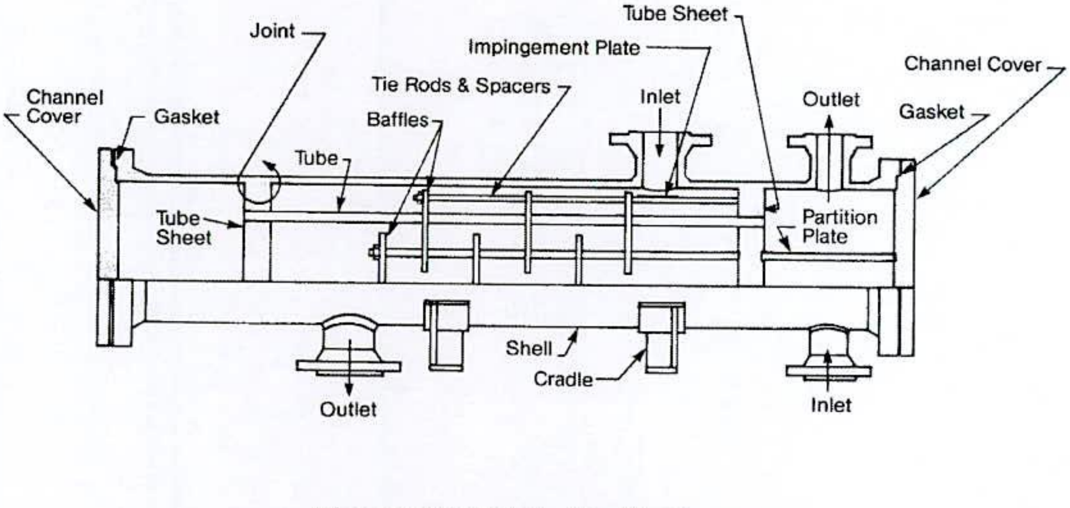
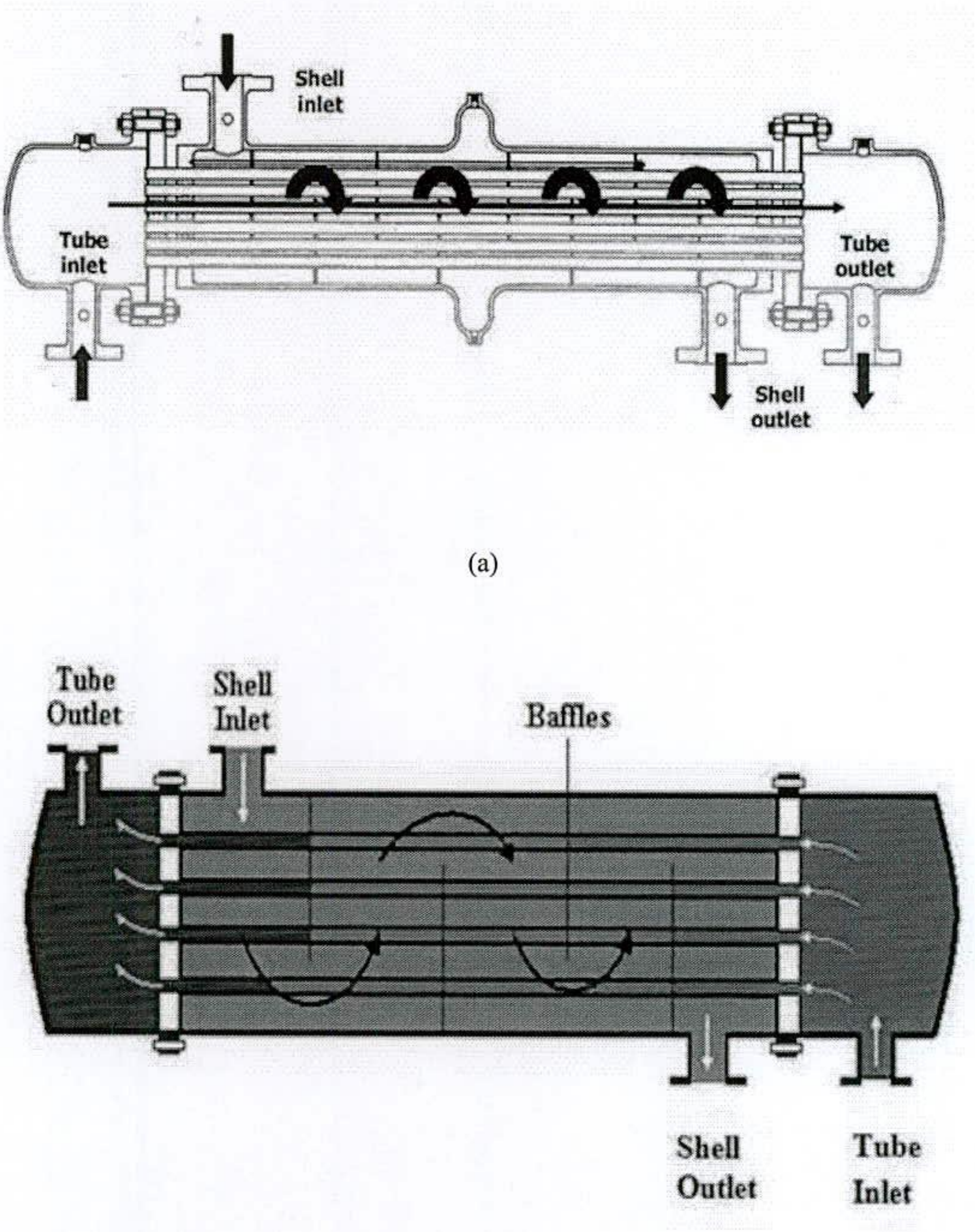


Figure 2.24: Tubular Heat Exchangers

Tubular exchangers can be designed for high pressures relative to the environment and high-pressure differences between the fluids. Tubular exchangers are used primarily for liquid-to-liquid and liquid-to-phase change (condensing or evaporating) heat transfer applications. They are used for gas-to-liquid and gas-to-gas heat transfer applications primarily when the operating temperature and/or pressure is very high or fouling is a severe problem on at least one fluid side and no other types of exchangers would work. These exchangers may be classified as shell and tube, double-pipe, and spiral tube exchangers. They are all prime surface exchangers except for exchangers having fins outside/inside tubes.

2.1.6.6 Shell-and-Tube Exchangers

Shell-and-Tube exchanger, shown in Figure 2.25 is generally built of a bundle of round tubes mounted in a cylindrical shell with the tube axis parallel to that of the shell. One fluid flows inside the tubes and the other fluid flows across the tubes surface continuously. The major components of this exchanger are tubes (or tube bundle), shell, front end head, rear-end head, baffles, and tube sheets. The three most common types of shell-and-tube exchangers are (1) fixed tube sheet design, (2) U-tube design, and (3) floating-head type. In all three types, the front-end head is stationary while the rear-end head can be either stationary or floating, depending on the thermal stresses in the shell, tube, or tube sheet, due to temperature differences as a result of heat transfer.



(a)

(b)

Figure 2.25: Shell-and-Tube Exchangers

2.1.6.7 Spiral Tube Heat Exchangers

These consist of one or more spirally wound coils fitted in a shell. Heat transfer rate associated with a spiral tube is higher than that for a straight tube. In addition, a considerable amount of surface can be accommodated in a given space by spiraling. Thermal expansion is no problem, but cleaning is almost impossible.

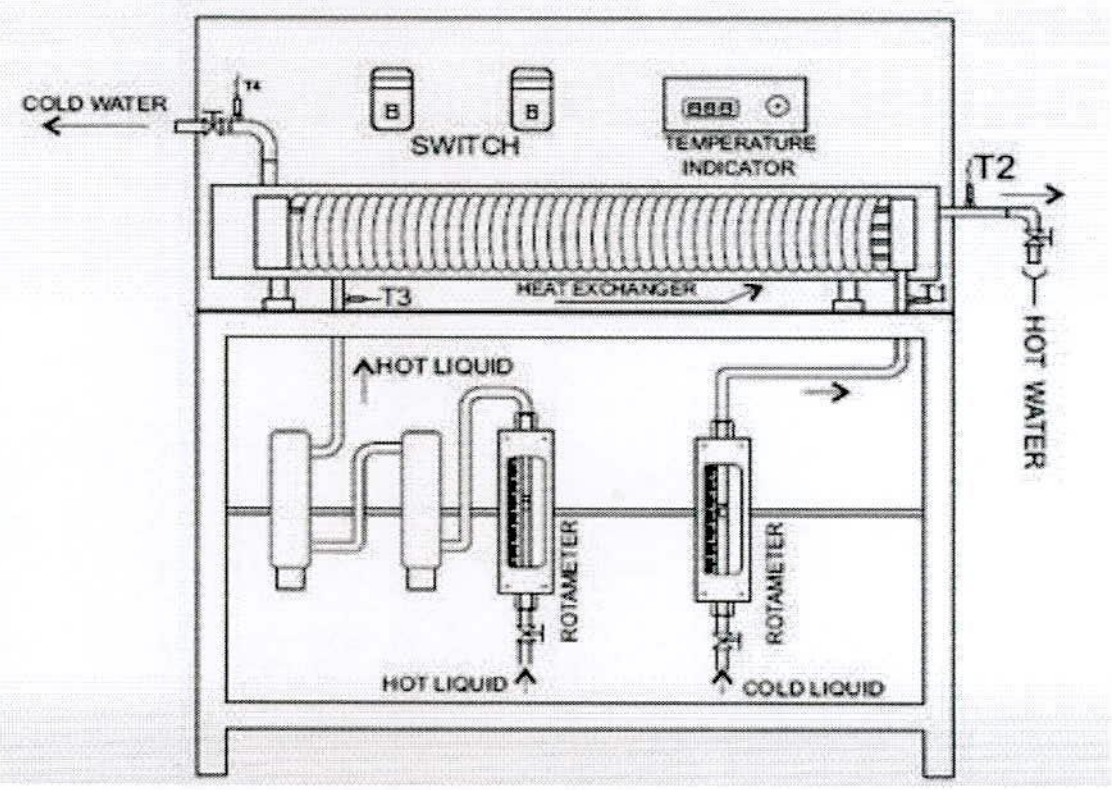
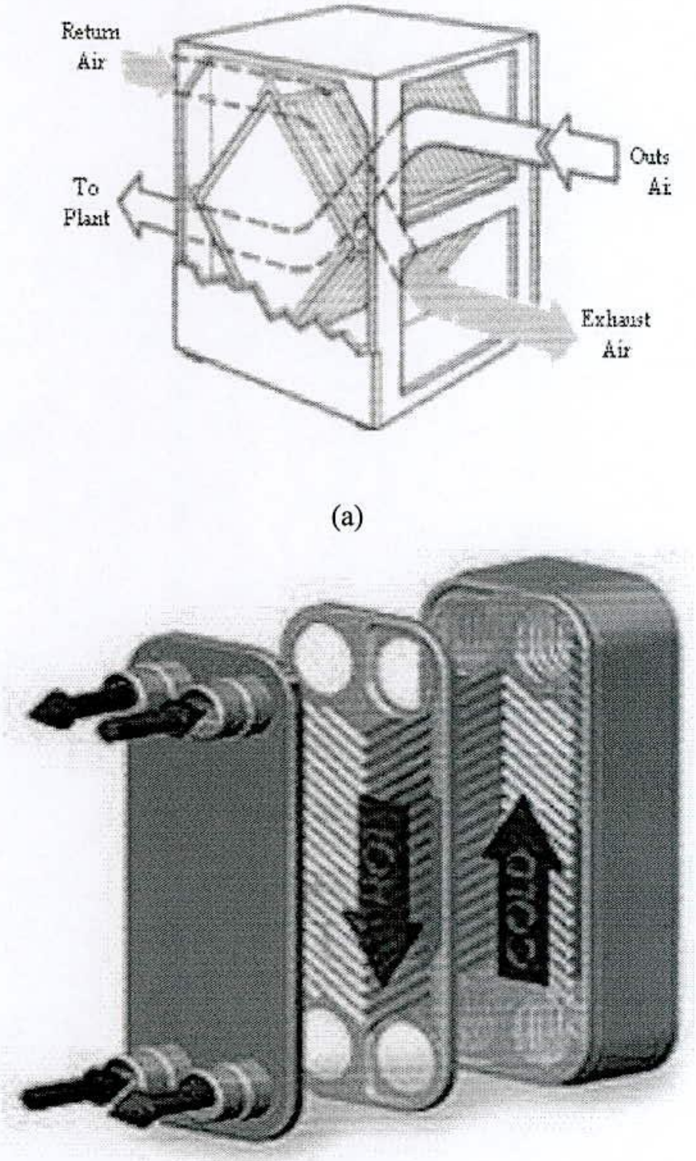


Figure 2.26: Spiral Tube Heat Exchangers

2.1.6.8 Plate-Type Heat Exchangers

Plate-type heat exchangers are usually built of thin plates. The plates are either smooth or have some form of corrugation, and they are either flat or wound in an exchanger. Generally, these exchangers cannot accommodate very high pressures.



(a)

(b)

Figure 2.27: Plate-Type Heat Exchangers

2.1.6.9 Plate Fin Type Heat Exchanger

Plate fin type heat exchanger is also another type of heat exchanger. This type is normally used for gas to gas applications.

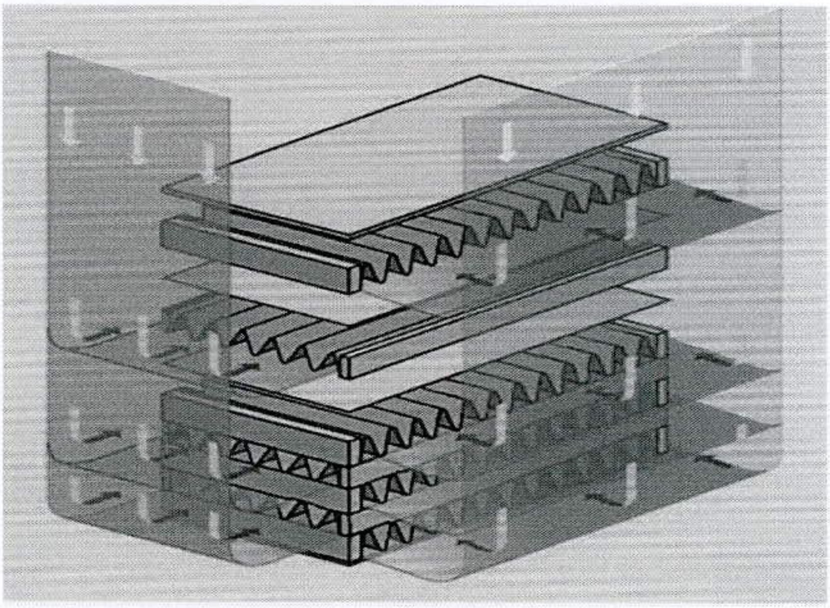


Figure 2.28: Plate fin type heat exchangers

2.1.6.10 Brazed Plate Heat Exchangers

Brazed plate heat exchanger which is an advanced form of compact plate heat exchangers designed for various types of applications in both process and hydroid heating and cooling. You will find the brazed plate heat exchangers in all kinds of refrigeration units, evaporators, condensers and coolers. The construction of the brazed plate heat exchanger is unique in that the plates are vacuum brazed together resulting in no gaskets and reduced leakage. They are compact in design, are lightweight and don't have a high deposit build up due to high turbulent flow. They are ideal if you are looking for a heat exchanger for your swimming pool, spa, radiant floor heating, domestic hot water or snow and ice melting.

2.2 Various Types of Refrigeration System

The major refrigeration systems are as given below:

1. Vapor compression system
2. Vapor absorption system
3. Vapor Adsorption System.

2.2.1 Vapor Compression System

A majority of big refrigeration systems in use nowadays use the Vapor Compression Refrigeration (VCR) cycle. VCR uses mechanical energy (through a compressor) as the driving force for refrigeration. The main components of VCR system are the compressor, evaporator condenser and expansion valve where condenser and evaporator are the high pressure & low pressure components respectively. It is the most widely system for domestic use. Units require stable, continuous electrical current or a fossil fueled mechanical power source to maintain the refrigeration process. The vapor-compression system is not a feasible process due to availability of an electrical grid or continuous fuel source.

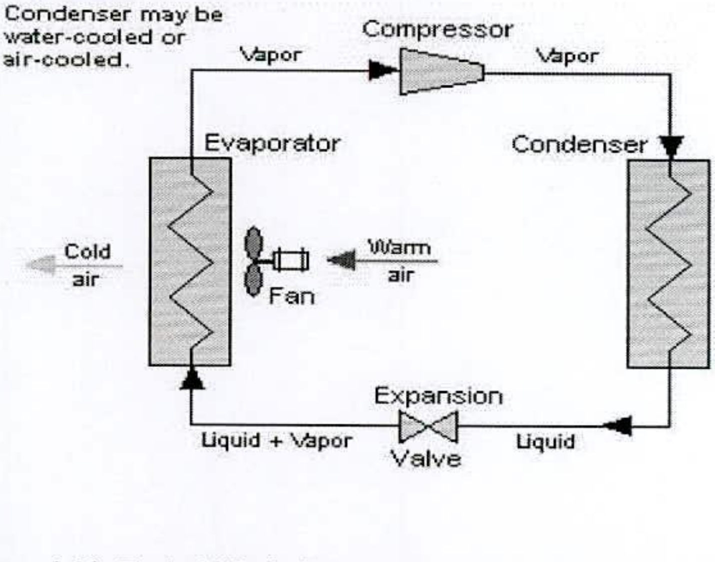


Figure 2.29: Typical Single Stage Vapor Compression Refrigeration

2.2.2 Vapor Absorption System

The absorption refrigeration system differs fundamentally from vapor compression system only in the method of compressing the refrigerant. An absorber, generator and pump in the absorption refrigerating system replace the compressor of a vapor compression system. The vapor absorption system consists of the following types.

a) Continuous vapor absorption system
b) Intermittent vapor absorption system
c) Double intermittent vapor absorption system

2.2.2.1 Continuous Vapor Absorption System

This system requires a continuous heat source and availability of waste heat in rural areas is normally limited. Batteries as a heating source are limited to an electrical charge; photovoltaic cells for recharging can be costly. Continuous absorption systems are also complex in nature and generally more moving parts (valves and float level devices).

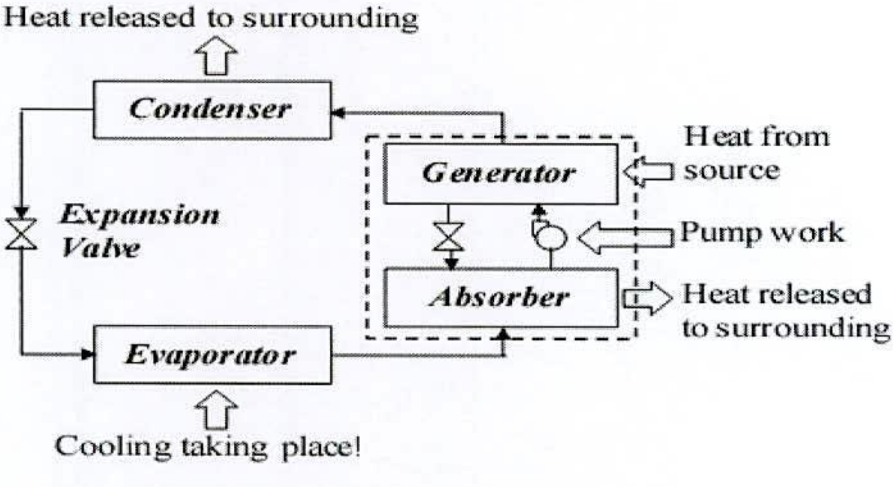


Figure 2.30: Continuous absorption cooling system

2.2.2.2 Intermittent Vapor Absorption System

This system has the widest array of functional designs; some systems are complex in nature and many other designs are very simple. Intermittent [1] Absorption Systems are able to use waste heat and solar energy as the primary heat source. The intermittent process works ideal with the intermittency of the sun. At night the natural ambient cooling environment is a practical means to complete the refrigerant process. This requires a working pair of a refrigerant and an absorbent. High pressure or heat separates the two elements during the generating phase and cooling/refrigeration takes place through the absorption/adsorption of the pair. Ambient cooling is an intermediate phase which takes place to reduce high pressure gas/vapor into a refrigerant working liquid.

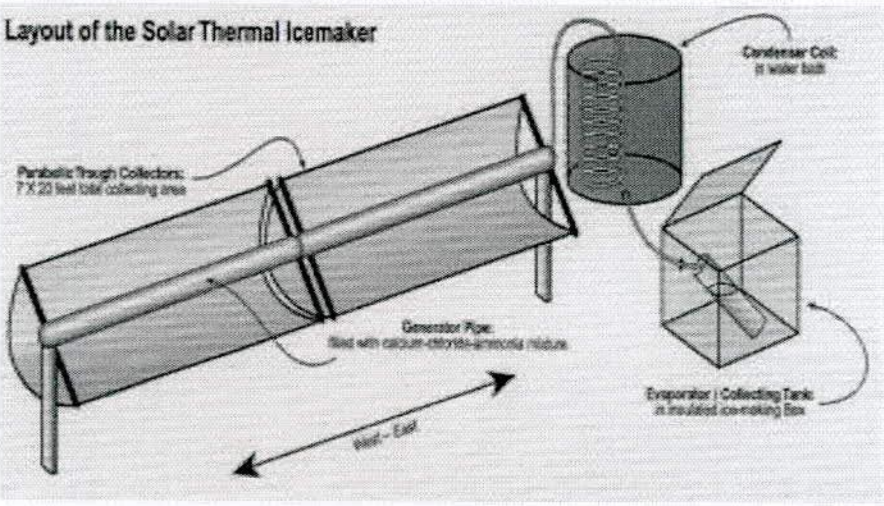


Figure 2.31: Intermittent Vapor Absorption System

There are four examples for this kind of design looked at are:

☐ Water& Ammonia

☐ Lithium Bromide & water

☐ Carbon& Methanol

☐ Calcium Chloride & Ammonia

2.2.2.3 Double Intermittent Vapor Absorption System

The third system is not only the most advanced system, but also the most expensive unit. The ISAAC solar Ice maker is produced by the Energy Concepts Co. and is a double intermittent solar ammonia-water absorption cycle. This system also operates on the Day/Night cycle heating the generator with a parabolic trough solar collector, but instead of using a condenser in a water bath it uses air condenser coils that are capable of condensing the ammonia into a liquid form inside of the evaporator tank. The ISAAC design requires a human operator that is needed to switch valves from the day to night cycle to allow the ammonia to evaporate back to the water. The critical component in this system is the use of a thermo syphon that operates during the night cycle to remove the heat from the generator instead of using just the ambient temperature of the night. The ISAAC has a higher coefficient of performance system yielding around 35 pounds of ice per day in the 37 foot parabolic trough collector. Energy Concepts also offers larger models: a 63 foot collector, and 125 foot collector that are capable of yielding 70 pounds and 150 pounds of ice per day.

2.2.3 Vapor Adsorption System

The application of solar adsorption both in air-conditioning and refrigeration with the improvement of the coefficient of performance is gradually on increase. Solar adsorption heat pump and refrigeration devices are of significance to meet the needs for cooling requirements such as air-conditioning and ice-making and medical or food preservation in remote areas. They are also noiseless, non-corrosive and environmentally friendly. For these reasons the research activities in this sector are still increasing to solve the crucial points that make these systems not yet ready to compete with the well-known vapor compression system. There is an increasing interest in the development and use of adsorption chillers due to their various economic and impressive environmental benefits, enabling solar energy or waste heat to be used for applications such as district networks and cogeneration plants.

Compared to adsorption [37] systems that requires heat sources with temperatures above

100°C (zeolite–water systems, activated carbon–methanol systems) or conventional compressor chillers, a silica gel/water adsorption refrigerator uses waste heat with temperature below 100°C.

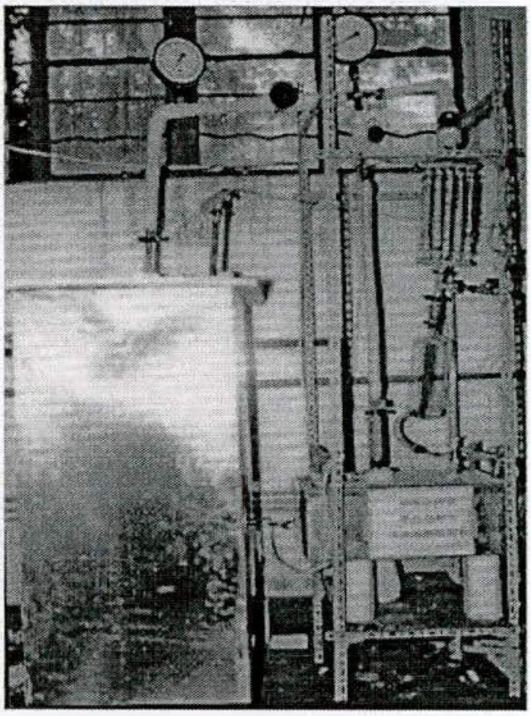


Figure 2.32: The prototype of hybrid water heating and adsorption ice maker

Common used Adsorption Pair-

☐ Zeolite-water
☐ Activated carbon- ammonia
☐ Z13X-water
☐ Silica gel-water
☐ Shanghai YKAC– methanol
☐ AC– methanol

2.3 Experimental Analysis of Different Vapor Absorption System

2.3.1 Experimental Analysis of Katejanekarn [2] Developed Intermittent Vapor Absorption System

The experiments were carried out in the actual climate of NakhonPathom, Thailand, during the summer month of April 2011. The system generated and stored high pressure refrigerant during the daytime and performed cooling during the nighttime. The performance of the system was considered in terms of cooling capacity, refrigeration COP (coefficient of performance), and system COP which were defined respectively as bellow:

$$Q = mc_{p,w}\Delta T_w + mL + mc_{p,i}\Delta T_i$$
$$COP_{ref} = Q / Q_{ref}$$
$$COP_{sys} = Q / Q_{sol}$$

Where,

Q = cooling capacity, MJ
m = mass of water (load), kg
$C_{p,\,w}$ = specific heat of water, kJ/kg-K
ΔT_w = temperature reduction of water, ^{0}C
L = latent heat of fusion of water, kJ/kg-K
$C_{p,\,i}$ = specific heat of ice, kJ/kg-K
ΔT_i = temperature reduction of ice, ^{0}C
COP_{ref}= refrigeration COP,
Q_{ref} = refrigeration heat input, MJ
COP_{sys} = system COP,
Q_{sol} = solar heat input, MJ.

2.3.2 Experimental Analysis of Moreno-Quintanar [3] Developed Intermittent Vapor Absorption System

Five main parameters were used in order to evaluate the experimental system: (i) the amount of the ammonia produced in the generator, (ii) the insulation, (iii) the solar energy received by the CPC, (iv) the cooling capacity and (v) the solar coefficient of performance.

The amount of the ammonia produced in the generator can be obtained as:

$$m_{NH_3} = \rho v_{NH_3}$$

The energy received from the solar radiation is calculated as the sum of the contributions of the product of the irradiation, the time and the aperture area.

$$Q_R = \sum_{i=1}^{n} G_i t A$$

The cooling capacity is the sum of the sensible heat to reduce the water temperature from ambient to 0°C plus the heat of fusion of ice.

$$Q_{EV} = m_{H_2O}\,(h_{fus} + C_p \Delta T)$$

Finally, the solar coefficient of performance is defined as the cooling capacity in the evaporator to the energy received from the solar radiation.

$$COP_S = \frac{Q_{EV}}{Q_R}$$

2.3.3 Experimental Analysis of Tangka [4] Developed Intermittent Vapor Absorption System

2.3.3.1 Determination of the refrigeration load

The refrigeration load (Q_T) was estimated as below equation. This load is the quantity of heat that must be extracted per day from the refrigeration compartment,

$$Q_T = Q_g + Q_p + Q_i + Q_a$$

Where,

The heat gain to the system (Q_g): The heat gain to the system was estimated using power gain as recommended by

$$Q_g = UA\Delta\theta$$

and: $$U = \frac{1}{\left(\frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_0}\right)}$$

therefore,

$$Q_g = \frac{A\Delta\theta}{\left(\frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_0}\right)}$$

The sensible heat (Qp): The sensible heat of the produce was taken as the heat produced during respiration (Qpr) and the heat removed to lower the temperature of the product to safe storage point (Qps)

$$Q_p = Q_{ps} + Q_{pr}$$

Where,

Q_{pr} = Respiration load

Q_{ps} = Product cooling load in here a typical perishable product in the tropics, tomato, was used to estimate the total heat to be removed from the product during cooling.

$$Q_{ps} = MC_3(\theta_3 - \theta_4)$$

Air infiltration load (*Qa*): This was estimated using the equation of

$$Q_a = QV_iE$$

The value of E chosen was 50 and the internal heat Q_i is zero in this case.

2.3.3.2 Determination of the Quantity of the Refrigerant

The system provides saturated ammonia liquid at 26°C/10.34 bar at condenser outlet. This liquid is then expanded as it enters the liquid receiver through a valve. The liquid now enters the evaporator at 1 bar / −33°C to provide the required cooling. The amount of heat absorbed by the evaporation process is,

$$Q_e = h_{g(-33.4^\circ\mathrm{C})} - h_{f(26^\circ\mathrm{C})}$$

The required quantity of ammonia was obtained by

$$M_a = \frac{Q_T}{Q_e}$$

2.3.3.3 Determination of the COP of the System

The following expression was used to calculate the surface area of the collector required

$$S_c = \frac{Q_T}{E_t}$$

Considering the efficiencies of the collector and that of the refrigeration process (0.8 assumed in this case), the equation was modified as follows:

$$S_c = \frac{\frac{Q_T}{\eta_r}}{E_t \cdot \eta_c}$$

The coefficient of performance *COP* was estimated by taking the ratio of the cooling effect to the heat supplied to the system

$$COP = \frac{Q_T}{E_T S_c}$$

CHAPTER III

Design and Construction

3.1 Design of an Intermittent Vapor Absorption System

3.1.1 Basic Working Principle of an Intermittent Absorption System

This system actually is not a true refrigeration system. It can be called as an intermittent refrigeration system because here the mass flow rate is not constant throughout the system. The mass flow rate increases when the temperature of the generator increases than the rate of flow of vapor ammonia also increases. The simplest design of this intermittent system consists of three major parts-a generator for heating the salt-ammonia mixture, a condenser coil, and an evaporator, where distilled ammonia collects during generation. Ammonia flows back and forth between the generator and evaporator [1]. This system operates in a day/night cycle, generating distilled ammonia during the daytime and reabsorbing it at night. Ammonia boils out of the generator as a hot gas at about 13.5 bar pressure. The gas condenses in the condenser coil and drips down into the storage tank where, ideally, 3/4 of the absorbed ammonia collects by the end of the day (at 105^0C, six of the eight ammonia molecules bound to each salt molecule are available). As the generator cools, the night cycle begins. The calcium chloride reabsorbs ammonia gas, pulling it back through the condenser coil as it evaporates out of the tank in the insulated box. The evaporation of the ammonia removes large quantities of heat from the collector tank and the water surrounding it. How much heat a given refrigerant will absorb depends on its "heat of vaporization," — the amount of energy required evaporating a certain amount of that refrigerant.

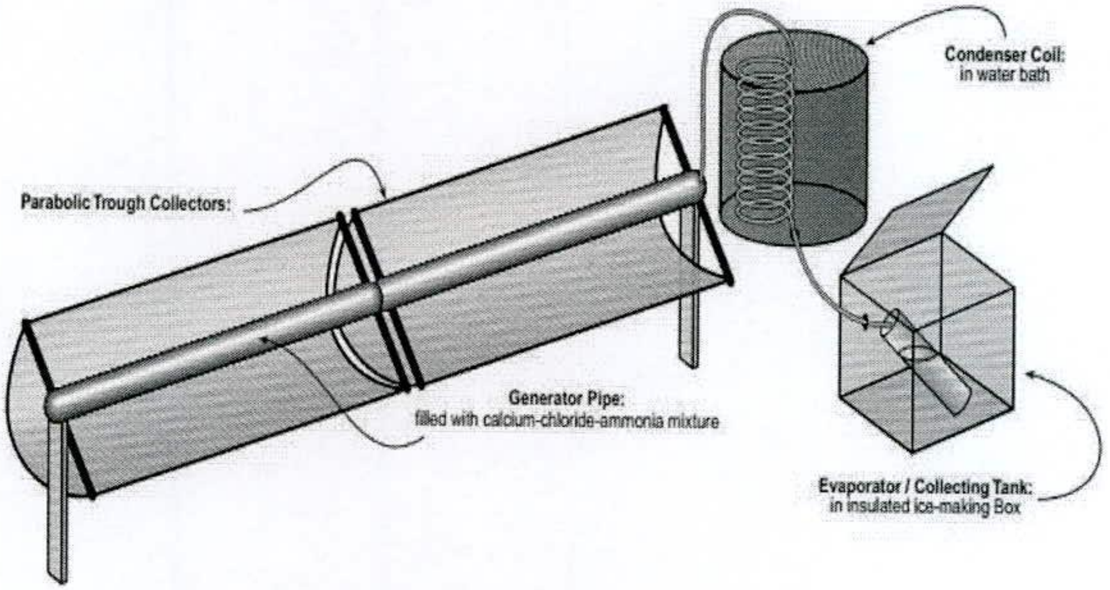


Figure 3.1: Layout of the intermittent vapor absorption system

3.1.2 The Proposed Design of Intermittent Vapor Absorption system

An intermittent vapor absorption system was designed and constructed in this project work. The system consists of a generator/ an absorber, a condenser, a storage tank and an expansion valve and evaporator. The storage tank and the evaporator located below the generator and condenser. The storage tank was located outside the evaporator and the evaporator was an insulated box where smaller diameter coil passes outside of it. A single generator/ Absorber were to be used in the system which acted as a generator in the generation and condensation period and acted as an absorber in evaporation and absorption period. The schematic diagram of the proposed was shown in Figure 3.2. Many absorbent /refrigerant combinations had been designed in the past. These include $CaCl_2/NH_3$, H_2O/NH_3, $SrCl_2/NH_3$, $LiBr/H_2O$, $LiNO_3/NH_3$, and $NaSCN/NH_3$ [8]. However in this project work ammonia and calcium chloride refrigerant-absorbent pair was used where ammonia used as the refrigerant while calcium chloride ($CaCl_2$) acted as the absorbent.

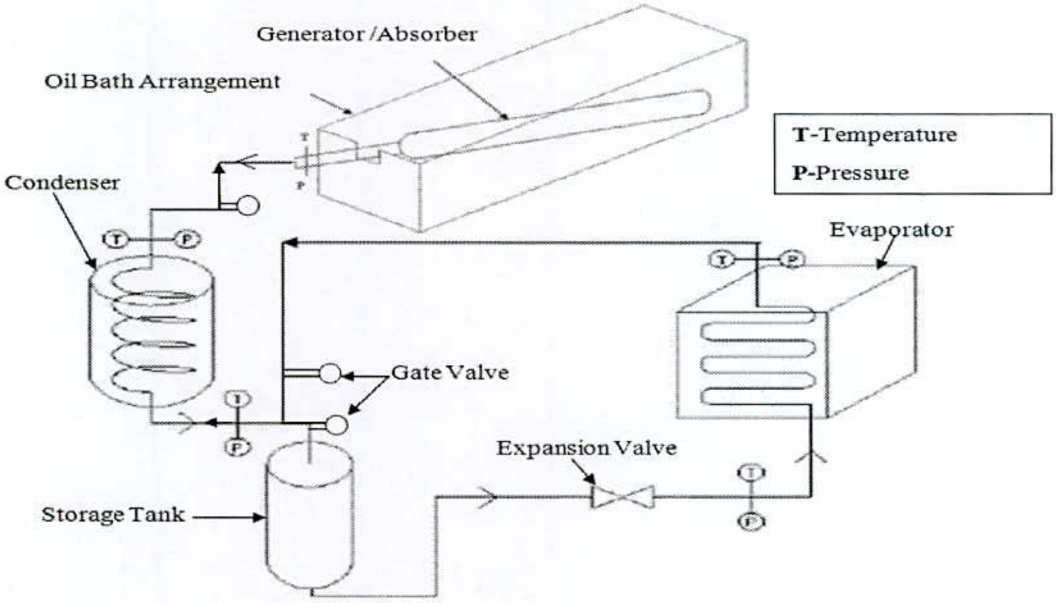


Figure 3.2: Schematic diagram of the proposed intermittent vapor absorption refrigeration system

The proposed system used a generator charged with ammonia and Calcium Chloride. The heat source was an oil bath where oil was heated up by the electric heater. The oil bath was made of a stainless steel sheet. During the generation cycle, at a certain time period the generator heated up the ammonia and Calcium Chloride and the ammonia vapor at the top of the system was condensed by the condenser into a saturated liquid. The liquid flowed by gravity through a gate valve into a storage tank. Then the liquid Ammonia flowed through an expansion valve into the evaporator. This produced an effect which creates a pressure differential. The pressure differential caused the liquid ammonia to change its boiling point from a high pressure flash of temperature to a lower pressure flash off temperature. Again a certain time period when the heat source did not work, then the generator acted as absorber and cooled down the system by using cooling coil or fan and hence caused the ammonia to evaporate at low temperature. The cooling effect of the evaporator causes a pressure drop, which made the ammonia to boil absorbing heat. The vaporized ammonia was reabsorbed into the generator/absorber [4].

3.1.3 Generator Analysis and Mass Calculation

In this project an oil bath was used to provide heat the energy for the generator. Schematic diagram of the oil bath and the generator arrangement is shown in Fig. 3.3. The length of generator is around 2.3368 m (92 inch) and the corresponding oil bath.

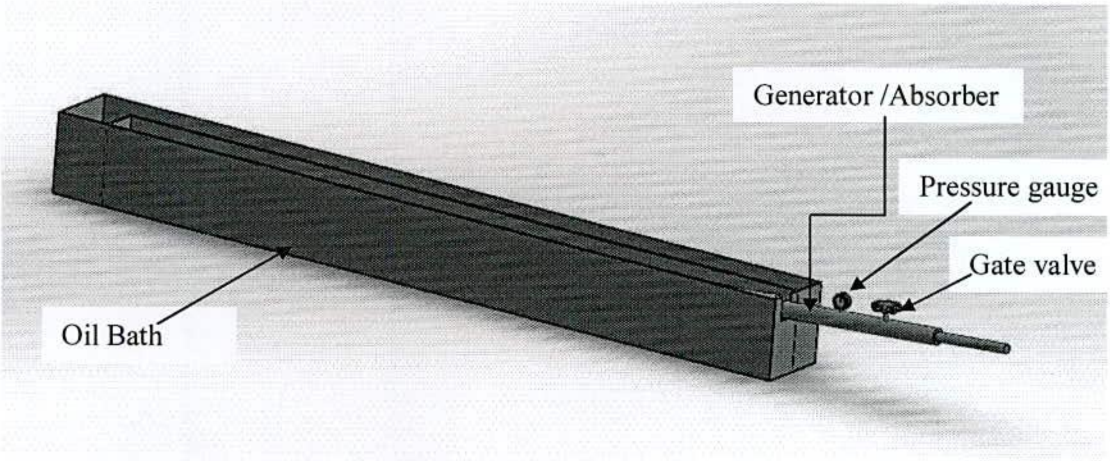


Figure 3.3: Schematic diagram of a generator/ absorber with the oil bath arrangement

The reaction between ammonia and calcium chloride areas follows-

1) $CaCl_2 + 2NH_3 \leftrightarrow CaCl_2.2NH_3$

2) $CaCl_2 + 4NH_3 \leftrightarrow CaCl_2.\ 4NH_3$

3) $CaCl_2 + 8NH_3 \leftrightarrow CaCl_2.\ 8NH_3$ $\Delta H = -78kJ/mol$

Thus, 8 moles of NH_3 may be absorbed per mole of $CaCl_2$, during initial charging, but only 6 moles will be available for refrigeration. It is assumed that no heat and pressure losses in components and lines. In order to analyze the system, mass and energy balance is performed at each component. The following values are used in the calculations:
Density of liquid NH_3: 681.9 kg/m^3

Density of anhydrous $CaCl_2$: 2150.0 kg/m^3

Molar Mass of liquid NH_3: 17.031 g/mol

Molar Mass of anhydrous $CaCl_2$: 110.98 g/mol

The volume of 1 mol ammonia is = $(17\times10^{-3})/681.9 = 2.5\times10^{-5}$ m^3

The net volume of ammonia = $8\times(2.5\times10^{-5}) = 2\times10^{-4}$ m^3

The net volume of $CaCl_2$= $(110.98\times10^{-3})/2150 = 5.162\times10^{-5}$ m^3

Total volume occupies by 1 mol$CaCl_2.8NH_3$= $(2\times10^{-4} + 5.162\times10^{-5}) = 2.5162\times10^{-4}$ m^3

The volume of generator = $\pi r^2 l = (3.1416\times 0.0381^2\times2.3368) = 0.01066$ m^3

Assume that 60% of the volume of generator is used for the purpose of well vaporization.

So the active volume = $0.01066\times40\% = 4.264\times10^{-3}$ m^3

Require $CaCl_2.8NH_3$= $(4.264\times10^{-3})/(2.5162\times10^{-4})$ =16.946 mol

= (16.946×247) g

= 4185.662 g

= 4.185662 kg

So the required NH_3 = $(136\times4.185662)/247 = 2.305$ kg

Required $CaCl_2$= $(110.98\times4.185662)/247$= 1.881 kg.

Assume that 1.8 kg ammonia out of 2.305 kg will be vaporized and stored in the evaporator after passing through the condenser coil. Hence, neglecting the losses and for the better performance from this system work ammonia is used as of 2.5 kg instead of 2.305 kg and calcium chloride is used as of 1.5 kg instead of 1.881 kg. For designing of the generator following designing parameters are used as shown in Table 3.1.

Table 3.1: Design parameters for the generator and mass of ammonia &calcium chloride refrigerant- absorbent pair

Parameters	Type/ Value
Tube diameter	Outside diameter, D_o= 0.0762 m (or 3 inch)
	Inside diameter, D_i= 0.0732 m (or 2.88 inch)
Tube length	2.3368 m (or 92 inch)
Material of the tube used	Stainless steel
Mass of Ammonia Used	2.5 kg
Mass of Calcium Chloride Used	1.5 kg

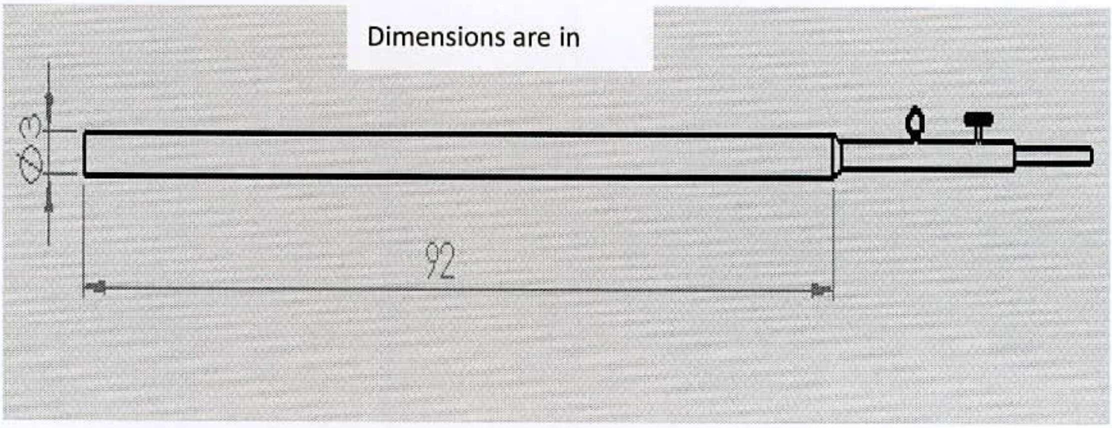


Figure 3.4: Schematic representation of Generator /Absorber

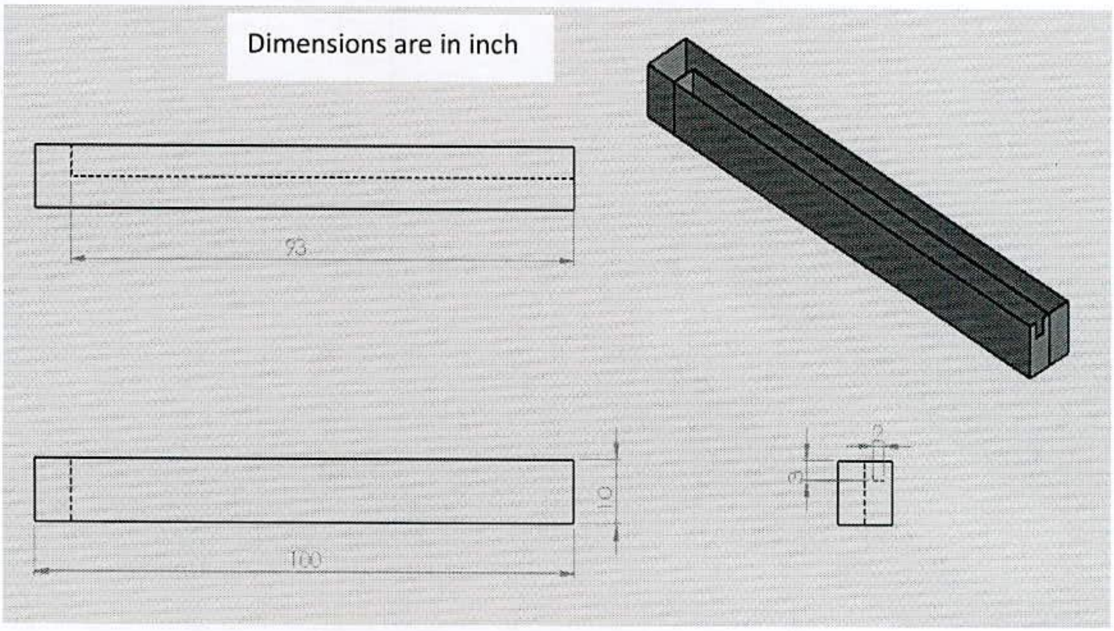


Figure 3.5: Schematic representation of Generator /Absorber tank

3.1.4 Design of the Condenser

At first the design of previously developed condenser has been studied. The design of the heat exchanger, therefore, requires values for the heat and mass transfer coefficient. In single-pass heat exchangers, the temperature difference ΔT between the hot and the cold fluids is not constant but it varies with distance along the heat exchanger. In the heat transfer analysis, it is convenient to establish a mean temperature difference (ΔT_m) between the hot and cold fluids such that the total heat transfer rate Q between the fluids can be determined from the following expression:

$$Q = AU\Delta T_m$$

Where, A(m^2) is the total heat transfer area and U (W/m^2-C) is the average overall heat transfer coefficient, based on that area.

$$\Delta T_m = F\Delta T_{ln} = F\left(\frac{\Delta T_0-\Delta T_L}{ln\,(\Delta T_0/\Delta T_L)}\right)$$

F= Correction factor depending on the type of the heat exchanger used.

The overall heat transfer [39] coefficient (U) based on the outside surface of the tube is defined as

$$U = \frac{1}{(D_0/D_i)(1/h_i)+(D_0/D_i)F_i+(1/2k)D_0 ln(D_0/D_i)+F_0+1/h_0}$$

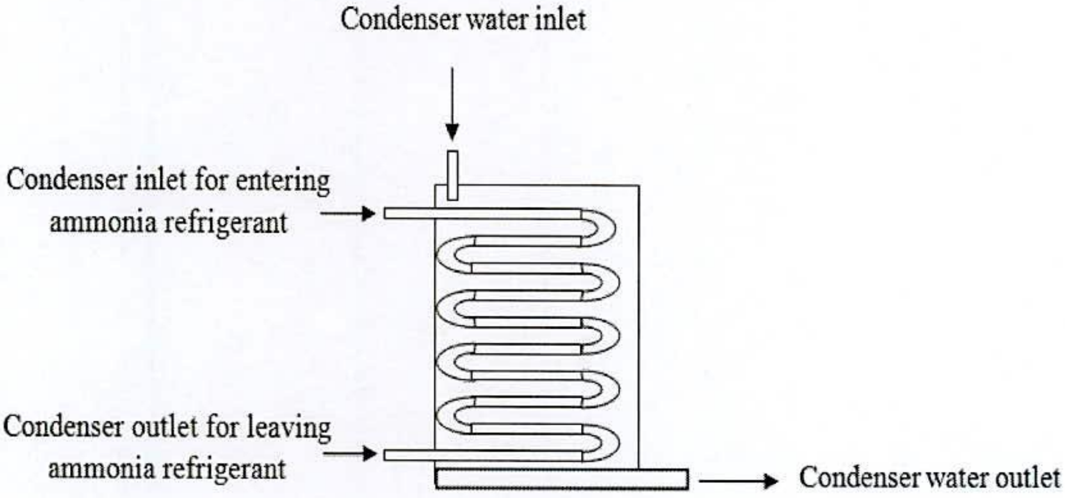


Figure 3.6: Schematic representation of condenser

For the design of the heat exchangers, the condenser water inlet and outlet temperatures are assumed. The condenser water inlet temperature depends exclusively on the available source of water, which may be a cooling tower or a well or a tube well. For designing of the condenser following designing parameters are used as shown in Table 3.2.

Table 3.2: Design parameters for Condenser

Parameters	Value
Condenser input ammonia temperature	90^0C
Condenser output ammonia temperature	32^0C
Condenser input water temperature	25^0C
Condenser output water temperature	35^0C
Mass flow of water	60 liter/hr
Specific heat of water	4200 J/kg-C

The value of the fouling factors ($F_{i,}$, F_o) at the inside and outside surfaces of the tube can be taken as 0.00009 m^2C/W [39] and k for stainless steel is as 17 (W/m-C) which are used to calculate the overall heat transfer coefficient. The heat transfer coefficients, h_i, ho, for the inside and outside flow can be taken as 7000 W/m^2-C. Again the correction factor of the heat exchanger can be chosen as, F=0.96. For mass flow rate of water at 60 liter/hr in the condenser, Heat gain by water from the condenser coil.

$Q= m\ C_p \Delta T_m$

$Q=0.0167\times4200\times(35-25)\,^0C$

Again, $Q = AU\Delta T_m$

$$U = \frac{1}{(D_0/D_i)(1/h_i) + (D_0/D_i)F_i + (1/2k)D_0 ln\,(D_0/D_i) + F_0 + 1/h_0}$$

$$Ro = (D_0/D_i)(1/h_i) + (D_0/D_i)F_i + (1/2k)D_0 ln\,(D_0/D_i) + F_0 + 1/h_0$$

$=(0.0127/0.0107)\times(1/7000)+(0.0127/0.0107)\times0.0009+(0.0127/(2\times17))\times\ln(0.0127/0.0107)+0.0009+(1/7000)$

$=0.0001695+0.001068+0.000064+0.0009+0.0001428$

=0.0023445

Again, U=1/R_0

U=426.53 W/$m^2.^0$C

$$\Delta T_{ln} = \left(\frac{(90-25)-(35-32)}{ln\ (65/3)}\right)= 20.15$$

$$\Delta T_m = F\Delta T_{ln} = F\left(\frac{\Delta T_0 - \Delta T_L}{ln\ (\Delta T_0/\Delta T_L)}\right) = 0.96 \times 20.15 = 19.34$$

Now, 0.0167× 4200 × (35-25) = A × 426.53× 19.34

Or, A=0.085 m^2

Now, A=3.1416×0.0127 ×L

L=2.13 m

Hence it will be used more length for more heat transfer of length 5.5 m. The length of our condenser is 18 feet which is higher than desired length.

Table 3.3: Obtained heat exchanger size for Condenser Coil

Parameters	Type/ Value
Tube diameter	Outside diameter, D_o= 12.7 mm (or 0.5 inch)
	Inside diameter, D_i= 10.7 mm (or 0.421 inch)
Tube length	Calculated length = 2.13 m
	Used length= 5.5 m
Material of the tube used	Stainless steel

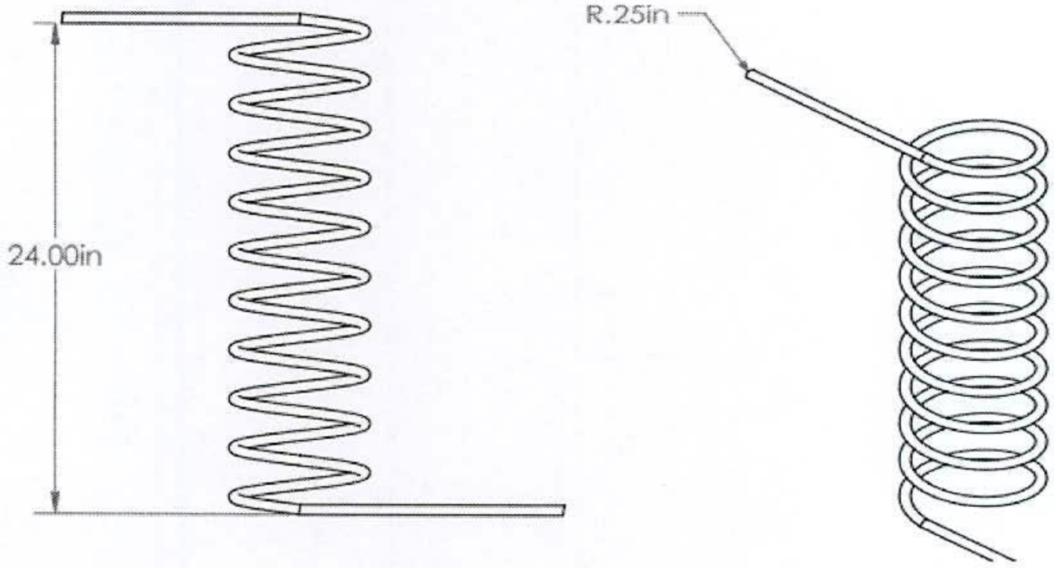


Figure 3.7: Schematic Representation of Condenser Coil and its parameter

3.1.5 Design of the Evaporator

The design of a refrigerator means designing the length, size, type and material of heat exchanger such as an evaporator. For this design, at first the design of previously developed of evaporator has been studied. The design of the heat exchanger, therefore, requires values for the heat and mass transfer coefficient. For the design of the heat exchangers, the evaporator entering and leaving temperatures of refrigerant are assumed with respect to the pressure of evaporator side. The evaporator temperature is also to be assumed as required on the basis of room temperature. For the designing such an evaporator it is taken into the consideration during absorption period and the cooling capacity or refrigerating effect of that evaporator. For this design the water is to be feed into evaporator compartment to measure the cooling capacity of evaporator. The entering temperature of the water depends exclusively on the available source of water, which may be a cooling tower or a well or a tube well or others. For designing of the evaporator following designing parameters are used as shown in Table 3.4

Table 3.4: Design parameters of evaporator

Parameters	Value
Room temperature	28^0C
Evaporator temperature	10^0C
Evaporator entering temperature	7^0C
Evaporator leaving temperature	20^0C
Pressure at evaporator	5.5 bar
Absorption period	2.50 hrs
Mass of water in the evaporator	3 kg
Cooling capacity	34 W

Now the overall thermal resistance can be obtained from the following equation-where convective heat transfer coefficient of ammonia is 7000 W/m^2-C and thermal conductivity of stainless steel pipe is 17 W/m-C. Again, from the ammonia properties table or pressure enthalpy (p-h) diagrams of R717 the bellow properties are taken corresponding to pressure and temperature.

For 7^0C and 5.5 bar saturated liquid enthalpy, h_3=232.503 kJ/kg

For 20^0C and 5.5 bar saturated vapor enthalpy, h_4 = 1488.30 kJ/kg

From the equation $Q_e= m_{ref}\,(h_4-h_3)$

$$\text{Or,}\quad m_{ref}=Q_e/\,(h_4-h_3)$$

$$=34/\,(1488.30\times10^3-232.503\times10^3)$$

$$=2.707\times 10^{-5}\text{ kg/s}$$

$R=1/Ah_i+t/KA+/Ah_o$

$=1/A\times\,(1/7000+0.002/17+1/15)$

$=0.0669/A$

Or,

$Q_e = [(T_0 - T_i) \times A] / R$

$= [(28-10) \times A] / R$

$= (18 \times A) / 0.0669$

Or, $A = (34 \times 0.0669)/18$

$= 0.1264\ m^2$

Here the used diameter of evaporator is 3/8 inch (or 0.009525 m)

So the heat exchanger surface, $A = \pi D_e L$

$A = 3.1416 \times 0.009525 \times L$

Or, $0.1264 = 3.1416 \times 0.009525 \times L$

Or, $L = 0.1264 / (3.1416 \times 0.009525) = 4.22$ m

Here it is used the evaporator tube of length is 6 m for safety and more heat transfer.

Table 3.5: Obtained Heat exchanger size for evaporator

Parameters	Type/ Value
Tube diameter	Outside diameter, D_o= 3/8 inch (or 0.009525 m)
	Inside diameter, D_i= 0.30 inch (or 0.007525 m)
Tube length	Calculated length = 4.22 m
	Used length= 6 m
Material of the tube used	Stainless steel

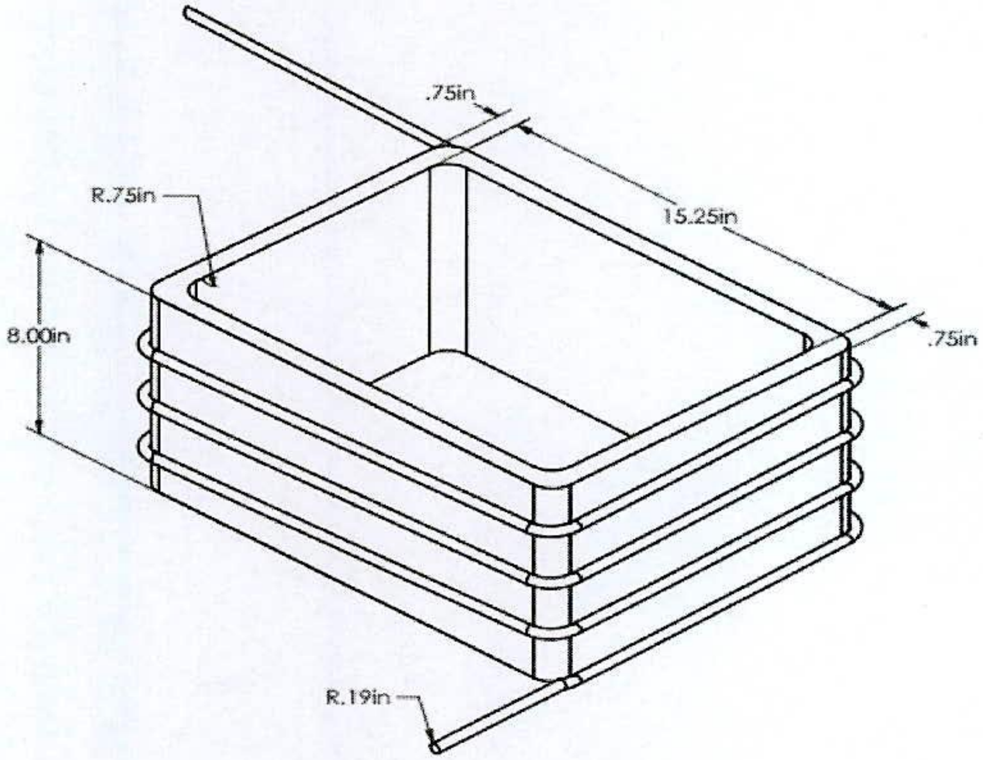


Figure 3.8: Schematic representation of Evaporator

3.2 Construction of an Intermittent Ammonia Absorption System

3.2.1 Construction of Generator/ Absorber

Initially the generator is a storage tank which contains ammonia and calcium chloride mixture. It is made of stainless steel. In desorption mode of the system the generator is heated up by the oil bath arrangement and during the absorption mode it acts an absorber in which absorbs ammonia from the system. The temperatures of the generator are depends upon the oil bath arrangement and however, in this experiment the temperature of oil bath arrangement varies from 90^0C to 110^0C.

Figure 3.9: Generator / Absorber located on an oil bath

3.2.2 Construction of Condenser

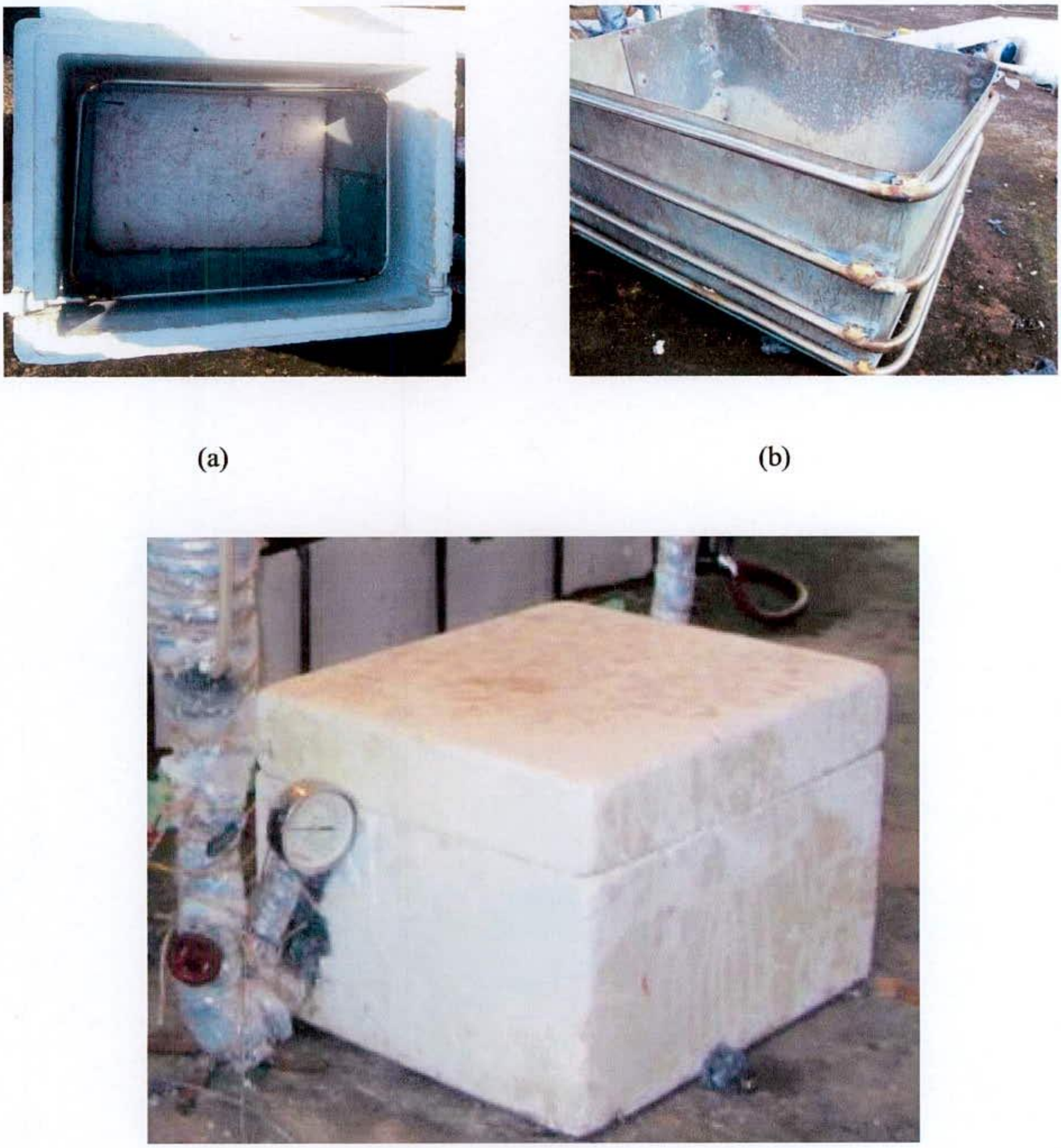
Condenser is a vital part of vapor absorption refrigeration system. In this project, condenser is made of stainless steel. It is considered as a shell & coil type heat exchanger and the convection process is free convection. The length is designed based upon the temperature drop from 90^0C to 32^0Cwhich required 2.13 m and integrate other losses actual length is given

5.5 m. The length is much greater than the calculated length. The condenser coil submerged into water. Here water takes the heat from the refrigerant and the liquid refrigerant stored into the storage tank due to gravitational flow.

Figure 3.10: Inside view of Condenser coil container

3.2.3 Construction of Evaporator

The evaporator is the component of a refrigeration system in which heat is removed from air, water or any other body required to be cooled by the evaporating refrigerant. Evaporator is placed on a thermocole box which insulated the evaporator from the outside medium. The evaporator raises heat from the 7^0C to 20^0C which required 4.22 m and integrate other losses actual length is given 6 m.



(a) (b)

(c)

Figure 3.11: Various view of Evaporator box; (a) Evaporator box in insulation box; (b) Evaporative coil; (c) Total Evaporation system

3.2.4 Construction of Ammonia Vapor Absorption Refrigerant System

While constructing the system first generator was constructed. Based upon the capacity of generator than condenser and evaporator is designed. For constructing the whole system plumbing stainless steel tube is used because it is more sustainable on high temperature and pressure. To carry out various measurements and performance tests pressure gauge, thermocouple are installed on various locations of the system.

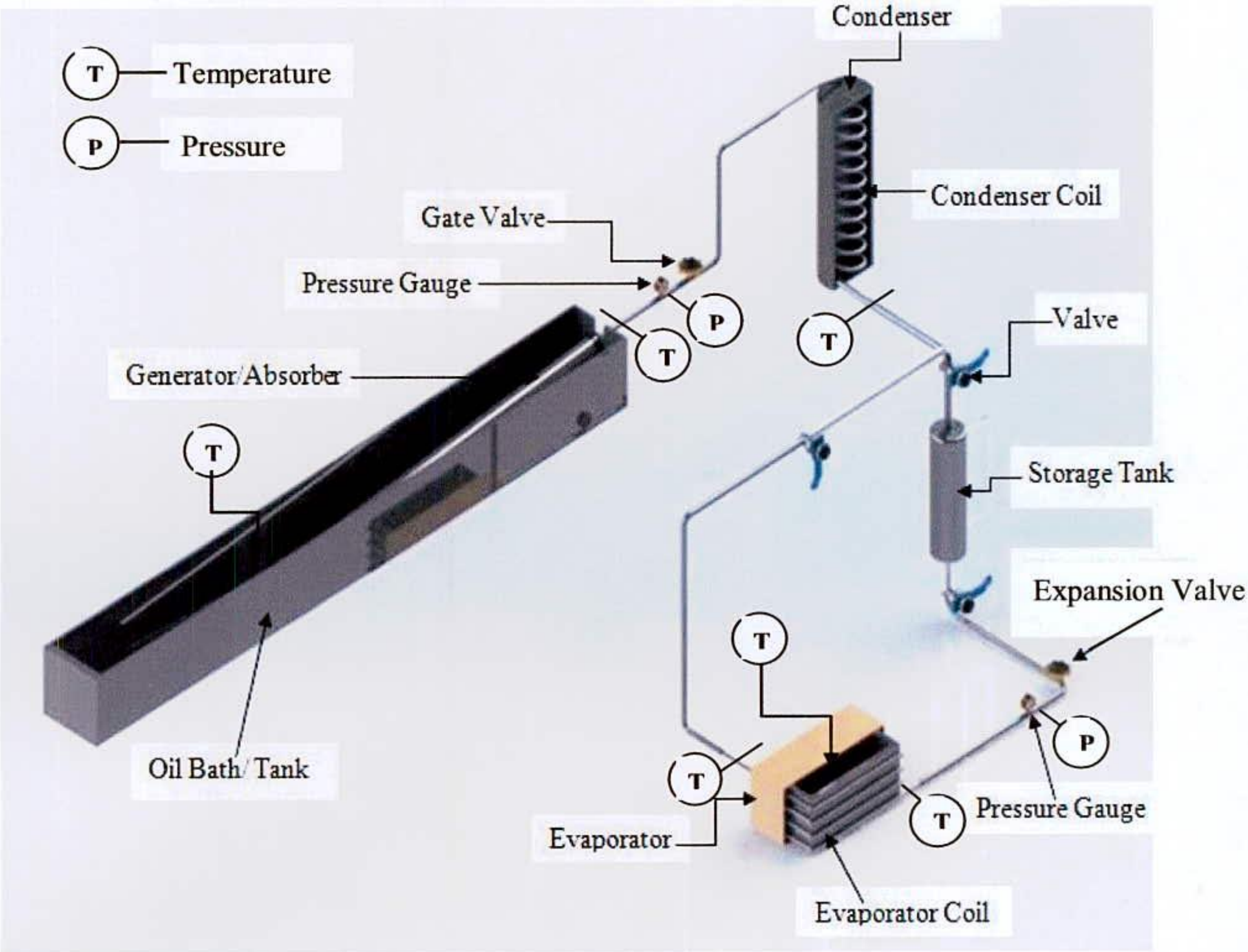
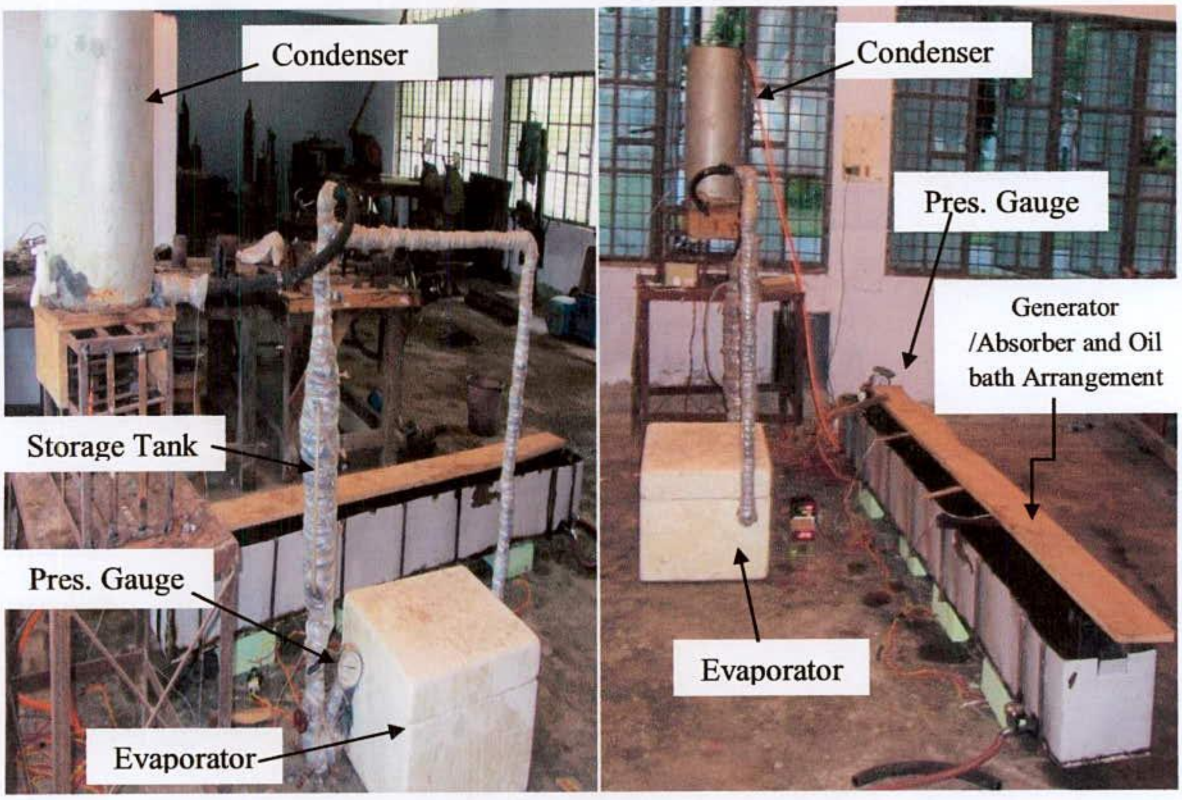


Figure 3.12: Layout of the Intermittent Vapor Absorption System



(a) (b)

Figure 3.13: Experimental Setup of the Intermittent Ammonia Absorption Refrigeration System

CHAPTER IV

Results and Discussion

4.1 Experimental Investigation

The results obtained from the experimental investigation of ammonia absorption refrigeration system were presented in this chapter. This system required some extra plumbing and valves to maintain working cycle during desorption mode and absorption mode of the system. The first justification was aimed at using the space as a refrigerated space during the whole cycle, where in the simplest design that got refrigerating effect only in absorption mode of the cycle. In the earlier, the condensed ammonia entering the storage tank during the desorption mode was cool, but not as cool as the temperatures one might hope to maintain in an icebox. The advantages of having a separate storage tank and evaporator is that are able to concentrate on specializing the evaporator to its purpose, that of absorbing heat from the water. By making it out of a long, flat coil of 1/4" stainless steel pipe, soldered to a sheet metal bucket of the same shape, heat can flow more easily into the evaporating ammonia within the system.

Refer to figure 4.1 during desorption mode, the heat source heated up the generator and the generator drives of the vapor form refrigerant around the system through a condenser to a storage tank when the valve 1 was only open and valve 2-3 remain closed. Before the absorption mode of the cycle fully starts the valve 3 may slightly open which will give some refrigerant effect to the evaporator. On the other hand, after starting the absorption mode the valves 2 and 3 remain open and valve 1 remains closed. The liquid refrigerant from the storage tank flowed through an expansion valve into the evaporator and absorbed heat from the surrounding water and performed cooling. Then the evaporated refrigerant vapor flowed back into the absorber where the concentrated absorbent awaits and both working substances are mixed.

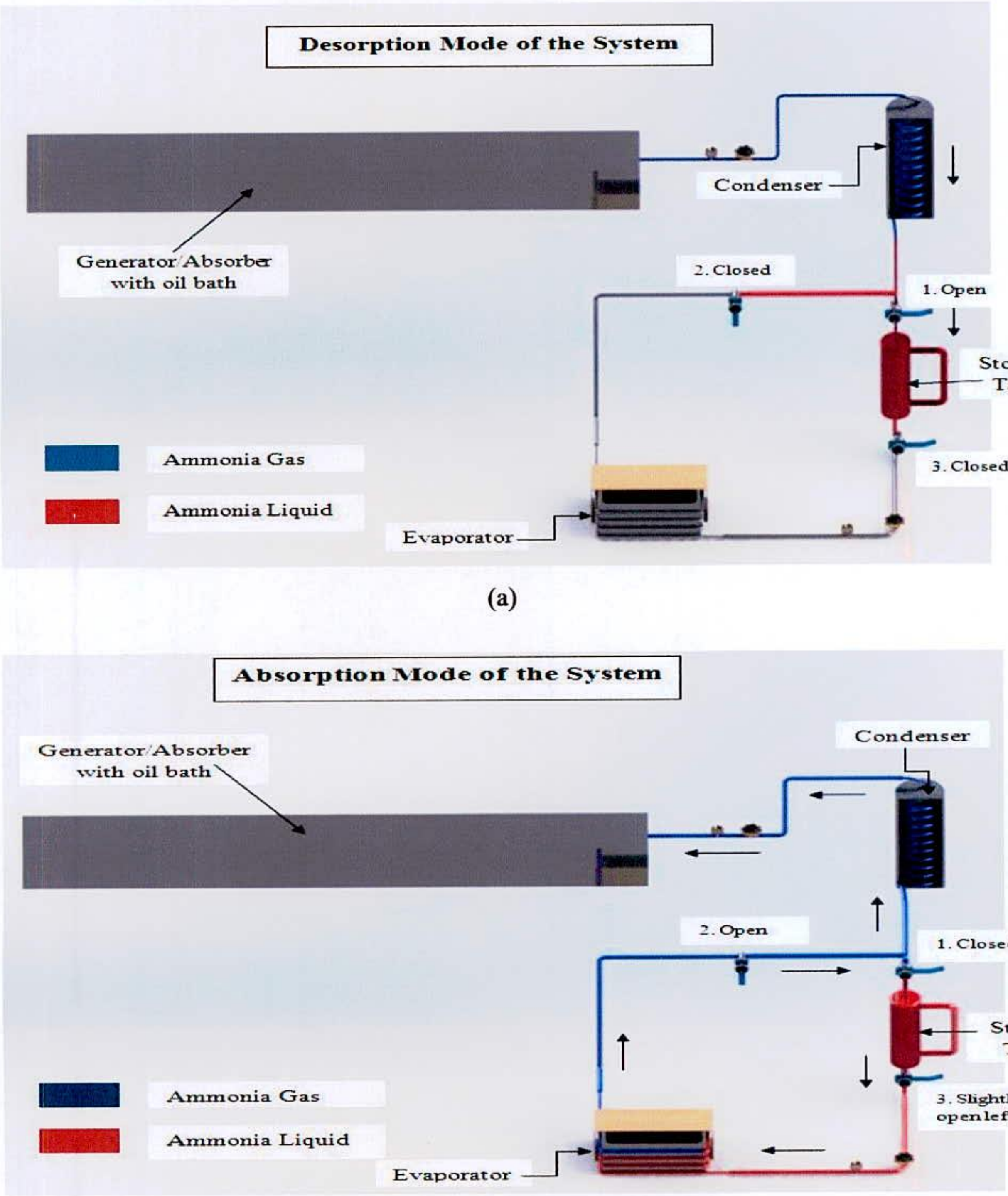


Figure 4.1: Simple layout of the Intermittent Refrigeration System (a) in Desorption Mode and (b) in Absorption Mode

4.1.1 Experimental Data

The data obtained from the experimental investigation of ammonia absorption refrigeration system were presented here. The mass flow rate of this system was to be constant due to investigate the intermittent system and during experiment the temperatures were measured by setting thermocouples at the different locations of the system. The pressure increment and decrement of the generator and the evaporator were measured by setting two pressure gauges. It was said that generator was the high pressure side element in where pressure increases with the heat addition but in low pressure side in evaporator, found decreased of pressure. The experiment was carried out about 2.5 hour both of desorption and absorption modes separately.

However, the data for the experimental investigation of this system were presented below.

Table 4.1: Experimental data for the heating of generator during the desorption mode (3:30 PM to 6:00 PM)

No. of obs.	Pressure of the generator (bar)	Temp. of Heat Source (Oil Bath) (^{0}C)	Temp. of Generator (Avg.) (^{0}C)	Outlet temp. of Condenser (Avg.) (^{0}C)	Room temp. (Avg.) (^{0}C)
1	8.50	90	76.70	25.30	30.64
2	9.50	100	83.70	26.50	31.04
3	10.00	105	86.50	27.00	30.82
4	10.50	110	91.40	28.30	30.80

Table 4.2: Experimental data during the absorption mode (9:00 PM to 11:30 PM)

No. of obs.	Room temp. (Avg.) (^{0}C)	Pres. of Evap. Side (bar)	Inlet temp. of Evaporator (^{0}C)	Temp. of Evaporator (^{0}C)		Outlet temp. of Evaporator (^{0}C)	
				Initial (water)	Final (water)	Initial	Final
1	29	6	13.5	27	15	20.5	25
2	29	6	12.5	27	14	21	24.5
3	29	5	11	27	12.5	20.5	25
4	29	5	10.5	27	11.5	20	24.5

4.1.2 Sample Calculation for the Presented Data

Considering for the no. of observation-1

During the 2.5 hours desorption mode, the heat gain by the generator $Q_g = Q_t - Q_{loss}$

$Q_t = 30 \times 1.75 \times 6 \times (150/20)$, where, $m_f = 30$ kg, $C_{p,f} = 1.75$ kJ/ kg-C and $\Delta T_f = 6^0$C per 20 min (45^0C), then obtained $Q_t = 2362.5$ kJ.

Again the heat loss on oil bath, $Q_{loss} = 30 \times 1.75 \times 25$, where $\Delta T_{f,} = 25^0$C and found $Q_{loss} =$ 1312.5 kJ. Then calculated the heat gain by the generator $Q_g = 2362.5 - 1312.5 = 1050$ kJ

Again at the 2.5 hours absorption mode, the performance of the system was considered as The refrigerating effect, $Q_e = Q_w + Q_s$

$Q_w = m\, C_{p,w} \Delta T_w$

or, $Q_w = 3 \times 4.2 \times (27-15) = 151.2$ kJ, where m = mass of water (load) = 3 kg, $C_{p,w}$ = specific heat of water = 4.2 kJ/ kg-C and ΔT_w = temperature reduction of water = (27-15) ^{0}C.

And $Q_s = m\, C_{p,w}\, \Delta T_s = 3 \times 4.2 \times 4 = 50.4$kJ, where ΔT_S= temperature reduction of water due to system heat gained = 4 ^{0}C

Hence finally obtained refrigerating effect, $Q_e = 201.6$ kJ

Again, the coefficients of performance of the system,

COP = $Q_e / Q_g = 201.6/1050 = 0.192$

Again, the required of ammonia was obtained from the below equation

$Q_e = m_a Ls$

Where, m_a= mass of ammonia, kg and

$Ls = h_g - h_f$

Therefore, corresponding the saturation table of R 717 (Ammonia) for evaporator pressure 6 bar, the values of h_f and h_g were found as 242.86 kJ/ kg and 1470.88 kJ/ kg respectively.

Then found, Ls= 1228.02 kJ/ kg and finally the mass of ammonia was obtained,

$m_a = Q_e/Ls$

or, $m_a = 201.6/ 1228.02$

or, $m_a = 0.164$ kg

4.2 Results and Discussion

In order to experimentally evaluate the COP of the system the ammonia and calcium chloride mixture at the generator/absorber, the evaluating parameters such as temperatures, pressures and heat input by the heat source everything were recorded time to time. A summary of the operating and testing parameters was presented in table 4.3 bellow.

Table 4.3: Operating and testing parameters of the system during the experiments

No. of obs.	Desorption Mode					Absorption Mode				COP
	Temp. of heat source (oil bath) (^{0}C)	Temp. of Gen. (^{0}C)	Outlet temp. of Cond. (^{0}C)	Heat loss (kJ)	Gen. heat gain (kJ)	Temp. of evap. (^{0}C)	Product (water) load (kJ)	Heat loss (kJ)	Ref. Effect (kJ)	
1	90	76.7	25.3	1312.5	1050.0	15	151.2	50.4	201.6	0.192
2	100	83.7	26.5	1312.5	1443.7	14	163.8	50.4	214.2	0.148
3	105	86.5	27.0	1312.5	1640.6	12.5	182.7	50.4	233.1	0.142
4	110	91.4	28.3	1312.5	1837.5	11.5	195.3	50.4	245.7	0.134
Avg.	101.25	84.58	26.78		1492.97	13.25			223.65	0.154

Under the project, an intermittent ammonia absorption refrigeration system was constructed and investigated. In desorption mode an oil bath arrangement heat source provided heat to the generator and the generator drives of the vapor form refrigerant around the system through a condenser to evaporator. When the heat source was removed in absorption mode and the absorbent cooled down, the vapor started moving back into the generator which acting as absorber. However, the table 4.3 shows that the generator pressure was varied from 8.5 bar to 10.5 bar during desorption mode when heat supplied from the oil bath arrangement. The average heat gained by the generator was about 1492.97 kJ during 2.5 hours desorption and also keep constant heat loss by the oil bath arrangement. It could also be seen that the average refrigeration time varies with time and during the absorption mode, the lowest evaporator i.e, refrigeration compartment temperature was found as 11.5^{0}C whereas average of 13.25^{0}C after a 2.5 hours performed. The coefficient of performance (COP) of the system was obtained at the average of 0.154 with a maximum value of 0.192.This coefficient of performance estimated was the higher coefficients of performance found by Vanek [1], Katejanekarn [2] and Moreno-Quintanar [3] developed intermittent solar absorption system and also lower than the COP found by Tangka [4] developed intermittent solar absorption system.

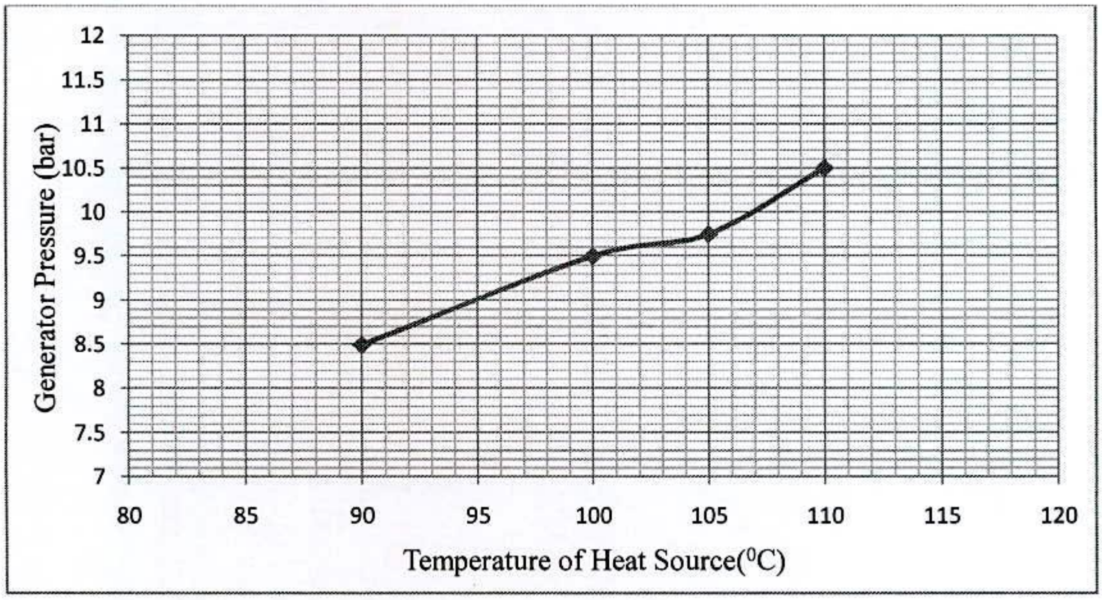


Figure 4.2: Variation of generator pressure with the temperature of heat source

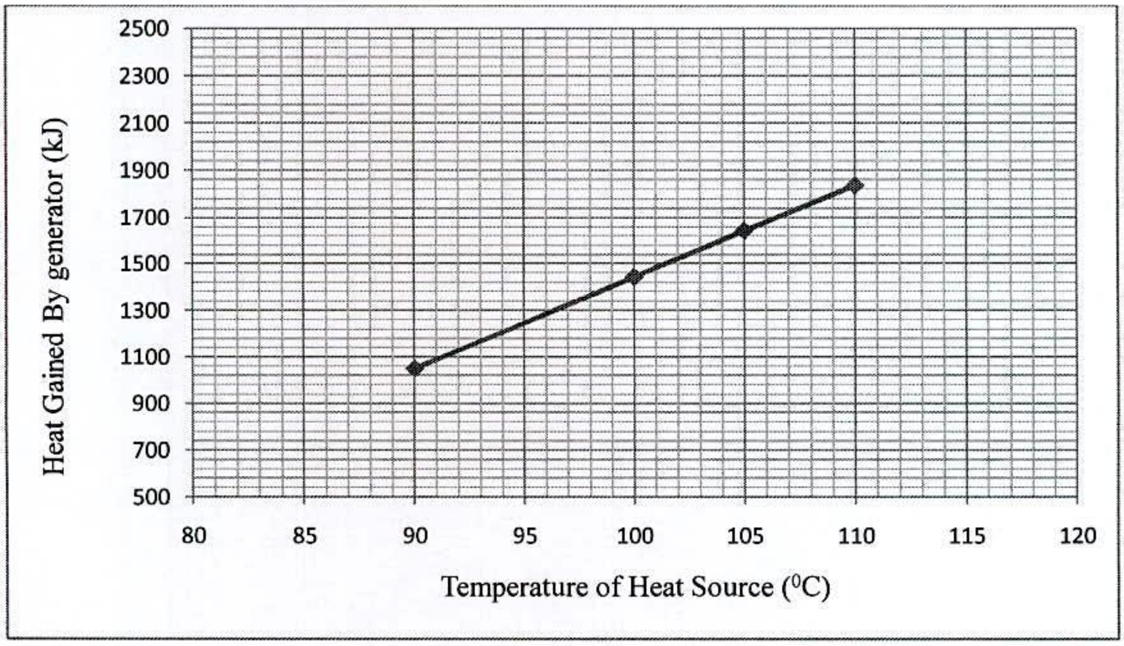


Figure 4.3: Variation of heat gained by the generator with the temperature of heat source

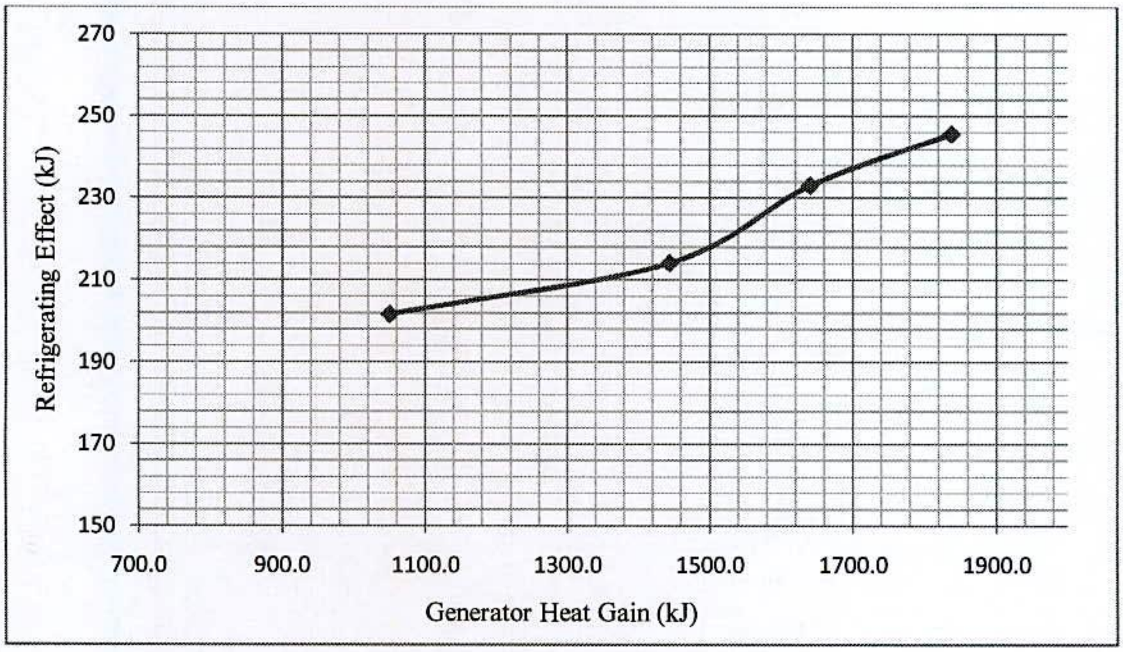


Figure 4.4: Variation of refrigerating effect with the generator heat gain

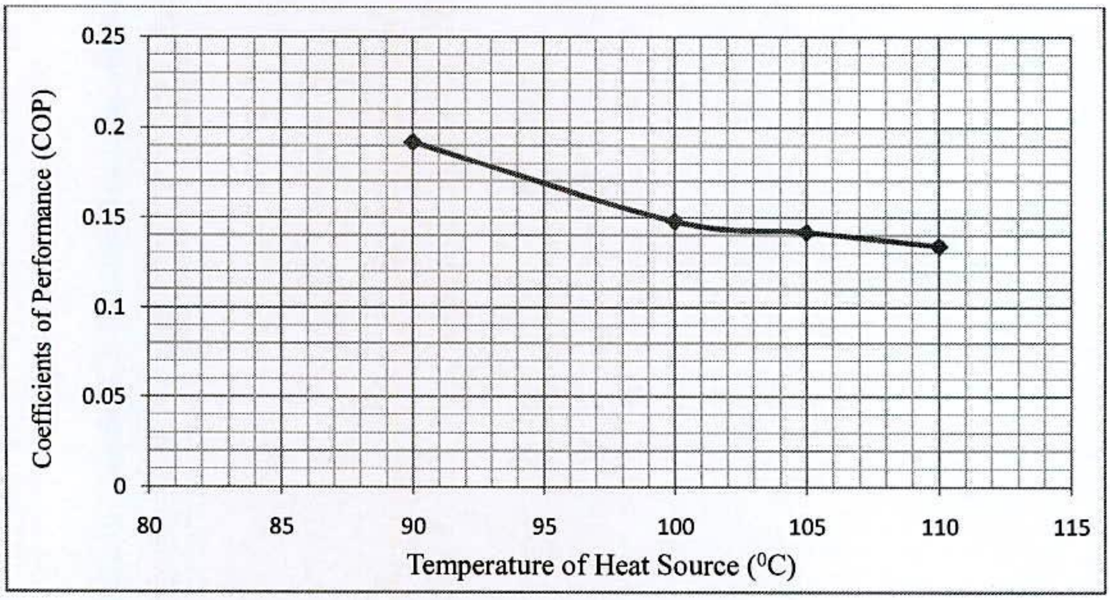
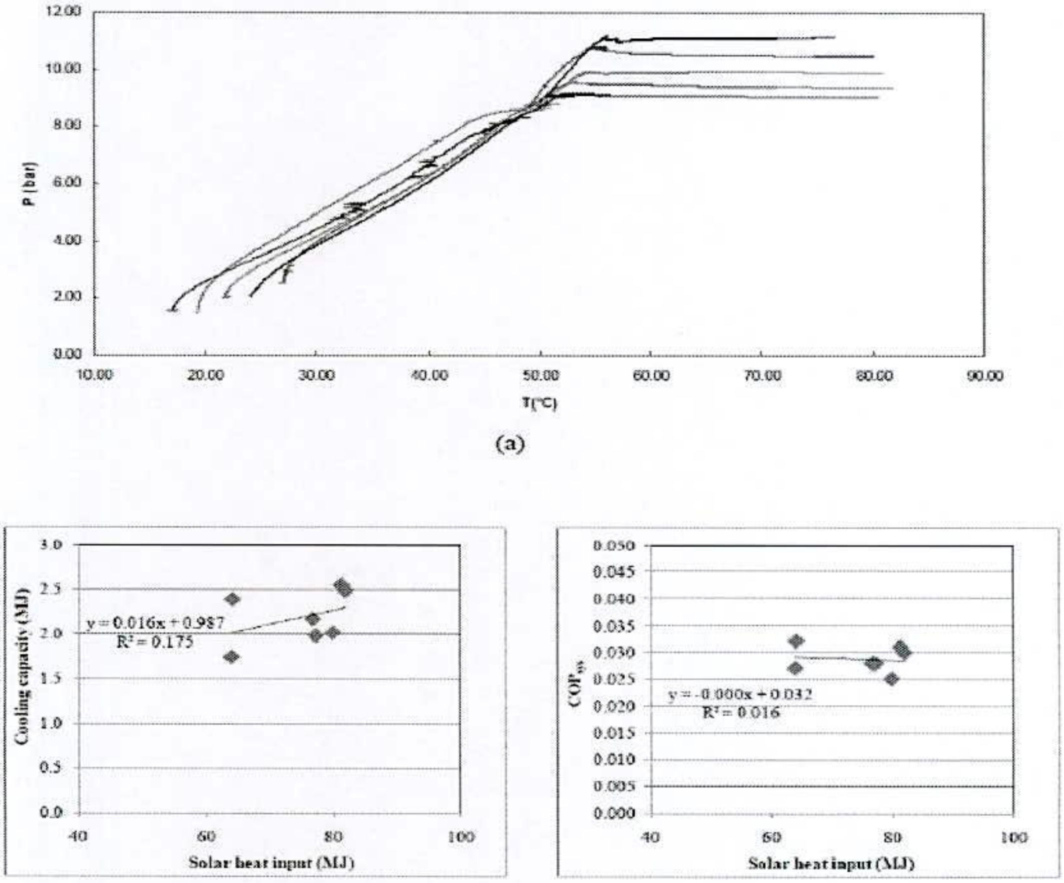


Figure 4.5: Variation of COP of the system with the temperature of heat source

From figure 4.2 to figure 4.3, it could be seen that the generator pressure varies with the heat input from the heat source of oil bath arrangement and also the generator heat gain was fairly related to the heat input by considering the heat losses from the heat source of the oil bath. From figure 4.4 and figure 4.5, it could be drawn that the refrigerating effect was affected with the generator heat gained. The refrigerating effect increased with the heat gained by the generator. But the coefficients of performance (COP) also decreased with the increase of temperature of heat source of oil bath.



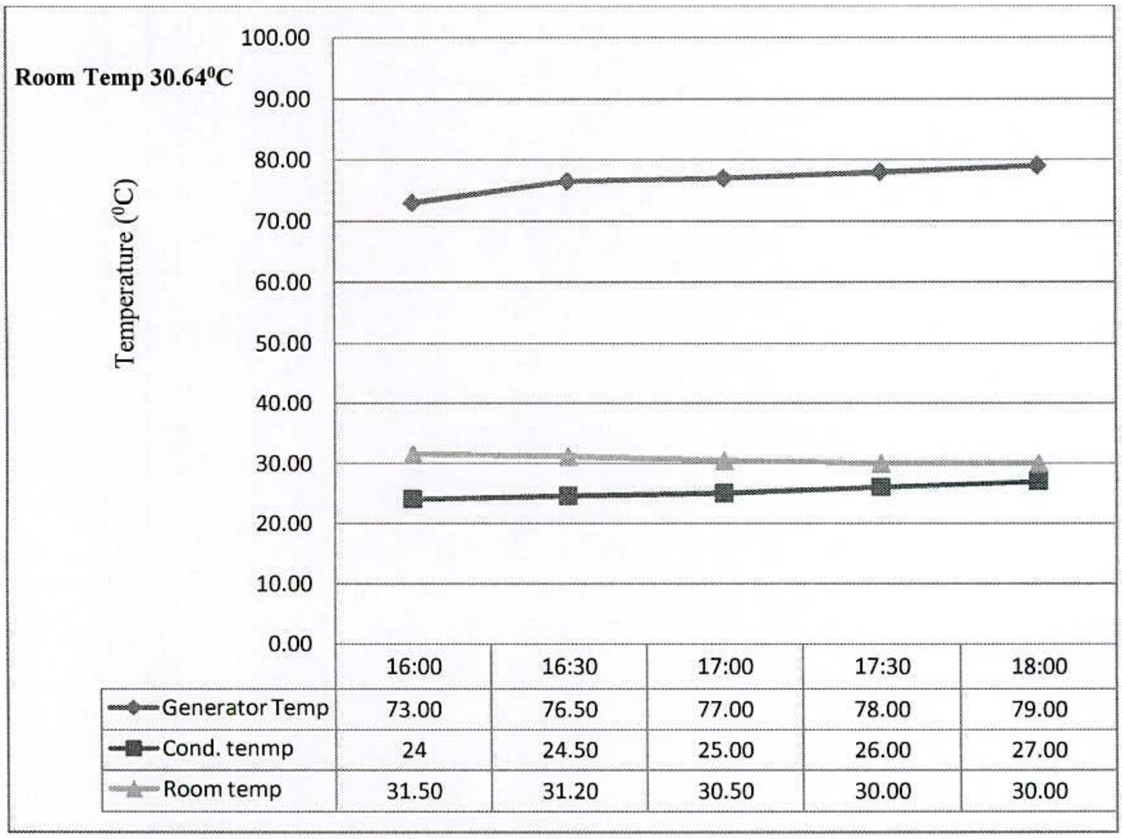
(a)

(b)

(c)

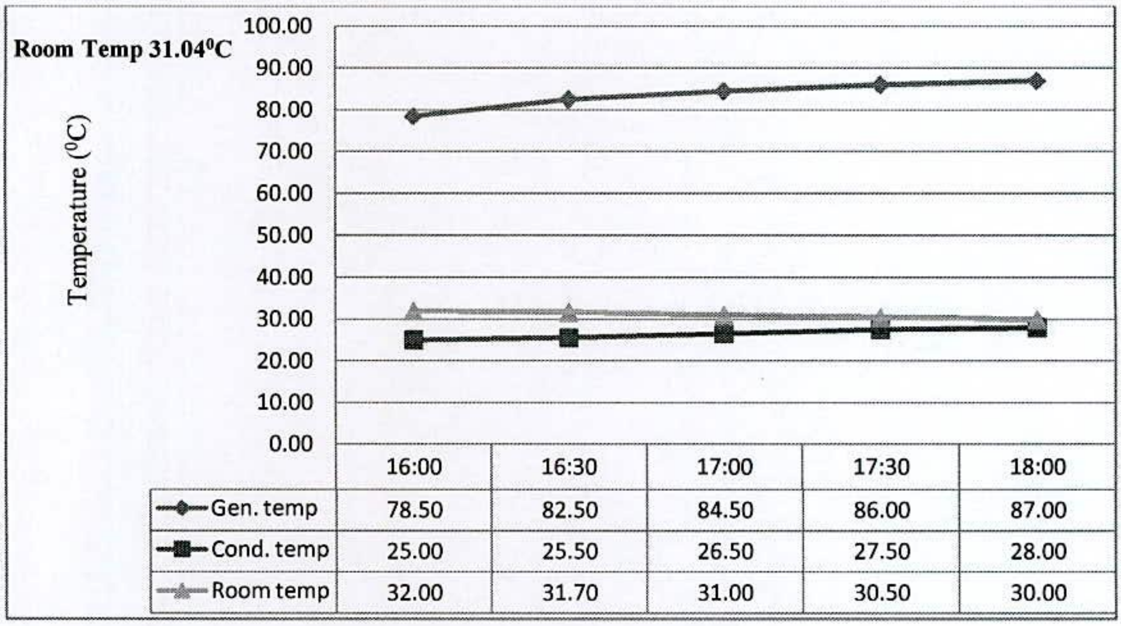
Figure 4.6: Carlos Rivera [38] examined the (a) generation temperature for different probes and Katejanekarn [2] examined the (b) Cooling capacity vs. solar heat input curve & (c) System COP vs. solar heat input

Again from figure 4.2 to figure 4.5 and figure 4.6, it could be drawn a comparison that the generator pressure increased with heat added and almost showed same phenomena with refereed of Carlos Rivera [38] developed the generating the pressure of generator. It also be compared that the cooling capacity or refrigerating effect increased with heat input and the system COP decreased with heat input which was almost the same nature of Katejanekarn [2]. The generation pressure is between 8.5 to 10.5 bar, and then the system generated 0.164 kg (or164 ml) of ammonia liquid in the storage tank during around 2.5 hours of desorption which is referred with Carlos Rivera [38] of 100 ml of ammonia liquid around 2 hours. During evaporation stage the ammonia pressure descends until 5 to 6 bar with temperature from 10.5^0C to 13.5^0C and the obtained 164 ml ammonia cools 300 ml of water from 27^0C to 11.5^0C.



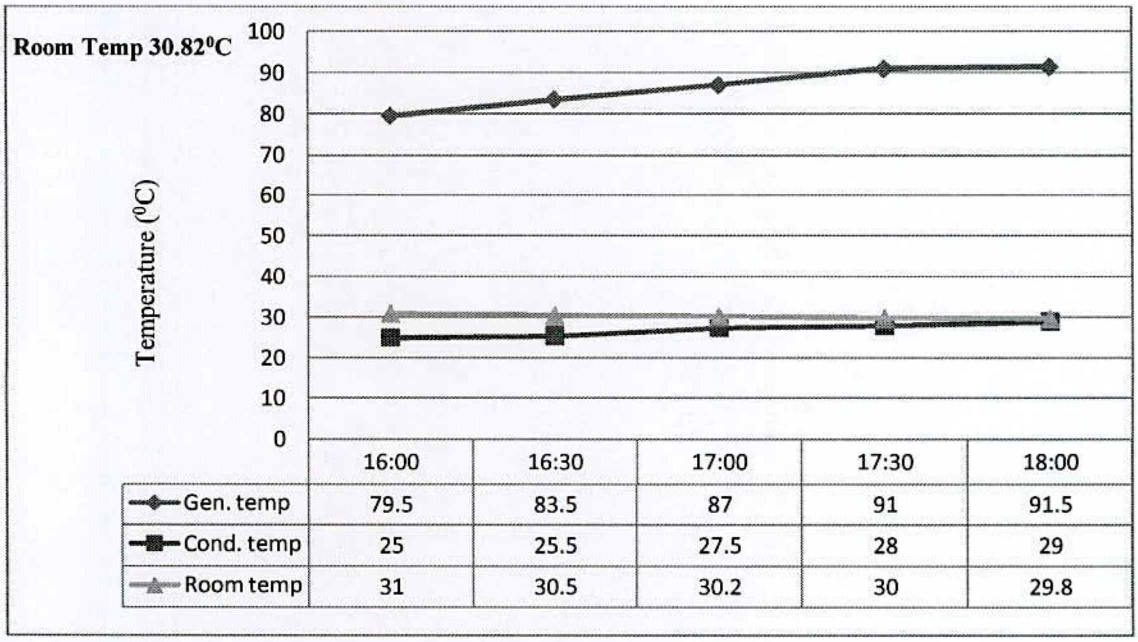
	16:00	16:30	17:00	17:30	18:00
◆ Generator Temp	73.00	76.50	77.00	78.00	79.00
■ Cond. tenmp	24	24.50	25.00	26.00	27.00
▲ Room temp	31.50	31.20	30.50	30.00	30.00

Figure 4.7(a): Temperatures profiles of the system during the desorption mode (1stobs.)



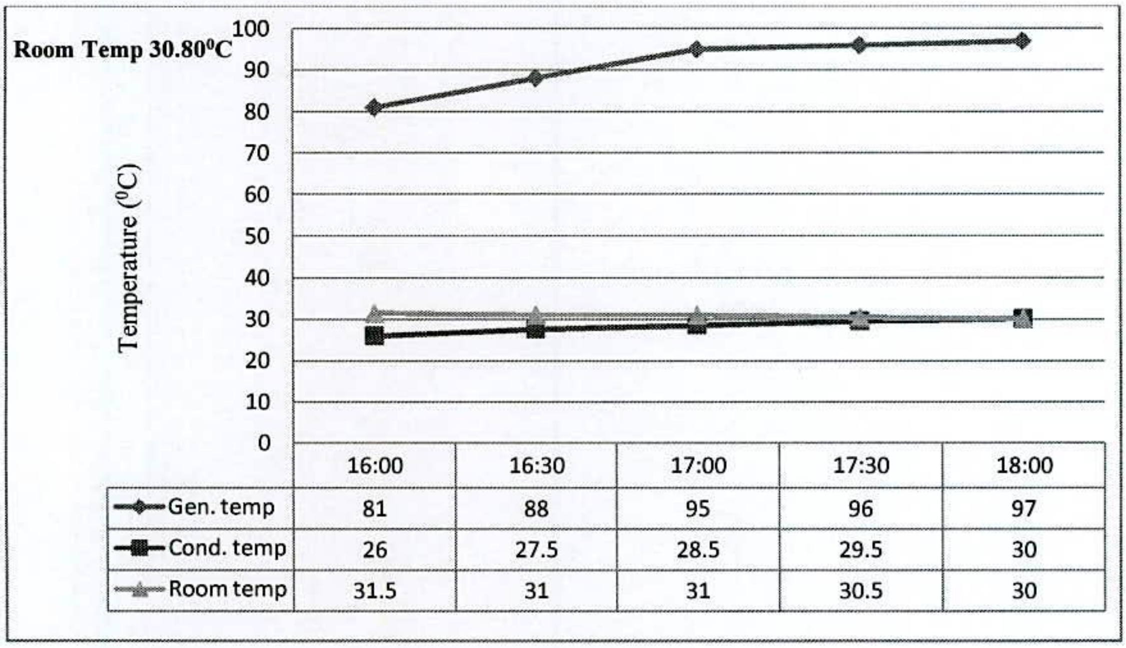
	16:00	16:30	17:00	17:30	18:00
Gen. temp	78.50	82.50	84.50	86.00	87.00
Cond. temp	25.00	25.50	26.50	27.50	28.00
Room temp	32.00	31.70	31.00	30.50	30.00

Figure 4.7 (b): Temperatures profiles of the system during the desorption mode (2nd obs.)



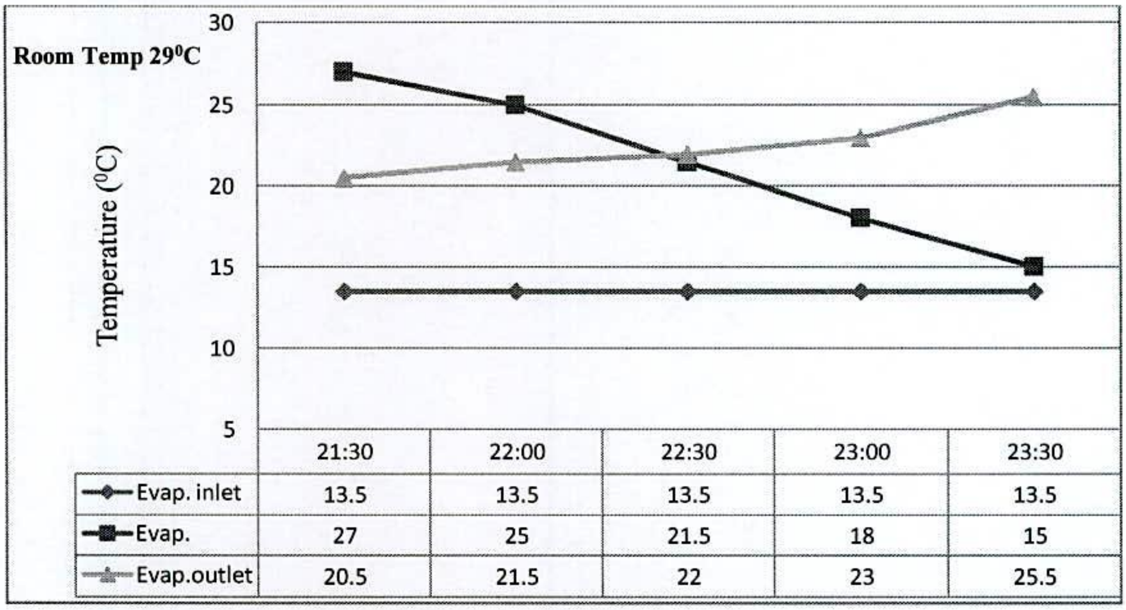
	16:00	16:30	17:00	17:30	18:00
Gen. temp	79.5	83.5	87	91	91.5
Cond. temp	25	25.5	27.5	28	29
Room temp	31	30.5	30.2	30	29.8

Figure 4.7 (c): Temperatures profiles of the system during the desorption mode (3rd obs.)



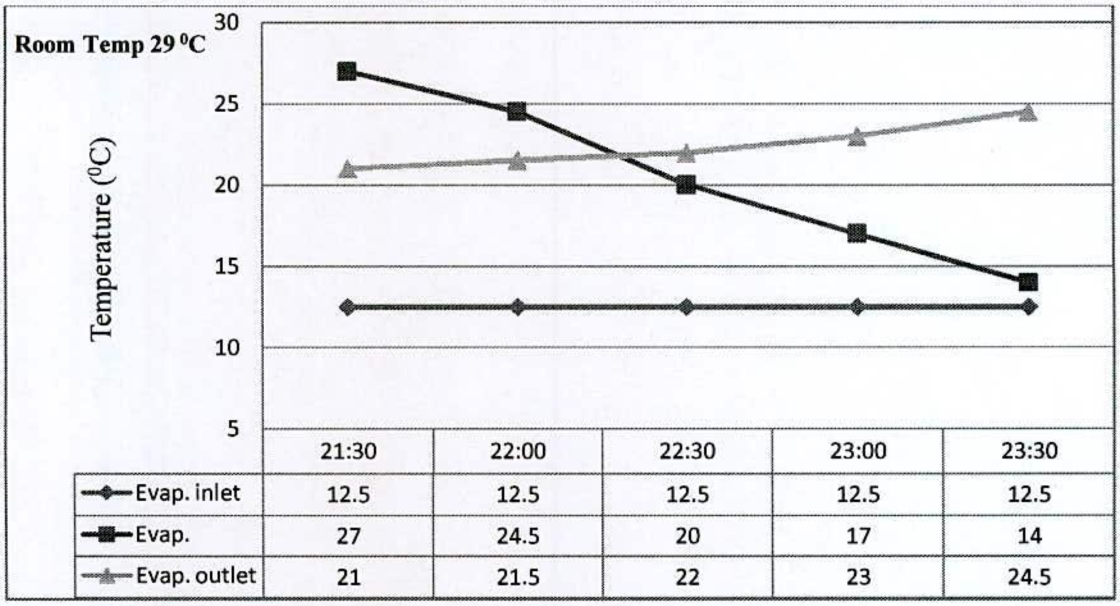
	16:00	16:30	17:00	17:30	18:00
Gen. temp	81	88	95	96	97
Cond. temp	26	27.5	28.5	29.5	30
Room temp	31.5	31	31	30.5	30

Figure 4.7(d): Temperatures profiles of the system during the desorption mode (4th obs.)



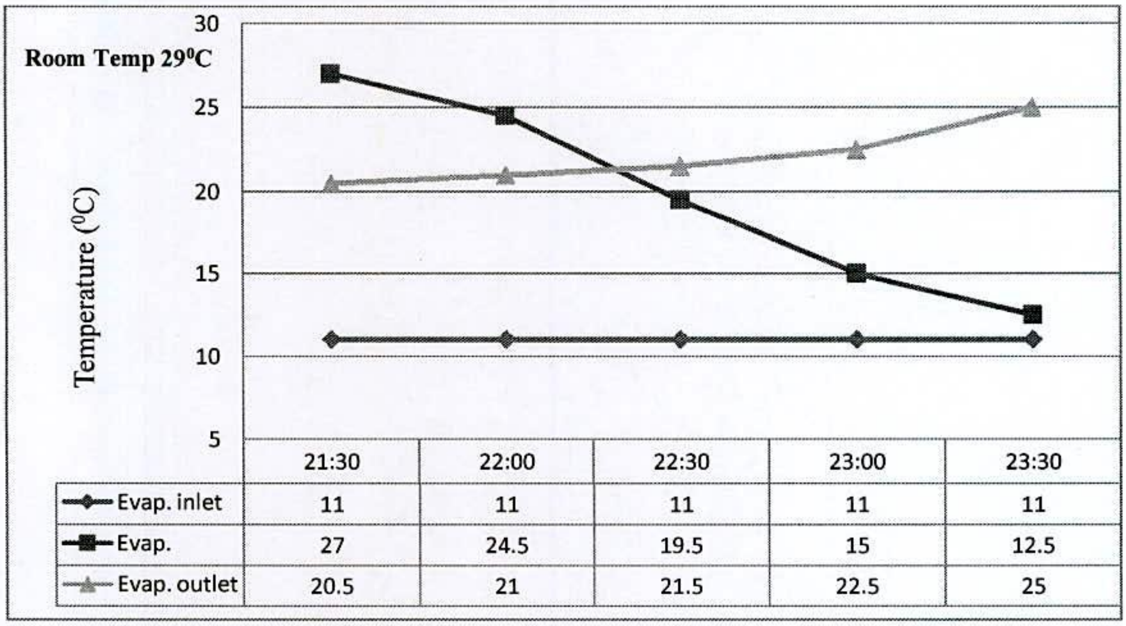
	21:30	22:00	22:30	23:00	23:30
Evap. inlet	13.5	13.5	13.5	13.5	13.5
Evap.	27	25	21.5	18	15
Evap.outlet	20.5	21.5	22	23	25.5

Figure 4.8 (a): Temperatures profiles of the system during the absorption mode (1st obs.)



	21:30	22:00	22:30	23:00	23:30
◆ Evap. inlet	12.5	12.5	12.5	12.5	12.5
■ Evap.	27	24.5	20	17	14
▲ Evap. outlet	21	21.5	22	23	24.5

Figure 4.8 (b): Temperatures profiles of the system during the absorption mode (2nd obs.)



	21:30	22:00	22:30	23:00	23:30
◆ Evap. inlet	11	11	11	11	11
■ Evap.	27	24.5	19.5	15	12.5
▲ Evap. outlet	20.5	21	21.5	22.5	25

Figure 4.8 (c): Temperatures profiles of the system during the absorption mode (3rd obs.)

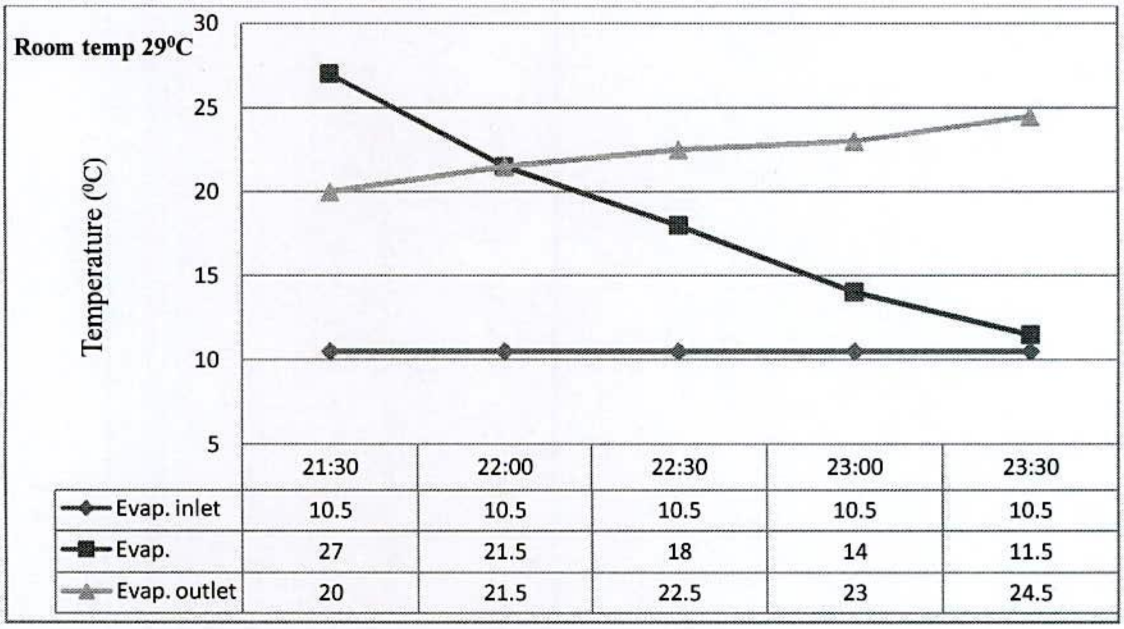


Figure 4.8 (d): Temperatures profiles of the system during the absorption mode (4th obs.)

From figure 4.7, it was said that the temperature of condenser increased with the increase of heat inputs to the system by almost constant room temperature. And from figure 4.8, it was seen that the temperature of evaporator (i,e, temperature of evaporator compartment water) almost decreased wherewith evaporator input temperature was same on average but evaporator output temperature increased with the certain limit.

CHAPTER V

Conclusions

5.1 Conclusions

An intermittent ammonia absorption refrigeration system was designed and constructed in this project work. Performances of the system with various generator temperature and heat addition were investigated. From the experimental investigations the followings could be concluded.

1. The average refrigeration temperature was found to be 13.25^{0}C with a minimum value of 11.5^{0}C and a maximum of 15^{0}C for a period of time up to 2.5 hours absorption.

2. The generator temperature was found to 91.40^{0}C corresponding to the oil bath temperature up to 110^{0}C during 2.5 hours desorption and the pressure was to be found 10.5 bar.

3. It could be seen that the generator pressure varies with respect to the heat input from the heat source of oil bath arrangement and also the generator heat gain was fairly related to the heat input by considering the heat losses from the heat source of the oil bath.

4. It could be drawn that the refrigerating effect increased with the generator heat gained. On an average the system could provide the heat gained by the generator of 1492.96 kJ and the refrigeration effect of 223.65 kJ.

5. The coefficients of performance (COP) also decreased with the increase of temperature of heat source of oil bath.

6. The coefficient of performance (COP) of the system was obtained at the average of 0.154 with a maximum value of 0.192. This coefficient of performance was higher than the coefficients of performance found by Katejanekarn [2] and Moreno-Quintanar [3] developed intermittent solar absorption system and also lower than the COP found by Tangka [4] developed intermittent solar absorption system.

5.2 Future Recommendation

1. Using an automatic control expansion valve can increase the pressure difference which also effect on the increment of the refrigerant effect though the high cost of it.

2. Instead of using felt on the outside of the pipe it would better for insulating the system with round thermocole.

3. By placing the evaporator pipes inside an insulated tank or container which is properly insulated and if the thermocouple wires are placed inside the heat exchanger tubes properly without any leakages and then recording the data will be the desired outcomes but it is quite difficult for our country for the shortage of properties and finding problems of goods and others shortcoming.

5.3 Suggestions and Safety Caution

1. The system could produce cold water but could not produce ice yet. The cooling capacity is fairly related to the heat input. Only 2.5 hour's absorption the evaporator temperature is to be maintained 11.5^0C and hence a greater time interval of absorption could produce ice yet and hence there is no solar radiation uncertainty problems in bad climates though of artificial heating source used inside the room or laboratory. It is expected to obtain the optimum temperature and heat addition in generator for better refrigeration effect in evaporator and also be helpful to design and construct of a

suitable solar collector for the practical solar operated intermittent vapor absorption system.

2. Working with pure ammonia can be dangerous if safety precautions are not taken. Pure ammonia is poisonous if inhaled in high enough concentrations, causing burning eyes, nose, and throat, blindness, and worse. Since water combines readily with ammonia, a supply of water (garden hose or other) should always be on hand in the event of a large leak. The current research work unit is a prototype and will not place it inside a dwelling until certain of its safety. Unlike some poisonous gases, ammonia has the advantage that the tiniest amount is readily detectable by its strong odor.

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