MODELING OF SMALL SCALE SOLAR ABSORPTION REFRIGERATION SYSTEM USED IN DAIRYING PROCESS

A thesis submitted to the school of Mechanical and Industrial engineering in partial fulfillment of the requirement for the Degree of Master of Science in Mechanical Engineering (Thermal Engineering Stream)

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This is to certify that the thesis prepared by Fikadu Geremu, entitled, "Modeling of small scale solar absorption refrigeration system used in dairying process" submitted in accordance with the requirements for the Degree of Master of Science (Thermal Engineering) complies with the regulation of the university and meets the accepted standards with respect to originality and quality.

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Declaration

I hereby declare that the work which is being presented in this thesis entitled "modeling of small scale solar absorption refrigeration system used in dairying process" is original work of my own, has not been presented for a degree of any other university and all the resource of materials used for this thesis have been duly acknowledged.

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This is to certify that the above declaration made by the candidate is correct to the best of my

Knowledge.

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Dr.-Ing. Demiss Alemu (Advisor) **Date**

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ABSTRACT

Dairy and milk are an important sub-sector in the national economies of many developing countries, providing food, regular incomes for households and sustainable livelihoods for rural farmers and small and microenterprises. Milk constitutes an important part of a balanced and healthy diet with its excellent nutritional properties. When we use milk direct for nutrition purpose it would affect human health, Refrigeration is crucial important to cool milk. Refrigeration and air conditioning consumes about 15% of world energy today, solar energy is one of the most efficient, clean and affordable energy alternatives available now a days and current concerns about global warming and ever-increasing energy rates. Absorption refrigeration system needs a low running cost on cheap fuel such as solar energy, biomass and waste heat while vapour compression refrigeration system need high cost due to electricity as energy input. Absorption refrigeration has been most frequently adopted for solar refrigeration and reducing the emission of greenhouse gases. To remove harmful bacteria that would affect human health, some processing is required. In rural area where electric grid is unavailable some of milk is wasted as bulky when they use traditional ways and they wouldn't pasteurize milk. To overcome such a problem pasteurizing and refrigeration of milk using solar energy is the important things. Solar absorption refrigeration system has the potential to improve the life quality for people who lives in remote areas or areas with insufficient electricity, with the objective of modeling small scale solar absorption Refrigeration system for dairying process for pastoralists of Afar region with capacity of 500 liters. The modeling of solar absorption refrigeration system used in dairying process is done, by using mathematical model and dynamic modeling of the system, using MATLAB software. The coefficient performance of the solar absorption refrigeration system varies from 0.45 to 0.94and milk temperature is cooled from 32 ^{o}c up to 4 ^{o}c using MATLAB simulation. the data used for modeling is gained from Chuye milk and milk processing, A two electromechanical companies and National metrology agency. The life cycle cost of solar absorption refrigeration system is calculated, the life cycle cost of solar absorption refrigeration is much less than that of vapour compression refrigeration system used in chuye milk and milk processing company. The project is economical feasible.

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LISTS OF TABLES

NOMENCLATURE

A	The capital recovery factor of the annual payment
Ao	The annual operating cost
a and b	Regression coefficients for the linear correlation of solar radiations.
B	The capital recovery factor of the once payment
COP	Coefficient of performance
CA	The initial capital cost
CAUX	The auxiliary energy cost
CE	The electrical energy cost
CM	The maintenance costs
CR	The running cost
CRE	The replacement cost
CS	The salvage cost
DX	Direct expansion
FTC	Flat Plate collector
ETC	Evacuated tube solar collector
G_{sc}	Solar constant with the value [w/m^2]
H	Monthly average of the daily global radiation on horizontal surface [kwhr./m^2/day].
H_d	Diffuse radiations [kwhr. /m^{2}day]
H_b	Beam radiations [Kwhr./m^2/day]
Ho	Average value of extraterrestrial solar radiation on a horizontal surface for Each Month [kwhr /m^{2}day].
LCCA	The life cycle costs analysis
Q_{cold}	The useful cold at evaporator, kwh
Q_{heat}	The driving heat at generator, kwh
PC	Pre - cooler
Po	The present value
P_{Re}	The future replacement cost

Rb	Ratio of beam radiation on the tilted surface to that on a horizontal surface
r_t	The ratio of hourly to daily total radiation
r_d	The ratio of hourly to daily diffuse solar radiation
STHE	Shell and tube heat exchanger
S	Sun shine hour (bright) in a day [hr].
ST	Solar time [hr].
SLCCA	The specific system life cycle costs, kwh-1
K_t	Clearance index
T_d	Sunrise and sunset hour angle [^{O}C].
δ	Declination angle of the earth [^{O}C].
VCR	Vapour compression refrigeration
VAR	Vapour absorption refrigeration
V_w	Wind speed [m/sec]
Z	Elevation of the site village above sea level [km].
ϕ	Latitude of the site [o].
PV	Solar photovoltaic

CHAPTER ONE

1. INTRODUCTION

1.1 Background

1. Dairying process

Milk production began as early as 6,000 years ago during the agricultural revolution. In this period, the ancient man learned to domesticate animals and recognized the nutritive value of their milk[1]. From the early years until now, milk is a source of both energy and necessary nutrients for growth. Milk also contains antibodies that are responsible to protect the young mammal against infection and diseases[1]**.**The dairy industry began when settlers brought cows to colonial America in the early 1600s. Until the mid-nineteenth century, a cow's owner consumed most of the milk it produced. Some milk and other dairy products were sold to people who lived nearby and did not have their own cows. In the nineteenth century, technological advancements led to the growth of the dairy industry. Refrigerated railroad cars allowed producers to ship their milk long distances to urban areas. Milk began to be pasteurized, which involves treating milk by heating it to kill disease-causing bacteria[2]. Commercially manufactured cheese became available as well**.** The dairy industry produces a wide variety of products from milk, using a range of different processing techniques. Internationally, approximately 60% of milk produced is used for fluid-milk products, 25% for cheese and whey products, while the remaining 15% is used for butter, milk concentrate, milk-powder products, and cream[2]. The global dairy industry accounted for 6.4% of the agricultural products and was valued at USD 69B in 2013 and this is likely to increase due to both the increasing wealth in developing nations and the realization that milk protein is a key nutritional aid. Dairying is an essential part of integrated farming systems in developing countries and countries in transition, where milk accounts for a significant share of livestock GDP[3]. Milk and dairy are an important sub-sector in the national economies of many developing countries, providing food, regular incomes for households and sustainable livelihoods for rural farmers and small and microenterprises[3].

The dairy sector in Ethiopia is expected to continue growing over the next one to two decades given the large potential for dairy development in the country, the expected growth in income, increased urbanization, and improved policy environment[4]. Ethiopia's estimated livestock population is often said to be the largest in Africa. The country currently manages the largest livestock population in Africa, estimated at 29 million cattle, 24 million sheep and goats, 18 million camels, 1 million equines and 53 million poultry[4]. Compared to other countries in Africa, Ethiopians consume less dairy products. Per capital consumption of milk in Ethiopia is as low as 17 kg per head while the average for Africa is 26 kg per head (Gebrewold et al. 1998).

In the first half of the 20th century, dairying in Ethiopia was mostly traditional. Modern dairying started in the early 1950s when Ethiopia received the first batch of dairy cattle from United Nations Relief and Rehabilitation Administration (UNRRA)[4]. Commercial liquid milk production started on large farms in Addis Ababa and Asmara (Ketema 2000). To facilitate growth of the sector, UNICEF established a public sector pilot processing plant at Shola on the outskirt of Addis Ababa in 1960. The plant started by processing milk produced by the large farms. Milk production in Ethiopia increased significantly during 1960s[5]. In the late 1980s, agriculture in Ethiopia contributed about 45% of national GDP while the Livestock sector contributed about 40% of agricultural GDP (18% national GDP) and 30% of agricultural employment[5]. The estimated milk production in the year 2008/09 is 2.76 billion liters from cattle and 162 million liters from camels (CSA 2009). The milk production systems in Ethiopia are categorized as pastoral, agro-pastoral, mixed crop livestock system, urban, peri-urban, and commercial[5]. One of the distinctions between the various systems is the type of breeds and purpose of holdings. Small-scale farmers in the various milk sheds are now maintaining crossbred cows that are capable of producing 800 to 1200 liters of milk/cow/lactation and sale milk to cooperative societies and commercial milk collectors. About 30% of the livestock population in Ethiopia is found in the pastoral areas; which comprise 50% of the total land area of the country. Pastoralist is the major dairy production system in the lowland. Livestock does not provide inputs for crop production but they are the very backbone of their owners providing all of the consumable and saleable outputs, like for example milk, and regarded as insurance against adversity. Milk production is dependent on season due to the rainfall pattern that influences feed availability (Hiwot, 2013) [6]. Urban and peri-urban system is largely found in the highlands where mixed-crop livestock farming is practiced as well as within urban centers. Economic factors have been dominant in determining the locations of exotic dairy cattle in these urban and peri-urban areas since the milk-production of exotic cattle far exceeds that of indigenous stock. Commercial systems are known for the adoption of modern technology and production of different types of dairy products including cheese, cream, yoghurt and butter [5] . The dairy sector in Ethiopia can also be categorized based on market orientation, scale, and production intensity[4]. Three major production systems are identified: traditional small holders privatized state farms and urban and peri - urban systems (GebreWold et al. 2000). The traditional smallholder system, roughly corresponding to the rural milk production system, produces 97 percent of the total national milk production and 75 percent of the commercial milk production. The urban and peri-urban milk production system, the third production system, includes small and large private farms in urban and peri-urban areas concentrated in the central highland plateaus (Felleke and Geda 2001). This production system is now expanding in the highlands among mixed crop-livestock farmers, such as those found in Selale and Holetta, and serves as the major milk supplier to the urban market (GebreWoldet al.2000).

2. Solar energy

Solar energy one important contribution for the reduction of fossil fuel consumption and harmful emissions to the environment, abundantly available[7] and it is also quite attractive due to its low operating cost[8].Solar technology is not new, its history spans from the 7th century B.C. We started out concentrating the sun's heat with glass and mirrors to light fires[9]. Today, we have everything from solar-powered buildings to solar powered vehicles. Solar energy is the result of electromagnetic radiation released from the sun by the thermonuclear reactions occurring inside its core. It has been calculated that the total solar radiation transmitted to the earth is approximately 1.74×10^{17}w, whereas the overall energy consumption of the world is approximately 1.84×10^{13}w[9]. Solar energy is currently a subject of great interest, solar electrical and thermal powered refrigeration systems can be used to produce cooling. The first is a photovoltaic-based solar energy system, in which solar energy is initially converted into electrical energy and then utilized for producing the refrigeration, similar to conventional methods. The second one utilizes solar thermal energy to power the generator of an absorption refrigeration system. Due to the advantages of solar thermal systems over solar photovoltaic systems [9].The photovoltaic phenomenon has been recognized since 1839, when French physicist Edmond Becquerel was able to generate electricity by illuminating a metal electrode in a weak electrolyte solution [10].

Solar thermal is currently providing only 0.5% of total primary energy need and solar photovoltaic has even lower energy supply ratio (0.04%) [11]. Solar thermal is one of the most cost effective renewable energy technologies and has huge market potential globally. It, representing more than

90% of the world-installed solar capacity, is utilized for various purposes including domestic hot water generation and space heating, solar assisted cooling and industrial process heating[11]. Solar thermal cooling is an interesting application because usually maximum cooling loads take place when most solar radiation is available. Solar thermal systems are especially interesting in remote areas or islands where conventional cooling is difficult and solar energy is always available[12][13]. Solar thermal refrigeration systems use heat energy from solar radiation to generate power to meet the cooling requirements. The heat energy collected from solar radiation is used to vaporize refrigerant. The cooled refrigerant vapour is then condensed and used to remove heat from warm food through an evaporator. The final cooling temperature required will depend on the combination of absorbent and refrigeration used in the system[3]. The utilization of solar energy in absorption refrigerating machines is limited by the condition that only heat collected at temperatures greater than a minimum value can be used in the refrigerating machine [14].Thus, the closed sorption refrigeration systems appear as a good alternative to replace the classical refrigeration and air-conditioning systems with more environmentally friendly systems that can be powered by renewable energy sources and contribute to sustainable development [9]. The overall efficiency of a solar refrigeration system is the product of the solar collection efficiency and the coefficient of performance of the absorption system [15].The advantage of solar refrigeration systems is that they displace some or all of the conventional fuel use. The operating costs of a solar refrigeration system should be lower than that of conventional systems, but at current and projected fuel costs, this operating cost savings would not likely compensate for their additional capital costs, even in a long-term life-cycle analysis. The major advantage of solar refrigeration is that it can be designed to operate independent of a utility grid[15].

3. Solar absorption refrigeration technologies

In the 1970s, solar refrigeration received great interests when the world suffered from oil crisis. The absorption cycle is a process by which refrigeration effect is produced through the use of two fluids and some quantity of heat input, rather than electrical input[16]. The first evolution of an absorption system began in the 1700s. It was observed that in the presence of H_2SO4 (sulphuric acid), ice can be made by evaporating pure H_2O within an evacuated container[9]. Initial experiments on absorption were conducted in 1810 by John Leslie with sulphuric acid. Two jars were connected by a tube, one containing H_2SO4 and the other water. The acid's very high affinity for water results in the absorption of water vapour evaporating from the second jar [17].In 1859, Ferdinand Care designed an installation that used a working fluid pair of ammonia/water (NH_3/H_2O). In 1950, a new system was introduced with a water/lithium bromide (H_2O/LiBr) pairing as working fluids for commercial purposes [9]. Most studies in solar absorption technologies have been limited to solar heating applications. The absorption cycle, however, has been known and used since the early 1900s for cooling purposes such as food refrigeration and building cooling [18]. Solar absorption systems are more suitable than conventional refrigeration systems because pollution-free working fluids (instead of chlorofluorocarbons) are used as refrigerants. Vapor compression cycle is described as work-operated cycle because it requires work to run the compressor for elevating the pressure of the refrigerant. On the other hand, absorption cycle is referred as heat operated cycle because most of the operating cost is associated with providing the heat that drives off the vapor from the high pressure liquid or solid [19]. In the vapor compression system, energy in the form of work is normally much more valuable and expensive than energy in the form of heat that can be obtained by solar energy, a fossil-fueled flame, waste heat from factories, or district heating. In fact, possible to design an absorption system that does not require any mechanical power input relying instead on gravity[15].The absorption refrigeration system may be driven directly by the heat transfer fluid in the refrigerant tank, or indirectly by another heat transfer fluid using a heat exchanger between the tank and the chiller. Originally used for refrigeration, the absorption cycle was invented in the mid-1800s but faded quickly after higher- performance is vapor compression refrigeration was introduced. Since the 1960s, absorption technology, especially the absorption refrigeration cycle for air conditioning, has been gradually developed because it primarily consumes low-level heat energy and can play an irreplaceable role in renewable energy use and waste heat recovery [8].The factors that make absorption technology attractive to both commercial and industrial markets namely the use of waste heat in order to provide cooling and heating have been investigated thus far.

1.2 Statement of Problem

Dairying process is important for human health and economics activity of one country such as Ethiopia in which the majority of economic activity is depend on agriculture and agricultural products. Milk inside a cow has a temperature of about 38°C. When we use directly without processing, the harmfully bacteria would affect human health, to eliminate such harmful things, some process is required. Most of milk in rural is processed using traditional ways, when milk is processed traditional, it is manually operated, affect the quality of milk, doesn't eliminate harmful bacteria and consume more energy. Milk has to be cooled to ensure that bacteria don't affect its quality. To overcome such difficult, Refrigeration is a vital important in dairying process. Milk cooling in dairying process account about 30-40% of the total energy costs of operating system. Every day to day of human activity depends on energy, the largest increases in energy demand. Refrigeration and air conditioning consumes around 35% world energy. In area where grid supply of electricity unavailable (in rural area), Energy from conventional sources is becoming increasingly expensive, adding to the difficulty of milk pasteurization, however, such area usually have a wealth of solar energy , free sources and does not affect the degradation of the environment. Solar energy can play big role in application of refrigeration division free and inexhaustible. Producing safe milk by pasteurization process is a vital issue for human health and increases their income. To overcome such a problem solar absorption refrigeration system has a potential to improve the life quality for people who live in rural areas or areas with insufficient electricity is available.

1.3 OBJECTIVE

4. General objective

Modeling of small-scale solar absorption refrigeration system used in dairying process for pastoralist area of Afar region, Ethiopia with capacity of 500 liters.

5. Specific objective

- Data collection on existing small- scale dairy processed company in Ethiopia.
- Energy demand analysis.
- Selection of appropriate of system configuration used for modeling.
- Mathematical modeling of systems component.
- Dynamic modeling overall system using appropriate software.
- Performance analysis of overall system

1.4 Organization of paper

This paper contains nine chapters. It starts with introductory part which deals with thebackground about dairying, solar energy and absorption refrigeration, statement of the problem and objectives of the thesis.The second chapter is a literature review. In this chapter review different materials about cooling systems used in dairying process, different type of refrigeration system, refrigerant used and as well as different type of energy used in dairying process. The third chapter deals with methodology used and selection of appropriate system used and data analysis. The fourth chapter deals solar energy and storage tank modeling. The fifth chapter deals with Mathematical modeling of systems component. The chapter seven is Economic analysis of solar absorption refrigeration system. Chapter eight is about the result and discussion. Finally end chapter is conclusion and recommendation of thepaper, which summarizes the result obtained and also there are references andappendixes.

CHAPTER TWO

2 LITERATURE REVIEW

2.1 Different milk cooling methods

Milk is cooled by means of heat conduction from the water–ice tank to the milk[3]. Milk inside a cow has a temperature of about 38°C but it usually drops to about 35°C once it passes through the collection can and some pipe work. This warm milk has to be cooled to ensure that bacteria do not affect its quality. When we use directly without processing, the harmfully bacteria would affect human health, to eliminate such harmful things, some processing is required. Cooled to and maintained at 7°C or below within 3 hours of the completion of milking and kept at or below 7°C until it is collected or the next milking. It is understood that in some other milk producing countries (e.g. Australia), storage at 4°C is the norm. There are different cooling methods from ancient up to know.

2.1.1 Cooling with basic facilities

Several systems are available for cooling milk. The simplest systems use water from a main or well. Simple systems of cooling that use water will bring the milk to a temperature only 3 – 5 °C above that of used water[20].However, is not advisable, where well water is also used for drinking, because the immersion of cans easily leads to contamination of the well.

2.1.1.1 Evaporate, heating method

Many cooling processes involve the evaporation heat of a liquid. If a liquid evaporates, it needs heat. This heat is taken from the surroundings of the evaporating liquid[20].An early example of cooling by evaporation can be found in ancient Egypt. Stone bottles, called Gandis, were filled with water. Because of the porous material, some of the water seeped through to the outside of the bottle wall and evaporated in the figure 1. This evaporation took the heat out of the bottle and therefore out of the water inside.

Figure 1: Evaporation cooling using gandis in ancient Egyptian[20].

2.1.1.2 Cooling rings

When running water is available, milk can be cooled by putting a perforated tubular ring around t0he neck of the can of warm milk[21]. After the ring has been connected to the mains, water will spray onto the can and flow over its surface.

2.1.1.3 Surface coolers

Surface coolers consist of a series of small-diameter horizontally arranged tubes. Mounted on top of each other, these tubes terminate at each end in a header. The headers connect the tubes, thus allowing the cooling agent to circulate through them; the warm milk is distributed over the surface of the cooler[3]. Surface cooling system, also called open cooling system, is simple, but requires a proper sanitization programmed. Special care must be taken to prevent airborne contamination.

2.1.1.4 Ice-cones

If small amounts of milk have to be collected and transported over long distances, and it is not technically or economically feasible to cool the milk in advance, metal ice-cones may be used in figure 2. These cones are inserted in the milk cans, so that the rim of the cone rests on the collar of the can and fits sufficiently tightly to prevent milk splashing out during handling and transport[21].

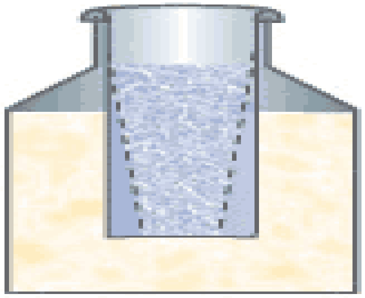

Figure2: Ice –cone cooling method[21].

2.1.1.5 Water tanks:

The simplest cooling system involves an open tank with cold water. Milk cans have been inserted into the tank as shown in figure 3, where they are immersed in the water up to their neck[3]. The tank may be covered with a lid to protect the milk from flies and dust. If well water or water from a main supply is used, this system only enables slow cooling to comparatively high temperatures. If well water or water from a main supply is used, this system only enables slow cooling to comparatively high temperatures[21].To limit losses of cold by radiation, the tank and its cover must be insulated.

Figure 3:**Cooling with water and almost always appropriate techniques**[3][21]**.**

6. Modern milk cooling systems:

It is important to choose the correct type of cooling system in relation to energy consumption and this choice will be affected by the source and cost of the energy supply[3]. The design and quality of the equipment will also have a direct impact on energy consumption. If milk is cooled in a modern way, electricity is needed to generate the temperature required[20].

7. Classification of modern cooling system

Two types of refrigerated milk cooling tanks are commonly used: the direct expansion and the ice bank tank system[3][20]. Ice bank tanks cool milk more quickly than DX tanks, all cooling systems are designed to cool any volume of milk to 10 °C within two hours and 4 °C within four hours[20].

2.1.1.6 Direct expansion cooling

This is the most common milk cooling system. Since direct expansion tanks do not have a cold buffer, energy must always be available. In DX cooling, the milk is cooled directly and agitated after arrival in the tank as represent in figure 4.Direct expansion milk cooling and storage tanks use refrigeration evaporator pipes or specially designed flat pillow-shaped plates attached to the outer side of the tank containing the milk[20]. DX systems can also be up to 50 percent more energy-efficient than ice bank cooling, but require larger compressors. Two types of tank are available for the DX system[3]: one in which the refrigerant gas flows inside copper pipes around the surface of the milk cooling tank allowing heat transfer from milk to the refrigerant; and the other in which the cooling gas flows through a specially designed stainless steel jacket (DX plates) of which one face serves as the milk holding chamber and the other is covered with insulation material. The bottom of the tank has been designed as an evaporator, while the heat of the milk goes through the stainless steel wall to the refrigerant. The refrigerant evaporates, which takes the heat away from the milk.

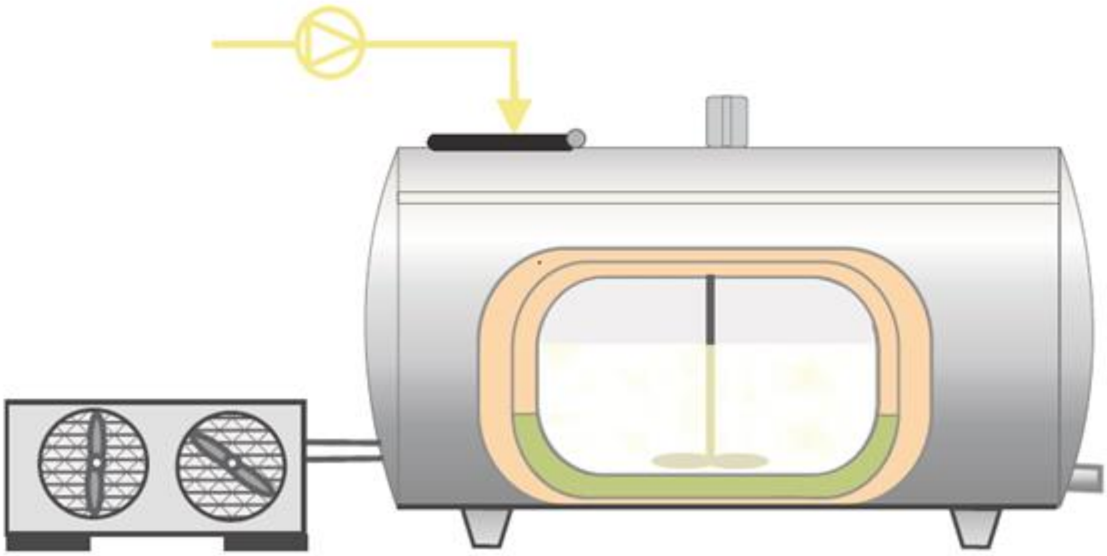

Figure 4: direct expansion milking cooling system [20].

2.1.1.7 Ice bank cooling

Ice bank milk cooling use an ice bank and chilled water reservoir located at the base of the tank assembly. This reservoir houses the evaporator section of the refrigeration system and, over a period of many hours, allows the build-up of ice on the copper pipes of the evaporator[3]. The biggest advantage of an ice bank system is that it allows the cooling capacity to be stored in an isolated reservoir with a heat carrier and cold buffer or ice buffer[3]. In areas where there is not sufficient energy, an ice bank system provides an efficient cooling solution. A device for controlling the thickness of the ice may be fitted to avoid over production of ice and the resulting damage to equipment and to control energy use and operating costs. To enable faster cooling of the warm milk, ice water is pumped to spray heads located on the outer side and the top of the inner tank. Ice bank tanks can cool up to 50 percent faster than DX tanks[20]. Solar panels can be used as a source of electricity for ice generation, but powering an ice bank cooling system using only a diesel generator would be costly[3]. Cooling with chilled water in an ice builder normally requires the use of plate heat exchangers or tubular heat exchangers as represents in figure 5 and 6.There are two types of chilled water equipment. The first is the ice builder, which accumulates ice between milk using a small condensing unit that runs up to 18 hours a day. The second is the package chiller, which has a large condensing unit that runs only during milking.

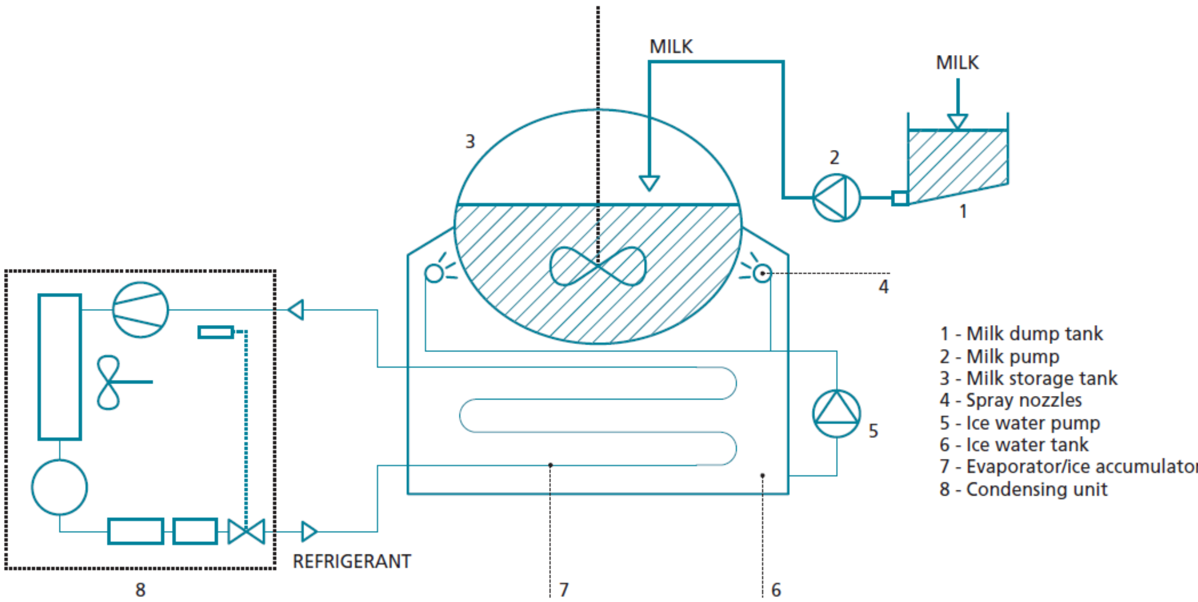

Figure 5: Integrated ice bank milk cooling[3].

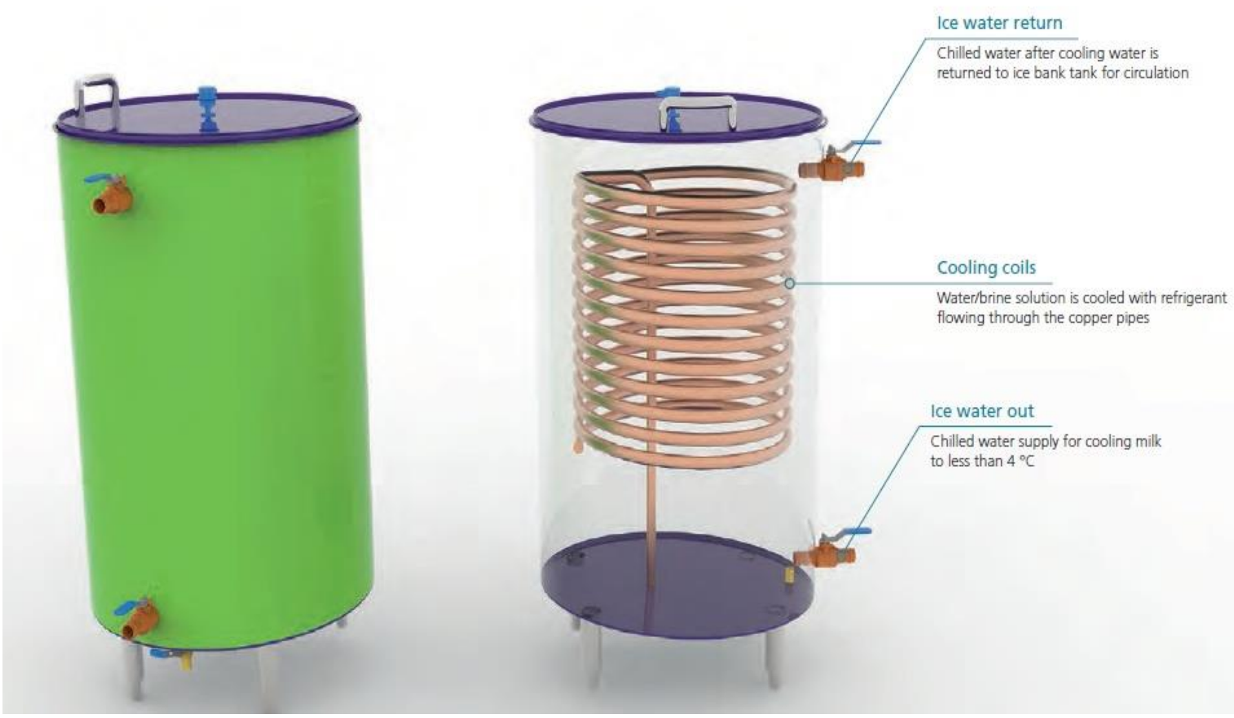

Figure 6: Ice making bank tank[3].

Refrigeration is the thermodynamic process to produce cooling effect below the atmospheric temperature. Its working is governed by reverse of the second law of thermodynamics, that is, when work input is given to system heat can be transferred from a low temperature body to a high temperature body.

2.2 Classification of refrigeration systems

Refrigeration systems fall into two main categories: those that require mechanical energy or its equivalent in order to operate, mechanical refrigeration systems, and those that consume essentially thermal energy, thermal refrigeration systems[22] and also Refrigeration based on solar energy is again classified as Photovoltaic (PV) driven Vapour compression refrigeration system and thermal energy driven vapour absorption Refrigeration system.

8. **Mechanical refrigeration systems**

2.2.1.1 Vapour compression refrigeration system

The vapour compression refrigeration cycle is still the most widely used system concerning refrigeration. In 1805, the American inventor Oliver Evans described a closed vapor-compression refrigeration cycle for the production of ice by ether under vacuum[23]. Heat would be removed from the environment by recycling vaporized refrigerant, where it would move through a compressor and condenser and would eventually revert to a liquid form in order to repeat the refrigeration process over again. This cycle constitution includes a compressor, condenser, an expansion device and an evaporator. All the devices are connected between them with a working fluid that circulates inside the system. The compressor pumps gas into the condenser. As long as the pressure remains below the pressure belonging to the condensing temperature, only the pressure will rise, at that time a heat transfer will start from the gas to the surrounding area. To make this process continuous, the liquid in the condenser must be fed back into the evaporator. Since the pressure in the condenser is always higher than in the evaporator, this can be easily done by establishing a pipe connection from condenser to evaporator. If expansion valve is mounted in this pipe, the amount of refrigerant can be adjusted. When the compressor starts, the gas above the liquid will be sucked away. Due to this, the pressure will decrease. The liquid starts to boil as soon as the pressure sinks below the pressure of the present temperature. The refrigerant will evaporate and take the heat out of the remaining cooling medium.

2.2.1.2 Gas refrigeration system

In these systems, the working fluid remains in the gaseous phase. The compressed gas heats up and is cooled under a lower pressure (lower than the pressure that exists in the system). Next, the gas is expanded, cooling down the system. The development of these systems has been hampered by their reduced efficiency, compared with the vapour compression systems. Their reduced efficiency is applied to refrigeration, freezing and air-conditioning fields. However, gas cycle systems are used in most cryogenic cycles in order to liquefy gases and produce lower temperatures[22].

9. Thermal refrigeration systems

The mechanical compressor used in the vapour compression cycle is replaced by a thermal compressor in the absorption system. The thermal compressor consists of the absorber, the generator, the solution pump, and the expansion valve. Thermal refrigeration system required small amount of heat compared to vapour compression refrigeration system and uses any type of heat[24]. There are three thermal driven refrigerationsystems:

2.2.1.3 Adsorption cooling system

In adsorption cooling system, instead of a liquid solution, solid Adsorption materials are applied[25]. Market available systems use water as a refrigerant and silica gel as absorbent. The machines consist of two absorbent compartments, one evaporator, and one condenser. The simple mechanical construction of adsorption chillers and their expected robustness is an advantage. There is no danger of crystallization and thus no limitations in the heat rejection temperatures. An internal solution pump does not exist and hence only a minimum of electricity is consumed. A disadvantage is that they are much heavier. A large potential for improvements of the heat exchangers in the adsorption compartments is expected[25].

2.2.1.4 Desiccant cooling for air based cooling systems

Desiccant cooling systems are open cycle systems, using water as a refrigerant in direct contact with air. The thermally driven cooling cycle is a combination of evaporative cooling with air dehumidification by a desiccant, i. e. a hygroscopic material. For this purpose, liquid or solid materials can be employed. Only water is possible as a refrigerant since a direct contact to the atmosphere exists[25]. The common technology applied today uses rotating desiccant wheels, equipped with silica gel or lithium chloride as absorption material. Solar assisted desiccant cooling uses solar thermal energy to dry out or regenerate the desiccant. Major benefit: increased comfort (since temperature and humidity are controlled independently), lower operating costs, heat recovery options, improved indoor air quality and reduced building maintenance as a result of high humidity levels. Desiccant systems are used to produce conditioned fresh air directly. Therefore, they can be used only if the air-conditioned ventilation system includes some equipment to remove the surplus internal loads by supplying conditioned ventilation air to the building[25].

2.2.1.5 Absorption cooling

Absorption refrigeration systems are thermodynamic cycles that mainly exchange heat with the surroundings while heat and mass transfer process occur inside the system[12].In these systems, instead of using a mechanical compressor to circulate the refrigerant, a pump is used to make the liquid absorbent circulate[22]. Mechanical work is reduced however, heat is still required.The absorption refrigeration cycle works due to absorption and desorption of the refrigerant fluid contained in a liquid solution. The absorption refrigeration cycle performs the heat exchanges that allow the conditioning of the environment. During the absorption process there is a heat transferred between the surrounding environment and the cycle[22] and the absorption refrigeration system may be driven directly by the heat transfer fluid in the refrigerant tank, or indirectly by another heat transfer fluid using a heat exchanger between the tank and the chiller [12].Absorption refrigeration has been most frequently adopted for solar refrigeration. There is no cycling loss during on-off operation while compression refrigeration produces a lot of waste heat. Absorption system need a low running cost on cheap fuel such as waste heat or biomass while compression refrigeration need high cost due to electricity as energy input. Compared to vapor compression refrigeration absorption refrigeration system ,it have low COP, less heat loss, very low maintenance cost due to no reciprocating parts , less noise, less moving parts[26].

10. Type of absorption refrigeration system

There are three types of absorption systems- Intermittent absorption system, continuous absorption system and double intermittent absorption systems. The difference is in the operating period of cycle [19].

2.2.1.6 Continuous operation cycle

Continuous operating cycle, are more widely used because they can provide lasting cooling and are therefore more advantageous over the intermittent absorption systems [8] and heat is added in the generator for 24 hours by an external heat source and evaporator maintains it temperature for 24 hours . The continuous operation systems belong to a specific classification of absorption refrigeration systems, in which both the generation and absorption processes take place simultaneously[9]. The Generator is powered by solar collectors in the case of a solar Powered absorption system. To ensure continuous operation and Reliability of the system, a hot-water storage tank is used. The pressurization process in such systems is carried out by the solution pump. Certain heat recovery components are added to the continuous operation absorption system to increase its COP. These heat recovery components are a solution heat exchanger and a refrigerant pre-cooler.

2.2.1.7 Intermittent absorption systems

Intermittent absorption systems are able to use waste heat and solar energy[20][19]. The intermittent operation systems comprise a specific class of absorption refrigeration systems in which the generation and absorption processes do not take place simultaneously but rather follow each other in an intermittent manner[9] . During the day, the mixture in the generator– absorber is heated by solar radiation, and subsequently, the ammonia is partially evaporated from the solution. The ammonia vapor is condensed in the condenser and stored in the refrigerant tank. During the night, the liquid ammonia passes through the expansion valve. In intermittent system, heat is added for a certain period. The pressurization process in the intermittent operation systems is carried out by isochoric heating of the NH_3/H_2O solution in the generator[20]. In this way, electrical energy is not required at all in the operation of intermittent absorption systems. Ambient cooling is an intermediate phase which takes place to reduce high pressure gas/vapor into a refrigerant working liquid[19].The systems are environmental friend, as they use eco-friendly refrigerants and run on clean and renewable solar energy. The intermittent solar absorption cooling system has two configurations: the first is a single stage and the second is a two-stage configuration. The single-effect system is the simplest type of these systems. Design of a single-effect absorption cooling system depends on the working fluid types. The system shows better performance with a non-volatile working pair, such as H_2O/LiBr[9]. For obtaining a higher COP, multi-effect systems such as double-effect and triple-effect absorption chillers are used, which are run by steam produced from concentrating solar collectors [47].

2.2.1.8 Double intermittent absorption systems

Double intermittent absorption systems are the refinement of the intermittent systems that work either in cascade, at a higher pressure and producing refrigeration more than single intermittent absorption process[19]. Double-effect absorption systems are commercially available. In this cycle, vapour produced in the high temperature generator flows to the high temperature condenser where changes phase by rejecting heat to the low temperature generator, driving vapour out of the solution the detail expression is clearly expressed in figure 7. With the same heat source, vapour is produced in two generators augmenting the flow of refrigerant per unit of energy introduced in the cycle, which results in an increase of the COP.

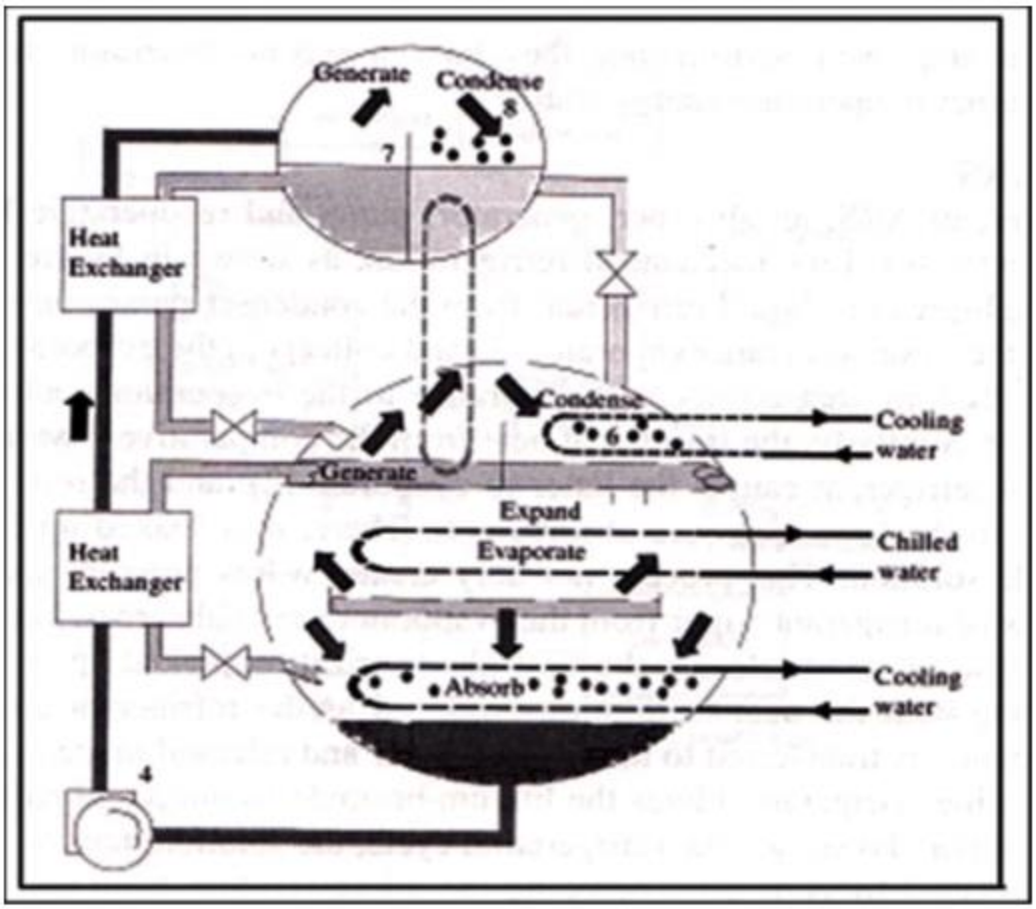

Figure 7:Schematic of a double-effect absorption system [26].

A number of barriers have prevented more widespread use of solar refrigeration systems: First, solar refrigeration systems necessarily are more complicated, costly, and bulky than conventional vapor compression systems because of the necessity to locally generate the power needed to operate the refrigeration cycle. Second, the ability of a solar refrigeration system to function is driven by the availability of solar radiation. Because this energy resource is variable, some form of redundancy or energy storage (electrical or thermal) is required for most applications, which further adds to the system size and cost[15]. According to Pilatowski.et al,proposed a monomethyl amine/H_2O absorption refrigeration system driven by evacuated tube collectors coupled with a conventional auxiliary heating system for milk cooling in the rural regions of Mexico. The results indicated that it is possible to obtain evaporation temperatures from -5 to 10 ^{O}C at a low generation temperature from 60 to 80 ^{O}C, condensation temperatures of 25 ^{O}C, and COP values from 0.15 to 0.7 [9].

2.3 Working Principle ofsolar absorption refrigeration system.

Absorption refrigeration uses physical or chemical attraction between a pair of substances to produce the refrigeration effect[27]. The absorption refrigeration technology consists of a generator, a pump and an absorber that are collectively capable of compressing the refrigerant vapour. The evaporator draws the vapour refrigerant by absorption into the absorber. The extra thermal energy separates the refrigerant vapour from the rich solution[9]. The refrigerant is condensed by rejecting the heat in a condenser, where high pressure vapour refrigerant rejects heat changing to liquid phase and then the cooled liquid refrigerant is expanded by the evaporator, thelow-pressure liquid refrigerant absorbs heat producing the cooling effect, and changing to vapour phase againand the cycle is completed. The refrigerant side of the absorption system essentially works under the same principle as the vapour compression system. However, the mechanical compressor used in the vapour compression cycle is replaced by a thermal compressor in the absorption system. The thermal compressor consists of the absorber, the generator, the solution pump, and the expansion valve. Where the vapour refrigerant is absorbed into another fluid, a pump that pressurizes the mixture and the generator can separates the refrigerant from the absorbent. In solution pump, the energy required for pressurizing a liquid is much less, than the energy required for pressurizing a gas[28].Any absorption refrigeration system can be operated only when the solution in the absorber is richer in refrigerant than that in the generator. The most important are the generator or desorbed where separation of refrigerant from absorbent occurs at the highest temperature found in the cycle and the absorber where at a temperature lower than that of the generator, refrigerant is re-absorbed into the absorbent medium. Because absorption is an exothermic reaction, the absorber requires cooling[17].The refrigerant - absorbent mixture is then transferred to the generator via pump. In this system, heat is transferred to the mixture in order to drive off the refrigerant from the mixture, leaving behind a weak mixture (low percentage of refrigerant) which is transferred back to the absorber via throttle valve. As the refrigerant-absorbent mixture is pumped into the generator, some of heat gained from the weak the mixture leaving the generator through a solution heat exchanger. This greatly enhances the coefficient of performance (COP)of the system[27].

2.4 Heat Exchange

A heat exchanger is a device that assists in the transfer of heat from one fluid to the other. In many cases, a solid wall or intermediate metallic wall may separate the fluids and prevent them from mixing and without moving parts[29]. Heat exchangers have a wide range of industry applications; they are widely used in space heating, refrigeration, power plants, petrochemical plants, petroleum refineries and sewage treatment. There are many types of heat exchanger designs for various applications[30]include double pipe, shell and tube, plate and shell, plate fin, and phase change heat exchangers. The flow in a heat exchanger arranged as parallel flow, counter flow, and cross flow. The counter-flow heat exchanger, both fluids enter the heat exchanger from opposite sides. In the parallel-flow heat exchanger, the fluids come in from the same end and move parallel to each other as they flow to the other side. The cross-flow heat exchanger moves the fluids in a perpendicular ways. Among the above flow arrangements counter–flow is the most efficient heat exchanger way because it transfers the greatest amount of heat. Now a day heat exchangers also designed for emerging thermal engineering fields, such as miniaturized heat exchanger for cooling electronics components and systems, miniaturized heterogeneously catalyzed gas-phase reactions, thermoelectric generators, etc. In the most efficient heat exchangers, the surface area of the wall between the fluids maximize and simultaneously minimizing the fluid flow resistance. Fins or corrugations are sometimes used with the wall in order to increase the surface area and to induce turbulence. Additionally, heat exchangers may be classified according to the type of fluid used (liquid-to-liquid, liquid-to-gas, gas-to-liquid, gas-to-gas), according to phase changes (vaporizers, condensers. There are two common designs of heat exchanger. These are shell and tube type, and plate type heat exchanger. The most commonly used type of heat exchanger is the shell and tube type. This heat exchanger can be design with bare tube or finned tubes. One of the fluids runs through the tubes while the other fluid runs over them, causing it to be heated or cooled. In the plate heat exchanger, the fluid flows through the baffles. This causes the fluids to be separated by plates with a large surface area.

11. Shell & Tube Heat Exchanger

Shell and tube heat exchanger is one of the heat exchanger groups, which one fluid runs through the tubes, and another fluid flows through the shell to exchange heat between the two fluids[31]. STHE is the most common type of heat exchanger. It originated from the jacketed-coil distiller, and is used in heavy industries (steam condensers, boilers), and residential hot water and heating systems (fire-tube water heaters). This exchanger, shown in Fig. 8, is generally built of a bundle of round tubes mounted in a cylindrical shell with the tube axis parallel to that of the shell. One fluid flows inside the tubes, the other flows across and along the tubes. The major components of this exchanger are tubes (or tube bundle), shell, frontend head, rear-end head, baffles, and tube sheets.). In this types of heat exchanger, where one flow goes along a bunch of tubes and the other within an outer shell, parallel to the tubes or in cross-flow. The design of STHE includes thermal design and mechanical design. The thermal design of STHE involves evaluation of required effective surface area (*i.e.* number of tubes) and finding out log mean temperature difference. Whereas, the mechanical design includes the design of main shell under internal & external pressure, tube design, baffles design gasket, etc[31]. Heat is transfer from one fluid to the other through the tube walls, either from tube side to shell side or vice versa.

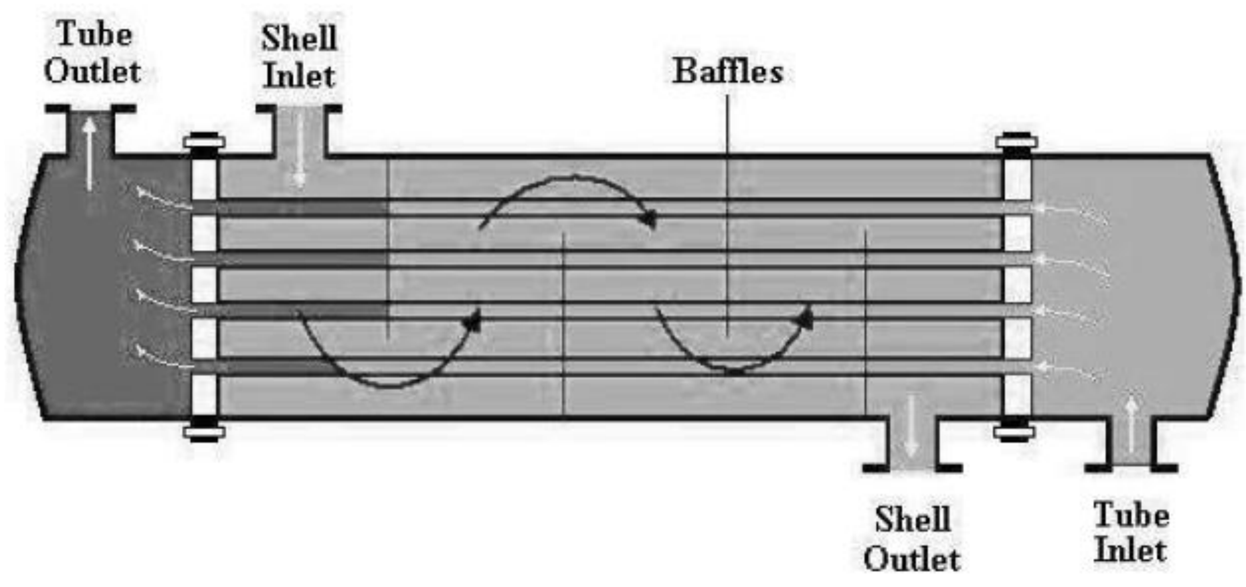

Figure 8:Shell tube type heat exchanger[29]

12. Plate type Heat Exchanger

A plate heat exchanger is a compact heat exchanger where thin corrugated plates are stacked in contact with each other, and the two fluids made to flow separately along adjacent channels in the corrugation[29].Plate and frame heat exchanger for general refinery service can be referring as gasket plate heat exchangers. The plate heat exchanger consists of a frame, which consists of a head follower, column, carrying bar, guiding bar, and a number of clamping bolts in figure 9. In between head and follower, a varying number of pressed plates are clamped together. Each plate is supplied with a gasket so that the plates form a closed system of parallel flow channels through which the mediums flow alternatively at every time interval. It is mostly used for heating and cooling applications and often for closed-loop applications.

Figure 9:plate heat exchanger (single plate, exploded assembly, assembled)[29].

2.5 Working fluid for solar absorption refrigeration system

The working fluid in an absorption refrigeration system is a solution consisting of refrigerant and absorbent[28].Refrigerant is separated from absorbent because of their difference in boiling point. Refrigerant produces the cooling effect when it vaporizes, in the evaporator taking heat from the water that is being chilled [12]. Therefore, Refrigerant should have high heat of vaporization and high concentration within the absorbent in order to maintain low circulation rate between the generator and the absorber per unit of cooling capacity. Performance of absorption refrigeration systems is critically dependent on the chemical and thermodynamic properties of the working fluid[28]. H_2O–Libr and NH_3–H_2O are the most widely used absorption working fluids, with the former being more popular due to its higher performance. In 1860 French engineer Ferdinand Care was the first person documented as inventing an aqua-ammonia absorption system that used water as absorbent together with NH_3, which in turn has a strong absorption potential for H_2O. These innovative systems were usually harnessed to produce ice in small amounts[17]. The NH_3–H_2O

mixture is still widely used due to the absence of crystallization and the low freezing point of the refrigerant [8]. Heat is supplied to a device called the generator. The generator temperature can be lower than 100°C, depending on the refrigerant-absorbent pair, e.g., ammonia/ water or water/lithium bromide. Low generator temperatures normally lead to significantly larger heat exchanger surfaces. But it is shown that an increase in the generator temperature increases the COP of the systems up to an optimum generator temperature [32]. To reach freezing temperatures below ^{0} C, the only available thermally driven chillers use NH_3-H_2O as working pair [33]. Although LiBr systems commonly present better COP than aqua-ammonia systems, they are limited to an evaporator temperature above that of the freezing point of water for the specific pressure chosen, or else the unit will cease to function[8]. Due to difficulties in heat transfer, higher temperatures are also needed in a LiBr desorbed to achieve optimal separation as well as greater absorber cooling requirements due to crystallization [17][8] when compared to liquid-gas systems, Another several critical limitation exist for H_2O-libr: The freezing point of water is above 0^{O}C, and therefore, an H_2O–LiBr uses water as a refrigerant cannot operate at an evaporation temperature below 0^{o}c, making it unusable for subfreezing refrigeration or heating/domestic hot water supplementation in cold regions and also High vacuum conditions should be maintained in the system for efficient operation of the H_2O–LiBr system. Absorption machines are commercially available today in two basic configurations. For applications above 32˚F (primarily air conditioning), the cycle uses lithium bromide as the absorbent and water as the refrigerant. There are various types of refrigerant and absorber pair is being used at present time. Among them NH_3/H_2O is widely used where low temperature is required and LiBr-H_2O systems are used where moderate temperature are required like air conditioning. Fluids mixtures are NH_3/H_2O (NH_3 is the refrigerant and H_2O is the absorbent) and LiBr-H_2O (H_2O is the refrigerant and LiBr is the absorbent). Both refrigerants have a high latent heat, but neither have ideal vapor pressure characteristic. Ammonia vapor pressures are much higher than those of water are. The freezing point of water restricts the use of H_2O /LiBr systems to temperatures above 0 °c, while the very low freezing point of ammonia, -77 °c, allows operating NH_3/H_2O systems at below zero temperatures. Ammonia also exhibit a lower pumping cost for liquid recirculation system, more tolerance of water contaminant and have no environmental impact such as global warming potential and greenhouse effect. NH_3/H2O systems have the advantage that the heat can be rejected directly into ambient air [12]. Since both NH_3 and water are volatility, the cycle requires a rectifier to strip away water that normally evaporates with

NH_3. Without a rectifier, the water would accumulate in the evaporator and offset the system performance [28]. Ammonia also exhibits some disadvantages such as it is poisonous and can burn with air. It is lighter than air and the leak of ammonia can easily detect smell at a concentration far below a dangerous level to get high performance solar collectors are needed to supply a sufficient solar energy. There are other disadvantages such as its high pressure, toxicity, and corrosive action to copper and copper alloy. However, NH3 –H_20 is environmental friendly and low cost. The advantage of ammonia overcomes the disadvantage. Therefore NH_3-H_2O can be selected as the refrigerant-absorbent in application of temperature required for low temperature below zero.

2.6 Different energy source to be used for cooling in dairying process

13. Wind power

Electrical power generators driven by wind turbines can be used to capture wind energy. In this system, the inherent kinetic energy in moving wind spins two, three or four propeller blades, which are connected to a rotor hub[34]. A wind turbinecaptures wind energy and converts it into electrical energy, as the wind passes over the blades it causes them to rotatewhich turns a generator to produce electricity which is fed into an inverter which converts the power to mainspower.Wind turbines generate direct current (DC) electricity and off-grid systems require battery storage and aninverter to convert the DC electricity generated to AC (alternating current) which is mains electricity[35].To cover periodsof low wind speed or no wind, a backup genset isalso required for off-grid installations. The initial capital costs of establishing a standalone wind turbine system to provide electrical power are high, but they are repaid in savings on energy costs over time. Batteries will need to be replaced every five to seven years. Use of conventional power supplies from a national grid, a backup genset and a DX milk cooling tank of 1000 liters would cost about US$10,000. The study concludes that such systems “will not be economically viable, even with generous feed-in tariff[35].

14. Geothermal power

Geothermal power is a renewable energy resource that uses heat from deep underground[35]. The element responsible for this heat energy is the radioactive substance in the earth’s core. Geothermal energy is accessible only in specific locations, but where it exists it provides a virtually infinite energy source at almost no cost. Exploitation of geothermal energy is limited mainlybecause of the up-front costs of exploration, with the risk that initial investments and geothermal energy resource is location limitation.

15. SOLAR COLLECTOR

A solar collector converts solar radiant energy into useful thermal energy, which is transferred to a fluid (water, air, glycol, oil, etc.) flowing through the collector[7][10]. Solar thermal energy collectors are special kind of heat exchangers that convert solar radiation into thermal energy through a transport medium and/or moving fluid. Solar collectors can be categorized in various ways based on working principle and other factors. A common categorization is as follows: non-concentratingand concentrating.Another categorization based on heat transfer fluid they uses, solar collectors into liquid heating and air heating types. Stationary solar collectors without tracking device are common for low- and medium-temperature (less than 100^{o}c) applications, whereas concentrating collectors with tracking systems are more often used for application sat medium to high temperature, i.e., (250–2500^{o}c). Stationary solar collectors are generally use for water and air heating applications, while most concentrating collectors with tracking are used for high temperature heating and solar thermal power generation[10]. Solar collector can be categorized into three based their application used in thermal technology.

2.6.1.1 Flat plate collector

The flat plate collector (FPC) is the heart of any solar energy collection system designed for operation in the low temperature range (less than 60 ^{O}C) or in the medium temperature range (less than 100 ^{O}C) [9][7] .The thermal efficiency of this collector type normally ranges from 40% to 60% for the low- to medium-temperature applications. The flat plate solar collector is most common solar thermal device and has been used for many decades. It is used to absorbed solar energy, convert it into heat and then to transfer that heat to stream of liquid or gases. Flat-plate solar collectors, being mechanically simpler than concentrating collectors, are mainly used for domestic and industrial purposes [7]. A flat plate collector consists of an absorber, a transparent cover, a frame and, commonly, insulation. Usually a low-iron tempered glass is used as a transparent cover. The absorber surface is the heart of the flat plate solar collector and is typically made of copper or aluminum and painted black or selectively coated [9]. When used, insulation is located at the back and sides of the collector to limit heat losses. The most common insulations material for flat plate collectors are polyurethane and mineral wool. A frame of tubes is attached to the absorber surface to allow for the flow of a heat transfer fluid. The insulation in the solar collector reduces the heat losses. The optical efficiency of a flat plate solar collector is product of the transmissivity (τ) of the glass cover and absorptive (α) of the absorber.Flat panel collectors typically have low efficiencies due to their limited water temperature outlet [36].

Figure 10: overview of flat plate collector[25].

2.6.1.2 Evacuated tube collector

Evacuated tube collectors are a non-concentrating solar collector, permanently fixed in position. Evacuated tubes offer the advantage that they work efficiently with high absorber temperatures and with low radiation[37]. Evacuated tube consists of two glass tubes made of borosilicate the inner glass tube and outer tubes are separated by vacuum space. The vacuum is a perfect insulator doesn't allow the short wave radiation to escape through vacuum characteristics of trapping radiation is better than other methods. The ETC provide the combined effects of a highly selective surface coating and vacuum insulation of the absorber element so they can have high heat extraction efficiency compared with flat plate collectors in the temperature range above 80 ^{o}C[25]. At present, the glass-evacuated tube has become the key component in solar thermal utilization and they are proved very useful especially in residential applications for higher temperatures.ETC use liquid–vapor phase change materials to transfer heat at high efficiency. These collectors feature a heat pipe (a highly efficient thermal conductor) placed inside a vacuum sealed tube. The pipe, which is a sealed copper pipe, attached to a black copper fin that fills the tube (absorber plate) is a metal tip attached to the sealed pipe (condenser). The heat pipe contains small amount of fluid that undergoes an evaporating-condensing cycle. In this cycle, solar heat evaporates the liquid, and the vapor travels to the heat sink region where it condenses and releases its latent heat. The condensed fluid return back to the solar collector and the process is repeated[7].Evacuated tube collectors consist of single tubes that are connected to a header pipe. To reduce heat losses of the water-bearing pipes to the ambient air each single tube is evacuated. The collector must always be mounted with a certain tilt angle in order to allow the condensed internal fluid of the heat pipe to return to the hot absorber[25]

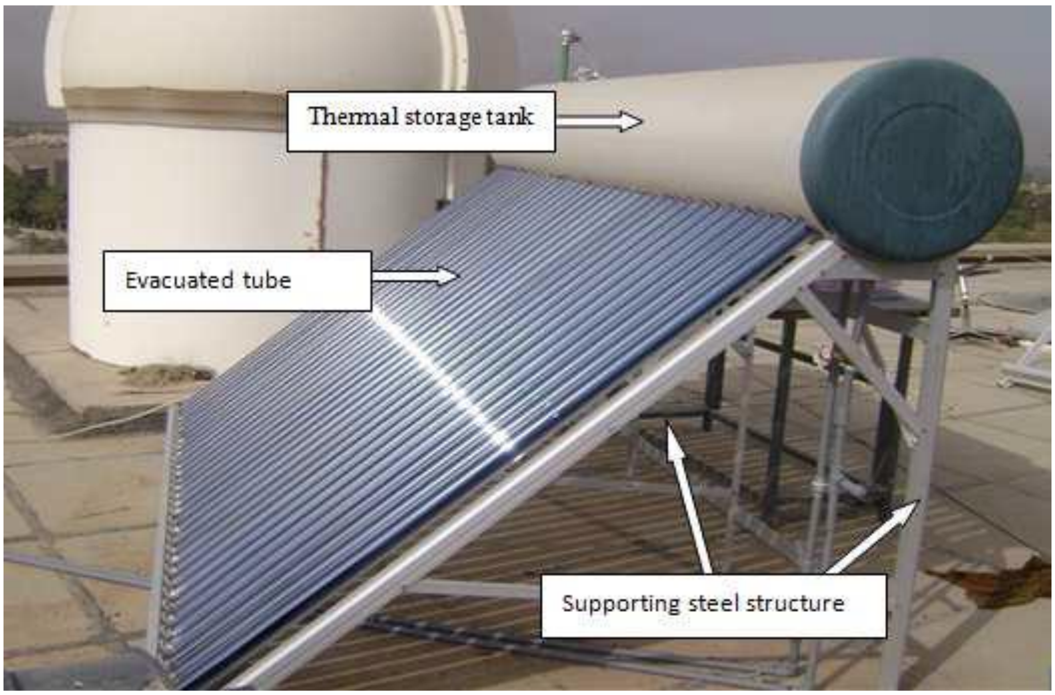

Figure 11:Evacuated –Tube Thermal Solar Collector Array[38].

Hot water in the tube moves by natural convection upward to be replaced by colder water. The hot water produced will be accumulated in the storage tank [38].

2.6.1.3 Concentrating collectors

Concentrating collectors provide energy at temperatures higher than those of flat plate Collectors do. They redirect solar radiation passing through an aperture into an absorber and usually require tracking of the sun[7]. In concentrating collectors, solar energy is optically concentrated before being transferred into heat. Concentrating collectors can also be classified into non-imaging and imaging depending on whether the image of the sun is focused at the receiver or not. The collectors falling in this category are: Parabolic trough collector, Linear Fresnel reflector, Central receiver and parabolic dish. Concentrating solar systems can improve the energy efficiency of solar cooling systems when high driving temperatures above 120 ^{O}C are required. They offer advantages compared to the solar collectors traditionally used in solar air-conditioning and refrigeration facilities. Those advantages of concentrating technologies are lower thermal losses and thus higher temperature of the fluid, high ground usage, and adequate recycling of component materials. The disadvantages of concentrating technologies are increased investment and installation due to the tracking system, higher effort for hydraulics and balance of plan [33]. The energy collected is enough to run a small absorption system with approximately a 3-ton cooling capacity where the hot water collection range is limited to approximately 80 ^{o}c[18]. When the desired hot source temperature requirement is greater than 100 ^{o}c, concentric collectors are recommended.Actual solar collector efficiencies will depend on the ambient air temperature and solar radiation [9] .

In this, paper focused on modeling of Evacuated tube due to it is efficient in medium temperature than other type of collector used in modeling of solar absorption refrigeration system for dairying process. The solar energy technology or synonym as photovoltaic and solar thermal technology has many advantages and disadvantages comparing to others energy. The potential advantages such as it works on noiseless environment; do not produce any unwanted waste such as radioactive materials, high performance and reliable system, clean technology, does not produce any toxic waste or radioactive material, highly credible system with life span expectation is between 20 and 30 years and low maintenance system. The disadvantages are; non-uniform cooling – need innovative absorber design, payback – less efficiency, longer payback period, production and installation cost–expensive and high cost, not suitable for integration with present roof system and need larger space for separate systems (hot water and electricity production)[39]. Refrigeration plays a very significant role in the overall energy requirements of a modern milk plant, often constituting above 40 percent of the total electric power consumption [40]. In modern systems, refrigeration in milk collecting and processing requires electric power and usually water. Absorption refrigeration has so far not gained any significance in the dairy industry. Solar based Refrigeration plant used for the chilling of milk in bulk milk cooler. It also used to run a vapour absorption system for refrigeration plant and maintain cooling system. Absorption systems have been extensively paid attention in recent years due to the potential for CFC and HCFCreplacements in refrigeration, heating and cooling applications[41]. Solar refrigeration systems are environmentally friendly and no harmful refrigerant is used. They are cheaper than vapour compression refrigeration system[42]. The International Institute of Refrigeration in Paris (IIF/ IIR) has 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[43]. Therefore high performance solar collectors are needed to supply a sufficient solar energy input[44].

CHAPTER 3

3 METHODOLOGY AND SELECTION OF APPROPRIATE SYSTEM

3.1 Methodology

To precede the modeling of small-scale solar absorption refrigeration used for dairying process several methods are used:

1. Data collection on existing small-scale dairying system companies and metrological data to help the real design of solar absorption refrigeration system for dairying process.
2. Analysis the amount of energy required to model system.
3. Identify the system capacity to be analyzed i.e. Sizing and selecting the appropriate components to used such as collector area and identify collector type, storage size, condenser, evaporator, the required temperature used for small capacity of absorption refrigeration system.
4. Mathematical model system components such as Absorber, Generator, Condenser, Evaporator and solar collector. Different modeling approaches with different levels of complexity have been proposed. Modeling is necessary for optimization, process control, fault detection, data reconciliation, etc. Not all models are useful for all tasks. Some are too complex to be used for control where rapid convergence is needed, and others are too simple to derive accurate conclusions about thermodynamic issues. We have classified the modeling approaches in three categories: thermodynamic, data driven, and semi empirical. Thermodynamic models are based on the mass and energy conservation principles, vapour-liquid equilibrium, as well as other physical constraints. Data driven models are based on data only and their equations have no physical meaning. Semi empirical models are a middle point between the former two. In semi empirical models some unknown parameters are estimated from experimental or manufacturer data; or some parts of a thermodynamic model are replaced by empirical correlations. The operating pressures at which the system is working needs to be determined to carry on further calculations, using an enthalpy

concentration chart. Once the pressure of the condenser and the pressure of the evaporator are determined the corresponding points can be fixed. The various other points and condition lines for components like absorber, generator, heat exchangers etc. can be subsequently fixed. A control volume is taken across each component i.e. the generator, absorber, evaporator, condenser and heat exchanger to analyze the working conditions of all components of the system. In this thesis focus on thermodynamic models are based on the mass and energy conservation principles

5. Dynamic modeling and simulation using appropriate software such as MATLABcoding. As absorption unit become more popular not only in industry but also in domestic level their simulation become more important. This enable of better understanding of complex thermodynamics behavior such as heat transfer analysis. There are many problems which simply do not have analytical solutions, or those whose exact solution is beyond our current state of knowledge. There are also many problems which are too long (or tedious) to solve by hand. When such problems arise we can exploit numerical analysis to reduce the problem to one involving a finite number of unknowns and use a computer to solve the resulting equations from those MATLAB coding is important software to solve complex mathematical system and to develop MATLAB program for the system.
6. Performance analysis of overall solar absorption refrigeration system.

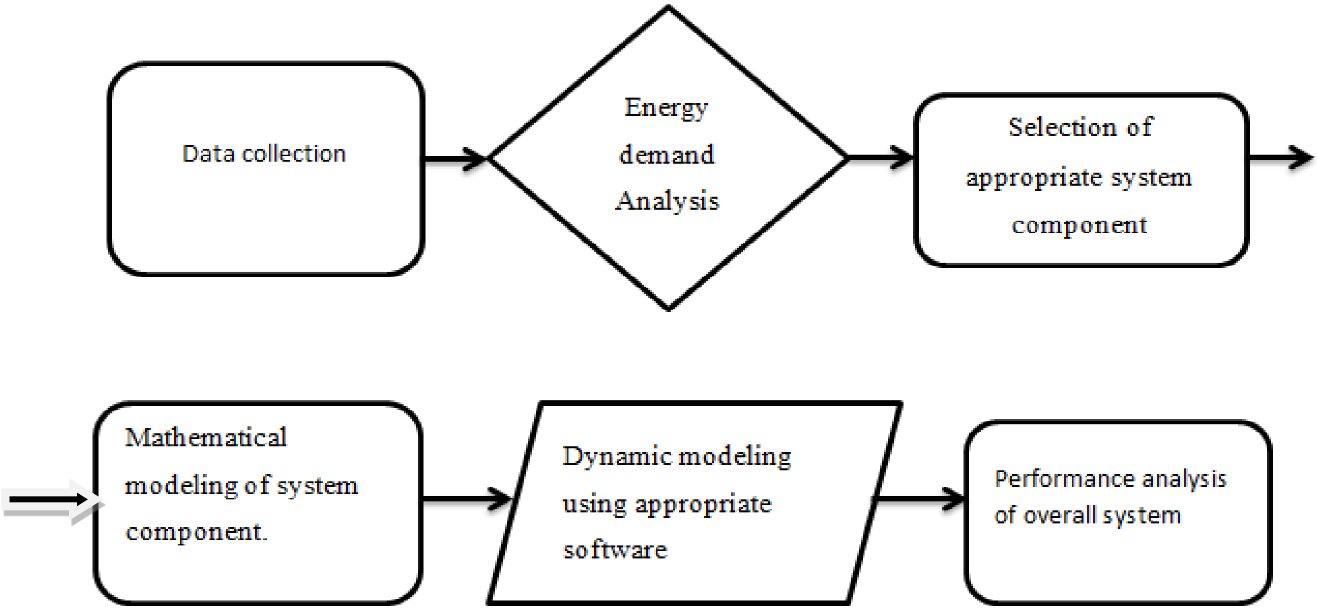

Figure 12: work flow the thesis

3.2 Energy demand and required data analysis

16. Energy demand analysis of chuye milk and milk processing company

Chuye milk and milk processing company is located in Addis Ababa city, Gulelle sub city. This company processes milk, around 500-800 liters of milk per day, depends on their customer's needs. The company is registered as one of small scale dairying process in Ethiopia based on their size. To cool milk up to 4^{o}c is 3-4 hours is required. To pasteurize the 500 volumes of liters of milk up to 80^{o}c it consumes 1:30-2:00 hours, the boiler uses two electrical heaters, they total energy consumption of boiler is around 10 kW.Refrigerated storage tank is used to cool milk up to 4^{o}c; uses vapour compression refrigeration system and initially the Refrigerated storage tank uses 3Hp powerto cool 500 liters. The Refrigerated storage tank uses R44type of refrigerant. This refrigerated storage tank manufactured by A Two electromechanical company, which are a supplier of boiler and refrigerated storage tank of chuye milk and milk processing. Chuye uses 500 liter refrigerated storage tank to proceeds their duty, with height of 1 m and area of 0.5m^2. The lists of processes proceed in chuye milk and milk processing is as follow:

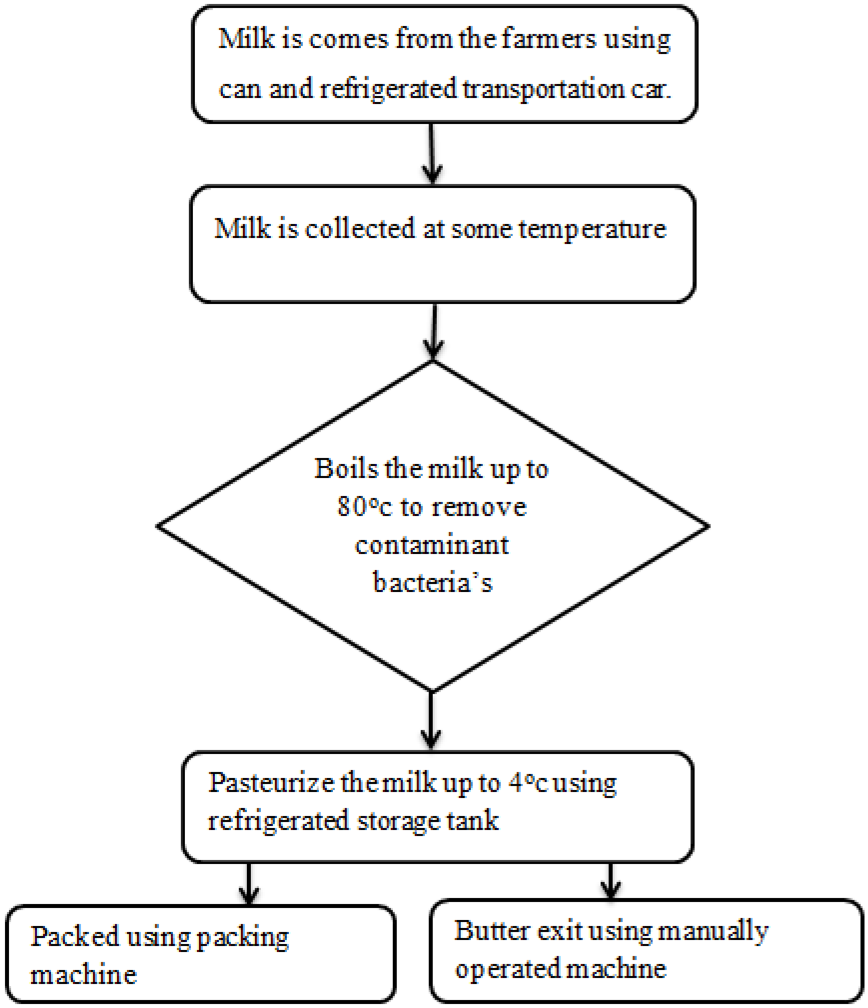

Figure 13: lists of processes proceed in chuye milk and milk processing company [source: chuye milk and milk processing company].

Chuye milk and milk processing uses electricity to run all processes in the factory, in average monthly electricity consumption is around 5575 KWh/KVA and yearly overall energy Consumption is drawn in graph from using MATLAB as below:

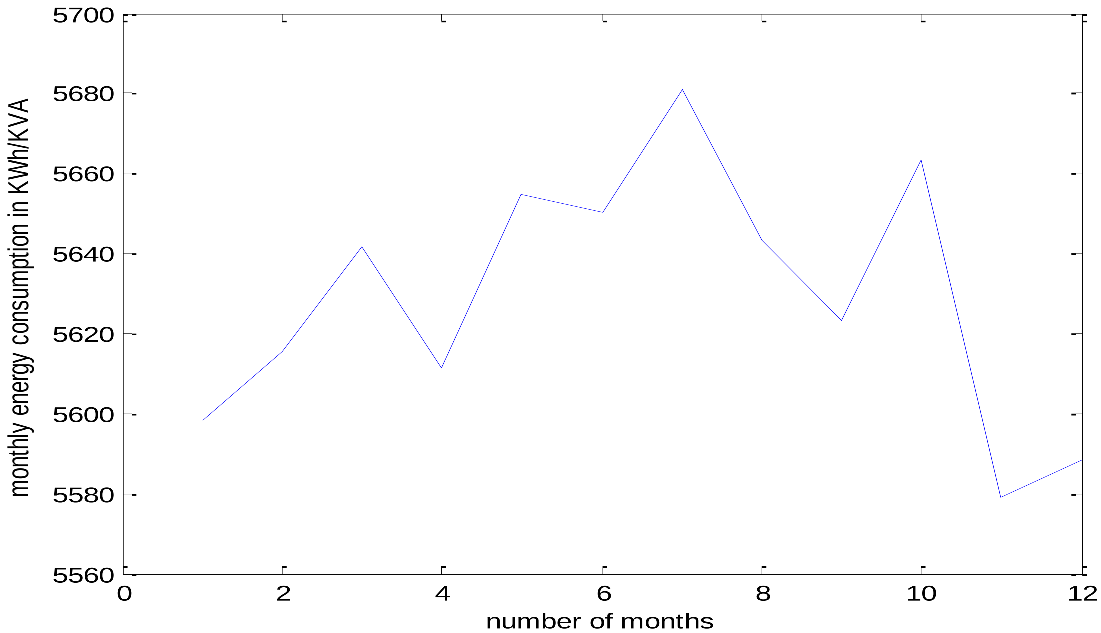

Figure 14: yearly overall energy consumption of chuyemilk and milk processing company [EEPCO energy bill].

3.3 Solar radiation and ambient temperature data analysis

17. Solar radiation analysis

The mathematical relations given below shows, beam and diffuse solar radiation data Based on Ethiopian National Metrological Service Agency record of bright sun shine hours in a day.

$$\frac{H}{H_o} = a + b\frac{s}{T_d} \quad \text{...1}$$

Where a and b are the empirical constant parameters, which express as a function of latitude and sunshine hours

$$a = -0.309 + 0.539 + \cos\phi - 0.0693xZ + 0.290x\frac{s}{T_d}$$

……………………………………….…...2

$$b = 1.449 - 0.553x\cos\phi - 0.694x\frac{s}{T_d} \quad \text{...3}$$

Where s is sunshine hour in a day, Z is elevation of the site above sea level = 433m and T_d isthe time of sun set or rise, its equation is given below

$$T_d = \frac{2}{15}\omega_s = \frac{2}{15}\cos^{-1}(-\tan\phi\tan\delta) \qquad \text{.....4}$$

Where hourly angle is:

$$\omega_s = \cos^{-1}(-\tan\phi\tan\delta) \qquad \text{.....5}$$

Latitude of the site =11.7°

Month	n for i_{th} Day of month	For Average Day of month date	n	δ
January	i	17	17	-20.9
February	31+i	16	47	-13.9
March	59+i	16	75	-2.4
April	90+i	15	105	9.4
May	120+i	15	135	18.8
June	151+i	11	162	23.1
July	181+ i	17	198	21.2
August	212+i	16	228	13.5
September	243+i	15	258	2.2
October	273+i	15	288	-9.6
November	304+i	14	318	-18.9
December	334+i	10	344	-23.0

Table 1: Recommended average day of month and value of n by month[45].

The declination δ can be found from the approximate equation of Cooper (1969)[45],

$$\delta = 23.5\sin\left(360x\frac{284+n}{365}\right) \qquad \text{.....6}$$

Where n = average number of the days in the month.

When we calculate daily solar radiation daily extraterrestrial solar radiation Ho (kW hr /m^2/day) are necessary, the equation of daily extraterrestrial solar radiation is given below [45]:

$$H_O = \frac{24 * G_{sc}}{\pi}\left(1 + 0.033\cos\frac{360n}{365}\right)x\left(\cos\phi\cos\delta\sin w_s + \frac{\pi w_s}{180}\sin\phi\sin\delta\right) \qquad \text{...........7}$$

Where GSC = 1367 w/m^2 is solar constant.

The sunrise and set hour angle is greater than 81.4^o and the clearance index (k_t) are in the range of 0.3 to 0.8. The average daily diffuse radiation on the surface is given by the following relations[45].

$$\frac{H}{H_d} = 1.311 - 3.022K_t + 3.427K_t^{\ 2} - 1.82K_t^{\ 3} \qquad \text{..8}$$

Therefore we can calculate the daily beam radiation from the total daily solar radiation:

$$H = H_b + H_d \qquad \text{...9}$$

Based on above daily solar radiation, estimate hourly solar radiation values from daily numbers as follow[45]:

$$r_t = \frac{I}{H} \qquad \text{..10}$$

Where: r_t is the ratio of hourly to daily total solar radiation and also expressed as follow:

$$r_t = \frac{\pi}{24}(a + b\cos\omega)\frac{\cos\omega - \cos\omega_s}{\sin\omega_s - \left(\frac{\pi\omega_s}{180}\right)\cos\omega_s} \qquad \text{.......................................11}$$

Where: ω is the hour angle in degree and ω_s is the sunset hour angle.

The coefficient of a, b and ω is given as:

$$a = 0.409 + 0.5016x\sin(w_s - 60) \qquad \text{...12}$$

$$b = 0.6609 - 0.4767x\sin(w_s - 60) \qquad \text{...13}$$

$$\omega = (12 - ST)x15^o \qquad \text{...14}$$

The diffuse solar radiation also determined as follow:

$$r_t = \frac{I_d}{H_d} \qquad \text{......15}$$

Where, r_d– is the ratio of hourly to daily diffuse solar radiation and it is expressed as:

$$r_t = \frac{\pi}{24}\frac{\cos\omega - \cos\omega_s}{\sin\omega_s - \left(\frac{\pi\omega_s}{180}\right)\cos\omega_s} \qquad \text{......16}$$

18. Ambient temperature analysis

The hourly ambient temperature is changed from daily ambient temperature as flow: The method is developed by Parton and Logan (1981). It will be referred to Temperature subroutine that a divides the day into two segments and utilizes a truncated sine wave in the daylight and an exponential decrease in temperature at night[46].

A, for day light hours

$$T(H) = (TMAX - TMIN)\sin\left(\frac{\pi m}{y+2a}\right) + TMIN \qquad \text{......17}$$

B, for night _time hours

$$T(H) = TMIN + (TSUNSET - TMIN)EXP\left(-\frac{bn}{z}\right) \qquad \text{......18}$$

Where T (H) is the temperature at any hour of the day or night period determined from m and n, Y is day length (h), z is night length (h), TSUNSET is temperature at sunset (^{o}c), m is number of hours between time at TMIN and sunset (h), n is number of hours from sunset to the time ofTMIN (h), a is lag coefficient for TMAX=1.80(h), b is night –time temperature coefficient =2.2.0(h), and c islag time of TMIN from time of sunrise = 0.880(h).

3.4 Selection of appropriate systemand configuration

It is important to choose the correct type of cooling system in relation to energy consumption. This choice will be affected by the source and cost of the energy supply. There are two type of modern cooling system used in dairying process. Direct expansion and ice bank cooling. Direct expansion tanks do not have a cold buffer, energy must always be available. DX systems can also be up to 50 percent more energy-efficient than ice bank cooling, but require larger compressors. Ice bank cooling system areas where there is not sufficient energy or continuous supply energy is unavailable. Ice bank tanks can cool up to 50 percent faster than DX tanks. In detail these two types of cooling system are discussed in literature review including their advantage and disadvantage. In the rural area where grid electricity is unavailable, but there is different alternative Intermittent energy used for the required application based on the resource of the location. Samara is one of the desert areas, in Ethiopia, reach in solar and slightly wind energy. In aspect of cost, solar thermal energy is more important than wind and PV solar energy.Ice bank cooling is more advantageous than direct expansion in areas where grid electricity is unavailable. Refrigeration is a device to produce cooling effect below the atmospheric temperature. Refrigeration systems categorized into two based on energy consumption: Those that require mechanical energy in order to operate, mechanical refrigeration systems (compression refrigeration system), and those that consume essentially thermal energy, thermal refrigeration systems (absorption refrigeration system). The vapour compression refrigeration cycle is still the most widely used system concerning refrigeration. Vapour compression uses more energy and efficient than thermal refrigeration system. The thermal refrigeration system uses thermal compressor in the absorption system. The thermal compressor consists of the absorber, the generator, the solution pump, and the expansion valve. Thermal refrigeration system required small amount of heat compared to vapour compression refrigeration system and uses any type of heat and free from any environmental emission. In this thesis focus on solar powered thermal refrigeration system due to its more advantageous than compression refrigeration, in terms of cost, energy available in the location and environmental impact. Thermal refrigeration (absorption refrigeration) system classified into three: intermittent absorption system, continuous absorption system anddouble intermittent absorption systems. The difference is in the operating period of cycle. In continuous operating cycle,heat is added in the generator for 24 hours by an external heat source and evaporator maintains it temperature for 24 hours[8].Intermittent absorption systems are able to use waste heat and solar energy[20][19]. The intermittent operation systems comprise a specific class of absorption refrigeration systems in which the generation and absorption processes do not take place simultaneously but rather follow each other in an intermittent manner[9] . During the day, the mixture in the generator– absorber is heated by solar radiation, and subsequently, the ammonia is partially evaporated from the solution. The ammonia vapor is condensed in the condenser and stored in the refrigerant tank. During the night, the liquid ammonia passes through the expansion valve. In intermittent system, heat is added for a certain period. Double intermittent absorption systems are the refinement of the intermittent systems that work either in cascade, at a higher pressure and producing refrigeration more than single intermittent absorption process[19]. In this cycle, vapour produced in the high temperature generator flows to the high temperature condenser where changes phase by rejecting heat to the low temperature generator, driving vapour out of the solution. A single-effect intermittent absorption refrigeration system is the simplest and most commonly used design in solar absorption refrigeration system. Single effect intermittent absorption refrigeration is selected for small scale solar absorption refrigeration system. Heat is provided to the generator where the refrigerant is evaporated out from the mixture. The high pressure vapour of refrigerant is liquefied in the condenser, rejecting heat to a thermal reservoir that is usually cooling water in the case of H_2O/LiBr, and cooling water or air in the case of NH_3/H_2O systems as represented in figure below. A throttling device expands the refrigerant to the low pressure level and at the same time reduces its temperature. In the evaporator the refrigerant removes heat from the chilling fluid and evaporates producing the cooling effect. The low pressure vapour refrigerant flows to the absorber where it is absorbed by the poor solution (low concentration of refrigerant) that comes from the generator.

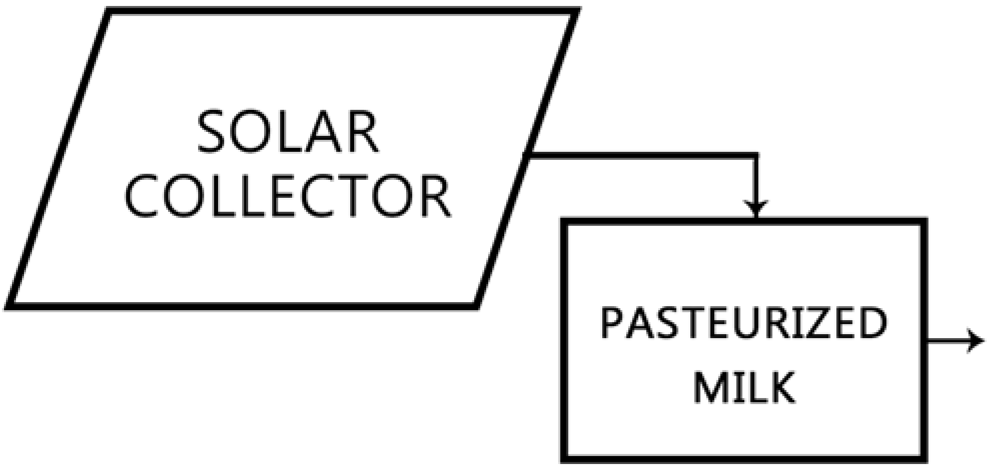

Figure 15: milk pasteurization using solar energy.

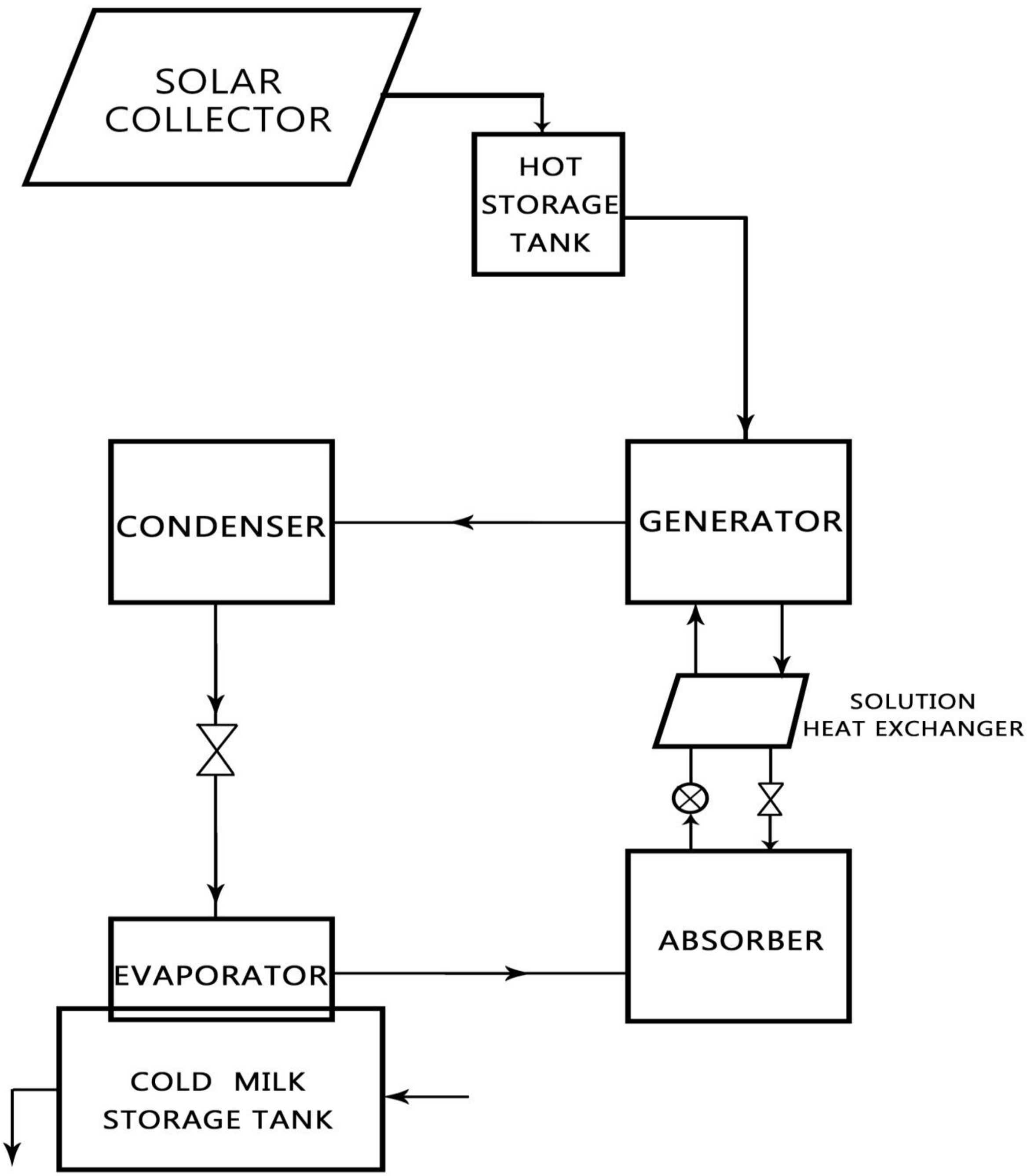

Figure 16: Single-effect absorption refrigeration cycle.

CHAPTER FOUR

4 SOLAR ENERGY AND STORAGE TANK MODELING

Demands on refrigeration and air conditioning have witnessed significant increase recently. Due to environmental awareness and technological advancement, improved standard living, and high oil price, the world is forced to utilize renewable energy sources to meet its growing demand[36]. In the solar thermal applications, the solar radiation is absorbed and transferred to the heat transfer fluid. The efficiency of solar thermal cooling system depends on the choice of solar thermal collector. Solar cooling can be divided into two main categories: passive and active solar cooling. In passive solar cooling method, the building is protected against heat by blocking sun radiation. The active solar cooling is classified into two categories: using traditional air conditioning system powered by photovoltaic panels and thermal cooling method. Absorption, adsorption, and desiccant are the main thermal cooling systems. In this paper focuses on absorption refrigeration system.

19. Working principle of Evacuated tube solar collector

Most evacuated tube collectors in the market use either a heat pipe design, which is popular in Europe, or a direct flow design which is more popular in China[47]. In the heat pipe design, an array of evacuated glass tubes -each holding a copper fin plate absorber fused to a copper heat pipe containing vaporizable fluid sealed in it- are connected to a manifold where heat is exchanged/transferred to other media such as water. Such a system is generally complicated by design, more expensive to manufacture and maintain, and heavy due to the copper plate and pipe use. The direct flow evacuated tube collector uses a collector encased in dual layer glass tubes with vacuum in between the layers to eliminate heat loss due to conduction[47]. The one end of the collector sealed with a layer of glazed coating and the other open end connected directly into a storage tank where the working fluid is stored. The open end and the storage tank are elevated with respect to the sealed end. Because of the circular geometry and the tilted installation setup, only the surface of the absorbing coating facing the sunlight irradiation will heat up and subsequently heats the adjacent working fluid. Under constant pressure, the density of the heated working fluid decreases and this causes the hotter fluid to float upward in the tube. Similarly, the working fluid with a lower temperature tends to move downward in the tube because of its relatively higher density. The temperature gradient between the absorbing coating surface facing the sunlight and the opposite side generates heat flow and thus circulation of the working fluid within the inner glass tube[47].Each evacuated tube consists of two glass tubes made from extremely strong borosilicate glass. The outer tube has very low reflectivity and very high transmissivity that radiation can pass through. The inner tube has a layer of selective coating that maximizes absorption of solar energy and minimizes the reflection, there by locking the heat [48]. The ends of the tubes connected to the copper header are fused together and a vacuum is created between them. This process is called as evacuation; it means that the air is pumped out from the cavity. The vacuum is created to recreate the thermoflask effect as vacuum acts as an insulator and does not allow short wave radiation to escape through the glass tube. This traps the solar radiation much more effectively and hence higher temperatures can be achieved. Evacuated tube collector receiver consists of a copper U-tube inside a glass vacuumed tube. The copper tube is surrounded by a cylindrical aluminum fin pressed on it. This fin enhances the heat transfer area between the inner glass absorber surface and the U-tube. The working fluid enters the collector inlet pipe, then it is evenly distributed to the U-tubes, absorbs heat and, at the end, it is returned to the outlet header pipe. The outer cylindrical glass transmits the rays to the inner glass tube, which conducts the energy to the absorber fin as shown figure 16. The energy transformed into heat is conducted by the fin to the copper U-tube and finally absorbed by the working fluid, which is water in this case [49]. Evacuated tube collectors are used for a variety of applications such as: space heating and cooling, industrial and domestic hot water requirements.

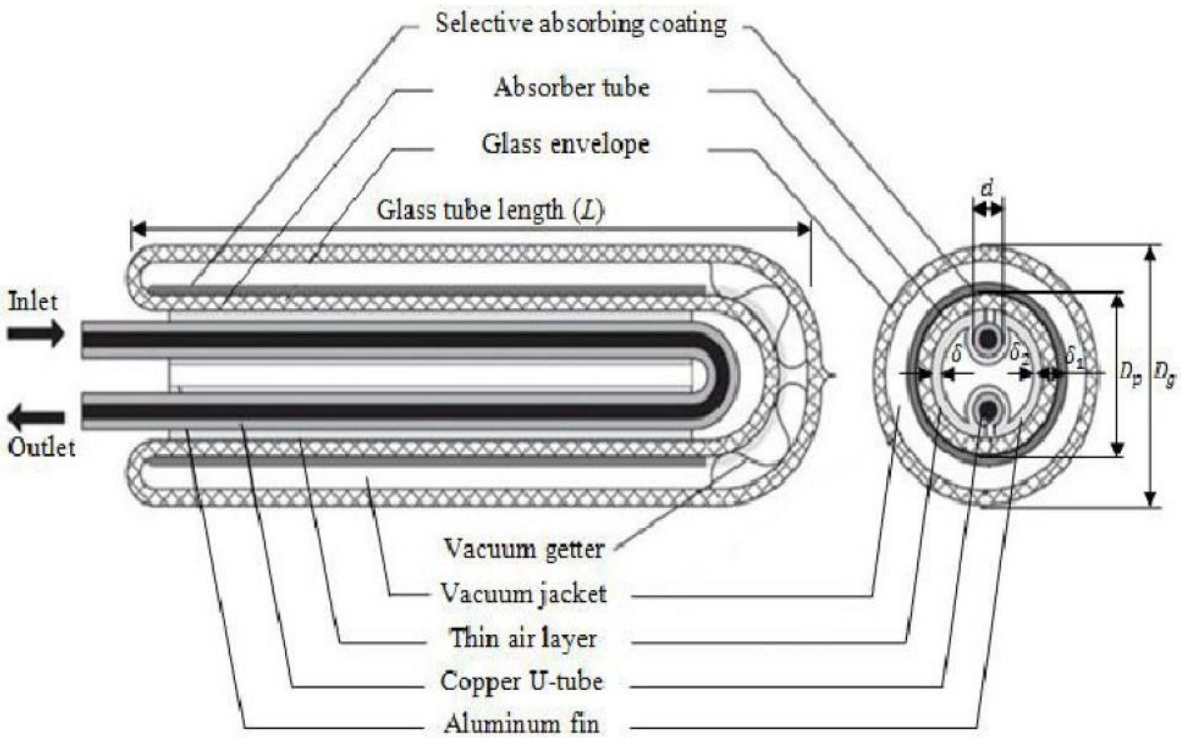

Figure 17: Schematic diagram of the glass evacuated tube solar collector with U-tube[49].

4.1 Mathematical modeling of evacuated-tube solar collector.

While modeling the evacuated tube collector some assumption made:

- Perfect insulation at the edges of the collector is assumed.
- No heat is supposed to be transported in the fluid moving direction, conduction is neglected.
- perfect vacuum is assumed between the two glass tubes, thus gas conduction is neglected
- Dust and pollution over the collector can be neglected.
- The gradients inside the glass cover and the absorber plate are assumed to be negligible.
- As the collector studied is a vacuum tube, free convection inside the glass tube is not taken into account.

We focus on, all heat transfer occurred in the solar collector are defined. The following diagram shows general description of thermal heat transfer in the solar collector.

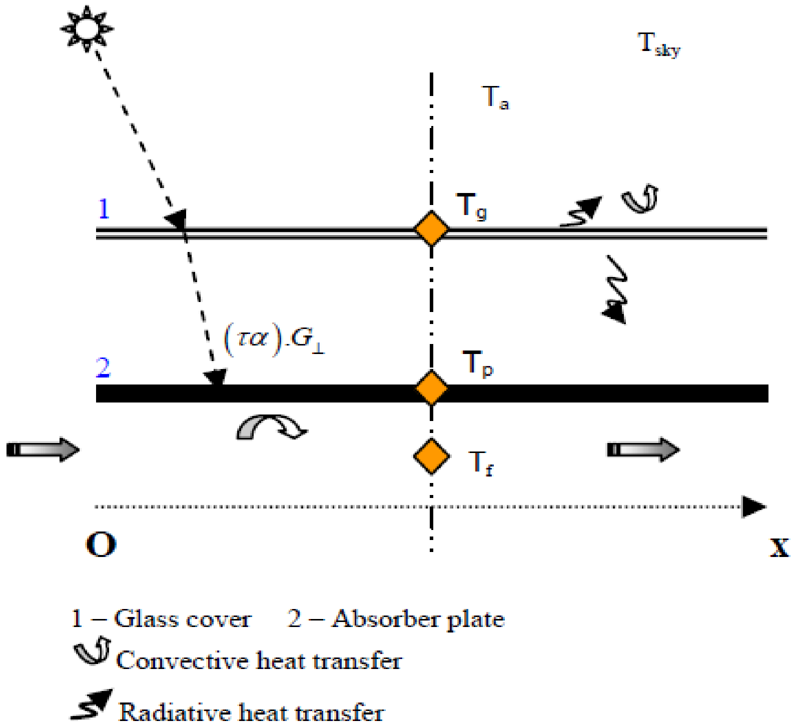

Figure 18: General description of the solar collector models[50].

20. Evacuated tube Collector differential equation system

The starting point of the model is a mathematical description proposed by Kamminga (1985). The model consists of three nodes corresponding to the fluid, the absorber plate and the transparent glass cover. It is considered that the temperature of the fluid is a function of x. The fluid is moving in a single channel with the velocity u, along x-axis. It results a 3-node collector model given by the following differential equation system:

$$C_g \frac{\partial T_g}{\partial t} = \varepsilon_g \sigma \left(T_{sky}{}^4 - T_g{}^4\right) + h_{g-a}\left(T_a - T_g\right) + \varepsilon_g \sigma \left(T_p{}^4 - T_g{}^4\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots...19$$

$$C_p \frac{\partial T_p}{\partial t} = \tau\alpha G_{\perp} + \varepsilon_g \sigma \left(T_g{}^4 - T_p{}^4\right) + U_{f-p}\left(T_f - T_p\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..20$$

Making an energy balance for the water stream, the temperature of the water leaving the solar collector at any time t is determined as:

$$\left(mc_p\right)_w \frac{dT_f}{d_t} = A_c x h_{f-p}\left(F_w x T_p - \frac{T_f + T_{fi}}{2}\right) - \left(\dot{m}cp\right)_w \left(T_f - T_{fi}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots...21$$

Where Fw = $2\eta_{f\ -1}, \eta_{f}$ is fin efficiencies, Cg is heat capacity of glass cover (KJ/m²·K), C_p is heat capacity of absorber (KJ/m²·K), G⊥ is global solar irradiance in the plane of the collector (KW/m²), $h_{f-p}$is heat transfer coefficient fluid – absorber (KW/m²·K), h_{g-a}– heat transfer coefficient glass – ambient (KW/m²·K), $h_{sky}$is heat transfer coefficient glass – sky (KW/m²·k), Ta– ambient temperature (°C), $T_{sky}$is sky temperature (°C), αis absorptive coefficient, ε is emissivity, σ is Stefan-Boltzmann constant (5.67*10-8 $W{\cdot}m^{-2}{\cdot}K^{-4}$)The total solar radiation absorbed, by absorber plate in evacuated tube solar collector is expressed as[49]:

$$\dot{Q}_{abs} = (\alpha\tau)GA_{ab} \qquad \text{.......... 22}$$

Where τ is the glass cover transmittance, α is the glass cover absorptance, $I_T$is the global solar irradiance on the collector surface and A_{ab} is the collector aperture (inner diameter of outer glass tube exposed tube length.

4.1.1.1 Solar Radiation on tilted surfaces

The ratio of beam radiation on the tilted surface to that on a horizontal surface[45].

$$Rb = \frac{G_{b,n}\cos\theta}{G_{b,n}\cos\theta_z} = \frac{\cos\theta}{\cos\theta_z}$$

γ=0°,
$$Rb = \frac{\cos(\phi-\beta)\cos\delta\cos\omega + \sin(\phi-\beta)\sin\delta}{\cos\phi\cos\delta\cos\omega + \sin\phi\sin\delta}$$

γ=180°,
$$Rb = \frac{\cos(\phi+\beta)\cos\delta\cos\omega + \sin(\phi+\beta)\sin\delta}{\cos\phi\cos\delta\cos\omega + \sin\phi\sin\delta}$$

4.1.1.2 Hourly Absorbed Solar Radiation

The total radiation on the tilted surface is the sum of the beam contribution calculated as $I_bR_b$and the diffuse on a horizontal surface, I_d[45]:

$$I_T = I_b R_b + I_d\left[\frac{1+\cos\beta}{2}\right] + I\rho_g\left[\frac{1-\cos\beta}{2}\right]$$

The useful heat that is transferred from the surface (absorber plate) to the thermal fluid is given by:

$$\dot{Q}_u = A_{abs}\left[G\tau\alpha - \varepsilon\sigma\left(T_p{}^4 - T_a{}^4\right)\right] \quad \text{……………………………………………………23}$$

Where: T_p is the absorber(plate surface) temperature, Ta is the ambient temperature, ε is the emissivity of the selective surface layer, G is the solar radiation,τ is the transmittance of the glass tube, α is the absorptivity of the selective layer and A_{abs} is the surface area for the absorption of solar energy [51].

The heat loss from absorber from absorber plate to ambient is:

$$\dot{Q}_r = A_{abs}\varepsilon\sigma\left(T_P{}^4 - T_a{}^4\right) \quad \text{……………………………………………………..24}$$

The useful energy gained can be evaluated by knowing the inlet and outlet fluid temperatures T_{in} and T_{out} and the mass flow rate of water $\dot{m}$[52][36] and given as:

$$\dot{Q}_u = \dot{m}cp_w\left(T_{out} - T_{in}\right) \quad \text{……………………………………………………………………25}$$

The useful energy may also be expressed in terms of the energy gained by the absorber and the energy lost from the absorber[36] as:

$$\dot{Q}_u = A_c F_R\left[\alpha\tau I - U_L\left(T_i - T_a\right)\right] \quad \text{……………………………………………………………..26}$$

The heat loss coefficient of the absorber tube is usually calculated based on the following correlation:

$$U_L = \frac{mc_P\left(T_i - T_f\right)_{av}}{A_a\left(T_m - T_a\right)_{av}} \quad \text{……………………………………………………………………27}$$

Where $\dot{m}$ is the mass flow rate of water inside the tube (kg/s); c_p is the specific heat of water (J/kg k); T_i and $T_f$are the inlet and outlet water temperatures, respectively, inside the tube; Aa is the surface area of the absorber; and (Tm-Ta)av is the average difference between the mean water temperature and the ambient temperature over some period of time. Budihardjo et al.16 showed that heat loss coefficient varies with tube quality. For good quality tubes, the heat loss coefficient varies from 0.5w/m^{2}k (at Tm-Ta=25^{0}c) to 0.65w/m^{2}k (at Tm-Ta=65^{0}c). For lower quality tubes, the coefficient varies from 0.7 to 0.9w/m^{2}k for the same temperature range[36]. The thermal performance of solar collector under steady state conditions can be calculated as follows[52]:

$$\frac{\dot{Q}_U}{A_C} = F_R(\alpha\tau)G - F_R U_L(T_m - T_{in}) \quad \text{..28}$$

From the above equation it is observed that the thermal performance of solar collector depends on the intensity of the sunlight striking the collector surface ,the temperature of the surrounding environment and the absorber plate and its optical and thermal performance represented by the values of (τα) and U_L respectively[53]. The transmittance (τ) of the glass cover and absorptance (α) of the absorber plate depend on the incidence angle of the collector and according to literature the product of the transmittance and absorptance(τα) is approximately 0.836. The useful solar energy collected is transferred to the hot liquid storage tank from which the generator of the absorption system is supplied with input thermal energy[54].

The useful energy output of an evacuated tube is then the difference between the absorbed solar radiation and the thermal loss

$$\dot{Q}_u = \dot{Q}_{abs} - \dot{Q}_{thermal\_loss} \quad \text{...29}$$

4.2 Thermal losses in tubes

Overall heat loss coefficient was very less in the collector due to the sealing of outer and inner glass together and by maintaining evacuated annular space between the tubes. Heat is lost from the outside of the outer glass tube by both radiation and convection [4], but from the absorber inner glass tube to the outer glass tube only by radiation heat transfers figure 19, as the heat lost by both convection and conduction are eliminated by vacuum (about 5×10-3 Pa). There is a small amount of conduction from the glass tubes walls through the structure that supports them but this effect has been assumed to be neglected [38]. The heat lost by the tube is given by:

$$\dot{Q}_{thermal-loss} = U_L A_{ig}\left(T_{ig} - T_a\right) \quad \ldots\ldots 30$$

Where U_L is overall heat loss coefficient of the tube, A_{ig} is area of the inner glass tube, T_{ig} is temperature of the inner glass tube and T_a is the ambient temperature.

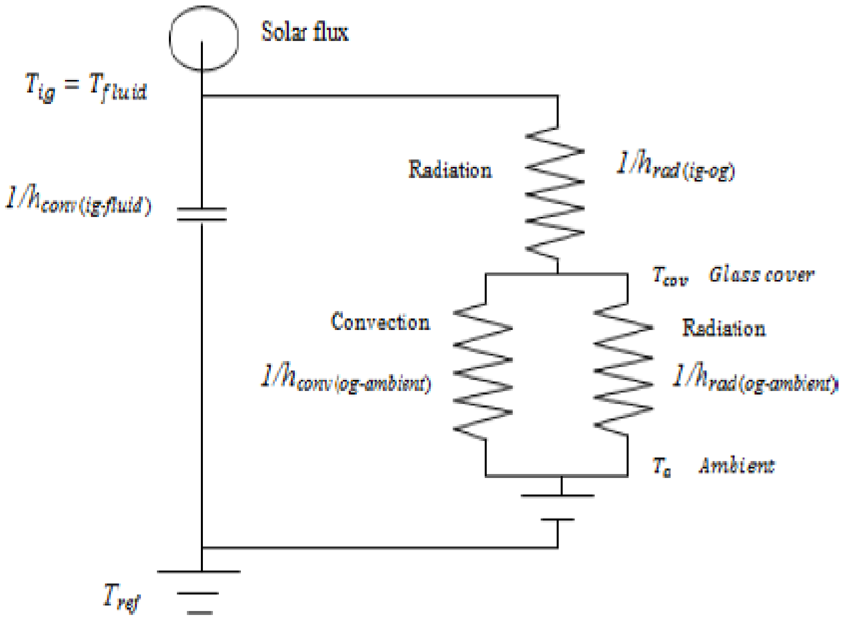

Figure 19: Thermal Circuit Analog of Evacuated Tube [38]

The heat loss coefficient U_L is calculated from thermal resistance between the absorber tube and the outer glass tube and between the outer glass tube and the surrounding such that[38]:

$$U_L = \left\{\left(\frac{1}{h_{rad\_og} + h_{wp}}\right) + \left(\frac{1}{h_{rad\_ig}}\right)\right\}^{-1} \quad \ldots\ldots 31$$

Where, h_{rad_og} is the radiation heat transfer coefficient of the outer glass tube h_{wp} is the adjusted convection heat transfer coefficient of the outer glass tube and h_{rad_ig} is the radiation heat transfer coefficient of the inner glass tube.

21. Radiation from the Outer Glass Tube

The radiate loss the outer glass tube surface accounts for radiation exchange with the sky at temperature (T_S)). The radiation heat transfer coefficient from the outer glass tube surface can be[38]:

$$h_{rad\_og} = \varepsilon_{og}\sigma \frac{\left(T_{Og}{}^4 - T_{Sky}{}^4\right)}{\left(T_{og} - T_{sky}\right)} \qquad \text{………………………………………………………………...32}$$

Where: ε_{og}isemissivity of the outer glass;$T_{og}$is the temperature of the outer glass tube.

Relates sky temperature to the local air temperature [38][5] can be:

$$T_{sky} = 0.0552 T_a^{1.5} \qquad \text{……………………………………………………………………………33}$$

The actual radiating surface of each outer glass tube is equal to its surface area, but in fact, much of the view of each outer glass tube is the neighboring outer glass tube or the building structure behind the outer glass tube Heat radiated only to the front. The area of the radiation, assume the product of the outer glass diameter and the length = Dog x L

22. Outer Glass Tube Convection

The adjusted convection heat transfer coefficient) of the outer glass tube surface is approximated by heat transfer coefficient around the outer glass tube. h_w can be determined from data illustrated by Holman for single tube. For the case of natural convection heat transfer [38]:

$$h_w = 5.8 + 3.8v \qquad \text{…………………………………………………………………………….34}$$

Where: h_w is convection heat transfer coefficient around the outer glass tube and V is the wind speed (m/s). The value of h_w represents the loss per unit area of the outer glass tube surface. For consistency, all losses are referenced to the net collector area.

$$h_{wp} = \frac{A_{og}}{A_{ig}} * 0.6 * h_w \qquad \text{………………………………………………………………………...35}$$

23. Radiation from Absorber Glass Tube to the Outer Glass Tube

The radiation heat transfer coefficient between the absorber tube and outer glass tube [38] can be:

$$h_{rad\_ig} = \varepsilon_{ig} - \varepsilon_{og}\sigma \frac{\left(T_{ig}{}^{4} - T_{og}{}^{4}\right)}{\left(T_{ig} - T_{og}\right)} \quad \text{.........36}$$

Where ε_{ig} is the effective emissivity between the absorber glass tube (inner tube) and the outer glass tube:

$$\varepsilon_{ig} - \varepsilon_{og} = \left\{ \frac{1}{\varepsilon_{ig}} + \frac{A_{ig}}{A_{og}} \left(\frac{1}{A_{og}} - 1 \right) \right\} \quad \text{.........37}$$

The glass temperature Tog is found by noting that the heat loss from the absorber tube to the outer glass tube is the same as from the absorber tube to the surrounding, it can be formulate as:

$$T_{og} = T_{ig} - U_{L} \frac{T_{ig} - T_{a}}{\left(h_{rad\_ig}\right)} \quad \text{.........38}$$

The collector efficiency can be formulated as below[53]:

$$\eta_{C} = F_{R}\left[(\tau\alpha) - U_{L} \frac{T_{in} - T_{a}}{I} \right] \quad \text{.........39}$$

In this equation, Where F_R is the collector heat removal factor, $(\tau\alpha)$ is transmittance-Absorptance product,T_{in} is the collector inlet temperature and T_a is the environment temperature.
According to Assilzadeh et al... 20004, for evacuated tube collectors simulated F_R $(\tau\alpha)$ = 0.82 and F_RU_L = 2.19 W/m^{2} ^{O}C.

The efficiency of the collector is by calculating the net output power by considering the heat losses[52]:

$$\eta = F_{R}(\alpha\tau) - F_{R}U_{L} \left(T_{m} - T_{a}\right) / G \quad \text{.........40}$$

The heat loss coefficient U_L is a function of the ambient temperature and the temperature of the absorber plate but in reality it is not constant's collector heat removal factor is a function of flow rate which is considered due to the fact that the average fluid temperature and the average absorber temperature are not same and the difference between these two temperatures is 0 to1. The efficiency of collector depends on the heat loss coefficient (U_L) and the design of the absorber plate of the collector.The solar system sizing strategy is to meet the load under the sunniest possible condition[55]:

$$A_C = \frac{L}{\eta_{solar} x G_{\max}}$$

……………………………………………………………………………..41

Where L = daily energy consumption load (kwh/day), η_{solar} is the efficiency of the solar system, and G_{max} isthe maximum daily solar radiation occurring in thesunniest summer month (kWh/m^2/day).

The second ways of find efficiency of an Evacuated tube solar collector can be calculated using:

$$\eta = \frac{\dot{Q}_u}{A_C G} = \frac{\dot{m} c_p (T_{out} - T_{in})}{A_C G}$$

………………………………………………..42

Where A_c is solar collector area (m^2), Gis solar radiation (w/m^2), M is water flow rate (kg/h), and $c_p$is specific heat of water (kJ/Kg.k), Tout water outlet temperature (^{o}c) and T_{in} is water inlet temperature (^{o}c). The above equation gives the efficiency of evacuated collector with the known value of fluid mass flow rate and the measured value of fluid inlet and outlet temperature.

4.3 Thermal storage and Thermal Storage tank modeling

Solar water heater is driven by natural circulation of the fluid in the collector and the storage tank. It consists of all-glass vacuum tubes, inserted directly into a storage tank, with water in direct contact with the absorber surface. The limitation of this concept is that it can only be used for a low-pressure system, as the tubes can only withstand a few meters of water head[38]. The higher the flow rate, more rapidly energy is extracted from the storage tank and the temperature of the inlet water to the collectors drops; as a result its efficiency increases. For this reason, the mass flow rate M_L, was obtained considering the energy extracted from the storage tank (Q_L) to correspond to the maximum energy transfer to the water in the collectors. Two type's storage tank modeling stratified and non-stratified tank modeling

24. Thermal stratification

4.3.1.1 Low Flow rate and Thermal Stratification

Holland's (1989) suggested that "By using low collector flow rates (roughly one-seventh of the standard value) with thermally stratified storage tanks, the calculated performance of solar hot water systems for domestic, commercial, and industrial use can be improved by as much as 38%, if one compares to a system having a fully mixed tank and a high flow rate [56].

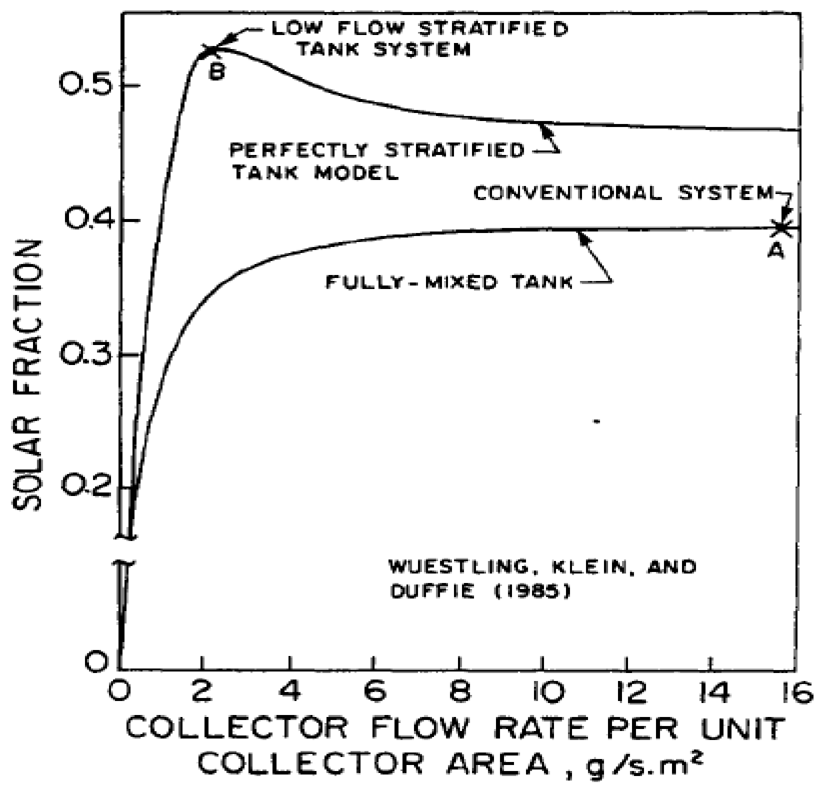

Figure 12: Annual solar fraction & collector flow rate per unit area [56]

Due to effect of gravity and the buoyant force, a high-density fluid or cold water will settle at the lower part of a tank and a low-density fluid or hot water at the upper part of the storage tank.

Furbo (1993) explains, A high degree of thermal stratification increases the thermal performance of solar hot-water systems because the return temperature to the solar collector is lowered. A lower return temperature to the solar collector will increase the efficiency of the solar collector. Solar water heater configuration based on natural convection of water between the solar collector panel and the hot water tank, called Thermosiphon system[56]. Thermal stratification in the storage tank is influenced by various parameters such as insulation thickness, flow rate at inlet and outlet, tank inlet and outlet geometry.

4.3.1.2 Thermal stratified storage tank modeling

To start the thermal modeling, a control volume approach is applied to the hot water tank. Energy balance is applied for the elemental volume i.e., rate of change of energy inside the elemental volume is equal to the net rate of input/output of energy to the elemental volume. The total volume is divided into finite control volumes[56].

On application of first law of Thermodynamics,

$$m_S x C_P x \frac{dT_S}{dt} = \dot{Q}_C - \dot{Q}_L - UAx(T_S - T_a) \quad \text{...43}$$

Where, $m_S$is mass of water in the storage tank, kg,Tsis the instantaneous tank temperature, K,$Q_C$is heat addition from the solar collector panel,$Q_L$is heat removal from the storage tank due to load to the utility, kw,Uis overall heat transfer coefficient of the storage tank, kw/ (m^2K),Ais the storage tank area, m^2,Tais the ambient temperature, K. the properties of water and their dependences on the temperature [57].

Density [Kg/m^3]:

$$\rho = 863 + 1.21xT - 0.00257xT^2 \quad \text{...44}$$

The heat removal from storage tank(hot water supply rate is expressed as[58]:

$$\dot{Q}_L = ALOAd(T_s - T_{ws}) \quad \text{...45}$$

$$ALOAD = (\rho c_p)xDWCxpp(j) \quad \text{...46}$$

25. Non-stratified storage tank

In a non-stratified tank model, a fully mixed tank model of water storage is developed using a lumped analysis. The tank is assumed to be a single unit and an energy analysis is applied to only one node representing the overall fluid temperature distribution in the whole tank [18]. The heat capacity of the fluid in the storage tank can be calculated using the mass, specific heat capacity and the temperature change that occurs in the tank as:

$$\dot{Q} = mxc_p x\Delta T \quad \text{......47}$$

Where; Q is heat capacity ΔT is temperature variation in the tank (K), m is liquid mass in the storage tank (kg) and C_p is specific heat capacity of the liquid (J/kg K).

The transient energy balance can then be formulated for the storage tank as:

$$mCp\frac{dT_s}{dt} = \dot{Q} - L - (UA)(T_s - T_a) \quad \text{......48}$$

Where m is mass of the liquid in the storage tank (kg), Q and L is energy addition/removal from the collector and to the load at a specified time (KJ), Ta is ambient temperature for the tank (K),(UA) is loss coefficient area product, (W/K) and T_s is the temperature of water in tank.

4.4 Thermal Storage Tank heat Losses

Heat loss from the tank helps to build up thermal stratification in the tank. Due to the loss to the surroundings, the fluid close to the wall of tank has lower temperature than the fluid at the center of the tank[57]. The amount of energy lost by the storage unit to its surroundings is represented as a function of time by $Q_{loss\,(t)}$ in the equation[59]:

$$\dot{Q}_{loss(t)} = U_S A_S (T_s - T_a) \quad \text{......49}$$

Where heat loss coefficient is calculated by:

$$U_{total} = 1.75 + 0.00198xT \quad \text{......50}$$

Where T is water temperature in the tank, Us are the heat transfer coefficient of the storage unit, As is the surface area of the storage unit.

CHAPTER FIVE

5 MATHEMATICAL MODELING

5.1 System efficiency and coefficient of performance

Compared with mechanical refrigeration system, absorption systems have a low coefficient of performance. However, absorption chillers can substantially reduce operating costs because they are powered by low grade waste heat[60]. It is recommended to use a solution heat exchanger in absorption systems as it allows the solution from the absorber to be preheated before entering the generator by using the heat from the hot solution leaving the generator. The temperature of the heat source is the most important factor in the thermal efficiency of an absorption refrigeration system. Efficiency of an absorption refrigeration system can be expressed by the coefficient of performance (COP), which is defined as the ratio between the amounts of heat/energy absorbed from the environment by the evaporator and the heat/energy supplied to the generator to operate the cycle [60].

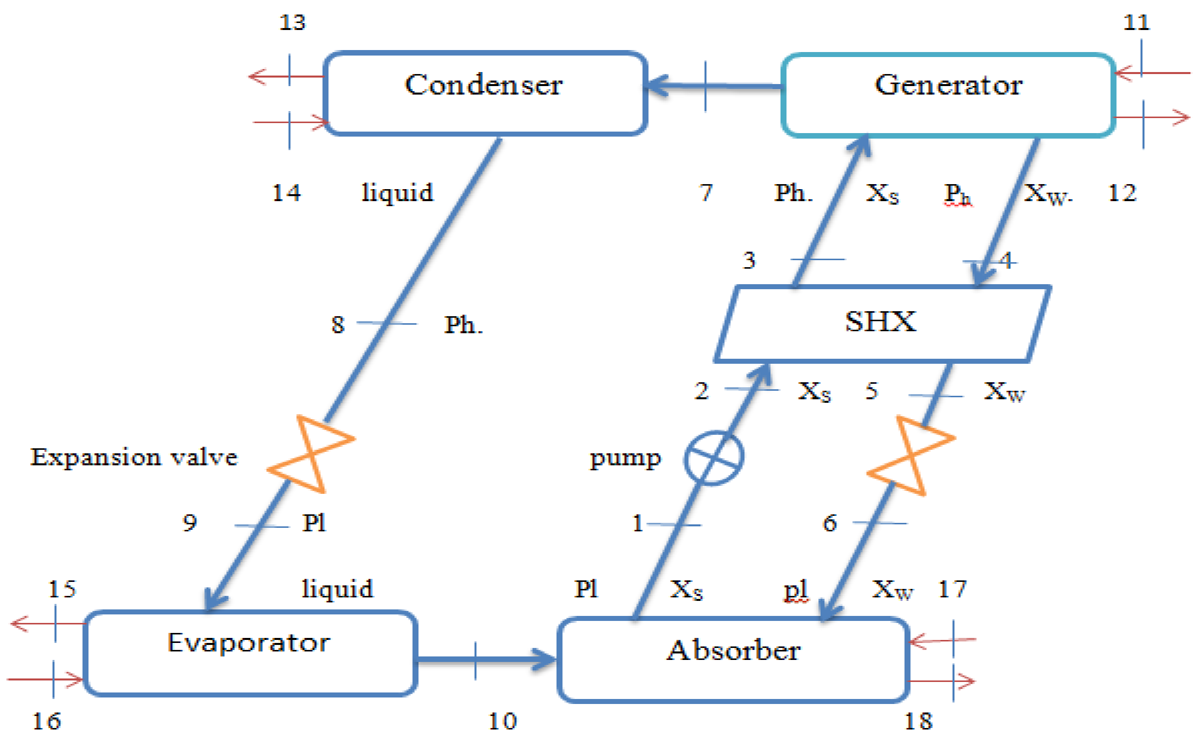

Figure 13: Single effect Ammonia -Water absorption refrigeration cycle.

5.2 Thermodynamic analysis of the system

The first and the foremost factor to be considered before thermodynamic analysis is selection of refrigerant absorbent mixture. Commonly used refrigerant- absorbent mixtures are Aqua-ammonia, LiBr- water, etc. One advantage with Vapour absorption refrigeration system is that it makes use of eco-friendly refrigerant[14].Ammonia – water absorption refrigeration system have a generator, absorber, condenser, evaporator, and a solution heat exchanger. The determination of the thermodynamic properties of each state point in the cycle, the amount of heat transfer in each component, and the flow rates at different lines depend on the generator temperature, evaporator temperature, condenser temperature, absorber temperature, liquid-liquid heat exchanger effectiveness, and the refrigeration load[60].A thermodynamic analysis of the proposed vapour absorption refrigeration system, the following assumptions were made:

- No pressure changes except through the flow pump and pressure expansion valve.
- At point 1, 4 and 8, there is only saturated liquid.
- At point 10, there is only saturated vapour. Pumping is isentropic. Assume a weak solution contains a larger percentage of refrigerants and a smaller percentage of absorbent and a

strong solution contains a larger percentage of absorbent and a smaller percentage of refrigerants. The percentages of the weak solution at state 4, 5 and 6 and the percentages of the strong solution at state 1, 2 and 3 will remain same. The temperatures at thermodynamic states 11, 12, 13, 14, 15, 16, 17 and 18 are theexternalcircuit for water which is used to input heat for the components of the system shown in Figure 13.

- The specific enthalpy of superheated refrigerant at inlet of condenser form generator is equal to the specific enthalpy of saturated refrigerant at the generator temperature.
- This system has two pressure limits; one is a high-pressure limit and the other is the low-pressure limit.

  P1 = P6 = P9 = P10 = Low pressure.

  P2 = P3 = P4 = P5 = P7 = P8 = High pressure.

5.3 Mathematical model of sub component

The thermodynamic equations have been derived in terms of mass flow rates and enthalpy by applying mass and energy balance for each component. Then the actual system conditions like temperature, pressures, enthalpies are substituted in the equations to finally obtain the COP value for the system[61].The assumptions for carrying out the Thermodynamic analysis of the system are as following:

a. No pressure drops due to friction
b. Only pure refrigerant boils in the generator.

The mathematical relationship used to model absorption refrigeration systems are based on mass and energy conversion and take it as follow:

$$\sum_i \dot{\mathbf{m}}=0 \qquad \text{..51}$$

$$\sum \dot{m}_{in} h_{in} = \sum \dot{Q} = \sum \dot{m}_{out} h_{out} \qquad \text{...52}$$

Where $\dot{m}$ is mass flow rate (kg/s),h is specific enthalpy (KJ/kgand Q is the heat transfer rate

26. Generator (Desorbed)

In generator Ammonia water, solution will be boiling using a hot water supply from the solar collector or storage tank. Once the weak ammonia water solution is brought to the boiling point, which is designed to take place at a certain operating temperature and pressure, the mixture will release the water in vapor form. For the single effect case, vapor from the desorbed will directly flow towards the colder region in the condenser.

The mass Balance across the generator:

$$\dot{m}_3 = \dot{m}_4 + \dot{m}_7 \qquad \text{.......53}$$

$$\dot{m}_3\, xX_3 = \dot{m}_4\, xX_4 + \dot{m}_7\, xX_7 \qquad \text{.......54}$$

$$\dot{m}_1 = \frac{\dot{m}_4\, xX_s}{X_w} \qquad \text{.......55}$$

Where X is Ammonia content in working media, X_S is Ammonia content of strong solution and X_w is Ammonia content of weak solution.

Combining Equation 54 and 55:

$$\dot{m}_6 = \frac{-\dot{m}_4}{1 - X_0 / X_1} \qquad \text{.......56}$$

On balancing the energy across the generator:

$$\dot{Q}_g + \dot{Q}_3 = \dot{Q}_4 + \dot{Q}_7 \qquad \text{.......57}$$

$$\dot{Q}_3 = \dot{m}_3\, xh_3 \qquad \text{.......58}$$

$$\dot{Q}_4 = \dot{m}_4\, xh_4 \qquad \text{.......59}$$

$$\dot{Q}_7 = \dot{m}_7\, xh_7 \qquad \text{.......60}$$

Using Equation 58-60:

$$\dot{Q}_3 = \dot{m}_4\, xh_4 + \dot{m}_7\, xh_7 - \dot{m}_3\, xh_3 \qquad \text{.......61}$$

Heat supplied to the generator is:

$$\dot{Q}_g = \dot{m}_6 \, x c_{pwater} (T_{11} - T_{12}) \quad \text{.......62}$$

$$\dot{m}_{11} = \frac{\dot{m}_4 \, x h_4 + \dot{m}_7 \, x h_7 - \dot{m}_3 \, x h_3}{\dot{m}_6 \, x c_{pwater} (T_{11} - T_{12})} \quad \text{.......63}$$

The circulation ratio (ƛ,) is defined as the mass flow rate of solution from the absorber to the generator to the mass flow rate of working fluid (refrigerant), that:

$$\lambda = \frac{\dot{m}_3}{\dot{m}_7} = \frac{\dot{m}_{ss}}{\dot{m}_r} \quad \text{.......64}$$

3.1.1. CONDENSER

In the condenser, incoming vapor from the concentrator is cooled to a liquid form in the high-pressure region. This is a natural process since heat is transferred from the hot to cold region. This causes the vapor to flow from the desorbed region to the condenser region.

The mass Balance across the condenser:

$$\dot{m}_7 = \dot{m}_8 \quad \text{.......65}$$

$$\dot{m}_8 \, x X_8 = \dot{m}_7 \, x X_7 \quad \text{.......66}$$

On balancing the energy across the condenser:

$$\dot{Q}_C + \dot{Q}_8 = \dot{Q}_7 \quad \text{.......67}$$

$$\dot{Q}_8 = \dot{m}_8 \, x h_8 \quad \text{.......68}$$

$$\dot{Q}_7 = \dot{m}_7 \, x h_7 \quad \text{.......69}$$

Then from above we get Q_C:

$$\dot{Q}_C = \left(\dot{m}_7\ x h_7\right) - \left(\dot{m}_8\ x h_8\right) \quad \text{……………………..…………………………………….70}$$

Heat supplied to the condenser is:

$$\dot{Q}_C = m_{13} x c_{pwater}\left(T_{13} - T_{14}\right) \quad \text{……..…………………………………………………..71}$$

Comparing with above 70 and 71 equation, we get:

$$\dot{m}_{14} = \frac{\left(\dot{m}_7\ x h_7\right) - \left(\dot{m}_8\ x h_8\right)}{c_{pwater}\left(T_{13} - T_{14}\right)} \quad \text{…………………………………………………………...72}$$

27. Evaporator

The evaporator forms the low-pressure side of the absorption refrigeration system and its function is to produce the chiller water. The low temperature liquid-vapor water exiting the condenser will evaporate to a low water vapor temperature due to the heat transfer from the water supply tubes. The mass Balance across the evaporator:

$$\dot{m}_{10} = \dot{m}_9 \quad \text{……………………………………………………………………………...73}$$

$$\dot{m}_{10}\ x X_{10} = \dot{m}_9\ x X_9 \quad \text{…………………………………………………………………….....74}$$

Balancing the energy across the evaporator:

$$\dot{Q}_e + \dot{Q}_9 = \dot{Q}_{10} \quad \text{………………………...……………………………………………...75}$$

$$\dot{Q}_9 = \dot{m}_9\ x h_9 \quad \text{……………………..…………………………………………………..76}$$

$$\dot{Q}_{10} = \dot{m}_{10}\ x h_{10} \quad \text{…...77}$$

On putting the energy values for points 75 and 76 into Equation 77:

$$\dot{Q}_e = \left(\dot{m}_{10}\ xh_{10}\right) - \left(\dot{m}_9\ xh_9\right)$$ ……………..………………………………………………...78

Knowing that heat extracted from the evaporator is:

$$\dot{Q}_e = \dot{m}_{15}\ xc_{p_{water}}\left(T_{16} - T_{15}\right)$$ ……………………………………………………………….79

Comparing Equations 28 and 29:

$$\dot{m}_{16} = \frac{\left(\dot{m}_{10}\ xh_{10}\right) - \left(\dot{m}_9\ xh_9\right)}{c_{pwater}\left(T_{16} - T_{15}\right)}$$ ……………………………………………………………80

28. Absorber

The absorber forms the lowest pressure side of the absorption refrigeration system. Its function is to absorb the water from the evaporator to be circulated back to the desorbed. The evaporation process from the evaporator causes the refrigerant to flow to the absorber. Once the Ammoniawater solution becomes diluted, it will be returned to the concentrator. The solution is also interrelated between the absorber and desorbed through the heat exchanger for better performance of the absorption system.

The mass Balance across the absorber:

$$\dot{m}_1 = \dot{m}_{10} + \dot{m}_6$$
………………………………………………………………………………...81

$$\dot{m}_1\ xX_1 = \dot{m}_{10}\ xX_{10} + \dot{m}_6\ xX_6$$ ……………………………………………………………..82

On balancing the energy across the absorber:

$$\dot{Q}_a + \dot{Q}_1 = \dot{Q}_6 + \dot{Q}_{10}$$ ……………………………………………………………………….83

$$\dot{Q}_1 = \dot{m}_1\ xh_1$$ ………………...……………………………………………………………..84

$$\dot{Q}_6 = \dot{m}_6\ xh_6$$ ………………………………………………………………………...…………..85

$$\dot{Q}_{10} = \dot{m}_{10}\, x h_{10} \quad \text{.......................................86}$$

The relation of effectiveness of the heat exchanger can be solved using:

$$T_5 = -\varepsilon - \frac{T_4}{T_4 - T_2} x\left(T_4 - T_2\right) \quad \text{.......................................87}$$

By putting energy values of equation 84, 85 and 86 into Equation 83, we get:

$$\dot{Q}_a = (\dot{m}_6\, x h_6) + \left(\dot{m}_{10}\, x h_{10}\right) - (\dot{m}_1\, x h_1) \quad \text{.......................................88}$$

We also know that the heat transfer from the absorber is:

$$\dot{Q}_a = \dot{m}_{17}\, x c_{pwater}\left(T_{18} - T_{17}\right) \quad \text{.......................................89}$$

On comparing Equations 88 and 89, we can get:

$$\dot{m}_{18} = \frac{\left(\dot{m}_6 x h_6\right) + \left(\dot{m}_{10}\, x h_9\right) + \dot{m}_1 x h_1}{c_{pwater}(T_{18} - T_{17})} \quad \text{.......................................90}$$

29. Heat-Exchanger

Heat Exchangers are used to exchange heat between the hot concentrated solution returning to the absorber and the cool dilute solution being pump to the desorbed. The heat exchangers reduce the heat energy required in the concentrator to raise the temperature of the dilute solution to the boiling point. Without heat exchangers, more heat needs to be supplied to the desorbed in order to boil the Ammonia-water solution. Heat exchangers also reduce the cooling load in the absorber section. The requirement of the heat exchanger area for the absorption system was calculated using the log mean temperature difference method. Since the rate of heat transferred, is a linear function of the overallcoefficient of heat transfer:

$$Q = UA * LMTD \quad \text{.......................................91}$$

$$\Delta T_{lm} = \frac{\left(T_{hot,1} - T_{cold,1}\right) - \left(T_{hot,2} - T_{cold,2}\right)}{\ln\left(\frac{T_{hot,1} - T_{cold,1}}{T_{hot,2} - T_{cold,2}}\right)} \qquad \text{……92}$$

The same as:

$$LMTD = \frac{(T_4 - T_3) - (T_5 - T_2)}{\ln\left(\frac{T_4 - T_3}{T_5 - T_2}\right)} \qquad \text{……93}$$

Solution Heat exchange efficiency becomes:

$$\eta_{she} = \ln\left(\frac{T_4 - T_3}{T_5 - T_2}\right) \qquad \text{……94}$$

Solution heat exchanger employs the transfer heat between two fluids Streams. Weak solution (absorbent alone) from the generator carries some refrigerant which is transferred to strong solution that's pumped from absorber[14].

30. Pump

Before we size pump, find Reynolds number, pressure in pipe and head loss in pipe[62].

$$\mathrm{Re} = \frac{V_{aV} x D}{\nu} = \frac{\rho V_{aV} D}{\mu} \qquad \text{……95}$$

For flow pipe

For Re < 2300 Laminar flow

For Re >4000 turbulent

The volume flow rate of overall system based on pressure is calculated by:

$$w = P\dot{V} \qquad \text{……96}$$

5.3.1.1 Pressure loss

In practice, it is found convenient to express the pressure loss for all types of fully developed internal flows (laminar or turbulent flows, circular or non-circular pipes, smooth or rough surfaces, horizontal or inclined pipes) as[62]:

$$\Delta P_L = f\frac{L}{D}\rho\frac{V^2{}_{aVg}}{2} \qquad \text{.........97}$$

$\rho\frac{V^2{}_{aVg}}{2}$: is the dynamic pressure and f is the Darcy friction factor.

5.3.1.2 Losses in pipe

In the analysis of piping systems, pressure losses are commonly expressed in terms of the equivalent fluid column height, called the head lossh_L. Noting from fluid statics that ΔP =ρgh and thus a pressure difference of ΔP corresponds to a fluid height of h = ΔP/ρg, the pipe head loss is obtained by dividing ΔPL by ρg to give[62]:

Head loss: $$h_L = \frac{\Delta PL}{\rho g} = f\frac{L}{D}\frac{V^2{}_{aVg}}{2g} \qquad \text{.........98}$$

The major losses, is calculated using moody chart, the friction factor in fully developed turbulent pipe flow depends on the Reynolds number and the relative roughness/*D*, which is the ratio of the mean height of roughness of the pipe to the pipe diameter. The losses that occur in piping components such as fitting, valves ,bends, elbows ,tees, inlets ,exists, enlargements and contractions are minor losses[62].

The head loss for a component is determined as:

Minor losses: $$h_L = K_L\frac{V^2}{2g} \qquad \text{.........99}$$

When all loss coefficients available, the total head loss entire piping systemis determined as:

$$h_{L,total} = h_{L,major} + h_{L,\min or} = \sum_i fi\frac{Li}{Di}\frac{Vi^2}{2g} + \sum_j K_L V^2/2g \qquad \text{.........100}$$

Where *i* represent each pipe section with constant diameter and *j* represents each component that causes a minor loss.

If the entire piping system being analyzed has a constant diameter, the above Eq. reduces to:

Total head loss (D=constant): $$h_{L,total} = \left(f\frac{L}{D} + \sum K_L\right)\frac{V^2}{2g} \qquad \text{.........101}$$

Many practical piping systems involve a pump to move a fluid from one reservoir to another:

$$\dot{W}_{pump} = \dot{v}\,\rho g h_{pump} = \dot{m}\, g h_{pump} \qquad \text{...102}$$

$$h_{pump} = Z_2 - Z_1 + h_L \qquad \text{...103}$$

Once the useful pump head is known, the mechanical power that needs to be delivered by the pump to the fluid and the electric power consumed by the motor of the pump for a specified flow rate are determined from.

$$\dot{W}_{pump,shaft} = \frac{\dot{v}\,\rho g h_{pump}}{\eta_{pump}}$$

....104

$$\dot{W}_{elect} = \frac{\dot{v}\,\rho g h_{pump}}{\eta_{pump\_motor}} \qquad \text{...105}$$

where $h_{pump–motor}$ is the efficiency of the pump–motor combination, which is the product of the pump and the motor efficiencies, The pump–motor efficiency is defined as the ratio of the net mechanical energy delivered to the fluid by the pump to the electric energy consumed by the motor of the pump, and it usually ranges between 50 and 85 percent.

Reynolds number of the entire flow system is:

The volume flow rate of overall system based on pressure is calculated by:

$$\dot{V} = \frac{w}{p} = 0.00112\, {m^3}/{s}$$

Where the recommended pressure is15-25 bar and the power required to run the overall is 3HP=2.235kw.

The average velocity of the flow is determined:

$$v_{avg} = \frac{\dot{V}}{A_c} = 1.58\, {m}/{s}$$

The pipe diameter of the entire system is constant =30mm, overall length of copper tubepipe (evaporator) = 60m.

$$\mathrm{Re} = \frac{V_{aV} x D}{\nu} = \frac{\rho V_{aV} D}{\mu} = 47262.77$$

From Appendix A, Using moody chart, the friction factor in fully developed turbulent pipe flow depends on the Reynolds number and the relative roughness/*D=0.00005* of copper, which is the ratio of the mean height of roughness of the pipe to the pipe diameter, the Darcy friction factor, f =0.0225.

The pressure drop: $\Delta P_L = 0.56kpa$

The total head loss of system is:

$$h_{L,total} = \left(0.0225\frac{30}{0.03} + 3.65\right)\frac{1.58^2}{2x9.81} = 3.325m$$

Assume, take head of pump is Z=8m

$$h_{pump} = 8 + 3.325 = 11.325m$$

Then work of pump is:

$$\dot{W}_{pump} = 122w$$

The mass flow rate of the system:

$$\dot{m} = \frac{\dot{W}_{pump}}{gh_{pump}} = 1.098 \, {kg}/{s}$$

The mechanical power that needs to be delivered by the pump to the fluid and the electric power consumed by the motor of the pump for a specified flow rate are determined from.

Take η_{motor} = 90%, η_{pump} = 75%, then $\eta_{pump\_motor} = \eta_{motor}$ x η_{pump} = 0.675

$$\dot{W}_{pump,shaft} = \frac{122}{0.7} = 174.3w$$

$$\dot{W}_{elect} = \frac{122}{0.675} = 180.74w$$

CHAPTER SIX

6 Economical analysis of small scale absorption refrigeration cycle

The absorption cycle can be economically attractive if the heat is sufficiently cheap. For a given ambient temperature, the COP will increase with an increase in the heating medium temperature input to the generator

6.1 The Specific Life Cycle Costs Analysis

The cost of any energy delivery process includes all of the items of hardware and labor that are involved in installing the equipment plus the operating expenses. The object of the economic analysis can be viewed as the determination of the least cost method of meeting the system life cycle costs to install cooling capacities was calculated as the SLCCA[63].

$$SLCCA = \frac{LCCA}{\dot{Q}_{cold}} \quad \text{...106}$$

The LCCA was the measurement of the total cost of a system through-out its life time

$$LCCA = AC_A + AC_M + AC_R + BC_{RE} - BC_S \quad \text{...107}$$

Where in this case

$$A = \left[\frac{(1+i)^n - 1}{i(1+i)^n}\right], \quad B = \left[\frac{1}{(1+i)^n}\right] \quad \text{...108}$$

We will assume that the installed cost of a solar cooling system includes the equipment and installation costs : it includes the cool storage unit ,the associated equipment cost(pumps, heat exchangers(CE) and piping, etc.), cost of installing the solar system(CI),storage tank(CS) , capital cost of the auxiliary cooling system(CA) , the installation cost of the solar heating system which is chargeable to cooling system(F) and necessary for interfacing the:

$$ACA = CE + CI + CS + CA + F \quad \text{...109}$$

6.2 The Present worth Cost

The present worth value of a cash flow over time is its value today. The total present worth value for any analysis is determined by summing the present worth value, both future single payment (i.e., replacement cost) items and series of equal future payments (i.e., annual operating cost). The cost or the value of money is a function of the available interest rate and inflation rate, *i*. The present value Poof a series of uniform end-of-period payments Ao(the annual operating cost) and the present worth value of a single payment in the future afterperiodRe,*n*, P(future replacement cost). The present worth method is normally used to evaluate and compare the life-cycle costs of systems[63].The uniform series present worth factor and the single payment present worth factor can be expressed as follow.

$$P_O = A_o\left(P/_A, i\%, n\right) \quad \ldots\ldots 110$$

$$P_{\mathrm{Re}} = F_{\mathrm{Re}}\left(P/_F, i\%, n\right) \quad \ldots\ldots 111$$

$$P_O = p_1 = A_o\left[\frac{(1+i)^n - 1}{i(1+i)^n}\right] \quad \ldots\ldots 112$$

$$P_{\mathrm{Re}} = F_{\mathrm{Re}}\left[\frac{1}{(1+i)^n}\right] \quad \ldots\ldots 113$$

In economic analysis of solar absorption refrigeration system is to analysis the life cycle cost, where the project is feasible or not. In general the feasibility of absorption refrigeration is depending on the type of heat input to use in system. The initial capital cost of the project is 224710 Birr (8642.7USD) and the detail of capital is shown in table2.

Component	Estimated specific cost
Solar collector array	92400 Birr
Heat storage unit	10000 Birr
Cooling storage unit	5000 Birr
Auxiliary components	50000 birr
Controlling units	30000 Birr
Installation	5000 Birr
Transport and logistics	3000 Birr

1 USD = 26Birr

Table 2: specific cost the project

When calculate the life cycle cost of the project, take the operation period 8 hours/ day, the life of time period, n=20 with the initial capitalcost of the project, CA= 224710 Birr (8642.7USD)and annual running cost, CR= 3% of CA, maintenance cost, CM= 1%of CA, replace cost, CRE= 7%of CA, salvage cost, CS= 3% and the inflation rate, i= 6%.

The specific life cycle cost is determined as:

$$SLCCA = \frac{LCCA}{\dot{Q}_{cold}}$$

$$LCCA = AC_A + AC_M + AC_R + BC_{RE} - BC_S$$

Where: $$A = \left[\frac{(1+i)^n - 1}{i(1+i)^n}\right] = 10.63$$

$$B = \left[\frac{1}{(1+i)^n}\right] = 0.2585$$

$LCCA = 2485373.373\,\text{Birr}$

$LCCA = 95636.05\,\text{USD}$

The specific life cycle cost of the project is:

In one hour $\dot{Q}cold = 5.36kW$

In one day $\dot{Q}cold = 42.88kW$

In 20 year $\dot{Q}cold = 313024kW$

$$SL\dot{C}C\,A = \frac{LCCA}{\dot{Q}_{cold}} = \frac{245373.3}{313024} = 0.785Birr$$

$$SL\dot{C}C\,A = \frac{LCCA}{\dot{Q}_{cold}} = \frac{95636.05}{313024} = 0.3Birr$$

The life cycle cost of vapour compression refrigeration system used in chuye milk and milk processing is:

$SLCCA = 113.135Birr$

The essence of the feasibility of solar absorption cooling systems is its technical and economically competitive position with respect to other space cooling system alternatives, including in particular conventional vapor-compression cooling units.

CHAPTER SEVEN

7 RESULT AND DISCUSSION

7.1 Result

Solar absorption refrigeration is vital important to improve energy shortage of the pastoralist area using local energy resource such as solar energy. Performance of solar absorption refrigeration systems is critically dependent on the chemical, thermodynamic properties of the working fluid and type of energy use. The usage is depends on the location, this paper focus Samara, Afar region, Ethiopia. Samara is one hot desert located in Ethiopia with average annual solar radiation of 6.4 Kwh/days and average ambient temperature of 38^{o}c. modeling of solar absorption is done using MATLAB software. The output of MATLAB represented in graph for hourly glass temperature, useful output heat arrive to fluid temperature, hourly milkpasteurized, water temperature in storage tank and hourly overall coefficient of performance (COP) of solar absorption refrigeration system in Samara site.

7.1.1 Hourly useful heat absorbed by absorber plate

The Useful heat absorbed by absorber plate depends on solar radiation and type of plate that can absorb solar radiation, this solar radiation varies from season to season and time to time. The maximum useful heat arrived is in December and the minimum one is in July.

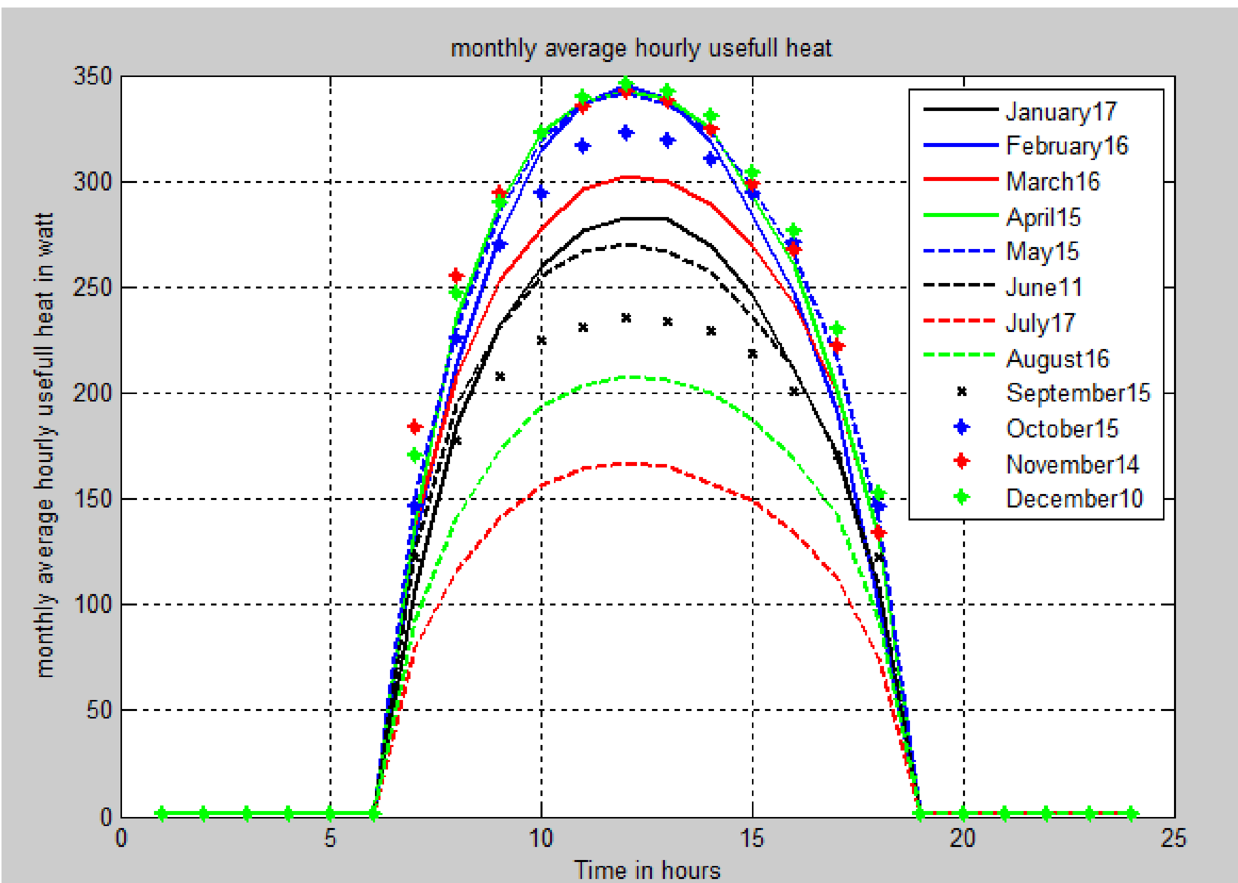

Figure 20:Hourly useful heat absorbed byabsorber plate.

7.1.2 Water temperature in storage tank

Hot water in storage is used as an input to pasteurized and cooling milk, when hot water in storage tank is increased, increases the coefficient of performance of the milk and reduces the time required to pasteurize the milk, the high water temperature is December and low temperature July .

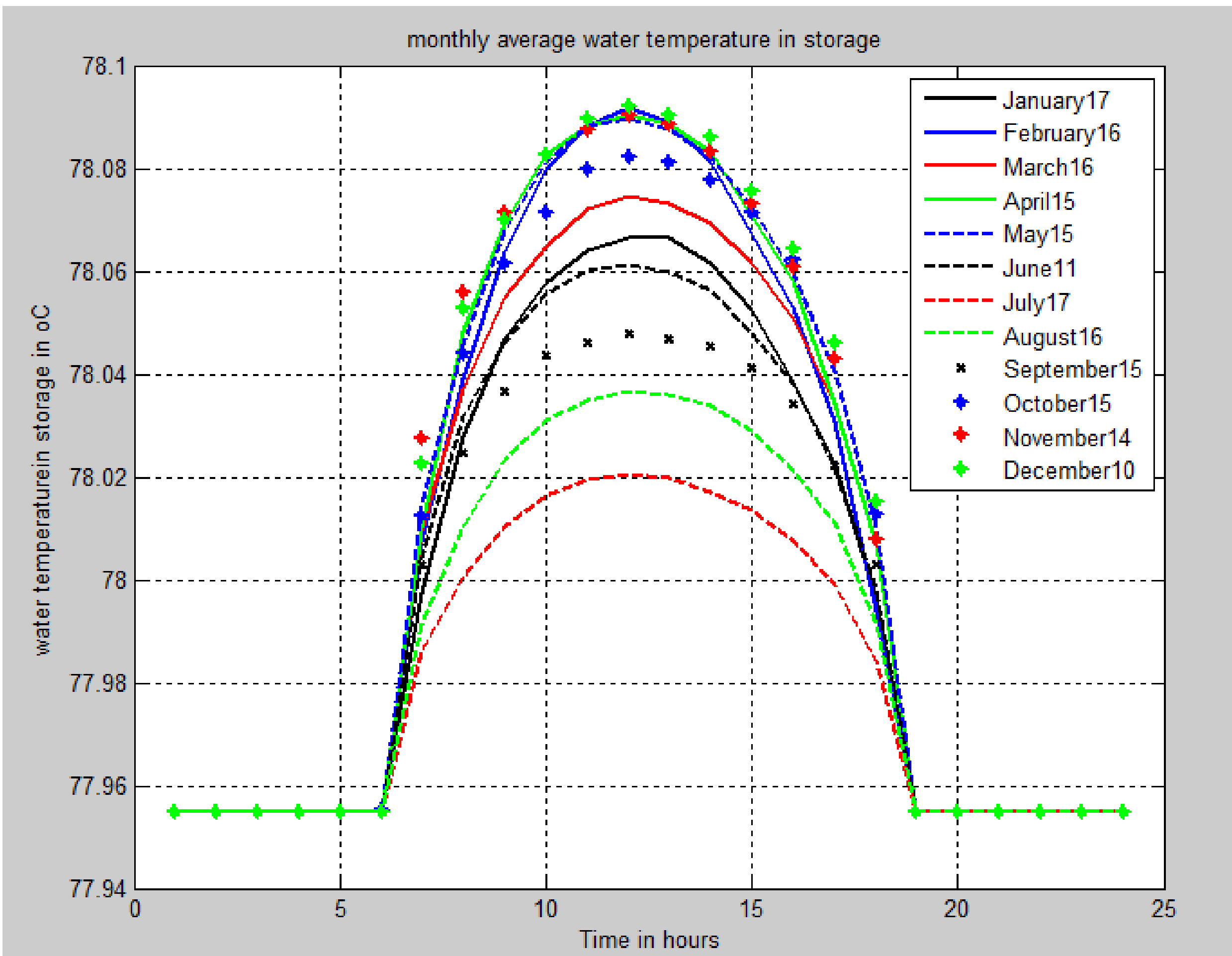

Figure 21: Water temperature in storage tank.

7.1.3 Hourly pasteurized milk

Temperature of pasteurized milk depends hot water temperature in storage tank and ambient temperature of the location, when the temperature of hot water temperature increase, the temperature of pasteurized milk is also increased; hourly temperature varies from hour to hour and season to season. Maximum pasteurized milk temperature is in June and minimum pasteurized milk temperature is in January.

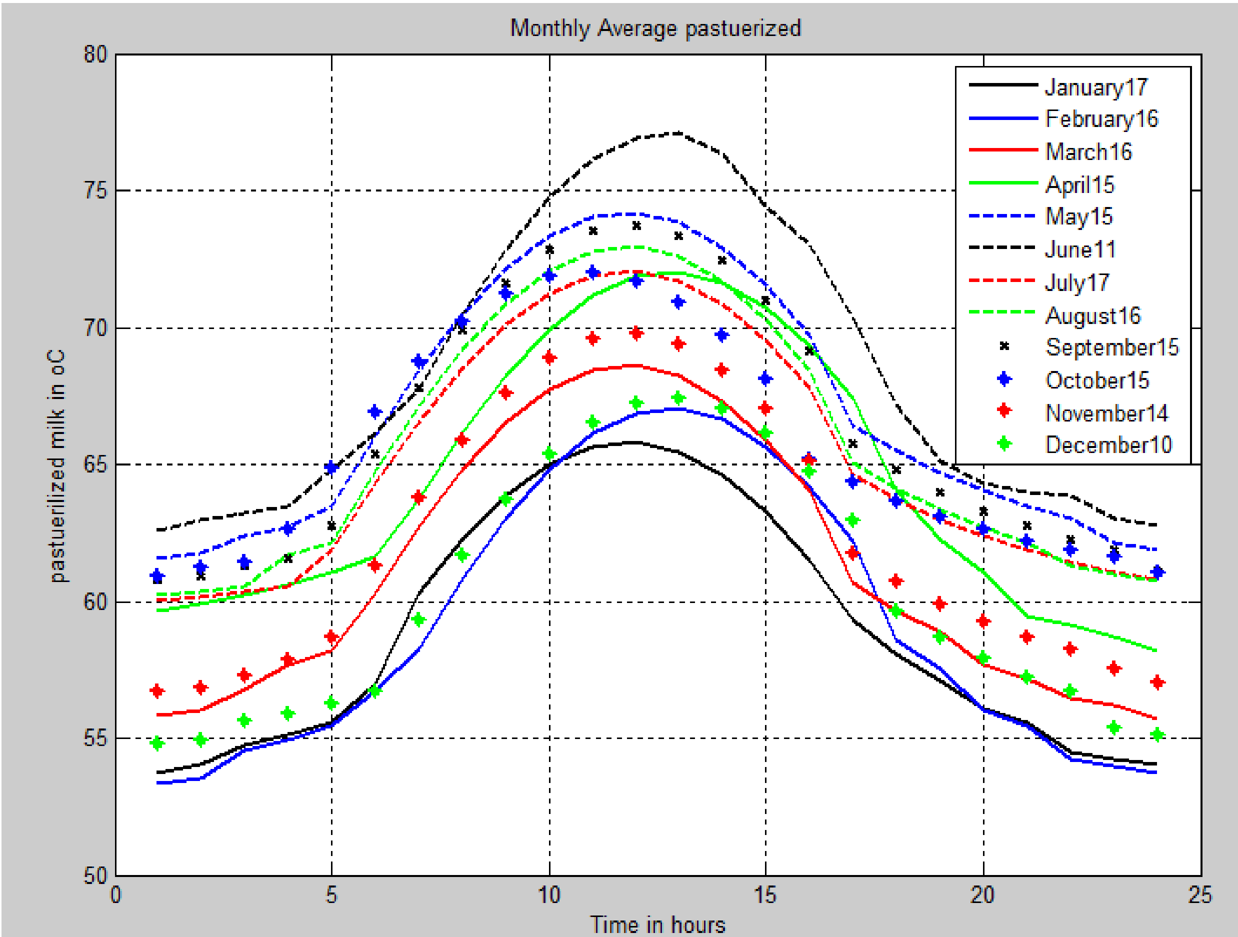

Figure 22: Hourly temperature of pasteurized milk.

7.1.4 Hourly coefficient of performance (COP)

coefficient of performance of solar absorption refrigeration system is depends on heat input in generator and the required cooling effect in evaporator .the heat input in generator is depend on solar radiation and ambient temperature . The COP Varies season to season and from hour to hour. The maximum coefficient of performance is in June and minimum coefficientof performance is in January.

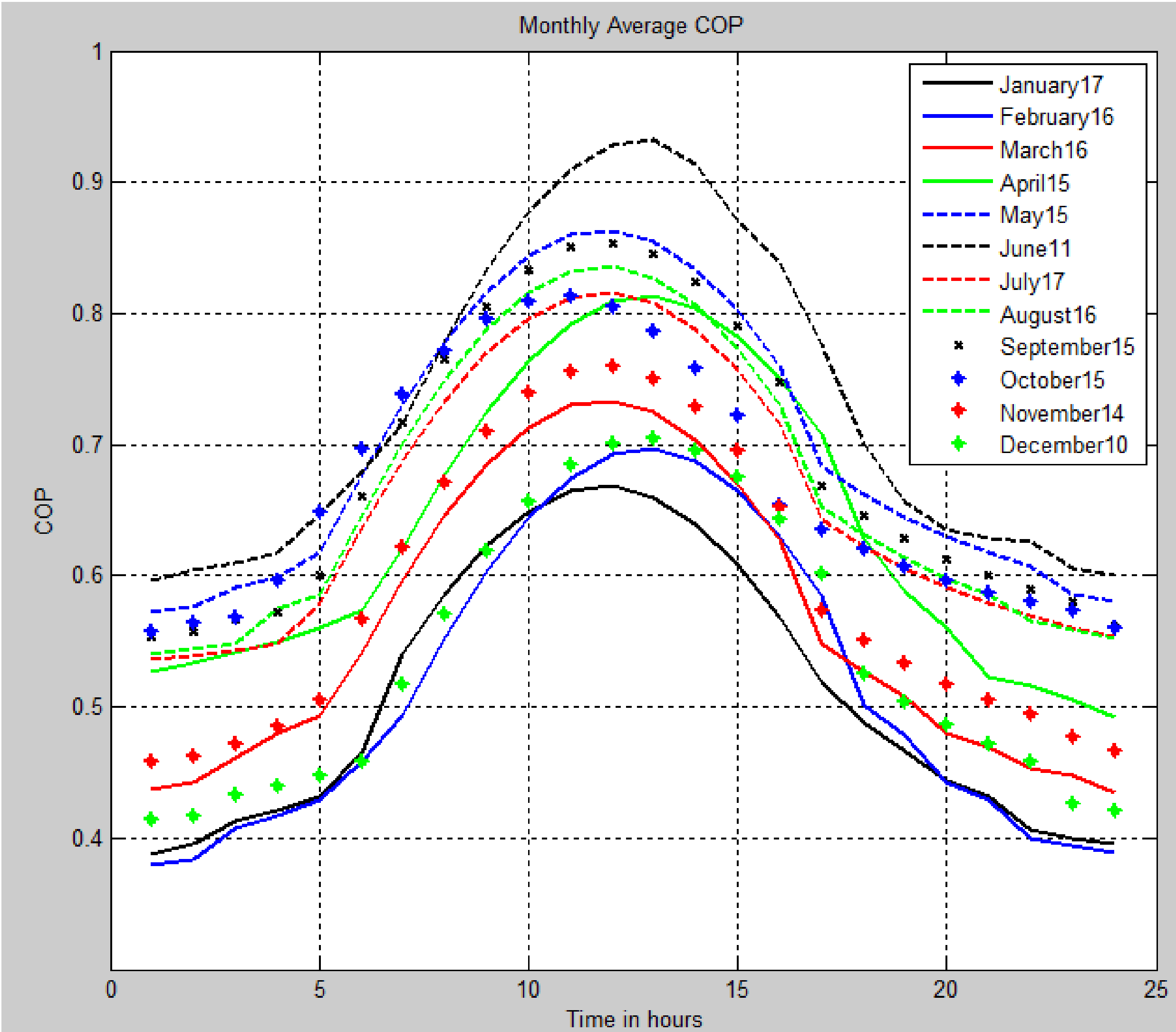

Figure 23: Hourly coefficient of performance (COP) of solar absorption refrigeration system.

7.1.5 Hourly Milk cooling Temperature

Milk cooling temperature is depends mainly Ambient temperature and evaporation temperature. When ambient temperature and solar radiation is increased, the cooling load of the milk temperature is increased;i.e the lower temperature of milk is gained.The required milk temperature is gained in noon time or mid-days.

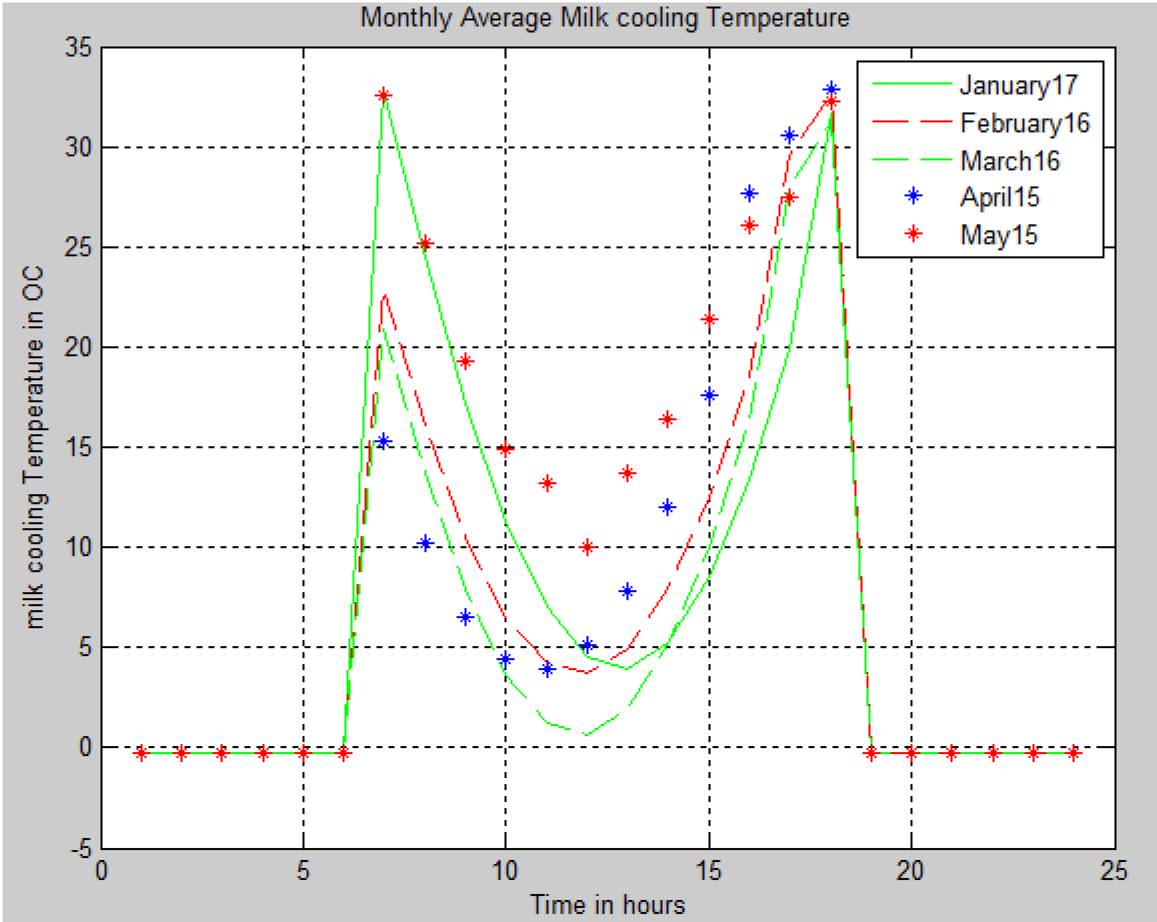

Figure 24: Hourly Milk cooling Temperature of solar absorption refrigerationsystem

7.2 Discussion

7.2.1 Factors affect milk pasteurization and cooling system

From the results shown in figures 23 and 24 the temperatureof milk pasteurized and coefficient of performance varies from hours to hour'sdueto variation of solar radiation and ambient temperature from season to season,high temperature of pasteurized always gained at mid-day, because of at noon time there is high solar radiation and also type of refrigerant used also affect the coefficient of performance. In result, the hourly data used is converted analytical, dueto lack of direct reading hourly solar radiation of the location is unavailable, while convert sunshine hour to solar radiation hourly there is error; such an error would affect the result.

7.2.1.1 Ambient temperature

The ambient temperature is vary from hour to hour and season to season .the peak daily temperature occurs at noon, as air keeps net absorbing heat even and the minimum daily temperature occurs substantially after mid night. Indeed occurring early morning in the hour around down, since heat is lost all night long.

7.2.1.2 Solar Radiation

Solar radiation directly heats the ground because of the land are ability to store heat. Land absorbs certain amount of heat before its temperature is rise, at noon when its reach peak height and is directly over head. After noon, intensity of incoming solar radiation continuously decline.Therefore solar radiation varies from hour to hour and season to season dueto sun direction and the land doesn't immediately warm.

7.2.1.3 Refrigerant

In absorption refrigeration system, there are some 40 refrigerant compounds and 200 absorbent compounds available. However, the most common working fluids are Water/NH3 and LiBr/water. In ammonia-water absorption refrigeration system, ammonia is used as the refrigerant, which is easily and cheaply available, which is necessary for efficient performance of the system. It can be used for low temperature applications. In the absorption refrigeration system there are no (or very few) leakages of the refrigerant and the refrigerant itself is very cheap. Thus there are almost zero refrigerant recharging costs. In case of the vapor compression systems there are lots of leakages of the refrigerant thus regular recharge of the refrigerant are required which is very expensive.Most of the halocarbon refrigerants used in the compression refrigeration system produces greenhouse effect. In the absorption refrigeration system no refrigerant produces the greenhouse effect, so their use won't be stopped in future. Finally NH_3-H_2O is environmental friendly and low cost.

Therefore, the result in figure pasteurized the milk and cools the milk using solar absorption refrigeration system is more advantage than other type of refrigeration system in case of cost and greenhouse effect. Thecoefficient of performance gained acceptable as compared to other paper done experimentally done.

CHAPTER EIGHT

8 CONCLUSION AND RECOMMENDATION

8.1 Conclusion

Pasteurized and refrigerated milk is important for all human being; unpasteurizing milk would affect human health. 90% of dairying process companies in Ethiopia located around Addis Ababa, to get market demand and full infrastructure to process a milk, those who far apart from central area; can't get such as access, most in pastoralists area, even if they have more resource demand's. To Pasteurized and refrigerated milk electrical energy or thermal energy is required. In pastoralists' area of Afar region where grid electricity is unavailable, this area is wealth in solar energy. In this area using solar energy and absorption refrigeration system to pasteurize and cool milk.The coefficient performance of the solar absorption refrigeration system varies from 0.45 to 0.94 and milk temperature is cooled from 32 ^{o}c up to 4 ^{o}c using MATLAB simulation. The pasteurizing and coolingmilk using solar energy is dependent on factors such as solar radiation, ambient temperature and type of refrigerant use.

Therefore temperature of milk pasteurized and coefficient of performance of solar absorption refrigeration system is varies from hour to hour and season to season due to variation of solar radiation and ambient temperature of the they location.

8.2 Recommendation

In this thesis the mathematical model, MATLAB simulation and economic analysis of the system is done, the future work includes the following things:

- Manufacturing of small-scale solar absorption refrigeration system using NH_3-H_20 asrefrigerant isnecessary to make it actually solve the problem of rural area societies.
- Experimental analysis based on prototype manufactured and lab test is important to improve the coefficient of performance of overall system.
- Designing and experimental test of small scale absorption refrigeration system using pv-t system and compare with solar thermal solar absorption refrigeration is necessary to select which is important one in case of energy efficient, coefficient of performance and cost of overall system.

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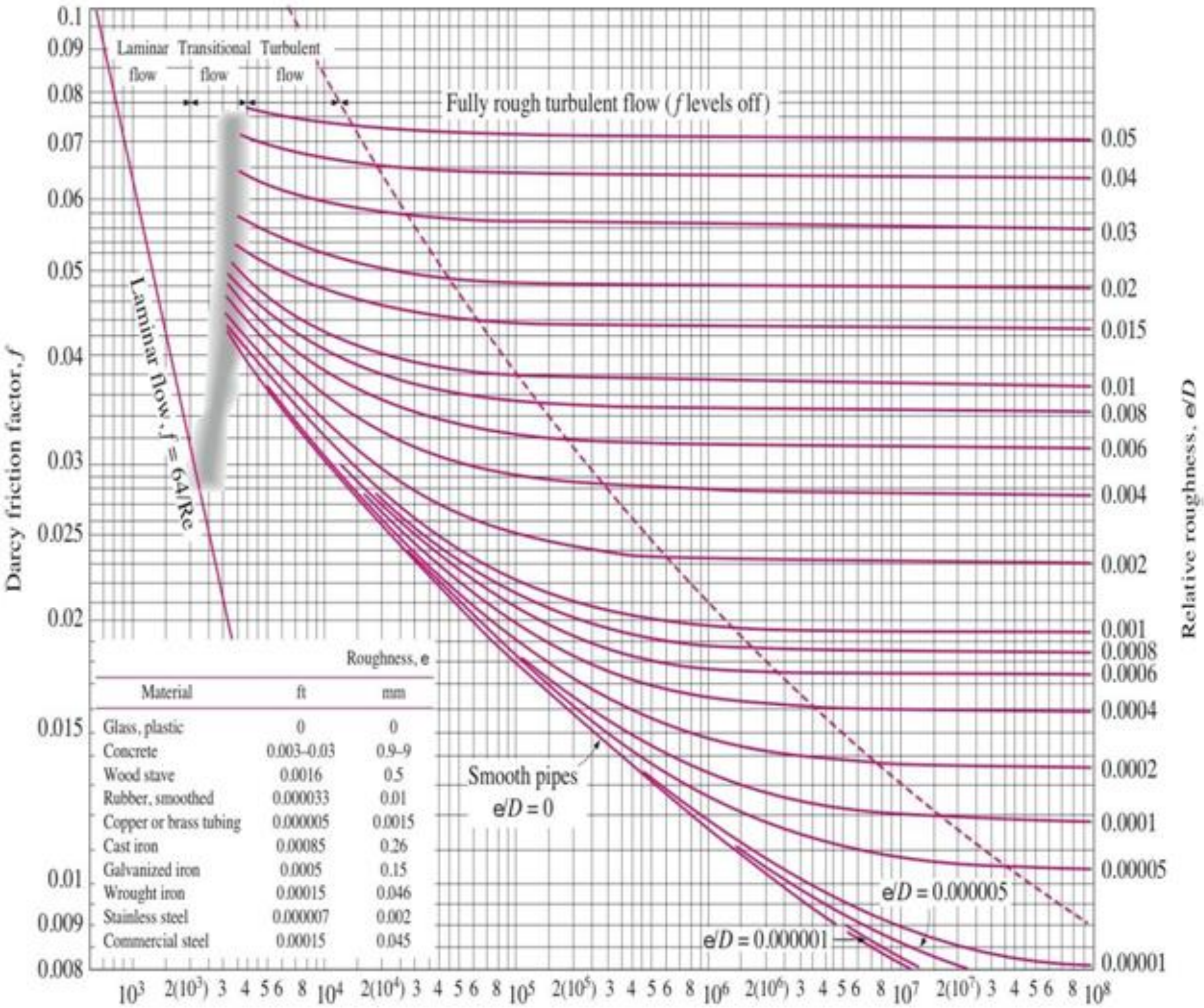
APPENDIX A

Moody chart



APPENDIX B

```
% Hourly Temperature variation of the useful heat, collector, hot water, pasteurized milk and coefficient of performance using the Global Radiation, Diffuse Radiation and ambient temperature variation data
Clear all
clc
Ac=5.89;              %collector area
Aab=7.8;              %collector aperture area
As=0.3995;            %heat transfer area of storage tank
eg=0.88;              %Emissivity of outer glass
ep =0.10;             %Emissivity of plate
Ks=0.045;             %thermal conductivity of insulation
t=0.05;               %Thickness of Insulation of Tank
KAl=0.202;            %thermal conductivity of aluminum fin
ag =0.94 ;            % absorption coefficient of the selective
%coating on the inner glass tube (absorber)
tg =0.96;             %transmissivity of glass
cp = 4187;            %specific heat of water
p=998;                %density of water
ca= 377;              %heat capacity of absorber plate / copper
cg= 830;              %heat capacity of outer glass
Vw=2.6;               %wind speed
m=499;                %mass of water in collector tube
Aabs = 5.96;          %area of collector aperture (absorber
s =5.67e-8;           %Stefan-Boltzmann Constant
kw =0.625;            %Thermal Conductivity of Water
Vs=0.5;               %Volume of Tank
tstep=60;             %time step
Rhow=998;             %Density of Water
dwc = 0.005787;       %daily water consumption m^3/ second
Tai = 37;             %average ambient temperature
Tgi=37;               %initial outer glass temperature
```

```
Tpi= 84.5;                %initial plate temperature
Tig=Tpi;                  %inner glass temperature
Tws= 80;                  %temperature of storage tank supply water
Ti= 30;                   %water inlet temperature
Tfi =85;                  %initial water exit from evacuated tube
To=85;                    %Temperature of Outlet Water
Tso= To;                  %initial instantaneous Temperature of Tanker
dgo =0.054;               %diameter of the outer glass tube
dgi = 0.047;              %diameter of the inner glass tube
dAlo=0.01525;             %diameter of the outer aluminum fin
dAli=0.015;               %diameter of the inner aluminum fin
dcuo=0.01;                %diameter of the outer copper tube
dcui=0.0095;              %diameter of the inner copper tube
L=1.8;                    %overall length of absorber tube and aluminum fin
Aog=7.2036;               %area of outer glass tube
Aig =5.9616;              %area of inner glass tube
UA=0.989;                 %heat loss coefficient and area of storage tank
mdot= 0.1389;             %mass flow rate of water
phi = 3.14;
cpl=4.187;
phii=11.74*pi/180;        %latitude
beta=25.74 *pi/180;       %Collector  slope
vs=0.5;                   %volume of storage tank
mr = 1.118;               %mass of refrigerant
mss = 2.118;              %mass flow rate of strong solution of refrigerant
mws = 1.118;              %mass of weak solution of refrigerant
Te=4;                     %Evaporator temperature
T3=55;                    %strong refrigerant temperature exitheat exchanger

T4=68;                    %weak refrigerant temperature inter heat exchanger
cpm=3.93;
```

```
mdotm=0.13861;
cpll=4.182;
cmin=cpm*mdotm;
cmax=mdot*cpll;
Tmilkin=32;
Tmilkin =Trf;
e=0.4;

n = xlsread ('HSR.xlsx');

forj=1:365
 data (:,:,j)=n(j+(23*(j-1)):j*24,:);   %Data given for the global radiation, diffuse radiation and
                                         %ambient temperature
end

for n=1:365
decl(n)=(pi/180)*(23.45*sin((2*pi/365)*(284+n)));   % Calculation of the declination angle )))
end

for  ii=1:24
    omega(ii)=(ii-12)*pi/12;          % Calculation of the hour angle for each hour of the day. )))
end


% This approximate the plate, glass, and exit temperature for Jan.01 at 1:00AM.

for i=1
     GRR=data(:,1,i);
     DRR=data(:,2,i);
 ATT=data(:,3,i);
for j=1
```

```
  GR=GRR(1);
        DR=DRR(1);
        AT=ATT(1);
%the ratio of beam radiation on the tilted surface
Rb=((cos(phii-beta))*cos(decl(i))*cos(omega(j))+sin(decl(i))*(sin(phii-
beta)))/(cos(phii)*cos(decl(i))*cos(omega(j))+(sin(decl(i))*sin(phii)));
It=Rb*(GR-DR)+(DR*0.5*(1+cos(beta)))+(0.6*Rhow*((1-cos(beta))*GR));
Tsky = AT- 6;                                    %sky temperature of the selected site
hp_g_rad = (ep-eg)*s*((Tpi^4-Tgi^4)/(Tpi-Tgi)); %radiation transfer coefficient between
%the absorber plate and glass
hp_g = hp_g_rad;
hw=2.8+3.8*Vw;                                   %convection heat transfer coefficient
  %around the outer glass tube
hg_a_rad=eg*s*(((Tgi). ^4)-((Tsky).^4))/(Tgi-Tsky);    %radiation heat transfer coefficient
                                                       %between the glass and ambient
hg_a = hg_a_rad + hw;   %overall the heat transfer coefficient between the glass and ambient
hf_p= 350;              %heat transfer coefficient between fluid and absorber tube
hst=ks/t;               %heat transfer coefficient of storage tank A=0.002826;
%the required pipe area
hp=350;                                  %heat transfer coefficient of pipe
eff=0.8;
fw=2*eff-1;
Tc=(Tfi+Ti)/2;
% glass temperature in evacuated tube solar collector attstep
Tg1= (tstep/cg)*(eg*s*(Tsky. ^4-Tgi. ^4) +hg_a*(AT-Tgi) + eg*s*(Tpi. ^4-Tgi.^4))+ Tgi;
% absorber temperature in evacuated tube solar collector at tstep
   Tp1= (tstep/cg)*(tg*ag*It+eg*s*(Tg1.^4-Tpi^4) + hf_p*(Tfi-Tpi))+ Tpi;
%fluid temperature in evacuated tube solar collector at tstep
Tf = (tstep/(m*cp))*((Ac*hf_p*(fw*Tp1-Tc))- mdot*cp*(Tfi-Ti))+Tfi;
%the instantaneous temperature of water in the storage tank at tstep
   s1=1-(As*hst*tstep/(m*cp));
```

```
   s2=0.2*(mdot*tstep/(m*cp));
 s3=As.*hst*tstep/(m*cp);
Tsi=s1*Tf+s3*Tai+s2*Tfi;
%coefficient of performance
Qgi = mr*cp*Tsi+mws*cp*T4 - mss*cp*T3;
Qpl = hp*A*(Tsi-ATT);
Qg = Qgi-Qpl;
Qe = mr*cp*(ATT - Te);
cop =(Qe/Qg);
Qmax=cmin*(Tsi-Tmilkin);
Qout=e*Qmax;
 Qin=mdot*cpll*(Tsi-AT);
Tout= (Qout/cmax) +AT;
QP=Qin-Qout;
Tm= cmax/cmin*(Tsi-Tmilkin);
Tn=Tout-AT;
Tpp=Tm-Tn+AT;
end
end
Tpo=(Tp1+Tpi)/2;
Tgo=(Tg1+Tgi)/2;
Tfo=(Tf+To)/2;
Tsoo =(Tso+Tsi)/2;

fori=1:365
    GRR=data (:, 1,i);
    DRR=data (: , 2,i);
    ATT=data (:,3,i);
for j=1:24
        GR=GRR (j);
        DR=DRR (j);
```

```
    AT=ATT (j);
%the ratio of beam radiation on the tilted surface
Rb=((cos(phii-beta))*cos(decl(i))*cos(omega(j))+sin(decl(i))*(sin(phii-
beta)))/(cos(phii)*cos(decl(i))*cos(omega(j))+(sin(decl(i))*sin(phii)));
It=Rb*(GR-DR)+(DR*0.5*(1+cos (beta))) +(0.6*Rhow*((1-cos(beta))*GR))
Forjj=1:60
Tsky= AT-6;            %sky temperature of the selected site
hw=2.8+3.8*Vw;          %convection heat transfer coefficient around the outer glass tube
hp_g_rad1 = (eg-ep)*s*((Tpo.^4-Tgo.^4)/(Tpo-Tgo));
hg_a_rad1 = eg*s*(((Tgo).^4)-((Tsky).^4))./(Tgo-Tsky); %radiation heat transfer coefficient
                                                        %between the glass and ambient
hg_a1 = hg_a_rad + hw;
hf_p= 350;
Tg= (tstep/cg)*(eg*s*(Tsky.^4-Tgo.^4)+ hg_a1*(AT-Tgo) + eg*s*(Tpo.^4-Tgo.^4))+ Tgo;
%glass temperature in evacuated tube solar collector  at tstep
Tp= (tstep/cg)*((tg*ag*It)+eg*s*(Tg. ^4-Tpo. ^4) + hf_p*(Tfo-Tpo)) +Tpo;
%absorber temperature in evacuated tube solar collector at change oftstep
hst=ks./t;                          %heat transfer coefficient of storage tank
eff=0.8;
fw=2*eff-1;
hpp=0.35;
Tev=-1.5;
 Tc1=(Tfo+Ti)/2;
A=0.002826;                         %the required pipe area
hp= 350;                        %heat transfer coefficient of pipe
UL=1./((1./hg_a_rad1+hw)+(1./hp_g_rad1));    %overall heat loss coefficient of the tube
Qabs= ag*tg*It*Aabs; %total solar radiationabsorbed by absorber plate
Qthermal_loss = UL.*Aig.*(Tp-It);             %heat lost by tubes
Qu=Qabs-Qthermal_loss;                        % useful output of evacuated tube solar collector
%theinstantaneous temperature of water in the storage tank at change of tstep
Tf1 =(tstep/(m*cp))*((Ac*hf_p*(fw*Tp-Tc1))-Qu)+Tfo;
```

```
for hwhf=[0.1,0.12,0.12,0,0,0.1,0.12,0.1,0,0,0,0,0.1,0.12,0.1,0,0,0,0,0,0,0,0,0,0.12];

Qload=Rhow*cp*dwc*hwhf*(Tsoo-Tws);
Q = mdot*cp*(Tf1-Ti);
if Tf1 < 348;
mdot=0.087;
else
mdot=0.087:0.08:0.1;
end
Qloss =As*hst*(Tsoo-AT);                  %storage tank heat loss
Q12=(Q-Qload-Qloss);
Ts =Tf1+(tstep./(m*cp*Q12));
        Qloss1= hst*As*(Tws-AT);
        Q11=(Q-Qload-Qloss);
        Ts1=Tf1+(tstep./(m*cp*Q11));
%coefficient of performance
Qgi = mr*cp*Ts1+mws*cp*T4 - mss*cp*T3;
Qpl = hp*A*(Ts1-AT);
Qg = Qgi-Qpl;
Qe = mr*cp*(AT - Te);
cop =(Qe/Qg);
%Temperature of milk pasteurized in storage tank
Qmax=cmin*(Ts1-Tmilkin);
Qout=e*Qmax;
 Qin=mdot*cpll*(Ts1-Tmilkin);
 Tout=(Qout/cmax)+AT;
QP=Qin-Qout;
Tm= cmax/cmin*(Ts1-Tmilkin);
Tn=Tout-AT;
Tpp=Tm-Tn+AT;
```

```
%Temperature of milk cooling temperature
mdotmm=0.1386;
    cpa=2.190;
    Qee=mr*cpa*(Trf-(AT-3));
Umm=1.3;
     Qmloss=Umm*Am*(Tmilkin-AT);
    Qinlet=mdotm*cpa*Tmilkin;
     Qmloss=hpp*At*(Tmilkin-AT);
Qmout = Qee-Qinlet-Qmloss;
     mm=mdotm*cpm;
     mmm=Qmout/mm;
     Tmilkout= Tmilkin+mmm;



end
end
% useful heat, glass temperature, water  temperature exit the collector, water temperature in storage tank, Temperature of milk pasteurized and coefficient ofperformance for each day of the yearare stored as;
Qum (i,j) = Qu;
Tgm(i,j) = Tg;
Tfm(i,j) = Tf1;
Tsm(i,j) = Ts1;
Tppm(i,j)= Tpp;
copm(i,j)= cop;
Tmmm(i,j)=Tmilkout;

end
end
```

```
 QumJ17=Qum(17,:);QumF16=Qum(47,:); QumM16= Qum(75,:);QumA15=Qum(105,:);
QumM15=Qum(135,:);
  QumJ11=Qum(162,:);QumJu17=Qum(198,:); QumA16=Qum(228,:);
QumS15=Qum(258,:);QumOct15=Qum(288,:);QumN14=Qum(318,:);QumD10=Qum(344,:);
jj=1:24;
% Plots mean monthly useful heat arrived to absorber plate
 plot(jj,QumJ17,'k-',jj,QumF16,'b-',jj,QumM16,'r-',jj,QumA15,'g-',jj,QumM15,'b--',
 jj,QumJ11,'k--',jj,QumJu17,'r--',jj,QumA16,'g--',jj,QumS15,'k x',jj,QumOct15,'b*',
jj,QumN14,'r*',jj,QumD10,'g*','linewidth',2);
xlabel (' Time in hours')
ylabel (' monthly average hourly useful heat in watt ')
Title ('monthly average hourly useful heat ')
 legend('January17','February16','March16','April15','May15','June11','July17',
'August16','September15','October15','November14','December10')
 grid on
pause
clf

% Plots mean monthly glass temperature
    TgmJ17=Tgm(17,:);TgmF16=Tgm(47,:);TgmM16= Tgm(75,:);TgmA15=Tgm(105,:);
   TgmM15=Tgm(135,:);TgmJ11=Tgm(162,:);TgmJu17=Tgm(198,:);TgmA16=Tgm(228,:);
   TgmS15=Tgm(258,:);TgmOct15=Tgm(288,:);TgmN14=Tgm(318,:);TgmD10=Tgm(344,:);

jj=1:24;


plot(jj,TgmJ17,'k-',jj,TgmF16,'b-',jj,TgmM16,'r-',jj,TgmA15,'g-',jj,TgmM15,'b--',
 jj,TgmJ11,'k--',jj,TgmJu17,'r--',jj,TgmA16,'g--',jj,TgmS15,'k x',jj,TgmOct15,'b*',
jj,TgmN14,'r*',jj,TgmD10,'g*','linewidth',2);
xlabel(' Time in hours')
ylabel(' glass temperature in oc sent ')
```

```
Title ('monthly average glass temperature ')
 legend('January17','February16','March16','April15','May15','June11','July17',
'August16','September15','October15','November14','December10')
 grid on
 pause
clf

% Plots mean monthly water temperature exit from collector
    TfmJ17=Tfm(17,:);TfmF16=Tfm(47,:);TfmM16= Tfm(75,:);TfmA15=Tfm(105,:);
   TfmM15=Tfm(135,:);TfmJ11=Tfm(162,:);TfmJu17=Tfm(198,:);TfmA16=Tfm(228,:);
   TfmS15=Tfm(258,:);TfmOct15=Tfm(288,:);TfmN14=Tfm(318,:);TfmD10=Tfm(344,:);

jj=1:24;


      plot(jj,TfmJ17,'k-',jj,TfmF16,'b-',jj,TfmM16,'r-',jj,TfmA15,'g-',jj,TfmM15,'b--',
    jj,TfmJ11,'k--',jj,TfmJu17,'r--',jj,TfmA16,'g--',jj,TfmS15,'k x',jj,TfmOct15,'b*',
jj,TfmN14,'r*',jj,TfmD10,'g*','linewidth',2);
xlabel(' Time in hours')
ylabel(' water temperature in oc ')
    title('monthly average water temperature ')
 legend('January17','February16','March16','April15','May15','June11','July17',
'August16','September15','October15','November14','December10')
 grid on
 pause
clf
% Plots mean monthly Temperature of water in storage tank
    TsmJ17=Tsm(17,:);TsmF16=Tsm(47,:);TsmM16= Tsm(75,:);TsmA15=Tsm(105,:);
   TsmM15=Tsm(135,:);TsmJ11=Tsm(162,:);TsmJu17=Tsm(198,:);TsmA16=Tsm(228,:);
   TsmS15=Tsm(258,:);TsmOct15=Tsm(288,:);TsmN14=Tsm(318,:);TsmD10=Tsm(344,:);

jj=1:24;
```

```
   plot(jj,TsmJ17,'k-',jj,TsmF16,'b-',jj,TsmM16,'r-',jj,TsmA15,'g-',jj,TsmM15,'b--',
  jj,TsmJ11,'k--',jj,TsmJu17,'r--',jj,TsmA16,'g--',jj,TsmS15,'k x',jj,TsmOct15,'b*',
jj,TsmN14,'r*',jj,TsmD10,'g*','linewidth',2);
xlabel(' Time in hours')
ylabel(' water temperaturein storage in oc ')
Title ('monthly average water temperature in storage ')
 legend('January17','February16','March16','April15','May15','June11','July17',.....
'August16','September15','October15','November14','December10')
 grid on
  pause
clf

% Plots mean monthly milk pasteurized;
  TppmJ17=Tppm(17,:);TppmF16=Tppm(47,:);TppmM16=
Tppm(75,:);TppmA15=Tppm(105,:);

TppmM15=Tppm(135,:);TppmJ11=Tppm(162,:);TppmJu17=Tppm(198,:);TppmA16=Tppm(22
8,:);

TppmS15=Tppm(258,:);TppmOct15=Tppm(288,:);TppmN14=Tppm(318,:);TppmD10=Tppm(3
44,:);...

jj=1:24;

 plot(jj,TppmJ17,'k-',jj,TppmF16,'b-',jj,TppmM16,'r-',jj,TppmA15,'g-',jj,TppmM15,'b--',
  jj,TppmJ11,'k--',jj,TppmJu17,'r--',jj,TppmA16,'g--',jj,TppmS15,'k x',jj,TppmOct15,'b*',
jj,TppmN14,'r*',jj,TppmD10,'g*','linewidth',2);
xlabel (' Time in hours')
ylabel ('pastuerilized milk in oc')
Title ('Monthly Average pastuerized')
```

```
 legend('January17','February16','March16','April15','May15','June11','July17',.....
'August16','September15','October15','November14','December10')
 grid on
 pause
clf
% Plots mean monthly coefficient of performance of overall system
   copmJ17=copm(17,:);copmF16=copm(47,:);copmM16= copm(75,:);copmA15=copm(105,:);
copmM15=copm(135,:);copmJ11=copm(162,:);copmJu17=copm(198,:);copmA16=copm(228,:);
copmS15=copm(258,:);copmOct15=copm(288,:);copmN14=copm(318,:);copmD10=copm(344,:
);

jj=1:24;

  plot(jj,copmJ17,'k-',jj,copmF16,'b-',jj,copmM16,'r-',jj,copmA15,'g-',jj,copmM15,'b--',
   jj,copmJ11,'k--',jj,copmJu17,'r--',jj,copmA16,'g--',jj,copmS15,'k x',jj,copmOct15,'b*',
jj,copmN14,'r*',jj,copmD10,'g*','linewidth',2);
xlabel(' Time in hours')
ylabel('COP')
   Title ('Monthly Average COP')
 legend('January17','February16','March16','April15','May15','June11','July17',
'August16','September15','October15','November14','December10')
 grid on
pause
 clf

% Plots mean monthly milk cooling Temperature
TmmmJ17=Tmmm(17,:);TmmmF16=Tmmm(47,:);TmmmM16=
Tmmm(75,:);TmmmA15=Tmmm(105,:);...


TmmmM15=Tmmm(135,:);TmmmJ11=Tmmm(162,:);TmmmJu17=Tmmm(198,:);TmmmA16=
Tmmm(228,:);...
TmmmS15=Tmmm(258,:);TmmmOct15=Tmmm(288,:);TmmmN14=Tmmm(318,:);TmmmD10
=Tmmm(344,:);...
```

```
jj=1:24;


    plot(jj,TmmmJ17,'k-',jj,TmmmF16,'b-',jj,TmmmM16,'r-',jj,TmmmA15,'g-',jj,TmmmM15,'b--
',...
  jj,TmmmJu17,'r--',jj,TmmmA16,'g--',jj,TmmmS15,'k x',jj,TmmmOct15,'b*',...
jj,TmmmN14,'r*',jj,TmmmD10,'g*','linewidth',2);
  xlabel(' Time in hours')
  ylabel('milk cooling Temperature in OC')
  title('Monthly Average Milk cooling Temperature')
 legend('January17','February16','March16','April15','May15','June11','July17',.....
'August16','September15','October15','November14','December10')
 grid on
```