SUDAN UNIVERSITY OF SCIENCE
& TECHNOLOGY
COLLEGE OF GRADUATE STUDIES

DESIGN, SIMULATION OF SOLAR REFRIGATOR OF ADSORPTION PRINCIPLE

تصميم ومحاكاة ثلاجة تعمل بالطاقة الشمسية على مبدأ الإمتزاز

A THESIS SUBMITTED IN FULFILLMENT OF THE REQUIREMENT OF THE DEGREE OF DOCTOR OF PHILOSOPHY (PhD) BY COURSES AND DISSERTATION IN MECHANICAL ENGINEERING

By

Mudathir Abbas Awadelkareem Jamrh

Supervisor

Dr. Ali Mohammed Hamadan Adam

August 2020

SUDAN UNIVETSITY OF SCIENCE AND TECHNOLOGY

COLLEGE OF GRADUATE STUDIES

DESIGN, SIMULATION OF SOLAR REFRIGATOR OF ADSORPTION PRINCIPLE

This thesis has been approved and accepted in partial fulfillment of the requirements for the degree of Doctor of Philosophy (Ph.D.) in mechanical engineering

Examination Committee

External Examiner	Signature
Prof. Mohammed Eltayeb Mansour	....
Rank: Associate Professor	Date: \ \ 2020
Internal Examiner	Signature
Dr.: Tawfig Ahmed Gamal Eldeen	....
Rank: Associate Professor	Date: \ \ 2020
Supervisor	Signature
Dr.: Dr. Ali Mohammed Hamdan Adam	....
Rank: Associate Professor	Date: \ \ 2020

SUDAN UNIVETSITY OF SCIENCE AND TECHNOLOGY
COLLEGE OF GRADUATE STUDIES

ASSIGNING THE COPY-RIGHT

STUDENT NAME: Mudathir Abbas Awadelkareem Jamrh

TITLE OFTHESIS: DESIGN, SIMULATION OF SOLAR REFRIGATOR OF ADSORPTION PRINCIPLE

☑ CONFIDENTIAL (Contains confidential information, accesses only possible to the abstract)

☒ OPEN ACCESS (Thesis to be published as online full text access)

I acknowledge that Sudan University of Science and Technology reserve sight as follows:

1. The project report is the property of Sudan University of Science and Technology
2. The library of Sudan University of Science and Technology has right to make copies of the thesis for purpose of research.
3. The library of Sudan University of Science and Technology has right to make copies of the thesis for academic exchange.

Student: Muhathir Abbas Awad	Supervisor: Ali Mohammed Hamadan
Signature …………………..…	Signature ……………………….……
Date ……….………………..	Date ……………………….……….

بسم الله الرحمن الرحيم

الآيـــــــــــة

{إِنَّ فِي خَلْقِ السَّمَوَاتِ وَالأَرْضِ وَاخْتِلافِ اللَّيْلِ وَالنَّهَارِ وَالْفُلْكِ الَّتِي تَجْرِي فِي الْبَحْرِ بِمَا يَنفَعُ النَّاسَ وَمَا أَنزَلَ اللَّهُ مِنْ السَّمَاءِ مِنْ مَاءٍ فَأَحْيَا بِهِ الأَرْضَ بَعْدَ مَوْتِهَا وَبَثَّ فِيهَا مِنْ كُلِّ دَابَّةٍ وَتَصْرِيفِ الرِّيَاحِ وَالسَّحَابِ الْمُسَخَّرِ بَيْنَ السَّمَاءِ وَالأَرْضِ لآيَاتٍ لِقَوْمٍ يَعْقِلُونَ }

(سورة البقرة - الآية (164)

Dedication

To the soul of my father, mercy of God upon him, To My

Mother

Brothers and Sister

To my wife, my sons

For their love and endless support

To all whom admired this piece of work

Acknowledgment

I would like to express my thanks and deep appreciation to my supervisor Dr. Ali Mohammed Hamadan Adam and for his continuous help and encouragement, for his patience and endless support. I would like to extend my special thanks to the committee members Prof. Mohammed Eltayeb Mansour and Dr Tawfig Ahmed Gamaleldin Abdalrhman, for their valuable time in reading and making comments on this thesis. I would like to express my thanks and respect to the Department of Mechanical Engineering and Supreme Council for Vocational Training and Apprenticeship Program, Finally, I am grateful to all my lovely family, for their support and encouragement.

ABSTRACT

*The objective of this thesis is to design and simulate Intermittent solar adsorption refrigeration machine. A solar heated charcoal / methanol of refrigeration machines as small unit of solar freezer on adsorption principle is used. The unit has a product capacity of about 5kg of ice daily. A flat-plate solar collector of (1*m^2*) area, and a water-cooled condenser of natural convection type is used. The evaporator was a flooded type evaporator, attached to the ice box to form a fully insulated unit. The machine was designed to work as intermittently; the night period is the productive period; while day period is the generative period. This type of machines has a great need in the society, especially in the rural areas, for general cooling purposes and in health centers for vaccines saving and others. The research satisfied its objectives and that it shall continue to the stage of constructing this machine in commercial amounts for the benefit or need of our society. Further development and optimization of the system design is still required to improve coefficient of performance (COP).*

المستخلص

الهدف الاساسى لهذا البحث هو تطوير المعرفة والخبرة العملية لتصميم وحدة تبريد شمسى متقطع تعمل على مبدأ الامتزاز وتستخدم مخلوط من الميثانول والفحم . تنتج هذة الماكينة المصغرة حوالي 5 كجم من الثلج يوميًا.وذلك يتم باستخدام مجمّع اللوحات الشمسية المسطحة، ويتم استخدام مكثف مبرد بالماء من النوع الحراري حيث المبخر عبارة عن مبخر من النوع المغمور بالمياه ، موصول بصندوق الثلج لتشكيل وحدة معزولة بالكامل. تم تصميم الجهاز للعمل بشكل متقطع حيث أن فترة الليل هي الفترة المنتجة ؛ في حين أن فترة اليوم النهار هي فترة التوليد. هذا النوع من الأجهزة لديه حاجة كبيرة في المجتمع ، وخاصة في المناطق الريفية للإستفاده منه فى إحتياجات التبريد العامة وفي المراكز الصحية لتوفير و حفظ اللقاحات وغيرها. لقد حقق هذا البحث أهدافه و ذلك بالعمل على بناء هذا الجهاز بكميات تجارية لصالح و حاجة المجتمع ويتم تطويره بهدف تحسين معامل الأداء ((COP)

TABLE OF CONTENTS

Chapter Four

Design ,Results & Discussion

Chapter Five

Conclusion & Recommendation

List of Tables

LIST OF FIGURES

LIST OF SYMBOLES

Symbol	Definition
A	Angstrom unit =10^{-10} m ,3.973x 10^{-11}
atm	atmospheric pressure (1.0135 Bar)
C^o	Degree Celsius
c	Base of Nigerian logarithmic system(2.7182)
G	Total Solar Radiation
G_{sc}	Solar Constant = 1357
I	Solar incident radiation
n	Number of the day of the year
X	Concentration
α	Absorptivity Coefficient
β	Temperature Coefficient
γ	Isentropic expansion(Cp /Cv)
δ	Solar declination Angle
ε	Emissivity
η	Efficiency
θ	Angle, temperature difference
μ	Dynamic viscosity
σ	Stefan- Boltzmann Constant = 5.67×10^{-8} (W/ $m^2 10^{-4}$)
φ	Latitude
Δ	Difference
∑	Summation o
τ	Transmissivity Coefficient
HCRE	Heat Coefficient Receiver Evaporator

ABBREVIATION

Symbol	Definition
AIT	Asian Institute of Technology , Bangkok,Thailand
A.C,A/C	Activated carbon
HVAC	Heating, Ventilating and Air Conditioning
Cal	Calories
Ch	Charcoal
COP	Coefficient of Performance
NPV	Net present value
Nu	Nusselt Number
Pr	Prandl Number
Re	Reynolds Number
Gr	Grashof Number
WHO	World Health Organization

CHAPTER I

INTRODUCTION

CHAPTER I

INTRODUCTION

1.1 Introduction

Solar refrigeration is a useful application of solar energy in different areas of the world where there is a demand for cooling high insulation levels and no firm electricity to supply conventional power systems. Although different adsorption pairs had been studied to build adapted solar ice maker, the activated carbon–methanol pair was found to be the most suitable for solar-powered refrigeration since it could be driven by heat engine, near ambient temperatures and it is less expensive than other pairs. The adsorption solar refrigerator in its simplest form is a closed system composed of the container of adsorbents and adsorb ate (sorption bed), which serves as a solar collector, a condenser and an evaporator.

The system is divided into two periods: First, the adsorbent is heated by solar during the day and the desorbed adsorb ate is condensed. The adsorbent energy during is cooled after sunset, thereby re-adsorbing the adsorb ate, the evaporation of which produces the refrigeration effect. As desorption is highly endothermic, the heat input to the adsorbed be large enough to allow for sufficient refrigerant to be desorbed. On the other hand, adsorption is highly exothermal; cooling down of the adsorbed is also a major concern. The heating and cooling during the cycle perfectly suits the intermittent nature of solar energy, yet efficient operation of the system requires high rates of heat transfer in and out of the adsorbent. In addition, all solar systems usually suffer from the large variations in ambient conditions between winter and summer, which makes these systems inefficient for part of the year.

In solar adsorption systems, while good heating is attained during day time in summer, cooling during night by ambient air will be limited. On the other hand, in winter, the system will attain good cooling during the night, but heating will be insufficient. In the last two decades, different approaches have been developed to improve heat transfer rates and enhance heating and cooling of the adsorbent bed. The use of composite adsorbent blocks and monolithic carbon are useful methods to increase both thermal conductivity and density of the bed. To enhance heating of the beds, flat plate solar collectors with selective coating surface, evacuated tubular collectors and simple concentration non-tracking collectors as compound parabolic concentrator collector have been used.

As cooling of the adsorbent, heat loss from the collector to the ambient could be enhanced by means of removable insulation, flaps, or dampers. Some designs combine the collector and finned condenser in one unit since outside fins have been located on the rear surface of the collector.. Recently, simple tubular modules are investigated to be used in the adsorption system. The adsorption generator is at one end of the tube and a combined evaporator and condenser are at the other end. In a stainless steel tube with a new monolithic carbon bed was used while in a glass tube with a composite block bed made of silica gel and coated with a thin layer of active carbon was suggested. Using of the refrigeration tube is a promising method to overcome some of the problems of the adsorption system. The work presents a novel module composed of a modified glass tube and simple system of plane reflectors. Investigations are focused to improve heat transfer in the adsorbent bed, thereby increasing Coefficient Of Performance (COP), and to improve external heating and cooling of the bed all year round, thereby realizing good performance of the system for most times of the year.

The sun is defined as a sphere of intensely hot gaseous matter with a diameter of 1.39×10^9m and distance from the earth is about 1.5×10^{11}m. At the center of this sphere, the temperature is estimated at 8×10^6 to 40×10^6K and the density about 10^5 kg/m^3 and it considered as a continuous fusion reactor. The energy produced in the central zone, is transferred to the outer surface, and radiated into space. At the outer surface, the temperature is estimated to be about5000 K. The intensity of energy radiated from the sun, is considered to be constant. The value of the solar constant adopted by the World Radiation Center, is 1367W/m^2, the earth rotates about its axis, which leads to variation in the distance between the earth and the sun, and, consequently, in the intensity of solar radiation.

When radiation passes through atmosphere, it is partly absorbed and scattered by the atmosphere. The radiation reaches the earth: is beam radiation and diffuse radiation. Beam radiation is the solar radiation received on the earth surface without being scattered by the atmosphere while diffuse radiation is the solar radiation received on the earth surface after it is scattered by the atmosphere.

Solar energy can be converted to heat energy by using solar collectors, which collect heat, and transfer it to a fluid. The heat is used, for heating, cooling, generating electricity and drying crops. Solar collectors are divided into two main categories: Flat-plate collectors and concentrating collectors. A flat-plate collector is, basically, an insulated box with a single or double glass cover. Flat plate collectors are used to provide heat energy at moderate temperatures, up to about 100°C. In a concentrating collector, an optical device is used to concentrate the incident radiation on a small area. Concentrating collectors are used to provide heat energy at a higher temperature compared with flat-plate collectors. Two main types of concentrating collectors are used: Parabolic dish collectors and parabolic trough collectors. Parabolic trough collectors, use parabolic reflectors in a trough configuration, to focus solar radiation on a tube running along the length of the trough. As the natural flow of heat is always from a high temperature region to lower temperature one, this means that, in the refrigeration process, an external energy is to be supplied, so as to force heat to flow in the opposite sense of its natural flow. The general applications of refrigeration are in food preservation and air conditioning. There are two common methods of refrigeration in general: vapor compression refrigeration, and absorption refrigeration.

Cooling process can take place naturally or artificially. It will be natural when it takes place due to the nature of heat of flowing from higher to lower temperature by conduction, convection or radiation. There are currently much stronger economic and political drives to promote solar cooling technology in the market. However, making a competitive solar cooling machine for the market still remains a challenge to the academic and industrial communities. The common goals of present day research activities on solar assisted cooling are to find, for each different application of cooling, an optimum combination of collector and cooling system that matches the special cooling demands and also the constraints of the available solar radiation in the best way, with only marginal need for fossil fuels

1.2 Problem Statement

Solar adsorption refrigeration on the bases of solar energy effect is one of an important application of solar energy usage its only major problem is how to reach a reasonable and simple design with good selection of materials and solution applicable successfully in Sudan where solar energy is abundant and the need for cooling is great.

1.3 Objectives

The aim of this study is to develop a commercially viable heat driven charcoal-methanol for making ice by use of energy from solar collectors. An adsorption system used because of their low operational cost and maintenance, higher reliability, simple and continuous operations and no crystallization, corrosion or chemical disposal issues. The main objectives are:

1) To improve heat, transfer in the adsorb ate there by reducing size and cost.
2) To improve regeneration heat, transfer between beds thereby increasing (COP).
3) To carry out a mathematical analysis of the intermittent adsorption refrigeration system, and then write a computer simulation program, which can be utilized in predicting the performance of the system.
4) To design an intermittent methanol adsorption refrigeration system.
5) The research can offer a good contribution in the field of solar refrigeration in Sudan and enrich this type application especially for the use in rural areas for diverse kinds of need.

1.4 Research Methodology:

Design and Simulation methods are both will be used for the purpose of the research to see to what extent the results are accepted. Designed and simulation are necessary sensors for data acquisition readings together with computer program for simulation so that both results analytical and simulation can be compared.

1.6 Significance of the study

The research can offer a good contribution in the field of solar refrigeration in Sudan and enrich this type application especially for the use in rural areas for diverse kinds of need.

CHAPTER II

LITERATURE REVIEW

&

Adsorption Solar

CHAPTER II

LITERATURE REVIEW & THEORETICAL BACKGROUND

2.1 Literature Review

Asian Institute of Technology (AIT) Designed the first solar powered refrigeration experiments in 1974. A small prototype intermittent ammonia-water unit was developed with flat-plate collectors each of 1.44 m^2 in area. By 1977 the modified unit was capable of producing 9 kg of ice in one day. [Hu 3mg and R. H. B. Exell2015]

.

Williams D. A. designed a solar refrigerator to be used in rural areas. He used a parabolic concentrator type of collector of total collecting area of 0.77 m^2, with pair of working fluid of ammonia/water. COP of this machine was 0.36. [University of Wales, 1989].

Swartman conducted a research on the use of a solution of ammonia and sodium cyanide for an intermittent absorption machine. From the results of his research, better COP (0.11 - 0.27) compared to (0.05 - 0.14) for ammonia/water pair and better economic factors were achieved. [University of Wales, 1989]

Swartman et al made a survey of the progress from 1936 to 1972 and show that early work in this field used designs with concentrating collectors but later more attention was given to flat- plate collectors, beginning with the pioneering work of Sri Lanka [susdesign.com] where the unit was constructed of welded steel pipe. It was concluded that about 1.4 kg of ice could be produced per square meter of collector area on a clear day, and this performance could be considerably improved by a selective radiation coating on the collector surface.

Swartmanand Swarminathan [R. H. B. Excel, Solar 2012] studied another solar powered intermittent ammonia-water absorption refrigerator similar to that tested by Chinnappa Their unit differed from that of Chinnappa in two respects. First, instead of using a boiler and a solution reservoir outside the collector, they used a large header of diameter 152 mm, which provided the required capacity for the solution and a large surface for the evaporation of ammonia. Second, for the refrigeration process they introduced the ammonia vapor from the evaporator into a tube connecting the lower and upper headers of the collector, and the glass cover of the collector was opened to facilitate cooling. Tests showed that, the rate of refrigeration was too slow because of slower re- absorption of ammonia vapor into the solution. They proposed using the collector to dissipate the heat of absorption to overcome this problem. [R. H. B. Excel, Solar 2012]

. Marrdov et al designed and developed solar absorption refrigerator using a solution of ammonia/sodium chloride as working pair. Flat-plate collector was used for heat generation, the collector total area was 2 m^2 and a double sheet of glass for glazing purpose was also used. [University of Wales, 1989]

Florida Machine: (University of Florida), an absorption solar refrigerator was also designed, the unit uses a flat-plate collector with an electric pump, it was stated that the COP was 0.1and the productivity of the machine was 12.5 Kg of ice daily.

University of Wales, 1989. During the period 1978 to 1980 a larger unit was designed and tested consist two flat-plate collectors mirrors attached to the eastern and western edges to enhance the solar heating. On a bright day 14 kg of ammonia could be distilled, and refrigeration the following night give over 25 kg of ice from water initially at 28 C. A preliminary economic evaluation showed that the cost of ice produced by the system would be 2 to 3 times the city price, which was acceptable in remote areas.

Aggarwal Research [University of Wales, 1989]**:** presented the possibility of using solid absorbing material of calcium chloride and ammonia or the pair of lithium nitrate for their suitability to use in absorption solar refrigerators. He concluded that both two pairs have good absorptivity suitable for application in tropical regions. [University of Wales, 1989]

Staicovici Research: [University of Wales, 1989] In a research conducted on a solar powered intermittent adsorption refrigerator, receiving 46 MJ in one complete cycle gives a COP of (O.09-0.152) during the period from May to September. [University of Wales, 1989]

Moore and Farber 2012 made a theoretical analysis of the heat transfer and flow characteristics of tube-in-sheet flat-plate collector used as generators in solar powered ammonia-water refrigerators. The work culminated in a collector 1.49 m^2 in area consisting of 25 mm diameter steel pipes running from a 32 mm diameter feeder to a 63 mm diameter header. The centers of the 25 mm pipes were spaced 102 mm apart and were soldered to the galvanized iron sheet of thickness 1.0 mm. A single glass cover was used. [R. H. B. Excel, Solar Air2012]

Farber's Research**:** Farber's refrigerator, using a continuous Ammonia-Water cycle with electrically operated circulation pumps, was able to produce 12.5 kg of ice per square meter of collector area per day. This yield is twice what can be obtained using the intermittent cycle, but more components are needed in the construction of the system, and it would not work in locations where electricity is not available.

Village size ammonia/water solar refrigerator**:** The latest type of this system was constructed and tested in MT, it has a collector area of 25 m^2 and is designed to provide 0.5 m^3 of food storage space or 100 kg of ice per day under favorable conditions. The principle of operation is shown schematically in the figure (2.4 and 2.5).

2.2 Adsorption Solar

a): Ammonia/water Continuous absorption solar refrigerators

If the intermittent ammonia-water absorption cycle was replaced by a continues cycle, the yield of the system for a given solar input can be increased ,smaller amounts of refrigerant and absorbent are circulated and solar energy is not wasted in heating the larger quantities of liquid required in the intermittent systems. A mechanical circulation pump, driven electrically by means of a small heat engine incorporated in the system, could be used to maintain the necessary pressure difference between the evaporator and the generator. Another possibility is to use solar heat to operate an absorption refrigerator of the Platen-Minter's type, with hydrogen in the evaporator to maintain a constant pressure in the whole system However, using the compact non-tracking evacuated-tube solar collectors now in the market, and a commercial Platen-Munoz absorption refrigerator, Pal et al. 2012 recently devised and tested a system that has no moving parts and is made up entirely of ready-made components. The only modification required is a change in concentration of the ammonia-water solution to match the temperatures of the output of the evacuated-tube collectors; This development seems extremely promising. [R. H. B. Excel, Solar Air2012]

b) Adsorption Solar Powered Refrigerators:

Adsorption is the process of absorbing material into another material without any chemical reaction-taking place like absorbing water in zeolite or ammonia in charcoal. Although the ammonia-water system is efficient, it has certain practical disadvantages as:

a. The problem of rectification (removing water from the ammonia during generation).
b. The corrosiveness of the fluid
c. The high pressure and consequent heaviness of the equipment

The problem of rectification can be avoided by using a solid absorber for ammonia instead of water. For many years Worse-Schmidt has been developing intermittent solar adsorption refrigerators using calcium chloride as the absorber. In one of his papers he concludes that such systems can now be built, which is so simple and reliable that, they can be operated for years without maintenance. It is unlikely that they will be competitive with conventionally powered plant, but in many cases a solar driven plant will do a much-needed job which in many countries could not be accomplished with a conventional system.

c) Zeolite Solar Adsorption Refrigerators:

The Zeolite solar adsorption Refrigerators is chemically similar to sand, however it is not made of compact molecules, but rather possesses a crystal structure with large interior surface nature, there are about 40 different types of Zeolite. These mostly come from volcanic activity and contain large amounts of impurities, and are therefore for technical cooling purposes. The chemical industry develops different types of Zeolite artificially. The synthetic zeolite is mostly used as phosphate substitute in washing detergent or for example, as dry substance in some special types of glass windows.

The application of large amounts of zeolite in washing detergents has led to countless studies concerning its environmental compatibility. Through this it has been proven that zeolite has no environmental damaging properties and has no toxic of facts [solar.com].

The problem of high pressures can be avoided by using other refrigerants than ammonia. One combination is the zeolite-water system; the concept for using it is shown in figure 2.1.The quantity of water absorbed by the zeolite is strongly dependent on temperature and only weakly dependent on vapor pressure. Therefore, during the refrigeration process at night, when the collector containing zeolite is near ambient temperature, the water is readily absorbed keeping the pressure below 0.55 kPa and making the refrigerant water freeze. During the day the generation process is only slightly affected by the condensation pressure.

Intermittent solid adsorption refrigeration cycles provide promising systems for solar cooling. The charcoal/methanol pair has proved to be the best pair among those studied so far because it is reasonably stable chemically, it has a higher coefficient of performance and it is less expensive than the other pairs.

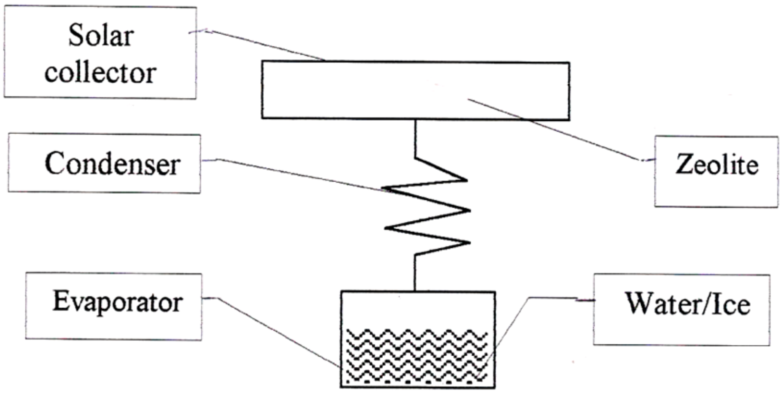


Fig. 2.1 ZEOLITE REFRIGERATOR

This type was developed by EG Solar in association with the firm ZEO-TECH of Munich, the project was promoted by the Bavarian Ministry of Economic Affairs. The solar-powered refrigerator consists of a cooler box with a built-in evaporator and uses water as the cooling agent. In addition to this, all that is needed is a manually operated vacuum pump and one or more canisters containing the non-toxic mineral Zeolite (naturally occurring mineral) and an SK 14 parabolic dish. To produce the desired cooling effect, one of the Zeolite containers is connected up to the evaporator. Air is extracted from the system with the aid of the vacuum pump. The vacuum created by the pump causes the boiling point of the water to drop so rapidly that the water boils at room temperature.

The advantages of the solar-powered refrigerator of this type are:

- Cordless refrigeration (which means that the cooler box can also be used to transport medicines).
- The use of simple and non-toxic materials
- The possibility to produce a refrigerator locally and extremely easily and for far less expenses than the conventional type [solar.com]

Bernig et al 1989 conducted a scientific research to establish thermodynamic functions that govern adsorption phenomena to be used as general models for thermodynamic equations for considerable number of usable absorbers in various applications. [Journal of Solar Energy, Vol.42, 1989]

Grenier Team Research**:** conducted another research for developing a solar powered cold store on adsorption principle using zeolite 13X/water pair intermittently. The unit uses a solar collector of total area 20 m^2 with 300 kg of zeolite material. The evaporator has a capacity of 190 kg of ice (fig 2.2) [Grenier and M. Pos, Experimental1992]

Meunier et al Research: Another important research in this field was conducted for comparing the COP of the two adsorption systems, zeolite/water and charcoal ethanol for their use in solar adsorption for ice-making. Their final result was that, COP for the pair of charcoal/methanol was higher (0.1 - 0.13) compared to(0.086 - 0.10) for zeolite/water pair. [“Revue Physics” Feb. 1986]

Pons et al 2011: designed and developed solar adsorption ice maker using activated carbon/methanol pair. The machine consists of four flat-plate collectors with area of 1.5 IN^2 each, and two air-cooled condensers of $24m^2$each, the unit has evaporator with a capacity of 35 kg ofice gave a COP of 0.12 and evaporator temperature of -3 °C . [M. Pos and Grenier, Experimental 2011].The collaborated teams of both Grenier and Pons (fig .2.2) have compared the thermodynamic performance of Zeolite-water and activated

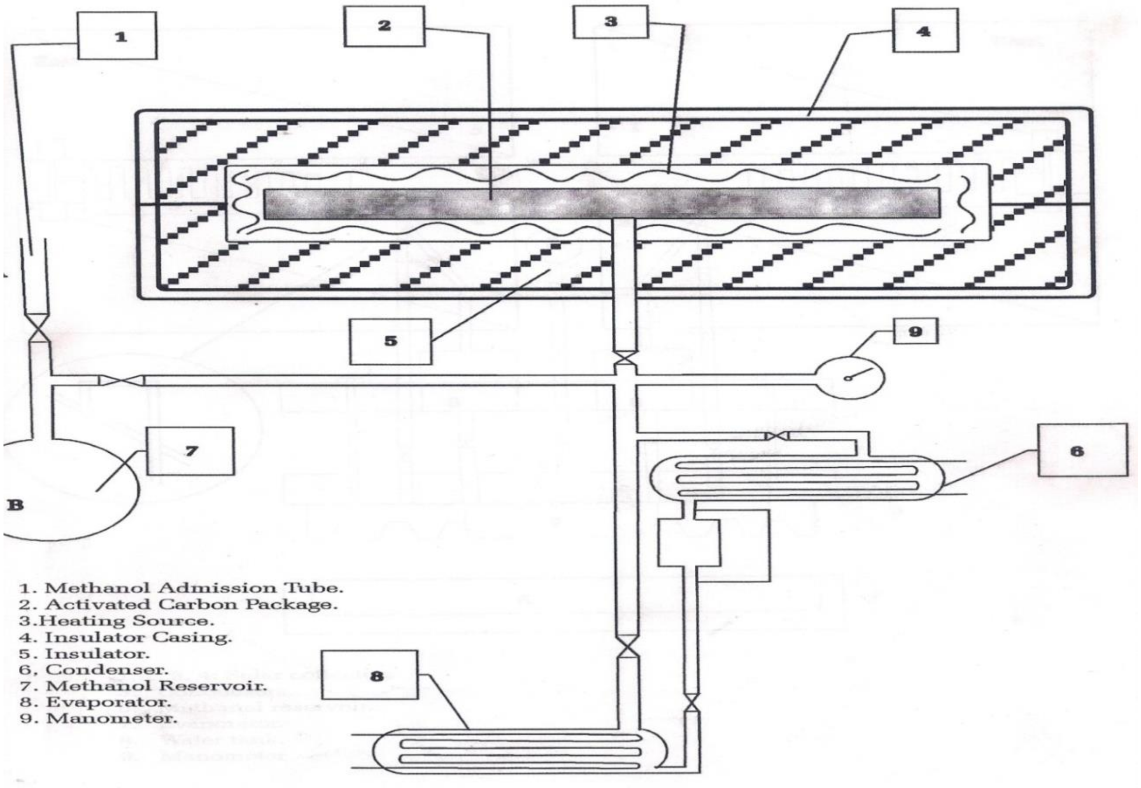
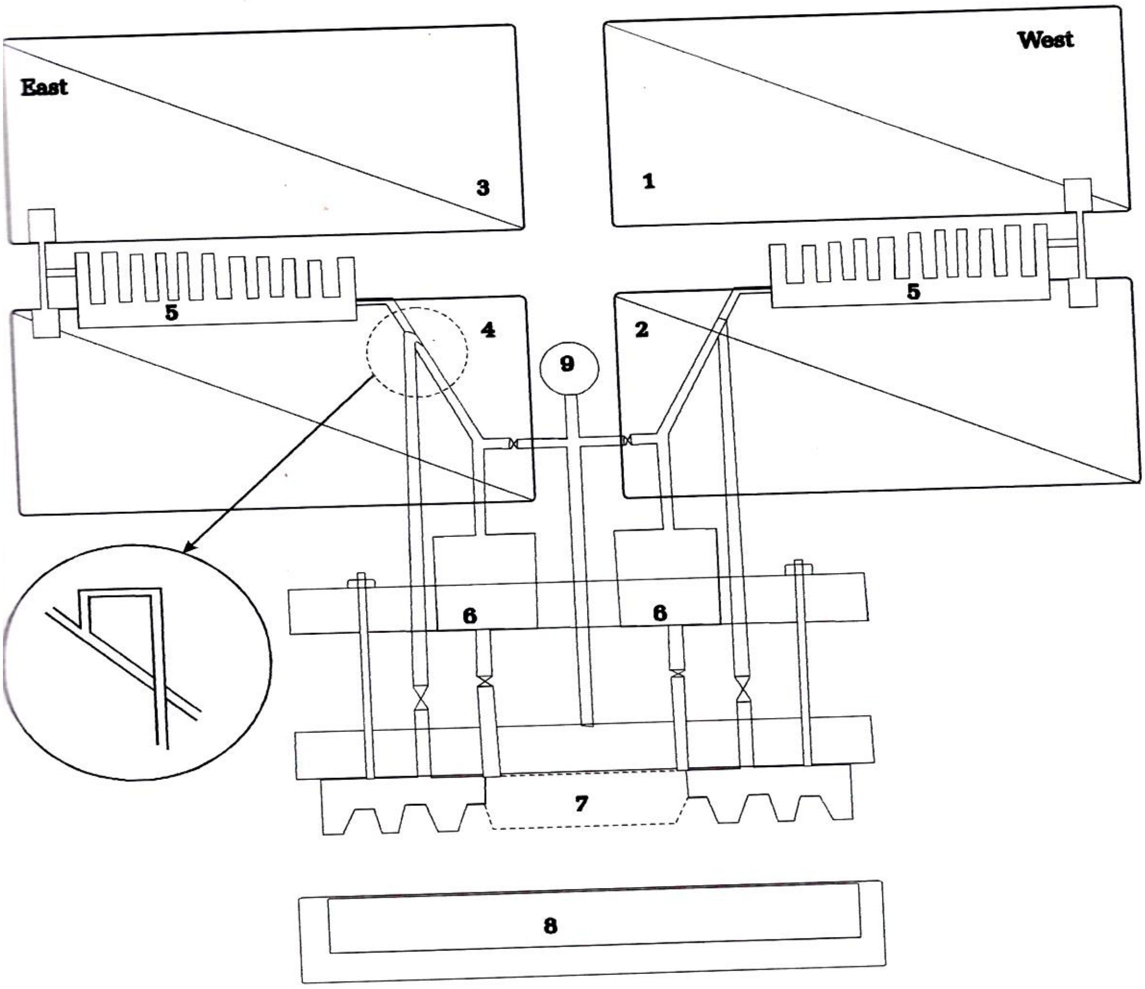


Fig 2.2 Experimental Unit for A Desorption Cycle [Grenier M. Pos, Experimental]

charcoal-methanol for application as solar ice maker. The results for condenser temperature T = 40 °C, evaporator temperature T = 0°C, adsorption temperature Ta = 25 °C and 35 °C, and a range of desorption temperatures T_d. At a desorption temperature of 100 °C, the charcoal-methanol combination has a collecting ratio in the range 0.4 - 0.5 as compared with 0.3 for zeolite-water. It is interesting to note that both activated charcoal and methanol are manufactured in developing countries, and that these substances are the most promising for solar cool storage systems.

Merghni Research 1989, in his research about the applicability of solar refrigeration systems, emphasized the suitability of charcoal/methanol pair. He stated that charcoal can absorb up to 30% of its weight under room temperature conditions, and is capable of desorbing up to 30% of the adsorbed amount under obtainable heating temperature on a flat-plate collector(fig 2.3) [University of Wales, 1989]



1, 2, 3, 4: Solar collectors
5: Condensers.
6: Methanol reservoir.
7. Evaporator.
8. Water tank.
9. Manometer

fig 2.3 Experimental Ice Maker (seen from the rear) [University of Wales, 1989]

- Critoph and Vogel : Another promising absorbent is activated charcoal. Critoph and Vogel have made measurements of the adsorbent properties of zeolite 5A, 1 3X and activated charcoal for the organic refrigerantsR11, R12, R22 and R 114 as possible pairs for solar cooling. Charcoal-R114 is interesting in that it operates at pressures near atmospheric pressure and units using it would be less prone to leakage. Critoph conducted a research on absorptive solar refrigerators, he focused on the hatchability of activated carbon with some available refrigerants like, ammonia and methanol solutions. Methanol has given better results, and with his studies about the thermodynamic cycles, he investigated their stability. He concluded that, activated carbon has good absorptive, and when it is used with methanol gave better COP but its only problem was that, it works under low pressures. (Sayigh., 2015).

CHAPTER III

SYSTEM & SIMULATION

CHAPTER III

System & Simulation

3.1 Introduction

Systems for utilization of solar energy may be classified into five categories

1 - Low temperature systems for space heating and cooling.

2- Concentrating collection systems, which produce temperature high enough for the generation of electric power.

3- Photovoltaic systems for direct conversion to electrical energy.

4- Photosynthesis systems based on the production of organic material, including, cropping systems and the organic waste material.

5- Systems that are direct drive from solar energy, including systems based on the wind, waves and temperature differences in the ocean. Each of these major systems are characterized by several different technical approaches.

3.2 Solar Energy

The sun's rays are the ultimate source of life on this planet and it is an obvious source of commercial energy. Certainly solar energy is plentiful enough. of course, the sun doesn't shine with the same intensity all the time moreover the suns availability differs widely among geographic areas. Nevertheless, solar energy has integrated individual scientists over the years, more practical applications involve the use of mirrors and absorbent materials that collect and concentrate solar energy on a small scale to heat or evaporate water. Solar energy sustains all the living organisms on earth in a sense

3.3- Economics of Solar Systems

The basic pieces of hardware needed to build a solar house are readily available. The private builders have little incentive for putting them into practice. The solar home is initially more expensive to build than conventional one, and consumers are largely ignorant of the potential savings on their energy bill. There is no incentive for the builders to innovate. The potential market for rapid solar energy development is the residential sector. The technology is simple, and enough solar energy reaches the roof of an average house to meet heating and cooling needs. It would, of course, be difficult to fix existing homes with the system economically, but in many cases it could be done. New homes being built could well be solar heated and cooled and the savings be measured in millions of barrels of oil each year. [Energy 2000 “k global strategy]

a. Solar Heating

Solar energy can be harvested economically to supply much of the space heating and cooling for new building. The opportunities of solar energy applications are well known, solar energy can no longer be laughed off. It is a source for immediate help in solving the problem of energy shortage. The technology is available to use solar energy to supply fraction of the hot water, space heating and air conditioning requirements of homes in many parts of the world indeed. Experimental solar-heated homes are already in existence in Boston, Massachusetts, Washington D.C. Denver, Colorado and elsewhere, and they have a satisfactory record of performance. These techniques must be perfected for mass production, but no new inventions are needed. [J. A. Duffie and W. A. Bekmaflfl,]

b. Electric Solar Cells :

These solar energy cells have been hand craft to meet stringent space requirements and are assembled much. There has been no concentrated effort to supply mass production techniques to reduce the cost. Solar enthusiasts believe that a research and development effort to improve efficiencies and develop. Fuel costs are much higher than solar energy equipment which is now being developed at a decreasing cost. if cells would be produced for $ 0.26 per watt the supplemental power can be economical for many uses. [Energy 2000 “k global strategy]

c. Solar Cooling

Solar heat can also be used to cool a building and experiments show that the concept is etiologically promising. The economics also seem attractive because there is a high degree of correlation between the availability of solar energy and the need for air-conditioning. The same basic collector system installed for solar heating can provide air-conditioning as well. In fact, the cost of heating and cooling a house with solar energy as compared with conventional sources may be extremely competitive

A comprehensive economic analysis with electrically heated homes even before the latest escalation in fuel prices shows that the fuel savings from solar energy more than pay for the extra investment. [Energy 2000 “k global strategy]

3.4- Solar Thermal Conversion

When a black body is exposed to solar radiation, it heats up, and as its temperature increases, the surface of that body loses heat to its surrounding at an increasing rate. At certain temperature, equilibrium condition will be reached where heat gain is equal to

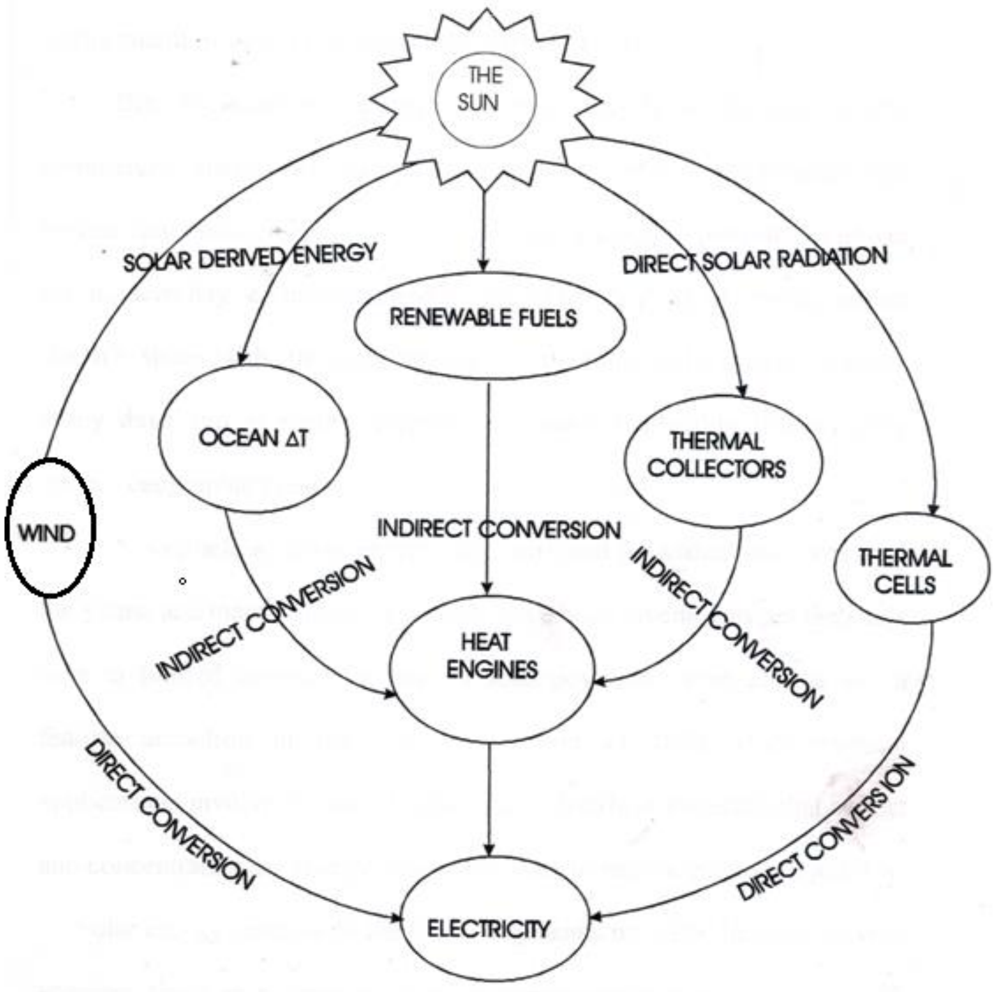


FIG3.1 Schematic Diagram of solar Energy Conversion

heat lost unless certain technologies are used to prevent heat loss or minimize its rate to allow continuity of heat gain, hence temperature increase. Solar energy utilization can take place through either passive or active systems. Passive systems are where solar energy are used directly; where active systems are those where solar energy is extracted using energy collecting means. There are two types of solar radiation, direct radiation and diffused radiation. Direct radiation is the one intercepted by a collector surface diffused radiation is portion scattered by dust particles or water vapor in the outer space and does not have a specified angle. Total global radiation is the summation of diffused and direct radiation.

3.5- Solar Energy Incident upon Horizontal Surface

The solar energy flux on a horizontal surface is given by the equation:

$$G_0 = G_{0,T}\mathrm{T}\cos\theta_Z \quad \ldots\ldots \qquad (3.1)$$

$$= G_{0,T}\,(\mathrm{Sin}\,\varphi\,\sin\delta\, + \cos\varphi\,\cos\delta\,\cos\omega\,)\ \ldots. \qquad (3.2)$$

Where $G_{0,T}$ Solar heat flux incident on the surface direct facing the sun

θ_Z = solar zenith angle

δ = solar declination

φ = Latitude

ω = hour angle

The total solar energy incident on an extra-terrestrial horizontal surface during the entire day is given by:

$$H_0 = \int_{tsr}^{tss} G_0\,\mathrm{dt}\ \ldots\ldots\ldots\ldots\ldots\ldots \qquad (3.3)$$

$$H_0 = \frac{24}{\pi}G_{sc}[\,1+0.033\cos\left(\frac{360n}{365}\right)]\times[\cos\varphi\ \cos\delta\,\cos\omega + \frac{\pi}{180}\,\omega\sin\varphi\sin\delta] \qquad (3.4)$$

Where:

tsr= Sunrise time

tss = Sunset time

G_{sc} = Solar constant = 1357 W/m2

s = Angle of tilt of the plate

n = day number of the year

3.6- Solar Refrigerators:

The concept of a solar home is really quite simple. A portion of the roof of the house is used as a "collector" made of a black surface to help absorb the heat and filled with water or air covered by sheets of glass to provide a "greenhouse" effect. The heats is transferred from storage into a large water tank or gravel's bin and circulate throughout the house in the same way as a conventional hot water or air system. A supplemental heating system may be required for cold days but the solar energy system supplies the greatest portion of the fuel supply, the portion varying, of course, with the region of the country.

In developing countries, there is an acute need for refrigerating food stuffs and medicines. However, normal refrigerators in order to work, they need electrical energy, often only available in large city centers, which are either difficult to access or simply impossible. However, the sun is shining everywhere and offers warmth in excess, especially in southern countries. With the assistance of a cooling aggregate and a parabolic solar collector, the warmth of the sun can be transformed into cold.

- **Choice of Solar Cooling Technology**

Solar cooling methods are reviewed by the World Health Organization(WHO) as part of its Expanded Programmer of Immunization in 1980.

Five alternative approaches were identified:

(1) Photovoltaic/vapor compression
(2) Photovoltaic/thermoelectric
(3) Solar thermodynamic - solid adsorption (zeolite /water)
(4) Solar thermodynamic - solid adsorption
(calcium chloride /ammonia)
(5) Solar thermodynamic - liquid absorption (water! ammonia)

- **Solar Absorption Refrigerator unit :**

The photovoltaic/ thermoelectric system which uses a solid state for cooling is of interest because no moving parts are involved although a fan is usually required to assist heat transfer from the hot side. Small thermoelectric refrigerators are available commercially for use in cars, and three examples were tested by WHO in 1979. These could not maintain sufficiently low internal temperatures or make ice.

3.7 Simulation Analysis:

The various studies of charcoal/methanol system were focused on prototype testing and the performance analysis of the ideal cycles. The numerical simulation is to describe the behavior of a solar powered charcoal/methanol refrigerator and, with the help of simulation model, the sensitivity of the performance to certain component parameters is also to be determined. The system modeled is based on the prototype charcoal/methanol adsorption refrigerator developed in [Solar Energy, Vol.42, 1989] where an intermittent adsorption cooling system composed of a container of adsorbent(charcoal), which serves as a solar heat collector, a condenser and a receiver/evaporator. The critical component of the system is the collector, the pressure modeling work focused on simulating the performance of the collector, which serves both as generator and absorber, in order to optimize its design. The model was validated by comparing the observed temperature and pressure histories of the solar powered charcoal methanol refrigerator

- **- Physical and Mathematical Model:**

The system modeled is shown in the fig. (3.2) shows with a the collector contains 14 stainless steel tubes of diameter 60 mm and length 1.2m arranged side by side oriented north south inside a casing. The casing has a single glass cover, well-insulated sides and back, and shutters at the top and bottom ends that can be opened to allow the tubes to be cooled by the natural convection of the ambient air. This configuration provided an effective collector area of 1.01 m^2.Within each collector tube there is another' concentric stainless steel tube of diameter 10 mm and perforated with small holes along its entire length. The annular space between the two tubes was filled with activated charcoal grains. The function of the small central tube is to ensure good distribution of the methanol on the charcoal, and eliminate pressure drops and temperature differences along the collector tube. The small tubes are connected to the rest of the system via a common header. The upper half of each collector tube receives heat from solar radiation and transfers it to the charcoal. The methanol adsorbed in the charcoal is then desorbed by the heat, and condensed in the condenser. Finally, the condensed methanol is collected in the receiver/evaporator. The collector tubes exchange heat with the glass cover, the casing, and the ambient air (when the shutters are opened).

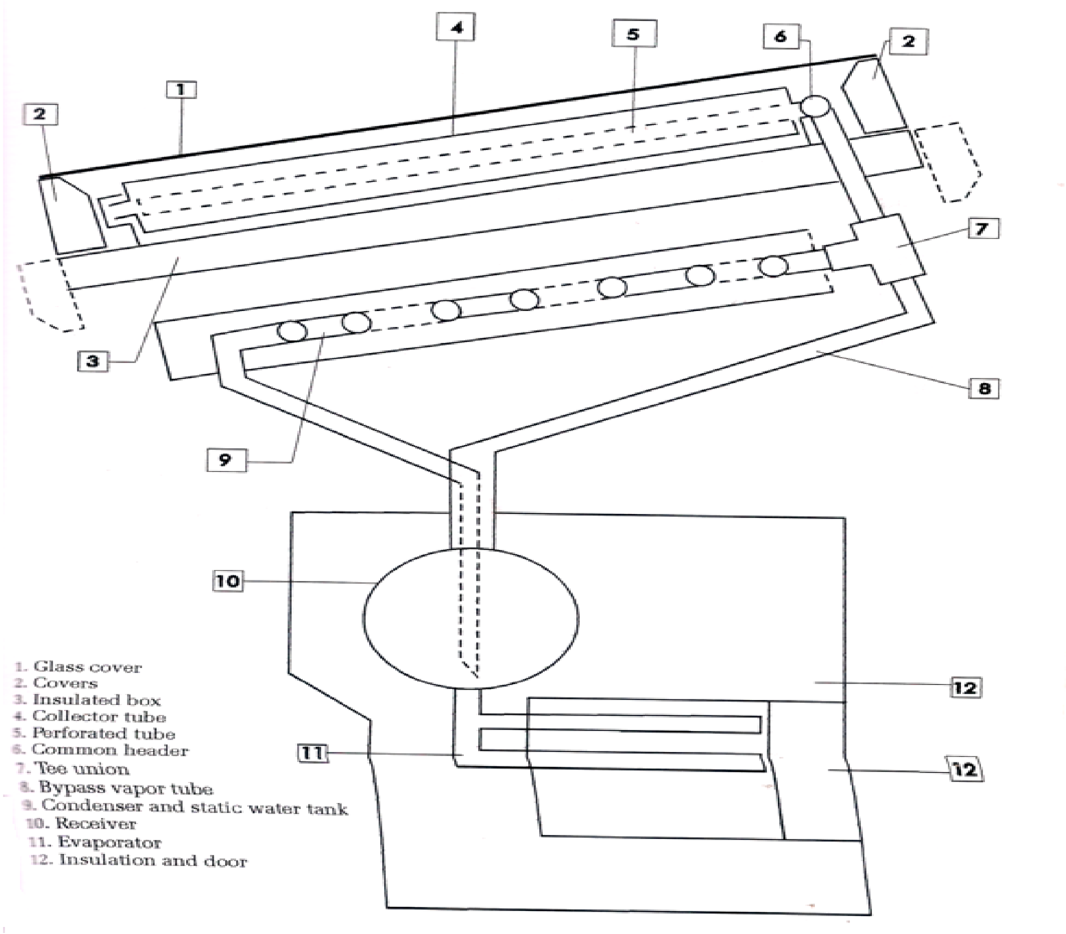


FIG 3.2 Sectional View of A. I. T. Refrigerator

- **Component Design**

Assumption

The model developed is based on the following assumptions:

1) There is no temperature gradient along the axis of the tube.

2) The charcoal and methanol are in local thermodynamic equilibrium.

3) The specific heat of adsorbed methanol is equal to that of the bulk liquid methanol.

4) Convective heat transfer and mass transfer resistance in the vapor-phase is neglected.

5) Side effects in the collector casing are neglected, and every collector tube is assumed to be in the same state.

a- Energy Balance:

Under the above assumptions, the energy balance equation is:

$$k\frac{\partial^2 T}{\partial r^2} + (k/r)\left(\frac{\delta T}{\delta r}\right) + (k/r^2)\left(\partial^2 T/\delta\theta^2\right) + q_{hs} = \rho C\left(\frac{dT}{dt}\right) \qquad (3.5)$$

T = Temperature (K)

r = radius (m)

k = thermal conductivity (W/mK)

θ = angle coordinate (radians)

q = heat source /unit volume (W/m3)

ρ = mass per unit volume (kg/rn3)

C = specific heat capacity (J/kgK)

t = time (second, hour)

For the adsorption heat source term per unit volume:

$q_{hs} = h_{ad}\rho_2$ (dX/dt) ………….[Hu 3mg and R. H. B. Exell2015] (3.6)

Subscript (2) = metal

For the metal tube; ρ C = $\rho_2 C_2$ + X $\rho_3 C_3$….. (3.7)

Subscript (3) = methanol

X = concentration (mass of methanol adsorbed on unit mass of charcoal) kg/kg %

Numerical solution of (eq. 3.5) is accomplished through the subdivision of the cross-section of the collector tube inside area into (M +2) concentric layers (with script i) along the radial coordinate, and N sectors (with script j) along the cylindrical co-ordinate. The first layer (i = 1) is the outside tube itself, and the layer (M + 2) is the inner tube itself. The layers of charcoal (from layer 1 to M+ 1) along the radius have the same volume except that the outmost layer i= 2 and the innermost layer i = M + 1 have half the volume of the outer layers.

b- **Dubinin-Astakhov** (D-A) equation [Solar Energy, Vol.42, 1989.]:

The equation states saturation of methanol in charcoal is written as:

X = ρ_3 W_0 exp [-D (T $\ln(\rho_s/\rho_0)^n$] ... (3.8)

W_0 = maximum volume available for adsorb ate in adsorbent (L/kg)

D (parameter) = K^n, K absorption coefficient & Subscript (s) = saturation

Chaperon equation and pressure conditions: $(\delta \ln P/\delta T)_x = h_{ad}/(RT^2)$ (3.9)

When x remains constant, i.e. before macro desorption starts, and after adsorption finishes, the system pressure P can be determined by the Chaperon equation:

$(\delta \ln P/\delta T)_x = h_{ad}/(RT^2)$ .. . (3.10)

h = heat transfer coefficient (W/m^2K) & subscript (ad) = adsorption

When the desorption starts, the system pressure is assumed to be equal to the saturation pressure of methanol at the condensing temperature T_{cond}:

$$P = Ps(T_{cond}) \ldots \quad (3.11)$$

The condensing temperature T_{cond} used in the model calculations as computed from the observed temperatures Tcw (subscript (cw) is for condenser water temperature) of the static water tank surrounds the condenser, with the help of an empirical coefficient (CCE): [Hu 3mg and R. H. B. Excel 2015]

$$T_{cond} = T_{cw}(H) + CCE\,[T_{cw}(H) - T_{cw}(l)] \ldots \quad (3.12)$$

H = Time interval

ECC= empirical condenser coefficient

T_{cw} (H) and T_{cw} (1) refer to the hour H and the initial hour (7:00 a.m.) respectively.

During the adsorption period when the receiver/evaporator is the coldest part, the receiver/evaporator temperature Trec controls the system pressure:

$$P = Ps(T_{rec}) \ldots \quad (3.13)$$

c- Initial condition:

Initially, the temperature and concentration of methanol in the charcoal is assumed to be uniform. Thus we have;

$$i=2, \ldots, M+1;\ j = 1 \ldots ,N; t = 0$$

$$T_{ij} = T_0 \text{ and } X_{ij} = X_0$$

And Po can be obtained from equation (3.8)

d- Glass collector

Equations for the Collector Casing and Receiver/Evaporator:

The glass cover is assumed to be at the same temperature T_g, whose change ΔT_g in the small time interval Δt is given by the equation:

$$(T_a - T_s)\, h_{ga} + (T_g - T_{sky})\, h_{gsky} + (T_g - T_{mp})\, h_{gmp}) + \delta\rho_g C\Delta T_g / \Delta t + Q_a = 0 \quad (3.14)$$

Where, T_a , T_{sky} are ambient temperature and sky temperature respectively.

T_{mp} is the mean temperature of the upper half of the collector tube.

h_{ga}, h_{gsky}, h_{gmp} are for air, sky and the upper-half of the collector tube respectively, and Q_a is the rate of absorption of solar radiation by the glass.

e- Casing:

The initial casing temperature assumed to be T_b, which satisfies:

$$(T_b - T_g)\, h_{ba} + (T_b - T_{md})\, h_{bmd} = 0.0 \ldots \quad (3.15)$$

Where; T_{md} is the mean temperature of the low half of the collector tubes. Because the insulation foam is a light material, its heat capacity is neglected.

f- Receiver/Evaporator:

During the evaporation /adsorption, it is assumed that receiver and the liquid methanol inside are at the same temperature T_{ev}, whose change ΔT_{rc} in the small time interval At is given by:

$$(m_i C_i + m_{rc} C_{rc}) \Delta T_{rc} + h_{rea} (T_{rc} - T_g) \Delta t + h_{rew} (T_{rc} - T_w) \Delta t = L_{ev} \Delta m_1 \quad (3.16)$$

Where, m_1, m_{rc} = mass of liquid methanol in receiver and mass of receiver/evaporator,

C_i, C_{rc} = specific heat capacity of the liquid methanol and the receiver/evaporator,

T_w = water temperature in ice tray,

L_{ev} = latent heat of evaporation of methanol at temperature T_{rc}.

g- Determination Parameters Values in the model Equation:

The solar heat absorbed per unit area per unit time (Qh_{gij}) by the element (ij) on the upper-part of the collector tube is given by:

$$Qh_{gij} = I_d \cos(\theta_j)\, \tau(\theta_g)\, \alpha + I_D\, \alpha\tau \quad (3.17)$$

Where θ_j = solar ray incidence angle to element (ij),

θ_g= solar ray incidence angle to glass cover,

$$\tau(\theta_g) = ((1-\tau)/(l+\tau)) \exp(-KL_g)) \approx (l-\tau)/(l+\tau) \quad (3.18)$$

$$\tau = 0.5\, (sin^2(\theta_g - \theta'')/ sin^2(\theta_g + \theta'')/tan^2(\theta_g + \theta'')) \quad (3.19)$$

$$\theta'' = sin^{-1}(\sin\theta_g)/n_2 \ldots. \,.\, \ldots \quad . \quad (3.20)$$

n_2 = index of refraction of glass,

$\alpha\tau$ = mean transmittance and absorption product for diffuse radiation.

The solar heat absorbed by the glass cover per unit area per unit time is:

$$Q_g = I_d \cos\theta_g\, \alpha\, (\theta_g) + I_D \alpha \quad (3.21)$$

Where the absorbance is given by:

$$(\theta g) = (1-\delta)\, (1-\exp(-KLg))/(l - \delta\exp(-KLg) \quad (3.22)$$

In the day time when the dampers are closed, the top heat loss coefficient h_{at}(including convective and radiative loss), is calculated within empirical formula developed by Klein and discussed by Duffie [Thermal Process, John Wiley (1980)].

$$h_{ta} = [\frac{1}{\frac{C}{T_{p,m}}[\frac{(T_{p,m} - T_a)}{I+f}]^e} + \frac{1}{h_w}]^{-1} + [\frac{\sigma(T_{p,m} + T_a)\,({T^2}_{p,m} + {T^2}_a)}{(\epsilon_p - 0.00591 h_a)^{-1} + \frac{2+f-1+0.133\epsilon_p}{\epsilon_g} - 1}]$$

(3.23)

Where:

$T_{p,m}$ = tube temperature,

f = $(1 + 0.089h_w - 0.1l6\ h_{w\epsilon_p})(1 + 0.07866N)$ .... (3.24)

e = $0.043(1 - 100 / T_{p,m})$ .... (3.25)

C= $520(1 - 0.000051\beta^2)$ .... (3.26)

β =horizontal tilt angle,

$\epsilon_{\boldsymbol{p}}$, $\epsilon_{\boldsymbol{g}}$ = emittance of collector tube and glass cover,

h_w= wind heat transfer coefficient = 2.8 + W * WCC (3.27)

W = wind speed (m /s)

WCC = Wind correction Coefficient

The determination of the parameter WCC is discussed below:

The back convective heat loss coefficient $h_{j-c,c}$from the warmer lower half of the collector tubes to the cooler back casing a distance, is calculated by the formula given by Bejan: [Ken-ichiKiniura, Solar Adsorption]

Nu $(\boldsymbol{\tau})$ = [I + [Nu $(\mathbf{90^0})$ -1]$\mathbf{sin}(\boldsymbol{\tau})$ (3.28)

Where $\boldsymbol{\tau}$= 180 collector tilt angle, and $h_{j-c:c}$= Nu (τ) h_{air}/L

The convective heat loss coefficient plus the radiative heat loss coefficient from the lower half of the tube to the casing (at temperature T_b) equals the total back loss coefficient. When the shutters are opened, wind correction factor (WCF) should be introduced to account for the total convective heat transfer coefficient

h_{tc} In the evening, which is defined as

h_{tc}= Natural convective coefficient + W * AI + WCE ....(3.29)

AI = cross-section area of shutter channel in the collector (m^2)

Following Guilleminot and Meunier [Heat and Mass Transfer in a non-isothermal], a heat transfer coefficient H_l between charcoal and the metal tube, and an equivalent conductivity of charcoal k, were introduced in the model to account for heat diffusion inside the collector tube. for the charcoal pair used, is determined experimentally The parameters. W_0, D and n (in eq 3.8) charcoal used in this refrigerator according to AIT’s design are 0.365 I/kg, 14.96 x 10^{-5} and 1.34 respectively [University of Wales, 1989.], The parameters such as WCE, WCED, HI , k, CCE, (see appendix D) etc. are difficult to estimate the theoretically, they were determined from experimental data by the tentative identification method of Grenier et al .in which actual temperature histories of the system under the sun are compared with temperature histories calculated by the model. The criterion of error chose in the method is the root mean square difference F($\propto$, $\boldsymbol{\beta}$) between the measured temperatures

$$F(\propto, \boldsymbol{\beta}) = (\textstyle\sum_{H=i}^{N} [\boldsymbol{T_i T_{chH-} T_{chH}})^2 + (\boldsymbol{T_i T_{upH-} T_{upH}})^2 + [\boldsymbol{T_i T_{loH-} T_{loH}})^2] / \mathbf{3\,N})^{\frac{1}{2}}.. \quad (3.30)$$

Where the subscripts have the following meanings:

ch = charcoal average

up = upper half part of the collector tube [Hu 3mg and R. H. B. Excel, Simulation]

lo = lower half part of the collector tube,

H = at the time step, and N = the number of time steps, which in this case is 48 per day for half-hour steps. The parameters are taken in pairs and are adjusted interactively until the values of F($\propto$, $\boldsymbol{\beta}$) are minimized. The values of most parameters were steady except for the WCE and WCED.

The value of HI is found to be (15 W/m^{2}K) this value depends on the exact packing of the charcoal container and different size of charcoal particles in the collector. The value of k was (0.19 W/mK).The parameters WCE and WCED are directly linked to the heat loss coefficient UT of the collector. For the three days' data, the hourly value of UT, which depends on wind speed and the temperature of the collector, was also calculated. In the day time when the shutters were closed, the hourly value of UT (upwards) was in the range 2.5 to 3.5W/m2K and at night when the shutters were opened, UT (upwards) was in the range 5.5 to 15 W/m2K [3mg and R. H. B. Excel, Simulation 1992]

- **Experimental Validation of the Model:**

Two other typical days' test data were selected to check the model and programmer. In the calculations, the values of AIT [3mg and R. H. B. Excel, Simulation 1992]used were those measured at the end of each half-hour time step, while the values of solar radiation, ambient temperature and wind speed were the mean values during the previous half-hour. The cross-section of the collector tube was divided into eight concentric layers (M = 6) and eight sectors (L = 8) in these calculations. The agreement was quite good between the observed and calculated values. [Simulation of an Intermittent Adsorptive Solar Cooling System, 1989]

- **- Sensitivity Analysis:**

To represent the system performance and to indicate the influence of different factors, four solar COP_s were introduced, they are defined below:

$$COP_1 = (Q_{ev} (M_{meth} - M_{left}) - Q_1) / S \tag{3.31}$$

This is the theoretical maximum net solar COP.

$$COP_2 = ((M_{max\,ice} L_{ice} + C_w M_w (30° - T_{min}))/S \tag{3.32}$$

This is the maximum useful solar COP.

$$COP_3 = (M_{ice\,morning} L_{ice} + C_w M_w (30° - T_{min}))/S \ldots \tag{3.33}$$

$$COP_4 = (M_{ice\,morning} L_{ice\,morning} + C_w (30° - T_{morning}))/S \tag{3.34}$$

The symbols in these equations are as follows:

Q_{ev} = heat of evaporation of methanol at design temperature,

M_{meth} = Maximum mass of liquid methanol collected during the day

M_{left} = liquid methanol left in the receiver/evaporator next morning.

Q_1 = heat necessary to cool the liquid methanol and the receiver/ evaporator from final condensing temperature to evaporating temperature

S = total solar radiation incident on the collector during the day

$M_{max\,ice}$ = maximum mass of ice produced at night

L_{ice} = latent heat of fusion of ice

$M_w C_w$ = mass and heat capacity of water in ice tray,

T_{min} = minimum water temperature in ice tray,

$M_{ice\,morning}$ = mass of ice left the following morning

$T_{morning}$ = Temperature in ice-tray the following morning.

- **Dependence of Cycle Performance**

For the actual system without valves, the main daily operation is to “close” and “open” the shutters in the morning and afternoon. The time at which the shutters are opened and closed affect the daily cycle, especially the adsorption part of the cycle. Besides operating the shutters, putting water in the ice trays for producing ice at night is also daily task. If too much water is put in the ice tray, no ice will be produced; with too little water the evaporator temperature will be too low and the heat leak to the evaporator will be high. From the above results, the influence loose is to produce ice, operating the shutters at about sunset and putting the proper amount of water in the ice-tray is important.

- **Sensitivity Observed:**

a) - Collector Tube:

With one square meter collector area, the performances of the system with the same inner gauge tube, the same thickness of tube wall, and the outside diameter (OD) of the tube ranging, the charcoal in the collector is less than 19.9 kg. As OD increases, the mass of charcoal contained inside the collector tube increases and the $\mathrm{COP_s}$ go up. As a peak value of $\mathrm{COP_s}$ occurs, at OD of about 0.1 m the total charcoal inside the collector was 35.4 kg while. in these calculations the thickness of the wall of the tube was assumed to be the same. However, in fact, the bigger OD normally has a bigger thickness for stainless steel pipe in the market. As the wall thickness increased the $\mathrm{COP_s}$decrease, especially for the smaller OD tube.

b) - Casings:

For the icebox casing, COP decrease when HCRE (Heat Loss Coefficient Receiver/ Evaporator) increases. For chilling water only, a bigger HCRE is acceptable; for ice making, the HCRE has to be less than 0.25 W/K in this particular case [3mg and R. H. B. Excel, Simulation 1992] . A small HCRE requires the use of costly insulation but the system performance will not improve much.

c) - Heat Transfer between Tube and Charcoal:

The values of the heat transfer coefficient between the metal and charcoal(H_1), are not sensitive with the COPs in the range of (20 - 80 W/m^2K). The sensitivity of k (charcoal conductivity) the bigger k seems better especially for the big OD tube. Both H_1 and k depend on the charcoal particle size, the packing of the charcoal grains and the shape of the container these two parameters in general, do not greatly affect the performance.

d) - Condenser Heat Transfer Coefficient:

The efficiency of the condenser is represented by the condenser correction coefficient (CCE). For fixed temperature of the static water surrounding the condenser, a large value of CCE means a poor heat transfers between the condenser and the water. COP_1

decreases but not much. In general, the lower the condensing efficiency (bigger CCE), the worse the performance of the system.

e) - Emittance of the Collector Tube Surface:

The long wave emittance of the collector tube surface depends on the nature of the surface used, which may be selective coating or non-selective black paint. Selective coating is recommended especially for ice making.

f) - Solar Radiation:

It has been proved that a threshold value of solar radiation is necessary to make ice or to cool water. Over this threshold value, the COPs increase sharply with increasing insulation. In fact, the only useful solar radiations that falling on the collector before the adsorption starts. For this reason, it is better to have the azimuth of the collector slightly towards the east (10 to 30 degrees) instead of due south for the best COPI [3mg and R. H. B. Excel, Simulation 1992]

g) - Selection of Activated Charcoal:

The initial charge of methanol is important for the system performance. The COPs increase as the initial concentration X (1) increases up to an optimum point. When X (1) is greater than the optimum point, the COPs decrease with the X (1) increasing. This is because there is some un evaporated liquid methanol remaining the following morning and the minimum evaporating temperature at night increases gradually with (l) increasing. For the different types of charcoal, values of optimum points are different. Among the property data (W_O, D, n) of the different types of charcoal, W_O has the greatest effect. Charcoal with bigger W_O can give a bigger maximum COP but need higher initial X(l). Beyond any optimum point, the decrease in COPI with increase in X(l)is not large. Therefore, slight 0verebargîflg of methanol would be guarantee for a good performance when the position of the optimum is not known exactly.

- **Summary:**

a. The daily operating conditions (the time the shutters are opened and how much water is put in the ice tray) will influence the performance of the system, especially for the ice making purposes. Generally, the suitable time to open the dampers is one to two hours after the collector reached its daily peak temperature, though there is still solar radiation at that time.

b. There is an optimum outside diameter for the collector tube (in other words an optimum rate per square meter of collector area). The actual value of the optimum OD depends on the meteorological condition in a particular area and on the material used.

c. The thickness of the wall of the tube would influence the COPs; the thicker the wall, the worse the COPs, but not significantly.

d. A selective absorbing surface is strongly recommended, otherwise the. COP will be insufficient for ice making.

e. The heat transfer coefficient Hl between the tube and the charcoal, and the equivalent conductivity k of charcoal do not influence the system's performance significantly.

f. The initial concentration of methanol charge into the system strongly influences the system COP. For different charcoals, the initial is quite different and it depends on whether the purpose is to chill water or to make ice. Slightly overcharging of methanol is recommended to ensure near-optimum performance.

g. The collector for such a solar refrigerator should face slightly towards the east (10 - 30^0) instead of due south for a Northern Hemisphere location.

- **Comment**

The major difference between the model and the developed machine is in the design of the collector, when AIT's design selected is a tubular collector, for this project, the selected design is a flat-plate type with attached charcoal housing box. Also the shutter in All'^{s} design was at the top of the collector on the glass side; whereas for this project they were designed to be at the bottom of the collector.

3.8- Refrigeration Process

Refrigeration is a process of removing heat from a cold space to relatively hotter one. Cycles that are receiving heat at low temperature and rejecting it at high temperature is called Refrigeration cycles.

a. Thermal Solar Refrigerators

These are thermal driven refrigerators, which are described in figures (3.3 that shows the principle of operation of the intermittent -water absorption refrigeration cycle. Initially the left-hand vessel contains ammonia-water solution, having with about 45% of ammonia by weight, at ambient temperature of 30°C.

Basic Concepts:

Initial state:

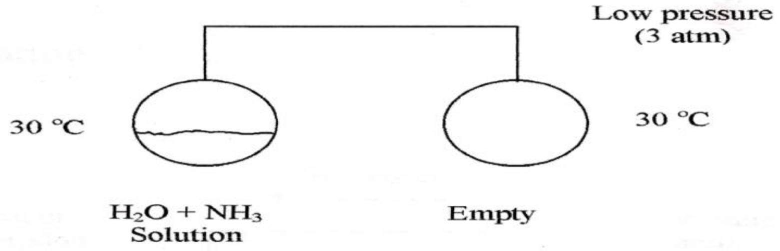


Fig.(3.3 a) Thermal Driven Refrigerator (Initial Stage)

Generation

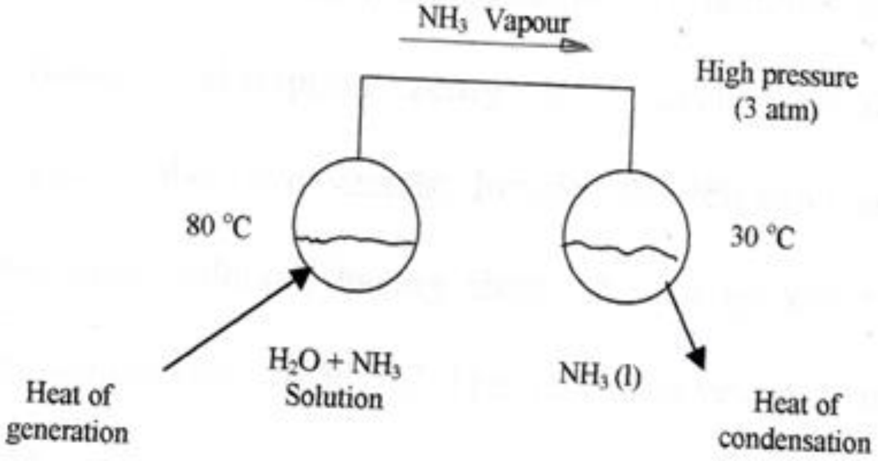


fig (3.3-b) Thermal Driven Refrigerator (Generation Stage)

During the generation process, heat is supplied at high temperature to the left-hand vessel, and the heat of condensation of ammonia is rejected to the surrounding at ambient temperature from the right-hand vessel. During refrigeration, the right-hand vessel absorbs heat from its refrigeration load at a low temperature, and the heat of absorption of the ammonia into the solution in the left-hand vessel is rejected to the surrounding at ambient temperature. The other way is to convert solar radiation into heat in a thermal solar collector and use the heat to drive an absorption/adsorption refrigerator.

.

Refrigerator

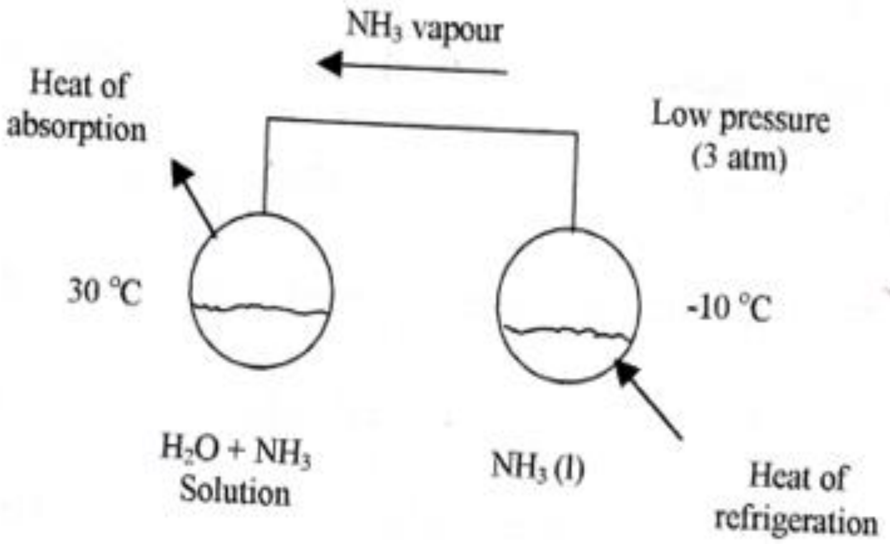


fig (3.3-c) **Thermal Driven Refrigerator (Refrigerator Stage)**

Efforts are being made to improve the efficiencies and to bring down costs, but there is little to choose between the various types, and it is still difficult to see which, if any, will ultimately secure appreciable market. This research is a humble trial in the line of finding out one simple technology using available resources in refrigeration field with utilization of solar energy. In view of the relatively large outlay required initially, the installation of a solar refrigerator and of the better technical efficiency obtainable with larger units, it is likely that village-size refrigeration systems would be more practicable and more economic than small domestic units. The Asian Institute of Technology has investigated the possibility of developing a village-size refrigerator with a solar collecting area of (20 - 25 m^2),. that provide 0.5 m^3 of food storage cooled below 10°C, or make about l00 kg of ice per day. An arrangement designed to overcome or use solar energy from a flat-plate collector to supply the heat of generation is sketched schematically in the figure (3.4). The vessels, pipes and valves are all made of steel; copper and brass are attacked by the ammonia-water solution and cannot be used. [Intermittent Adsorptive Solar Cooling System, , 1989.]

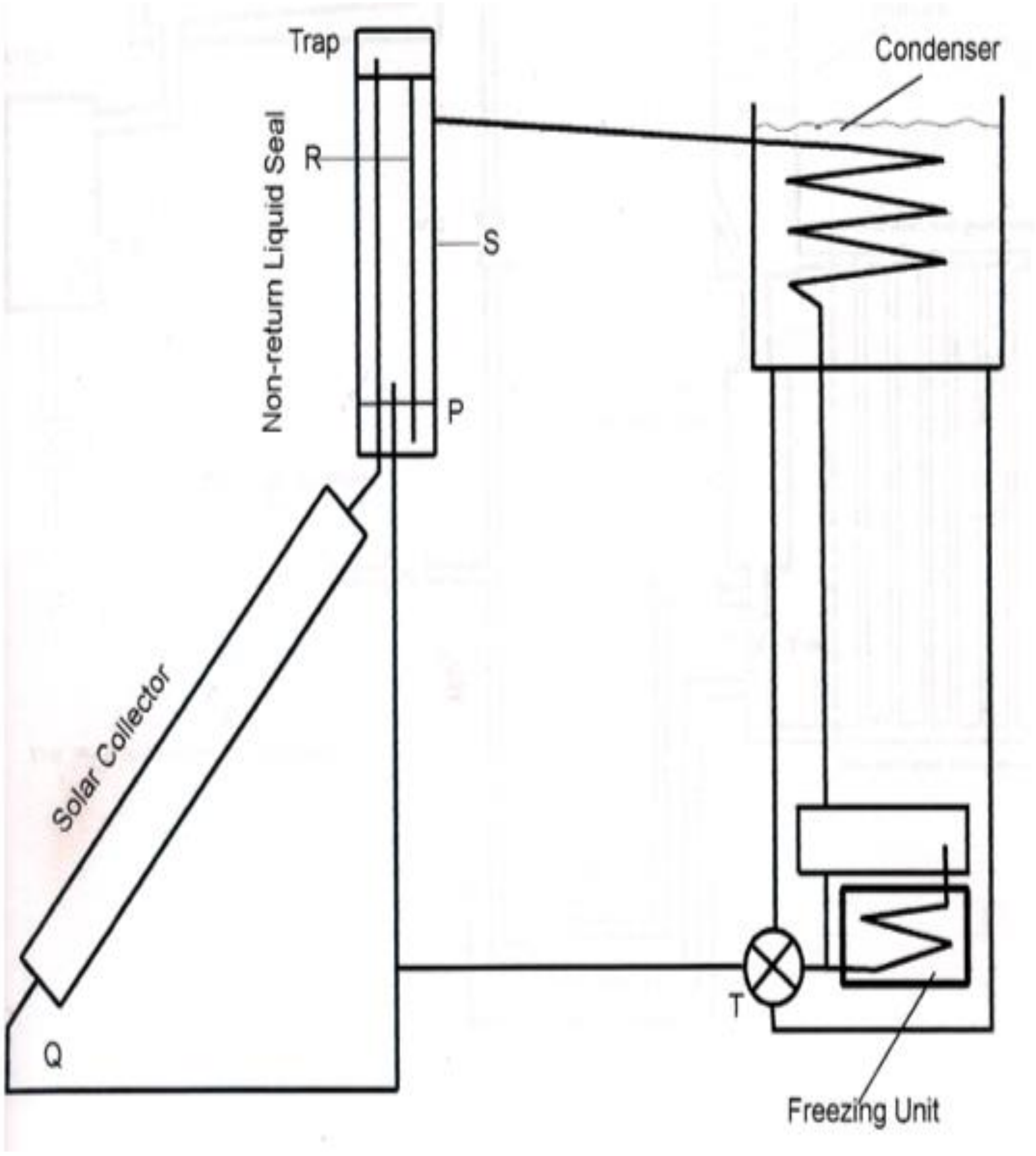


Fig (3.4) Ammonia Solar Refrigerator [Ken-ichiKiniura, Solar Adsorption]

The ammonia-water solution contained in the solar collector, and tall air-cooled rectifier pipe serves to remove water from the generated vapor. The vapor is condensed in a condenser cooled by a tank of water, and the liquid ammonia is collected in the receiver. An evaporator coil provides a heat transfer surface between the liquid ammonia and the refrigeration load, which may be a quantity of water for making ice. There is a drain, closed by a valve C, from the evaporator to the solar collectors that if unwanted water accumulates inside the evaporator it can be returned to the solar collector from time to time. During the refrigeration process the ammonia vapor is directed to the bottom of the solar collector to ensure good mixing and uniform re-absorption throughout the solution. The intermittent ammonia-water absorption cycle has been used because the thermodynamic properties of this combination are well-known, and the technology is reliable. The type of solar collector employed was determined by the fact that, in the tropics the proportion of diffuse solar radiation in the total insulation is usually high. A flat plate collector was therefore used in preference to a concentrator, which would require accurate orientation and tracking. It seems, however, that the use of flat auxiliary mirrors to enhance the heating by direct solar radiation might be worthwhile. A diagram of Chinana's unit is shown in fig. (3.5), it was constructed of welded steel pipe. The generating system consists of a tube-in-sheet collector 1.52 m by 1.07 m painted black with three glass covers and containing 9 kg of ammonia-water solution of concentration 0.46 ammonia by weight. Ammonia could be distilled from the solution. For the refrigeration process, water-cooled absorber was used, and a minimum temperature of -12℃ was obtainable. It was concluded that about 1.4 kg of ice could be produced per square meter of collector area on a clear day, and this performance could be considerably improved by a selective radiation coating on the collector surface. [M. Sayigh., 2015, PP (203 -223)] Swartman and Swarminathan [(paper," Revue Physics" Feb. 1986).] .Another solar powered intermittent ammonia-water absorption refrigerator similar to that tested by Chinnapa. fig.(3.6) and differed from that of Chirmapa in two respects Firstly , instead of using a boiler and a solution reservoir outside the collector , they used a large header of diameter 152 mm, which provided the required capacity for the solution a large surface for the evaporation process was introduced and the ammonia vapor from the evaporator into a tube connecting the lower and upper headers of the collector and the glass cover of the collector was a solution reservoir outside the collector, they used a large header of diameter 152 mm, which provided the required capacity for the solution a large surface for the evaporation process was introduced and the ammonia vapor from the evaporator into a tube connecting the lower and upper headers of the collector and the glass cover of the collector was opened to facilitate cooling. Although 1.9 kg of ammonia could be distilled per square meter of the collector area, and refrigeration temperatures as low as -12℃, could be reached, the rate of refrigeration was too slow because of slow re-absorption of ammonia vapor into the solution They proposed using the collector to dissipate the heat of absorption to overcome this problem. A theoretical analysis of the heat transfers and flow characteristics of tube in-sheet

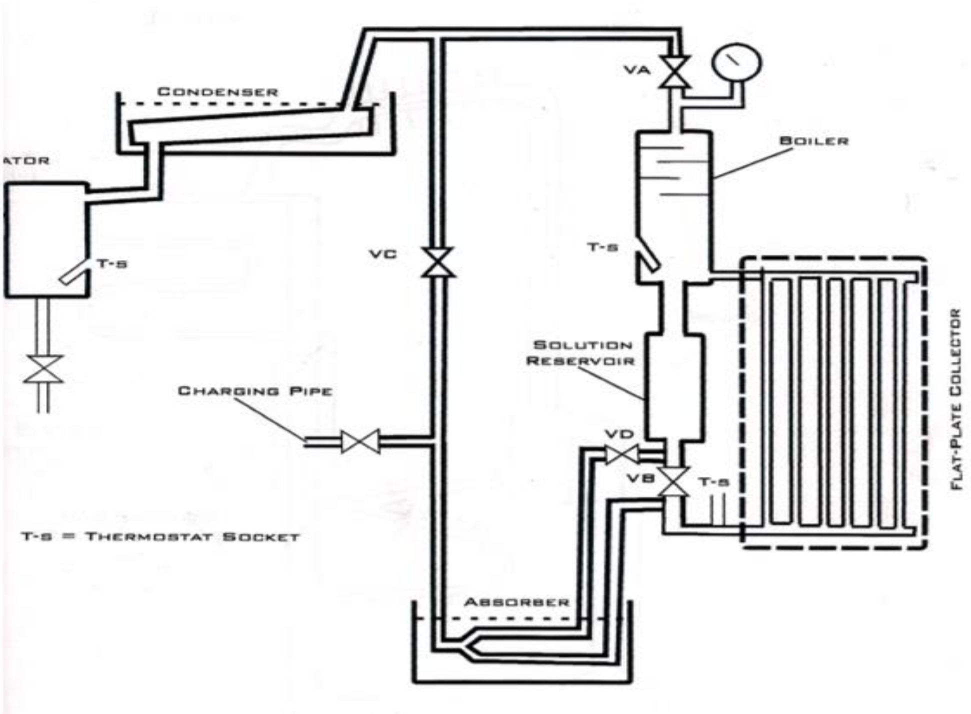


Fig (3.5) Ammonia Solar Refrigerator [Hu 3mg and R. H. B. Excel]

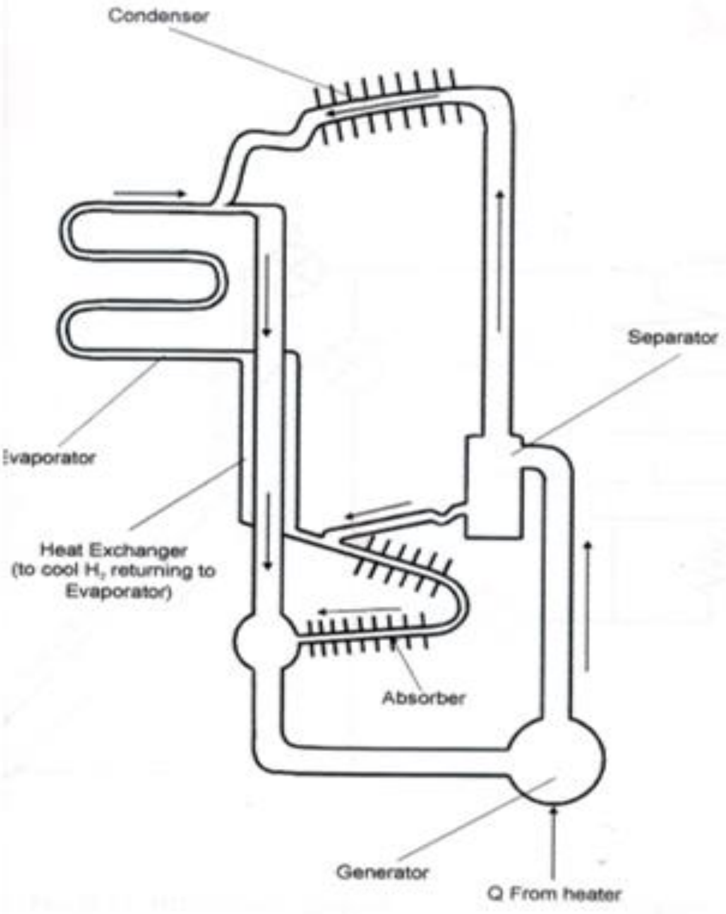


Fig (3.6) Ammonia Solar Refrigerator [Hu 3mg and R. H. B. Excel] flat-plate collectors used generators in solar powered ammonia-water refrigerators was made by Moore and Farber [(paper, “Revue Physics” Feb. 1986).].

The work culminated in a collector 1.49 m^2 in area consisting of 25mm diameter steel pipes running from a 32 mm diameter feeder to a 63.5 mm diameter header. The centers of the 25 mm pipes were spaced 102 mm apart and were soldered to the galvanized iron sheet of thickness 1.0 mm. A single cover was used. [Web Sites]

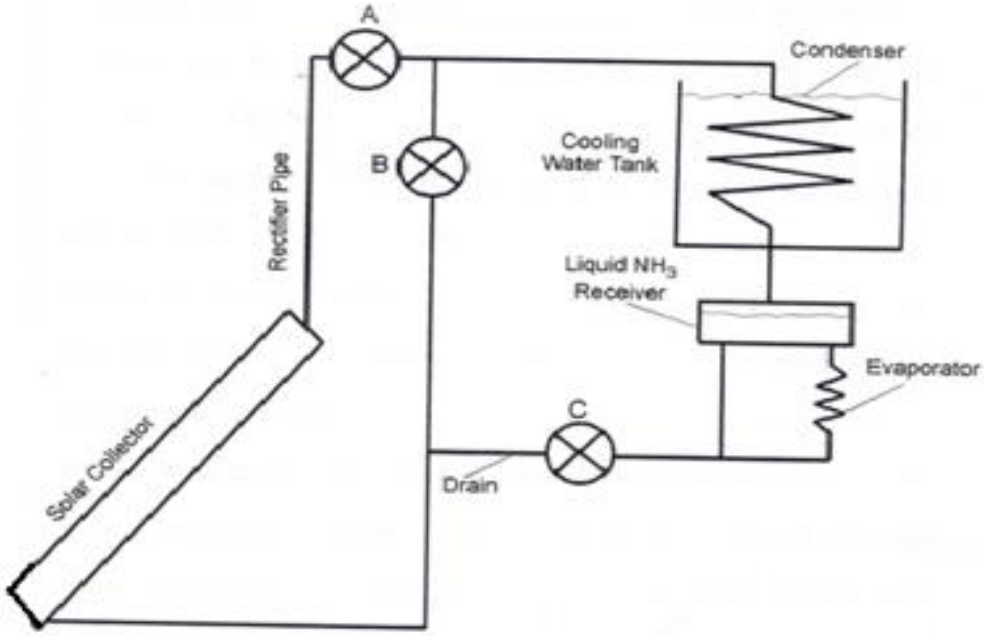


Fig (3.7) Ammonia/ water Solar Refrigerator [Hu 3mg and R. H. B. Excel]

3.9- Solar Adsorption Refrigerator:

Solar refrigeration on the bases of the solar thermal energy is the simplest among other refrigeration techniques The only problem is how to obtain a reasonable, simple design with good selection of materials and well-calculated amount of both; working fluid and absorbing material. Solid adsorption cycles are well adapted for solar cooling, but some more important precautions to be taken are:

- Absolute tightness: This is an important technological difficulty for those systems.
- Good Sizing of the Components: (Heat Exchanger Area, Pipes Diameter, mass of Adsorbent...).
- Development of valves. Check valve is needed between the condenser and evaporator (except in the case of icemaker), thermal valve to open and close ventilation shutters.

Some important points concerning adsorption solar refrigeration attained through various researches can be summarized as:

- Adsorption solar refrigerators can work without auxiliary energy.
- Active carbon-methanol pair yields higher efficiencies when night temperature is low
- With Zeolite-water pair, the refrigeration storage is performed through ice production inside the evaporator.
- With the active carbon-methanol pair, the ice is produced outside the evaporator.
- Active carbon-methanol pair can be used in competition with Zeolite-water pair for cold stores and can be used moreover force making. It is adapted for small units, up to middle size units (up to 1000 kg of ice per day). It seems particularly well suited for hot and arid climates where the Nocturnal temperature is low due to nocturnal radiation.

- **The Pair Selection:**

A pair consists of a solid adsorbent and a refrigerant fluid. Amongst the solid adsorbents only micro-porous adsorbents are considered, since the macro-porous adsorbents cannot provide the high temperature lifts necessary for cooling. A lot of micro-porous adsorbents ,some of them are Zeolite, activated carbon, silica gels and activated alumnae. The refrigerant fluids, which can be associated in priority, are those, which have small polar molecules since they have high latent heat, some of these, are: water, alcohol, and ammonia. The possible combinations lead to a large variety of possible pairs but two more conditions to be considered are:

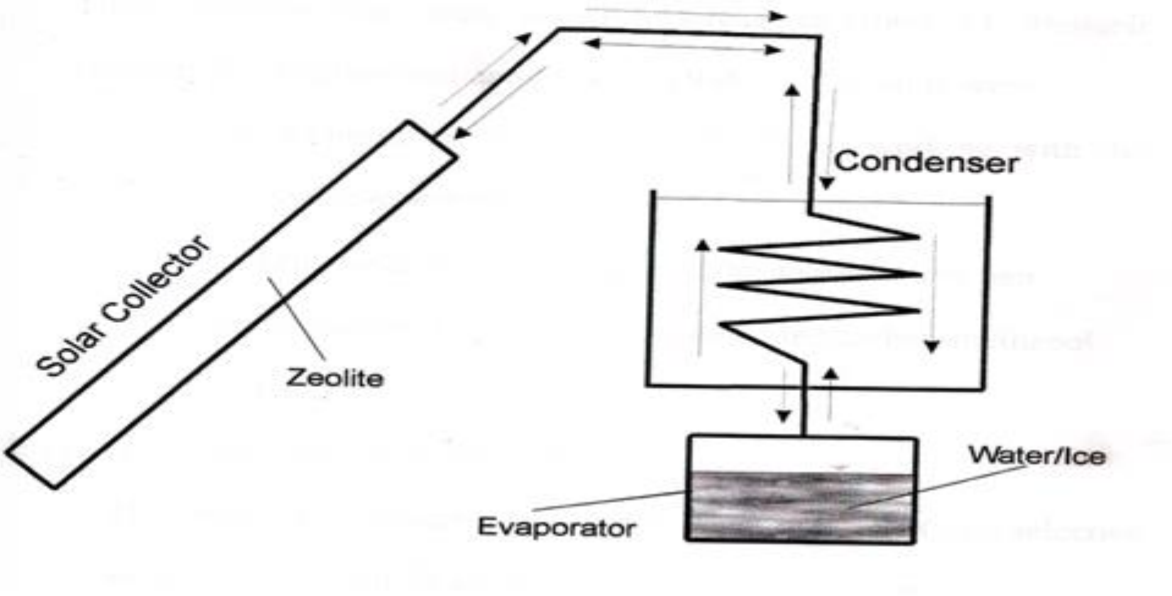
- Chemical stability.
- Large adsorptive capacity for cycling.

The main pairs, which have been proposed by various authors, are Zeolite- water, Zeolite-methanol, active carbon-methanol; silica gel-water, active carbon-R12.The pair Zeolite 13X-CH_3OH has been tested, but is not chemically stable. It would probably be possible to find another Zeolite, which would not have a catalytic reaction with methanol. It seems that silicalite which is a zeolite like structure adsorbs alcohol without catalytic reaction. But the adsorptive capacity of those silicates is much less than that of the Zeolite so that their use for thermodynamic applications is not obvious. The other pairs are claimed to be chemically stable. These pairs are equally suited for all purposes since they differ by their temperature rise. is the temperature difference between temperature of condensation/adsorption and the evaporating temperature.

To illustrate this point ,the pairs efficiencies are compared using second law .Defined as the ratio between the thermal COP and the ideal Carnot COP for the same operating temperature. The results are in table (3.1) & Fig(3.8)

Table(3.1) Temperature and efficiencies for
Various Refrigeration pairs [Grenier and M. Pos, Experimental]

Adsorbent	Fluid	temperature	2nd law thermodynamics	Thermal COP	Regeneration
AC-DEG	CH_3OH	15 ºC	0.48	0.54	34 ºC
AC-35	CH_3OH	15-30 ºC	0.36	0.49-0.43	38-76 ºC
AC-PKST	CH_3OH	25 ºC	0.29	0.38	66 ºC
Z-13X	H_2O	65 ºC	0.26	0.33	210 ºC
Z 4A	H_2O	75-90 ºC	0.22	0.26-0.28	265-325 ºC



Fig(3.8) Water Solar Adsorption Refrigerator

The advantage of activated carbon-methanol pairs for solar powered cycles are their optimum corresponds at low generating temperature compatible with flat plate collectors. On the opposite, the optimum for the Zeolite-water pair corresponds to high temperature and high regenerating temperatures. The development of that technology lies upon the control of two components the collector and the evaporator

- **The Domestic Refrigerator:**

The basic unit consists of a 0.8m^2 solar collector (with selective surface) filled with 23 kg Zeolite. This unit has been tested with various operating conditions (water cooled or air-cooled; with and without mirror boosters). Some results of the overall solar COP presented in the following table (3.2)

Table(3.2) The effect of cooling form of the Condenser on Cop

[Grenier and M. Pos, Experimental]

Conditions	Solar flux KJ/m^2	T_{coned} C	T_{ev} ºC	T_{max} ºC	T_{min} ºC	Cop
W/cooled condenser	27310	31	0.0	104	29	0.10
W/cooled condenser	17980	25	0.0	81	24	0.14
Air- cooled condenser	26040	31- 39	0.0	109	29	0.10
E-w Booster	288880	31	0.0	134	20	0.14
N-S Booster	18020	25	0.0	97	30	0.17

- **The 12m^3 Cold Store (Zeolite-Water)**

The unit is tested with water-cooled condenser. In spite of the number of elementary adsorbent solar reactor (24 units; 0.83m^2 each, total area 19.92m^2) and a length of the pipes (the pipes between the evaporator and some reactors reach 8-m long), the outcome results are summarized in table (3.3):

Table (3-3) The Effect of solar flux on COP [Grenier and M. Pos, Experimental]

S .F(KJ/ m2)	$T_{cond.}$ ºC	T_{ev} ºC	T_{max} ºC	T_{min} ºC	COP
23100	32	1.0	137	15	0.096
24800	35	-2.5	132-144	25	0.086
17800	26	-0.6	93-103	13	0.105
25300	33	-1.0	124-145	28	0.095

- **Charcoal/Activated Carbon:**

The machine consists of three main parts: Flat plate collector, condenser & evaporator .Flat plate thermal collector is to collect the thermal solar power energy to heat up some amount of charcoal particles in a lower box. The charcoal is to act as an adsorption material for the working fluid (methanol alcohol). This unit works with an air-cooled condenser, with no valve between the evaporator. The condensation starts in the evaporator, and generates heat, which facilitates the removal of the ice from the evaporator box. When the temperature of the evaporator reaches ambient temperature, the condensation occurs normally at the condenser. The result of the test is summarized in table (3.4):

Table (3.4) The Effect test of solar Flux on the Machine's productivity

[Grenieand M. Pos, Experimental]

S. F (KJ/ m^2)	$T_{cond.}$°C	T_{ev} °C	T_{max} °C	T_{min} °C	COP	Ice Prod. (kg-m^2)
19500	20-20	-3.0	79-92	19-26	0.13	6.0
18500	25-35	-3.0	77-91	24-30	0.10	4.7

Among many criteria studied, most important are:

a- Optimal thickness of the Zeolite for best COP

b- The orientation: Performance is not sensitive to the azimuth orientation.

c- Solar irradiation:

$Es > 5000\ kJ/m^2 \rightarrow COP_s = (0.625\ Es/2500 + 0.375)\ COP_o$

$Es < 5000 kJ/m^2 \quad COP_s = 0$ Where;

Es is the input solar energy (kJ/m^2) on the plane solar collector. Cop_0 is the COP corresponding to 25000 kJ/m^2 irradiation.

d- The active carbon pair: It is well adapted for ice making application but the problem of nocturnal ventilation is much critical in using A/C pair than it when using Zeolite-water pair.

- **- Various Components**

The main components of the solar adsorption refrigerator can generally be described as follows:

a) – Collecting Plate

The originality of the solar collector process these are:

- During the day, it collects solar energy and serves as the heat exchanger between the absorber and the collector.
- During the night, radiator serves as a heat exchanger between the absorber and the radiator. The fact that the solar collector is a multi-functions component requires a careful sizing of the mass of absorber as well as of the area of heat exchanger. If the sizing is not carefully performed, one of the functions may be miss-fitted.

b)- The Evaporator

It provides the cooling effect, the evaporator together with the ice moulds form an insulated box to ensure efficient heat exchange between the working fluid and the water in the moulds. In the intermittent cycle, the cold production occurs during the night only. It is important to limit the heat transfer between the evaporator and the store so as to limit the cooling effect of the evaporator. Refrigeration storage between the evaporator and cold space is needed. For good sizing of heat exchange surface, the volume of the refrigeration storage must be enough to maintain the store at the required temperature during several sunny days..

c) – The Condenser

It is an opened box containing water and a coil of tube carrying the working fluid (vapor). It condenses the vapor of the working fluid back to liquid The problem of sizing the condenser is a well-known problem of natural convection condenser. If the condenser is an air-cooled built-in on the back of the collector plate, the solar collector serves in an additional capacity and the sizing of that component becomes much more difficult since, thermal losses due to heat transfer between solar collector and the condenser is to be considered.

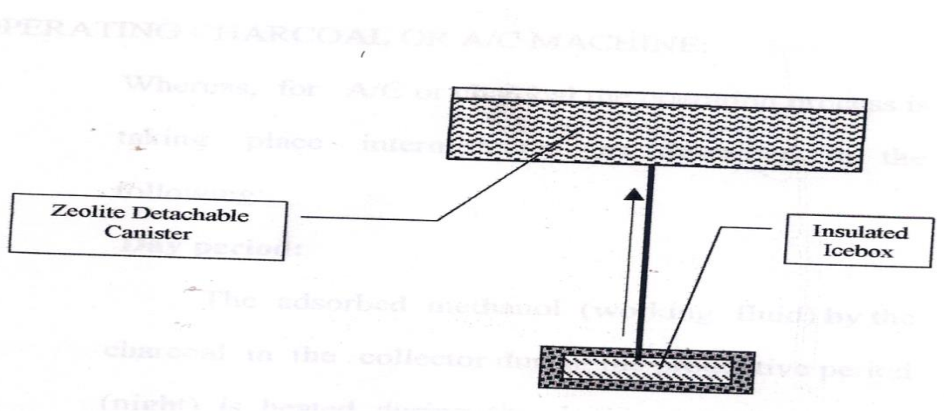


Fig 3.9 Schematic Drawing of Zeolite Solar Freezer

- **- Operating Charcoal Machine**

Charcoal Machine (A/C) the operation process is taking place intermittently in the day period and the heating process of the alcohol will evaporate and hence the pressure in the whole system will increase to the condensation pressure and starts to condense back to liquid collected in the evaporator box which will act now in reverse manner as a condenser. The ice moulds are full with water attached to the evaporator lower surface and the whole combined unit (evaporator and the ice moulds) is well insulated from the outside environment. The working fluid will start evaporating according to the heat receives from the water in the moulds. The cold charcoal now has high adsorptive and will soon absorb the generated vapor from the evaporating effect and keeps the lower pressure of the system and hence the continuity of the evaporating process which in turn generates the cooling effect on the water in the moulds throughout this period. (Zeolite, charcoal). [paper, "Revue Physics" Feb. 1986).]

The main advantages of solid adsorption over solid absorption cycles or liquid absorption cycles are:

1. The solid adsorption, suffer no volume alteration during the cycles (which is not the case with liquid absorption cycles).
2. There is no need for vapor rectification (to prevent the gradual accumulation of water) as in the case of liquid absorption in the ammonia-water cycle. The two points mentioned above strengthen the idea of using solid adsorbent in solar cooling application, the option adopted in this project.

3.10 System Description:

The proposed unit consists of three main parts described as follows:

- **Generator/absorber**:

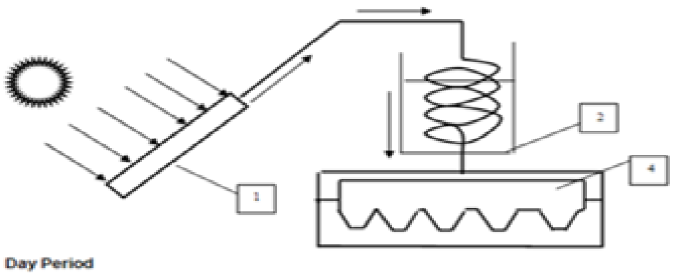
This part is in a form of a flat-plate collector, its purpose is to absorb the solar thermal power from the sun to heat up charcoal particles packed in a box attached to the lower part of the collector-plate, the heat will generate refrigerant vapor (methanol) which will then be condensed to liquid in the condenser and collected for use in the productive time later.

- **Condenser:**

It condenses the generated methanol vapor coming from the collector using immerged coil in an open container having water.

- **Receiver/evaporator unit:**

This part works as a storage tank for the condensed methanol liquid during the day and as an evaporator during the night (productive time) where the flow of the refrigerant will reverse. There will be an ice mould box attached to this unit at the bottom, water filled in will act as the load, ice produced there is the product.



1- Collector
2- Condenser
3- Ice mould

fig (3.10 a) Schematic Drawing of Solar Refrigerator(Day Period)
[Excel, Simulation and Analysis of an intermittent Solar]

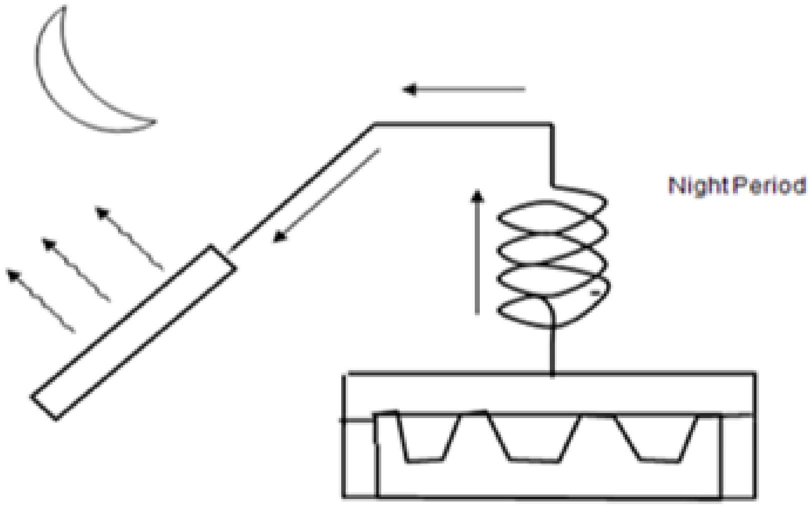


fig (3.10 b) Schematic Drawing of Solar Refrigerator (Night Period)
[Excel, Simulation and Analysis of an intermittent Solar]

CHAPTER FOUR

DESIGN, RESULTS

&

DISCUSSION

CHAPTER IV

DESIGN, RESULTS & DISCUSSION

4.1- **Introduction**

Generally, two design approaches can be considered for the design of namely:

a- **Bottom-up** Approach: Where the starting point would be of the required amount of the product, i.e. to say for example, (10 Kg) of ice is required daily as the target amount of product.

b- **Top-down approach**:

Where the starting point would be the area of the collector, to say for example, the area of the collector is ($5m^2$).To start with a reasonable area of the collector is much better especially if the cost factor is considered, top-down approach method would be followed for the purpose of this design. total useful area of ($1\ m^2$) for the collector is to be taken.

- **4.1 - Design:**

The following factors are to be considered for the design

- Simplicity of operation and manufacturing technology.
- Availability of materials locally.
- Low manufacturing cost.
- Safe use.

4.2 - Material selection:

a. - Collector plate:

A steel plate (4mm thick), (Net useful area of $1\ m^2$), with black paint on the outer face. The lower side is to be prepared with fins on it to accelerate the heat transfer rate to the charcoal.

b. Plain glass to act as the upper cover for the plate. (4 mm).

c. - Wire net (fine mesh):

To hold the charcoal particles against the collector plate and allows diffusion of the methanol vapor into the charcoal. This has another importance of constraining the charcoal particles coming out during the evacuating process.

d. - Thin steel sheet (1 mm):

is to be used for the lower case of the collector (insulator casing).

e - Charcoal particles

This is the adsorbing material for the methanol vapor. Charcoal was packed in a sealed welded box at the back surface of the collector plate. The type of charcoal used is the available are in the local market, it is a mixture of various types of carbonized hard woods. The total mass required is about 19.25 kg.(Calculated)

f - Galvanized pipe (1/4 in dia):

for the entire piping system joining the three main parts, the collector, the condenser and the evaporator.

g - Insulation layer:

Cotton is used as thermal insulator on the lower part of the collector and also the evaporator box. The insulator of the collector was packed in shutters form cases and can be opened and closed at required times.

h - Methanol:

For best results, laboratory grade of methanol is used. The required charging amount is about 7.5 litters, as for our design below.

4.2 - Assumptions and design:

- Maximum temperature of the plate =90℃.
- Mean ambient temperature = 30℃.
- Stability of state for both condensation and evaporation processes.
- Steady State Conditions for each process.
- Average Temperatures are calculated to specify each point.
- Heating Process is divided into two stages (ab) and (be), the intermediate point is just approximated in thermodynamics Cycle
- Also for the cooling processes is divided into (cd) and (da), point(d) is the intermediate point chosen in thermodynamics Cycle
- Properties of methanol are taken from thermodynamic tables
- Empirical Equations are used in finding out heat-transfer
- Radiation and some minor heat losses are ignored as a first approximation.
- Let the mass of steel in the welded fins m_{sc}= (33.0 kg)

- Let the mass specific heat capacity of steel C_s = 0.46 kJ/Kg K
- Let the mass of charcoal in the fins' m_{CHC} = 19.25 kg
- Let the specific heat capacity of charcoal C_{CH} = 0.709
- **Loads Calculations**

a. **Heating Loads** (**Collector Day Period**)

- Sensible heating load to raise the temperature of the plate from the
- Ambient temperature to the designed temperature.
- Sensible heat required to heat up the absorbed methanol to the evaporation temperature corresponds to the designed pressure.
- Latent Heat: Required to evaporate the methanol at the given pressure.
- Sensible Heat required to heat up the mass of the charcoal under
- The collector plate to the desorption temperature.

b. Cooling Load (Col1ector Night:

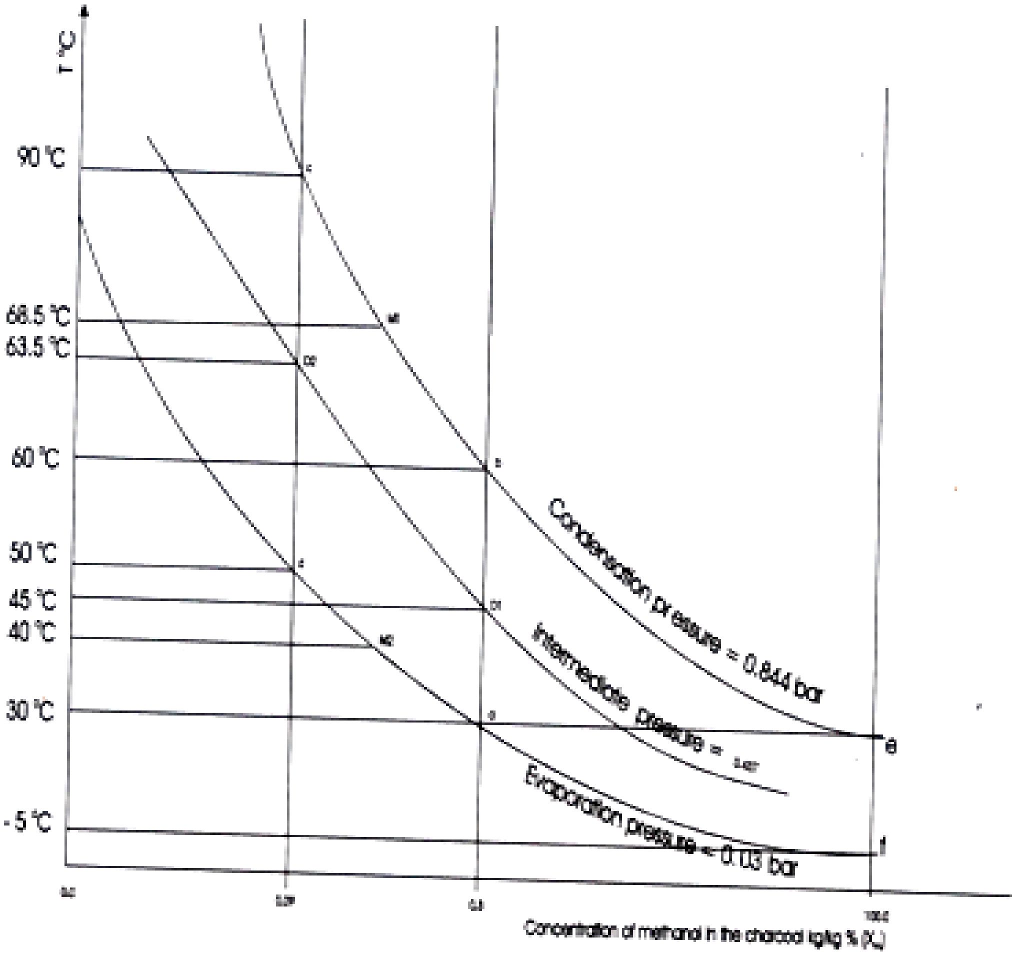
Heat dissipated from the collector to the surrounding to increase the absorptive of the charcoal particles to methanol vapor (For time modeling, this is to be checked against the available time).

4.5 Thermodynamics Cycle:

Although the thermodynamic states in the various processes are not stable, but for the purpose of the design work some assumptions are taken to simplify the work as follows:

- a - b: (pressurizing Process) Heating of the methanol in the collector (charcoal) the concentration remain constant X_m only pressurize methanol is released and the pressure increase.
- b - c: Heating of the methanol in the collector (charcoal). During this process, which takes place at constant pressure, methanol evaporates from the charcoal reducing its concentration, condenses at ambient temperature (point e), and accumulates in the receiver (evaporator).
- c - d: Cooling of the charcoal (the collector), during which the concentration of methanol in the charcoal X remains constant and the pressure P decreases. Only small amount of methanol vapor is absorbed to reduce the pressure of the system.
- d - a: Evaporation of methanol from the evaporator (point f). During this process, which takes place at constant pressure, the methanol vapor is re-adsorbed

into the charcoal increasing its concentration. This is the productive process of the system. (Fig. 4.1)



Fig(4.1) thermodynamics Cycle the solar adsorption

on CH_3CH/ CharcoalT- X_mDiagram

The heating process (I) and (2) are carried out during the day under the sun. The cooling process (3) and (4) are carried out at night with the back insulation on the collector removed to accelerate the cooling of the system. Taking ambient temperature = 30 °C and the required refrigeration temperature 5.0°C. Point (f) corresponds to saturation (evaporator) pressure of 0.03bar and the concentration X_m= 1. Point (a) is the intersection of the 0.03bar pressure (evaporation pressure) line and the 30 °C isotherm. Point (e) is fixed by condensing pressure line (0.844 bar) at a Temperature = 30 °C(aassumed ambient temperature) and the concentration(X =

1.0). Point (b) now is the intersection of 0.844 bar isobar line (T_s =60 °C) and the line X_m 0.3 (concentration of the methanol in the charcoal) from point (a).The position of point (c) on the 0.844 bar line depends on the amount of solar heating that occurs on any particular day, and on the amount of methanol contained in the charcoal per unit of the collecting surface. Point (c) is therefore, a floating point according to the heating radiation available (90 °C was assumed).Assuming T_c90°c(superheating situation) and X_c= X_d= 0.09(methanol in charcoal kg/kg as assumed by Merghni [Solar Refrigeration in Sudan University of Wales, 1989]. .Finally point (d) is the intersection of the line X_m from point (c) down to0.03 bar line. Assume T_d = 50 °C, for the design purpose. It is convenient to have an expression for the mass of the liquidCH_3OH m_l in the solar collector, and the mass of methanol vapour m_v thathas been distilled from the charcoal in terms of the liquid concentrationX_M at any point. A complete theory must take into account the small amount of water in the vapor. However, to simplify the calculations

$M_L = m_l(a)\,(1 - X_M(a)) / (1 - X_M)$, &

$m_v = m_L(a)\,(X_M(a) - X_M)$ I $(1 - X_M)$. where

M_L = Total mass of the liquid methanol in the collector.

m_l = instantaneous mass of liquid methanol in collector at point (a)

(Almost for the entire points methanol in the charcoal state is found to be superheated vapor)

$X_M(a)$ = concentration of methanol at point (a)

X_M = initial concentration of methanol in the collector

m_v = mass of methanol vapor in the collector

It should be noted that, the thermodynamic system used, is a binary materials system (has two chemical components), methanol and charcoal, and two phases, liquid and vapor. Therefore, by the Gibbs phase rule, there are two degrees of freedom. The two independent variables that can be used to specify the thermodynamic state of the system are:

X_M, the mass fraction of methanol in the charcoal (dimensionless) and

T, the temperature (K)

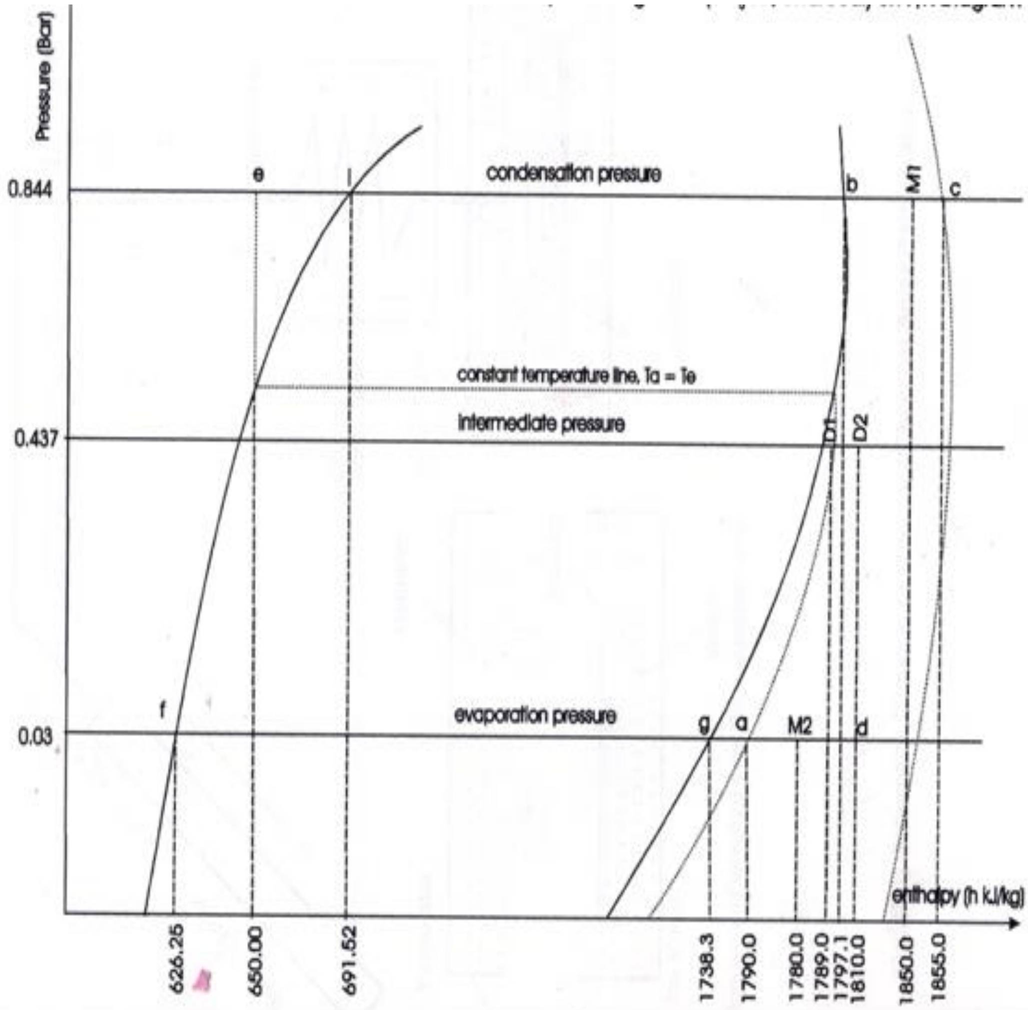
All other properties of the system are functions of X_M and T, for example:

P, the pressure kPa),

X_v, the mass fraction of methanol vapour (dimensionless)

v, the mass specific volume of the liquid methanol (L/Kg),

H_l, h_v the mass specific enthalpieS of the liquid and vapour (kJ/kg), and S_l, S_v the mass specific entropies of the liquid and vapor (Methanol) (kJ/kg K) respectively. The cooling water of the condenser controls the upper pressure. The pressure will keep increasing (during the heating process of the collector in day period) but when the pressure reaches the condensation pressure of methanol at corresponding temperature of the cooling water, it will stop increasing and condensation of methanol will continue.



(fig 4.2) thermodynamics Cycle the solar adsorption onCH_3CH/ Charcoal P/h Diagram

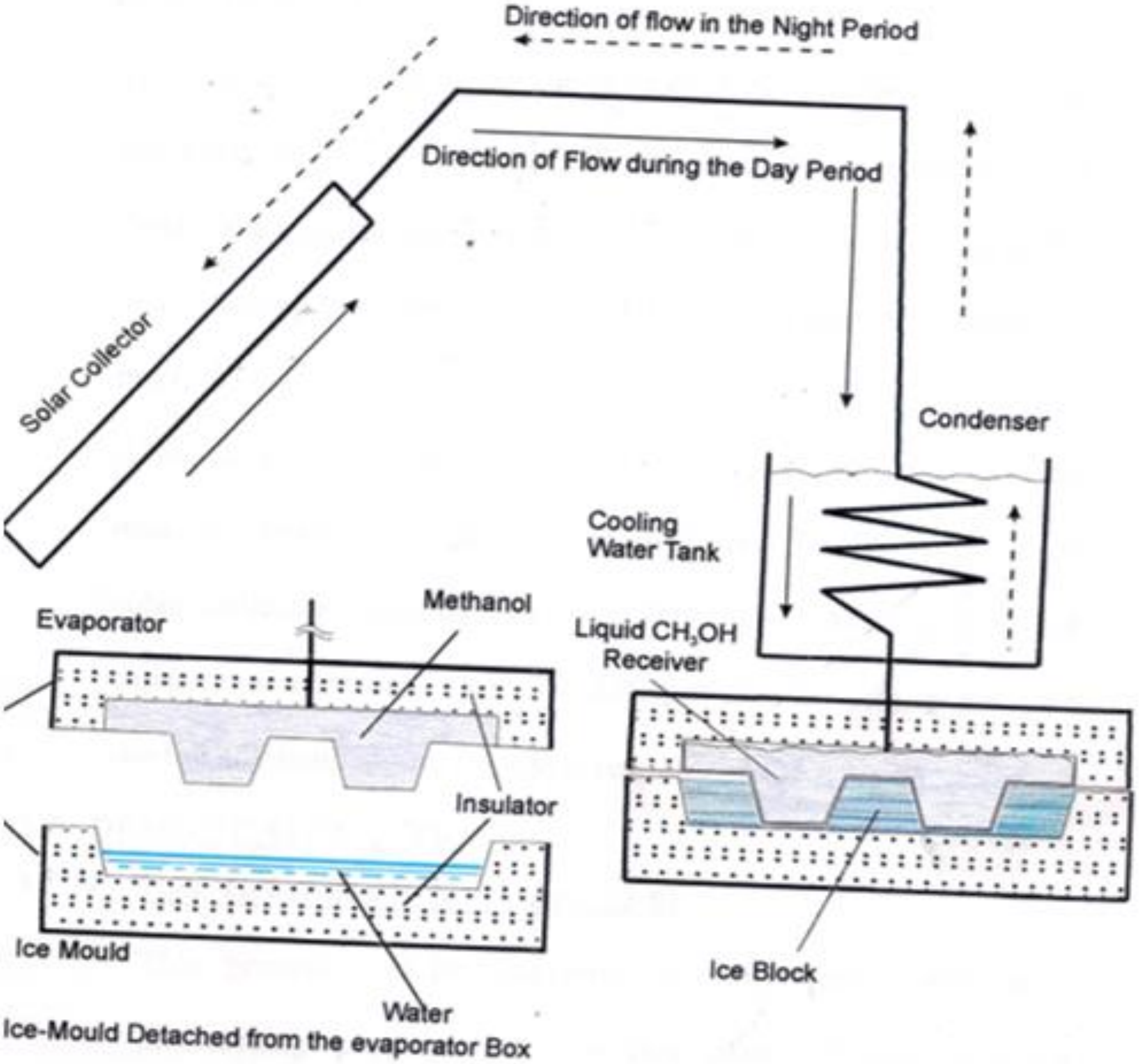


fig (4.3) Schematic Drawing of Methanol/ Charcoal Solar Refrigerator
[Hu 3mg and R. H. B. Excel]

4.6 **HEAT TRANSFER PROCESSES:**

$m_{sc} =$ (33.0 kg) denote the mass of steel in the welded fins and the black plate of the solar collector, and C_s (= 0.46 kJ/Kg K) denote the specific heat capacity of steel. Then the differential of the heat entering the steel collector plate in any temperature change is$m_{sc}C_s\,dT$. Also, let m_{CHC} = (19.25 kg) denote the mass of charcoal in the fins' vacancies and the black plate of the solar collector, and C_{CH} (= 0.709) denote the specific heat capacity of charcoal. Then the differential of the heat entering the steel in any temperature change is ($m_{CHC}C_{CH}dT$) kJ.

4.7 DESIGN CALCULATIONS:

- **Heat Generated in the Process (a-b):**

This process can be considered as partly heating and partly pressurizing process. The collector plate, charcoal and the methanol in the charcoal particles of the collector, all these are to be heated from the ambient temperature T_a to evaporation temperature (T_b). Additional amount of heat is required for generating pressurizing vapor of methanol that will raise the pressure of the system from evaporation pressure (0.03 bar) to

$$Q_{ab}=\int_a^b(m_{sc}C_{CH}+m_{CHC}C_{CH})\,dT+m_{a-b}\,dh_{a-b}+mdh_{D1-b})\ldots.. \qquad (4.1)$$

Condensation pressure (0.844 bar). This can be calculated as follows: for practical calculations the integral may be approximated by:

$$Q_{ab} = (Q_{(a-b)\,1}) + (Q_{(a-b)\,2}) + (Q_{(a-b)\,3}) \qquad (4.2)$$

Where:

$Q_{(a-b)\,1}$= sensible heat required for the collector plate and the charcoal (KJ)

$Q_{(a-b)\,2}$= sensible heat required for the liquid methanol (KJ)

$Q_{(a-b)\,3}$= heat of desorption required for pressurizing (KJ)

The values of m_{sc}, m_{CHC} and m_l(a) depend on the design of the system. In accordance with this machine;

(collector plate + fins)

Calculated as follows:

$$m_{sc} = 7833[(1.02 \text{ x } 0.004 \text{ x } 1.02) + (0.004 \text{ x } 0.002\text{X}0.9 \text{ x } 10)] = 33.16 \text{ Kg}$$

$$m_{CHC} = 19.25 \text{ kg (assumption)}$$

This was calculated as follows: Basic principles are:

(1) Charcoal can absorb 30% of its weight at room temperature [Zed Books Ltd. 2013].

(2) It can adsorbs up to 30% of the absorbed amount in normal collector day temperature.

(3) density of charcoal is 0.709 Kg/m^3

(4) Allowing 20% increase in volume for crushed charcoal; density of Charcoal is 0.567 Kg/m^3

Based on above, calculated mass of charcoal for the circulating methanol is 14 Kg. Additional charcoal for the pressurizing methanol is 5.2324 Kg.

Thus total mass of charcoal is:

$$14 + 5.2324 = 19.25 \text{ Kg.}$$

c_s=0.46kJ/kgK

C_{CH} = 0.709 kJ /kg k

For evaluating the various thermodynamic properties of methanol, basic principles can be used as follows:

$$R = \frac{\tilde{R}}{\tilde{m}} \text{kJ/kgK} \ldots\ldots\ldots \quad (4.3)$$

$$R_{math} = \frac{8.3145}{32.04} = 0.2595 \text{ kJ/kg K}$$

Where: R = specific gas constant,

$\tilde{R}$= Universal gas constant = 8.3145 kJ/kmol

$\tilde{m}$= Molar mass, for methanol $\tilde{m}$= 32.04 kJ/kmol

$$h = u+RT, \ u=C_\gamma T„ \qquad C_\gamma = C_p - R,$$

$$C_{p,1,\text{meth}} = 65.44\text{kJ/kmol}, \quad C_{\gamma,\text{g,meth}} = 3.2195 \text{ kJ /kmol},$$

$$C_{p,\text{g,meth}} = 111 .46 \text{ kJ/kmol}, \quad C_{\gamma,\text{l,meth}} = 1 .7825\text{kJ / kmol},$$

$$C_{pg} = 111.46/32.04 = 3.479 \text{ kJ/kg}, \quad C_{\gamma g} = 3.479 - 0.2595 = 3.22 \text{ kJ/kg}$$

$$C_{pi} = 56.44/32.04 = 1.761 \text{ Id/kg}, C_{\gamma i} = 1.761 - 0.2595 = 1.502 \text{ kJ/kg}$$

Properties are either calculated analytically from the equations above or taken from thermodynamic tables [Thermal Process, John Wiley (1980).] At point (a); T_a = (30 °C, super heated methanol vapour in charcoal)

T (temperature) = 30 °C

P(pressure) = 0.03 bar

h_a (enthalpy of methanol vapour) = 1790 kJ /kg

At point (b); T_b = 60 °C, (methanol vapour is superheated)

P (pressure) = 0.844 bar

h_b (enthalpy of methanol vapour) = 1797.1 kJ/kg

At point (D1); (methanol is in superheated vapor state)

T (temperature) = 45 °C

h_{D1} (enthalpy of methanol vapor) = 1789 kJ /kg

$$Q_{ab} = Q_{(a-b)\,1} + Q_{(a-b)\,2} + Q_{(a-b)\,3} \quad (4.4)$$

$Q_{(a-b)\,1}$ = Heat the temperature of the steel plate and the charcoal from the ambient temperature at (a) to 60 °C at (b)

$$Q_{(a-b)\,1} = (m_{CH}C_{CH} + m_S.C_S)\,(T_a - T_b) \text{ kJ} \ldots \quad (4.5)$$

m_{CH}=mass of charcoal 19.25 kg

C_{CH} = specific heat of charcoal = 0.709 kJ /kg K

m_S= mass of steel plate and the fins 33 kg

C_S = specific heat of steel = 0.96kJ /kg [Holman, Heat Transfer, McGrraW-Hffl, 1999]

Substituting for the various values for the equation above,

$Q_{(a-b)\,1}$ = 864.85 kJ

$Q_{(a-b)\,2}$= Heat required to raise the temperature of the methanol vapor in the collector from ambient temperature at (a) to T_b(60 °C)

$$Q_{(a-b)\,2} = m_{a-b}\ \Delta h_{a-b}\text{kJ} = 5.1424\ (1797.1\text{- }1790) = 36.511\text{ kJ}$$

$m_{a-b,}$= total mass of the methanol minus pressurizing evaporated methanol

$$m_{a-b,} = 5.1424\text{ kg (see P. 98)}$$

$Q_{(a-b)\,3}$ = Heat required toevaporate the pressurizing amount of methanol

$$Q_{(a-b)3} = m_p.\ \Delta h_{D1-b} = 0.09\ (1797.1\ 1789.0)\ 0.729\text{ kJ}$$

$$Q_{(ab}\ 864.85 + 36.511 + 0.729 = 902.09\text{ kJ}$$

b. **Heat Generated in the Process (b-c):**

The process b-c takes place at constant pressure, so the differential of the heat supplied to the methanol-charcoal system in the solar collector is the differential of the enthalpy of the liquid-vapor system with the term for the sensible heating of the vapor omitted.

The complete differential of the enthalpy of the liquid-vapor system is;

$$d(m_L h_L + m_V h_V) = d(m_L h_L) + h_V dm_V + m_V dh_V \qquad (4.6)$$

Adding the term ($m_S C_S$ + $m_{CH} C_{CH}$)dT for the beating of the steel plate and charcoal, omitting the termm$_V dh_V$for the heating of the vapor, and integrating from (b) to (c), we obtain;

$$Q_{ab} = \int_b^c \quad (m_{sc}C_s + m_{CHC}C_{CH})dT + m_{b-c}\,dh \ \ + m_{desp}dh_{M1-C}) \qquad \ldots\ldots\ldots.\ (4.7)$$

$$Q_{bc} = (T_c\text{-}T_b)(m_{sc}C_s + m_{CHC}C_{CH}) + 0.5(m_c + m_b)(h_c - h_b) + m_{despM1}h_{M1}\text{–}h_c\ldots.\{4.8)$$

Here we have:

At point (b); T_b = 60 °C (methanol is in saturated liquid condition)

h_b= 1797.1 kJ/kg

At point (c); T_c = 90 °C, (methanol is in superheated vapour state)

P(pressure) 0.844 bar

h_c (enthalpy of methanol vapour) = 1855 id/kg

At point ($M1$); T$_{M1}$ 68.5 °C (methanol is in superheated vapour state)

h$_{M1}$ (enthalpy of methanol vapour) = 1850 kJ/kg

$$Q_{b-c} = Q_{(b-c)\,1} + Q_{(b-c)\,2} + Q_{(b-c)\,3} \qquad (4.9)$$

$Q_{(b-c)\,1}$= Heat required to raise the temperature of the steel plate and the charcoal from the temperature at (b) to 90 °C at (c)

$$Q_{(b-c)\,1} = (m_{CH}C_{CH} + m_S.C_S)\,(T_c - T_b)\text{ kJ...} \qquad (4.10)$$

Substituting for the various values for the equation above,

$$Q_{(b-c)\,1} = 864.84\text{ kJ}$$

$$Q_{(b-c)\,2} = m_{b-c}\,\Delta h_{b-c}\text{kJ .} \qquad \text{.. (4.11)}$$

$$Q_{(b-c)\,2} = 3.585\ (1855 - 1797.1) = 207.57\text{ KJ}$$

m_{b-c}= total mass of the methanol minus total evaporated methanol

$$m_{b-c} = 3.585\text{ kg}$$

$Q_{(b-c)\,3}$Heat required to evaporate the working amount of methanol

$$Q_{(b-c)\,3} = m_u(h_c - h_{m1}) = 1.557\ (1855 - 1850) = 7.785\text{ kJ}$$

$$Q_{(b-c)} = 864.84 + 207.57 + 7.785 = 1080.195\text{ kJ}$$

$$X_{m,c} = 0.09$$

$$\text{Gives } m_L(c) = 3.6754\text{ kg}$$

summarizing, we find the total heat of regeneration is

$$Q_{ab} + Q_{bc} 902.09 + 1080.195\ 1982.285\text{ kJ}$$

And this produces 5.77 kg of distilled methanol (desorbed from the charcoal at the end of the heating process) per square meter of solar collector area. Charging amount of methanol: Above calculations lead to the calculations of the required charging amount of methanol as follows:

Since the charcoal releases only 30% of the total adsorbed, then the charged amount should be 2/3 greater to ensure the circulated amount.

Net charged amount of methanol = 5.77/0.3 = 19.21kg -

Gross amount = Net + (pressurizing amount + losses reserve)

$$19.2 + (0.2324 + 0.69)\text{ kg} \qquad = 19.211\text{kg} = 20\text{ kg}$$

Volume of the methanol charged = m/p= 20 /747.6 = 26.75 x $10^{-3}m^3$Where p of methanol at atmospheric pressure, From the tables (p) = 747.6 kg/m^3

Thus, the required charge of methanol = 26.75 x 10^{-3}x 1000 $\approx$ 27liters

The shape of the evaporator is formed to satisfy the volume required, provide the least heat transfer surface and forms the ice molds shape

c.- **Heat Rejected in the Process (c-d':**

By similar calculations we may find the amount of heat released from the solar collector in the process (c-d) and (d-a). Thus, by replacing (a) by (c) and (b) by (d) in equation (3.1) leads to;

$$Q_{c-d}=\int_c^d \quad (m_{sc}C_s+m_{CHC}C_{CH})dT+m_{c-d}\,dh \;\; +m_p dh_{c-D2}) \ldots \qquad (4.12)$$

$$Q_{cd}= (T_d-T_c)(m_{sc}C_s+m_{CHC}C_{CH})+m_{c-d}\,(h_d - h_c)+m_p h_{c-}h_{D2} \qquad (413)$$

At point (c); (methanol is in superheated vapor condition)

T (temperature) = 90 °C

P(pressure) = 0.844 bar

h_c (enthalpy of methanol vapor) = 1855 kJ/kg

At point (d); T_d = 50 °C, (methanol is in superheated vapor state)

h_d= 1810 kJ/kg

At point ($D2$); T_{D2} = 63.5 °C, (methanol is in superheated vapour state)

h_{D2}=1810kJ/kg

$$Q_{c-d} = Q_{(c-d)\,1} + Q_{(c-d)\,2} + Q_{(c-d)\,3} \ldots. \qquad (4.14)$$

$Q_{(c-d)\,1}$ = Rejected heat required to lower the temperature of the steel plate and the charcoal from the temperature at (c) to 50°Cat (d)

$$Q_{(c-d)\,1}= (m_{CH}C_{CH} \; + m_S.C_S)\,(T_c - T_d) \text{ kJ} \ldots \qquad (4.15)$$

Substituting for the various values for the equation above,

= [(19.25 x 0.709) + (33 x 0.46)] (50 -90) - 1153.12 kJ

$Q_{(c-d)\,2}$ = Sensible heat rejected to lower the temperature of the liquid methanol in the collector from temperature at (TC) to T_d (50 °C)

$$Q_{(c-d)\,2}=m_{c-d}\,\Delta h_{c-d}\text{kJ} \ldots\ldots. \qquad (4.16)$$

m.d = total mass of the methanol remained in the collector plus the Pressurizing amount.

$$m_{a-b} = 3.6754 \text{ kg}$$

Substituting for the other values,

$Q_{(c-d)\,2}$= - 3.6754 (1855 - 1810) = - 165.39 kJ

$Q_{(c-d)\,3}$ = Heat released to desorb the pressurizing amount of methanol

$Q_{(c-d)\,3} = m_p(h_c - h_{D2})$ = 0.09 (1855 - 1810) 4.05 kJ

$Q_{(c-d)}$ =-(1153.12 + 165.39+4.05) = - 1322.56kJ

3.7.4- Heat Rejected in the Process $(b - a)$: -

At point (d); T_d = 50 °C (methanol is in superheated vapour condition)

h_d= 1810 kJ/kg

At point (a); T_a = 30 °C, (methanol is in superheated vapor state)

h_a= 1790 kJ! kg

At point $(M2)$; T_{M2} = 40 °C, (methanol is in superheated vapor state)

h_{M2} = 1780 kJ/kg

$Q_{d-a} = Q_{(d-a)\,1} + Q_{(d-a)\,2} + Q_{(d-a)\,3}\cdots$. (4.17)

$Q_{(d-a)\,1}$ = Heat rejected to lower the temperature of the steel plate and the charcoal from the temperature at (d) to 30°Cat (a)

$Q_{(d-a)\,1}= (m_{CH}C_{CH} + m_S.C_S)\,(T_a - T_d)$ kJ

Substituting for the various values for the equation above,

$Q_{(d-a)\,1}$ = [(19.25 × 0.709) + (33 × 0.46)] (30- 50) = -576.56 kJ

$Q_{(d-a)\,2}$ = Heat rejected to lower the temperature of the liquid methanol in thecollector from T_d (50 °C) to ambient temperature at (a)

$Q_{(d-a)\,2}=m_{d-a}\,\Delta(h_{d-a})$kJ.. . (4.18)

m_{d-a} = Remaining mass of the methanol plus pressurizing mass adsorbed

m_{d-a}= 5.2324 kg

Substituting for the other values,

$Q_{(d-a)\,2}$ = -4.212 (1810 - 1790) = -84.24 kJ

$Q_{(d-a)\,3}$= Heat rejected to desorbed the evaporated amount of methanol from the cooling process in the evaporator.

$Q_{(d-a)\,3} = m_u(h_d - h_{M2})$ .. .. (4.19)

= 1.577 (1780- 1810) = -47.31 kJ

$Q_{(d-a)}$ =-(576.56 + 84.24+47.31) = - 708.11 kJ

(The negative sign indicates heat losing) Therefore the total amount of heat released is;

$Q_{cd} + Q_{da}$ = -(1322.56 + 708.11) -2030.67 kJ,

The negative sign indicating that the heat is lost by the system.

4.8 COMPONENT DESIGN;

Various components of the system design are explained in the following:

a. **Collector Plate:**

From the assumptions and the calculations of solar collector plate is of (lm x 1 m) steel plate i.e. a net collector area of $1m^2$, and for insulation purpose, 10 cm is added to each sides. The important design work apart from that of specifying the collector dimensions is the charcoal box sizing, since this box is designed to exchange heat with the collector plate. Charcoal boxing turn depends on the amount of methanol., it will be seen that total mass of methanol required for charging the system is 5.775 kg. According to U. Merging [University of Wales, 1989.], the charged amount of methanol should be three time greater than the circulating mass required. The importance of the excess mass is to reserve for circulating and pressurizing amount of methanol, the mass should be increased by 30% which makes up a total of 19.25 kg. Taking the density of the solid charcoal as 481kg/rn^3, and allowing 25%increase in volume for the crushed charcoal, the density will be25% less i.e. 385 kg/$\boldsymbol{m^3}$. Therefore, the volume required for accommodating the 19.25 kg of charcoal can be calculated as:

V = m\p = 19.25/385 0.05 m^3Taking the area of the charcoal box to be the same as the effective area of the collector, the height of the charcoal box (t) will be:

t=V/A=0.05/1.0=0.05 m=5.0 cm

b- Condenser Design:

In designing the condenser, the heat released to the tank of water by the condensing methanol, this is simply the difference between the enthalpy of the methanol vapor entering the condenser $m_M h_{MV}$and the enthalpy of the methanol liquid leaving the condenser $m_M h_{ML}$. Hence we have;m_M = 1.557 kg, and making the simplifying assumptions that the methanol is pure and that the whole process takes place at T_s= 60°C, we have from previous calculations (c - e).

h_c= 1855 kJ/kg & h_e==650kJ/kg

The total amount of heat released by 1.557 kg of methanol condensing

(c - e) is therefore=$m_M(h_c\text{-}h_e)$ (4.20)

= 1.557 × (l855-650) =1876.185kJ

In designing the system, it should be ensured that there is sufficient water in the tank to prevent its temperature from being raised more than a few degrees by this heat. Condenser pipe calculations: For estimating heat transfer coefficients for sides, (water and methanol)

methanol side:

Using empirical equation for natural heat transfer coefficient in a horizontal tube [Holman, Heat Transfer, McGrraW-Hffl, 1999.]

$$h_m = 0.55\{ \frac{g\rho_{i\,(}\rho_l - \rho_{v\,)}K^3 h_{fg}}{u_1\,(T_v - T_w)D_{pi}} \}^{0.25} \ldots\ldots \tag{4.21}$$

Where:

h_m= heat transfer coefficient on the methanol side W/m^{2}K

g = gravitational acceleration = 9.81 m /s^2

ρ_l= density of liquid methanol at the condensation temperature = 752.67kg/m^3

$\rho_{v\,=}$ density of methanol vapour 0.757 kg/m^3

K = thermal conductivity of liquid methanol 0.2033 W/mk

h_{fg}= latent heat of condensation of the methanol = 1110.28 kJ/kg

μ_1= Viscosity of the liquid methanol = 752.67 × 7.37 ×10^{-7}

=5×547 ×10^{-4}(N.s)/m^2

T_v = Temperature of the methanol vapour °C = 50 °C

T_w=Temperature of the water °C =60 °C

D_{pi}= Inner diameter of the pipe (m) = 0.007 m

Substituting for various quantities,

h_m= 449.122 W/m^{2}K

2. Water side:

$$h_w = \frac{k_{w,}Nu_w}{D_{p,o}} \ldots\ldots \tag{4.22}$$

Where:

h_w =Heat transfer coefficient of the water layer on the outer surface of the tube

$k_{w=}$ Thermal conductivity of the water

Nu = Nusselt Number

$D_{p,o}$= Outer diameter of the tube (m) = 0.008 m

$$\text{Nu}=\text{c}(\text{Pr. Gr})^{\text{m}}\ldots\ldots\ldots \tag{4.23}$$

Where:

c& m = constants selected from a table according to the value of (Pr. Gr)

Pr= Prandtl Number

Gr = Grashof Number

(Pr.Gr) = $\frac{g\beta\rho^{2}{}_{,}C_p}{uk} \times x^2 \Delta\text{T}$……. (4.24)

Where:

g = gravitation acceleration 9.81 m/s^2

β= thermal coefficient = the inverse of the average temperature of the water.

ρ =Density of the water at the given temperature 999.6 kg/m^3

C_p= Specific heat of water 4.178 kJ/kgK

μ= Dynamic Viscosity of the water at the given temperature = 6.82×10^{-4}kg/m.s

K= thermal Conductivity of the water

x = the thickness of the tube 0.001 m

ΔT =Temperature gradient of the water = (30 - 35)

$\frac{g\beta\rho^{2}{}_{,}C_p}{uk}$ =this is taken direct from tables according to the value of β= 33 x10^{10},so

(Pr. Gr) = 3.3 xl10^{10}× (0.001$^{)3}$) × (35 -30) = 165

But since the value (Pr. Gr) is in the range: 10^4<Gr Pr>10^9, experimental formula can be used [Ref. 16) as follows:

h_w = 1.32$(\frac{\Delta\text{T}}{d})^{\frac{1}{4}}$…….. (4.25)

ΔT= temperature difference = 5 °C

d=0.007m,

h_w =1.32 $(\frac{5}{0.007})^{0.25}$= 6.82 KJ/Kg

k$_c$ = 0.644 W/m.k, then

$\frac{1}{U} = \frac{1}{h_m} + \frac{1}{h_w} + \frac{X}{K} + R_{f,w} + R_{f,m}$……… (4.26)

Where:

U = overall heat transfer coefficient W/m^{2}K

$R_{f,m}$= fouling factor for methanol side = 0.0002

$R_{f,w}$ = fouling factor for water side = 0.0005 [Holman, Heat Transfer, McGrraW-Hffl, 1999] Substituting for values above,

$\frac{1}{U} = \frac{1}{338} + \frac{1}{6.82} + \frac{0.001}{43}$ + 0.0002 + 0.0005 = 0.15m^{2}K/W

U = 6.65 W/m^{2}K

But U = Q/(Aθ)

Where:

Q = heat transfer rate = Q/t = (1876(eq4.20) ×10^3) /(8 hr× 3600) = 65 W

A = heat exchange area

θ = logarithmic temperature difference = LMTD

$$\theta = \frac{(T_c - T_{w1}) - (T_c - T_{w2})}{\ln\left[\frac{T_c - T_{w1}}{T_c - T_{w2}}\right]} \qquad (4.27)$$

Substituting for various values,

θ=16.83K

A = Q/Uθ = 65/(6.65 × 16.83) = 0.58 m^2

But, A =$\pi . D . L$

D = diameter of the tube

L= length of the pipe,

1=A/($\pi . D$) = 0.58/(π× 0.008) = 23.1 m

A cylindrical container is used for the purpose with a helical coil copper pipe.

C- **Evaporator Design:**

The liquid methanol in the evaporator container, is supposed to fall rapidly to temperature -5 °C. This is represented by the transformation from point (e) to point (f). Let m_{sr} denote the mass of steel in the receiver and evaporator; then the differential of the heat flow from the receiver to the liquid methanol is;

$Q = m_{sr} C_s dT$

$dh_{M-} V_M dp = d(m_{ML} h_{ML} + m_{MV} h_{MV}) - (m_{ML} V_{ML} + m_{MV} V_{MV}) dp$ (4.28)

Where the subscripts M, L and V refer to methanol, liquid and vapor respectively.

$Q_{bc} = (T_c - T_b)(m_{sc} C_s + m_{CHC} C_{CH}) + 0.5(m_c + m_b)(h_c - h_b) + m_{despM1} h_{M1-} h_c \ldots$.(4.29)

$Q_{bc} = (T_c - T_b)(m_{sc} C_s + m_{CHC} C_{CH}) + 0.5(m_c + m_b)(h_c - h_b) + m_{despM1} h_{M1-} h_c \ldots.$ (4.30)

Setting $dh_{MV} = -dm_{ML}$, omitting the terms containing dh_{MV} and V_{MV}, equating the result to the differential of the heat from the receiver, and integrating from the state e to state f, we obtain;

$\int_e^f (m_{ML} dh_{ML} + (h_{ML} - h_{MV}) dm_{ML} - m_{ML} v_{ML} + m_{desp} dp) = \int_e^f (m_{sr} C_s dT$ .. (4.31)

Finally, approximating the integrals in the same way as obtained before,

$0.5(m_{ML}(e) + m_{ML}(f)) (h_{ML}(f) - h_{ML}(e)) + 0.5(h_{ML}(e) - h_{ML}(e) + h_{ML}(f) -$
$h_{ML}(f)) (m_{ML}(f) - m_{ML}(e)) - 0.5(m_{ML}(e) + m_{ML}(f)) (0.5)(v_{ML}(e) +$
$v_{ML}(f)) P(f) - P(e) = m_{sr} C_s (T(f) - T(e))$ ………. (4.32)

All the quantities in this equation are known from the system design and from thermodynamic analysis except m_{ML}(f), which may therefore be calculated.

Taking m_{sr} = 9.0 kg and the remaining quantities from thermodynamic tables we obtain m_{ML}(f) = 1.58 Kg per square meter of the collector, the useful heat of refrigeration q_f is the heat required to vaporize the remaining liquid methanol at constant temperature -3.0 ℃ and pressure

0.03327 bar, which is given by;

$Q_f = m_{ML_f}(f)\,(h_{MV_f} - h_{ML_f})$ ...... (4.33)

In designing the evaporator, the heat gained by the boiling-methanol in the evaporator pool must be equal to the heat released by the freezing water.

M_u = 1.578 kg,

And making the simplifying assumptions that the methanol is pure and that the whole process takes place at the evaporating temperature of -5 ℃,

We have from previous calculations;

$h_{f,-5}$= 531.5 kJ/kg and

$h_{g,-5}$= 1209.2 kJ/kg

The total amount of heat absorbed by 1.578 kg of methanol boiling at -5 ℃, (0.02 bar) is therefore:

$Q = m_M\,(h_{g,-5} - h_{f,-5}) = 1.557(1738.3 - 626.25) = 1731.46$ kJ

(per square meter of the collector)Dimensioning of the evaporator box. For estimating heat transfer coefficients for both sides, (water and methanol) following equations are used :

1**. methanol side:**

Using empirical equation for natural heat transfer coefficient as before: [Thermodynamics,4th Edit., Longmafl 1992.]

$$h_m = 0.58\{\frac{g\rho_1(\rho_l - \rho_v)K^3 h_{fg}}{u_1\,(T_v - T_m)L_{pi}}\}^{0.25} \ldots\ldots$$

h_m= heat transfer coefficient on the methanol side W/m^{2}K

g = gravitational acceleration = 9.81 m/s^2

ρ_l = density of liquid methanol at the evaporation temperature813.0 kg/m^3

ρ_v = density of methanol vapor = 0.0478 kg/m^3

K = thermal conductivity of liquid methanol = 0.206 W/mk

h_{fg} = latent heat of condensation of the methanol = 1209.2 kJ/kg

u_1 = Viscosity of the liquid methanol = 8.17 x10^{-7}

T_v = temperature of the methanol vapor °C = 20.0 °C

T_m = temperature of the methanol °C = -3 °C

L_{pi} = Characteristic dimension (m) = (A/P)

A = Area of the plate (m^2)

P = Perimeter of the surface (m)

AssumeL_{pi} = 0.03 145 m

Substituting for various quantities,

h_m = 338.466 W/m^2K

2. For the water side:

$$h_w = \frac{k_{w,}Nu_w}{L_{p,o}} \quad \ldots \quad \ldots \quad (4.34)$$

Where:

h_w= heat transfer coefficient of the water layer on the outer surface of the evaporator.

$k_{w,}$ = thermal conductivity of the water

Nu_w = Nusselt Number

$L_{p,o}$= characteristic dimension (m) = 0.03145 m

$$\mathrm{Nu} = \mathrm{c}\,(\mathrm{pr.\,Gr})^{\mathrm{m}} \ldots\ldots. \quad (4.35)$$

Where:

c, m = constants selected from a table according to the value of (Pr.Gr)

[Thermodynamics,4th Edit., Longmafl 1992, see the appendix]

Pr = Prandtl Number

Gr = Grashof Number

$$(\mathrm{Pr.\,Gr}) = \frac{g\beta\rho^2{}_{,}C_p}{uk} \times x^2\Delta\mathrm{T} \quad (4.36)$$

Where:

g = gravitational acceleration = 9.81 m/s^2

β = thermal coefficient = the inverse of the average temperature of the water

ρ = density of the water at the given temperature = 1000 kg/m^3

C_p = specific heat of water = 4.178 kJ/kgK

u= viscosity of the water at the given temperature

v.ρ= 6.82 ×10^{-4}kg/m.s

k = thermal conductivity of the water

x = the thickness of the evaporator sheet-metal = 0.001 m

ΔT= temperature gradient of the water = 5 °C

$\frac{g\beta\rho^{2}{}_{,}C_{p}}{uk}$ this is taken direct from tables according to the value of , β= 8.6 ×10^{9}, [see the appendix]

(Nu.Gr)= 8.6 ×10^{9}× $(0.001)^{3}$× (5) = 43

For horizontal plate heated with uniform flux source; But since the value (Pr. Gr) is in the range:10^{4}, <Gr Pr>10^{9},, experimental formula can also be used [16] as follows:

$$h = 1.23\,(\frac{\Delta T}{L})^{1/4} \qquad (4.37)$$

ΔT=5°C

L=0.03l45m (P. 116)

hence:

h = 1.23 $(\frac{5}{0.03l45})^{1/4}$= 4.37 W/m^{2}K

Taking k =43 W/m. k, then (4.38)

$$\frac{1}{U} = \frac{1}{h_m} + \frac{1}{h_w} + \frac{X}{K} + R_{f,w} + R_{f,m} \;\ldots\ldots\ldots$$

Where:

U = overall heat transfer coefficient

$R_{f,m}$ = fouling factor for methanol side 0.0002 [J. P. Holman, Heat Transfer

$R_{f,w}$ = fouling factor for water side 0.0005 [Thermodynamics,4th Edt., Longmafl 1992.]

Substituting for values above,

$$\frac{1}{U} = \frac{1}{338} + \frac{1}{4.37} + \frac{0.001}{43} + 0.0002 + 0.0005 = 0.2325 \text{ m}^2\text{K/W}$$

U =4.3 m^{2}K/W

But U = Q/(A.θ), Q = 1731.46 K J(P. 114)

Where:

q = heat transfer rate = Q/t Watt

= 1731.46 ×10^{3}/(8 ×3600) = 60.12 W

A = heat exchange area

θ = log mean temperature difference = LMTD

$$\theta = \frac{(T_c - T_{w1}) - (T_c - T_{w2})}{ln\frac{(T_c - T_{w1})}{(T_c - T_{w2})}} \;\ldots\ldots\ldots\ldots\ldots \quad . \; (4.39)$$

Substituting for various values,

θ=12.0K

therefore, A = Q/U θ = 60.12/(4.3 × 12.0) = 1.156 m^{2}

But, A = V/d

d = depth of the box, V = the volume of the box = 5 litters = 0.005 m^3

d = depth of the box, from geometry of the box to form the ice moulds;

d=5 cm= 0.05 m,

Therefore, a = 0.005/0.05 = 0.1 m (rejected, since it is less than the heat transfer area required calculated above) for inconvenience of the ice blocks' design

Calculations of the product (Ice):

Taking the total mass of steel sheet forming the evaporator box

$$=1.15\ m^2 \times 0.001\ (\text{thickness}) \times 7833\ (\text{density}) = 9.0\ kg$$

Total cooling load (useful) Q_u can be calculated as:

$Q_u = Q_{e1} + Q_{e2} + Q_{e3}$………….. (4.40)

Where:

Q_{e1} = Cooling load of the ice mould material

Q_{e2} = Cooling load of the water from ambient temperature to 0 °C

Q_{e3}=latent heat of transformation of the water to ice (solidification process)

Taking C_w specific heat of water = 4.18 kj/kg

$Q_{e1} = m_{sr}C_s dT_e = 9.0 \times 0.46 \times (30 - 0.0)\ 124.2\ KJ$

$Q_{e2} = m_{sr}C_s dT_w = m_{wf} \times 4.1818 \times (30 - 0) = 125.45\ m_{wf}$

$Q_{e3} = m_{wf}h_{fgw} = m_{wf} \times 332.4 = 332.4\ m_{wf}$

Total productive heat $Q_{e2} + Q_{e3} = m_w(125.45 + 332.2)\ 457.65 m_w$

But total cooling effect $Q_u = Q_{e1} + Q_{e2} + Q_{e3}$

Therefore;

$Q_u - Q_{e3} = Q_{e1} + Q_{e2} +$

Qr:1783.917W=457.65 $\times m_w$

Then;m_w= 1783.917/457.65 = 3.9 kg of water

Total projected area (A) =39 × 29 1131.0 cm^2

Perimeter (P) = 2(39 + 7.5) = 93 cm

Characteristic Heat Transfer Length (L) = A/P = 292.5/93 = 3.145 cm = 0.03145 m

Total lower Sheet Area (Actual Heat Transfer Area) = (7.0 × 4) + (5 ×5)

= 53 cm^2 = 0.0053 m^2

Ice-Blocks' Sectional Area = (10.8 × 4) + (5 × 6 × 3) = 133.2 cm^2

3.9 kg of ice = 3.9 litter of water = 3900 cm^3

Therefore, depth of the ice mould =Volume (V)/Sectional Area = 3900/1131.0 =3.45 cm. Comparing the obtained area with the heat transfer area required, it would be seen that the designed dimensions are acceptable since designed area is greater. To allow for the increase in volume because of the difference intensity between the ice and the water, the depth is increased to 5.0 cm to accommodate this variation.

Allowing for other losses through the insulator, net ice production can be taken as;

3.8 Kg of ice daily

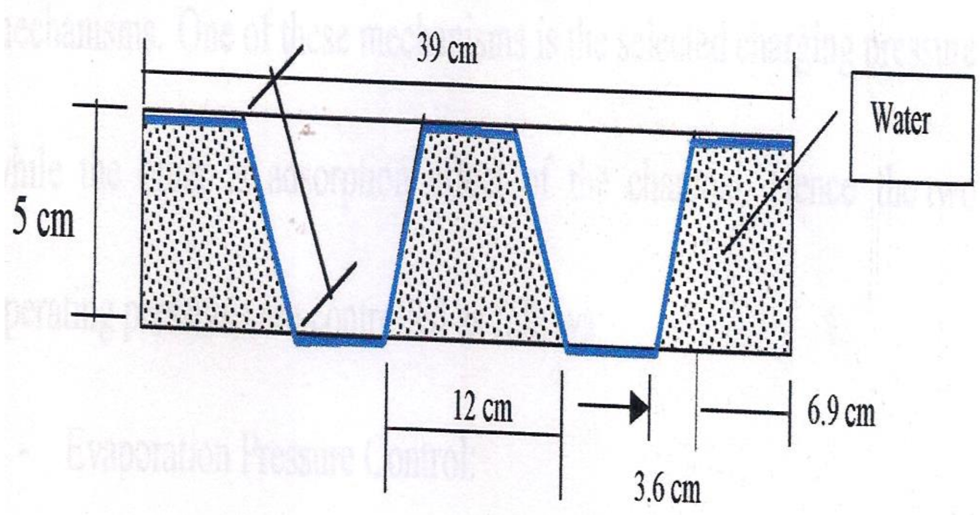


Fig. (4.4) Dimensions of the Evaporator

4.9 Cooling Ratio:

Substituting the known values on the right hand side we obtain;

$$CR=\frac{Q_f}{(Q_{ab+(Q_{bc})}} \qquad (4.41)$$

Q_f= 1731.67 KJ per square meter of the collector.

$Q_{ab+(Q_{bc}}$ = 902.09 + 1080.195 = 1982.285 KJ

Assuming collector efficiency 30%, heat supplied

$Q_T = Q_{a-c}/\eta$ = 1982.283/0.3 = 6607.61 KJ

The cooling ratio for this cycle is defined as;

Using the results calculated above we obtain a cooling ratio of,

CR = 173 1.67/6607.61 = 0.262

4.10 Pressure control:

The machine is. on control valves are needed on the operation on the system both evaporation and condensation pressure are controlled automatically in mechanisms.

One of these mechanisms is the charging pressure while the other adsorption effect of the charcoal. Hence the operating pressures are controlled

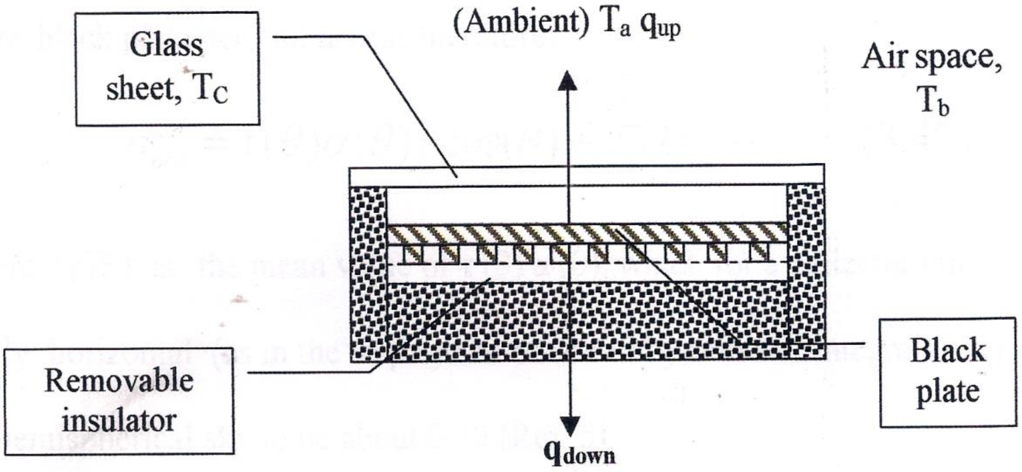


Fig {4.5} Heat Flow on the Collector

4.11 Collector Heat Losses:

The rates at which the process in the cycle takes place are controlled by the energy flow rates from the solar collector. In discussing these rates, that solar flat-plate collector with one glass cover at a distance 50 mm from the black plate, and foam insulation 100 mm thick at the back which can be removed at night to allow the system to cool quickly is considered.

The transmissivity $\tau(\theta)$ of the glass cover, and the absorptivity $\alpha(\theta)$ of the black plate, for solar radiation both depend on the angle of incidence (θ) of the radiation. The total solar radiation .

$q_{in} = I \cos\theta + D$ (4.42)

Where (1) is the direct solar irradiance at the angle of incidence 0, and D is the diffused solar irradiance The rate g_{abs} at which solar radiation by the black plate per unit is therefore:

$q_{abs} = \tau(\theta)\,\alpha(\theta) I \cos(\theta) + \overline{\tau\alpha} D$ (4.43)

Where $(\overline{\tau\alpha})$ is the mean value of $\tau(\theta)\ \alpha\ (\theta)$, which for a collector that is nearly horizontal (as in the tropic) can be found by mean of integrals over the hemispherical sky to be about 0.70 [Solar Energy, Vol.42, 1989.].

There are several processes by which a flat-plate solar collector losses heat, as illustrated in fig above. they are as follows (steady state heat transfer is assumed)

1. Convection and radiation $q_{up\ from}$ the glass cover at temperature T_a to the surroundings and at ambient temperature T_a.
2. Convection and radiation q_{ap} between the black plate at temperature T_h and the glass cover.
3. Conduction q_{down} through the back insulation.
4. Heat losses through the edges of the collector and through the connecting pipes These losses may not be negligible, but in good designs they should be small and here are being neglected.
5. Convection and radiation q_{down} from the back of the black plate when the back insulation is removed at night to cool the collector and release the heat of absorption. The total heat loss rate q_{loss} from the collector is given by;

$q_{loss}= q_{up} + q_{down}$ (4.44)

as function of T with the help of standard equations for the heat transfer in solar collectors. The calculation of q_{loss} must be done numerically. Some results obtained for ambient temperature 30 °C give for selective black plate and one glass cover:

q_{loss} = 33.4(T_b -30) + 0.0064(T_b- 30)^2 (W/m^2) ... . (4.45)

Where T_b is in degree Celsius

When the insulation is removed from the back of the solar collector one obtains;

q_{loss} =3+ l0.4(T_b -30) + 0.0064(T_b - 30)^2 (W/m^2) ………. (4.46)

The instantaneous efficiency i of the solar collector is defined by the equation6

$\eta = (q_{abs}- q_{loss}) / q_{in}$ (4.47)

This depends on the intensities of the direct/diffuse solar irradiance,(I) and (D), on the angle of incidence θ of the direct solar radiation, and onthe temperature T_b of the black plate.

The overall performance of the system may be defined to be the ratio of the useful heat of the refrigeration Q_f , to the total solar incident on the

Collector per day. This equal the cooling ratio multiplied by the average efficiency of the solar collector throughout the day.

4.12 MODLING THE SYSTEM THROUGH TIME:

Although, we have seen how to calculate the total quantities of heat in a particular cycle, and the heat transfer rates in the solar collector that controls the process in the cycle require a method of calculating the responses system to a given solar input throughout the day in older to do this, we must write down the differential quantities for the heat transfer that occurs in each of the processes (a-b), (b-c), (c-d), and (d-a), and integrate them with respect to time the calculations involved are simple in principle. But detailed and numerous. therefore be performed on a computer. the basic ideas can be explained. in the process (a-b) differential of the absorbed solar radiation minus the losses from the collector equals the differential of heating of the collector and the fluid in it.

$$(qabs- q_{loss})\ dT= (m_S c_{S+}\ m_{cH\ c\ cH})\ dT +m_L T d_{sll}\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad .\ (4.48)$$

Here q_{abs}in given by equation (3.42) with I, D and specified as functions of time: while q_{loss} is given by equation (4.45). for simplicity we assume that temperature T of the methanol – charcoal combination and the black plate temperature T are same. In real designs these. temperatures differ slightly, and the temperature of the solution is not uniform, but we shall ignore these difficulties. The thermodynamic properties of the combination of methanol-Charcoal are assumed to obey functions of X_M and T. the right hand side of (4.49) is then written as:

$$F(X_M)=(m_{sc}C_s+m_{CHC}C_{CH})dT+m_L\ (X_M,T)T(\delta_{S_L}(X_M T)/\delta T)dT)\ . \quad ..(4.49)$$

And the resulting equation when integrated numerically step-by-step gives (T) as a function of time (t). At each step the pressure $P(X_M,T)$ is found and the integration is continued until the condensation pressure0.844 bar is reached. On a cloudy day the solution may not be strong enough for this to occur, and then no methanol is distilled. [5] In the process b-c the differential equation is;

$$(q_{abs}\ \text{-}\ q_{loss})\ dT = (m_{sc}C_s+m_{CHC}C_{CH})dt + d(m_L h_L) + h_V d_{M_V}\ldots \quad ..(4.50)$$

Where

m_L , h_L , $m_V$and hv are known functions of X_M and T. Itfollows that the right-hand side can be written in terms of the differentialof dX_M and dT. In order to follow the isobar we need $dP(X_M,T) = 0$; so the differentials dX_M and dT must satisfy the equation

$$[\delta P(X_M,T)\delta X_M\ ] + [\delta P(X_M,T)\delta T\ ]dT=0\ \ldots\ldots.. \quad (4.51)$$

Equations (4.48) and (4.49) are now solved simultaneously step-by-step to determine how far the process goes. On a good day the final temperature may be high and a large quantity of methanol may be distilled. It is assumed that when the temperature stops rising, due to Reduction of the solar heating towards the end of the day, insulation removed from the back of the collector and start the process c-d. At this point we have calculated the amount of distilled methanol available for refrigeration for the calculation of the process c-d one uses equation (4.47) again with q_{abs} small at the afternoon and zero after dark, and g_{loss} given by equation (4.46) the process d-a depends on the refrigeration load. further development of the model is needed to calculate this process. The variability of the experiment yields by the manner in which cloudiness varies throughout the day if the weather is fine in the morning and cloudy in the afternoon the yield of methanol may be good due to the strong heating of the solar collector during the critical generating period. on the other hand, when the morning is cloudy the collector many not become hot enough by middy to generate methanol, even if the weather is fine in the afternoon

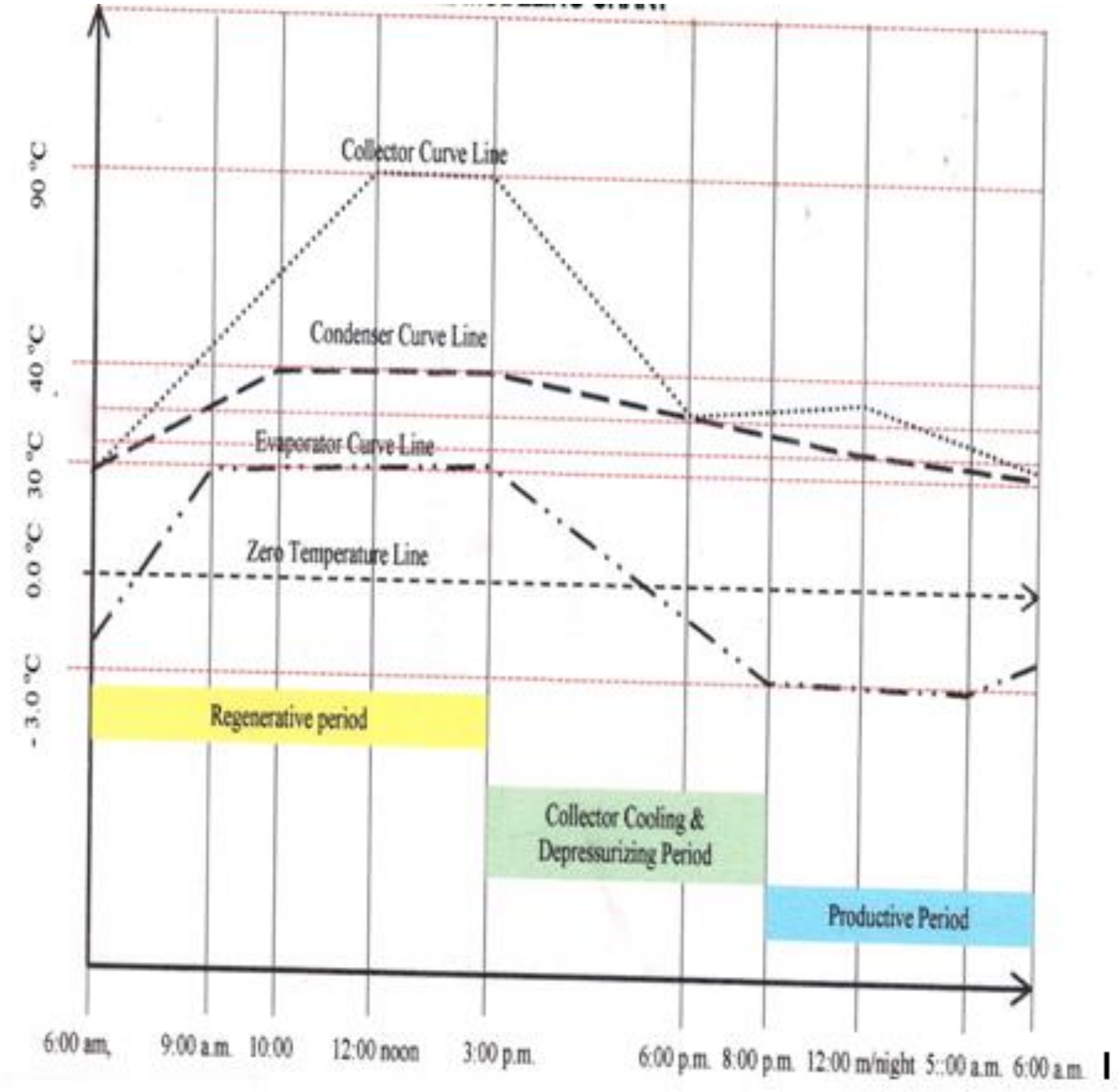


Fig 4.6 Time molding chart

CHAPTER IV

CONCLOUSION &RECOMNDATION

CHAPTER IV

CONCLOUSION & RECOMNDATION

5.1 EOCONOMICALCOMPARISON:

Adsorption Solar Freezer can be compared with either ordinary electric refrigerator or with a photocells electric solar refrigerator form the economical point of view as follows:

TABLE (5.1) Comparison

Comparing factor	Adsorption S.F	Conventional Refrigerator	Photocells Refrigerator
Cost	Cheep	Fairly cheep	Expensive
Reliability	Good	Bad	V. good
Maintenance	Simple	Difficult	V. difficult
Running cost	V. low	High	Low
Productivity	Low	V. good	Good
Simplicity	V. simple	Complicated	Complicated

5.2 CONCLUSION

The objective of this study is to show the feasibility of an adsorption refrigerating system driven by solar collector, which is coupled with an annular heat pipe for transferring heat towards the adsorbent bed. A theoretical model and a numerical program have been developed, in order to evaluate the performance of the adsorption- cooling machine. The numerical results show a great sensitivity of the performance coefficient of the machine to the radius of the adsorber and the aperture width of collector. Simple structure, low cost, solar adsorption refrigeration had been designed and focused to improve heat transfer in the adsorbent bed and to improve external heating and cooling design and simulation of solar freezer unit on adsorption principle has become possible. The idea is a promising one for its simplicity and the availability of its materials in the local market and the simplicity of manufacturing technology and lab our skills needed. The research conducted enable simple design, which if it is applied perfectly can help in constructing a solar freezer of daily productivity of 5 kg of ice It brings out how the machine will look like also this makes it possible to detect any structural fault for another construction if needed next time.

One of the advantages of adsorption is operation with thermal heat source . The COP of adsorption is not high, compared to the other. They have other problems and difficulties during the design and production such as leakage, high vacuum, poor mass, and heat transfer. However, many researches are performed to develop innovative designs and cycles in order to eliminate those problems and enhance COP values. The researchers are also focused on the solutions for the problem of leakage and high vacuum in adsorption systems , since the cycle has to be repeated for many times under the same operation pressures . adsorption systems will become widespread and popular in near future.

Recommendations:

1. Further research is needed concerning the local charcoal for specifying its type and properties.
2. Nocturnal temperature is claimed to have great effect on the performance of adsorption solar refrigeration, this is yet to be proved for the local climate of Khartoum and other Sudan regions. Further research is needed to verify this.
3. This research is promising. more investigation is needed in order to produce this of machine in commercial quantities.
4. it is the hope of The research that this project shall be continued to confirm its performance.
5. Methanol and charcoal as working pair in the adsorption solar freezer are recommended but investigations are still needed for their adaptation to use with our local products.
6. The researcher foresees the success of this research for future application and recommends enhancing the research to cold store units and village size units of the same method.
7. Protect Collector Plate Distortion due in construction:
8. Heaviness of the evaporator material: For good heat conduction between the evaporator and the water in the tray attached to hence the productivity of the machine.
9. Joining Method of the Water Tray to the Evaporator by using the best method .
10. Evaporator Pipe Length: This is the short pipe joining the condenser and the evaporator. A serious mistake has been committed on welding this pipe to the evaporator –box. Instead of welding it just slightly below the inner surface of the evaporator upper plate it is welded into the evaporated, this can lower the evaporation

REFERENCES

REFERENCES

1. Hu 3mg and R. H. B. Exell, Simulation and Analysis of an intermittent Solar Powered Charcoal/Methanol Refrigerator, Asian Institute of Technology, Bangkok, (paper, 1992).
2. Usman Merghni, (Solar Refrigeration in Sudan) Ph.D. Thesis, University of Wales, 1989.
3. Web Sites:
   a- efunda.com
   b- methanol.org
   e- susdesign.com
   d- wattsun.com
   e- efunda. coml ./methanol
   f- solar.com
4. Excel R. H. B., Solar Air Conditioning and Refrigeration, Pergamon Press, 2012.
5. Grenier and M. Pons, Experimental and Theoretical Results on the Use of an Activated Carbon. /CH3CH intermittent Cycle for the Application to a Solar Powered Icemaker. Solar Adsorption Cooling, Renewable Energy Series, Editor-in-chief, A. A. M. Sayigh, 1992.
6. Meunier Feb. F., Guilleminot J. J., Pons M., Solar powered Refrigeration Using Intermittent Solid Adsorption Cycles, (paper, "Revue Physics" 1986)
7. Experimental Data on a solar powered Icemaker Using Activated Carbon and Methanol Adsorption pair, Journal of Solar Energy, Vol. 109, 9. R. H. B. Exell, Ice-Making for Rural Applications, Renewable M. Pons and Grenier,
8. Energy Series, Editor-in-chief, A. A. M. Sayigh., 2015, PP (203 -223)
9. Energy 2000 "k global strategy for sustainable development Energy New Era" Report submitted by energy to the United Nation World Commission on Environment and Development, PP (177- 187).
10. Beckman, W. A., Duffie J. A., Solar Engine of Thermal
11. E. F. Passons and J. F. Escobedo, Simulation of an Intermittent Adsorptive Solar Cooling System, Journal of Solar Energy, Vol.42, 1989.
12. Thermal Process, John Wiley (1980).
13. Heat and Mass Transfer in a non-isothermal fixed bed solid adsorbent reactor, J. J. Guilleminot and Meuner,
14. Peter E. Liley, Thermodynamic Properties of Methanol, Chemical Engineering Magazine,

15. Stephen K. & Timothy Ranja, AFREPREN, (African Energy Policy Research Network), Zed Books Ltd. 2013
16. Heat Transfer, McGrra W- J. P. Holman, 1999.
17. Convection of Heat Transfer, Wiley - inter-science publications. A. Bejan,
18. and Y. R. Mayhew, Engineering Thermodynamics, 4th Edt., Longmat G. F. Rogers, 1992.
19. Ken-ichi Kimura, Solar Adsorption Cooling, Renewable Energy.

APPENDIX A

MATLAB PRORAMME

```
clc, clear
%DESIGN CALCULATIONS:
%Heat Generated in the Process (a-b):
%This process can be considered as partly heating and partly pressurizing process. The collector plate, charcoal and the methanol in the charcoal particles of the collector, all these are to be heated from the ambient temperature T_a to evaporating temperature (T_b). Additional amount of heat is required for generating pressurizing vapor of methanol that will raise the pressure of the system from evaporation pressure (0.03 bar)
m_Ch = 19.25; % kg mass of charcoal
C_Ch = 0.709; % kJ /kg K (specific heat of charcoal)
m_S = 33; % kg (mass of steel plate and the fins)
C_S = 0.96; % kJ /kg (specific heat of steel) [Ref.16]
m_sc= 33.16; %Kg (collector plate + fins)
m_CHC = 19.25; % kg
m_p=0.09; % kg     pressurizing amount of methanol
m_bc = 3.585; %kg   total mass of the methanol minus total evaporated methanol
m_d =m_bc+m_p; %kg   total mass of the methanol remained in the collector plus the Pressurizing amount.
m_u=1.557; %Kg working amount of methanol
hr=8; % working hours
m_ab = 5.1424; %kg
m_da = 4.212; %kg Remaining mass of the methanol plus pressurizing mass adsorbed
c_s =0.46; %kJ/kg K
C_CH= 0.709; % kJ /kg k
%For evaluating the various thermodynamic properties of methanol, basic principles can be used as follows:
```

```
%R= R^/m^ kJ/kg K………                          (3.3)
R_math= 0.2595; %= (8.3145/32.04) kJ/kg K
% Where: R = specific gas constant,
% R^= Universal gas constant = 8.3145 kJ/kmol
% m^= Molar mass, for methanol m? = 32.04 kJ/kmol
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
SP_w=1; % m steel plate width
M_meth=19.25;  %kg  total  mass  of  methanol  required  for charging the system
Ch_den=385; %density of charcoal
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% h = u +RT      u=Cv*T     Cv=Cp-R
C_p1meth= 65.44; % kJ/ kmol,
C_vgmeth= 3.2195; %kJ /kmol,
C_pgmeth= 111.46; % kJ/ kmol,
C_vlmeth= 1.7825; %kJ / kmol,
C_pg = 3.479; % kJ/kg
C_vg = 3.22; % kJ/kg
Cpi = 1.761; % Id/kg,
C_vi= 1.502; %kJ/kg
%  Properties  are  either  calculated  analytically  from  the equations above or
%  taken  from  thermodynamic  tables  (tables  are  used)  [Ref. 11]:
% At point (a);
T_a = 30; % dC super-heated methanol vapor in charcoal
P_a = 0.03; % bar
h_a = 1790; % kJ /kg (enthalpy of methanol vapor)
%At point (b)
T_b = 60; %dC (methanol vapor is superheated)
P_b = 0.844; % bar
h_b = 1797.1; % kJ/kg (enthalpy of methanol vapor)
```

```
%At point (D1); (methanol is in superheated vapour state)
T_D = 45; % dC
h_D = 1789; % kJ /kg (enthalpy of methanol vapour)
%Q_ab = Q_ab1+ Q_ab2+ Q_ab3 ...............(3.4)
% Q_ab1 = Heat required to raise the temperature of the
steel plate and the
% charcoal from the ambient temperature at (a) to 60 at (b)
Q_ab1 = (m_Ch*C_Ch+ m_S*c_s) *(T_b- T_a)

% Q_ab2 = Heat required to raise the temperature of the
methanol vapour in
% the collector from ambient temperature at (a) to T_ (b)
(60 °C)
Q_ab2 = m_ab*(h_b-h_a)

% Q_ab3 = Heat required to evaporate the pressurizing
amount of methanol
Q_ab3 = m_p*(h_b-h_D)
Q_ab = Q_ab1 +Q_ab2 +Q_ab3
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%
% Heat Generated in the Process (b-c):
% The process b-c takes place at constant pressure,

% At point (c)
T_c = 90; %dC (methanol is in superheated vapour state)
P_c= 0.844; % bar
h_c = 1855; % kj/kg (enthalpy of methanol vapour)
% At point M
T_M= 68.5; %dC (methanol is in superheated vapour state)
h_M = 1850; % kJ/kg (enthalpy of methanol vapour)
% Q_bc= Q_bc1+Q_bc2+Q_bc3        ...              (3.9)
% Q_bc1 = Heat required to raise the temperature of the
steel plate and the
```

```
% charcoal from the temperature at (b) to 90? at (c)
Q_bc1= (m_Ch*C_Ch+ m_S*c_s) *(T_c- T_b)

Q_bc2=m_bc*(h_c- h_b)

% Q_bc3 Heat required to evaporate the working amount of
methanol
Q_bc3 = m_u*(h_c- h_M)
Q_bc = Q_bc1+Q_bc2+Q_bc3
Q_T=Q_ab+Q_bc
% 3.7.3- Heat Rejected in the Process (c-d):
% At point (c); (methanol is in superheated vapour
condition)
T_c = 90; % °C
P_c = 0.844; % bar
h_c = 1855; % kJ/kg (enthalpy of methanol vapour)
% At point (d); (methanol is in superheated vapour state)
T_d = 50; % °C
h_d= 1810; % kJ/kg
% At point (D2); (methanol is in superheated vapour state)
T_D2 = 63.5; % °C
h_D2=1810; %kJ/kg
% Q_cd= Q_cd1 + Q_cd2+Q_cd3 +....                (3.14)
% Q_cd1 = Rejected heat required to lower the temperature
of the steel plate
% and the charcoal from the temperature at (c) to 50 °Cat
(d)
Q_cd1= (m_Ch*C_Ch+ m_S*c_s) *(T_c- T_d)
% Substituting for the various values for the equation
above,
% = [(19.25 x 0.709) + (33 x 0.46)] (50 -90) - 1153.12 kJ
% Q_cd2 = Sensible heat rejected to lower the temperature
of the liquid
```

```
% methanol in the collector from temperature at (TC) to T_ (d)
Q_cd2=m_d*(h_c-h_d)

% Q_cd3 = Heat released to desorb the pressurizing amount of methanol
Q_cd3 = m_p*(h_c - h_D2)
Q_cd= -(Q_cd1+Q_cd2+Q_cd3)

% 3.7.4- Heat Rejected in the Process (b-a): -
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

% At point (M2);
T_M2 = 40; % °C, (methanol is in superheated vapour state)
h_M2 = 1780; % kJ/kg
% Q_da = Q_da1 + Q_da2 + Q_da3 +......
% Q_da= Heat rejected to lower the temperature of the steel plate and
% the charcoal from the temperature at (d) to 30 °Cat (a)
Q_da1 = (m_Ch*C_Ch+ m_S*c_s) *(T_a- T_d)

% Q_da2 = Heat rejected to lower the temperature of the liquid methanol in the collector from T_ (d) (50 °C) to ambient temperature at (a)
Q_da2=-m_da*(h_d-h_a)

% Q_da3 = Heat rejected to desorbed the evaporated amount of methanol from the cooling process in the evaporator.
Q_da3 = -m_u*(h_d - h_M2)
Q_da=Q_da1+Q_da2+Q_da3

Q_Tot=Q_cd+Q_da
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%
% DESIGN;

% COLLECTOR PLATE:
SP_w=1; % m steel plate width
M_meth=19.25; %kg total mass of methanol required for
charging the system
Ch_den=385; %density of charcoal

%the volume required for accommodating the charcoal can be
calculated as:
V = M_meth/Ch_den
% the height of the charcoal box (t) will be:
t=V/SP_w;
Box_Thickness_cm=t*100
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%
%CONDENSER DESIGN:
h_e=650; %kJ /kg
% The amount of heat released by m_u kg of methanol
condensing (c - e) is
Q=m_u*(h_c-h_e)
% Condenser pipe calculations: For estimating heat transfer
coefficients
%for both sides, (water and methanol) following equations
are used:
%1-methanol side: Using empirical equation for natural heat
transfer coefficient in a horizontal tube [Ref. 16]:
%================
% h_m= heat transfer coefficient on the methanol side
W/m^2K

g =9.81; % m /s^2 gravitational acceleration
```

```
pl=752.67; %kg/m^3 density of liquid methanol at
condensation temperature
pv = 0.757; % kg/m^3 density of methanol vapour
K = 0.2033; % W/m kthermal conductivity of liquid methanol
h_fg = 1110.28; % kJ/kglatent heat of condensation of the
methanol
u_1 = 2735e-4; %(N.s)/ m^2 Viscosity of the liquid methanol
Tv = 50; %Temperature of the methanol vapour °C
Tw = 60; % Temperature of the water °C
Dpi= 0.007; %Inner diameter of the pipe (m)
h_m=0.55*((g*(pl-pv) *(K^3) *h_fg)/(u_1*(Tw-Tv) *Dpi))
^0.25

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%
% 2. For the water side:
%   ====================
D_po = 0.008; % outer diameter of the tube (m)
% Nu=c(XX)^m ………                              (3.23)
x = 0.001; %the thickness of the tube m
dt= 5; % temperature gradient of the water = (30 - 35
YY=3.3e10; % =(g*beta*rho^2*cpw*/(u*k0))
XX= YY*(x^3*dt); % XX=pr.Gr
rho=999.6; % Density of the water at the given temperature
kg/m^3
cpw= 4.178; %specific heat of water kJ/kgK
u = 6.82*10^ (-4); %dynamic Viscosity of the water kg/m.s
K=43; % thermal Conductivity of the water)
do=0.007;
h_w = 1.32*(dt/do) ^ (1/4)
k_c = 0.644; % W/m.k,
R_fm= 0.0002; %fouling factor for methanol side
R_fw = 0.0005; %   fouling factor for water side [Ref. 16]
h_mw=338;
```

```
HH= (1/h_mw) +(1/h_w) +(x/K) +R_fw+R_fm
U=1/HH
ti=hr*3600; % working time
Qp = Q*1e3/ti %heat transfer rate (w)
 theta=16.83; %K logarithmic temperature difference

% A = heat exchange area
A = Qp/(U*theta)
D =0.008; %diameter of the tube
% L= length of the pipe, from which;
L= A/(pi*D)
% EVAPORATOR DESIGN:
% =================
% Taking m_sr= 9.0 kg and the remaining quantities
% from thermodynamic tables we obtain m_ML(f) = 1.58 Kg per
square meter of the collector,
M_u5     = 1.578; % kg
h_f5= 626.25; % kJ/kg and
h_g5= 1738.3; % kJ/kg
Qo = M_u5*(h_g5-h_f5)
plm=813.0; %kg/m^3 density of liquid methanol at
evaporation temperature
pvm = 0.0478; % kg/m^3 density of methanol vapour
Km = 0.206; % W/m kthermal conductivity of liquid methanol
h_fgm = 1209.2; % kJ/kglatent heat of condensation of the
methanol
u_1m = 8.17e-7; %(N.s)/ m^2 Viscosity of the liquid
methanol
Tvm = 20; %Temperature of the methanol vapour °C
Twm = -3; % Temperature of the water °C
Lpi= 0.03145; %Characteristic dimension (m) = (A/P)
h_mm=0.55*((g*(plm-pvm)   *(Km^3)   *h_fgm)/(u_1m*(Tvm-Twm)
*Lpi)) ^0.25
```

```
% 2. For the water side:
%    ===================
h_we = 1.23*(dt/Lpi) ^.25
HHe= (1/h_mw) +(1/h_we) +(x/K) +R_fw+R_fm
Ue=1/HHe
q=Qo*1e3/ti
theta_e=12
Ae = q/(Ue*theta_e)
% A = heat exchange area
V_ac = 0.005; % m^3 the volume of the box = 5 litters
d_ac=0.05; % m depth of the box
A_ac = V_ac/ d_ac
%Check Design
D_A=A_ac-Ae
% Therefore, (A_ac-Ae) <0 (rejected design)
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%
% Calculations of the product (Ice):
% ===================================
t_sh=0.001; %m
rho_sh=7833; %sheet density
M_sh=Ae*t_sh*rho_sh
C_wsh =4.18; % specific heat of water kj/kg
dT_e=30;
C_sh=0.46;
h_fgw=332.4;
A_coll=1:10;
QR=1783.917*A_coll; %watt
% QR=1783:1800;
Qe1=M_sh*C_sh*dT_e;
Qe2 = C_wsh*dT_e;
Qe3 =h_fgw;
Qe23=Qe2+Qe3;
M_Ice=(QR/Qe23);
```

```
plot(M_Ice,A_coll)
% slim ([0 15])
xlim([0 40])
xlabel('Ice(Kg)')
ylabel ('Collector Aera(m^2)')
title ('COLLECTOR CHART')
grid on
X_time= [0;5;6;9;10;12;15;18;20;24]
y_coll= [37;34;31;45;75;90;90;35;36;37]
y_cond= [32;31;30;30;35;40;40;40;35;33]
y_eva= [-3; -3; -2; -2;30;30;30;30; -2; -3]
y_zer= [0;0;0;0;0;0;0;0;0;0]
y_max= [90;90;90;90;90;90;90;90;90;90]
figure
plot(X_time,y_coll)
lime ([0 95])
xlim([0 24])
xlabel('Time(Hour)')
ylabel('Temperature(DC)')
title ('COLLECTOR CHART')
grid on
figure
plot(X_time,y_cond)
ylim([0 45])
xlim([0 24])
xlabel('Time(Hour)')
ylabel('Temperature(DC)')
title ('CONDENSOR CHART')
grid on
figure
plot(X_time,y_eva)
ylim ([-5 40])
xlim([0 24])
xlabel('Time(Hour)')
```

```
ylabel('Temperature(DC)')
title ('EVAPORATOR CHART')
grid on
figure
plot(X_time,y_coll)
ylim ([-5 95])
slim ([0 24])
xlabel('Time(Hour)')
ylabel('Temperature(DC)')
title ('TIME CHART')
hold on
plot(X_time,y_cond)
hold on
plot(X_time,y_eva)
hold on
plot (X_time, y_zer,'r')
 text   (15.5,88,'\leftarrow                      Collector
Curve','FontSize',9)
 text   (10.5,35,'\leftarrow                      Condenser
Curve','FontSize',9)
 text   (10,20,'\leftarrow                       Evaporator
Curve','FontSize',9)
hold on
plot(X_time,y_max)
grid on
```

APPENDIX B

MATLAB RESULTS

```
Q_ab1 =

  864.8475

Q_ab2 =

   36.5110

Q_ab3 =

    0.7290

Q_ab =

  902.0875

Q_bc1 =

  864.8475

Q_bc2 =

  207.5715

Q_bc3 =

    7.7850
```

```
Q_bc =

   1.0802e+03


Q_T =

   1.9823e+03


Q_cd1 =

   1.1531e+03


Q_cd2 =

  165.3750


Q_cd3 =

    4.0500
Q_cd =

  -1.3226e+03

Q_da1 =

 -576.5650

Q_da2 =
```

```
  -84.2400

Q_da3 =

  -46.7100

Q_da =

 -707.5150

Q_Tot =

  -2.0301e+03

V =

    0.0500

Box Thickness cm =     5

Q =

   1.8762e+03

h_m =

   23.9480

h_w =

    6.8240

HH =
```

```
    0.1502


U =


    6.6568


Qp =


   65.1453


A =
    0.5815



L =
   23.1363



Qo =
   1.7548e+03



h_mm =


  338.0092



h_we =
    4.3676



HHe =
    0.2326
```

```
Ue =
    4.2985


q =
   60.9311

theta_e =
    12

Ae =
    1.1813

A_ac =
    0.1000

D_A =
   -1.0813

M_sh =
    9.2528
```

APPENDIX C

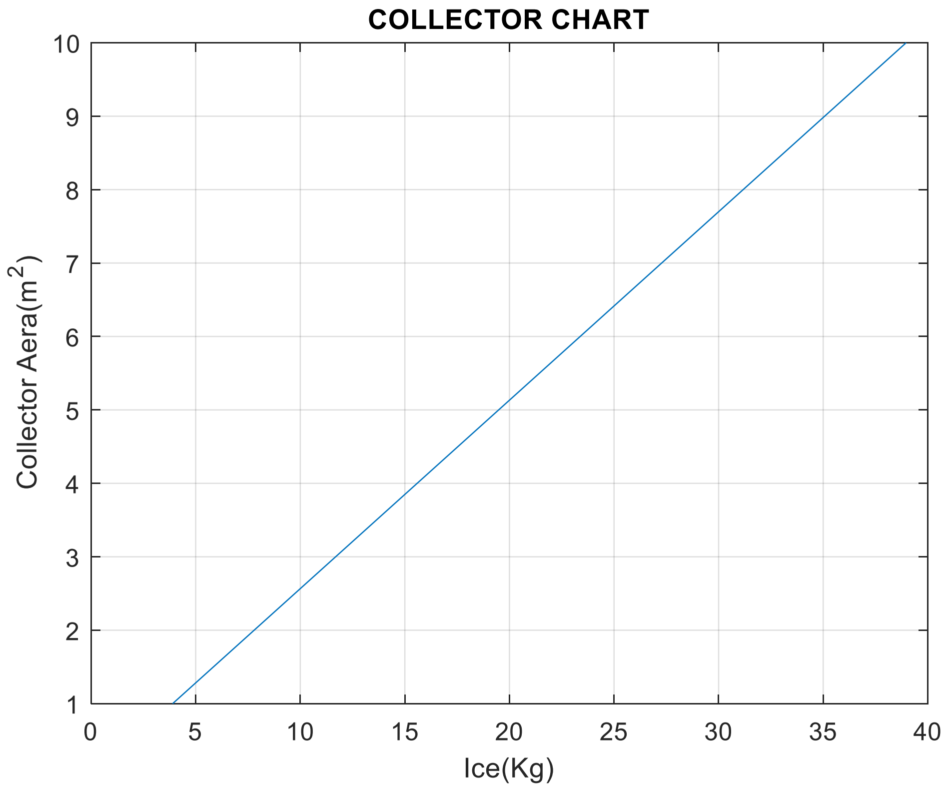


Fig 5.1 collector chart

This part is in a form of a flat-plate collector, its purpose is to absorb the solar thermal power from the sun to heat up charcoal particles packed in a box attached to the lower part of the collector-plate, this heat will generate refrigerant vapor (methanol) which will then be condensed to liquid in the condenser and collected for use in the productive time later.

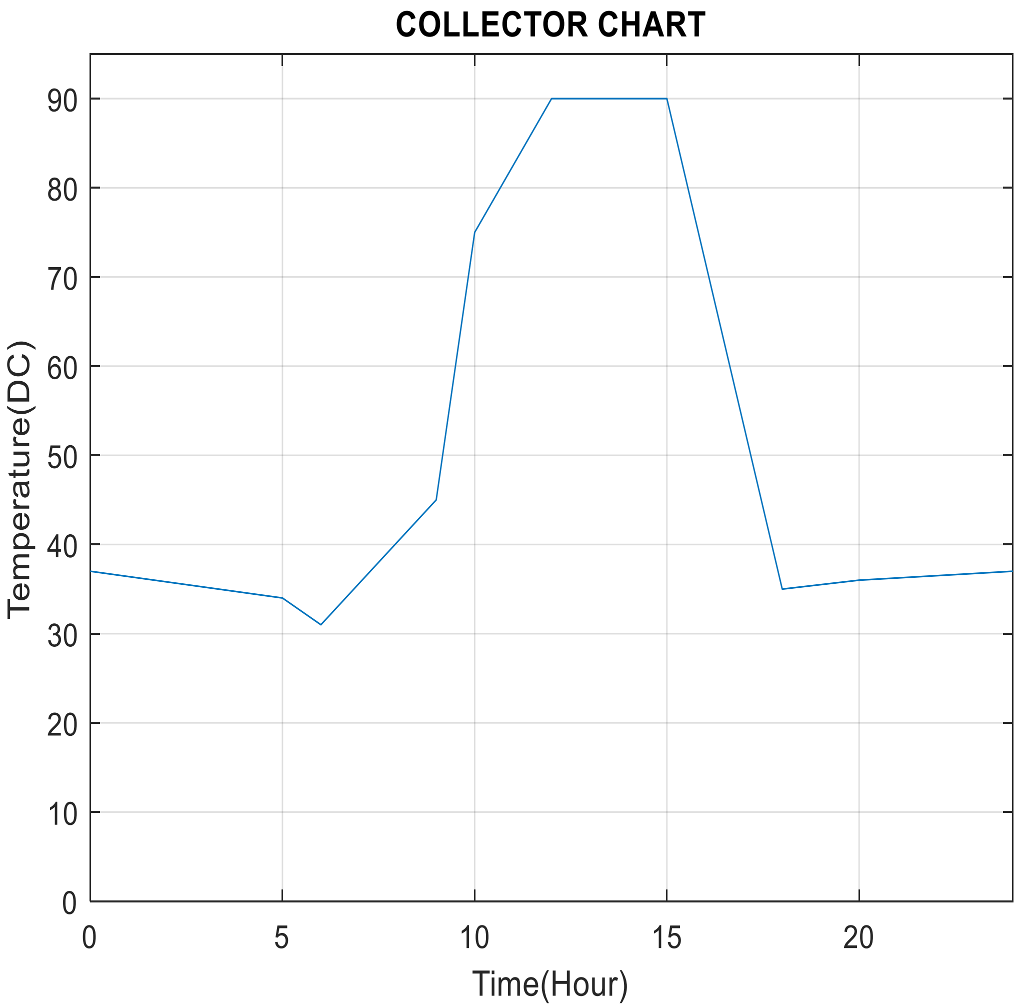


Fig 5.2 collector chart

the quantity of ice making is 5 kg when the temperature reached 90 degrees in collector this heat will generate refrigerant vapor (methanol) which will then be condensed to liquid in the condenser and collected for use in the productive time later.

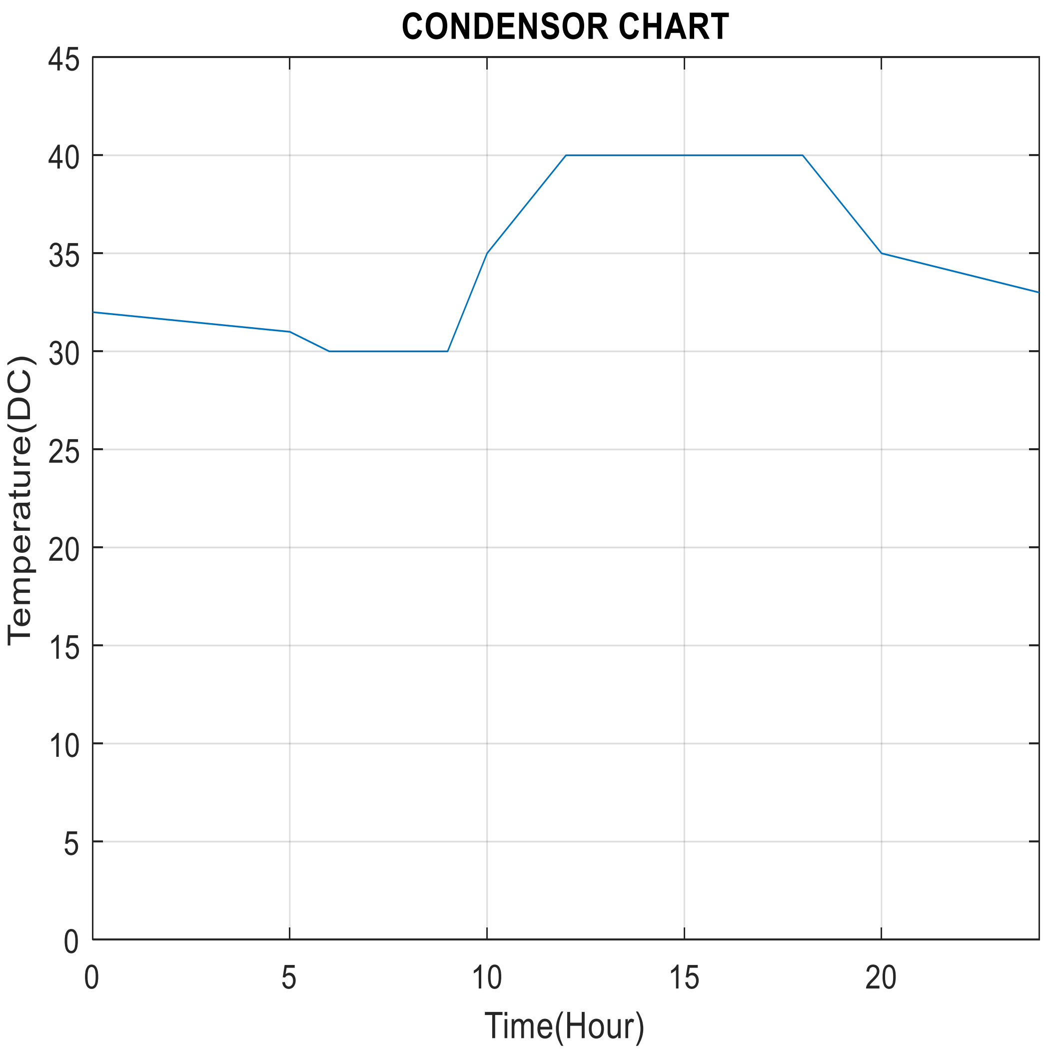


Fig 5.3 condenser chart

In condenses the generated methanol vapor coming from the collector using immerged coil in an open container having water.

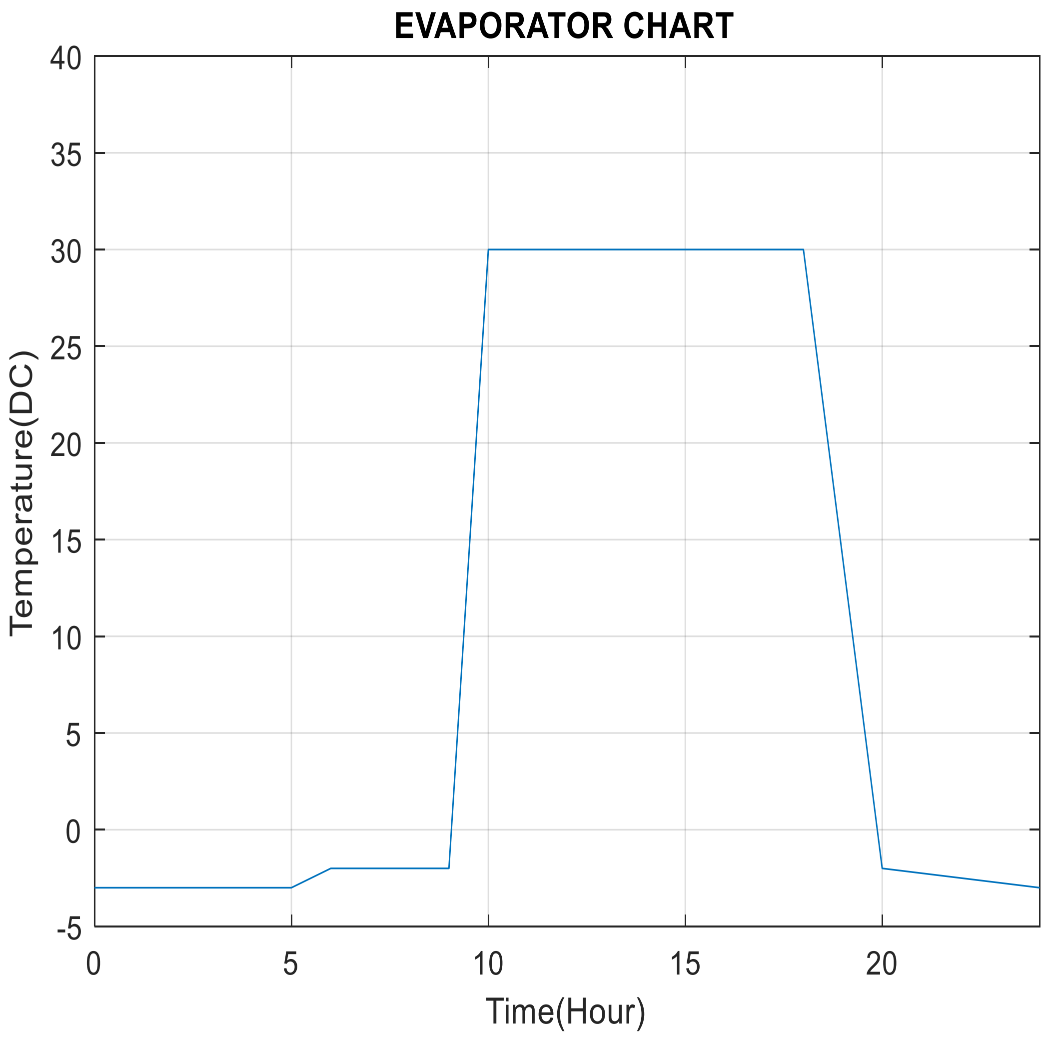


Fig 5.4 evaporator chart

This part works as a storage tank for the condensed methanol liquid during the day and as an evaporator during the night (productive time) where the flow of the refrigerant will reverse. There will be an ice mould box attached to this unit at the bottom, water filled in will act as the load, ice produced there is the product.

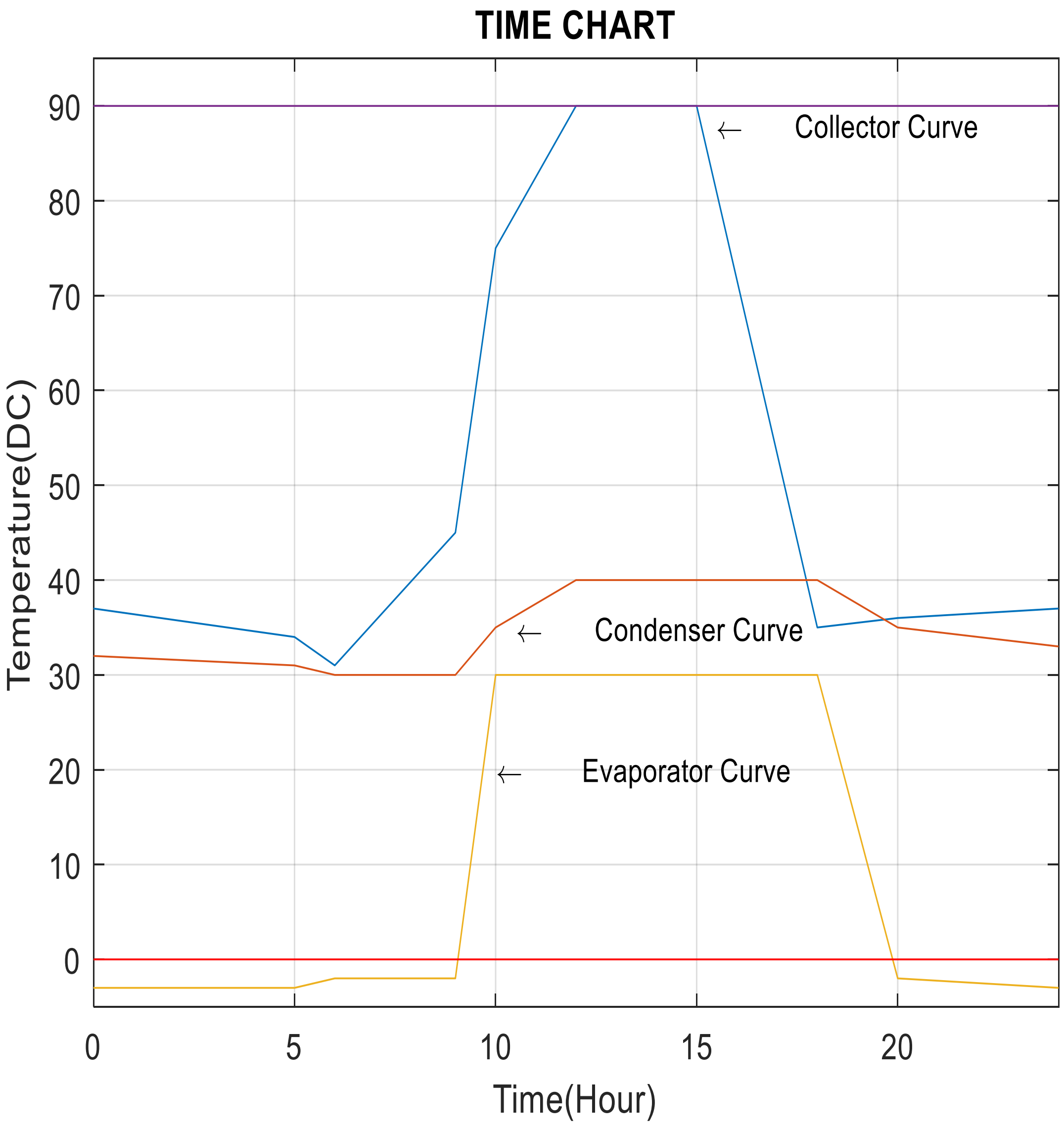


Fig 5.5-time chart

The unit will work in an intermittent cycle of operation. During the day when the sun is shining, generation is taking place and the whole unit will be at condenser pressure and the direction of flow of methanol will be from collector to the receiver across the condenser. During the night, shutters are to be opened to cool down the collector, and methanol will flow back to collector at evaporation pressure, and this

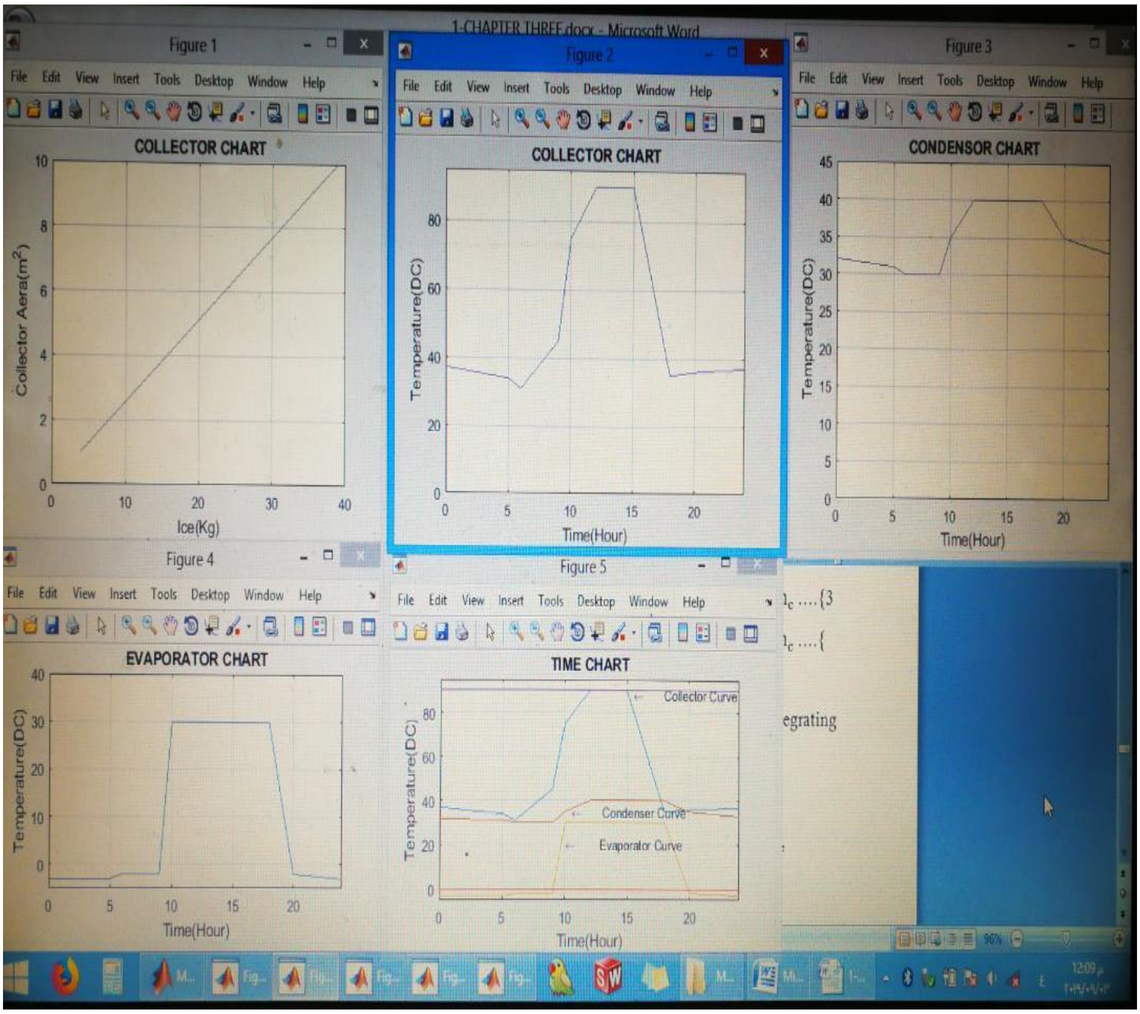


fig 5.6 mat lap charts

APPENDIX D

(D)A.I.T. SUMMARY OF RESULTS TABLES **[**Simulation Solar Powered Charcoal/Methanol, 1992**]**

Table (1) Constant used in the Calculations

Constant	**DATES**			**Mean**
	14 June	**15June**	**12 July**	
WCE	55.259	737.75	542.97	611.0
WCED	50.121	26.136	43.162	40.4
HI(W/M ^{2}K)	50.47	45.5	45.4	47.5
K(W/mk)	0.175	0.175	0.175	0.175
a_1	0.953	0.95	0.953	0.953
CCE	1.242	1.250	1.25	1.247
F(°C)	2.65	2.626	3.063	

A. I. T. [Excel, Simulation Solar Powered Charcoal/Methanol, 1992**]**

(D) Table(2)AIT`S MACHINE COPs AT VARIOUS CONDITIONS

Condition	cops				Te_v (max)	Note
	Cop1	Cop2	Cop3	Cop4		
Shutter operation: (1)**Shutters Opened**						
13:30	0.075	0.036	0.027	0.018	-8.40	1.2 kg Water in Ice tray
15:00	0.080	0.036	0.029	0.020	-8.70	
16:00	0.080	0.036	0.030	0.021	-8.95	
18:00	0.080	0.036	0.030	0.021	-8.30	
Not open	0.080	0.024	0.024	0.015	-1.10	
(2)Water ice tray :						
0.5 kg	0.076	0.016	0.015	0.012	-11.60	Shutters opened At 16.00
1.2 kg	0.080	0.036	0.030	0.021	-8.95	
2.5 kg	0.080	0.051	0.037	0.018	-6.40	
4.5 kg	0.081	0.052	0.037	0.003	-8.10	

*The actual climatic data on june1992,in AIT were used in the calculations .