DESIGN AND DEVELOPMENT OF A SOLAR REFRIGERATION SYSTEM

By

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(2013 – 18-101)

THESIS

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the requirement for the degree of

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2015

DECLARATION

I, hereby declare that this thesis entitled **"Design and Development of a Solar Refrigeration System"** is a bonafide record of research done by me during the course of research and the thesis has not previously formed the basis for the award to me of any degree, diploma, associateship, fellowship or other similar title, of any other University or Society

Tavanur

Date: 13/08/2015

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CERTIFICATE

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Rakhi, J. F

DEDICATED TO AGRICULTURAL ENGINEERING PROFESSION

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SYMBOLS AND ABBREVIATIONS

AM	-	Ante Meridiem
ARS	-	Absorption Refrigeration System
ASHRAE	-	American Society of Heating, Refrigerating, and Air Conditioning Engineers
CFCs	-	Chloro Fluoro Carbons
CIESOL	-	Center for Solar Energy Research
COP	-	Coefficient of Performance
CPC	-	Compound Parabolic Concentrator
DC	-	Direct Current
DAR	-	Diffusion Absorption Refrigeration
E-W	-	East-West
et al.	-	and others
g	-	gram
h	-	Hour
HCFCs	-	Hydro Chloro Fluoro Carbons
HTF	-	Heat Transfer Fluid
HFCs	-	Hydro Fluoro Carbons
H_2O	-	Water
I	-	Current
i.e.	-	that is

J	-	Joule
K	-	Kelvin
Kg	-	Kilo gram
KCal/h	-	Kilo Calorie per hour
KJ/Kg	-	Kilo Joule per Kilo gram
KJ/min	-	Kilo Joule per Minute
KN/m^2	-	Kilo Newton per Square meter
KW	-	Kilo Watt
KWh	-	Kilo Watt hour
L	-	Liter
LiBr	-	Lithium Bromide
$LiNO_3$	-	Lithium Nitrate
LPD	-	Litre Per Day
LPG	-	Liquefied Petroleum Gas
m	-	metre
m^2	-	Square metre
m^3	-	Cubic metre
Min	-	Minute
mg	-	milli gram
MJ	-	Mega Joule
ml	-	mililitre
MW	-	Mega Watt

NH_3	-	Ammonia
NIST	-	National Institute of Standards and Technology
N-S	-	North-South
P	-	Power
PM	-	Post Meridiem
Psi	-	Pounds per square inch
PV	-	Photo - voltaic
REFPROP	-	REFerence fluid PROPerties.
R -22	-	Monochloro-difluoro-methane
R-134a	-	Tetrafluoro- ethane
SACE	-	Solar Air-conditioning in Europe
SESEC	-	Sustainable Energy Science and Engineering Center
SPV	-	Solar Photo-Voltaic
TMY	-	Typical Meteorological Year
TR	-	Tone of Refrigeration
TMY	-	Typical Meteorological Year
UK	-	United Kingdom
V	-	Voltage
VAR	-	Vapour Absorption Refrigeration
VARS	-	Vapour Absorption Refrigeration System
W	-	Watt
w/w	-	Weight/Weight

°C	-	Degree Celsius
%	-	Percent

INTRODUCTION

CHAPTER 1

INTRODUCTION

Solar energy, a vast and inexhaustible source of energy, is one of the most promising sources of nonconventional energy currently available. The power from the sun received by the earth is approximately 1.8×10^{11} MW which is much larger than the present consumption rate on the earth from all commercial energy sources, so solar energy could supply all the present and future energy needs of the world. When solar power, either thermal or photovoltaic, is used to provide energy to any refrigeration system, it is called as solar refrigeration system.

Many agricultural products like vegetables, fruits, dairy products, meat, fish etc., can be maintained in fresh conditions for longer periods of time, if they are stored under refrigeration conditions. Large quantities of agricultural products are lost annually due to lack of refrigeration facilities. Refrigeration is the process of extracting heat from a substance or space by any means usually at a low temperature (ASHRAE). Refrigeration has a large effect on agriculture, industry, lifestyle and settlement patterns of people. Refrigeration has many applications that include; storing and preservation of perishable food products, keeping medicines at low temperatures, for human comfort, etc. and also have many industrial applications such as in manufacturing sector, freezing of food products, in textile industry etc.

Refrigeration system are commonly based on vapour compression refrigeration system in which a compressor that is operated by mechanical power is a major component and also it is used in most household refrigerators and large commercial and industrial refrigeration systems. The vapour absorption cycle using water-ammonia refrigerants were popular and widely used before the popularization of the vapour compression cycle but lost its importance because of its low coefficient of performance (COP) which is about one fifth of that of a vapour compression cycle.

The principle of vapour absorption refrigeration system was developed by Michael Faraday while performing a set of experiments to liquefy certain gases. In the case of vapour compression refrigeration systems mechanical energy is used to drive the system but the vapour absorption refrigeration systems use heat energy instead of mechanical energy. Thus this system can be used where the electric power is not available, difficult to obtain or it is very expensive or where noise from the compressor is problematic, or where surplus heat is available and also it is a best alternative to regular compressor refrigerators.

The solar energy is the new source of energy for refrigeration applications in future. Solar refrigeration can revolutionize the global refrigeration scenario in future. For solar refrigeration systems operation costs will be minimum and the only operating costs would be the maintenance associated with the refrigeration system. Many regions around the world are not electrified even today and hence the application of refrigeration is limited in those regions due to the non-availability of electricity. In recent years, many countries have been facing difficulties in using refrigeration systems, as price of electricity has been exorbitantly high. The present energy crisis, high cost of fossil fuel, has opened up the door for solar energy to handle not only peak electricity demands but also cooling issues of the world. Therefore solar cooling technologies are currently a worldwide focal point.

The use of solar energy for environmental control is receiving much attention as a result of the limited storage and environmental issues of fossil fuels. The traditional vapour compression refrigeration cycles are driven by electricity or mechanical power, which increases the consumption of electricity and fossil energy. It is estimated that approximately 15% of all the electricity produced in the whole world is employed for refrigeration and air-conditioning processes of various kinds, and the energy consumption for air-conditioning systems has recently been estimated to 45% of the whole households and commercial buildings (Anirban and Randip, 2010). The conventional vapour compression system use refrigerant fluids, like ChloroFluoro Carbons (CFCs), Hydro

ChloroFluoro Carbons (HCFCs) and Hydro Fluoro Carbons (HFCs). These directly affect the ozone layer depletion and global warming. But in vapour absorption refrigeration systems use only natural refrigerants like ammonia, lithium bromide, water etc. and therefore this technique is eco – friendly and also it has no adverse effect on environment.

Solar thermal refrigeration system harnesses solar heat directly using collectors or concentrators to heat a transfer fluid in a vapour absorption refrigeration system. Solar photovoltaic refrigeration system uses solar panels that operate using the photo-electric effect to produce direct current (DC) to give mechanical energy to operate a compressor in a vapour compression refrigeration system. Thus solar energy can be used to operate both vapour compression refrigeration systems and vapour absorption refrigeration systems. However, the direct utilization of solar thermal energy, for running refrigeration systems, is more efficient. Thus heat operated vapour absorption technology has been adopted for solar refrigeration systems. It requires very low or no electric input and also the absorbent refrigerant used in this system have high heat transfer coefficient.

Absorption refrigeration system operates without any mechanical or electrical power so it may prevent the spoiling of agricultural products during storage in remote rural regions without electricity. A solar refrigerator is a cooling appliance that is operated completely with energy harnessed from the sun. A solar refrigerator can store food, medications, and other products that require cold temperatures. Solar-powered refrigerators are useful to keep perishable goods such as vegetables, meat and dairy products and also they very suitable for keeping vaccines at their appropriate temperature to avoid spoilage. Solar refrigeration systems can be effectively used in ice making, chilling of products and air conditioning systems. In view of shortage of energy production and fast increasing energy consumption, there is a need to minimize the use of energy and conserve it in all possible ways. Refrigeration systems form a vital component for the industrial growth and affect the energy requirements of the country at large.

Therefore, it is desirable to provide a way for energy conservation and energy recovery through vapour absorption refrigeration systems. Mechanical vapour compression refrigeration requires high grade energy for their operation. Apart from this, recent studies have shown that the conventional working fluids of vapour compression system are causing ozone layer depletion and green house effects. However, vapour absorption refrigeration systems are harmless, inexpensive as it uses waste heat, solar energy, biomass or geothermal energy sources for which the cost of supply is negligible. Moreover, the working fluids of these systems are environmental friendly.

Considering the above facts and situations, the present study was undertaken to Design and Develop a Solar Refrigeration System with the following objectives:

- Design and development of a Solar Refrigeration System.
- Testing of the developed Solar Refrigeration System.
- Economic analysis of the developed Solar Refrigeration System.

REVIEW OF LITERATURE

CHAPTER 2

REVIEW OF LITERATURE

A critical comprehensive review of literature is inevitable for any scientific investigation. A brief report of research works carried out on the design and development of the solar refrigeration system is presented in this chapter. This chapter provides the background information on the issues to be considered in the present research work and to focus the relevance of the present study.

2.1 Methods of refrigeration

Before the invention of refrigeration systems, people cooled their food with ice transported from the mountains and also used snow cellars and pits dug into the ground and insulated with wood and straw to store the ice. Stored ice was the principal means of refrigeration until the invention of refrigeration. In India and Egypt evaporative cooling was used before the invention of refrigeration systems (Anderson and Edward, 1953).

Methods of refrigeration can be classified as non-cyclic, cyclic, thermoelectric and magnetic. In non-cyclic refrigeration, cooling is accomplished by melting ice or by subliming dry ice (frozen carbon dioxide). In cyclic refrigeration heat is removed by means of phase change of refrigerant, it circulates through a refrigeration system and alternatively absorbs and rejects heat. Cyclic refrigeration also classified as vapour cycle, and gas cycle. Vapour cycle further classified as vapour-compression refrigeration and vapour-absorption refrigeration. In gas cycle, working fluid is a gas, air is most often used that is compressed and expanded but doesn't change phase (Burstall, 1965).

2.1.1 Vapour compression refrigeration system

In a vapour compression refrigeration system, low temperature and pressure vapour refrigerant from the evaporator is drawn into the compressor, where it is converted into high pressure and high temperature vapour refrigerant, then it is discharged into a condenser. In the condenser the vapour refrigerant gives latent heat to the surrounding medium and it is converted into liquid refrigerant. This high pressure, temperature liquid refrigerant is again converted into low pressure and low temperature liquid while passing through the expansion valve and in the evaporator it is again converted into low pressure and low temperature vapour refrigerant by receiving latent heat of vaporization from the medium which is to be cooled (Khurmi and Gupta, 2011).

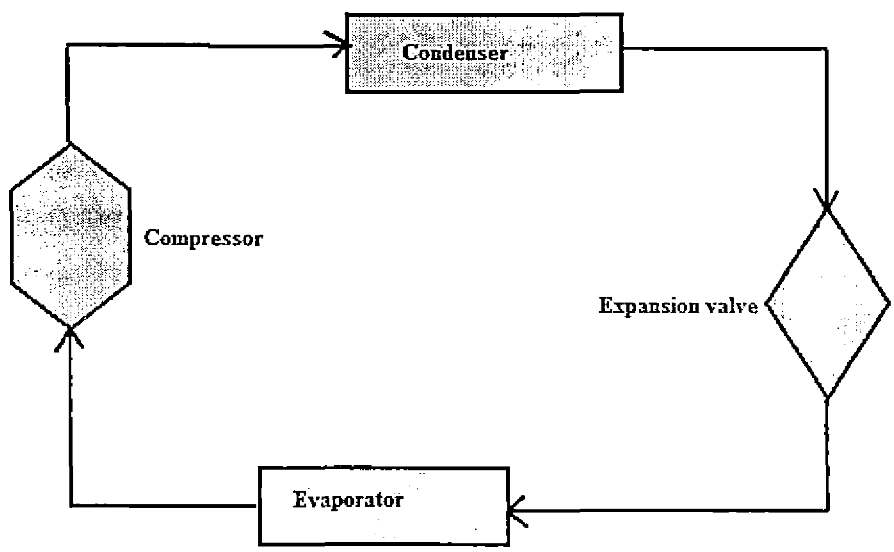


Figure 2.1 Schematic diagram of vapour compression refrigeration system

2.1.2 Vapour absorption refrigeration system

The vapour absorption system uses heat energy to produce cooling effect and also in this system the compressor is replaced by an absorber, a pump, a generator and a pressure reducing valve. Working of this system is such that vapour refrigerant from the evaporator is absorbed by an absorbent in the absorber forming a strong solution. This strong solution is pumped to the generator where it is heated by using an external heat energy source. During the heating process, the vapour refrigerant is driven off from the concentrated aqua - ammonia solution and the vapour refrigerant enters into the condenser where it is liquefied. Then the liquid refrigerant flows into the evaporator where it takes latent heat of vaporization from the surroundings and changes from liquid phase to vapour phase. Cooling takes place here. The vapour refrigerant then passes to absorber and it is absorbed by the liquid in the absorber making it a concentrated solution. Also, after the separation of refrigerant in the generator, the weak solution return back to the absorber and thus the cycle is completed (Bell *et al.* 1996)

In ammonia vapour absorption refrigeration system, ammonia is used as the refrigerant and water is used as the absorbent, because ammonia is highly soluble in water and form aqua- ammonia solution. Working of the system is based on the vapour absorption refrigeration cycle (Raghuvanshi and Maheswari, 2011).

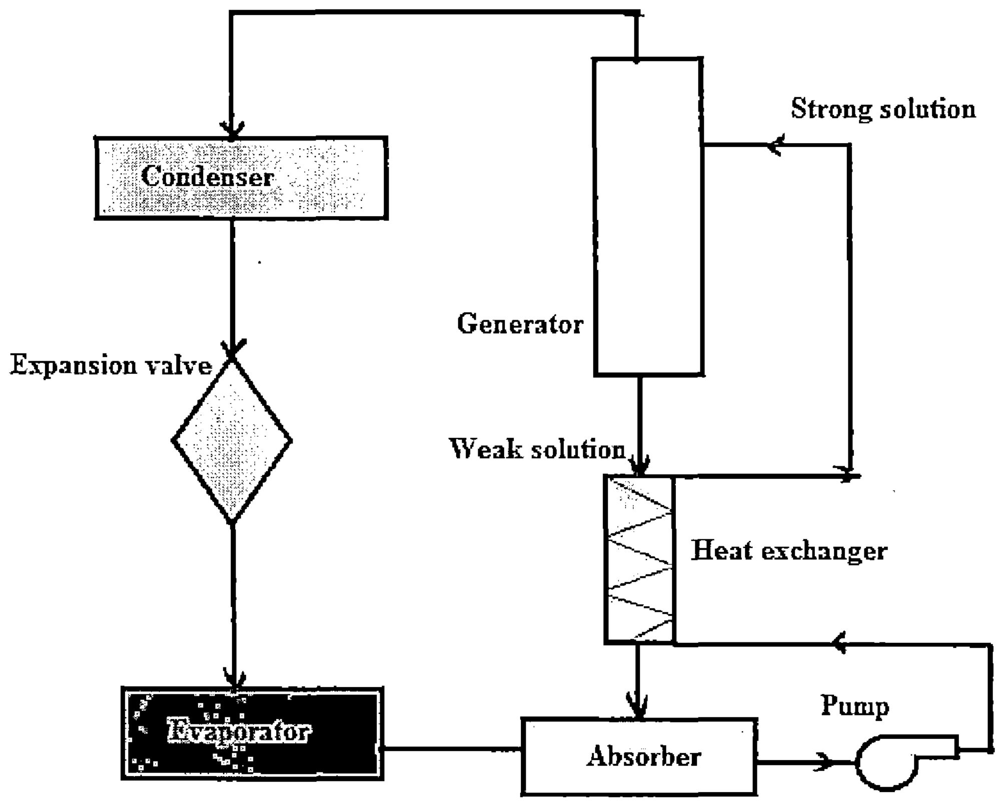


Figure 2.2 Schematic diagram of vapour absorption refrigeration system

Vapour absorption refrigeration systems using water-lithium bromide pair are extensively used in large capacity air conditioning systems. In these systems water is used as refrigerant and a solution of lithium bromide in water is used as absorbent. This system was not suitable for producing refrigeration at sub-zero temperatures because water is used as the refrigerant so these systems are used for applications requiring temperatures above 0°C like air conditioning applications. In ammonia absorption refrigeration systems, ammonia and water vapour are generated in the generator but in this system vapour generated in the generator is almost pure refrigerant (water), unlike ammonia-water systems. (Kalogirou *et al.* 2001).

In two-fluid vapour absorption cooling system the strong solution in the absorber is pumped to the generator by the liquid pump to increase the pressure of solution, which requires some amount of mechanical work. But in the three fluid vapour absorption refrigeration system by adding a third fluid, the necessity of pump is eliminated thus completely eliminating all moving parts (Khurmi and Gupta, 2011).

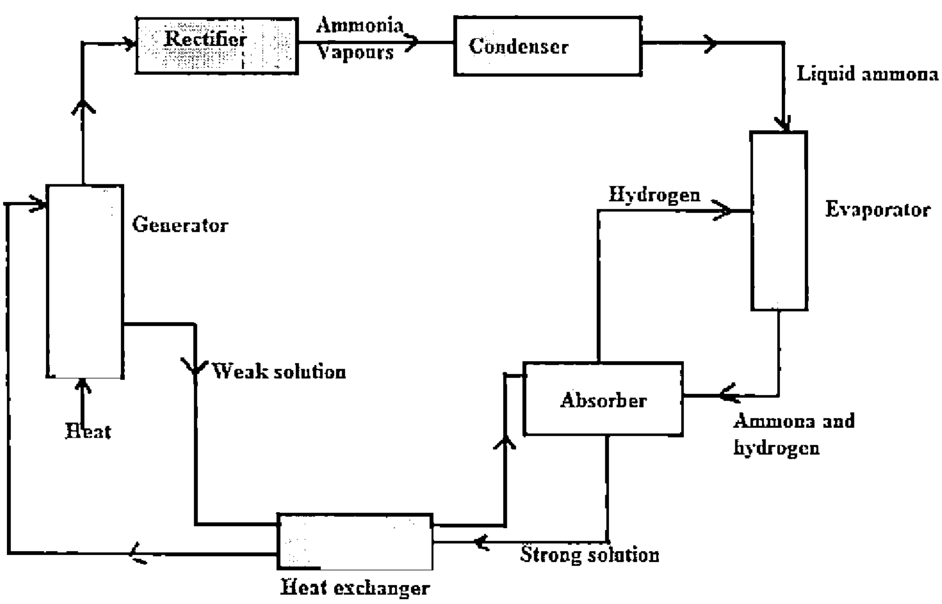


Figure 2.3 Schematic diagram of three fluid absorption refrigeration system

Three fluids used in three fluid absorption refrigeration systems are ammonia, hydrogen and water. Ammonia is used as a refrigerant, hydrogen is used to increase the rate of evaporation of the liquid ammonia passing through the evaporator and water is used as a solvent because it has the ability to absorb ammonia readily. The partial pressures of the refrigerant vapour (in this case ammonia) must be low in the evaporator, and higher in the condenser. The total pressure throughout the circuit must be constant so that the only movement of the working fluid is by convection currents. The refrigeration system working was same as two fluid vapour absorption refrigeration cycle but hydrogen circulate in between evaporator and absorber (Pongsid and satha, 2002).

2.2 Solar refrigeration systems

When solar power, either thermal or photovoltaic, is used to provide energy to any refrigeration system, it is called as solar refrigeration system. Solar thermal refrigeration system harnesses solar heat directly using collectors or concentrators to heat a transfer fluid in a vapour absorption refrigeration system (Sorensen, 2004).

Solar photovoltaic refrigeration system uses solar panels that operate using the photo-electric effect to produce direct current (DC) to give mechanical energy to a compressor in a vapour compression refrigeration system. Thus solar energy can be used to operate both vapour compression refrigeration systems and vapour absorption refrigeration systems. However, the direct utilization of solar thermal energy for running refrigeration systems is more efficient. Thus heat operated vapour absorption technology has been frequently adopted for solar refrigeration (Horuz, 1999).

2.2.1 Solar absorption refrigeration systems

In the developing countries there is a growing interest in refrigeration for food preservation. Especially in rural areas, simple solar refrigerators working independently, without using electrical energy has been very suitable. Mechanical refrigerators powered by solar photo voltaic cells are available, but are too expensive and hence low cost heat operated vapour absorption technology has been frequently adopted for solar refrigeration (Abdul, 1984).

2.2.2.1 Ammonia – water absorption refrigeration system using solar energy

In solar ammonia - water absorption refrigeration systems, hot water or steam from flat- plate collector array or concentrating collectors is passed through a heat exchanger, where it transfers heat to the aqua-ammonia solution in the generator and ammonia is liberated from the aqua ammonia solution by using this heat energy (Rai, 2010).

Chinnappa (1962) built a simple intermittent refrigerator operated with a flat-plate collector. An ammonia-water solution was used as the working fluid. He reported that the flat-plate collector would be more suitable for the lower temperature of generation required in air conditioning, and also he indicated that it is possible to use a flat-plate collector incorporated with the generator to produce refrigeration at a temperature as low as -12°C. It is noted that ice can be produced in this refrigerator at one kg a day per 0.7 $m^2$of solar collecting surface.

The major project on a solar absorption refrigeration system was undertaken by Trombe and Foex (1964). Ammonia-water solution allowed to flow from a cold reservoir to a pipe placed at the focal line of a cylinder-parabolic reflector having area of 1.5 m^2 and this pipe acted as a generator. Heated ammonia-water vaporized in this generator pipe and condensed in a cooling coil. The container surrounding the evaporator coil was used as an icebox. Daily production was about 6 to 4 kilograms of ice per square meter of collecting area for four-hour heating.

Farber (1970) has built one of the most successful solar refrigeration systems. It was reported that an average of about 42,200 KJ of solar energy was collected by the collector per day andthe ice produced was about 18.1kilograms. This gave an overall coefficient of performance of about 0.1and 12.5 Kg of ice per m^2ofcollector surface per day.

Swartman and Swaminathan (1971) built a simple, intermittent refrigeration system incorporating the generator-absorber with a 1.4 m^2 flat plate collector. Ammonia water solutions of concentration varying from 58 to 70 percent were tested. Tests were relatively successful; evaporator temperatures were as low as -12°C.

Exell *et al.* (1976) designed and developed a small ammonia-water intermittent absorption refrigerator with a 1.44 m^2 flat plate solar collector for ice making. In the generator 15 Kg of solution containing 46% ammonia in water are used. On a clear day the solution temperature rises from 30°C, to 88°C and 0.9 Kg of pure ammonia is condensed at 32°C. During refrigeration the temperature of the ammonia drops to -7°C. The estimated overall solar coefficient of performance (cooling effect divided by solar heat absorbed) was 0.09.

Staicovici (1986) made an intermittent single-stage H_2O-NH_3solar absorption system of 46 MJ/cycle. Solar collectors were used to heat the generator. The system coefficient of performance (COP) varied between0.152 and 0.09 in the period of May–September. Actual (COP) system values of 0.25–0.30 can be achieved at generation and condensation temperatures of 80°Cand 24.3°C respectively.

Tyagi (1988) carried out the detailed study on an aqua-ammonia VAR system and calculated the coefficient of performance, mass flow rates as a function of operating parameters i.e. absorber, evaporator and generator temperatures. He showed that COP and the work done are the functions of evaporator, absorber, and condenser and generator temperature and also depends on the properties of binary solution.

Kouremenos *et al.* (1990) made a laboratory modal of absorption refrigeration. Using ammonia - water solution at 52 %concentration by weight and with a total weight of 38Kg, this system was operated using heat source. A heat source at temperature no higher than 80°C was used to simulate the heat input to absorption refrigeration from solar pond. In this system the temperatures of the generator was 73°C and evaporator temperatures as -2°C. Tap water was used to remove the heat generated from the condensation of the ammonia vapour and the absorption of the refrigerant in the water. The temperature of the tap water was near the ambient laboratory temperature of 28°C. The COP for this unit working under such condition was in the range 0.24 to 0.28.

Rogdakis and Antonopoulos (1992) studied a two-stage solar $NH_3 - H_2O$ absorption-refrigeration system, which was used to produce refrigeration temperatures as low as -7°C.

Hammad and Habali (2000) designed a solar-powered absorption refrigeration cycle using NH_3 –H_2O solution to cool a vaccine cabinet. Results indicated that thermal COP ranged between 0.5 and 0.65 with generation temperature at 100- 120° C and cabin inside temperature was 0-8°C.

Francisco *et al.* (2002) developed and tested a proto type of 2 KW NH_3 – H_2O absorption system in Madrid for solar-powered refrigeration in small rural operations. The test results showed that unsatisfactory operation of the equipment with COP lowers than 0.05.

The modeling, thermodynamic simulation and second law analysis of an ammonia-water solar double-effect, double generator absorption chiller was presented by Ezzine *et al.* (2004). Computer simulation was carried out in order to determine the stream properties and the amount of heat and work exchanged with the surroundings. Also, simulation results were used to study the influence of the various operating parameters on the performance coefficient of the chiller. Results indicated that the absorber, solution heat exchangers, and condenser have the most potential to improve chiller energy efficiency.

Alili *et al.* (2008) modeled and optimized a solar powered air conditioning system in Abu Dhabi. The system used evacuated tube collectors to drive a 10 KW, NH_3-water absorption chiller. The results showed that the minimum heater consumption was 1845 KWh. It was also found out that cost savings and heater consumption reduction of up to 24.5% can be achieved compared to the vapour compression system.

Jia *et al.* (2009) proposed a novel solar-powered absorption air conditioning system used a solar collector area of 19.15m^2. COP of the air conditioning system was obtained as 0.4372 and the efficiency of entire system (including the solar panel) was obtained as 0.7771 and 0.4372.

Jacob and Eicker (2010) developed a solar powered ammonia water absorption cooling machine. This machine was designed for 2.5 KW cooling capacity at temperature between -10°C and 5°C with indirect heat through compound parabolic concentrator vacuum tubes collectors. The indirectly heated solar generator represents the main features of the cooling machine. Measurements were done with different evaporator temperature from 0 – 25°C and COP obtained between 0.2 and 0.3.

Bajpai (2012) designed a solar powered vapour absorption system of 1TR capacity, the heat input required to run the 1 TR vapour refrigeration system was 304.2 KJ/min. This heat in the generator is supplied by the hot water coming from the solar flat plate water heater. For this system the coefficient of performance is also calculated. COP of refrigerating unit is 0.69 the feasibility of the solar powered vapour refrigeration system has been proved.

Patel *et al.* (2012) experimentally developed a solar vapour absorption refrigeration system of 0.25 TR capacity. Results indicated that the refrigeration system of designed capacity was not completely achieved during the practical design as compared to the theoretical design due to certain parameters such as less number of turns of condenser and tube length resulted in inefficient heat rejection. This caused the hot vapour from the generator to enter the evaporator coil without changing its phase completely and thus reduced the cooling effect and also the system couldn't sustain desired pressure. Another limitation was concentration of ammonia in this system, was 50% but in the commercially available system the concentration was 25%.

Hyginus (2012) developed and studied a solar micro-absorption refrigerator; this study contributes an important knowledge and method in the development, fabrication and application of an absorption refrigerator as a better alternative to the commonly used compressor refrigerators. Also, it focuses on the selection of a suitable refrigerant that is economically friendly in order to reduce or eliminate its ozone depleting effect. Consequently, the design was fabricated using adapted locally sourced materials. The performance of the machine generally is very efficient as its calculated coefficient of performance (COP) is 1.21. Also, the total cost including an over-head of 30% of the machine was estimated. Cost estimation indicated that the machine was affordable to all, and was highly recommended for local entrepreneurs for mass production because of its cost effectiveness, simplicity and availability of spare parts.

Lavanya *et al.* (2013) designed a solar water cooler of 200 litres capacity. In this aqua ammonia absorption system they used ammonia as the refrigerant and water as the absorbent. In this system, ammonia is evaporated from the solution and then it is condensed in an outdoor condensing coil. The refrigerant is then expanded and evaporated in the evaporator at low pressure, producing cooling effect.

Labus *et al.* (2013) reviewed on absorption technology that focused on small capacity absorption machines. The aim of this study was to review the past achievements in the field of absorption systems, their potential and possible directions for future development. Various types of absorption systems and research on working fluids are discussed in detail. Among various applications, solar assisted air-conditioning and refrigeration, and combined cooling heating and power are identified as two most promising applications for further development of absorption machines.

2.2.2.2 Water-lithium bromide absorption refrigeration system using solar energy

Lithium bromide absorption refrigeration system use water as refrigerant and lithium bromide as absorbent so heat energy available from sun was used to regenerate water from the lithium bromide solution in generator. Water is used as the refrigerant so this system and it was only applicable for the temperature required above 0° C. Mainly it was used for air – conditioning applications (Ercan, 1991).

Mahieddine (1989) developed an absorption refrigeration cycle employing a lithium bromide-water solution as the working fluid. Solar thermal systems providing a low temperature heat source for absorption cooling units have been also examined. An aqueous Li-Br absorption cooling cycle, with hot water as a heat source and cooling water as a heat sink was then modeled and optimized. The generator, condenser, evaporator, absorber and solution heat exchanger cycle components have been designed.

Assilzadeh *et al.* (2005) carried out simulation and modeling of a solar absorption system in Malaysia and found out that continuous operation and increase in system reliability could be achieved through use of a $0.8m^3$ hot water tank. They concluded that the optimum system for Malaysia's climate for a 3.5 KW system consists of $35m^2$ evacuated tube collectors tilted at 20^0. It was also observed that air collectors are not cost effective for solar cooling applications because heat exchange surface areas required are very large

Sencan *et al.* (2005) carried out the exergy analysis of a single effect water lithium bromide absorption refrigeration system and calculated the exergy losses in the system components. The effect of heat source temperature on COP and exergetic efficiency was computed. They did not analyze the effect of variation in absorber and condenser temperature. They concluded that the cooling and heating COP of the system increases slightly when increasing the heat source temperature but the exergetic efficiency of the system decreases when increasing the heat source temperature for both cooling and heating applications.

Mittal *et al.* (2006) modeled and simulated a solar absorption cooling system using a flat plate collector, a10.5 KW LiBr-water absorption machine, using weather data for Haryana, India. The effects of hot water inlet temperatures on the COP were investigated and it was found out that high hot water inlet temperature increases the COP of the system and decreases the surface area of the components.

Guozhen *et al.* (2006) developed a mathematical model for predicting the performance of the absorption system with lithium bromide as absorbent and water as refrigerant by increasing absorption pressure. In this experiment, improved absorption cycle was adopted to raise the pressure inside the absorber of the machine in order to intensify the absorption effect of thick Lithium Bromide solution and enhance the COP of the absorption refrigeration system. A mathematical model used for predicting the performance of the system and the influence of pressure change on the overall performance of the machine.

Balaras *et al.* (2007) surveyed and analyzed over 50 solar-powered cooling projects in different climatic zones as part of the Solar Air-conditioning in Europe (SACE) project. This project was aimed to assess the state of the art and to provide a clear picture of the potential, future needs and the overall perspectives of this technology. They carried out a parametric study to examine the current cost situation for solar assisted air conditioning. For southern European and Mediterranean areas, it was found that solar air-conditioning systems can lead to primary energy savings in the range of 40-50%. It was also concluded that the cost of solar powered air conditioning in buildings and market integration promotion can be achieved through increased research and development.

Ali *et al.* (2008) carried out a performance study of an integrated cooling plant having both free cooling and solar powered LiBr-water absorption chiller. This system has been in operation since August 2002, in Oberhausen, Germany. A floor space of 270m^2 is air conditioned by the plant, which includes a 35.17 KW absorption chiller, vacuum tube collectors with total aperture area of 108m^2, hot water storage capacity of 6.8m^3 and 134 KW cooling tower. The results show that COP varies between 0.37 and 0.81.

Posiek and Batlles (2009) analyzed the solar absorption cooling system installed at the Solar Energy Research Centre (CIESOL), at the University of Almeria. A single effect 70 KW absorption chiller was used for analyzing the performance using heat energy from 64m^2 flat plate solar collector arrays. When the available solar energy is insufficient to run the chiller, heat is provided by a 100 KW auxiliary heat source. During summer the system solar field could sufficiently supply all the heating energy required. The value for the COP was 0.6 and the cooling capacity was 40 KW.

Tsoutsos *et al.* (2009) designed a solar absorption cooling system for a hospital in Greece. Four alternative scenarios were analyzed with different heating and cooling solar fractions. The scenario that optimized the financial and environmental benefits consisted of 500m^2 of solar collector surface driving a 70 KW absorption chiller, with 15m^3 hot water storage. The system had a solar fraction of 74.73% and a payback period of eleven and a half years without funding subsidies and this could be reduced to 6.9 years with a 40% subsidy. They concluded that the financial investment for these systems was high and further work needed to be carried out to reduce this cost. It was discovered that use of the solar energy powered system resulted insignificant primary energy savings.

Zhai and Wang (2009) carried out a review of absorption and adsorption solar cooling systems in China and concluded that because of high initial costs and high solar specific collector area the large scale residential application of solar cooling technologies was unfeasible, however that it was highly desirable to design solar powered integrated energy systems for public buildings. In public buildings there is enough roof areas to install solar collectors and as a result integrated energy systems are capable of supplying cooling and heating and can even enhance natural ventilation. It was further concluded that solar energy systems used only for cooling purposes are not economically viable because capital costs are very high.

Agyenim *et al.* (2010) developed a LiBr-water experimental solar cooling system and tested it during the summer months of 2007 at Cardiff University, UK. The system was made up of 12m^2 evacuated tube collectors, a 4.5KW LiBr-water absorption chiller, a 1000liters cold storage tank and a 6 KW fan coil. A system performance evaluation was carried out and it was found out that the average COP was 0.58. They proved the feasibility of the concept of cold storage and concluded that this type of arrangement had potential use for domestic cooling systems.

Calise *et al.* (2010) carried out transient analysis of heating and cooling systems in various configurations. They analyzed the case studies results on a monthly basis, concentrating on the energy and monetary flows of the standard and optimized configurations. The simulation results showed that the solar collector pump flow rates and storage tank volume have to be properly sized if satisfactory economic performance and energy efficiency is to be obtained. The results were found to encourage in gas for the potential of energy savings but appeared still far from economic profitability. The economical profitability of the systems could only be improved by significant public funding.

Marc *et al.* (2010) implemented a solar absorption system in Reunion without a back-up system to cool school class rooms. Experimental analysis showed that it is very difficult to design a solar energy absorption cooling system without back-up and to define the appropriate refrigeration capacity of the chiller.

Ming *et al.* (2010) at Carnegie Mellon University designed, installed and studied a 16 KW double effect LiBr-water chiller, with 52m^2 of linear parabolic trough collectors. This system also happens to be the smallest high temperature cooling system in the world. It was found that the system could potentially supply 39% of the cooling load of the building space in Pittsburgh, if it includes a properly sized storage tank

Moorthy (2010) carried out performance analysis on a system made up of 84 heat pipe evacuated tube collectors with a total absorber surface area of 235m^2, a 105 KW double effect LiBr-water chiller and two8m^3 hot and chilled water storage tanks. It was concluded that solar energy is able to produce sufficient energy to power the solar air conditioning system. The solar collector efficiency varied from 26 to 51%during day time and stored energy can be used for several hours during night time. The system COP varied from 0.7 to 1.2 throughout the system operation time and overall system efficiency varied from 27 - 48%.

Adhikari *et al.* (2012) investigated and evaluated the feasibility of an absorption refrigeration unit on solar power. The system designed here functions with the principle of absorption refrigeration cycle having water as a refrigerant and Lithium Bromide as an absorbent. The cooling load for the office building is 5 KW. The designed absorption system has 0.77 average coefficient of thermal performance (COP). They also analyzed the effect of COP in the variation of the refrigeration mass flow rate and generator temperature.

Balghouthi *et al.* (2012) carried out studies on a solar powered cooling system made up of parabolic trough solar collectors, a 16 KW LiBr-water double effect absorption chiller, cooling tower, back-up heater, two tanks for storage and a set of fan-coils in the building to be conditioned. In 2008 and 2009, water was used as the heat transfer fluid (HTF) and the system was run without thermal energy storage. The COP ranged from 0.8 -0.9 during system operation. Even though the nominal capacity of the chiller was 16 KW, the maximum output obtained was 12 KW. It was found that the back-up night storage improves the solar fraction of the system from 0.54 to 0.77. The avoided carbon dioxide emissions during a cooling season were approximately 3000 Kg corresponding to an equivalent energy saving of 1154 litres of gasoline.

Darkwa *et al.* (2012) carried out an evaluation of a typical integrated solar absorption system to determine its overall performance. Their study was intended to evaluate the performance of the system located in China, so as to find out the operational and technical barriers facing large scale systems. The results were analyzed and the solar collectors were found to have an operational efficiency of 61%. The chiller achieved a COP of 0.69. They concluded that the integrated solar powered absorption system has proven potential as a viable cooling technology for buildings. To maintain the hot water supply temperature for a system relying solely on solar energy, it is necessary to add secondary heat sources.

Samanta and Basu (2013) theoretically designed and analyzed the performance of a solar flat-plate collector-driven water-lithium bromide absorption refrigeration system for comfort air -conditioning of an office room during working hours of summer months. Absorption system exhibits an optimum generator temperature corresponding to maximum COP for any condenser-evaporator temperature combination. Integrated system COP has also been found to attain maxima for a particular solar insolation level. It is possible to maintain collector efficiency greater than 40% almost throughout the day. Supply tank water temperature has been found to be slightly higher at the end of the day compared to the early morning for summer months, whereas it remains slightly lower during post-monsoon months. Use of auxiliary heaters has been suggested rather than augmenting the number of collectors.

Robert *et al.* (2014) conducted an investigation to summarize the development status of air-cooled lithium bromide (LiBr)-water absorption chillers to guide future efforts to develop chillers for commercial buildings. The key technical barrier to air-cooled operation is the increased tendency for LiBr solutions to crystallize in the absorber when heat-rejection temperatures rise. Developers have used several approaches, including chemistry changes to inhibit crystallization, improving heat and mass transfer to lower overall temperature lift, modifying the thermodynamic cycle, combining absorption with vapour-compression to lower the temperature lift for each system, and advanced control systems to sense the onset of crystallization and take corrective action.

2.2.2.3 Three fluid absorption refrigeration system using solar energy

Narayankhedkar and Maiya (1985) investigated the working of the solar three fluid absorption refrigeration systems by varying the concentration of the strong solution and the charge pressure of inert gases with the generator's working temperature kept at above 150°C. The conclusion of their investigation was that increasing concentration of solution decreases the generator's working temperature and increases the evaporator's temperature. It was also established that an excessive inert gas charge pressure will reduce the refrigerating effect and as a result affect the overall system COP.

Pongsid and satha, (2002) used the same principles as those of Platen – Munters cycle with the exception that their system was of a different configuration with hydrogen/helium as one of the working fluids. The system has three working fluids with known concentrations i.e. ammonia 35 %, water 65 % concentration and initially hydrogen. To further investigate the characteristics of the system, variations of some operating parameters were considered. Among these parameters, helium was selected as an auxiliary gas for lowering the partial pressure of liquid ammonia instead of hydrogen mainly for safety reasons. The solution in the generator was heated to 180 °C and also they reported that the temperature below 90° C was not sufficient for the working of bubble pump.

Karthikeyan *et al.* (1995) studied the performance of absorption refrigeration system by adding a solution heat exchanger; the purpose is to increase the generator's heat input. The results showed that COP increases with increasing generator temperature.

Chen *et al.* (1996) considered a different approach by adding a solution heat exchanger and an auxiliary gas heat exchanger attached to the evaporator. Although a higher generator's working temperature of around 210 °C was used, Chen et al achieved a system COP of around 0.35, which was higher than that experienced in the original cycle.

Saravanan and Maiya (2003) also added to the original system a solution heat exchanger with the intention of increasing the system's efficiency by recovering heat from a solution; and therefore decreasing the amount of heat input to the generator. However in conjunction with another parameter considered and discussed above, a COP of 0.5 at a generator operating temperature of 85 °C was reported.

Srikhirin (2003) developed a mathematical model of a diffusion absorption refrigerator. The system was tested with heat input values between 1000 and 2500 W for a helium pressure of 6.1 bar. The system cooling capacities were found to be between 100 and 180 W with a COP between 0.09 and 0.15. A simple mathematical model was developed to analyze the experimental refrigerator.

The study carried out by Zohar *et al.* (2005) showed that helium as a working fluid was superior to hydrogen as the inert gas and therefore the coefficient of performance of a DAR unit working with helium was higher by up to 40 % than a cycle working with hydrogen. A gas heat exchanger and a solution heat exchanger attached to the evaporator in a shell-and tube configuration, and also they vary the generator's operating temperature in the range of 195 to 205 °C.

They reported the COP in the order of 0.09 to 0.15 corresponding to the generator temperatures.

Chaouachi and Gabsi (2007) designed a solar diffusion absorption refrigeration unit for domestic use. The obtained result showed that the developed mono pressure absorption cycle of ammonia was well suitable for producing refrigeration effect by using parabolic dish collector, which produced power in the order of 900 watts so it was very suitable for domestic use.

Sheridan (2009) reported that the generation of refrigerant ceases several hours before the sunset because of decrease in solar radiation intensity and also cooling rate continually decreases necessitating additional heating source when the solar radiation was not available

Velmurugan *et al.* (2011) investigated solar-based refrigeration system using three fluid ammonia-hydrogen/water (NH_3-H_2/H_2O) vapour absorption systems with solar point collector. The system was tested with helium pressure of 1.6 bar and heat input values between 1000 W and 2500 W. The system cooling capacities were found to be between 100 and 180 W with a COP between 0.09 and 0.15. A simple mathematical model was developed to analyze the experimental refrigerator. Comparisons between actual and calculated values show that the evaporator and absorber mass transfer have a strong effect on the system performance. The system produces maximum performance when all refrigerant is completely evaporated in the evaporator. This requires sufficient mass transfer surface in the absorber and evaporator.

Karthik (2014) designed and analyzed a solar vapour absorption refrigeration system. The setup was successfully made and the testing was done and vapour absorption system was successfully run using hot water as source of heat obtained from a solar collector of area $0.64m^2$. The mechanical modifications made to the mini fridge of 40 L capacity to accommodate the vapour absorption system and performance of the refrigerator was tested by using hot water from the solar collector, testing was done on a sunny day and the duration of the experiment was 9 hours. Lowest cabin temperature obtained was 8°C and COP obtained was 0.1675. The tonnage of the system was computed as 0.0164 TR. The mass flow rate calculated was 0.0306 Kg/min.

2.3 Heat generation by solar collectors

Soteris (2004) conducted a survey on the various types of solar thermal collectors and applications and also described about the various types of collectors including flat-plate, compound parabolic, evacuated tube, parabolic trough, Fresnel lens, parabolic dish and heliostat field collectors. Typical applications of the various types of collectors are presented in this paper According to him flat plate collector produced temperature in the range of 30 - 80° C, evacuated tube collector produced temperature in the range of 90 to 130° C and compound parabolic concentrator produced temperature in the range of 110 to 200°C. It is also reported that parabolic trough or parabolic dish type collectors were used for the applications like refrigeration, desalination, thermal power systems etc.

Muhammad *et al.* (2005) provided an up-to-date review of solar concentrators and their benefits to make solar technology affordable and also analyzed the some of the existing solar concentrators used in the solar technology for the past four decades. The design and performance of each concentrator was explained and compared. They concluded that solar concentrators could bring down the total cost of the solar cell, thus making the solar technology cheaper and affordable.

Shireesh *et al.* (2006) successfully installed and commissioned a large solar concentrator that can generate process heat at about 200°C, store it and supply it at desired process temperature any time of the day or night. A unit has been installed for pasteurization of about 30,000 L of milk daily by using solar energy at Mahanand Dairy, Latur, without firing the conventional Furnace Oil boiler. It has potential to save about 25 to 100% of process heat in industrial units. The technology when fully exploited in industrial process heat sector holds potential to save up to 6 to 10% of India's oil imports. It also opens up the doors for solar power generation by thermal route at less than half the cost of the Solar Photo Voltaic (SPV) systems available presently.

Christopher (2007) constructed a concentrated solar thermal energy system using a parabolic concentrator with the receiver placed along the line between the center of the concentrator and the sun. This allows for effective collecting and concentrating of the incoming solar radiation. The concentrator receives approximately 1.064 KW/m^2 of solar insolation (dependent upon time of year), which is concentrated and reflected to the receiver. By concentrating the incoming radiation, the operating temperature of the system is increased significantly, and subsequently increases the efficiency of the conversion from sun light to electricity. For the current system, with a concentration ratio was 96 and the concentrator is theoretically capable of producing temperatures above 712 degrees centigrade.

Zanis (2008) classified the solar collectors according to the temperature range. Flat-plate collectors are used typically for temperature requirements up to 75 °C although higher temperatures can be obtained from high efficiency collectors. According to him a concentrating collector utilizes a reflective parabolic-shaped surface to reflect and concentrate the sun's energy to a focal point or focal line where the absorber is located. To work effectively, the reflectors must track the sun. These collectors can achieve very high temperatures because the diffuse solar resource is concentrated in a small area. And also he mentioned that parabolic cylinder collectors were producing temperature in the range of 150 - 500°C but in the case of parabolic trough collectors producing temperature higher than 500° C.

Dascomb (2009) built an economic parabolic dish concentrating system for steam generation at the Sustainable Energy Science and Engineering Center (SESEC) at Florida. The goal of the project was to provide 6.67 kW of thermal energy. This is the amount of energy required to produce 1 kW of electricity with a conventional micro steam turbine. A 14 m^2fiberglass parabolic concentrator was made at SESEC to ensure simplicity of production and operation. The concentrator was coated with a highly reflective polymer film. The cavity type receiver was filled with sodium nitrate to act as a heat storage and transfer medium. The collection efficiency of the cavity was estimated at70%. The gross thermal conversion efficiency of the system was 39%, which represented a major improvement over the first concentrator assembled at SESEC. At peak insolation 5.46 KW of thermal energy was produced.

Joshua (2009) reported the design, construction and testing of a parabolic dish solar steam generator. Using concentrating collector, heat from the sun is concentrated on a black absorber located at the focus point of the reflector in which water is heated to a very high temperature to form steam. It also describes the sun tracking system unit by manual tilting of the lever at the base of the parabolic dish to capture solar energy. The whole arrangement is mounted on a hinged frame supported with a slotted lever for tilting the parabolic dish reflector to different angles so that the sun is always directed to the collector at different period of the day. He reported that on the average sunny and cloud free days, the test results gave high temperature above200°C.

Ouedemi *et al.* (2009) experimentally studied the parabolic solar concentrator. This experimental devise consists of a dish of 2.2 m opening diameter. Its interior surface is covered with a reflecting layer, and equipped with a disc receiver in its focal position. The orientation of the parabola is ensured by two semi-automatic jacks. Experimental measurements of solar flux and temperature distribution on the receiver have been carried out. The solar flux concentrated on receiver has been experimentally determined. The temperature in the center of the disc reached about 400 °C.

Romero *et al.* (2011) built and tested a solar parabolic concentrator prototype for the production of steam that can be coupled to a turbine to generate electricity or a steam engine in any particular industrial process. On a clear day temperatures above 200 °C and pressures up to 12 Kg/cm^2 was produced. They reported that the design could have achieved a higher temperature and pressure but this design is constrained by the errors associated with the shape of the parabolic surface of the parabolic solar concentrator prototype.

Robert (2012) investigated the three dimensional concentration concepts using two axis tracking solar collectors, since they offer an application temperature clearly beyond 500 °C. Three technical concepts for high temperature solar concentrators were also presented in this paper. For all three options the system design and a description of different components are presented together with an analysis of the system performance and loss mechanisms. The state of the art of existing facilities is demonstrated and further development directions are pointed out.

Saad *et al.* (2013) designed and constructed a parabolic trough collector for moderate heat load processes for instructional and demonstrative purposes. The solar tracking collector was designed to be a self-powered system so it can operate remotely and independently under moderate radiation levels. Once the system is oriented to a certain axis, N-S or E-W, it can operate continuously with minimal technical supervision. The manufacturing and operation of this device was simple. It used stainless steel as a reflecting surface and a closed loop control system for the solar tracking axis. The average efficiency of the collector when operated afternoon was found to be about 60% and produced process heat at moderate temperatures (up to 150°C), which is required by most of small factory processes, such as food canning, paper production, air-conditioning, refrigeration, sterilization, etc.

MATERIALS AND METHODS

CHAPTER 3

MATERIALS AND METHODS

This chapter deals with the details of the materials and the methodology followed during the course of the present research work.

3.1 Experimental outline

Design and development of solar refrigeration system was done in three stages. In the first stage the solar radiation intensity at Tavanur and the variation in temperature of water from the solar water heater were studied. In the second stage 40 L capacity absorption refrigeration system was designed and commercially available three fluid refrigerator using electricity was procured and it was modified for using heat energy. In the third stage experiments were conducted for assessing the performance of the developed Vapour absorption refrigerator using various heat energy sources. The Coefficient of Performance (COP) of the developed refrigerator was calculated.

3.2 Modification of the VARS

An electrically operated three fluid vapour absorption refrigerator (*CelfrostWellway (MB- 40)*) of 40 L capacity was modified for using heat energy for producing refrigeration effect.

Heat exchanger coil was constructed using copper tube of 6 mm outside diameter and 1 mm thickness. The copper tube was coiled and wound around the generator tube to transfer heat energy from heating medium. This heat exchanger coil had an inlet at the bottom for admitting hot water and an outlet at the top for the colder water. Inlet was connected to the heat resistant rubber tube of 6 mm diameter to carry heating medium from source. The outlet was also connected to a rubber tube for the discharge of medium after transfer of heat energy.

3.2.1 Components and working of the VARS

Absorption Refrigerator consists of generator, absorber, condenser and evaporator. In this system three fluids were used, ammonia, hydrogen and water.

Ammonia used as a refrigerant because it possesses most of the desirable properties. Water is used as the absorbent because it has the ability to absorb ammonia readily. In this system water and ammonia together form an aqua-ammonia solution. Hydrogen was used to increase the rate of evaporation of the liquid ammonia passing through the evaporator by lowering the partial pressure of ammonia in evaporator. Main components are;

a) **Copper heat exchanger coil**

Heating medium enters into the heat exchanger coil through inlet tube and it was circulated through the heat exchanger copper coil; convective heat transfer takes place from hot water to heat exchanger wall. Conductive heat transfer takes place between heat exchanger wall and generator wall, and then convective heat transfer takes place between generator wall and aqua ammonia solution. This heat was used to liberate ammonia gas from aqua- ammonia solution.

b) **Generator and bubble pump**

The strong ammonia solution from the absorber was heated in the generator by applying heat through heat exchanger coil. Heat input to the generator was used for two purposes: to vaporize and separate ammonia from the aqua- ammonia solution, and to stimulate the pumping effect through the pump-tube. The circulation of the weak solution between the generator and absorber is maintained by vapour bubbles which are generated at the lower part of the lifting pipes and forms at best a slug flow regime to push up a liquid column. This pushing action of bubbles causing lifting of weak solution through bubble tube. Once the weak solution and ammonia vapours reaches the top of bubble tube, ammonia vapours will enter into condenser but the weak solution again return back to absorber tubes to absorb ammonia vapours from the evaporator. This separation of ammonia vapours and weak solution were based on the density difference. The pressure compensation between high and low pressure level is achieved by the inert auxiliary gas hydrogen. The auxiliary gas circulates between the evaporator and absorber because of the temperature and density differences.

c) Condenser

Ammonia vapours are removed from the solution and passed to the condenser. Ammonia vapours in the condenser are condensed by using external cool air because condenser exposed to air. The liquid refrigerant leaving the condenser flows under gravity to the evaporator. There are no mechanical moving components inside the cooling unit and total pressure is constant at all points inside the cooling unit.

d) Evaporator

Liquid ammonia enters into the evaporator, where it meets the hydrogen gas. The hydrogen gas which is being fed to the evaporator permits the liquid ammonia to evaporate at a low pressure and temperature because liquid ammonia exposed to a hydrogen inert atmosphere in the evaporator, it will evaporate until the atmosphere is saturated with the vapour of the liquid. During the process of evaporation, the ammonia absorbs latent heat from the refrigerated space and thus produces cooling effect. The mixture of ammonia vapour and hydrogen is passed to the absorber tubes.

e) Absorber unit

Absorber unit consists of absorber tubes and absorber vessel. The mixture of ammonia vapour and hydrogen enters into the absorber tubes, where the weak solution from the generator meets with the ammonia vapour and hydrogen, ammonia is absorbed in the weak solution form strong aqua- ammonia solution then it flows into absorber vessel while the hydrogen rises to the top and flows back to the evaporator through narrow tube.

3.3 Design of the VARS

Vapour absorption refrigeration system mainly consists of generator tube, absorber unit, and evaporator and condenser unit. In this case 40 L capacity absorption refrigerator of each part was designed and dimensions were calculated. For design of the VARS following parameters are calculated;

a) Concentration of ammonia in each stage
b) Enthalpies of ammonia at saturated points

c) Enthalpies of strong and weak aqua- ammonia solutions

d) Mass flow rate of ammonia in the evaporator

e) Mass flow rates of strong and weak solution

f) Absorber heat rejection

g) Heat added in the generator (heat given to the refrigerant in the generator)

h) Heat rejection in condenser

i) Heat absorbed by the refrigerant in the evaporator

a) Concentration of ammonia in each stage

Concentration of ammonia per kilo gram of solution in each stage obtained from Enthalpy-Concentration Diagram for Ammonia/Water Solutions (2009 ASHRAE Handbook—Fundamentals (SI)).

x_1= Concentration of ammonia in vapour leaving generator

x_2= Concentration of ammonia in strong solution leaving absorber

x_3= Concentration of ammonia in weak solution leaving generator

b) Enthalpy of ammonia at saturated points

Enthalpy of ammonia at saturated points was determined using one software program called REFPROP. REFPROP is an acronym for REFerence fluid PROPerties. This program, developed by the National Institute of Standards and Technology (NIST), calculates the thermodynamic and transport properties of industrially important fluids and their mixtures. These properties can be displayed in tables and plots through the graphical user interface. Steps for calculating enthalpy of ammonia at saturated points using REFPROP software program were given below

a) Start the REFPROP program by double-clicking on its icon.

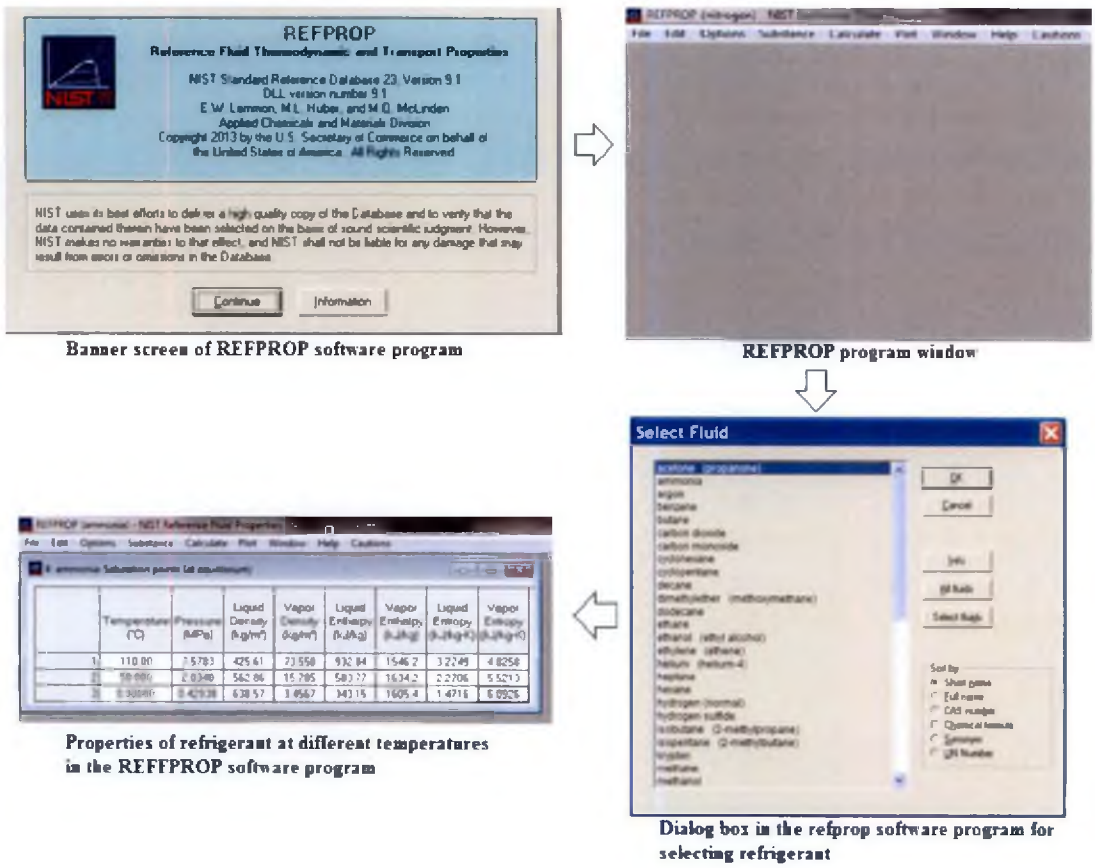
b) A banner screen displays the title, credits, and a legal disclaimer.

c) Clicking the "Continue" button starts the program.

d) REFPROP program window will open.

e) For selecting ammonia as pure fluid Selecting the Pure Fluid item in the Substance menu brings up the dialog box. From this dialog box select ammonia as working fluid, click OK button.

f) For calculating enthalpy, select Saturation points (at equilibrium) from Calculate menu brings up one table. Enter the temperature of generator in temperature column click enter button then enter condenser temperature in next row click enter then enter temperature of evaporator then we will get corresponding pressure, liquid density, vapour density, liquid enthalpy, vapour enthalpy, liquid entropy and vapour entropy of corresponding temperatures.



Banner screen of REFPROP software program

REFPROP program window

Dialog box in the refprop software program for selecting refrigerant

Properties of refrigerant at different temperatures in the REFFPROP software program

Plate 3.1 Flow chart for getting refrigerant properties in the REFPRP software program

Enthalpies obtained from REFPROP software program are;

Enthalpy of vapour ammonia leaving evaporator = (h_1)

Enthalpy of ammonia vapours leaving generator = (h_4)

Enthalpy of liquid ammonia leaving condenser = (h_5)

c) Enthalpy of aqua- ammonia solution

Enthalpy of aqua- ammonia solution determined using mathematical equations. The relation between saturation pressure and temperature of an ammonia-water mixture is given as (Bourseau and Bugarel, 1986):

$$\log P = A - \frac{B}{T} \qquad \text{.... 3.1}$$

Where; $A = 7.44 - 1.767\,X + 0.982X^2 + 0.362\,X^3$ 3.2

$B = 2013.8 - 2155.7\,X + 1540.9\,X^2 - 194.7\,X^3$ ... 3.3

X = concentration of ammonia

The relation among temperature, concentration and enthalpy is as follows (Patek and Klomfar, 1995);

$$h(T,\bar{X}) = 100 \sum_{i=1}^{16} a_i \left(\frac{T}{273.16} - 1\right)^{mi} \bar{X}^{ni} \qquad \text{... 3.4}$$

h = Enthalpy

$\bar{X}$ = ammonia mole fraction

$$\bar{X} = \frac{18.015\,X}{18.015X + 17.03(1-X)} \qquad \text{... 3.5}$$

Using these equations enthalpy of strong and weak solutions was calculated with the help of MS Excel.

d) Mass flow rate of solution in the evaporator (m_l)

Mass flow rate of liquid ammonia in the evaporator was calculated using formula given below.

$$m_l = 210 * Q_E/(h_l - h_5) \qquad \text{....3.6}$$

h_l = Enthalpy of vapour ammonia leaving evaporator (h_l)

h_5 = Enthalpy of liquid ammonia leaving condenser (h_5)

Q_E= Heat absorbed by the refrigerant in the evaporator

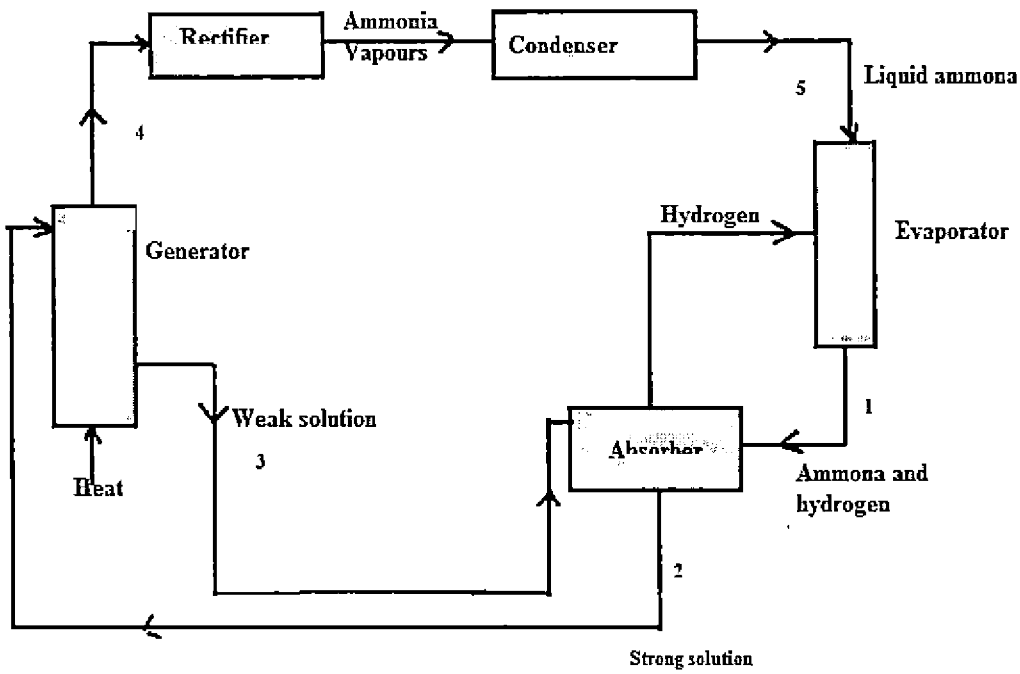


Figure 3.1 Vapour absorption refrigeration cycle

e) Mass flow rates of strong and weak solution

Mass flow rate of weak aqua- ammonia solution and mass flow rate of strong aqua- ammonia solution were calculated by considering the overall mass balance and material balance of NH_3in the absorber. Overall mass balance of NH_3 in the absorber was given by (Figure 3.1);

$$m_1 + m_3 = m_2 \qquad \ldots 3.7$$

m_1 =Mass flow rate of solution in the evaporator

m_2= Mass flow rate of strong aqua- ammonia solution leaving absorber

m_3= Mass flow rate of weak aqua- ammonia solution leaving generator

Material balance of NH_3in the absorber was given by (Figure 3.1);

$$m_1x_1 + m_3x_3 = m_2x_2 = (m_1 + m_3)\ x_2 \qquad \ldots 3.8$$

x_1= Concentration of ammonia in vapour leaving generator

x_2= Concentration of ammonia in strong solution leaving absorber

x_3= Concentration of ammonia in weak solution leaving generator

f) **Absorber heat rejection**

Strong aqua- ammonia solution in the absorber rejects heat to the atmosphere. Heat rejected to the atmosphere was calculated by considering the energy balance for the absorber. Heat rejected to the atmosphere was given by;

$$Q_A = m_1h_1 + m_3h_3 - m_2h_2 \qquad \ldots 3.9$$

Q_A= Absorber heat rejection to the atmosphere

m_1 =Mass flow rate of solution in the evaporator

m_2= Mass flow rate of strong aqua- ammonia solution leaving absorber

m_3= Mass flow rate of weak aqua- ammonia solution leaving generator

h_1= Enthalpy of vapour ammonia leaving evaporator

h_2= Enthalpy of strong solution leaving absorber

h_3= Enthalpy of weak solution leaving generator

g) **Heat added in the generator (heat given to the refrigerant in the generator)**

Heat given to the generator was to regenerate ammonia refrigerant from aqua – ammonia solution. Heat given to the refrigerant in the generator was calculated by considering the energy balance for generator; therefore heat added in the generator was given by;

$$Q_G = m_3h_3 + m_4h_4 - m_2h_2 \qquad (\text{ie } m_4 = m_1) \qquad \ldots 3.10$$

Q_G=Heat given to the generator

m_2= Mass flow rate of strong aqua- ammonia solution leaving absorber

m_3= Mass flow rate of weak aqua- ammonia solution leaving generator

m_4= Mass flow rate of ammonia vapours leaving generator (i.e. m_4= m_1)

h_2= Enthalpy of strong solution leaving absorber

h_3= Enthalpy of weak solution leaving generator

h_4= Enthalpy of ammonia vapours leaving generator

h) Heat rejected from the condenser

Air – cooled condenser was used in this system, therefore heat rejected from the condenser to air was calculated by considering energy balance for condenser, it was given by;

$$Q_c = m_4 (h_4 - h_5) \quad \ldots 3.11$$

Q_c =Heat rejected from the condenser

m_4 = Mass flow rate of ammonia vapours leaving generator

h_4= Enthalpy of ammonia vapours leaving generator

i) Heat absorbed by the refrigerant in the evaporator

Heat absorbed by the ammonia refrigerant in the generator was calculated by considering the energy balance for evaporator, it was given by,

$$Q_E = m_5(h_5 - h_1) \quad \ldots 3.12$$

Q_E=Heat absorbed by the refrigerant in the evaporator

m_5 = Mass flow rate of liquid ammonia leaving condenser ($m_4 = m_1 = m_5$)

h_1= Enthalpy of vapour ammonia leaving evaporator

h_5= Enthalpy of liquid ammonia leaving condenser

3.3.1 Generator design

Generator was designed based on the mass flow rate of strong and weak solution in the generator and amount ammonia vapours produced in the generator. While designing the generator original mass of the solutions is taken 15% more, total volume of generator was given by;

Total volume of generator

= [volume of water vapour] + [volume of strong Solution] …. 3.13

Volume of water vapour generated was calculated by;

Total volume of vapour = m× Specific volume … 3.14

Specific volume of water vapour calculated based on the generator temperature and pressure. Volume of strong solution was also calculated. Therefore dimensions of the generator tube were calculated by using equation;

$$V_G = \Pi/4 D_G^2 L_G \qquad \ldots 3.15$$

V_G = Volume of the generator tube

D_G = Diameter of the generator tube

L_G = Length of the generator tube

3.3.2 Design of Absorber unit

Absorber unit consists of absorber tube and absorber vessel. Absorber tube is an integral part of the vessel. Absorber vessel was designed based on the volume of NH_3vapour and volume of weak solution of NH_3 in the absorber vessel. Volume of absorber vessel was calculated by;

Volume of the absorber vessel

= [Volume of NH_3vapour] + [volume of weak NH_3 solution] ... 3.16

Volume of the ammonia vapour was calculated by;

Volume of NH_3vapour = $[(1/S_v) \times$ mass flow rate] ... 3.17

Where, S_v= Density of NH_3vapour

Volume of liquid ammonia was calculated by;

Volume of liquid ammonia = $[(1/S_L) \times$ mass flow rate] ... 3.18

Where, S_L =Density of liquid NH_3

Therefore dimensions of the absorber vessel were calculated by using equation;

$$V_{AV} = \Pi/4 \times D_{AV}^2 \times L_{AV} \qquad \ldots 3.19$$

V_{AV} = Volume of the absorber vessel

D_{AV}= Diameter of the absorber vessell

L_{AV}= Length of Absorber vessell

Design of absorber tube was based on the heat removed from the absorber unit and heat transfer between the solution and the air. Heat removed from the absorber unit was given by;

$$Q_A = [U \times A_{AT} \times LMTD] \quad \ldots 3.20$$

Q_A= Heat removed from the absorber

A_{AT} = Area of the Absorber tube

LMTD = Logarithmic Mean Temperature Difference

Logarithmic Mean Temperature Difference was calculated by using the equation

$$LMTD = [((\Delta T1 - \Delta T2))/\ln[\![(\Delta T1)/(\Delta T2)]\!]] \quad \ldots. 3.21$$

$$\Delta T_1 = T_{hi} - T_{co} \quad \ldots. 3.22$$

$$\Delta T2 = T_{hi} - T_{ci} \quad \ldots 3.23$$

T_{hi}= Temperature of NH_3vapour at inlet

T_{co} = Outlet temperature of air

T_{ho}= Temperature of NH_3 liquid at outlet

T_{co} =Outlet temperature of air

U = Overall heat transfer coefficient

$$1/U=\{1/h_o+(D_{AO}/k)\times[(D_{AO}D_{Ai})/(D_{AO}+D_{Ai})]+(1/h_i)\times D_{AO}D_{Ai}\} \quad \ldots 3.24$$

h_i= Convective heat transfer at the inlet (W/m^2K)

h_o= Convective heat transfer at the outlet (W/m^2K)

k = Thermal conductivity (W/m K)

D_{AO} = Outer diameter of absorber tube (m) (assumed value)

D_{Ai}= Inner diameter of absorber tube (m)(assumed value)

Therefore dimensions of the absorber tube were calculated by using equation

$$V_{AT} = \Pi/4 \times D_{AT}^2 \times L_{AT} \quad \ldots 3.25$$

D_{AT}= Diameter of the absorber tube

L_{AT}= Length of absorber tube

3.3.3 Design of evaporator

An evaporator should transfer enough heat from as small size as possible. It should be light, compact, safe and durable. The pressure loss should be as low as possible. Evaporator was designed based on evaporator temperature, refrigerant properties and refrigeration effect. Heat absorbed by the refrigerant in the evaporator was given by;

$$Q_E = [U \times A_E \times LMTD] \qquad \ldots 3.30$$

Q_E =Heat absorbed by the refrigerant in the evaporator (KJ/min)

A_E= Area of the evaporator coil

Q_A= Heat removed from the absorber

A_{AT} = Area of the Absorber tube

LMTD = Logarithmic Mean Temperature Difference

Logarithmic Mean Temperature Difference was calculated by using the equation

$$LMTD = [((\Delta T1 - \Delta T2))/\ln [(\Delta T1)/(\Delta T2)]] \qquad \ldots. 3.31$$

$$\Delta T_1 = T_{hi} - T_{co} \qquad \ldots. 3.32$$

$$\Delta T2 = T_{hi} - T_{ci} \qquad \ldots 3.33$$

T_{hi}= Temperature of air surrounding evaporator

T_{ho} = Temperature of air after 1 min

T_{ci}= Temperature of NH_3 liquid entering evaporator

T_{co}= Temperature of NH_3 liquid leaving evaporator

U = Overall heat transfer coefficient

$$1/U=\{1/h_o+(D_{EO}/k)\times[(D_{EO}D_{Ei})/(D_{EO}+D_{Ei})]+(1/h_i)\times D_{EO}D_{Ei}\} \qquad \ldots 3.34$$

h_i= Convective heat transfer at the inlet (W/m^2K)

h_o= Convective heat transfer at the outlet (W/m^2K)

k = Thermal conductivity (W/m K)

D_{EO} = Outer diameter of evaporator coil (m) (assumed value)

D_{Ei} = Inner diameter of Evaporator coil (m) (assumed value)

Therefore dimensions of the evaporator coil were calculated by using equation

$$V_E = \Pi/4 \times D_E^2 \times L_E \quad \ldots 3.35$$

V_E = volume of evaporator coil

D_E = Diameter of the evaporator coil

L_E = Length of the evaporator coil

3.3.4 Design of condenser

Condenser was designed based on the heat transfer between Ammonia vapours and air. An air cooled fin type condenser is used in this refrigerator. Heat rejected from the condenser was given by;

$$Q_C = [U \times A_C \times LMTD] \quad \ldots 3.36$$

Q_C = Heat rejected from the condenser

A_C = Area of the condenser

LMTD = Logarithmic Mean Temperature Difference

$$LMTD = [((\Delta T1 - \Delta T2))/\ln [(\Delta T1)/(\Delta T2)]] \quad \ldots 3.37$$

$$\Delta T_1 = T_{hi} - T_{co} \quad \ldots 3.38$$

$$\Delta T2 = T_{hi} - T_{ci} \quad \ldots 3.39$$

T_{hi} = Temperature of NH_3 vapor at inlet

T_{ho} = Temperature of NH_3 liquid at outlet

T_{ci} = Inlet temperature of air

U = Overall heat transfer coefficient

$$1/U = \{1/h_o + (D_{CO}/k) \times [(D_{CO} D_{Ci})/(D_{CO} + D_{Ci})] + (1/h_i) \times D_{CO} D_{Ci}\} \quad \ldots 3.40$$

h_i = Convective heat transfer at the inlet (W/m^{2}K)

h_o= Convective heat transfer at the outlet (W/m^{2}K)

k = Thermal conductivity (W/m K)

D_{CO} = Outer diameter of condenser tube (m) (assumed value)

D_{Ci}= Inner diameter of condenser tube (m) (assumed value)

Therefore dimensions of the condenser tube were calculated by using equation

$$V_C = \Pi/4 \times D_C^2 \times L_C \quad \dots 3.41$$

V_C = volume of evaporator coil

D_C= Diameter of the condenser tube

L_C= Length of the condenser tube

3.4 Methodology

3.4.1 Measurement of solar radiation

Solar radiation intensity was measured using digital lux meter in the month of March and every day readings were taken from morning 10' O clock to evening 4' O clock at an interval of 1 h. By using lux meter solar radiation intensity was measured directly in lux. Using equation 3.42 solar radiation intensity in lux is converted into watt.

$$P_{(W)} = \Phi_{V(lm)} / \eta_{(lm/W} \quad \dots 3.42$$

Where, $P_{(W)}$= solar radiation intensity in W

$\Phi_{V(lm)}$= Luminous flux in lumen

$\eta_{(lm/W)}$= Luminous efficacy in lumen per watt

Solar efficacy was calculated by Paul, 1985 as;

Solar efficacy ($\eta_{(lm/W)}$) = 105 lm/W

3.4.2 Temperature measurement of solar water heater

Study was conducted at KCAET, Tavanur in Malappuram district. This place is situated at 10° 52' 30" north latitude and 76° east longitude. Water heater was installed at the roof top of the college guest house, inside the campus. Inlet and outlet water temperatures were measured using digital thermometer .Measurements were taken every day continuously for a month , from morning 10' O clock to evening 4 ' O clock in an interval of 1 h.

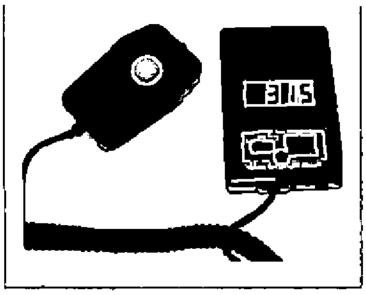


Plate 3.2 Digital Luxmeter

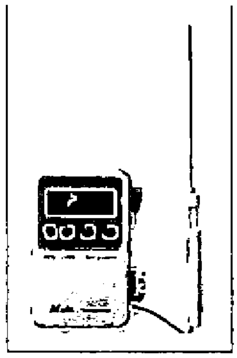


Plate 3.3 Digital Thermometer

3.4.3 Performance evaluation of VARS using hot water from solar water heater

Solar water heater outlet was connected to the bottom side of heat exchanger tubes and hot water continuously circulates through copper coil and exit water was collected at the top side of heat exchanger. Convective heat transfer takes place from hot water to heat exchanger walls and conductive heat transfer takes place from heat exchanger walls to generator wall. Heat resistant rubber tubes are used to connect water heater outlet to heat exchanger inlet. Temperature of water at heat exchanger inlet, temperature of water at heat exchanger outlet, heat exchanger wall temperature, generator temperature and refrigerator cabin temperature were measured using digital thermometer.

3.4.4 Performance evaluation of VARS using steam (hot water supplied by the solar flat plate collector)

Solar water heater outlet was connected to the steam generator inlet through flow control valve. Hot water to the steam generator was supplied by the solar flat plate collector of 200 L capacity. Steam outlet through pressure control valve was directly connected to the heat exchanger inlet at the bottom side of the generator tube with the help of heat resistant rubber tube. Steam at 115°C continuously circulated through heat exchanger coil wound around the generator tube. Outlet steam was collected at the top side of the heat exchanger tube. Inlet steam temperature, outlet steam temperature, heat exchanger wall temperature, generator temperature, condenser temperature, absorber temperature and evaporator temperature were measured using digital thermo meter. Performance of the refrigerator tested by measuring the temperature of water kept inside the ice tray.

3.4.5 Simulated studies on VARS using hot water

Performance of the refrigerator was tested using hot water in three stages. In the first stage performance was tested using hot water from blancher, which simulated the conditions of hot water from solar water heater. In the second stage water at 100° C generated in a steel vessel was used for the performance evaluation of the refrigerator. In the third stage performance was tested using hot water at 100° C generated in a hot water blancher.

3.4.5.1 Performance evaluation of VARS using hot water at 100°C

a) Hot water generated in a steel vessel

Water was heated up to 100°C in a steel vessel using a LPG gas burner. Hot water inlet was connected to top side of the heat exchanger. Hot water circulates through copper coil turns of the heat exchanger and exit water was collected at the bottom side of heat exchanger. The inlet and outlet temperature of hot water was measured using digital thermometer. Generator and refrigerator cabin temperatures were also noted.

b) Hot water generated in a hot water blancher

Hot water blancher temperature set as 100°C and water was heated up to 100°C in the hot water blancher. The flow rate of water to the inlet of blancher and outlet hot water from the blancher was set as same for maintaining uniform flow rate .Blancher outlet was connected to the inlet of heat exchanger tube by using heat resistant rubber tube. Hot water at 100°C was circulated through heat exchanger tubes. Convective heat transfer takes place between hot water and heat exchanger wall and conductive heat transfer takes place between heat exchanger wall and generator wall. Temperature of water at heat exchanger inlet, heat exchanger wall temperature, generator temperature, condenser temperature, absorber temperature and evaporator temperature were measured using digital thermo meter. Performance of the refrigerator was tested by measuring the temperature of water kept inside the ice tray.

3.4.6 Simulated studies on VARS using steam

Since solar water heater could not produce steam, it was artificially produced in 10L capacity pressure cooker at four different pressures and four different temperatures. Steam outlet from pressure cooker was connected to the heat exchanger inlet at the bottom side of the generator tube with the help of heat resistant rubber tube. Steam continuously circulated through heat exchanger coil wound around the generator tube. Outlet steam was collected at the top side of the heat exchanger tube. Inlet steam temperature, outlet steam temperature, heat exchanger wall temperature, generator temperature, condenser temperature, absorber temperature and evaporator temperature were measured using digital thermo meter. Performance of the refrigerator tested by measuring the temperature of water kept inside the ice tray.

Experiments were repeated using steam for seven trials at four different temperatures and four different pressures.

Table 3.2 Performance evaluation of VARS using steam in different trials

No. Trail	Temperature of steam, °C (7 h circulation)
1	103
2	106
3	116
4	121

3.4.7 Performance evaluation of VARS using electric heater

Performance of the refrigerator was tested by using electrical heater fitted with the generator tube. Generator, condenser, absorber, evaporator temperatures and corresponding refrigerator cabin temperatures were measured using multi thermometer. Refrigerator cabin temperature measured at regular intervals, time-cabin temperature graph were plotted.

a) Measurement of power consumption of electric heater in the VARS

Power consumption of electric heater in a refrigerator was measured using watt meter. Current coil in a watt meter was connected to one end of the load and potential coil was connected to other end of the load. Then two wires connected from each coil were connected to the electrical input and wattmeter readings were measured in one hour.

b)Measurement of energy consumption of electric heater in the VARS

Connections were made same as that of watt meter. Energy consumption was measured by counting the rotation of disc. Number of rotation of disc is measured at specified interval of time 10min, 20 min, 30 min, 40 min, 50 min and 1 hour. Number of rotation of disc in 1 hour noted from that energy consumption was calculated. In this 1 kilowatt hour equals 1200 no. of rotation of disc. The results were noted in an interval of 1 h.

3.5 Statistical Analysis

The effect of temperature of different heating mediums on generator temperature, Refrigerator cabin temperature and ice tray temperature in each hour was statistically analyzed by ANOVA. The statistical analysis of data was carried out using SPSS software (Version 16.0; SPSS Inc. Chicago).

3.6 Calculation of coefficient of performance (COP) of the refrigerator

COP of the refrigeration system indicates the performance of the given refrigeration system. Coefficient of performance of the VARS was calculated for each generator temperature by using each heating medium. COP was calculated as;

$$COP = \frac{\text{Heat absorbed by the refrigerant in the evaporator (QE)}}{\text{Heat given to the generator (QG)}}$$

Figure 3.4 Setup for the performance of VARS using hot water from solar flat plate collector

Figure 3.5 Set for the performance of VARS using steam (Hot water supplied by solar flat plate collector)

Plate 3.6 Setup for the performance of VARS using hot water from blancher

Plate 3.7 Setup for the performance of VARS using hot water generated in a steel vessel

Plate 3.8 Setup for performance evaluation of VARS using steam

Plate 3.9 VARS with Electric heater

Plate 3.10 VARS with energy meter

3.7 Economic analyses of the developed refrigeration system

Economic analysis was done by comparing the modified 40 L capacity solar absorption refrigerator with 40 L capacity absorption refrigerator using electricity. Fixed cost for each refrigeration system was calculated and operational cost of absorption refrigerator using electricity was calculated with 10 percent increase in electric cost in every year. Payback period was also calculated. The calculations are given in Appendix IV.

RESULTS AND DISCUSSION

CHAPTER 4

RESULTS AND DISCUSSION

The results of the work, ***design and development of solar refrigeration system*** are detailed in this chapter. The results of solar radiation measurements at KCAET, Tavanur, and performance evaluation of the developed solar refrigeration system are also discussed in detail this chapter.

4.1 Monthly average solar radiation intensity

Measurements of solar radiation intensity at Tavanur were taken during the month of March, from morning 10 AM to evening 4 PM. Monthly measurements of solar radiation intensity were given in Appendix I. Table 4.1 shows the monthly average solar radiation intensity with time at KCAET Tavanur.

Table 4.1 Variation of solar radiation intensity with time

Time, h	Monthly average solar radiation intensity, lux	Monthly average solar radiation intensity, W/m^2
10	54300	517.14
11	64900	618.11
12	77600	739.04
1	82300	783.81
2	76900	732.38
3	64700	616.19
4	49200	468.57

Figure 4.1 shows the variation of solar radiation intensity with time. It shows that solar radiation intensity increases with time up to a certain period then it starts decreasing. It was observed that maximum solar radiation intensity obtained at 1.00 PM and it was 783.81 W/m^2. Variation of solar radiation intensity at Tavanur was studied in previous years also as reported by Biju in 2000 and Jijimon in 1993 and the above values are almost identical.

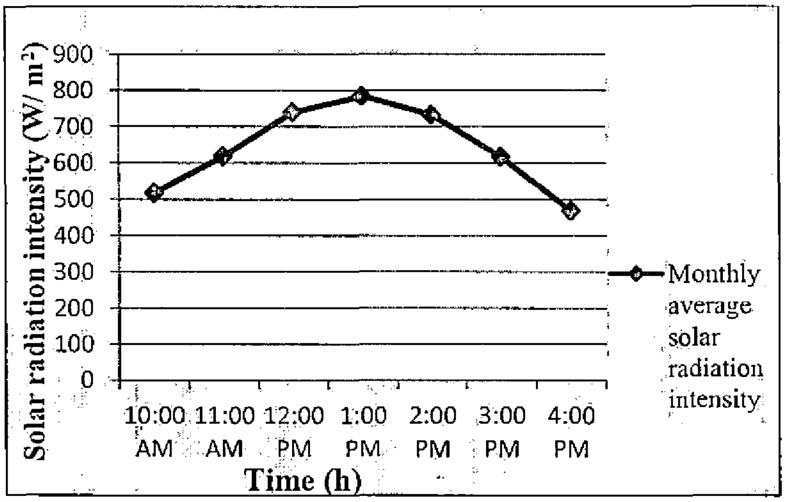


Figure 4.1 Variation of solar radiation intensity with time

Positions of the Sun have a direct effect on the intensity of solar radiation. The intensity of solar radiation is largely a function of the angle at which the Sun's rays strike the Earth's surface. If the Sun is positioned directly overhead or 90° from the horizon, the incoming insolation strikes the surface of the Earth at right angles and is most intense. If the Sun is 45° above the horizon, the incoming insolation strikes the Earth's surface at an angle. This causes the rays to be spread out over a larger surface area reducing the intensity of the radiation (Carg 1982). This is the why more solar radiation is present during midday is more than during either the early morning or late afternoon. At midday, the sun is positioned high in the sky and the path of the sun's rays through the earth's atmosphere is shortened. Consequently, less solar radiation is scattered or absorbed, and more solar radiation reaches the earth's surface (Bhattacharya, 1996)

4.2 Monthly average outlet temperature of solar water heater

Experiments were conducted with the solar water heater installed at KCAET, Tavanur. Measurements were taken during the month of March. Monthly average water heater outlet temperatures with time were given in Table 4.2. Monthly readings of ambient temperature and solar water heater outlet temperature were given in Appendix II and Appendix III.

Table 4.2 Variation of outlet water temperature from solar water heater and ambient temperature with time

Time, h	Monthly average ambient temperature, °C	Monthly average water heater outlet temperature, °C
10:0	33.1	41.3
11:0	33.3	47.9
12:0	33.8	57.3
1:00	34.3	66.8
2:00	34.5	74.6
3:00	34.5	78.2
4:00	34.3	77.84

Table 4.2 shows the variation of water heater outlet temperature with time. The outlet temperature of water heater is mainly depends on solar radiation intensity. If the solar radiation intensity increases, temperature of outlet water also increases.

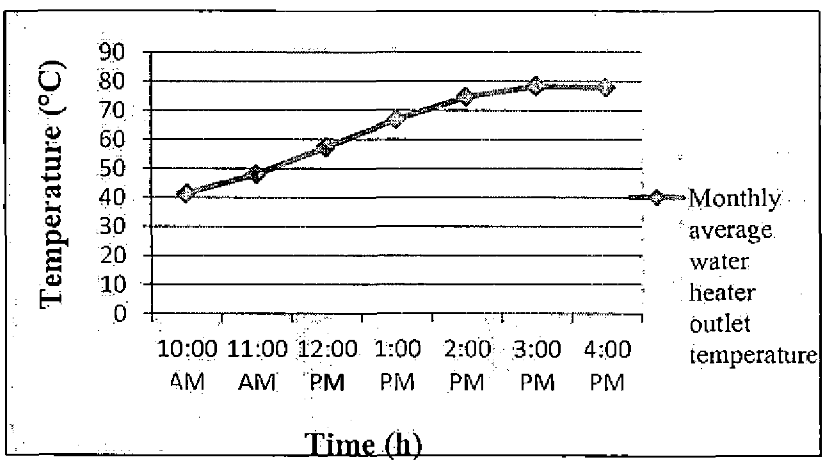


Figure 4.2 Variation in temperature of outlet water from solar water heater with time

Comparing Figure 4.1 and Figure 4.2 solar radiation intensity increased up to certain period then it started decreasing but the temperature of outlet water increased with time, even if the solar radiation decreased. This temperature rise of water is due to the storage of water inside the insulated storage tank of the solar water heater. Storage tank in the solar water heater is highly insulated to reduce the heat loss and therefore hot water inside the storage tank maintained at high temperature even though the solar radiation intensity decreased.

4.3 Modified vapor absorption refrigeration system using heat energy

Vapour absorption refrigeration system mainly consists of generator tube, absorber unit, and evaporator and condenser unit. In this case 40 L capacity absorption refrigerator (*Celfrost-Wellway (MB- 40)*) of 40 L capacity was modified for using heat energy. Heat energy was provided by the hot water or steam. Absorption refrigerator of each part was designed and dimensions were calculated.

Vapour absorption refrigeration system mainly consists of generator, absorber, condenser and an evaporator. Additional one part used in this system was copper heat exchanger coil. Dimensions of each component was calculated and summarized in Table 4.3.

Table 4.3 Designed dimensions of the components of the VARS

<table>
<tr><th colspan="2" rowspan="2">Components</th><th colspan="5">Dimensions</th></tr>
<tr><th>Volume, m^3</th><th colspan="2">Diameter, m</th><th colspan="2">Length, m</th></tr>
<tr><td colspan="2" rowspan="2">Heat exchanger copper coil</td><td>4.24×10^{-5}</td><td colspan="2">0.006</td><td colspan="2">1.5</td></tr>
<tr><td colspan="5">No. of turns = 28</td></tr>
<tr><td colspan="2">Generator tube</td><td>7.85×10^{-5}</td><td colspan="2">0.02</td><td colspan="2">0.25</td></tr>
<tr><td rowspan="2">Absorber unit</td><td>Absorber vessel</td><td>3.30×10^{-4}</td><td colspan="2">0.06</td><td colspan="2">0.125</td></tr>
<tr><td>Absorber tube</td><td>2.5×10^{-3}</td><td colspan="2">0.03</td><td colspan="2">3.6</td></tr>
<tr><td colspan="2">Evaporator</td><td>4.6×10^{-5}</td><td colspan="2">0.014</td><td colspan="2">0.30</td></tr>
<tr><td rowspan="3">Condenser unit</td><td>Condenser tube</td><td>4.9×10^{-5}</td><td colspan="2">0.014</td><td colspan="2">0.32</td></tr>
<tr><td rowspan="2">Condenser fins</td><td rowspan="2">3.5×10^{-6}</td><td>Length, m</td><td>Width, m</td><td>Thickness, m</td><td>No. of fins</td></tr>
<tr><td>0.07</td><td>0.05</td><td>0.001</td><td>35</td></tr>
</table>

A designed dimension of the 40 L capacity vapour absorption refrigerator was identical to the commercial three fluid vapour absorption refrigeration system. Hence it was procured for modification and further evaluation.

Table 4.4 Technical specifications of electrically operated VARS

Technical Specifications	
Dimension	400 × 440 × 575 mm
Net weight	18 Kg
Available Capacity	40 L
Rated Voltage	220 V AC
Rated electric Current	0.32 A
Rated frequency	50 Hz
Total input power	70 W
Power Consumption	0.085 KWh/24 h
Cooling Medium	Ammonia, Water, Hydrogen
Quantity ammonia filled	280 g

4.3.1 Properties of aqua – ammonia solution

In this system ammonia was used as the refrigerant and water was used as the absorbent therefore it form aqua – ammonia solution and hence concentration, enthalpy and mass flow rate of aqua – ammonia solution in each stage was changing because of heat absorption and rejection of this fluid. Properties of aqua- ammonia solution in each stage was calculated and given in Table 4.5.

Table 4.5 Properties of aqua – ammonia solution in each stage

Particulars	Concentration, Kg of NH_3/Kg of solution	Enthalpy, KJ/Kg	Mass flow rate, Kg /min
Strong aqua- ammonia solution leaving absorber	0.538	93.695	0.02241
Weak aqua- ammonia solution leaving generator	0.434	261	0.0183
Ammonia vapor leaving generator	1.00	1546.2	0.00411
Liquid ammonia leaving condenser	1.00	583.77	0.00411
Ammonia vapor leaving evaporator	1.00	1605	0.00411

4.3.2 Heat rejection/absorption of aqua – ammonia solution

In the generator, strong aqua- ammonia solution absorbs heat from the heating medium for separating ammonia from aqua – ammonia solution. In condenser vapour ammonia reject heat to surrounding for changing the phase to liquid state; in the evaporator liquid ammonia absorb heat from the product or space for producing refrigeration effect. In the absorber, heat was rejected to atmosphere for increasing the absorption of vapour refrigerant. Temperature of each component of the VARS was measured by considering the evaporator temperature of 0°C. Heat absorbed/ rejected and temperature of the each components of VARS were obtained are given below;

Table 4.6 Heat absorption / rejection and temperature of components of VARS

Components	Heat absorption/ rejection, KJ/min	Temperature, °C
Generator	Heat absorption = 9.03	110
Absorber	Heat rejection = 9.274	35
Evaporator	Heat absorption = 4.2	0
Condenser	Heat rejection = 3.956	40

4.4 Performance of VARS using hot water from solar water heater

Hot water from the solar water heater was passed through heat exchanger coil, variation of temperatures of heat exchanger tube; generator and ice tray with time were given Table 4.7

Table 4.7 Performance of VARS using hot water from solar water heater

Time, h	Water heater outlet temperature, °C	Heat exchanger surface temperature, °C	Generator temperature, °C	Ice tray temperature, °C
0	-	-	-	-
1	41.3	38	35	29.5
2	47.9	43	39	29.5
3	57.3	53	48	29.5
4	66.8	64	59	29.5
5	74.6	69	63	29.5
6	78.2	74	68	29.5
7	77.8	71	65	29.5

In this case maximum generator temperature obtained as 68°C at the hot water temperature of 78.2°C, but this temperature has no effect on ice tray temperature, it maintained 29.5°C even after 7 hours, and this indicates that no refrigeration effect was produced.

Figure 4.3 shows the performance of VARS using hot water from the solar water heater. It reveals that heat exchanger surface temperature and generator temperature increases with increase in hot water temperature but no cooling effect on ice tray temperature.

Cooling effect mainly depends on generator temperature and effective working of bubble pump. Cooling effect increases with increase in generator temperature because heat energy given to the generator is used for the regeneration of ammonia from the aqua ammonia solution and pumping effect.

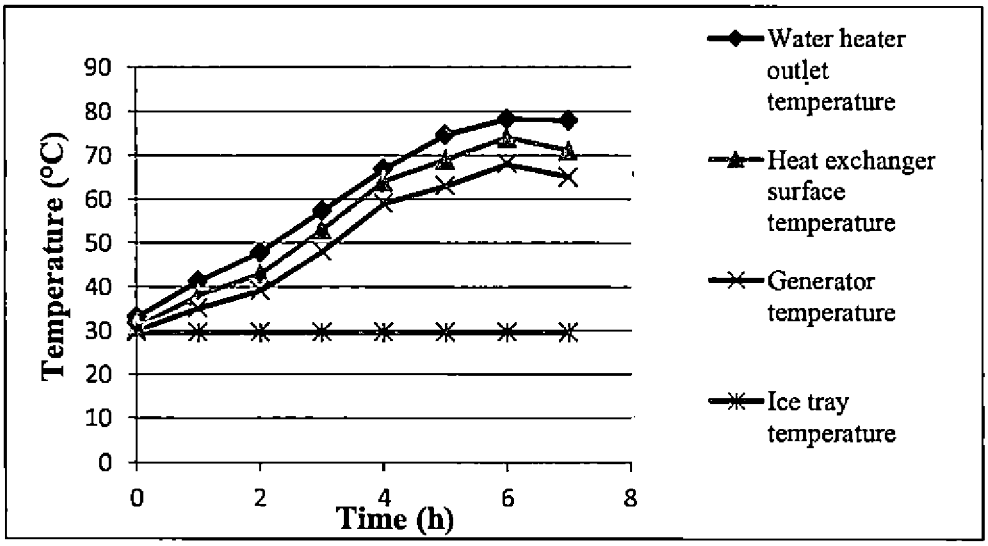


Figure 4.3.Performance of VARS using hot water from the solar water heater

To operate the refrigerator there was a minimum generator input temperature. When the heat input was lower than the minimum value, the system was not able to produce any cooling effect. This is due to the bubble pump characteristics. To obtain a pumping effect, a minimum vapour generation is required. When the heat input is too low, there is not enough quantity of vapours n generated to drive the pump. When the heat input increases, the bubble-pump begins to operate. Further increase in heat input produces a higher cooling effect as more refrigerant vaporizes and more liquid is pumped to the absorber as reported by Velmurugan *et al.*, 2011.

From this result it was found that 68°C generator temperature was not sufficient for vapour generation for effective working of bubble pump and also concluded that the available flat plate collector based water heater was not sufficient for producing refrigeration effect because it can produce hot water temperature maximum of about 80°C and this temperature was not sufficient for producing cooling effect.

4.5 Performance of VARS using steam (Hot water supplied by the solar flat plate collector)

Steam outlet from the steam generator was directly connected to the inlet of the heat exchanger tube. Ice tray temperature and refrigerator cabin temperature were measured at the steam temperature of 115°C. Table 4.8 shows the variation of ice tray temperature and refrigerator cabin temperature with temperature of steam at 115°C.

Table 4.8 Performance of VARS using steam (Hot water supplied by the solar flat plate collector)

Time (h)	Solar water heater outlet temperature (°C)	Steam temperature (°C)	Generator temperature (°C)	Ice tray temperature (°C)	Refrigerator cabin temperature (°C)
1	42	115	96	29	29
2	46	115	98	25	27
3	58	115	101	19	24
4	67	115	103	15	20
5	75	115	105	11	18
6	78	115	105	9	14
7	77	115	105	7	12

Table 4.8 reveals that generator temperature increases up to 105°C with the steam temperature of 115°C and refrigerator cabin temperature and ice tray temperature were decreased in these generator temperatures. In this case ice tray temperature attained was 7°C after 7 h circulation steam at 115°C. Because of this increase in generator temperature more cooling effect was produced. It was concluded that at higher generator temperatures, higher will be the cooling as reported by Karthik, 2014.

4.6 Simulated studies on VARS using hot water

Performance of the refrigerator was tested using hot water from the solar water heater and also using hot water at 100°C heated in a steel vessel and hot water blancher

4.6.1 Performance of VARS using hot water at 100°C

In the first stage, water heated in a steel vessel using LPG gas burner was circulated through heat exchanger tube. In the second stage hot water generated continuously in a hot water blancher was circulated through heat exchanger tubes. Variation of heat exchanger surface, generator, ice tray and refrigerator cabin temperatures with time were given in Table 4.9.

Table 4.9 Performance of VARS using hot water at 100° C

Time (h)	Heat exchanger surface temperature (°C)		Generator temperature (°C)		Ice tray temperature (°C)		Refrigerator cabin temperature (°C)	
	Steel vessel	Hot water blancher	Steel vessel	Hot water blancher	Steel vessel	Hot water blancher	Steel vessel	Hot water blancher
0	-	-	-	-	-	-	-	-
1	79	78	74	71	29.8	29.8	30	30
2	80	80	75	72	29.8	29.8	30	30
3	80	84	75	75	29.8	29.8	30	30
4	74	85	70	75	29.8	29.8	30	30
5	72	85	69	75	29.8	29.8	30	30
6	76	85	70	75	29.8	29.8	30	30
7	79	85	74	75	29.8	29.8	30	30

In this case generator temperature was increases up to 75°C with increase in hot water temperature but no effect on ice tray temperature. Generator attained very low temperature even though the hot water temperature was maintained at 100 °C. It may be due to heat loss through inlet tube and heat exchanger wall.

In this case also no refrigeration effect was produced, because of low generator temperature. Minimum generator temperature was required to produce ammonia vapours for effective working of bubble pump. Above 90° C generator temperature was required for the effective operation of bubble pump as well as separation of ammonia from aqua- ammonia solution as reported by Pongsid and satha, 2002.

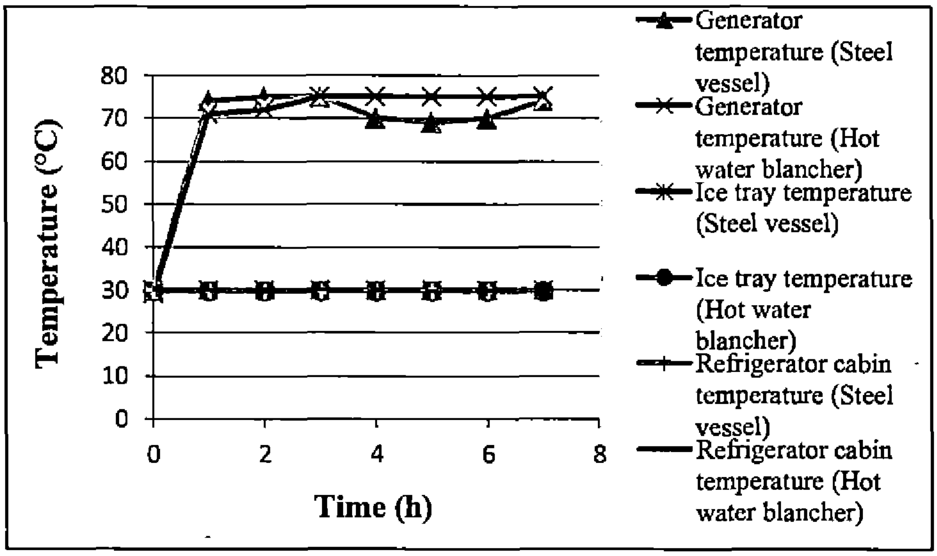


Figure 4.4 Performance of VARS using hot water at 100°C (Simulated Condition)

It was clear that 75°C generator temperature was not sufficient to regenerate ammonia from aqua-ammonia solution and also working of bubble pump so solar water heater producing water with temperature of 100°C was not sufficient. It may be sufficient if losses are reduced by using heating jacket instead of copper coil heat exchanger coil, because heating jacket have more heat transfer capability than copper tube heat exchanger tubes. When heating jacket around the generator tube, the heating medium is directly contact with generator tube so convective heat transfer will be more because generator pipe was is heated from all directions resulting good transfer of heat energy to generator tube.

4.7 Simulated studies on VARS using steam

Steam outlet of the pressure cooker was directly connected to the inlet of heat exchanger tube by using heat resistant rubber tube. Inlet steam temperature, heat exchanger surface temperature, generator temperature, absorber temperature, condenser temperature and ice tray temperature were measured. Experiments were repeated at different temperatures with different pressures.

4.7.1 Steam at 103°C and 106°C

In this steam was generated at 103°C with the pressure of 1.5 psi and 106°C with the pressure of 5 psi using 10 L capacity steam generator (pressure cooker) was continuously circulated in 7 hours through heat exchanger coil. Heat exchanger wall temperature, generator temperature, ice tray temperature and product temperature were measured in an interval of 1 hour. Temperatures obtained are given in Table 4.10.

Table 4.10 Performance of VARS using steam at 103°C and 106°C

Time, h	Heat exchanger wall temperature, °C		Generator temperature, °C		Ice tray temperature, °C		Refrigerator cabin temperature, °C	
	Steam at 103°C	Steam at 106°C	Steam at 103°C	Steam at 106°C	Steam at 103°C	Steam at 106°C	Steam at 103°C	Steam at 106°C
0	-	-	-	-	-	-	-	-
1	80	87	79	80	29.8	29.8	30	30
2	88	89	82	85	29.8	29.8	30	30
3	90	92	82	86	29.8	29.8	30	30
4	92	92	82	87	29.8	29.8	30	30
5	92	92	82	87	29.8	29.8	30	30
6	92	92	82	87	29.8	29.8	30	30
7	92	92	82	87	29.8	29.8	30	30

It reveals that after seven hour circulation of steam at 103°C and 106°C corresponding generator temperature obtained as 82°C and 87°C respectively but no effect on ice tray temperature and refrigerator cabin temperature.

Figure 4.5 shows the variation of generator temperature, refrigerator cabin temperature and ice tray temperature with time while using steam at 103°C and 106°C as heating mediums. It could be observed that generator temperature increases but product temperature and ice tray temperature showed straight line, indicating no change in temperatures.

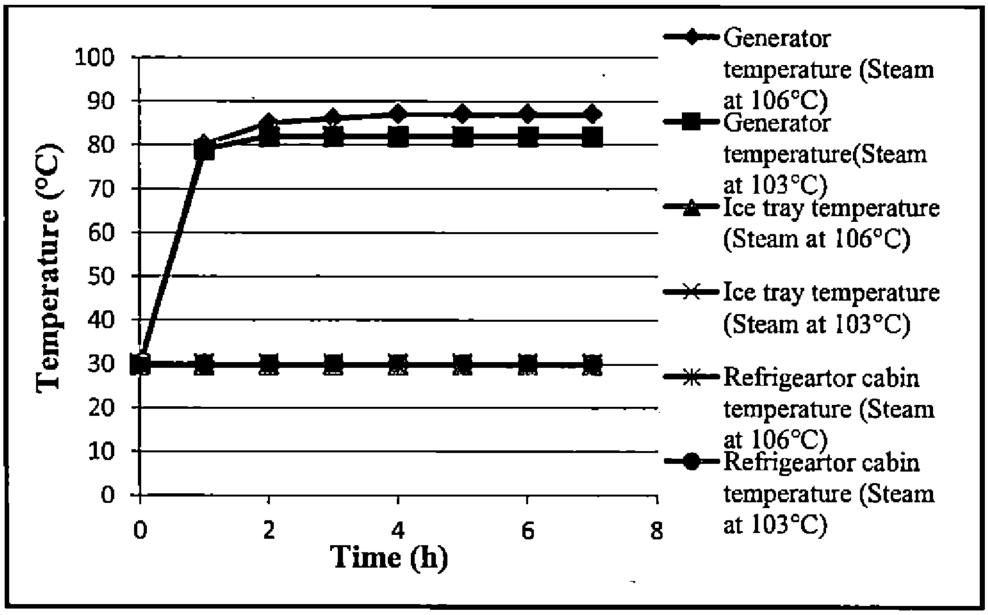


Figure 4.5 Performance of VARS using steam at 103°C and 106°C

Temperature of steam affects the generator temperature but these generator temperatures were not causing any change in ice tray temperature and product temperature. It was concluded that these generator temperatures were not sufficient for producing refrigeration effect because heating medium temperatures were not causing effective change in generator, for effective working of bubble pump and separation of ammonia from aqua- ammonia solution. Therefore it was concluded that steam at 103°C and106°C where not sufficient for producing refrigeration effect so this experiment was repeated for higher steam temperature.

4.7.2 Steam at 116°C and 121° C

Earlier results with steam at 103°C and 106°C showed that the temperatures were not causing any change in ice tray temperature and product temperature so this experiment was repeated for higher steam temperatures.

Steam at 116°C with pressure of 10 psi and steam at 121° C with pressure of 15 psi were generated in a steam generator of 10 L capacity and circulated through heat exchanger coil continuously for seven hours. Performance of refrigerator was also noted.

Table 4.11 Performance of VARS using steam at 116°C and 121°C

Time (h)	Heat exchanger surface temperature, °C		Generator temperature, °C		Ice tray temperature, °C		Refrigerator cabin temperature, °C	
	Steam at 116°C	Steam at 121°C	Steam at 116°C	Steam at 121°C	Steam at 116°C	Steam at 121°C	Steam at 116°C	Steam at 121°C
0	-	-	-	-	-	-	-	-
1	98	112.3	96	109.3	27.6	27.8	29.1	29.3
2	106	112.7	98	110.0	24.3	22.7	27.2	24.7
3	106	113.5	101	110.0	20.7	15.00	24.4	19.4
4	108	114.2	103	110.1	17.8	10.3	19.5	14.3
5	108	114.4	105	110.2	14.4	5.40	17.3	11.9
6	108	114.6	105	110.2	10.3	2.80	15.6	9.6
7	108	114.6	105	110.2	6.8	1.9	10.9	8.1

Table 4.11 reveals that generator temperature increases up to 105°C with the steam temperature of 116°C and 110.2°C with the steam temperature of 121°C, refrigerator cabin temperature and ice tray temperature decreased in these generator temperatures. In this case ice tray temperature attained 6.8°C after 7 h circulation of steam at 116°C but in the case of steam at121°C ice tray temperature attained 1.9°C. It was observed that minimum ice tray temperature obtained was 1.9°C with the generator temperature of 110°C after 7 h circulation of steam at121°C through heat exchanger tubes. It could be observed that higher generator temperature was obtained using steam at 121°C than steam at 116°C. Because of this increase in generator temperature, more cooling effect was produced. It was concluded that at higher generator temperatures higher will be the cooling as reported by Karthik, 2014. Absorber, condenser and evaporator temperatures were also measured when the temperature of ice tray attained lowest temperature. Temperatures of generator, absorber, condenser and evaporator were measured; corresponding values obtained are 110°C, 35°C, 50°C and 0°C

At higher generator temperature of 110.2°C ice tray temperature and product temperature decreased up to 1.9°C and 8.1°C respectively. It was concluded that generator temperature of above 100°C was sufficient for regenerating ammonia from aqua ammonia solution and effective working of bubble pump.

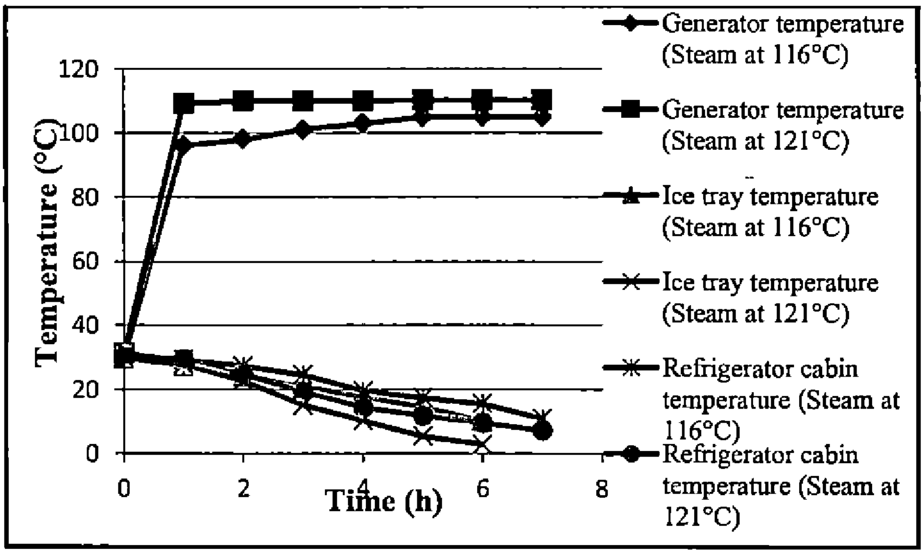


Figure 4.6 Performance of VARS using steam at 116°C and 121°C

Figure 4.6 shows the variation of generator temperature, ice tray temperature and refrigerator cabin temperature with time. It shows that refrigerator cabin temperature and ice tray temperature decreases with increase in generator temperature.

Ammonia from the evaporator coil extract heat from the ice tray. The quantity of heat extracted from the ice tray depends on the availability or amount of ammonia in the evaporator. More quantity of ammonia in the evaporator may cause more heat extraction from the inner compartment of the refrigerator. Quantity of ammonia in the evaporator depends on the amount of regeneration of ammonia from the generator and amount of regeneration of ammonia from the generator depends on the generator temperature. So cooling increases with increase in generator temperature.

4.8 Performance of refrigerator using electrical heater

Performance of the refrigerator using electric heater was tested. Temperatures of different components of refrigerator were measured using digital thermo meter. Variation of ice tray temperature and refrigerator shelves temperature with time are given in Table 4.12.

Table 4.12 Performance of VARS using electric heater

Time, h	**Generator temperature, °C**	**Ice tray temperature, °C**	**Refrigerator cabin temperature,**
0	29	29	29
1	100	13	24
2	110	6	19
3	115	2	16
4	125	1	15
5	138	1	12
6	140	1	9
7	142	0	6

From Table 4.12 it was observed that maximum generator temperature obtained was 142°C at the electrical heater temperature of 150°C. Lowest ice tray temperature and refrigerator cabin temperature obtained was 0°C and 6°C respectively. It was observed that higher generator temperature will produce more cooling effect (Bajpai 2012).

Figure 4.7 shows the variation of ice tray temperature and refrigerator shelves temperature with time. In this case ice tray temperature was reduced to 0° C. It was concluded that this higher generator temperature was sufficient for effective operation of bubble pump and separation of ammonia from aqua-ammonia solution.

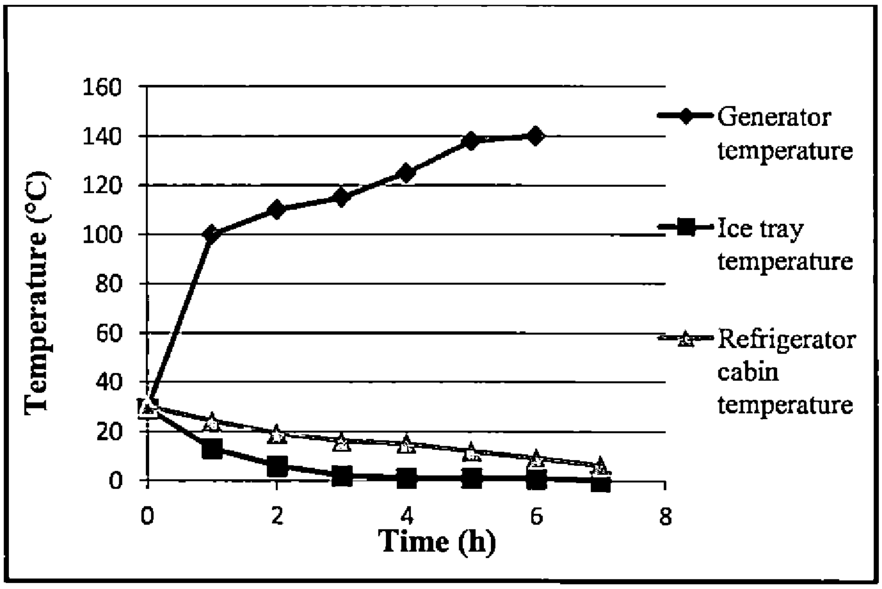


Figure 4.7 Performance of VARS using electric heater

Very high heat flux is required for effective function of bubble pump, which is not attainable by exchanging heat with a hot fluid. Higher temperatures are obtained in the generator because of the higher rates of heat transfer to the generator and also it produce more cooling as reported by Velmurugan *et al.* 2011.

4.8.1 Power consumption of electric heater

Power consumption was measured by using the watt meter. Measurements were taken for one hour at an interval of 10 minutes. Power consumption of the electric heater used for heating the generator tube was 70 Watts

4.8.2 Energy consumption of electric heater

Energy consumption was measured by counting rotation of disc in watt-hour meter. Readings were taken for 1 h at an interval of 10 min.

Table 4.13 Energy consumption of electric heater in the VARS

Time	No. of rotation of disc	Energy consumption, KWh
10 min	7	0.00583
20 min	14	0.01167
30 min	21	0.01750
40 min	28	0.02333
50 min	35	0.02917
1 h	42	0.03500

4.9 Performance of VARS using various heat sources

Performance of VARS using various heat energy sources were compared statistically. Performance of VARS was analyzed by using hot water below 100°C, hot water at 100°C, steam at 103°C, 106°C, 116°C, and 121°C and electric heater at 150°C. In these, steam at 116°C and 121°C and electric heater at 150° C produced refrigeration effect

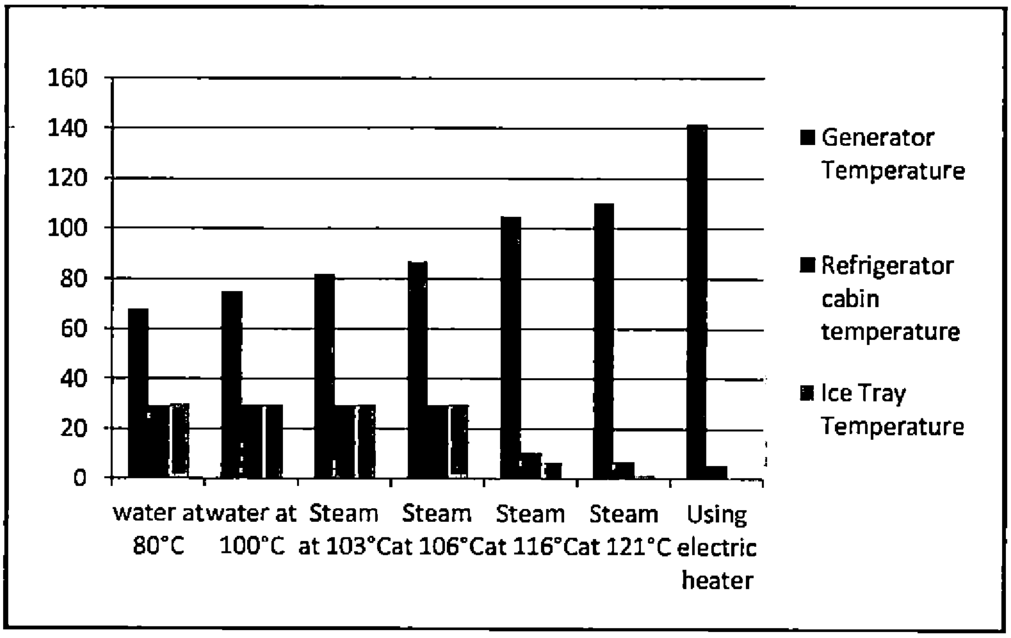


Figure 4.8 Performance of VARS using various heat sources

Figure 4.8 shows the variation of ice tray temperatures with generator temperatures by using various heating sources. Among this higher performance of the refrigerator was obtained by using electric heater because of the higher generator temperatures. In this case generator temperature obtained was above 140°C and the water in the ice tray became ice after 7 h.

In other cases steam at 116°C and 121°C produced refrigeration effect because in these steam temperatures the generator temperature attained were above 100°C. Higher generator temperatures were sufficient for effective operation of bubble pump as well as the separation of ammonia from aqua ammonia solution (Velmurugan *et al.* 2011)

No effect on ice tray temperature and refrigerator temperature while using water at 100°C and below 100°C as heating mediums. It was concluded that generator temperatures below 90°C was not sufficient to produce any cooling effect in refrigerator. The low generator temperatures were not sufficient for effective operation of bubble pump and since solar flat pate produced hot water of 80°C only, this temperature was not sufficient for producing cooling effect. So it was concluded that flat plate collector based solar water heater was not suitable for producing refrigeration effect because the heat input for this refrigeration system required a temperature higher than 90°C. Generator temperatures above 100°C were possible using solar collectors but construction becomes complex and expensive (Velmurugan *et al.* 2011). Therefore high performance solar collectors are needed to supply sufficient solar energy input as reported by Carg, 1982. Parabolic dish concentrating collectors will produce steam at higher temperatures as reported by Christopher, 2007, and Dascomb 2009. The concentrating solar collectors are suitable for producing refrigeration effect as reported by Karthik, 2014.

Table 4.14 Effect of temperature of heating mediums on the performance of VARS

Heat source	Generator temperature, °C	Ice tray temperature, °C	Product temperature, °C
Steam at 116°C	105[c]	6.8[c]	10.9[c]
Steam at 121°C	110 [b]	1.9[b]	8.1[b]
Electric heater at 150°C	142[a]	0[a]	6[a]

Higher performance of the refrigerator was obtained by using electric heater because of the higher generator temperature, this was proved statistically also. Experiments were conducted for a maximum of seven hours, statistically compared the temperature of generator, ice tray and refrigerator cabin in each hour repeated three times by using steam at 116°C, 121°C and at 150°C. The analysis indicated that ice tray temperature and refrigerator temperature were significantly varying from 1 hour to 7 hour but generator temperature was found to significantly vary up to 5 hours and after temperature difference was not significantly different. Which indicated that ice tray temperature and refrigerator cabin temperature depends on generator temperature. Increase in generator temperature causes decrease in ice tray temperature and refrigerator cabin temperature.

4.10 Coefficient of performance of the developed refrigeration system

COP of the refrigeration system indicates the performance of given refrigeration system. Higher value of the COP indicates the good performance of the refrigerator. Coefficient of performance of the absorption refrigerator mainly depends upon the generator heat input.

$$\text{Coefficient of Performance (COP)} = \frac{\text{Heat absorbed by the refrigerant in the evaporator (QE)}}{\text{Heat given to the generator (QG)}}$$

Heat absorbed by the refrigerant in the evaporator and heat given to the generator were calculated for different heating mediums. COP at various generator temperatures by using various heating mediums are given in Table 4.15.

Table 4.15 COPs of VARS with various heat energy sources

Heating medium	**Generator temperature, °C**	**Ice tray temperature, °C**	**COP**
Hot water at 80°C (Hot water from solar water heater)	68	No refrigeration effect produced	-
Hot water at 100°C	75	No refrigeration effect produced	-
Steam at 103°C	82	No refrigeration effect produced	-
Steam at 106°C	87	No refrigeration effect produced	-
Steam at 116°C	105	6.8	0.39
Steam at 121°C	110	1.9	0.47
Electric heater at 150°C	142	0	0.68

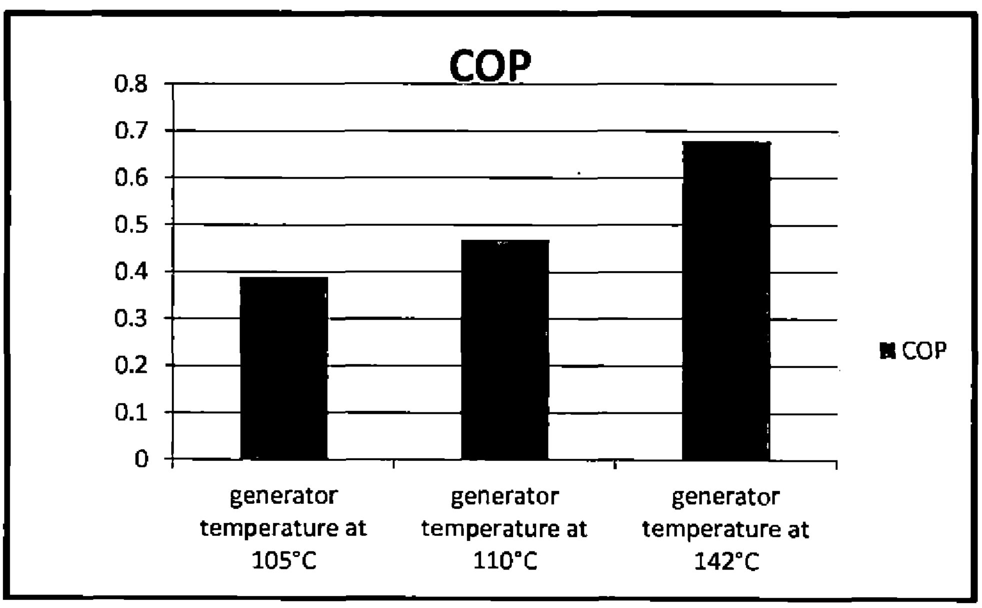


Figure 4.9 Variation of COPs with generator temperature

COP of the refrigeration system indicates the performance of given refrigeration system. Higher value of the COP indicates the good performance of the refrigerator. Coefficient of performance of the absorption refrigerator mainly depends upon the generator heat input. Since COP of the system was the ratio of heat absorbed by the refrigerant in the evaporator to the heat given to the refrigerant in the generator, COP of the refrigerator increases with increase in generator heat input (Pongsid and satha, 2002). Maximum Coefficient of performance (COP) of the refrigeration system was obtained was 0.68 at the higher generator temperature of 142°C.

In this case actual maximum COP obtained was only 0.68, it was very low compared to the maximum COP range of the absorption refrigerator. It may be due to the low temperature of generator.

4.11 Economic analysis of the developed absorption refrigeration system

Solar absorption refrigerator of 40 L capacity with 20 year life period was compared with vapour absorption refrigeration system of same capacity using electricity. Economic analysis was conducted to find the energy saving, cost saving and also to calculate payback period of the vapour absorption refrigeration system. Cost analysis was given in Appendix IV. Initial cost of absorption refrigerator including a concentrating type steam generator was calculated as Rs 57,500 /-. The payback periods is found as 12.21 years.

SUMMARY AND CONCLUSION

CHAPTER 5

SUMMARY AND CONCLUSION

Solar energy, a vast and inexhaustible source of energy, is one of the most promising sources of nonconventional energy currently available. The power from the sun intercepted by the earth is approximately 1.8×10^{11} MW which is much larger than the present consumption rate on the earth from all commercial energy sources. Thus, in principle, solar energy could supply all the present and future energy needs of the world on a continuing basis. When solar power, either thermal or photovoltaic, is used to provide energy to any refrigeration system, it is called as solar refrigeration system.

Many agricultural products like fruits, vegetables, milk, meat, fish etc., can be maintained in fresh conditions for significantly long periods of time, if they are stored under refrigeration or at low temperatures. Large quantities of these products are lost annually due to poor storage and lack of refrigeration facilities. Refrigeration is the process of extracting heat from a substance or space by any means usually at a low temperature.

Vapour compression refrigeration cycle is the most commonly used refrigeration system. It is used in most household refrigerators as well as in many large commercial and industrial refrigeration systems. The vapour absorption cycle using water-ammonia refrigerants were popular and widely used before the popularization of the vapour compression cycle. It lost much of its importance because of its low coefficient of performance (COP) which is about one fifth of that of a vapour compression cycle. VARS can be effectively used in industrial environments where plenty of waste heat is available. Since these systems run on low-grade thermal energy, they are preferred when low-grade energy such as waste heat or solar energy is available.

The main objective of the study was to design and develop a solar absorption refrigeration system and performance evaluation of the developed system. In this study 40 L capacity absorption refrigeration system was designed.

A commercially available three fluid refrigerator working on electricity was procured and it was modified for using heat energy. In this system three fluids were used, ammonia, hydrogen and water. Ammonia is used as a refrigerant, hydrogen being the lightest gas and inert, was used to increase the rate of evaporation of the liquid ammonia passing through the evaporator and water is used as the absorbent. Heat energy was used for operating this system. This Vapour absorption refrigeration System consists of generator tube, absorber unit, condenser and evaporator. Ammonia vapours from the evaporator was absorbed by water in the absorber forming aqua ammonia solution. The generator unit comprises of a bubble pump. Aqua-ammonia solution then goes to the bubble pump, Heat energy was given to the generator; by using this heat ammonia vapours were generated from the generator. This ammonia vapours push the weak solution through generator tube, this pushing action of bubbles cause weak solution return back to absorber tubes. Ammonia vapours separated from the generator tube enters to the condenser where it is converted to ammonia liquid. This liquid ammonia receives latent heat of vaporization from the product thereby cooling the product and converting the ammonia liquid into ammonia vapours. Hydrogen is incorporated with the evaporator to increase the rate of evaporation of the liquid ammonia by reducing partial pressure. Ammonia vapours with hydrogen enters into absorber tubes, from where hydrogen is separated and ammonia is returned back to absorber.

The solar radiations incident at KCAET Tavanur was studied using a lux meter. It was observed that maximum solar radiation intensity was obtained at 1.00 PM and it was 783.81 W/m^2. Variation of solar radiation intensity at Tavanur was studied in previous years by many and the results are in concurrent with their observations.

The commercially available, electrically operated, a three fluid absorber refrigerator was modified to operate using heat energy. A copper coil having 6mm outer diameter and 1 mm thickness (28 turns) was used as the heat exchanger through which hot water and steam at different temperatures were passed for assessing the performance of the refrigerator.

Maximum outlet temperature of solar water heater was obtained as 80°C with the maximum solar radiation intensity of 783.81 W/m^2. Performance of the refrigerator was tested in one hour duration using hot water from blancher that simulated the conditions of hot water from solar water heater.

In this case there is no change in ice tray temperature with the increase in generator temperature. Maximum generator temperature obtained was 68° C with the hot water temperature of 80° C. But this temperature had no effect on ice tray temperature as it maintained 29.5°C even after 7 hrs, and this showed that no refrigeration effect was produced during the experiment.

Again the experiment was repeated using higher water temperature. Water at 100°C generated in a steel vessel was circulated through heat exchanger tube and this experiment was not satisfactory and hence this experiment was repeated by using hot water blancher.

Water at 100°C was generated continuously in a hot water blancher was circulated through heat exchanger coil. In this experiment the maximum generator temperature obtained was 73°C but there was no effect on ice tray temperature. Hence it is presumed that this temperature (100^0C) is also not sufficient for the operation of the bubble pump. So these experiments were repeated using steam.

Steam at 103°C, 106°C, 116°C and 121°C were generated in a 10 L capacity steam generators at different pressures. It was found that steam at 103° C and steam at 106°C did not produce any refrigeration effect. But steam at 116°C and 121° C produced refrigeration effect. The generator temperature obtained were 105°C and 110°C and the corresponding ice tray temperatures were 6.8°C and 1.9°C respectively and refrigerator cabin temperature obtained was 10.9 and 8.1 respectively. This showed that the generator temperature above 105°C can produce refrigeration effect and it may be also noted that flat plate collector based solar water heater was not suitable for refrigeration purposes in three fluid absorption refrigeration systems because it produced hot water temperature up to 80°C only.. For effective working of bubble pump above 90°C temperature is required, so flat plate collectors are not suitable for this purpose.

Solar Concentrating collectors such as parabolic dish or parabolic trough collectors are well suitable for this application as they are able to produce steam temperatures higher than 100°C.

Experiments were conducted to understand the performance of the refrigerator using electricity, in the refrigerator a 70 W capacity of electric heater was used for normal operation. In this case after seven hours, generator temperature obtained as 142°C water kept inside the ice tray became ice. This revealed that by increasing generator temperature cooling also will increase.

Vapour absorption refrigeration system mainly consists of generator tube, absorber unit, and evaporator and condenser unit. In this case 40 L capacity and 70 W absorption refrigerator of each part was designed and dimensions were calculated. For calculating dimensions of the refrigerator concentration of ammonia in vapour (Kg of NH_3 / kg of solution) leaving generator (x_1), concentration of ammonia in strong solution (Kg of NH_3 / kg of solution) leaving absorber (x_2) and concentration of ammonia in weak solution (Kg of NH_3 / kg of solution) leaving generator (x_3) were taken from standard chart. Corresponding values obtained are 1.00, 0.538 and 0.434.

Enthalpy of ammonia at saturated points was also calculated using REFPROP software program based on temperature at various refrigerator components. Enthalpy of vapour ammonia leaving evaporator (h_1) corresponding generator temperature 0°C was obtained as 1605 KJ/Kg. Enthalpy of ammonia vapours leaving generator (h_4) for generator temperature of 110°C obtained was 1546.2 KJ/Kg and enthalpy of liquid ammonia leaving condenser (h_5) corresponding condenser temperature of 50°C was obtained as 583.77 KJ/Kg.

Enthalpy of aqua- ammonia solution determined using mathematical equations, These equations shows the relation among temperature, concentration and enthalpy by using this relation enthalpy of strong solution leaving absorber (h_2) and enthalpy of weak solution leaving generator (h_3) were calculated corresponding values obtained are 93.695 KJ/Kg and 261 KJ/Kg.

Mass flow rate of ammonia in evaporator and mass flow rate of weak and strong solutions were calculated. Corresponding flow rates obtained are 0.00411 Kg /min, 0.0183Kg /min and 0.02241 Kg /min. And also heat absorbed by the refrigerant in the evaporator (Q_E), heat given to the refrigerant in the generator (Q_G), heat rejected from the condenser (Q_c) and heat rejected from the absorber (Q_A) were calculated, corresponding values obtained are 11.16 KJ/min, 4.2 KJ/ min, 10.92 KJ/min and 3.956 KJ/min.

Coefficient of performance (COP) of the refrigerator was calculated at the different generator temperatures by using different heating mediums. Maximum Coefficient of performance (COP) of the refrigeration system obtained was 0.68 at the higher generator temperature of 142°C.

Economical analysis was conducted to find out the energy saving, cost saving and also to calculate payback period of the vapour absorption refrigeration system. The Solar absorption refrigerator of 40 L capacity with 20 year life period was compared with vapour absorption refrigeration system of same capacity using electricity. Initial cost of absorption refrigerator including a concentrating type steam generator was calculated as Rs 57500/- The payback period is found as 12.21 years. For an ordinary type solar water heater (200LPD) the payback period is calculated was 10 months.

Application of steam in food related industries is very common and a lot of heat is often wasted .As the present vapour absorption refrigeration system runs on low-grade thermal energy or steam, this is the best alternative for vapour compression refrigeration systems in food industries.

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APPENDICES

Appendix I

Daily solar radiation intensity in March, 2015

Date	Solar radiation intensity (Lux × 1000)						
	10:00 AM	11:00 AM	12:00 PM	1:00 PM	2:00 PM	3:00 PM	4:00 PM
01-03-15	675.00	725.00	810.00	895.00	860.00	795.00	743.00
02-03-15	655.00	796.00	852.00	895.00	873.00	810.00	750.00
03-03-15	588.00	746.00	850.00	889.00	860.00	795.00	768.00
04-03-15	683.00	824.00	865.00	895.00	864.00	815.00	786.00
05-03-15	565.00	710.00	840.00	890.00	860.00	785.00	695.00
06-03-15	792.00	846.00	888.00	896.00	830.00	760.00	445.00
07-03-15	800.00	865.00	836.00	870.00	810.00	752.00	590.00
08-03-15	538.00	754.00	845.00	860.00	760.00	685.00	492.00
09-03-15	684.00	813.00	885.00	860.00	710.00	645.00	510.00
10-03-15	485.00	695.00	800.00	820.00	720.00	630.00	500.00
11-03-15	593.00	655.00	812.00	833.00	685.00	640.00	495.00
12-03-15	618.00	675.00	815.00	865.00	726.00	460.00	430.00
13-03-15	710.00	838.00	859.00	865.00	725.00	610.00	485.00
14-03-15	469.00	610.00	750.00	810.00	685.00	565.00	390.00
15-03-15	375.00	485.00	610.00	650.00	590.00	510.00	440.00
16-03-15	368.00	420.00	785.00	760.00	560.00	465.00	385.00
17-03-15	388.00	450.00	725.00	775.00	685.00	610.00	555.00
18-03-15	500.00	756.00	852.00	860.00	752.00	664.00	510.00
19-03-15	435.00	510.00	785.00	820.00	865.00	750.00	475.00
20-03-15	352.00	425.00	693.00	785.00	810.00	710.00	395.00
21-03-15	453.00	485.00	596.00	653.00	785.00	645.00	315.00
22-03-15	510.00	562.00	710.00	825.00	860.00	652.00	412.00
23-03-15	395.00	456.00	690.00	785.00	810.00	596.00	510.00

24-03-15	415.00	465.00	685.00	765.00	795.00	685.00	345.00
25-03-15	685.00	745.00	825.00	855.00	758.00	595.00	410.00
26-03-15	412.00	700.00	810.00	850.00	764.00	520.00	490.00
27-03-15	710.00	820.00	860.00	855.00	710.00	520.00	315.00
28-03-15	520.00	584.00	650.00	755.00	765.00	510.00	320.00
29-03-15	485.00	550.00	685.00	796.00	810.00	610.00	420.00
30-03-15	452.00	525.00	615.00	765.00	785.00	625.00	410.00
Average	543.67	649.67	776.10	823.23	769.07	647.13	492.87

Appendix II

Date	**Ambient temperature**						
	10:00 AM	11:00 AM	12:00 PM	1:00 PM	2:00 PM	3:00 PM	4:00 PM
01-03-15	32.90	33.10	33.50	33.90	34.50	34.40	34.20
02-03-15	33.00	33.20	33.60	34.00	34.60	34.30	34.10
03-03-15	32.80	33.10	33.80	34.10	34.60	34.40	34.20
04-03-15	32.90	33.20	33.70	33.70	34.30	34.50	34.40
05-03-15	33.10	33.30	33.80	34.40	34.60	34.60	34.40
06-03-15	33.00	33.30	33.70	34.60	34.70	34.50	34.30
07-03-15	33.00	33.30	33.60	34.10	34.50	34.50	34.30
08-03-15	33.20	33.40	33.80	34.50	34.70	34.80	34.60
09-03-15	32.90	33.10	33.60	34.40	34.60	34.70	34.50
10-03-15	33.00	33.20	33.60	34.20	34.50	34.60	34.40
11-03-15	33.10	33.20	33.80	34.40	34.60	34.60	34.30
12-03-15	33.00	33.20	34.00	34.60	34.70	34.80	34.50
13-03-15	33.30	33.40	34.10	34.60	34.60	34.70	34.40
14-03-15	33.20	33.20	33.90	34.60	34.70	34.80	34.60
15-03-15	32.30	32.50	33.40	34.20	34.30	34.30	34.20

16-03-15	32.10	32.30	33.30	34.00	34.10	34.30	34.10
17-03-15	32.20	32.50	33.00	33.90	34.10	34.30	34.10
18-03-15	33.30	33.40	34.30	34.80	34.80	34.70	34.70
19-03-15	33.20	33.40	34.10	34.70	34.70	34.60\	34.50
20-03-15	33.30	33.40	34.20	34.80	34.70	34.60	34.60
21-03-15	33.20	33.30	34.00	34.20	3430	34.30	34.20
22-03-15	33.30	33.50	34.00	34.30	34.40	34.50	34.30
23-03-15	33.10	33.30	33.90	34.50	34.40	34.40	34.30
24-03-15	33.30	33.40	33.80	34.20	34.30	34.30	34.20
25-03-15	33.20	33.40	33.90	34.60	34.50	34.50	34.40
26-03-15	33.20	33.30	33.80	34.20	34.70	34.50	34.20
27-03-15	33.90	34.10	34.10	34.50	34.5	34.20	\34.20
28-03-15	33.2	34.20	34.20	34.40	34.6	34.50	34.30
29-03-15	33.8	34.00	34.20	34.30	34.40	34.60	34.40
30-03-15	33.9	34.00	34.20	34.20	34.30	34\10	34.10
Average	**33.10**	**33.31**	**33.83**	**34.33**	**3451**	**34.50**	**34.33**

Appendix III

Daily outlet water temperature of solar water heater per day in March, 2015

Date	**Outlet water temperature of solar water heater**						
	10:00 AM	11:00 AM	12:00 PM	1:00 PM	2:00 PM	3:00 PM	4:00 PM
01-03-15	39.80	47.00	56.60	65.00	73.50	76.20	79.90
02-03-15	44.00	49.00	58.00	66.28	75.00	79.30	79.30
03-03-15	39.00	46.00	55.60	68.20	75.90	80.40	80.00
04-03-15	40.90	48.00	59.30	68.00	77.40	81.00	80.80
05-03-15	40.60	47.60	59.90	68.20	75.00	79.80	79.60

06-03-15	44.00	51.00	60.80	73.00	81.10	81.30	80.80
07-03-15	41.00	50.00	60.30	70.60	76.60	78.90	78.00
08-03-15	39.50	46.00	58.70	67.90	75.60	79.60	79.40
09-03-15	40.40	48.00	58.90	69.10	76.30	79.00	78.60
10-03-15	37.80	45.00	56.20	68.30	76.30	79.20	79.50
11-03-15	39.60	46.00	52.30	63.30	74.40	78.20	78.00
12-03-15	39.00	46.00	55.30	67.30	76.30	79.90	79.50
13-03-15	40.90	49.00	58.30	67.40	76.40	80.30	79.80
14-03-15	39.00	46.00	53.60	64.10	73.60	77.60	76.50
15-03-15	39.20	44.00	56.30	64.90	74.30	79.60	79.00
16-03-15	40.60	45.00	53.50	63.20	71.60	76.20	76.00
17-03-15	38.60	41.00	52.40	63.20	70.30	78.90	79.00
18-03-15	41.00	47.00	56.30	68.20	76.30	80.20	79.30
19-03-15	44.20	52.00	61.20	70.30	76.90	79.90	79.60
20-03-15	40.80	48.80	56.30	65.40	74.60	78.60	78.60
21-03-15	41.30	49.00	58.60	69.60	76.30	80.00	79.80
22-03-15	43.20	49.00	58.60	67.20	76.30	79.80	79.00
23-03-15	44.00	51.00	60.70	69.30	76.50	79.30	79.10
24-03-15	42.60	48.30	57.10	68.20	75.80	77.30	77.00
25-03-15	41.00	49.60	56.90	66.20	73.50	78.60	78.30
26-03-15	37.00	39.00	43.00	49.00	50.00	51.30	49.90
27-03-15	46.30	51.60	59.30	67.00	76.00	79.30	78.60
28-03-15	45.00	53.00	62.00	69.00	77.00	80.30	79.00
29-03-15	44.00	52.00	63.00	69.60	76.20	79.00	78.80
30-03-15	46.00	53.00	62.00	68.30	73.30	78.90	78.00
Average	41.34	47.93	57.37	66.84	74.61	78.26	77.84

Appendix IV

Economic analysis of solar refrigeration system using a parabolic dish collector

Cost of solar refrigeration system

= cost of refrigerator + cost of heat exchanger + cost of parabolic dish collector

15000+2500+40000= Rs 57500/-

Year	Cash inflow	Discounted cash inflow	Year to be recovered
			57500
1	2885	2622.72	54877.27
2	3173.5	2885	51992.27
3	3490.85	3173.5	48818.77
4	3839.93	3490.85	45327.92
5	4223.92	3839.93	41487.98
6	4646.32	4223.92	37264.05
7	5110.95	4646.32	32617.73
8	5622.83	5110.95	27506.78
9	6184.27	5622.04	21884.73
10	6802.69	6184.25	15700.48
11	7482.97	6802.67	8897.80
12	8231.24	7482.94	1414.85
13	9054.37	8231	

'* Commercial cost of electricity per unit @ rate Rs 9.3/- is considered for calculation

= 1414.86/6816.39

=0.21*12

= 2.49

Payback period = 12.2 years

Cost analysis for solar water heater

Energy required for rise the temperature of 200L water from 29°C to 80°C =200000* 4.186*(80-29) = 42697200J= 42.7 MJ

1 KWh =1000W*3600s = 3.6 MJ

Hence energy needed = 42.7/3.6 =11.86 KWh/day

300 sunny days, therefore,11.86*300=3558 KWh/year

Cost of electricity =Rs 9.3/-

Total cost =3558*9.3 =Rs 33089.4/-

Payback period =27300/33089.4= 0.825*12= 10 month

Appendix V

Variation of different temperatures with time

Time (h)	Mean generator temperature, °C	Mean ice tray temperature, °C	Mean refrigerator cabin temperature, °C
0	30.1[f]	30[h]	30[h]
1	102[e]	23[g]	27[g]
2	106[d]	18[f]	24[f]
3	108[c]	13[e]	20[e]
4	113[b]	10[d]	16[d]
5	117[a]	7[c]	14[c]
6	118[a]	5[b]	11[b]
7	119[a]	3[a]	7[a]

DESIGN AND DEVELOPMENT OF A SOLAR REFRIGERATION SYSTEM

By

RAKHI. J. F

(2013 - 18 - 101)

ABSTRACT

Submitted in partial fulfillment of
the requirement for the degree of

MASTER OF TECHNOLOGY
IN
AGRICULTURAL ENGINEERING
(Agricultural Processing and Food Engineering)

Faculty of Agricultural Engineering & Technology
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2015

ABSTRACT

When solar power, either thermal or photovoltaic, is used to provide energy to any refrigeration system, it is called as solar refrigeration system. The main objective of the study was to design and develop a solar absorption refrigeration system and performance evaluation of the developed system. In this study 40 L capacity three fluid vapour absorption refrigeration system (VARS) was designed and a commercially available three fluid absorption refrigerator working on electricity was procured and modified for using heat energy.

The solar radiation at KCAET Tavanur was measured and it was observed that a maximum solar radiation intensity of 783.81 W/m^2obtained at 1.00 PM at Tavanur. The performance of the modified VARS was tested using hot water obtained from solar water heater. It was found that the hot water from the solar water heater was not sufficient to produce any cooling effect. Hence a hybrid system using hot water from solar water heater and subsequent heating of the hot water using other heating sources such as electricity and LPG were utilized and the hot water was converted to steam at high temperature. Under this new set up the system worked perfectly and produced refrigeration. The performance of the modified VARS was then tested in the laboratory under simulated conditions using water at 100°C, steam at 103°C, 106°C, 116°C and 121°C and using electric heater. The experiments with low temperatures could not produce any cooling whereas steam at116°C and 121°C and electric heater at 150°C produced refrigeration effect. The corresponding generator temperature obtained were 105°C and 110°C and the ice tray temperatures were 6.8°C and 1.9°C. The temperature obtained in the cabin of the refrigerator was 10.9°C and 8.1°C which was ideal for keeping fruits vegetables and other perishable items. For effective working of this system using only on solar energy, instead of flat plate collector, a solar steam generator that could produce steam can be used.

സംഗ്രഹം

സൗരോർജ്ജ ശീതീകരണ സംവിധാനത്തിന്റെ ഡിസൈനും വികസനവും

ശീതീകരണത്തിനായി ഫോട്ടോവോൾട്ടായിക് ഊർജ്ജം അല്ലെങ്കിൽ സൗര താപോർജ്ജം ഉപയോഗപെടുത്തുന്നതിനെയാണ് സോളാർ ശീതീകരണ സംവിധാനം എന്ന് പറയുന്നത്. ഈ ഗവേഷണത്തിന്റെ പ്രധാന ലക്ഷ്യം സൗരോർജ്ജം ഉപയോഗിച്ചുള്ള ഒരു സോളാർ ശീതീകരണ സംവിധാനം ഡിസൈൻ ചെയ്തു പരീക്ഷിക്കുക എന്നതായിരുന്നു. ഇതോടനുബന്ധിച്ച് നാൽപത് ലിറ്റർ ശേഷിയുള്ള ഒരു സൗരോർജ്ജ ശീതീകരണ സംവിധാനം ഡിസൈൻ ചെയ്യുകയുണ്ടായി. വൈദ്യുതി കൊണ്ട് പ്രവർത്തിക്കുന്ന ഒരു ശീതീകരണ സംവിധാനം വാങ്ങി അത് സൗരോർജ്ജം / താപോർജ്ജം കൊണ്ടു പ്രവർക്കുന്ന രീതിയിൽ മാറ്റിയെടുക്കുകയും ചെയ്തു.

കെ. സി. എ. ഇ. ടി തവനൂർ ക്യാമ്പസിൽ ലഭ്യമായ സൗരവികിരണ തീവ്രത അളക്കുകയും, ഇവിടെ ലഭ്യമായത് 783.81 W/m^2 ആണെന്ന് കണ്ടെത്തുകയും ചെയ്തു. പുതുതായി രൂപകൽപന ചെയ്ത റെഫ്രിജിറേറ്ററിൽ കെ. സി. എ. ഇ. ടി യിൽ ലഭ്യമായിരുന്ന സോളാർ വാട്ടർ ഹീറ്ററിൽ നിന്നുമുള്ള ചൂട് വെള്ളം ഉപയോഗിച്ച് പ്രവർത്തിപ്പിക്കാൻ ശ്രമിച്ചെങ്കിലും ചൂട് വെള്ളത്തിന്റെ ഊഷ്മാവ് കുറവായതിനാൽ ശീതീകരണം നടക്കുകയുണ്ടായില്ല. എന്നാൽ ചൂടു വെള്ളം ഒരു പൈപ്പ് വഴി ഒരു പ്രഷർ കുക്കറിൽ എത്തിച്ച ശേഷം ചൂടാക്കി, നീരാവിയാക്കി മാറ്റിയ ശേഷം റെഫ്രിജിറേറ്ററിൽ നല്കിയപോൾ ശീതീകരണം മികച്ച രീതിയില് നടക്കുന്നതായി കാണപെട്ടു.

വികസിപ്പിച്ചെടുത്ത റെഫ്രിജിറേറ്ററിന്റെ പ്രവർത്തനം വിവിധ ഊഷ്മാവിൽ തുടർ പഠനം നടത്തിയത് കെ. സി. എ. ഇ. ടി ലാബിലാണ്. 100°C ലുള്ള ചൂട് വെള്ളം ഉപയോഗിച്ചും 103°C, 106°C, 116°C, 121°C നീരാവിയിലും, ഇലക്ട്രിക് ഹീറ്റർ ഉപയോഗിച്ച് 150°C താപനില ഉയർത്തിയും പരീക്ഷണങ്ങൾ ചെയ്തപ്പോൾ ആദ്യത്തെ മൂന്ന് പരീക്ഷണങ്ങൾക്ക് ശീതീകരണം ഉണ്ടാക്കാൻ കഴിഞ്ഞില്ല. എന്നാൽ 116°C, 121°C, 150°C എന്നീ ഊഷ്മാവുകളിൽ ശീതീകരണം നടക്കുന്നതായി കാണപെട്ടു. ശീതീകരിണിയിലെ ജനറേറ്റർ എന്ന ഭാഗത്ത് 105°C ഉം 110°C ഉം ഐസ് ട്രേയിൽ 6.8°C

ഉം 1.9°C ലഭിക്കുന്നതായി കാണപെട്ടു. തണുപ്പിക്കാനുള്ള കാബിൻ ഊഷ്മാവ് 10.9°C ഉം 8.1°C യുമായും കാണപെട്ടു. ഈ ഊഷ്മാവ് പഴങ്ങളും പച്ചക്കറികളും കേട്ടുകൂടാതെ സൂക്ഷിക്കുന്നതിന് ഏറ്റവും അനുയോജ്യമായതാണ്. അതായത് വൈദ്യുതി ഇല്ലാതെ ചൂട് കൊണ്ട് മാത്രം പ്രവർത്തിക്കുന്ന ഈ റെഫ്രിജിറേറ്റർ ഭക്ഷ്യ വസ്തുക്കൾ കേട്ടുകൂടാതെ സൂക്ഷിക്കാൻ ഉപകരിക്കുന്നു.

ഒരു സോളാർ കോൺസെൻട്രേറ്റർ ഉപയോഗിക്കുകയാണെങ്കിൽ നീരാവി ഉല്പാദിപ്പിക്കാനും അതുപയോഗിച്ച് ഈ റെഫ്രിജിറേറ്റർ മറ്റ് ഊർജ്ജങ്ങൾ ഒന്നും ഉപയോഗിക്കാതെ, സൗരോർജ്ജം മാത്രം ഉപയോഗിച്ച് പ്രവർത്തിപ്പിക്കാനും കഴിയുന്നതാണ് .