Addis Ababa Institute of Technology

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Post Graduate Program in Thermal Engineering Stream

COMPUTATIONAL AND EXPERIMENTAL ANALYSIS OF ICE THERMAL STORAGE WITH SOLAR REFRIGERATOR

A thesis Submitted for the Partial Fulfillment of Degree of master in Mechanical and Industrial Engineering (Thermal stream), in Addis Ababa University, Addis Ababa Institute of Technology (AAIT)

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CERTIFICATION

I, the undersigned, certify that I read and hear by recommend for the acceptance by Addis Ababa University institute of Technology, Mechanical Engineering (Thermal Engineering Stream) a thesis entitled "**COMPUTATIONAL AND EXPERIMENTAL ANALYSIS OF ICE THERMAL STORAGE WITH SOLAR REFRIGERATOR**." This certificate used as a partial fulfillment of the requirement of Masters of Science in Mechanical Engineering (Thermal Engineering)

Advisor:

Dr. Yilma Tadesse ________________________ ________________

Signature Date

DECLARATION

I, Kamil Nuredin, I declare that this thesis is the result of my own work and that all source or material used for this thesis have been duly acknowledged. This thesis is submitted in partial fulfillment of the requirement for Master's Degree in Mechanical Engineering (Thermal Engineering) at Addis Ababa University and to be made available at the university's Library under the role of the Library. I confidently declare that this thesis has not been submitted to any other institutions anywhere for the award of any academic degree, diploma, or certificate.

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Computational and Experimental Analysis of Ice Thermal Storage with Solar Refrigerator

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ABSTRACT

In rural areas of Ethiopia the grid electricity supply coverage is very low, which is about 2%. As result of this dairy product preservation before supply to local market is difficult especially for night time collected milk. However, utilization of direct coupled PV powered solar refrigerator incorporated with Ice thermal storage able to preserve the products until market. The use of Ice thermal storage can avoid the use of solar batteries. The study includes the design, manufacture the prototype and experimental testing and computational modeling of PV powered solar refrigerator.

As a case study Adama city Ethiopia is selected for the experimental test, the solar radiation of the site is analyzed. The minimum daily average solar radiation is determined at a month of August which is $4873.2\text{Wh}/m^2/day$. To preserve 5L milk, MT3.5DC compressor and 140W PV panel are selected. The operating point of compressor is obtained and the minimum solar radiation required for the solar refrigerator is $315\ \text{W}/m^2$.

Computational simulations using ANSYS-Fluent is conducted to determine the amount of ice produced as a thermal storage for night time preservation and the average temperature of water-ice during solidification process. The system produced nearly 1kg solid ice with a mean temperature of 271.78K . During night time which is the discharging process of the ice the milk got chilled from $310\ \text{K}$ to 274K and maintained this temperature for more than 12hours.

Then, the model is manufactured and tested for its performance. During the experimental test solar radiation was simulated virtually with DC power supply due to frequent intermittent nature of the radiation. From the experiment, it was observed that the average water-ice temperature was -12.5^{o}C. In the discharging process the milk temperature was reduced from 310K to 274K within 12 hours. Also, the milk temperature is reached to a range of $0°\text{C}$ to $4°\text{C}$ with in 4hours duration and satisfies the FAO standard. CFD simulation was validated by experimental test, with 6% maximum deviation.

Keywords: Solar refrigerator; Variable speed DC compressor; Computational fluid dynamics; Experimental testing; PV panel: Ice-thermal storage.

ACKNOWLEDGMENTS

First of all, I am very grateful to Allah, this study would not have been possible without his blessing and graces.

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ACRONYM

PV	Photo voltaic panel
G	Solar radiation
h	Specific heat of enthalpy
hconv	Convective heat transfer coefficient
L	Latent heat of fusion
K	thermal conductivity of a material
m	Mass
$\dot{m}$	Mass flow rate
P	Power
R	Resistance
$\dot{Q}$	Heat transfer rate
STC	Standard test condition
PCM	phase change material
V	Voltage
Nu	Nusselt number
Ns	Number photo voltaic cells in series
C_p	Specific heat at constant pressure
PSH	Peak sun hour
VDC	Variable direct current
$\dot{Q_g}$	Heat gain rate from the surrounding
PGF	$Power\ generating\ facor$
COP	Coefficient of performance of refrigerator
h_{iev}	Convective heat transfer coefficient inside surface of evaporator tube
U_{oev}	The global heat transfer coefficient between water-ice and the refrigerant fluid

CHAPTER ONE

1. Introduction

1.1 Background and Significant of Study

The depletion of fossil fuels, environmental concern and difficulty in expansion of the power grid to cover the rural areas is costly; due to that people are trying to use standalone energy supply. Solar panels are most common form of portable renewable energy that can be used for off grid application. It does not require a connection to the power grid; environmental friendly therefore solar energy is an attractive alternative solution to power the rural or remote areas.
Since Ethiopia is located near to equator, it has high potential for solar energy. It has been estimated 3000 hour of sunshine per year and solar energy exceeding 5 kW/m^2 per day[1]. From its population about 85 percent the population lives in rural area where only less than 2 percent of this population has an access to the electricity[2][3]. In 2018 the grid electricity coverage in the country was 20% and the off grid access rate 10% [4]. The total percentage of electricity coverage in the country is 45% of its population[5].

In the present day, solar panels have a wide variety of applications. Such as; water pumping for irrigation, household appliance from homes to shopping malls to office buildings, solar energy powers everything from lighting to automatic doors to wall sockets, telecommunication system remote mission etc.

Solar powered refrigerator is a cooling device that works fully with energy supplied by the sun. And mostly used in hot climates to keep or preserve vaccine medicine.
Solar-powered refrigerator is useful for many reasons:

- ✓ Areas with no access to the national power grid
- ✓ Areas with limited or intermittent power supply
- ✓ Locations with at least 3.5 kWh/m^2 of solar irradiation per day

Some solar powered refrigerators use combination of solar panels and lead batteries to store energy when the cloudy days and at night as shown Figure 1.1. The batteries work on electricity but it is supplied from solar PV panels. These refrigerators are expensive and have heavy lead acid batteries which deteriorate quickly in hot weather. It must be replaced approximately every three years which makes them more costly. However it is possible to replace the battery with thermal storage shown in Figure 1.2 and 1.3. Thermal energy can be stored using either sensible or latent heat through phase change materials during the day time and can be utilized during the night time. In this study ice thermal storage is used to preserve the night time milk for the rural area around Adama city. In the morning, the preserved milk is supplied to nearby market.

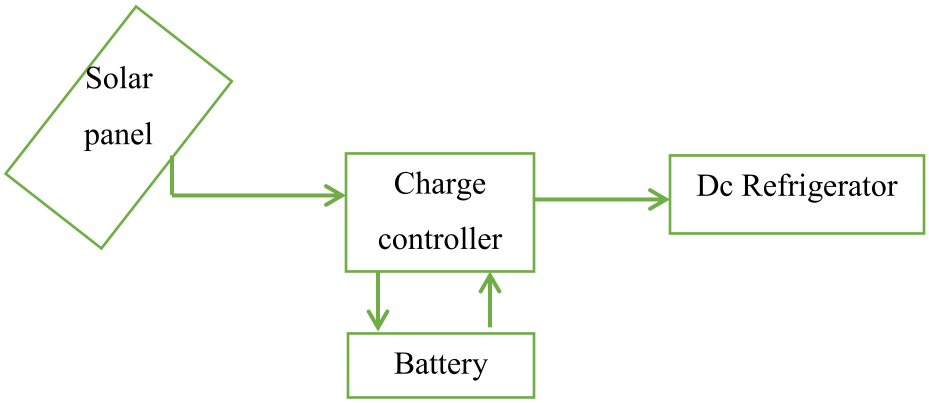


Figure 1.1: Schematic of the Solar Powered DC Refrigeration System with battery

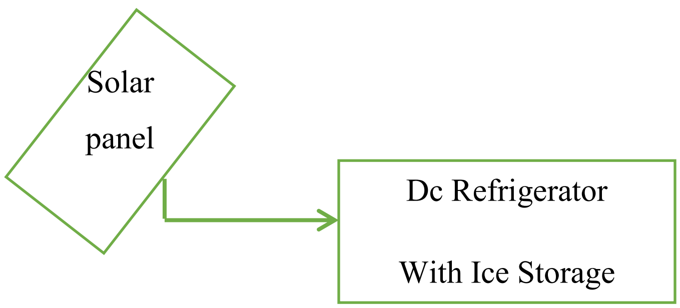


Figure 1.2: Schematic of the direct coupled Solar DC Refrigeration System without battery.

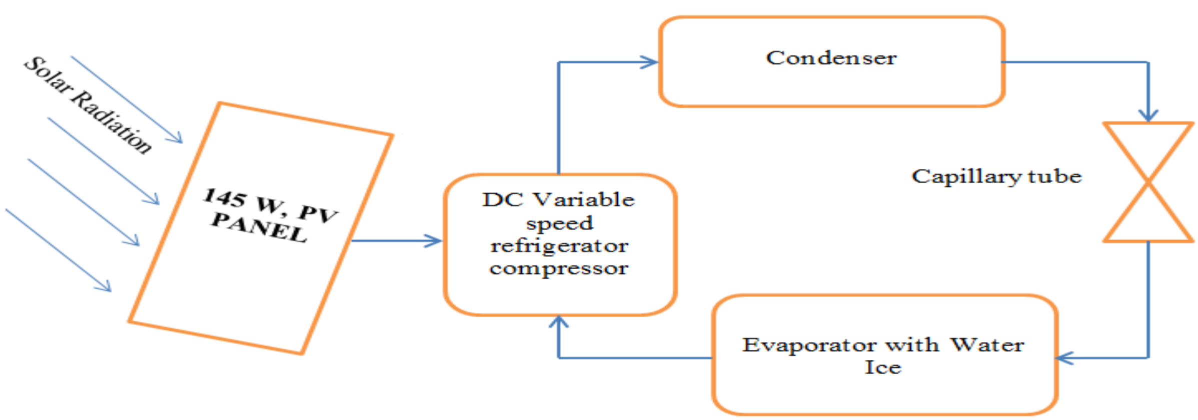


Figure 1.3: Schematics of Direct-Coupled Photo Voltaic Refrigeration System

1.2 Statement of the Problem

In rural areas of Ethiopia milk collection is done in day and night time milking events. For day time milking event, it can be directly sold or supplied to nearby market whereas for the night time milk it should be preserved until the morning. Therefore, some kind of milk preservation is required during the night time so that the night time milk is to be supplied to the market. In Ethiopia most of the population lives in rural area and only 45% of the country's population has electricity access and rural area electricity coverage is very low[5]. The energy source for dairy and food preservation in rural areas requires off grid power source. And Ethiopia has high potential for solar energy with daily average solar radiation 5 kw/m^{2}day[6]. However solar energy supply is intermittent and requires energy storage device.

A complete off-grid solar PV cooling system normally consists of 4 basic components: photovoltaic modules, a battery, an inverter circuit and a vapour compression refrigerator[7]. However this study is aimed to design and develop to replace the AC compressor of the refrigerator with direct coupled with solar panel and variable speed DC compressor with the help of ice thermal storage to serve as a DC refrigerator without electrical battery for off grid refrigeration.

1.3 Objective

1.3.1 General Objective

The objectives of this research is to design, analysis, manufacture and experimental testing of direct coupled Photo voltaic refrigerator with ice thermal storage for dairy preservation at rural areas of Ethiopia.

1.3.2 Specific objectives

- Collecting & analyzing solar data for the study area
- Design PV refrigerator components for the required cooling load
- Simulate the PCM (water-ice) using Ansys Fluent for its performance
- Conduct the experiment on the solar refrigerator model with ice thermal storage

1.4 Methodology

The methodology followed in this study is based on the objectives listed the above section. Literature review, collect site solar radiation data, design, manufacture prototype, perform simulation to size thermal storage, experiment and processing of data are methodologies of this work to arrive at conclusion. The logical steps of activities in the methodology used in this work is given in Figure 1.4

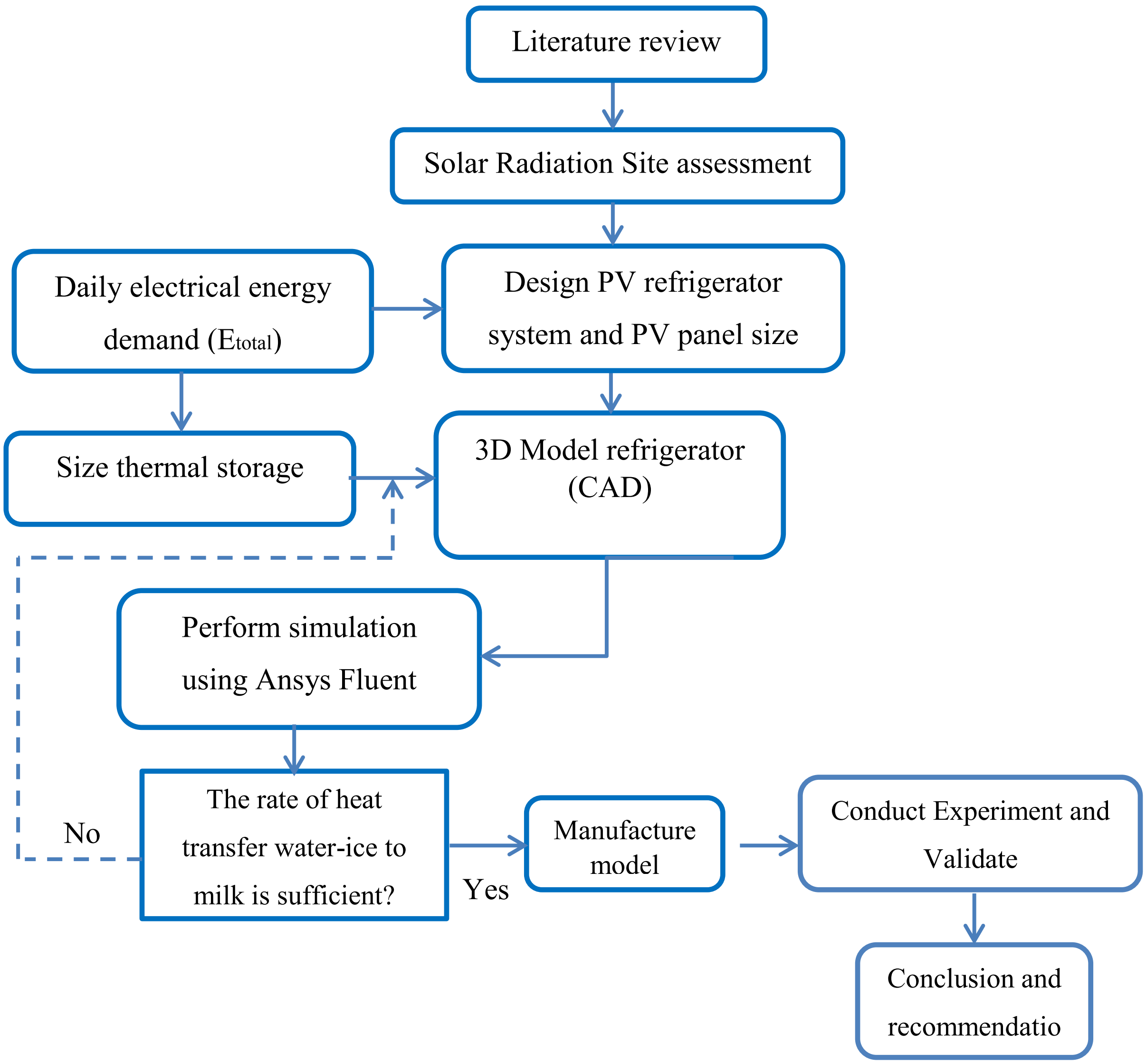


Figure 1.4: Flow chart for methodological approach

CHAPTER TWO

2. LITERATURE REVIEW

Literature reviews on combining the vapour compression refrigeration system with a solar photovoltaic system for the purpose of domestic refrigeration for remote or off grid area.

2.1 SOLAR ELECTRIC REFRIGERATION

Solar powered refrigerator is a cooling appliance that works fully with energy supplied by the solar energy.

Several researchers studied deferent types of solar cooling system, such as Critoph and Thompson [8]discussed the three classes of cycle that can be used for renewable energy powered refrigeration systems. These are: A standard photovoltaic panel driven mechanical vapour compression cycle; Adsorption cycles; Adsorption refrigeration cycles rely on the adsorption of a refrigerant gas or solid into an adsorbent at low pressure and subsequent desorption by heating. The adsorbent acts as a chemical compressor‟ driven by heat. And desiccant cooling is used in air-condition system to cool air. Chin [9]conducted a review on different types of solar energy utilizing methods such as solar electric method, solar mechanical method and solar thermal methods of utilizing solar energy for refrigeration. A comparison is made from the point of view of economic and environmentally. The results of the review concluded that Solar Electric Method is the most promising method for utilizing solar energy. In this study Photovoltaic panel driven system is used.

2.2 Types of Solar Electric Refrigerators

Depending on the type of power supplied to the compressor used by the Solar Electric Refrigerator (SER), could be categorized in to direct current (DC) and alternating current (AC) refrigerators. Solar panels supplies direct current, so to run AC compressor an inverter is needed to convert DC to AC. Direct coupled solar refrigerators are used for preservation of vaccines and other biological components with DC compressors for remote areas.

A complete off grid solar PV cooling system typically consists of four basic components: photovoltaic modules, a battery, an inverter circuit and a vapour compression refrigerator.

2.3 Solar Electric Refrigerators with thermal storage

Many researchers studied solar electric refrigerators with thermal storage, Zhongbao [10] experimentally studied on photovoltaic dc refrigerator with thermal storage and the result showed an improvement of the refrigerator system performance compared the one with the conventional PV refrigerator without thermal energy storage. Efficiency obtained from photovoltaic refrigerator with ice thermal storage is similar to those obtained by other commercial refrigeration facilities powered by a photovoltaic array with lead batteries[11]. For equal cooling capacity less space is required for ice storage compared with a lead battery [9]. Topo et al. [12] described with experimentally using 300 watt photovoltaic cell and ice gel for the house hold 165L refrigerator t store food. Here in this study also ice thermal storage is used to replace battery which is to store energy for night time.

Some researchers eliminated the use of electric power invertor in PV solar refrigeration by directly coupling PV panel with VDC compressor.[7] The coefficient of performance (COP) of the refrigeration system studied (direct-coupled PV array) is comparable to values obtained for commercial refrigeration equipment containing batteries. In this study also direct coupled PV refrigeration system is used.

[13] Aktacir studied experimentally the reliability of PV-powered multi-purpose refrigerator system for semi-arid climatic conditions of Sanliurfa province in Turkey. The results revealed system can be reliably used in places where the local grid was unreliable.

[14]Jordan et al. proposed a methodology for modeling of an ice bank for thermal storage of energy at low temperature to be used in milk cooling and estimated of ice formation, temperature of the ice bank for the selected refrigeration system.

2.4 The relationship between solar panel output power and solar irradiance

The solar panel output power (P_{out}) for the selected PV panel varies with the intensity of the instantaneous radiation(W/m^2). At $1000W/m^2$ it produces maximum power (P_{peak}). When the radiation is less than the reference value ($1000W/m^2$) and without considering effect of operating temperature , it can be estimated the following expression [15].

$$P_{out} = P_{peak}\, X\, \frac{I_p}{I_r} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 1$$

Where: I_p $is\ solar\ radition\ in \frac{W}{m^2}$,

I_r $is\ reference\ solar\ radition\ (1000\ W/m^2), P_{out}\ is\ output\ power$

2.5 Insulating Materials

It is used to prevent the flow of heat from outside warm space to the cold refrigerating space. The thermal conductivity of the insulating material is low. Spray polyurethane or polyurethane foam (PUF) insulation is flexible in terms of the surfaces they can be applied on[16]. In this study Polyurethane foam was used.

Tablet 2.1: The R-Values of Some Insulation Materials

Insulation material	Thermal resistance (Km^2/W)
Polyurethane foam	0.986-1.356
Fiber glass	0.546-0.616
Polystyrene	0.634-0.9333

CHAPTER THREE

3. Photovoltaic refrigerator system design

3.1 Estimation of Solar Radiation of the study area

Assessment of solar radiation availability is significant in the design input variable for any solar system. Solar radiation data are available in several forms for a variety of purposes. The beam and diffuse solar radiation is on horizontal surface. The instantaneous measurement is called irradiance. The integrated value of irradiance over some period of time is known as irradiation usually per hour or day. The estimated solar radiation is carried out in study using measured data from Ethiopian national metrology (Figure 3.1).

3.1.1 Analysis of solar radiation data of the site

Four years solar radiation and temperature data which is collected every 15 minutes was converted into monthly average daily as shown Figure 3.1and Figure 3.2 below.

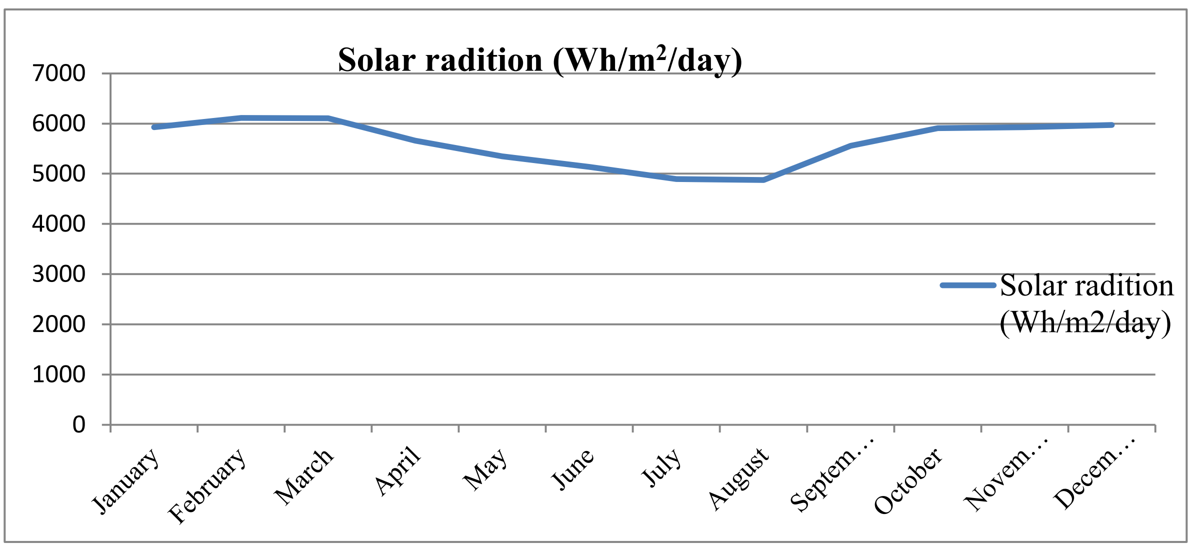


Figure 3.1: Four years monthly average daily, solar radiation on horizontal surface of Adama Ethiopia

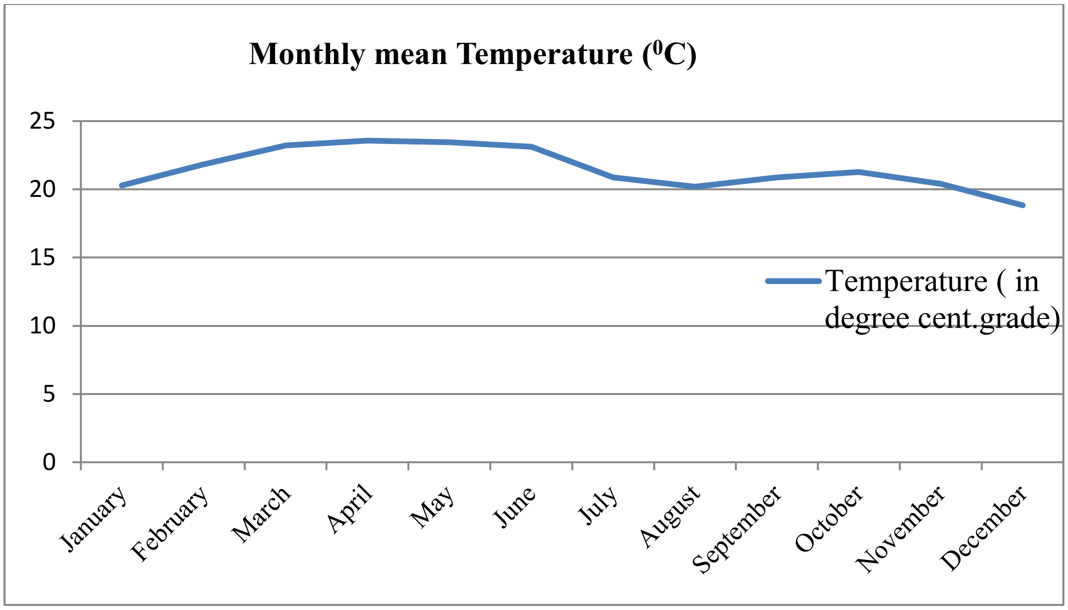


Figure 3.2: Four years monthly average daily, ambient temperature of Adama Ethiopia

From the above Figure 3.1 the month of August had lowest monthly average daily radiation. So the Photo voltaic panel is designed with the month having the lowest monthly average daily solar radiation. And also monthly average daily radiation is for horizontal placed panel, for optimally tilted panel it is more than the value depicted.

3.1.2 Optimum Tilt Angle of the solar panel

The researchers state that for small values of latitude, the optimum annual tilt angle is closer to the location's latitude while for higher ones the optimum tilt angle is smaller[17]. So, the optimum panel tilt angle of the site is the local latitude angle which is $\beta = 8.54\ ^0$.However, the radiation flux on optimally tilted panel is not better than that of horizontally installed panel at all times. Table 3.1 shows radiation flux for July and August months less than that of horizontally positioned panel, so for those cases it requires tracking.

Table 3.1: Monthly Average Hourly Radiation on Optimally Tilted Surface (kWh/M2/Day)

Month	Monthly Average Hourly Radiation On Optimally Tilted Surface(kWh/M2/Day)
January	6.40
February	6.41
March	6.19
April	5.55
May	5.13
June	4.87
July	4.67
August	4.74
September	5.56
October	6.12
November	6.36
December	6.5

August is the design month for the selected site and the 4 years every 15 minutes metrological data of that month was used to draw average hourly global radiation as shown Figure 3.3 below.

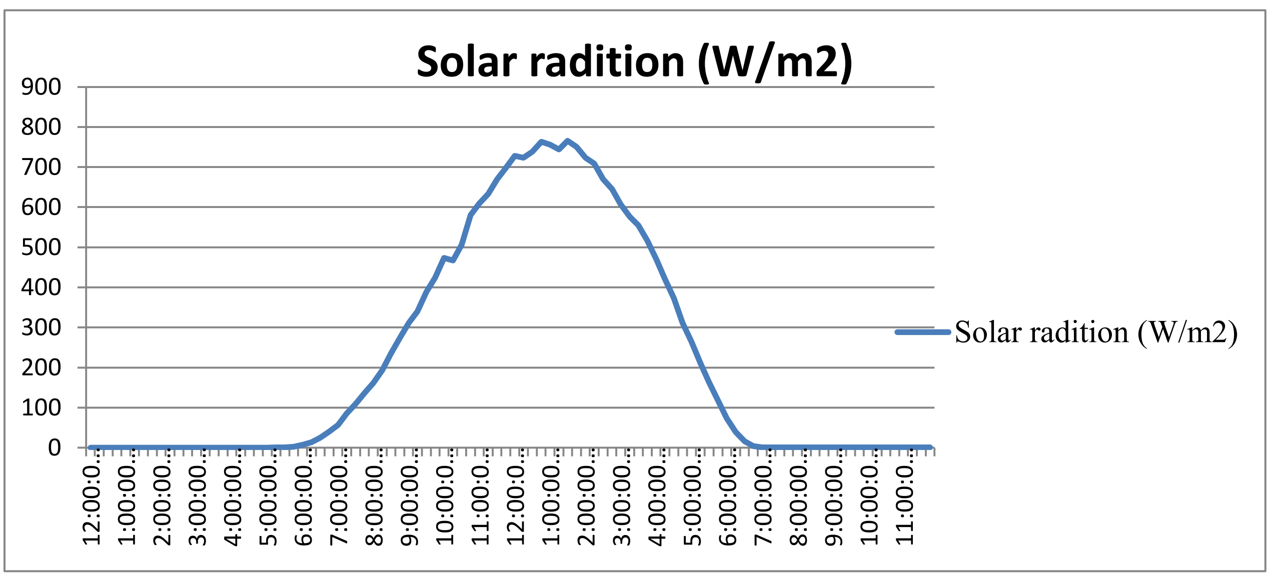


Figure 3.3: Four years average hourly global radiation for the month of August

3.1.3 The peak solar hours (PSH)

Solar PV systems are sized normally based on the month with the lowest irradiation that has lowest peak sun hours (PSH). Both the solar irradiation and the load energy requirement are considered and matched accordingly for optimization purposes. Due to their linear relationship, it can be assumed that the variation in daily power requirement of the refrigerator is negligible. The month with the lowest number of peak sun hours, which occurs in August would be considered in sizing the solar panel. The peak solar hours is the equivalent number of hours per day at reference irradiance which is $1000W/m^2$ which is equal with daily radiation as the actual radiation levels distributed through the day[18].

After analyzing the metrological data of Adama, August month has lowest average daily solar insolation $4.87kWh/m^2/day$.Therefore PV system is designed to meet the daily energy required by the system, based on the minimum average monthly daily available solar radiation in this case August with irradiance of $4873.217\frac{\mathrm{W}}{m^2}/day$.

$$PSH = \frac{Irradiation\ (\frac{Wh}{m^2}/day)}{Peak\ sun\ irradiance\ (W/m^2)} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots. (1)$$

Where Peak sun irradiance or reference irradiance is $1000\ W/m^2$

$Irradiation\ \frac{Wh}{m^2}/day$ is monthly average daily irradiation of the site

$$PSH = \frac{4.873(\frac{kWh}{m^2}/day)}{1000(W/m^2)} = 4.87\ hours/days$$

3.2 Sizing of Photovoltaic Panel of the System

Generally off-grid PV systems are sized to meet specific load requirements and involve the following main steps:

- Determining the average daily load requirements for each month
- Conduct a critical design analysis to determine the month with highest load to solar insolation ratio
- Size the required energy storage capacity

- Size PV array to meet average daily load requirements during period with lowest sunlight and highest load.

3.2.1 Determining the average daily load requirements for the system

Five liters of milk per day is cooled by means of heat conduction from the water – ice tank to milk chamber, refrigerant copper tube cool and freezes the water water-ice tank. The temperature of the milk poured into the tank is 37°C and it is cooled to 0 to 4°C the conservation temperature. The required energy for cooling the milk from the milking event is calculated as:

$$Q_{milk} = \mathrm{m}\mathrm{c}_p(\mathrm{T}_1 - \mathrm{T}_2) \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots . \ldots \ldots (2)$$

Where;

m is the milk mass (kg), ρ is density of milk, c_p is the specific heat (sensible heat)of milk $\left(\frac{3790\mathrm{J}}{\mathrm{kg\,k}}\right)$,

$T_1$is the milk input temperature (37°C), $T_2$is conservation temperature (0°C − 4°C)

During the morning milking event, the energy required for cooling 5 liter milk (Q_{milk}) :

$$Q_{milk} = \rho\mathrm{Vc}(\mathrm{T}_1 - \mathrm{T}_2)$$

To avoid freezing of milk conservation temperature T_2 shall be in a range of 0°C − 4°C.

$$Q_{milk} = 5x10^{-3}x1035x3790(37-0)J$$

$$\text{Daily milk load } (Q_{milk}) = 725690.250J/day$$

In the day time milk chamber is empty, so it is filled with air.

Energy for cooling the air in the milk chamber:

$$\mathrm{Q}_{\mathrm{air}} = \mathrm{m}_{air}\mathrm{c}_{air}(\mathrm{T}_{\mathrm{ambiant}} - \mathrm{T}_2) \ldots \ldots \ldots \ldots \ldots \ldots \ldots (3)$$

Where;

m_{air} *is mass of air in milk chamber*, $\mathrm{c}_{air}$is the specific heat of air

(c_{air} = 1006.43j/kgK), T_2 *cold chamber temprature*

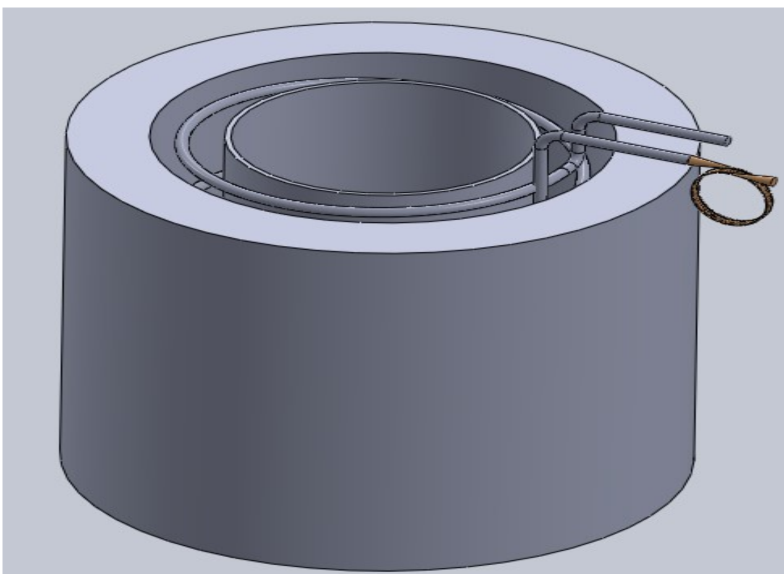
$$\mathrm{Q_{air}} = \mathrm{m}_{air}\mathrm{c}_{air}(\mathrm{T_{ambiant}} - \mathrm{T_2})$$

$$\mathrm{T_2} = 273\ K, \mathrm{T_{ambiant}} = 298K, \rho = \frac{1.225kg}{m^3}, V = 0.005m^3(milk\ chamber\ volueme)$$

$$\mathrm{Q_{air}} = \frac{1006.43\mathrm{j}}{\mathrm{kgK}} * 0.005m^3 * \frac{1.225kg}{m^3}(298 - 273) = 154.11J$$

3.2.2 A heat gain from ambient

The refrigerator gains heat from the surrounding due to conduction heat gain through the insulator and convective heat transfer from the external surface of the refrigerator. In general, as a rule of thumb thermal insulation thickness of 50mm is used.



Dimension Height (H) =185mm

Outer diameter (Do) =335mm

Insulation thickness =50mm

Milk chamber diameter=185mm

Figure: 3.4 Geometry of the refrigerator

Heat flow rate across the insulator surface ($\dot{Q}_{surface}$) is expressed as:

$$\dot{\mathrm{Q}}_{\mathrm{surface}} = \mathrm{U_o}\mathrm{A}(\mathrm{T}_\infty - \mathrm{T_{ice}}) = \frac{\mathrm{T}_\infty - \mathrm{T_{ice}}}{\mathrm{R_{total}}} \dots\dots\dots\dots\dots\dots\dots\dots\dots.(4)$$

where;

$\mathrm{U_o}$ is the overall heat trasfer coeficient based on outside surface area of refrigerator

T_{∞} is ambiant temprature (°C), T_{ice} is the water − ice tank temprature (°C)

R_{total} is the total thermal resistance or is the sum of conductive and convective thermal resistance

The conductive thermal resistance of the polystyrene insulation is given by:

$$R_{cond} = \frac{\ln \frac{r_0}{r_i}}{2\pi H k_{insu}} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots.. (5)$$

Where:

k_{insu} Thermal conductivity of the polystyrene insulator ($0.041 W/mk$)

r_0 the outside radius of the insulator

r_i the inside radius of the insulator

r the radius of the insulator

$$R_{cond} = \frac{\ln \frac{16.5}{11.15}}{\pi * 0.185 * 2 * 0.041} = 7.777 k/W$$

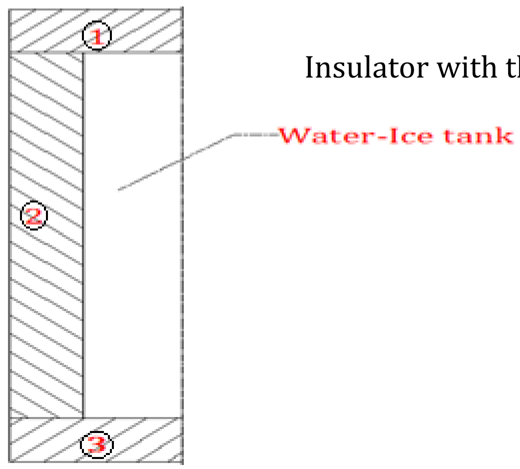
The convective thermal resistance between insulator and air is given by:

$$R_{conv} = \frac{1}{\bar{h}_{cov} A} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots.. (6)$$

where $\bar{h}_{conv}$ is convective heat tranfer coeficient

3.2.3 Determination Natural Convective Heat Transfer Coefficient between Air and Insulator Surface

Insulator with thickness $t_{ins} = 50$mm



T_∞ (ambiant temprature)

Figure 3.5: Insulator surface

Assumptions: in order to obtain natural convective heat transfer coefficient air and refrigerator, at the bottom of refrigerator convection heat transfer is insignificant and the surface of the refrigerator (part 2 in the above Figure 3.5) is considered as vertical plate. The top face of refrigerator as horizontal flat plate is assumed to estimate convective heat transfer coefficient.

Chu [19] developed an empirical correlation to express buoyant convective heat transfer for vertical plate immersed in external flow

$$\overline{Nu_L} = 0.68 + \frac{0.670 {R_a}^{\frac{1}{4}}}{\left(1 + (\frac{0.492}{P_r})^{\frac{9}{16}}\right)^{\frac{4}{9}}} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (7)$$

Where; Pr is prandtl number. R_a is Rayleigh number, and

$\overline{Nu_L} =$ is average nusselt number for characteristic length

Air properties are evaluated at film temperature (T_{film}).

$$T_{film} = \frac{T_s + T_\infty}{2} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (8)$$

where T_s is surface temprature of the insulator surface

The average nusselt number for vertical plate is given by:

$$\overline{Nu_L} = \frac{\bar{h}_{vet} L}{K_{film}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots. (9)$$

$\bar{h}_{vet}$ is average natural convective heat tranfer coeficient

K_{film} is thermalconductivity of film of air at the surface

Rayleigh number is expressed as:

$$R_{aL} = \frac{g\bar{\omega}(T_s - T_\infty)L^3}{\alpha\vartheta} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (10)$$

$$Where\ \ \bar{\omega} = \frac{1}{T_{film}}$$

Heat rate through the insulator ($\dot{Q}$)

$$\dot{Q} = \frac{(T_\infty - T_{ice})}{R_{cond} + R_{conv}} = \frac{(T_s - T_{ice})}{R_{cond}}$$

$$T_s = \frac{(T_\infty - T_{ice})}{R_{cond} + R_{conv}} R_{cond} + T_{ice} \ldots\ldots\ldots\ldots. (11)$$

The value of $\bar{h}_{cov}$ is calculated by the method of iteration.

The Refrigerator surface temperature (T_s) is not known, an arbitrary value of T_s is assumed. Obtain T_{mean} or T_{film} then calculate expansion coefficient ($\bar{\omega}$) at the T_{film} read air properties of ϑ, α, P_r and then calculate Nusselt number, Rayleigh number, the convective heat transfer coefficient, convective thermal resistance using equations 4 to10 . And a new value of T_s is obtained from above equation 11 with this value of T_s the above calculations are repeated till the two consecutive values of T_s obtained are nearly the same.

Using the above procedure the convective heat transfer coefficient obtained is $\bar{h}_{cov} = 3\ W/m^2$

The convective thermal resistance at the external surface of the refrigerator is given by:

$$R_{convection} = \frac{1}{\bar{h}_{cov}A}$$

Where:

$\bar{h}_{cov}$ Convective heat transfer coefficient at external surface of the refrigerator ($\bar{h}_{cov}$=3 w/m^{2}k)

A is normal surface area of the heat flow (A=πDiH= 0.06477m^2)

$$R_{convection} = \frac{1}{3 * 0.06477} = 3.356k/w$$

Total thermal resistance is the sum of conductive and convective thermal resistance

$$R_{total} = R_{conduction} + R_{convection}$$

$$R_{total} = 7.777\text{k/W} + 3.356k/w = 11.1333k/w$$

Heat flow rate across the insulator surface area ($\dot{Q}_{surface}$) is expressed as:

$$\dot{Q}_{surface} = \frac{T_\infty - T_{w-ice}}{R_{total}} \ldots\ldots\ldots\ldots..(12)$$

Where $T_\infty = 25^0C, T_{w-ice} = 0^0C$

The ambient temperature varies seasonally and with hour within a day. The site dry bulb temperature varies 6℃ to 28℃ and also the water-ice tank temperature (5^{0}c to 0^{0}c). The maximum heat gain is given as:

$$\dot{Q}_{surface} = \frac{25^0c - 0^0c}{11.133\ k/w} = 2.24W$$

3.2.4 Heat gain from top and the bottom of refrigerator compartments

It is assumed that the heat transfer at the refrigerator top and bottom surface is due to conduction, radiation and convective heat transfers are not considered. Since the surface temperature and heat transfer area of the surface is small the above assumption

$$\dot{Q}_{bottom} = \frac{T_\infty - T_s}{R_{bottom}} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots . (13)$$

$$\dot{Q}_{top} = \frac{T_\infty - T_{ice}}{R_{top}} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots .. \dots\dots\dots\dots (14)$$

$$R_{top} = \frac{x_{thickness}}{k_{insu}A_t} + \frac{1}{A_{t*}\,\bar{h}_{cov}}, \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (15)$$

$$R_{bottom} = \frac{x_{thickness}}{k_{insu}A_t}, \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots . (16)$$

$There\ is\ no\ \ convection\ heat\ transfer\ from\ bottom\ of\ refrigerator\ , air\ velocity\ is\ nearly\ zero$

$where\ A_b\ \&A_t\ area\ of\ base\ and\ top\ of\ the\ refrigerator\ respectively$

$$A_t = A_b = \pi\frac{Do^2}{4} = \pi\frac{0.323^2}{4} = 0.082m^2$$

$$R_{top}\ = \frac{0.05}{0.041 * 0.082} + \frac{1}{0.082 * 4.6} = \frac{17.54k}{W}, \qquad R_{bottom} = \frac{0.05}{0.041 * 0.082} = 14.8k/W$$

$$\dot{Q}_{top} == \frac{T_\infty - T_{ice}}{R_{top}} = \frac{25^0c - 0^0c}{\frac{17.5k}{W}} = 1.42W$$

$, \dot{Q}_{bottom} \approx 0\ can\ be\ approximated$

$since, the\ temprature\ difference\ between\ the\ ambiant\ and\ insulator\ surfceis\ negligeble$

The total heat gain rate from the surrounding to refrigerator is:

$$\dot{Q_g}\ \ = \dot{Q}_{top} + \dot{Q}_{bottom} + \dot{Q}_{surface} \dots\dots\dots\dots (17)$$

$$\dot{Q_g}\ = 1.42W + 2.24W\ = 3.66W$$

Daily energy loss to the surrounding (Q_g) given by:

$$Q_g = \dot{Q}_g * 24\text{hour} = (3.66W)X24X3600s$$

$$Q_g = 316224\, Joules \text{ perday}$$

The total daily load on the refrigerator is (Q_{load}):

$$Q_{load} = Q_{milk} + Q_g + Q_{air} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (18)$$

$$Q_{load} = \frac{725690.25J}{day} + \frac{316224J}{\text{day}} + \frac{154.11\text{J}}{\text{day}}$$

$$Q_{load} = 1042068.36J/day$$

During the night time the compressor is not running, but in the day time the solar refrigerator will freeze the water to ice. The ice produced in day time will cool the milk at night time from 37^0C to 0°C − 4°C and maintains the cooled milk until it supplied to the market (about 14 hours).

The amount of ice required for the system is obtained by considering the heat gain from the environment and the night milking load (Q_{milk}) and taken care by the latent heat of ice in ice-water chamber.

The required amount of ice is:

$$m_{ice} = \frac{Q_{milk} + Q_{g@14hr} + Q_{air}}{L} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (19)$$

Where; L is the latent heat of fusion $\left(335\frac{\text{kJ}}{\text{kg}}\right)$,

$Q_{g@14hr}$ is daily heat gain from environment while the ice − water is cooling the milk

$$m_{ice} = \frac{Q_{milk} + Q_g + Q_{air}}{L}$$

$$m_{ice} = \frac{\frac{910308.36J}{day}}{\frac{335000J}{kg}} = \sim 3kg of\ ice$$

Therefore, the refrigerator should have to make $3kg$ of ice to cool 5L of milk from to $0°C - 4°C$ range.

3.3 Selection of DC Compressor

The total cooling load required by the refrigerator per day is Q_{load} is $1042068.36J/day$. According to FAO the milk temperature should have to be cooled from milking event temperature (37^{0}C) to 4^{0}C within three to four hour time period. Then the cooling rate of refrigerator is given by:

$$\dot{Q}_{capacity} = \frac{Q_{milk} + Q_{air} + Q_{tot.gain}}{4hr} = \frac{1042068.36J/day}{4 * 3600s} = 73\text{W}$$

The system requires **73** watt compressor, but the milk is not directly cooled by the refrigerant rather it is cooled by thermal storage (ice). In this case, the compressor cooling capacity is determined by the amount of thermal storage (Ice) required that can cool the milk as per FAO requirement.

By assuming compressor running time to 6 hours to produce the required ice of 3kg:

$$\dot{Q}_{capacity} = \frac{\frac{910308.36J}{day}}{6hr} = 42.2\text{W}$$

Therefore, a compressor capacity of greater than 42.2W is selected. And Model MT35DC compressor is selected. It has cooling capacity of up to 62-102 Watt and more if evaporator temperature is maintained above -23^{0}C. It is designed for connection to 12V or 24V DC (9.6V～31.5V) power supply and refrigerant R134a. It can be used in stationary applications powered by photovoltaic solar panels.[20]

Table 3.2: Specification of MT35DC compressor

Application		LBP(Low Back Pressure)
Evaporating temperature range		-30 ^{0}C to0(10)^{0}C
Voltage range/maxi. Voltage		9.6V DC /31.5V DC
Displacement	cm^3	3.5
Oil quantity	cm^3	150
Maximum refrigerant charge	g	300
Free gas vol. in compressor	cm^3	870
Weight: Compressor /Electronic unit	kg	4.3/0.25

Table 3.3: Performance characteristics of MT35DC compressor

Model	**Disp.**	**Speed**	**Cooling Capacity**	**cop**	**Current**	**Power Supply**	**Cooling**	**Refrigerant**
Properties	cm3	rpm	Watt	w/w	12v/24v	volts	FC	R134a
MT35DC	3.5	2000	62	1.3	4.1/2.0	12/24 V	FC	R134a
		2500	75	1.3	5.0/2.5			
		3000	88	1.3	5.9/2.9			
		3500	102	1.3	7.3/3.6			

Figure 3.6: MT35DC Variable speed direct current compressor

Testing condition :

Evaporating Temp -23.3^0C Suction Temp 32.2^0C

Condensing Temp $54.4\ ^0C$ Sub cooling Temp 32.2^0C

Ambient Temp 32.2^0C testing voltage 12V/24V DC

3.3.1 Load characteristics of compressor

The power consumption of the compressor is a function of RPM the compressor, evaporator and condenser temperature.it is constant with respect to input voltage.

$$P_{in}V_{in} = contant \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots. (20)$$

$$P_{in} = V_{in} x I$$

where; P_{in} is power consumption of compressor, V_{in}, is input voltage and I is current consumption .The input current and voltage characteristic of the compressor is given as:

$$I = \frac{P_{in}}{V} = \frac{constant}{V} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots. (20.a)$$

Power consumption of the compressor is constant with respect to source voltage but current consumption is inversely proportional to source voltage.

3.4 Selection of Condenser Temperature

The condensing temperature depends on the type of refrigerant, the type of condenser, and the compressor capability. The condensing pressure and the gas temperature in turn affect the power consumption and the cooling capacity of the system. It is, therefore, necessary to select a condensing temperature based on the recommendation of compressor manufactures.

The ambient temperature of the site can reach up to 28 ^{0}C.There should be minimum temperature difference of 5^{0}C between the ambient and the refrigerant condensing temperature. Therefore the condenser temperature could be 32 to 55^{0}C.

3.5 Selection of Evaporator Temperature

The milk temperature shall be maintained at temperature of 0 to 4^{0}C within the refrigerator. This is achieved by ice storage. The ice temperature could be 0 to -10^{0}C and minimum 5^{0}C temperature difference shall be maintained for effective heat transfer between refrigerant and ice-water chamber. Therefore evaporator temperature -15^{0}C is selected.

3.6 Mass flow rate of the refrigerant can be obtained mathematically:

$$\dot{m} = \frac{\rho V_{displ} RPM \eta_{vol}}{60} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots. (21)$$

Where, ρ is density of refrigerant, V_{displ} is the displacement volume, RPM compressor revolution per minute, η_{vol} is volumetric efficiency . The volumetric efficiency (η_{vol}) of reciprocating compressors is between 65% to 85% [21] .

$$\dot{m} = \frac{\rho V_{displ} RPM \eta_{vol}}{60}$$

Mass flow rate refrigerant can be estimated using variable speed compressor manufacturing data; refrigerant mass flow rate varies with the compressor rpm.

, for 2000-3500 RPM, displace Volume of 3.5 cm^3 density of refrigerant at inlet of compressor is $\rho = 22.5 kg/m^3$, $\eta_{vol} = 85\%$

Mass flow rate the refrigerant at varies compressor rpm:

$$\dot{m}_{2000} = \frac{10.04 * 2000 * 3.5 * 10^{-6} * 0.80}{60} = 0.00094 kg/s$$

$$\dot{m}_{2500} = \frac{10.04 * 2500 * 3.5 * 10^{-6} * 0.80}{60} = 0.00117 g/s$$

$$\dot{m}_{3000} = \frac{10.04 * 3000 * 3.5 * 10^{-6} * 0.80}{60} = 0.00141 kg/s$$

$$\dot{m}_{3500} = \frac{10.04 * 3500 * 3.5 * 10^{-6} * 0.80}{60} = 0.00164 kg/s$$

The refrigeration capacity is expressed as:

$$Q_{ref} = \dot{m}(h_1 - h_4) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(22)$$

From manufacturer test condition the refrigeration capacity at the maximum Compressor speed ($Q_{ref\ @3500\text{rpm}}$) is 102 watt

$$Q_{ref\ @3500\text{rpm}} = \dot{m}_{3500}(h_1 - h_4)$$

Where;

$Q_{ref\ @3500\text{rpm}}$, The refrigeration capacity

$\dot{m}_{3500}$ *is mass flow rate of refrigerant at* 3500 *rpm*

h_1 enthalpy of refrigerant at the inlet of the compressor $(\frac{\mathrm{kJ}}{\mathrm{kg}})$

$\mathrm{h}_1 = h_g@\ evaporator\ temprature\ (T_{ev})$

h_4 enthalpy of refrigerant at exit of the capillary tube $\left(\frac{\mathrm{kJ}}{\mathrm{kg}}\right)$

The compressor work (w_{inp})

$$w_{inp} = \frac{\dot{\mathrm{m}}_{3500}(h_2 - h_1)}{\eta_c} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (23)$$

$$where\ \eta_c = efficiency\ of\ compressor, 90\%$$

The actual and cycle coefficient of performance (COP) of the refrigerator is given as:

$$COP_{actual} = \frac{Q_{ref}}{w_{inp}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (24)$$

From compressor manufacturer, performance test data refrigerant condensing and evaporating temperature of 32^0C and -23^0C respectively. The COP of the compressor is 1.3

$$w_{inp} = \frac{102W}{1.3} = 78.46W$$

$$\mathrm{h}_2 = \mathrm{h}_1 + \frac{\eta_c w_{comp}}{\dot{\mathrm{m}}_{3500}}$$

$$\mathrm{h}_2 = 244.46\mathrm{kJ/kg} + \frac{0.9 * 59.2\mathrm{J/s}}{0.00117 kg/s}$$

$$\mathrm{h}_2 = 290kJ/kg$$

Delivery temperature is obtained from P_2=condenser pressure is 8 bar , $\mathrm{h}_2 = 290\mathrm{kJ/kg}$
and corresponding temperature is$\mathrm{T}_2 = 54^0 C$.

Table 3.7: Refrigerator cycle Temperature and Enthalpy at different state

State	Temperature (℃)	Enthalpy (KJ/kg)
1	-15	244.46
2	54	290
3	32	96
4	-15	96

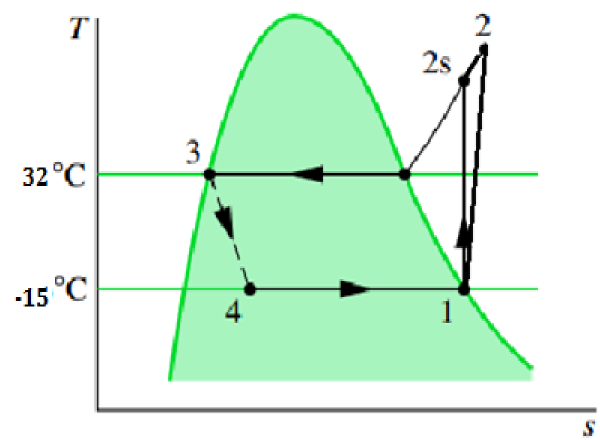


Figure 3.7: T-S diagram of vapor compression cycle

Heat absorbed by the refrigerant at the evaporator

$$\dot{Q}_{ref} = \dot{\mathrm{m}}_{3500}(\mathrm{h}_1 - \mathrm{h}_4) = 0.00117kg/s(\frac{244.46\mathrm{kj}}{\mathrm{kg}} - \frac{96\mathrm{kj}}{\mathrm{kg}}) = 173.7wat$$

Heat rejection rate at the condenser is:

$$\dot{Q}_{cond} = \dot{m}(\mathrm{h}_2 - \mathrm{h}_3) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(25)$$

$$\dot{Q}_{cond} = \dot{\mathrm{m}}_{3500}(h_2 - h_3) = 0.00117kg(290 - 96)\frac{kJ}{kg} = 226.9W$$

3.6 Sizing of evaporator

Evaporator is sized based on the heat load, the linear average mean temperature difference between the refrigerant and water-ice temperatures, and the overall heat transfer coefficient (U_{oev}). The selected evaporator copper coil is having outside diameter of 0.008m with thickness of 0.0005m, Thermal conductivity of the copper pipe tube k_t (385.47 w /m $^{2\ 0}$C), h_{wo} is natural convection for the external heat transfer coefficient between the water and evaporator tube. The evaporator tube is immersed in water - ice storage tank and its temperature varies from 15^{0}C to -15^{0}C).

The rate of heat removed by the refrigerant at the evaporator is given by.

$$\dot{Q}_{ref} = \dot{m}(h_1 - h_4) = U_{oev}A_{ev}(T_{average(water-ice)} - T_{ev}) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (26)$$

$Where;$ A_{ev} Evaporator heat transfer area, $T_{average(water-ice)}$ is -2.5°C ,

T_{ev} is evaporator temrature (-15°C)

The global heat transfer coefficient (U_{oev}) between water-ice and the refrigerant fluid is given by:

$$\frac{1}{U_{oev}} = \frac{1}{h_i A_i} + \frac{A_i \ln\frac{r_0}{r_i}}{2\pi L k} + \frac{A_i}{h_{wo} A_0} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (27)$$

3.6.1 Estimation of Convective heat transfer coefficient inside (h_{iev}) surface of evaporator tube

The refrigerant entering and leaving evaporator temperature is -15 ^{0} C, average water ice temperature at evaporator surface is -2.5 ^{0}C, ^{0} C at this temperature properties are.

From R-134(a) refrigerant property tables film temperature (-6.25^{0}C) [22]

$$\rho = 1327.5 kg/m^3$$

$$D_i = 0.007m$$

$$\mu = 3.0487x\ 10^{-4} Ns/m^2$$

$$V = 3.5m/s$$

$$K = 0.09655W/mk$$

$$P_r = 4.14$$

$$R_{eDi} = {\rho V D_i}/{\mu} = {(1327.5 * 3.5 * 0.007)}\Big/{(3.0487x\ 10^{-4})} = 1.067x10^5$$

$$For, R_{eDi} = 1.067x10^5 \qquad n = 0.3$$

$$N_{uDi} = 0.023 R_{eDi}^{\frac{4}{5}} P_r^{\ n} \ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (28)$$

$$N_{uDi} = 0.023 * (1.067x10^5)^{\frac{4}{5}} * 4.14^{0.3} = 370.99$$

$$N_{uDi} = \frac{h_{iev} D_i}{K} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (29)$$

Where, R_{eDi} is Reynolds number, N_{uDi} is nusselt number K is thermal conductivity of refrigerant

$$h_{iev} = \frac{K N_{uDi}}{D_i} = \frac{0.09655 * 371}{0.007} = 5120 W/m^2$$

3.6.2 Estimation of Convective heat transfer coefficient between surface of evaporator tube and water-ice (h_{wo})

$h_{wo,}$ is natural convection for the external heat transfer coefficient between the water and evaporator tube. Water temperature in water-ice tank varies from 15^0c to -10^0c. So its mean temperature is 2.5^0c.

Churchill and Chu [15]

Properties of water at mean temperature:

$$\rho = 1000 kg/m^3$$

$$\vartheta = 1.307 \ x10^{-6} m^2/s$$

$$P_r = 9.45$$

$$K = 0.580 W/mk$$

$$\varpi = 0.733 \text{ x}10^{-3}$$

Rayleigh number is expressed as:

$$R_D = \frac{g\overline{\omega}(T_\infty - T_s)D^3}{\vartheta^2} P_r \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots . (30)$$

$$R_D = \frac{9.81x0.733\ \text{x}10^{-3}(10℃ - 0℃)(0.008)^3}{(1.307\ \ x10^{-6}m^2/s)^2}x9.45 = 2.036x10^5$$

$$\overline{N}_{ud}{}^{1/2} = 0.60 + 0.387\left\{\frac{R_D}{\left[1 + \left({}^{0.559}\!/_{P_r}\right)^{9/16}\right]^{16/9}}\right\}^{1/6}\quad for\ 10^{-5} < R_D < 10^{12} \dots 35$$

$$\overline{N}_{ud} = 1.8465$$

$$h_{wo} = \frac{K\overline{N}_{ud}}{D} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots.(31)$$

$$h_{wo} = \frac{0.580 * 1.8465}{0.007} = 133.87W/m^2k$$

$$\frac{1}{U_0} = \frac{0.008}{0.006 * 5342.6} + \frac{0.008\ln\frac{0.008}{0.007}}{2 * 385.47} + \frac{1}{133.87}$$

$$U_0 = 129W\ /m^2k$$

Since the change in temperature difference between copper tube surface and water ice is varies linearly with time, mean temperature difference (ΔT_{mean}) is used.

$$\Delta T_{mean} = \frac{\Delta T_2 - \Delta T_1}{2} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots..(32)$$

Where; ΔT & ΔT_2 temperature difference at evaporator initially and final (after ice is formed)

$$\Delta T_1 = T_{wi} - T_{ev} = 15^0c - (-15^0c) = 30^0c$$

$$\Delta T_2 = T_{wf} - T_{ev} = 0^0c - (-15^0c) = 15^0$$

$$\Delta T_{mean} = \frac{30 - 15}{2} = 7.5^0c$$

Then the heat transfer surface area of the evaporator is determined using equation

$$A_{s.evap} = \frac{\dot{Q}_{ref}}{U_{0ev}\Delta T_{mean}}$$

It is determined that the maximum cooling capacity of the compressor 102 watts and work input is78.46 $watt$.

From energy conservation:

$$\dot{Q}_{ref} + W_{in} = \dot{Q}_{cond} \dots\dots\dots\dots\dots\dots\dots (33)$$

$$\dot{Q}_{cond} = 78.46watt + 102watt = 313.66watt$$

$$A_{s.evap} = \frac{102W}{129\ \frac{w}{m^{2\ 0}c} * 15^{0}c} = 0.0527m^{2}$$

The length of the copper tube will be (L)

$$A_{s.evap} = \pi d_0 L \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (34)$$

$$L = \frac{A_{s.evap}}{\pi(d_0)} = \frac{0.0527m^{2}}{\pi * 0.008} = 2.1\text{m}$$

3.7 Sizing of condenser

Condensers are sized based on the heat load, the logarithmic mean temperature difference between the refrigerant and the ambient temperatures, and the overall heat transfer coefficient.

The condenser tube is cooled by natural convection of air.

The rate of heat rejected by the refrigerant at the condenser is given by.

$$\dot{Q}_{cond} = \dot{m}(h_2 - h_3) = U_0 A_0 \Delta T_{mean} \dots\dots\dots\dots\dots\dots.. (35)$$

The global heat transfer coefficient between condenser tubes with in which refrigerant fluid is flowing and the environment is given by:

$$\frac{1}{U_0} = \frac{A_0}{h_i A_i} + \frac{A_0 \ln\frac{r_0}{r_i}}{2\pi L k} + \frac{1}{h_{air}} \dots\dots\dots\dots\dots\dots.. (36)$$

Where; $U_0$is global heat transfer coefficient between condenser tubes with in which refrigerant fluid

Convective heat transfer coefficient at the surface of condenser tube is h_{air}:

Convective heat transfer coefficient inside refrigerator surface of condenser tube is :

Condenser entering temperature= 54^0 C

Condenser leaving temperature = 32 ^{0}C

Air temperature at condenser inlet and outlet =25 ^{0}C

Mean temperature of refrigerant in the condenser = (54+32)/2 = 43^0C

Mean temperature = 43 ^{0}C at this temperature properties are

From R-134(a) refrigerant property tables at mean temperature;

For fully developed turbulent flow in smooth tubes the following relation is recommended by Dittus and Boelter [23].

$$N_{uDi} = 0.023 {R_{eDi}}^{4/5} {P_r}^n N_{uDi} = \frac{h_i D_i}{K} \dots\dots\dots\dots\dots\dots\dots..(37)$$

The properties in this equation are evaluated at the fluid bulk temperature, and the exponent n has the following values:

$$n = \begin{Bmatrix} 0.4 & for\ heating \\ 0.3 & for\ cooling \end{Bmatrix}$$

$$\rho = 1020 kg/m^3$$

$$D_i = 0.008m$$

$$\mu = 1.118x\ 10^{-4} Ns/m^2$$

$$V = 3.5m/s$$

$$K = 0.062/mk$$

$$P_r = 3.05$$

$$R_{eDi} = {}^{\rho V D_i}/_{\mu} = {}^{(1020 * 3.5 * 0.006)}\Big/_{1.118 * 10^{-4})} = 1.916x10^5$$

$$For, R_{eDi} = 1.916x10^5 \qquad\qquad n = 0.3$$

$$N_{uDi} = 0.023 R_{eDi}{}^{4/5} P_r{}^n = 0.023 * (1.916x10^5)^{\frac{4}{5}} * 3.05^{0.3} = 547.7$$

$$N_{uDi} = \frac{h_{con} D_i}{K} \ldots\ldots\ldots\ldots\ldots\ldots\ldots.\ldots\ldots\ldots\ldots\ldots.(38)$$

$$h_{con} = \frac{K N_{uDi}}{D_i} = \frac{0.0943 * 547.7}{0.006} = 5{,}586.95 W/m^2 k$$

3.7.1 Estimation of natural convective heat transfer coefficient between air and condenser coil.

The following relations are used[22].

The properties evaluated at the mean temperature;

$$G_r = g\beta \left(\frac{T_s - T_\infty}{\vartheta^2}\right) D^3 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(39)$$

Where, D is diameter of the condenser tube, $\beta = \frac{1}{T_{film}}$, T_s is condenser tube surface temperature

$$\overline{N_u} = \frac{\bar{h} D}{K} \ldots\ldots\ldots\ldots\ldots\ldots\ldots..(40)$$

Where, $\overline{N_u}$ average nussselt number, K thermal conductivity of air,

The properties of the air evaluated at the film temperature(T_{film})

$$T_{film} = \frac{T_s + T_\infty}{2}$$

$$T_{film} = \frac{43 + 25°\mathrm{C}}{2} = 34°\mathrm{C} = 307K$$

$$\beta = \frac{1}{T_{film}} = \frac{1}{307} = 3.257 * 10^{-3}/k$$

$$K = \frac{0.02690W}{mK}$$

$$P_r = 0.7245$$

$$\vartheta = 1.742 * 10^{-5} m^2/s$$

$$G_r = g\beta\left(\frac{T_s - T_\infty}{\vartheta^2}\right)D^3 = 9.81 * 3.257 * 10^{-3}\left(\frac{62.5 - 25}{(1.742 * 10^{-5})^2}\right)0.008^3 = 1957.65$$

$$G_r P_r = 1.418x10^3$$

For horizontal cylinder $10^2 < G_r P_r < 10^4$

$$\overline{N}_u = 0.85(G_r P_r)^{0.188} \dots\dots\dots\dots\dots\dots\dots\dots.\dots\dots\dots\dots\dots\dots.(41)$$

$$\overline{N}_u = 0.85(1.418x10^3)^{0.188} = 3.33$$

$$\bar{h}_{air} = \frac{K\overline{N_u}}{D} = \frac{0.02690 * 3.81}{0.006} = \frac{15W}{m^2}$$

Therefore overall heat transfer coefficient is:

$$1/\mathrm{U}_0 = 0.008/(0.006 * 5{,}586.95) + (0.008 \ln(\frac{0.008}{0.006}))/(2 * 385.47) + 1/15$$

$$U_0 = 15W/m^2$$

Mean temperature difference (ΔT_m)

$$\Delta T_{mean} = \frac{\Delta T_2 + \Delta T_1}{2} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots.\dots..(42)$$

Where;

$\Delta T_2 = T_{ref\ in} - T_{amb} = 54^0c - (25^0c) = 29^0c$

$\Delta T_1 = T_{ref\ 0ut} - T_{amb} = 32^0c - (25^0c) = 7^0c$

$$\Delta T_m = \frac{29^0c + 7^0c}{2} = 18^0c$$

$$A_0 = \frac{\dot{Q}_{cond}}{U_0 \Delta T_m} \qquad A_0 = \frac{313.66watt}{17W/m^2 * 18^0c} = 1.025m^2$$

$$length\ of\ condenssor\ tube\ , \qquad L = \frac{A_0}{\pi d_0} = \frac{1.025m^2}{\pi 0.008m} = 40.8m$$

Standard condenser is selected based on the capacity of the compressor.

3.8 Selection of capillary tube

The primary function of the capillary tube is to supply liquid refrigerant to the evaporator. It also, because of the high frictional resistance to liquid flow through the capillary tube, meters the refrigerant flow enough to maintain the required pressure difference between the high and low sides of the system. They are best suited to close connected packaged systems that use hermetic compressors and have relatively constant loads. Danfoss capillary tube selector software is used to size the capillary tube as shown Figure 3.9.

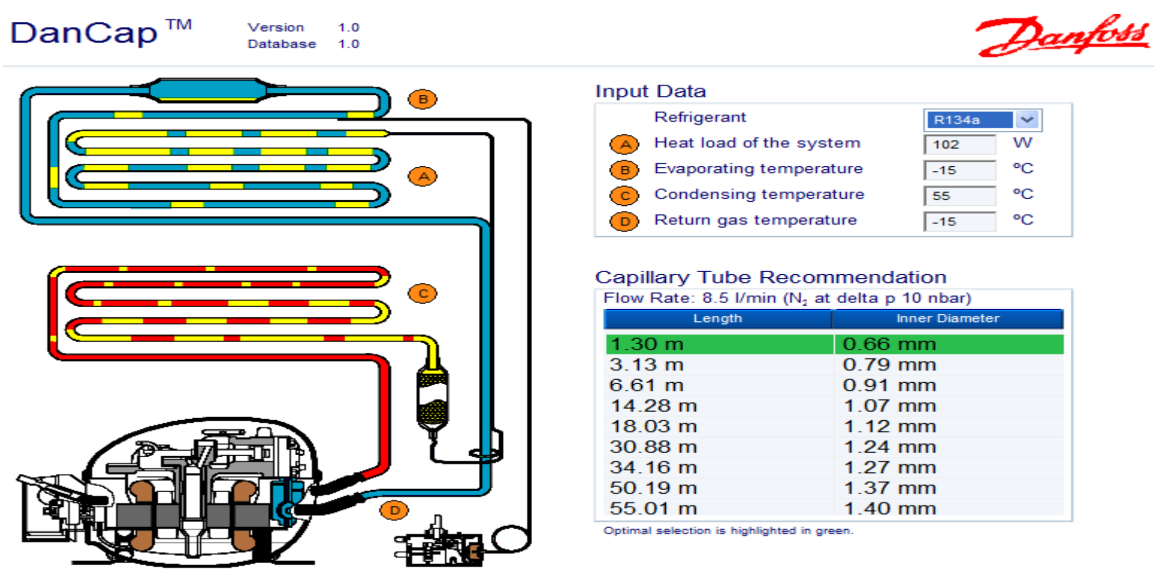


Figure 3.8: Danfoss capillary tube selector

3.8.1 Specification of the capillary tube for the system

Length of capillary tube is 1.30 meter with inside diameter of 0.66mm.

Figure 3.9: capillary tube

3.9 Sizing of PV Panel for the Refrigerator System

3.9 .1 Estimation of daily load on PV system

The total electrical energy requirement of the system (E_{load}), the photovoltaic system supplies a power to the compressor. The input power to the compressor varies with the refrigeration capacity as shown in Table 3-3.Therefore for cooling capacity of 62 to102 watts and the maximum input power to MT35DC is 78.46W as per manufacturer test condition.

The electrical energy required by the system per day(E_{load}@maximum rpm of compressor)

$$= \frac{\text{average work input to compressor xDuty hour of copmressor}}{\text{efficiency of compressor}} \ldots\ldots\ldots.(43)$$

In this case the average input power required running compressor is used to size the PV panel. And compressor runs for 7 hours per day. Efficiency of compressor is 90%.

$$\mathrm{E}_{load} = \frac{65.9\text{W x7 h}}{0.90} = 512\text{Wh/day}$$

The energy demand from the solar panel depends on[24]:

- ✓ The power consumption of the refrigerator
- ✓ The duty hours of compressor (D_H)
- ✓ Efficiencies in the solar PV system and
- ✓ The peak sun hours of the location.

$$W_p = \frac{E_{load}}{PGF} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.. (44)$$

$$PGF\ is\ power\ generating\ facor$$

$$PGF = f_{PV} xPSH \quad where\ f_{PV}\ PV\ panel\ derating\ factor \ldots\ldots. (45)$$

The solar PV derating factor is a scaling factor that applies to the Solar PV array power output to account for reduced output in real-world operating conditions compared to the conditions under which the solar PV panel was rated. Use the derating factor to account for such factors as soiling of the panels, wiring losses, shading, snow cover, aging, and so on.

PV derating factor is used to estimate the actual output from a PV module. The operating factor between 0.60 and 0.90 (implying the output power is 60 to 90% lower than rated output power) in normal operating conditions, depending on temperature, dust on module, etc. [25]

$$PGF = f_{PV} xPSH, PSH = 4.87, f_{PV} = 0.75$$

$$PGF = 0.75x4.87 = 3.652h/day$$

$$W_p = \frac{512.5\text{Wh/day}}{3.652h/day} = 140W$$

3.9.2 Selecting of PV devices for the refrigerator system

Specification of Suntech photo voltaic module

Table 3.8: Electrical characteristics of PV panel

Model	**STP140S-12/Tb**
Number of cell	36
Open-Circuit Voltage (Voc)	22.4
Maximum power voltage (Vmpp)	17.6Volts
Short- circuit current (Isc)	8.33 Amps
Maximum power current (Impp)	7.95 Amps
Maximum power at STC (Pmax)	140Watts
Operating temprature	-40 to 85^{0}C
Maximum system voltage	1000 Volts
Total efficiency	14%
NOCT	$48^0C \pm 2^0C$
Temperature coefficient of $\boldsymbol{I_{sc}}$	0.055%/^{0}C
Temperature coefficient of $\boldsymbol{V_{oc}}$	-0.35%/^{0}C
STC: irradiance 1000 W/m^2 , Module temperature 25^{0}C, AM=1.5	

3.1 I-V characteristics of the PV module

The above information shown in Table 3-8 provided with The PV array manufacturer, at reference standard test conditions (irradiance of 1000 W/m^2, at 25^{0}C air mass 1.5); therefore it is required to drive I-V characteristics equation to find I-V characteristics at different irradiance and temperature condition.

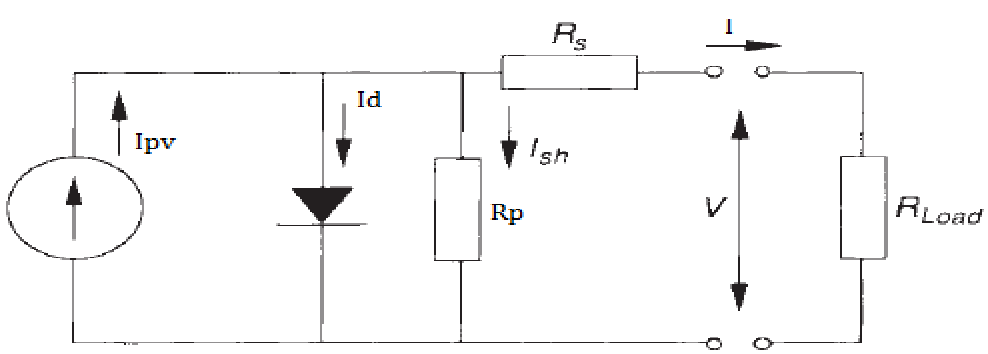


Figure 3.10: Equivalent circuit for PV generator

3.10.1 Governing equations

A single PV module is made of PV cells connected in series. And the terminal current of the module is given as follows.

$$I = I_{pv} - I_o \left[\exp\left(\frac{V+R_sI}{V_t a}\right) - 1\right] - \frac{V+R_sI}{R_p} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(\mathbf{a})$$

$$V_t = \frac{N_s KT}{q} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots....(\mathbf{b})$$

Where

$\mathrm{I_{pv}}$ is light current , $\mathrm{I_0}$ is diod reverse saturation current, $\mathrm{N_s}$ number of cells connected in series,

$\mathrm{V_t}$ is thermal voltage of array

The diode saturation current I_o is expressed as:

$$I_o = I_{on}(\frac{T_n}{T})^3 \exp\left[\frac{qE_g}{aK}\left(\frac{1}{T_n} - \frac{1}{T}\right)\right] \ldots\ldots\ldots\ldots\ldots\ldots..,\ldots\ldots(\mathbf{c})$$

$\mathrm{E_g}$ is the band gap energy of semiconductor ($\mathrm{E_g} = 1.12\mathrm{eV}$)

$\mathrm{T_n}$ referance temprature (nominal temprature), a is the diode ideality constant whose value lies in the range, 1≤ a≤1.5, and for silicon-poly material the value of a is around 1.3 the polycrystalline Si at 25 °C) and $\mathrm{I_{on}}$is the nominal saturation current,

The nominal diode reverse saturation current is given as follows:

$$I_{on} = \frac{I_{scn}}{\exp\left({}^{V_{ocn}}/_{aV_{tn}}\right) - 1} \ldots\ldots\ldots\ldots\ldots\ldots.(\mathbf{d})$$

When the voltage, V is zero, short circuit conditions exist. The current at this point is called short circuit current (Isc) .If the load resistance is infinite, the current is zero; the voltage is at its maximum

The light-generated current of PV cell depends linearly on the solar irradiation and also affected by the temperature.

$$I_{pv} = \left(I_{pvn} + K\Delta\mathrm{T}\right)\frac{G}{G_n} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..\ldots(\boldsymbol{e})$$

The nominal light generated current is expressed as:

$$I_{pvn} = \frac{R_p + R_s}{R_p} I_{scn} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..\ldots\ldots..(\boldsymbol{f})$$

The maximum current output of solar panel at nominal temperature and solar radiation at STC is given as:

$$I_{mpp} = I_{pvn} - I_{on}\left[\exp\left(\frac{V_{mpp} + R_s I_{mpp}}{V_{tn}a}\right) - 1\right] - \frac{V_{mpp} + R_s I_{mpp}}{R_p} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(\boldsymbol{g})$$

Where V_{mpp} $voltage\ at\ at\ maximum\ power\ poit\ (MPP), I_{mpp}\ is\ current\ at\ MPP.$

Rearranging the above equation

$$R_p = \frac{V_{mpp} + R_s I_{mpp}}{I_{pvn} - I_{on}\left[\exp\left(\frac{V_{mpp} + R_s I_{mpp}}{V_{tn}a}\right) - 1\right] - I_{mpp}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots..\ldots(\boldsymbol{h})$$

By solving the above governing equation with MATLAB code (Appendix D) using the algorithm used [26] the following electrical characteristics of the selected Photo voltaic panel of graphs are generated and shown below in Figure 3.11 and Figure 3.12.

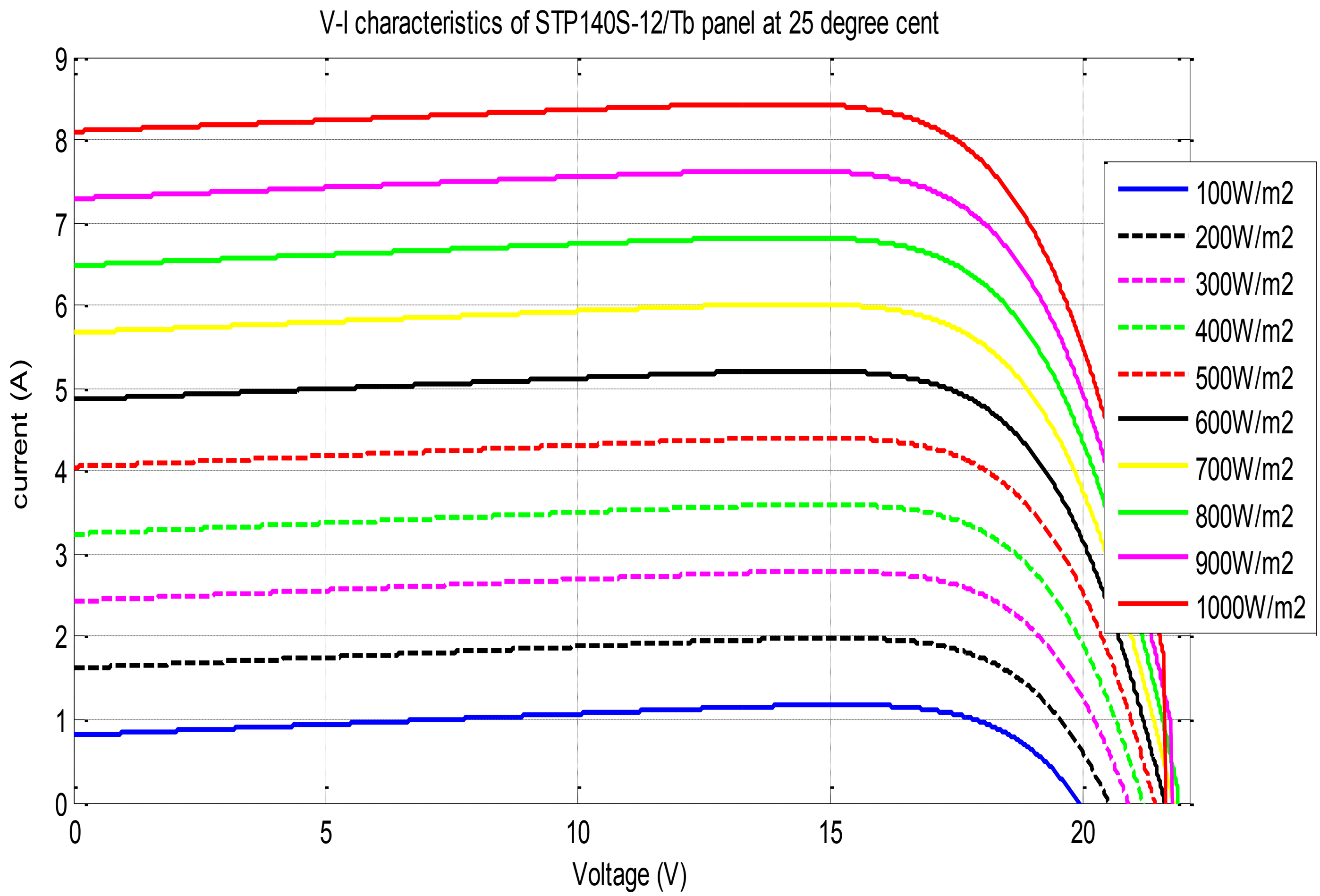


Figure 3.11: V-I characteristics STP140S-12/Tb solar array at different irradiance

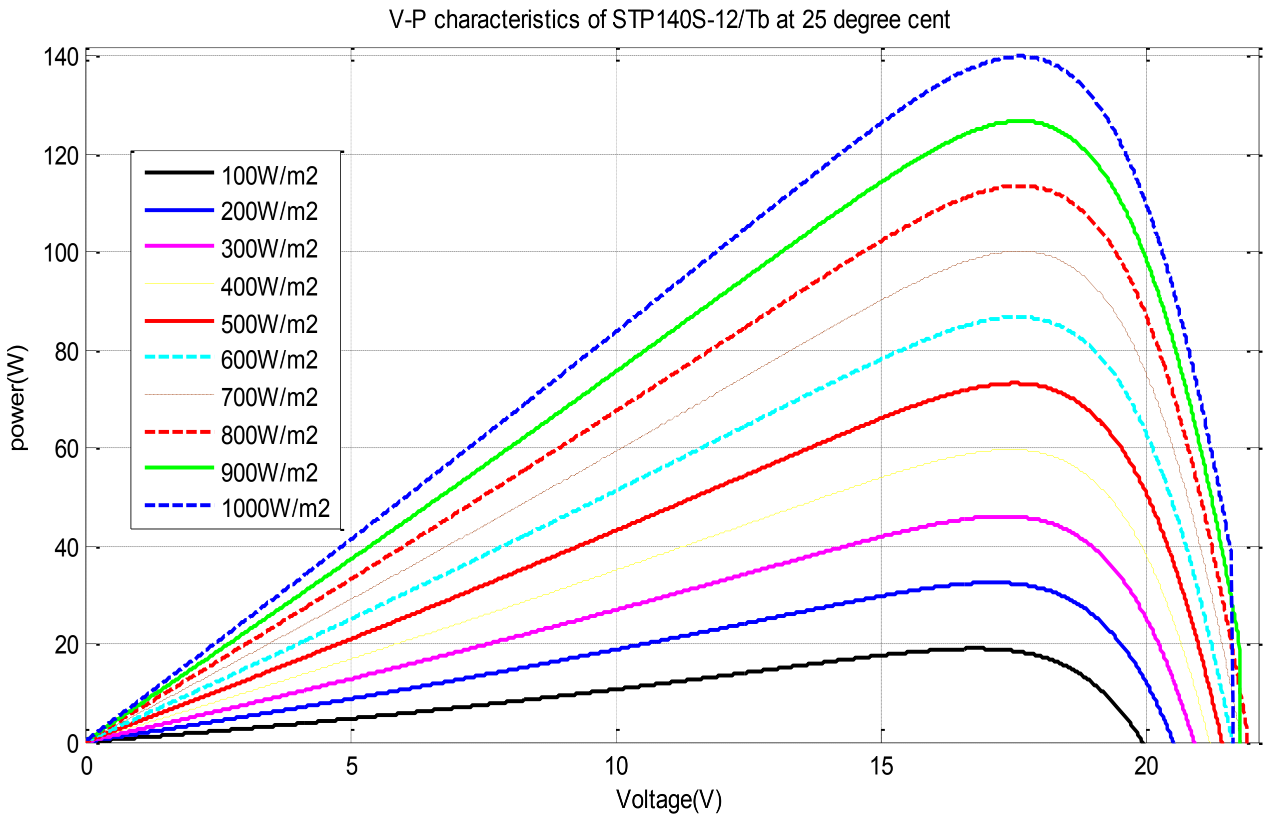


Figure 3.12: P-V characteristics of STP140S-12/Tb solar array at different irradiations

3.11 Minimum solar irradiance required driving the compressor

The selected DC compressor requires minimum power of 47.69 watt to run with lowest rpm (2000rpm).

I-V characteristics of compressor at lowest rpm expressed as:

$$I = \frac{P_{cons}}{V} = \frac{47.69}{V}$$

MATLAB code (Appendix E) is used determine the system operating point. The system will operate with minimum Insolation of 315W/m2 irradiance as depicted in Figure 3.13 below. For a solar radiation of 315 W/m2, the operating voltage is 17.44 V and the current 2.735 A. The power provided from the system is 47.698W. At this operating point the refrigeration has capacity of 62 W.

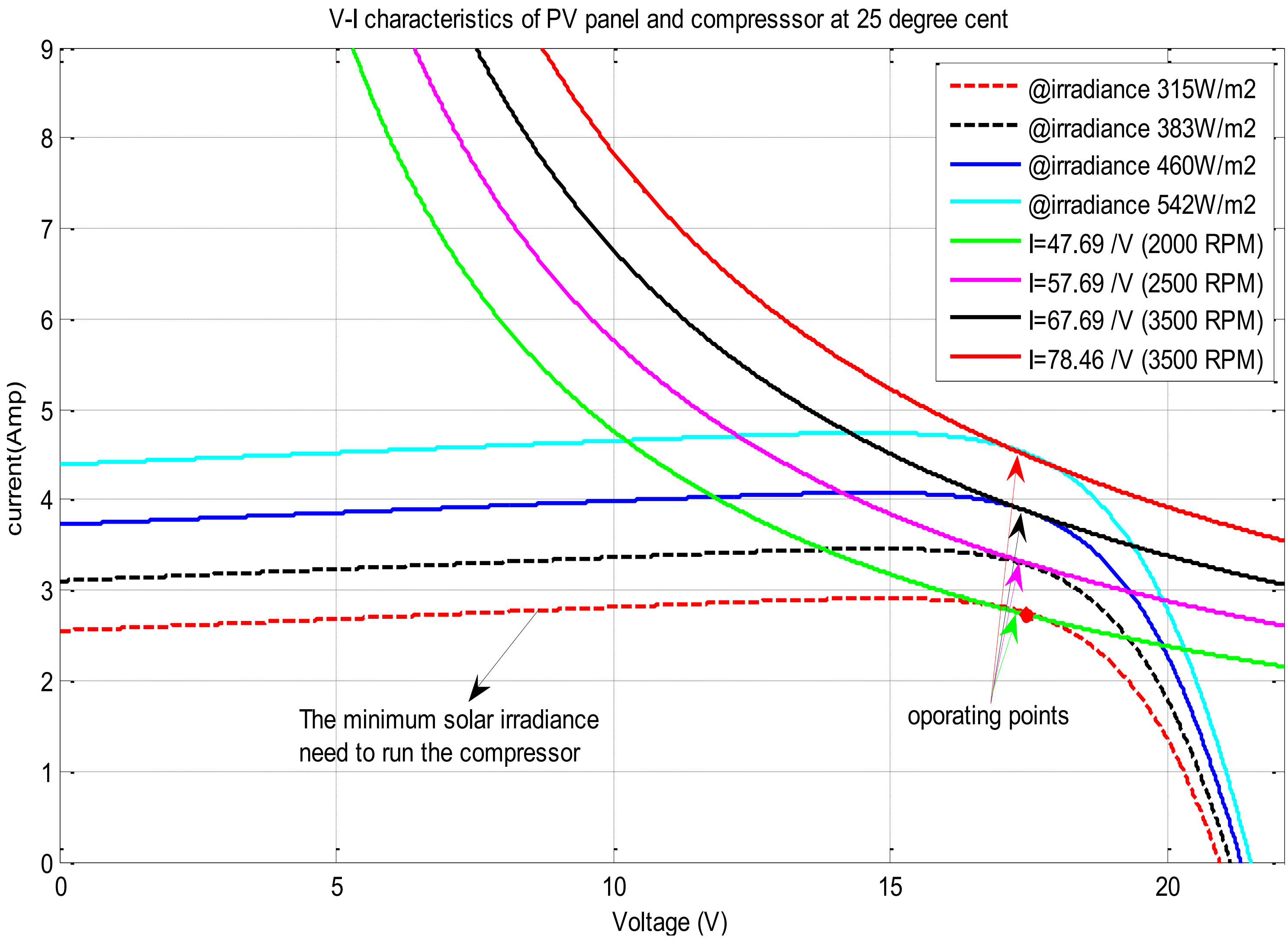


Figure 3.13: I-V characteristics STP140S-12/Tb PV panel and compressor at operating temperature 25 degree cent and compressor at 2000, 2500, 3000 and 3500rpm

CHAPTER 4

4. Ice Formation and Melting

In this section, the amount of ice formed by solar refrigerator during day time and melting of ice to cool 5 liters of milk at night time is simulated. The day time produced ice will melt or discharge the stored energy during the night time to cool milk to a desired temperature ($0^0C - 4^0C$) and maintain the temperature for about 14 hours. Simulation was carried out using Ansys Fluent 18.0.

In the simulation PV panel will supply variable input power to the DC variable speed compressor. The cooling capacity of solar refrigerator varies from 62W to 102W with respect to power in put compressor as shown Figure 4.1.

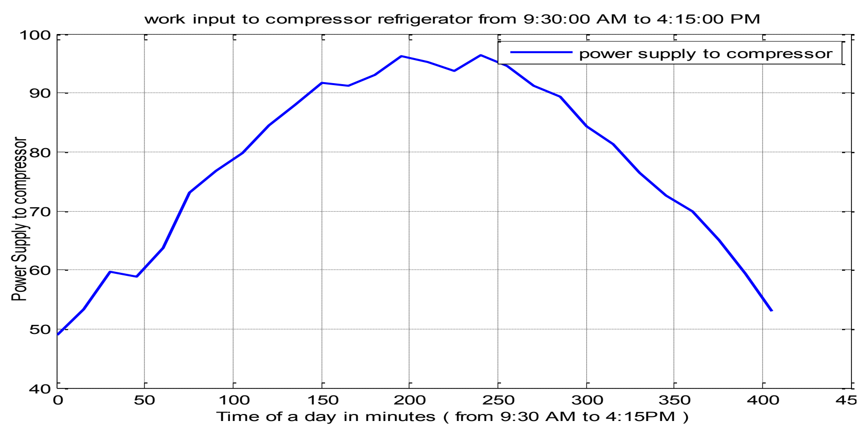


Figure 4.1 Power input to refrigerator compressor

In the evaporator, the refrigerant flowing in a copper tube absorbs heat energy from the thermal storage (water) and the water first cools to freezing temperature and undergoes phase change process and solidifies as it loses its latent heat of fusion. Due to the nonlinear governing equation of the heat transfer and fluid dynamics the Ansys fluent software is used to analyze ice formation and melting in solar refrigerator evaporator section.

4.1 Assumptions

The assumptions made during the numerical modeling are:

- The motion of water in liquid state is incompressible and laminar.

- The density, viscosity and thermal conductivity of the water-ice vary as piecewise linear with temperature.
- The flow of refrigerant in the copper tube is not considered in the simulation rather it is assumed copper tube wall as refrigerant evaporating temperature of -12.5^{0}C.
- The copper coin in the evaporator is turned around milk chamber and formed helix, it is assumed to be circular in the simulation.
- The initial water temperature in the water -ice chamber is to be not more than 4°C.

4.2 Energy for refrigerant flow in the copper tube at the evaporator

4.2.1Governing equation

For the energy equation derivation, an energy balance is completed for a differential volume within the refrigerant flowing in the copper tube in the evaporator system. By balancing the energy transfer in and out of the system with the change in energy of the system over time, the energy equation is derived as follows;

$$\sum \dot{E}_{in} - \sum \dot{E}_{out} = \Delta \dot{E}_{System} \dots \dots \dots \dots \dots \dots \dots (4.1)$$

Energy may be transferred through mass, heat, and work. Energy contained within the fluid flow is represented by three quantities: enthalpy, potential energy, and kinetic energy. Hence contributions from potential and kinetic energy are small in comparison to the enthalpy, the potential and kinetic is neglected. It is convenient to represent enthalpy in terms of temperature such that the energy per unit mass is:

$$h = c_p dT \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots (4.2)$$

where $c_p$is the specific heat capacity of the refrigerant and T is the temperature at a given position

The energy transfer rate due the movement of mass thus becomes

$$\dot{E}_{mass} = \dot{m} c_p dT \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots .. (4.3)$$

In refrigeration system evaporator section consists of refrigerant flowing in a copper tube which is immersed in water. Components of energy transfer are mass energy transfer in, mass energy transfer out, and convective heat transfer through the walls of the copper tube.

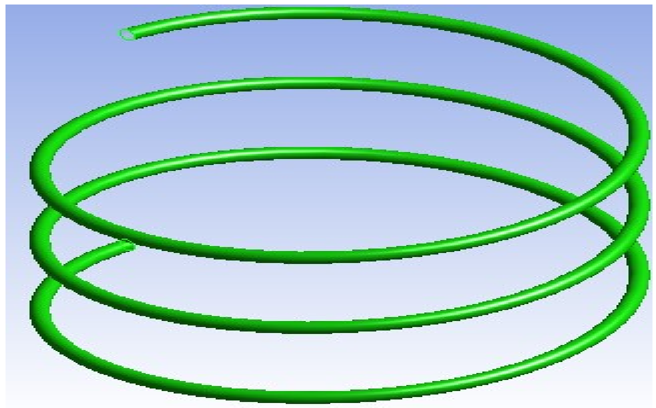


Figure 4.2: Refrigerant flowing in the helical copper tube in the evaporator

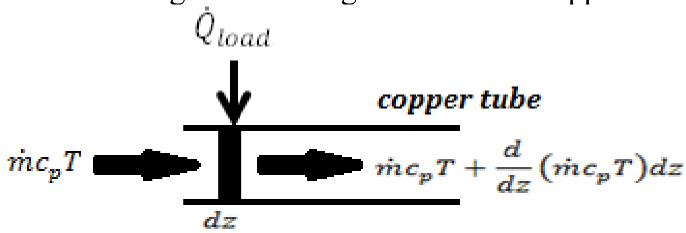


Figure 4.3: Differential volume in the refrigerant flow domain in Z-direction

The governing equations of refrigerant flow in the copper tube are partial differential equations that describe the Conservation of mass, momentum and energy these equations can be written as:

Continuity equation

$$\frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0 \ldots\ldots\ldots\ldots.\ldots\ldots\ldots.(4.4)$$

Momentum equation

In x direction

$$\frac{\partial(\rho u^2)}{\partial x} + \frac{\partial(\rho uv)}{\partial y} + \frac{\partial(\rho uw)}{\partial z} = -\frac{\partial(p)}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) - \frac{2}{3}\frac{\partial}{\partial x}(\rho K) \ldots\ldots\ldots(4.5)$$

In y direction

$$\frac{\partial(\rho v^2)}{\partial y}+\frac{\partial(\rho uv)}{\partial x}+\frac{\partial(\rho vw)}{\partial z}=-\frac{\partial(p)}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)-\frac{2}{3}\frac{\partial}{\partial y}(\rho K)\ldots\ldots\ldots.(4.6)$$

In z direction

$$\frac{\partial(\rho w^2)}{\partial z}+\frac{\partial(\rho uw)}{\partial x}+\frac{\partial(\rho vw)}{\partial y}=-\frac{\partial(p)}{\partial z}+\mu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right)-\frac{2}{3}\frac{\partial}{\partial z}(\rho K)\ldots\ldots\ldots.(4.7)$$

The energy balance within the differential volume is

$$\sum \dot{E}_{in}-\sum \dot{E}_{out}=\Delta\dot{E}_{System}=0\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(4.8)$$

$$In\ x\ direction:\quad \dot{m}c_pT-\left(\dot{m}c_pT+\frac{d}{dx}(\dot{m}c_pT)dx\right)-\dot{Q}_{reff}dx=0\ldots\ldots\ldots\ldots\ldots..(4.9)$$

$$In\ y\ direction:\quad \dot{m}c_pT-\left(\dot{m}c_pT+\frac{d}{dy}(\dot{m}c_pT)dy\right)-\dot{Q}_{reff}dy=0\ldots\ldots\ldots\ldots\ldots(4.10)$$

$$In\ z\ direction:\quad \dot{m}c_pT-\left(\dot{m}c_pT+\frac{d}{dz}(\dot{m}c_pT)dz\right)-\dot{Q}_{,reff}dz=0\ldots\ldots\ldots\ldots\ldots..(4.11)$$

$$\dot{Q}_{reff}=-\frac{d}{dx}(\dot{m}c_pT)dx-\frac{d}{dy}(\dot{m}c_pT)dy-\frac{d}{dz}(\dot{m}c_pT)dz\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(4.12)$$

The convective wall heat transfer at the surface the copper wall is also given as:

$$\dot{Q}_{reff}=Ah_{cov}(T_{reff}-T_{wall})\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.(4.13)$$

For the given mass flow rate of refrigerant the heat absorbed by the refrigerant from copper tube wall is taken from compressor manufacturer test data.

Table 4.1 the cooling capacity of compressor at the given rpm

Compressor (rpm)	**Refrigerant flow rates (kg/s)**	**Heat absorption rate ($\dot{Q}_{reff}$)of Refrigerant**	$\frac{\dot{Q}_{reff}}{A_{evaporator}}$
2000	0.000670	62watts	1209.242 W/m^2
2500	0.000836	74watts	1462.792W/m^2
3000	0.001040	88watts	1716.343W/m^2
3500	0.001170	102watts	1989.398W/m^2

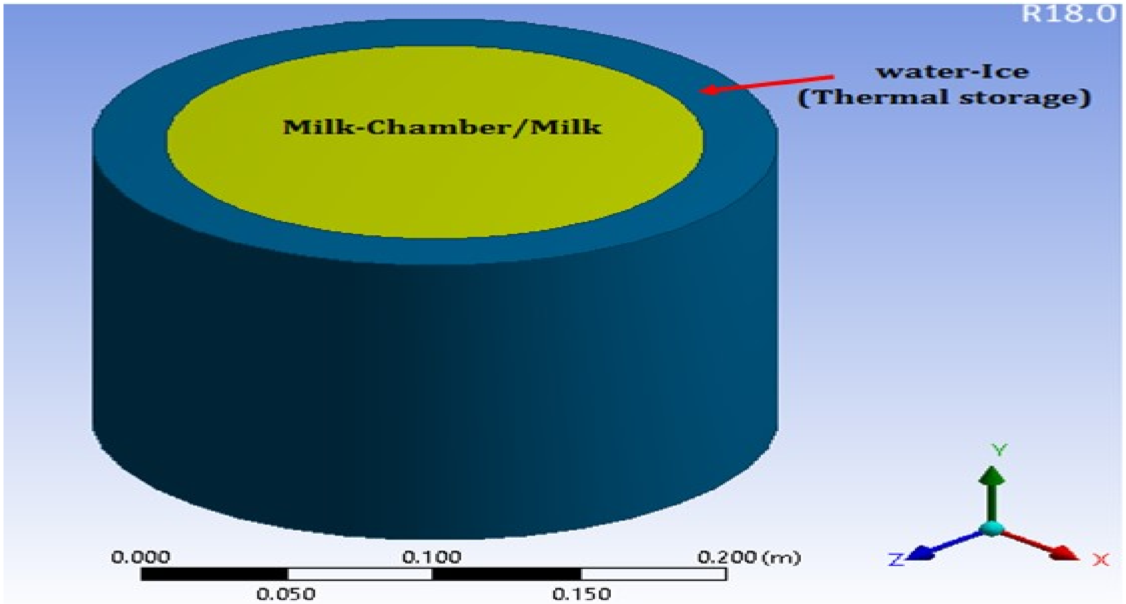


Figure 4.4: Geometry of model (evaporator section of refrigerator where copper tube is immersed in water-ice)

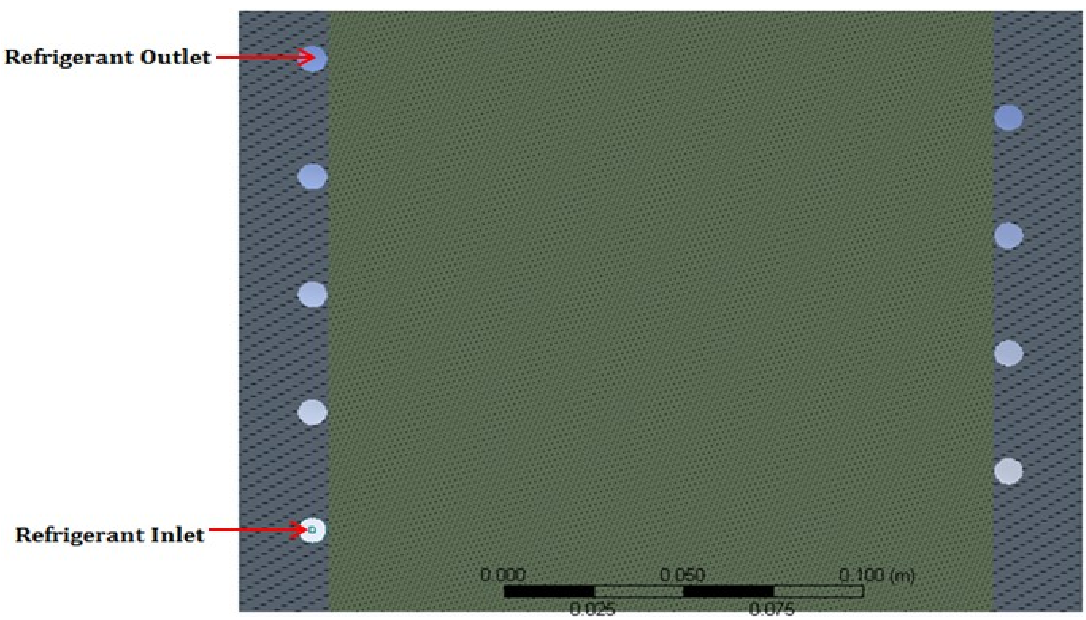


Figure 4.5: Sectional view of the evaporator

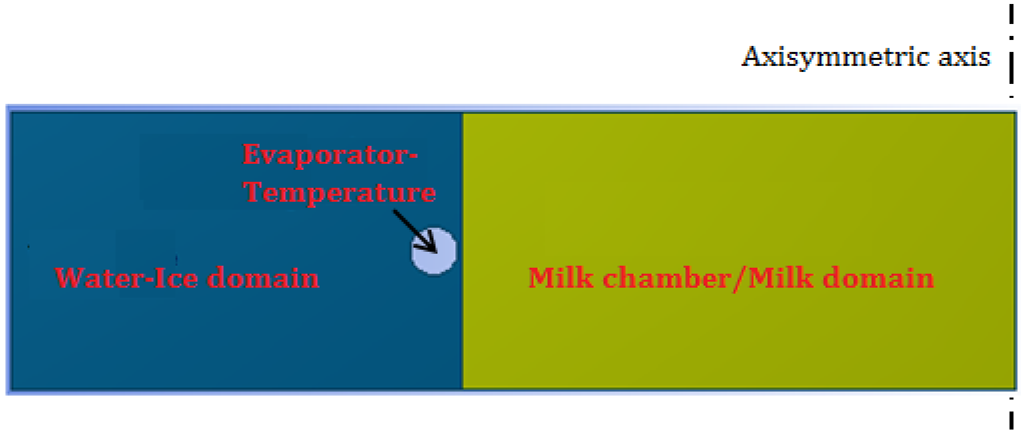


Figure 4.6: The simplified Computational model

To simplify the copper coil is turned in circular fashion rather than helix.as a result the model can be treated as 2D axisymmetric about X axis. And computational model can be simplified as fig 4.6 which is of the domain in Figure 4.5.

4.3 The water cooling time

The melting and solidification model of ANSYS Fluent 18.0 software was used for modeling, simulation of the freezing and melting of water-ice as thermal storage material to cool milk.

Cooling of the liquid water in a water –Ice tank in the evaporator from initial temperature (T_i) to its melting point T_f, will occur before the initiation of ice formation. The time required to cool the liquid from its initial temperature to the solidification point can be determined by solving transient heat conduction model in cylindrical co-ordinates for the cooling of the stagnant fluid in the water-ice tank.

4.3.1The governing equation

For stagnant water in water- ice tank as shown figure 4.6.

Assuming one dimension

$$\frac{1}{r}\frac{\partial}{\partial r}\left(rk_w\frac{\partial T}{\partial r}\right) = \rho c_p\frac{\partial T}{\partial t}, \qquad r_i \leq r \leq r_0, t \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..\ldots..(4.15)$$

Where:

K *thermal conductivity of water*, ρ & C_p *density and specific heat of water at constant pressure*

$$r_i\ radius\ of\ copper\ tube\ r_0\ outer\ radius\ of\ water\ tank$$

$$\frac{1}{r}\frac{\partial}{\partial r}\left(rk_{ins}\frac{\partial T}{\partial r}\right)=\rho c_p\frac{\partial T}{\partial t},\qquad (r_0\leq\ r\leq r_0+t_{ins}\ ,t)\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(4.15)$$

$$K\ thermal\ conductivity\ of\ insulator,$$

$$\rho\ \&\ C_p\ density\ and\ specific\ heat\ of\ insulator\ at\ constant\ pressure$$

The boundary and initial conditions are:

$$\left(k_{ins}\frac{\partial T}{\partial r}\right)=\mathrm{h}_{conv}(T_s-T_\infty)\quad for\ (r\leq r_0+t_{ins}\ ,t)\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(4.16)$$

Where:

h_{conv} $is\ the\ convective\ \ heat\ transfer\ coeficient\ between\ insulator\ surface\ to\ surrounding,$

$T_\infty\ \ ambiant\ temprature, k_{ins}, t_{ins}\ \ thermal\ conductivity\ of\ insulator\ and\ insulator\ thickness$

$$\left(\mathrm{k}_w\frac{\partial \mathrm{T}}{\partial \mathrm{r}}\right)=\mathrm{U_r}\left[\mathrm{T_{copper\ wall}}-\mathrm{T_{reff}}\right],=\dot{Q}_{reff}\quad r\leq r_i\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(4.18)$$

Where: $U_r\ is\ the\ global\ heat\ transfer\ rate\ between\ copper\ to\ refrigerant\ ,$

$$\mathrm{T_{copper\ wall}}\ \ temprature\ of\ copper\ wall,\qquad T_{reff} is\ reffrigerant\ temprature$$

4.4 Phase Change and Freezing Time of Water

Once the water temperature in the water-ice tank reached to its solidus temperature, ice formation and deep freezing is commenced in water ice-tank.

ANSYS Fluent uses enthalpy porosity method for modeling the solidification and melting process. This method indirectly tracks the advancement of the solid-liquid interface by using a liquid fraction; Liquid fraction of cell volume is fraction of cell volume which is in liquid form associated with each cell in the water –ice domain. The porosity in each cell is set equal to the liquid fraction in that cell. The value of liquid fraction changes from 0 to 1 and it is called Mushy zone or partially solidified region [27][28].

4.4.1 Governing equations

The liquid fraction (β) is expressed as:

$$\beta = \begin{cases} 0 & T < T_{solidus} \\ \dfrac{T - T_{solidus}}{T_{liquidus} - T_{solidus}} & T_{solidus} < T < T_{liquidus} \\ 1 & T > T_{liquidus} \end{cases} \quad \ldots\ldots(4.20)$$

Where T is the local temperature, and T_s and T_l means solid state and liquid state temperature of the phase change material (water), respectively.

$$\frac{\partial(\rho)}{\partial t} + \nabla.\left(\rho\vec{V}\right) = 0 \ldots\ldots\ldots\ldots\ldots\ldots(4.21)$$

$$\frac{\partial}{\partial t}\left(\rho\vec{V}\right) + \nabla.\left(\rho\vec{V}\vec{V}\right) = -\nabla P + \nabla.\left(\bar{\bar{\tau}}\right) + \rho\vec{g} + S \ldots\ldots\ldots\ldots\ldots(4.22)$$

$$\text{Source term}(S) = \frac{(1-\beta)^2}{(\beta^3+\varepsilon)} A_{mush}\left(\vec{V} - \vec{V_p}\right) \ldots\ldots\ldots\ldots\ldots(4.23)$$

where ε is a small number (0.001) to prevent division by zero, A_{mush} is the mushy zone constant, its value is 10^5 *to* 10^8 and $\vec{V}$ is fluid velocity, $\vec{V_p}$ is pull velocity for this case is not required

The enthalpy of the water is computed as the sum of the sensible enthalpy and the latent heat:

$$H = h_{ref} + \int_{T_{ref}}^{T} C_p\, dT + \beta L \ldots\ldots\ldots\ldots\ldots\ldots\ldots(4.24)$$

Where h_{ref} is the reference enthalpy, T_{ref} is the reference temperature C_p is the specific heat at constant pressure and L is latent heat of fusion of water

$$\frac{\partial(\rho H)}{\partial t} + \nabla.\left(\rho\vec{V}H\right) = \nabla.(k\nabla T) + S \ldots\ldots\ldots\ldots\ldots(4.25)$$

$$\bar{\bar{\tau}} = \mu\left[\left(\nabla\vec{V} + \nabla\vec{V}'\right) - \frac{2}{3}\nabla.\vec{V}I\right] \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(4.26)$$

Where $\bar{\bar{\tau}}$ the stress tensor is μ is the molecular viscosity of water , and the second term on the right hand side is the effect of volume dilation [Ansys 18.0]

Boundary Conditions axisymmetric boundary condition was applied as described in Figure 4.6. The water and milk domain chamber are insulated with Polyurethane foam, so adiabatic boundary condition was used. To accommodate the slightly refrigerant temperature variation in the evaporator, copper surface temperature of $-12.5°C$ was used as boundary condition. And the two fluid domains thermally coupled by interface wall.

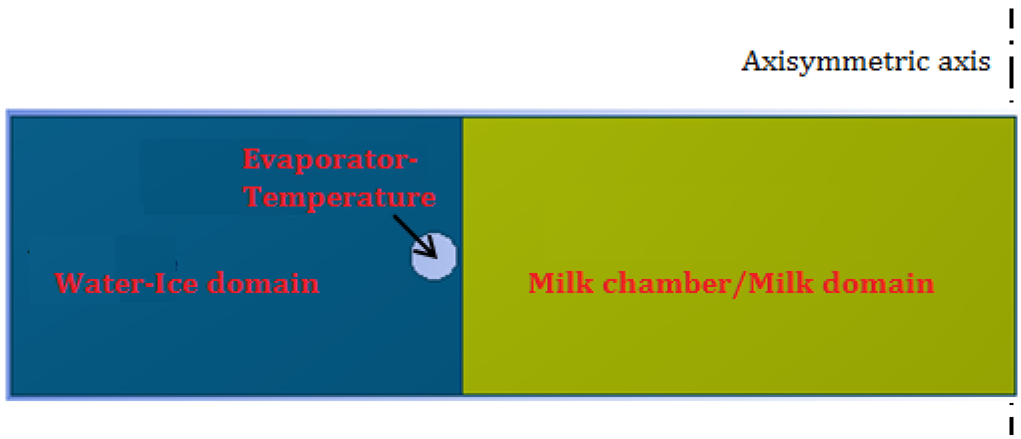


Figure 4.6: Schematic of solidification of water around circular copper tube in water ice chamber

Table 4.2: Properties of water in the study

Property	$T < 273.15K$ (solid)	$\Delta T(phase\ change\ temp.$ 0.3°C)	$T > 273.45K$ (liquid)
Thermal conductivity	1.918W/mK	-	0.579W/mK
Specific heat capacity	2217J/kg K	-	4180J/kg K
Density	Piece wise linear	-	Piece wise linear
Viscosity	-	-	0.001003kg/m s
Heat of fusion	-	335kJ/kg	-

Table 4.3: Properties of Milk in the study

Property	
Thermal conductivity	0.6W/mK
Specific heat capacity	3790 J/kg K
Density	1035kg/m3
Viscosity	0.00179 kg/m s

4.5 Mesh generation

Discretization of the computational model is developed using ANSYS Meshing software. Structured mesh consisting of quadrilateral elements with inflation layer was created. Inflation layers with 1.2 growth rate, maximum layers 4, transition ratio of 0.272, were created on the wall of solid and fluid domain.

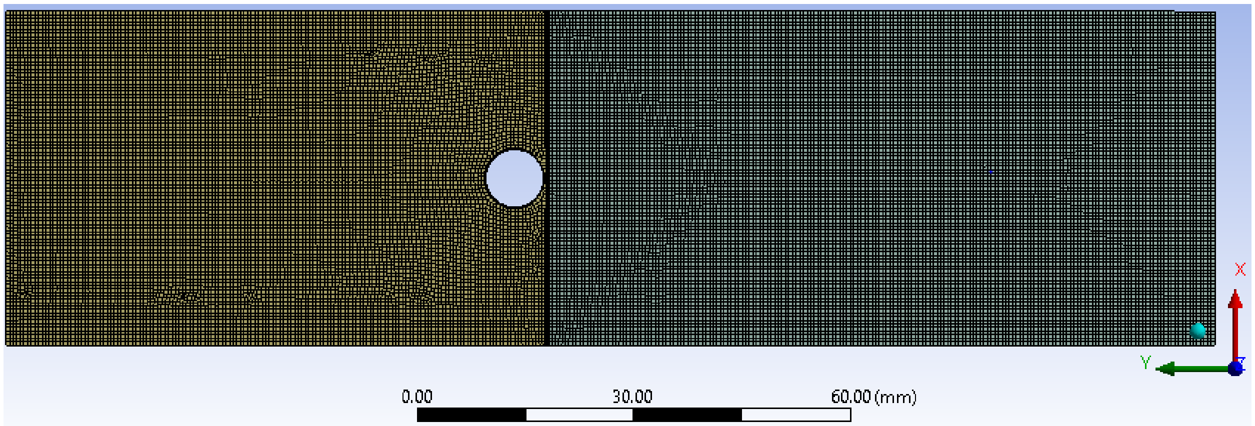


Figure 4.7: Schematics of mesh

The quality the generated mesh was shown in figure below, in this analysis skewness and orthogonal mesh metrics was used. The recommended values for skewness is [0.9 >] and [>0.1] for orthogonal quality. In both cases mesh quality is within the recommended rage.

Quality	
Check Mesh Quality	Yes, Errors
Target Skewness	0.8
Smoothing	Medium
Mesh Metric	Orthogonal Quality
Min	0.71434
Max	1.
Average	0.999
Standard Deviation	7.0152e-003

Table 4.5: Mesh metric

Convective terms in momentum equations are discretized using second order upwind interpolation scheme. Convective terms in energy equations are discretized using first order upwind interpolation scheme. The coupling between pressure and velocity is done by simple algorithm and presto is adopted for pressure interpolation. The convergence criteria for residuals were set 10^{-3} for velocity and continuity and 10^{-6} for the energy equation. And minimum evaporator temperature set to $-12.5\ ^{0}C$. Throughout the simulation 0.1second constant time step was used.

4.8 Grid independence test

The grid independency test was done to observe the numerical results with the mesh size. Four mesh sizes were created, from a finer mesh size which consists of 150289 cells to a coarser 5506 cells. The four mesh sizes were tested for a total of 2 hours flow time of the freezing cycle of water in water ice chamber as shown figure 4.7 below. And details on the three grids are listed in Table 4.6.

Table 4.6 Details of the mesh sizes

Mesh size	Number of cells	Number of nods
0.3mm	85960	86868
0.4mm	48558	49242
0.5mm	31131	31679
0.8mm	12366	12708

Average water temperature result obtained at a mesh size of 0.4mm nearly the same as that of 0.3mm as shown Figure 4-8 below. In the simulation 0.5mm mesh size was selected by accepting 6% error with the result obtained in 0.3mm mesh size. An element size of 0.5 mm was selected. And also 0.5 mm mesh size is the general element size for water freezing and melting process according to several literatures. Michalek, T. and Kowalewski [29] obtained an error of less than 1% in local temperature and velocity with mesh size of 0.1 mm during the water freezing process. Dallaire, J. and Gosselin [30] reported on their studies that with a mesh size of 0.5mm obtained good agreement between the experimental and numerical results considering the phase change interface position.

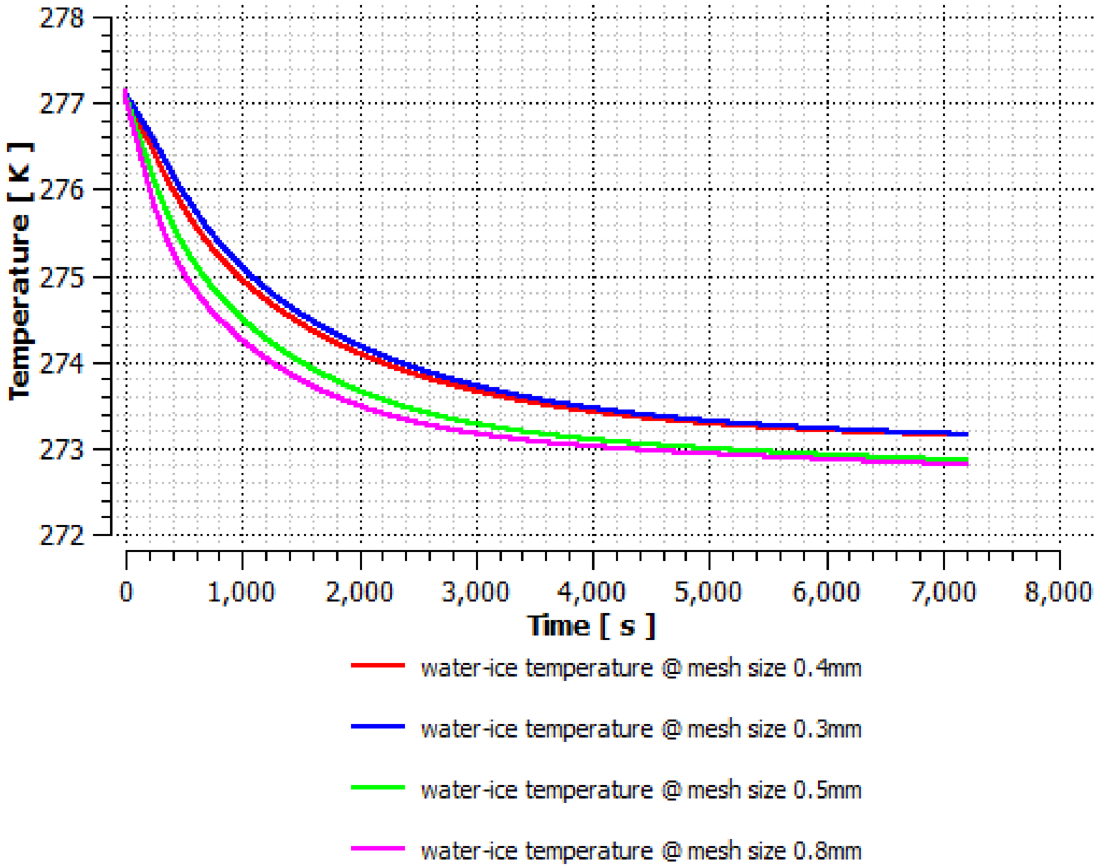


Figure 4.8 Mesh independence verification result

4.6 Result and discussion

4.6.1 Resultes of simulation of ice formation in water-ice chamber with noload in milk chamber

As shown Figure 4.9 below, the minimum average temperature of water-ice was-1.22℃. The first 25 minutes of simulation time temperature of the water is reduced rapidly this is due to the higher temperature difference and heat transfer mainly by convection. As the cooling of water progressed the conduction heat transfer dominates over convection and the temperature difference between refrigerant and water is reduced resulting and low heat transfer takes place also conduction heat transfer mechanism will dominates as the ice is accumulates more. During ice formation in the water-ice chamber, the milk chamber (cold chamber) is empty and the minimum mean temperature of the cold chamber achieved was -9.25℃.

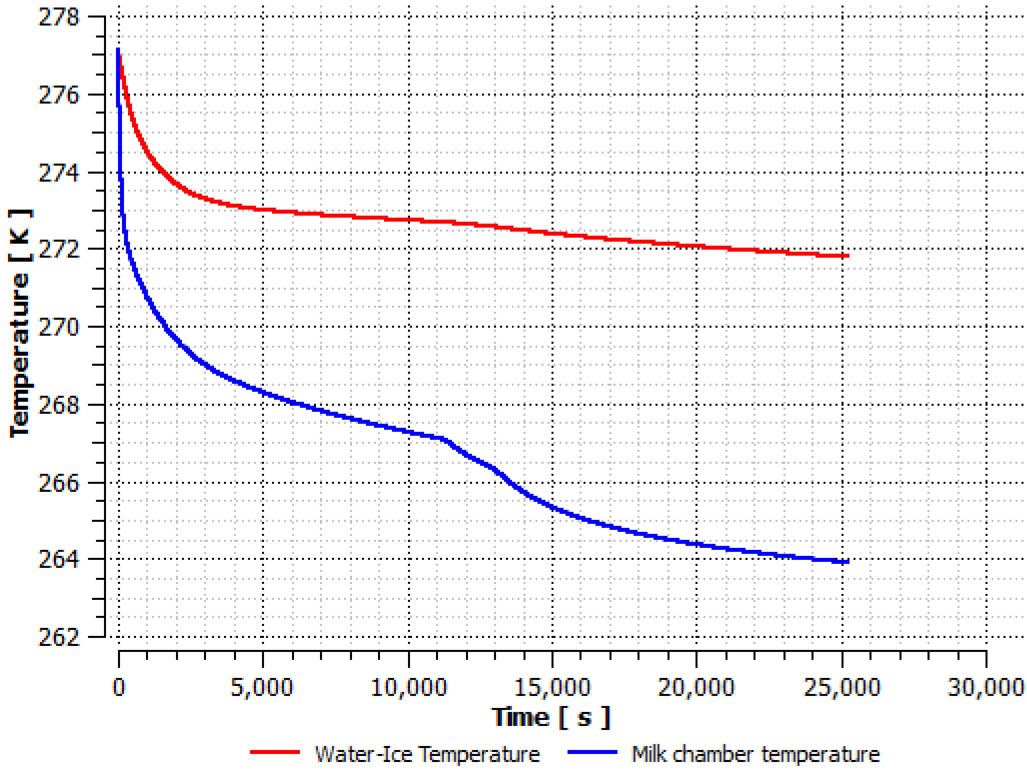


Figure 4.9: water-ice and milk chamber temperature variations while the compressor running.

As depicted Figure 4-10 below, at the end of simulation time 67.6% (nearly 1kg) water in the water-ice tank is converted to solid ice. As time is elapsed the water having lower temperature ice formation is increased linearly with time.

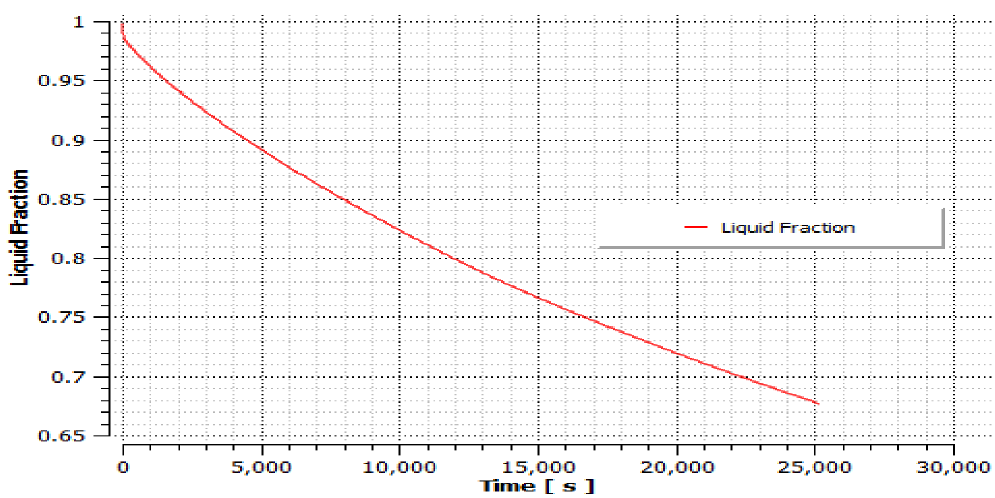


Figure 4.10: Ice formation rate

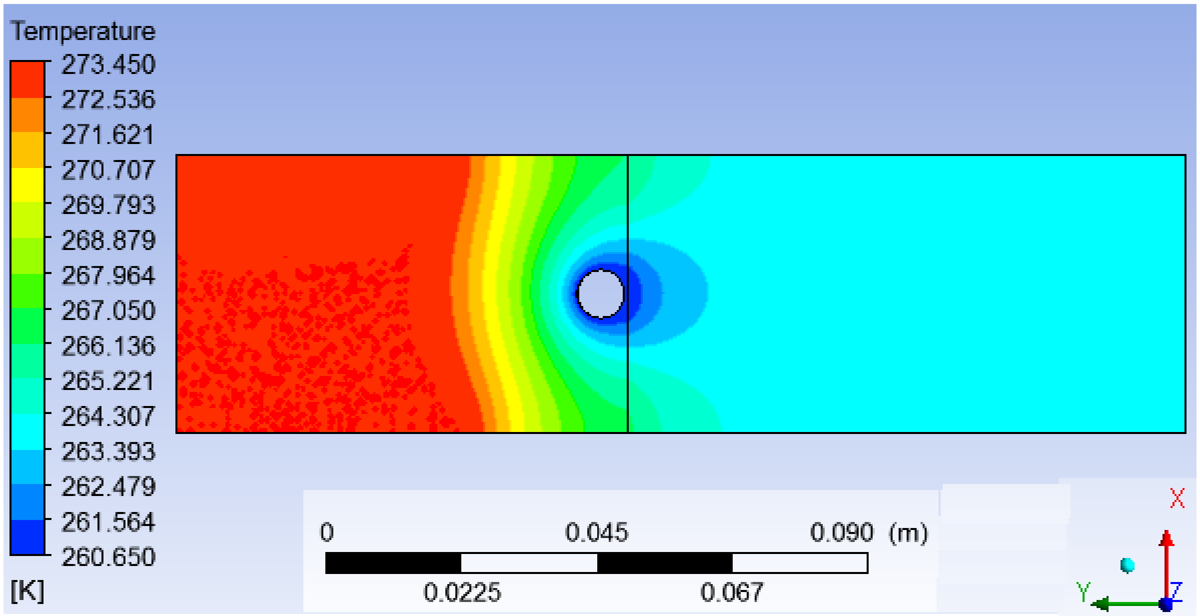


Figure 4.11: Ice-Water and Milk chamber Temperature contour after 7 hours

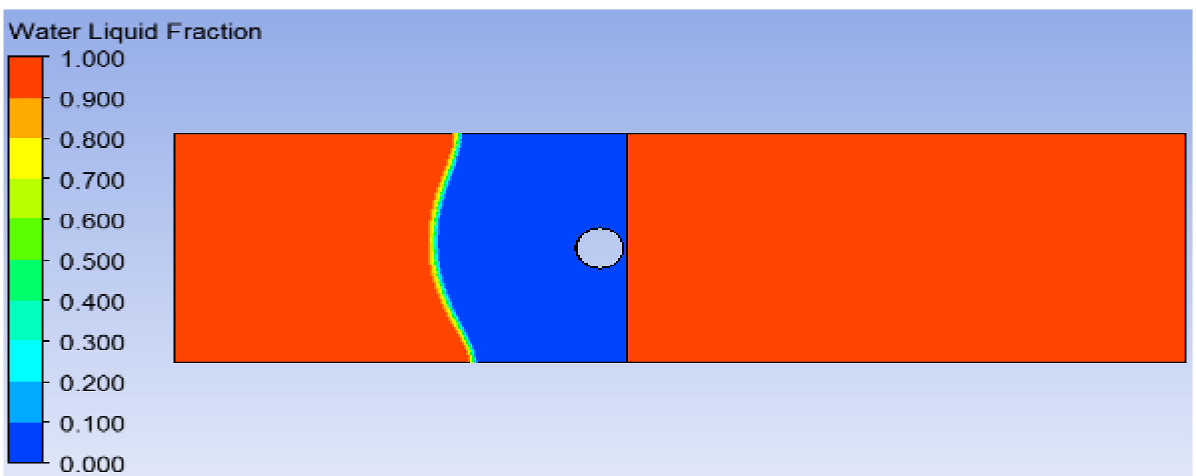


Figure 4.12: Liquid fraction contours in the water-ice tank

4.6.2 Results of simulation, melting of ice in water-ice chamber while cooling milk

As Figure 4-13 shown below, the milk is chilled from its initial temperature 37^0C to a temperature of 0.857^0C. During the first 4 hours of simulation time the milk temperature is reduced to 6.45^0C. It is observed that nearly 1kg of ice at average temperature of -1.37^0C is sufficient to cool or preserve 5 kg of milk for 12 hours and more. Also it shown that the water – ice temperature is raised from -1.37^0C to 0.21^0C after cooling the milk for the period of 12 hrs.

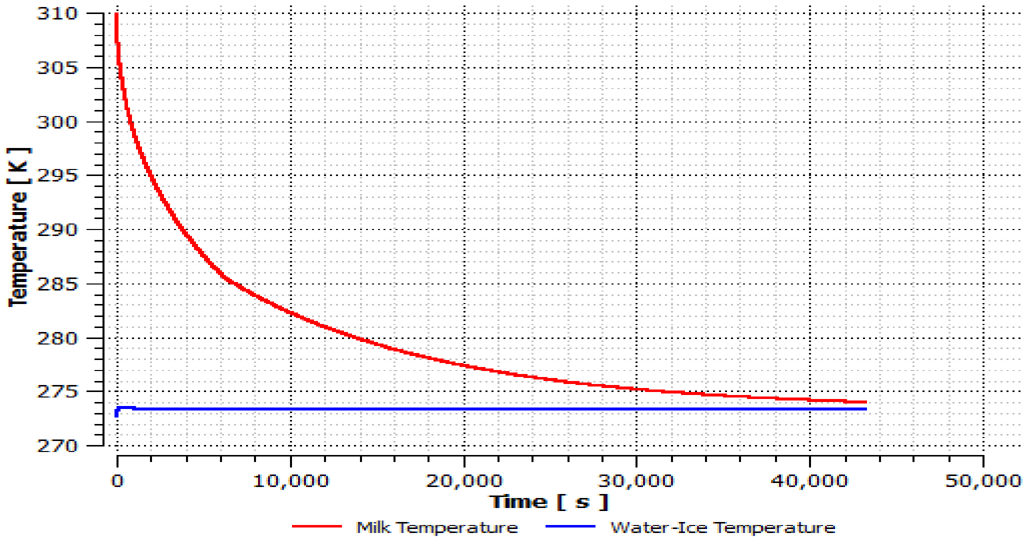


Figure 4.13: the water-ice and milk temperature while cooling the milk

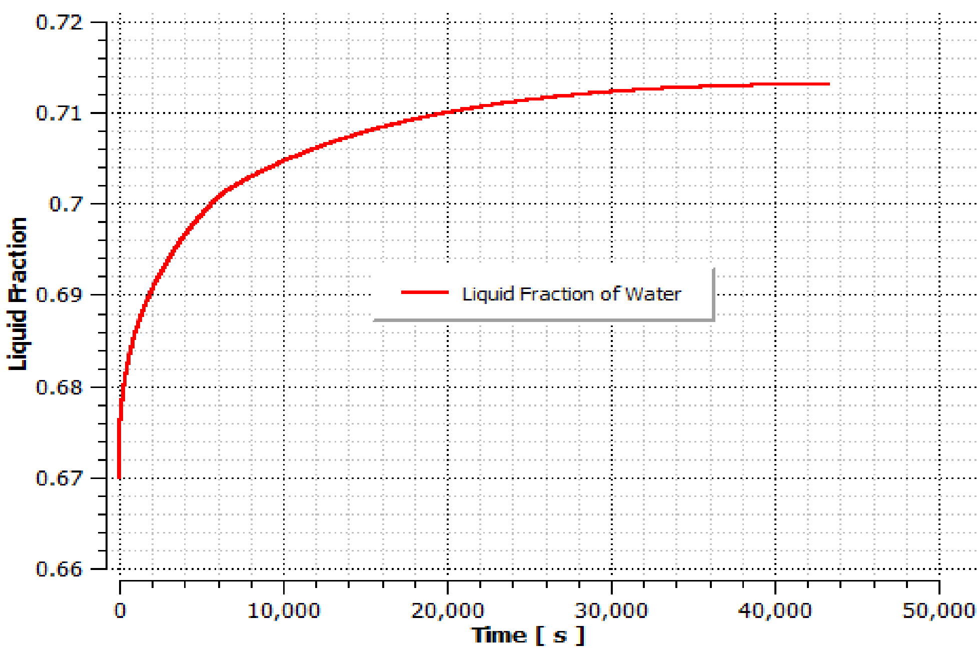


Figure 4.14: Ice melting rate in water-ice chamber

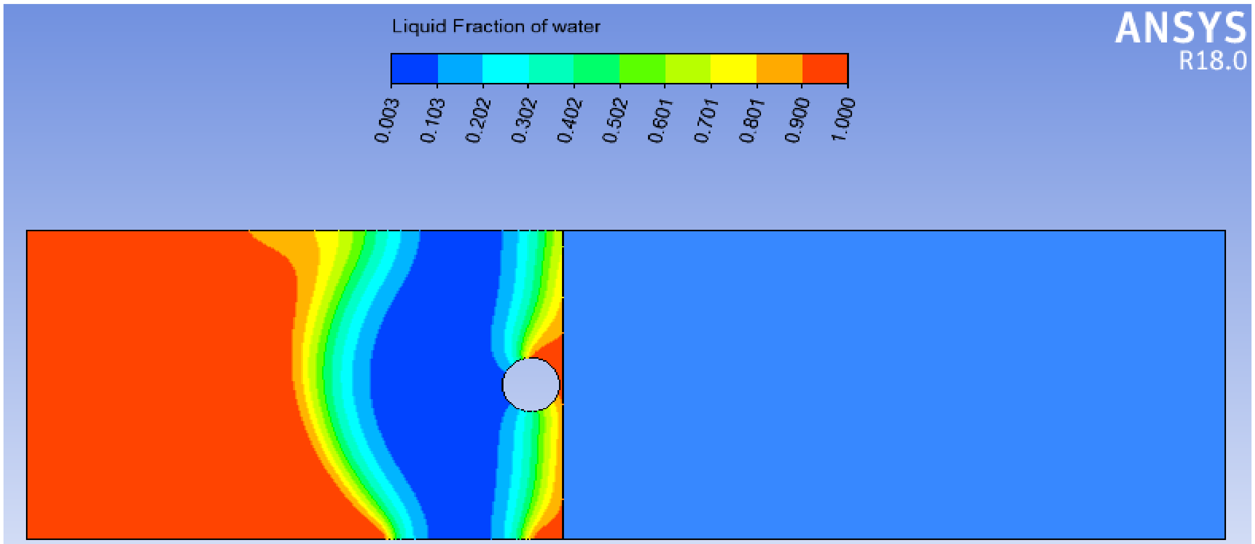


Figure 4.15: Liquid fraction contour in water Chamber as it cools Milk

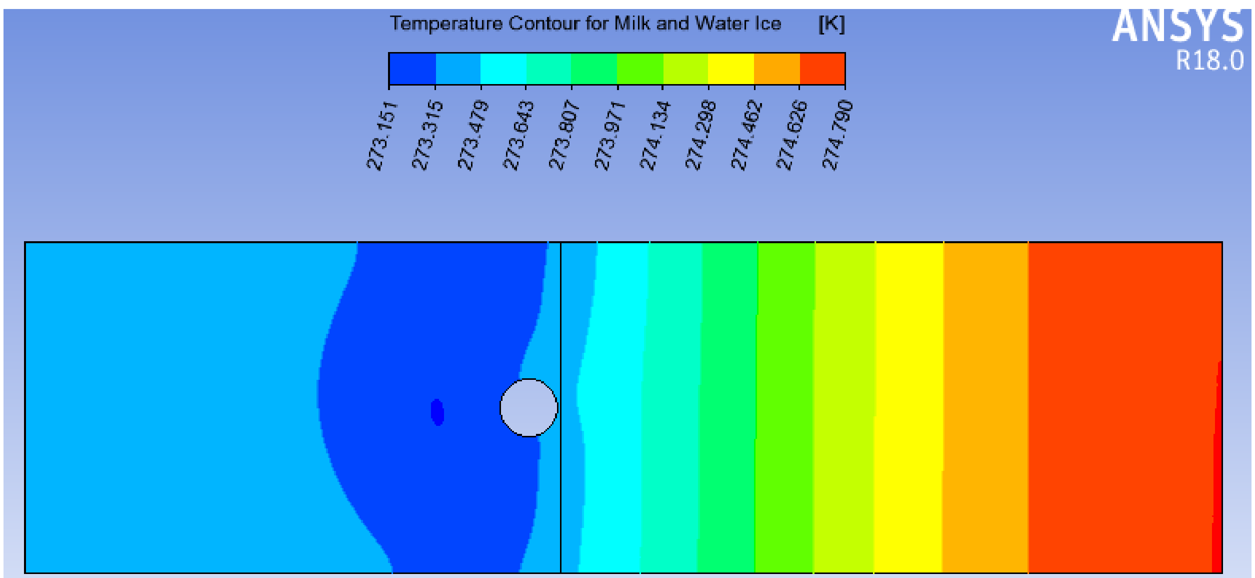


Figure 4.16: temperature contours in water-ice and milk while cooling milk

CHAPTER FIVE

5. Experimental testing of the system

To construct the experimental set up (Figure 5.6) each components of the solar refrigerator was designed, manufactured and selected for 8 liters capacity refrigerator. In this work 8 liters of capacity prototype (5liters milk and 3 liters of water space) was used for simulation and testing purpose.

5.1 Main components of PV direct coupled refrigerator:

- ✓ Main Body (water-ice chamber with evaporator coil, milk chamber)
- ✓ Thermal insulator
- ✓ Variable Dc refrigerator compressor with control unit
- ✓ Solar panel
- ✓ Condenser
- ✓ Expansion or throttling device

5.1.1 Manufacturing of components

The water- ice chamber was manufactured using 1mm thick sheet metal. The sheet metal was cut and rolled in to circular cylinder and having volume of $3m^3$. The water-ice chamber is thermally insulated all the surfaces from the surrounding with 50mm thick polyethylene foam.

Figure 5.1 water-ice and milk-chamber

The milk-chamber was manufactured using 1mm thick aluminum cylinder and manufactured using sand casting with a capacity of 5 liters. Around a circular aluminum cylinder 8mm diameter cupper tube turned around a long the height of cylinder in which the refrigerant is flowing through as shown Figure 5-2.

Figure 5.2 Refrigerator component contains of DC compressor, evaporator and condenser tubes

5.2 Materials used to conduct experiment are:

- Temperature measuring data logger was used to measure water-ice and milk temperatures and it is K type thermocouples

Figure 5.3 Temperature Data logger

- Clump meter and Multi meter were used to measure the power consumption of compressor

Figure 5.4 Photograph view of clamp meter

Figure 5.5 Photograph view of multimeter

- DC power supply

  It is required to conduct virtually simulate solar panel. The PV output for specific site was estimated. According to this the input power the compressor is varied. Matching of PV output with dc power supply is carried out by varying the speed of the dc-compressor with the help of compressor control

Figure 5.6 Dc power supply

- Refrigerant Charging device

Charging of the refrigeration system with a specific amount of refrigerant was achieved with the help of R134a two-way valve, double headed low and high side pressure gauge with a hand wheel.

Figure 5.7 the assembly of the experimental set up

5.3Testdata measurement procedure

5.3.1 Ice formation during the day time (solidification process)

- ✓ 3 L of water is filled in water-ice chamber initially its temperature was 4℃ and milk chamber kept empty.
- ✓ According to the selected solar panel size and site irradiance, the compressor rpm is varied to simulate the PV panel power supply virtually using DC compressor control unit.
- ✓ By using temperature data logger, the water-ice and milk chamber temperature was recorded for every 15 minutes and this was carried on for five hours (PV panel able to run the compressor)

5.3.2 Melting of ice to cool 5L milk during night time (melting process of ice)

This time the compressor is not running and the milk chamber is filled with 5L of milk with milking time temperature (37°C). Using temperature data logger the water-ice and milk temperature was recorded for every 15 minutes for 12 hours period.

5.4 Experimental Results and discussions

Figure 5.8 below shows the temperature of water-ice during solidification process. The water temperature was reduced from$4°\mathrm{C}\ to - 12.5°\mathrm{C}$. And 3 liters of water was solidified within 5 hours of running the refrigerator. It was also observed that the milk chamber temperature$-13°\mathrm{C}$.

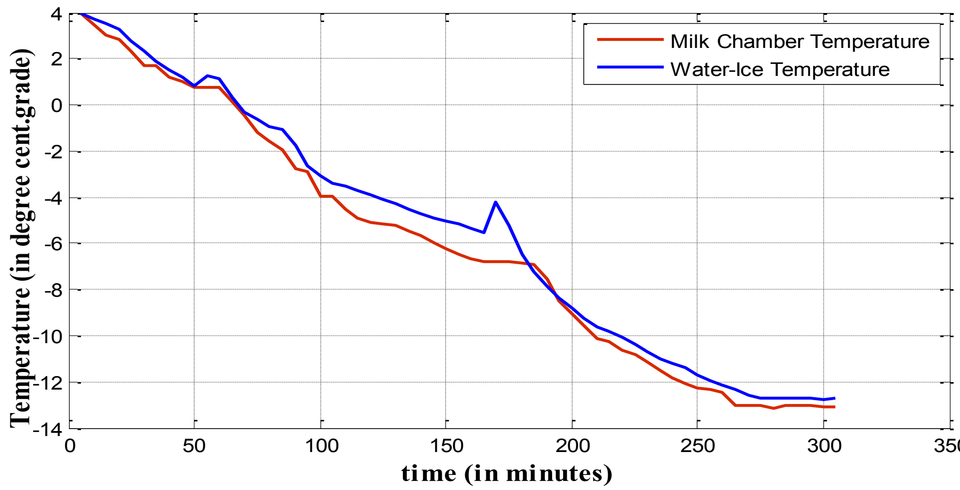


Figure 5.8 Solidification of water

During ice-water discharging process the milk was cooled from its initial temperature of $37°\mathrm{C}\ to 4°\mathrm{C}$ within time period of 4hours. And maintain the milk temperature $4°\mathrm{C}\ to 0°\mathrm{C}$ for 12 hours as shown Figure 5.9.

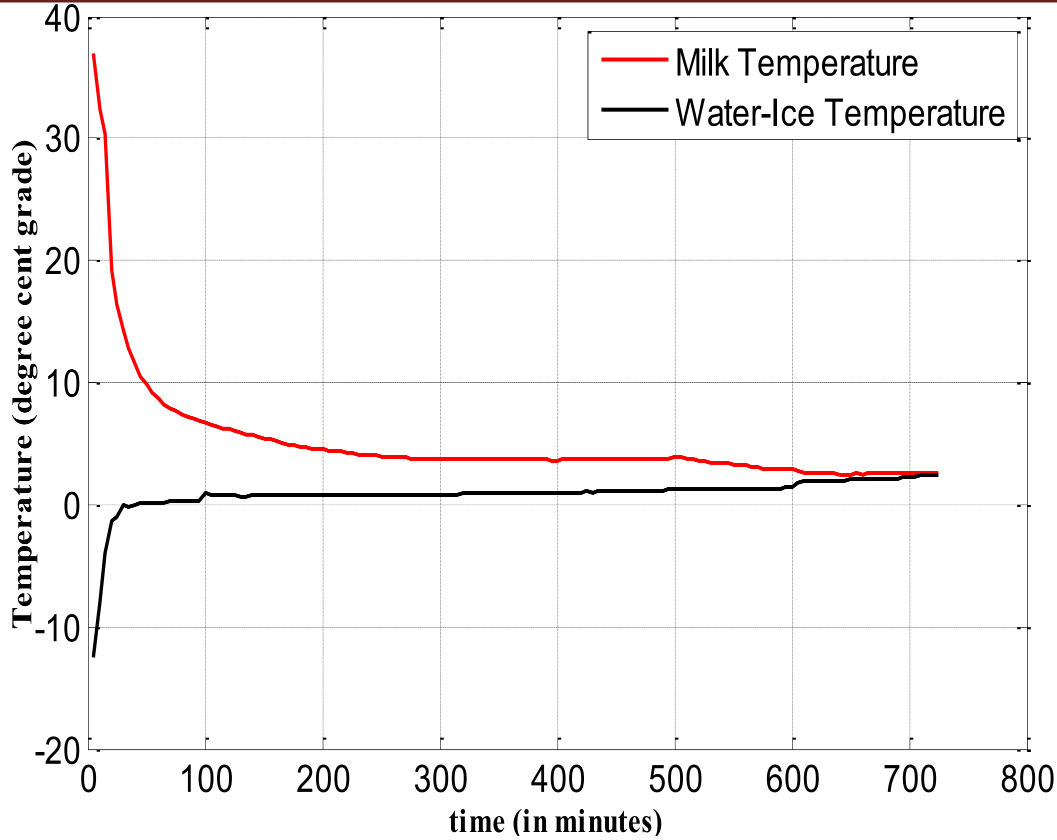


Figure 5.9 Milk cooling using ice-water (PCM)

5.5 Comparison of the CFD and the experimental result

5.5.1 The results of water-ice temperature for solidification process

In both experimental and CFD simulation, the solidification and discharging process of water-ice (PCM) was conducted. Milk and water-ice temperature were recorded in every 5 minutes PCM melting for 12 hours and solidification processes for 5 hours respectively and drawn in graphs as shown below in Figure 5-10 and Figure 5-11. It is observed in that there is a good agreement between experimental and computation results with the maximum percentage deviation of 3.6% for milk cooling and 6% for water-ice temperature.

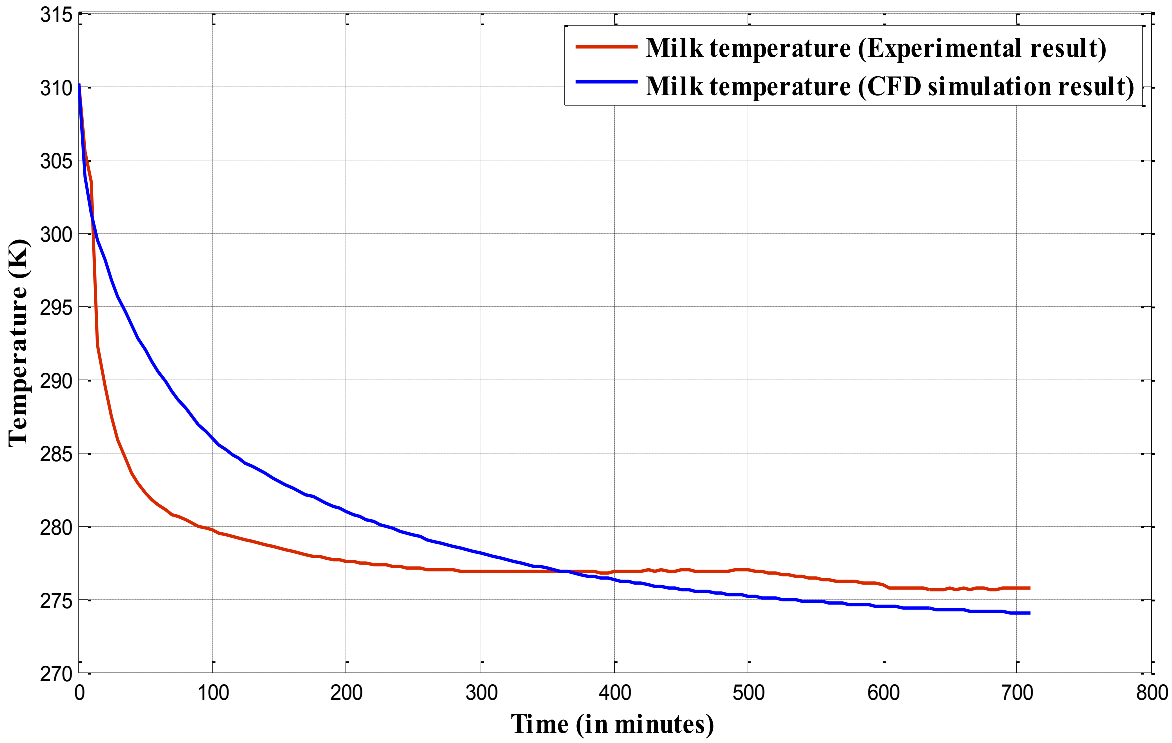


Figure 5.10 Comparisons of milk temperature during melting process of water-ice

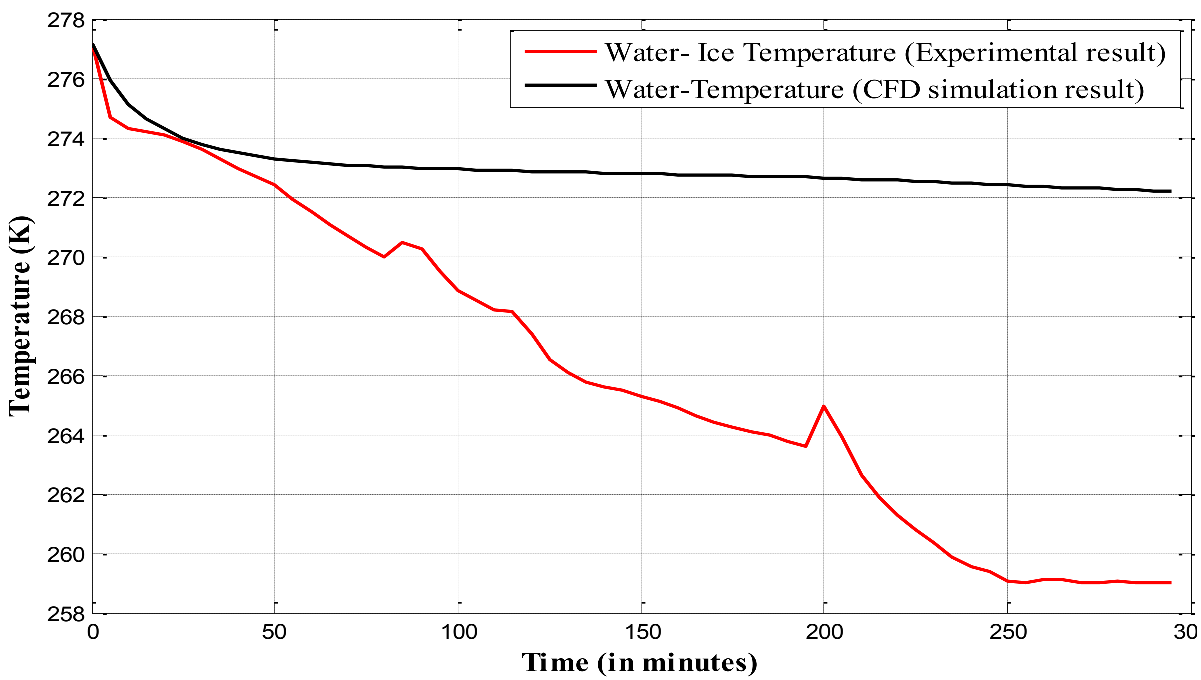


Figure 5.11 Comparisons of water-ice temperature during solidification process

5.6 Coefficient of performance of the system (COP)

COP of the system is expressed as:

$$COP_{system} = \frac{\dot{Q}_{ref}}{W_c} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.(4.1)$$

Where $\dot{Q}_{ref}$ is removal rate of refrigerant, W_c is compressor work

The rate of heat removed by the refrigerant at the evaporator is also given by.

$$\dot{Q}_{ref} = U_{oev}A_{ev}\left(T_{(water-ice)} - T_{ev}\right) \ldots\ldots\ldots\ldots(4.2)$$

$Where;$ $\mathrm{A_{ev}}$ Evaporator heat transfer area, $T_{(water-ice)}$ is temperature of water − ice ,

$\mathrm{T_{ev}}$ is average evaporator temrature (−12.5℃)

$$W_c = \eta_c * P_{peak}\, X\, \frac{I_p}{I_r}$$

Whereη_c $efficiency\ of\ compressor$(90%), $P_{peak} = 140W, I_r = {}^{1000W}/_{m^2}$

I_p $site\ radition \geq 315\mathrm{W}/m^2$

The overall heat transfer r coefficient ($U_{oev} = 129\mathrm{W}/m^2\mathrm{K}$) between copper tube and water-ice is obtained while sizing evaporator in design section. The water temperature is obtained in the simulation result and it varies from 277.15 K to 271.78K. Using MATLAB cod in appendix F COP of the system is obtained and plotted in Figure 5.13.

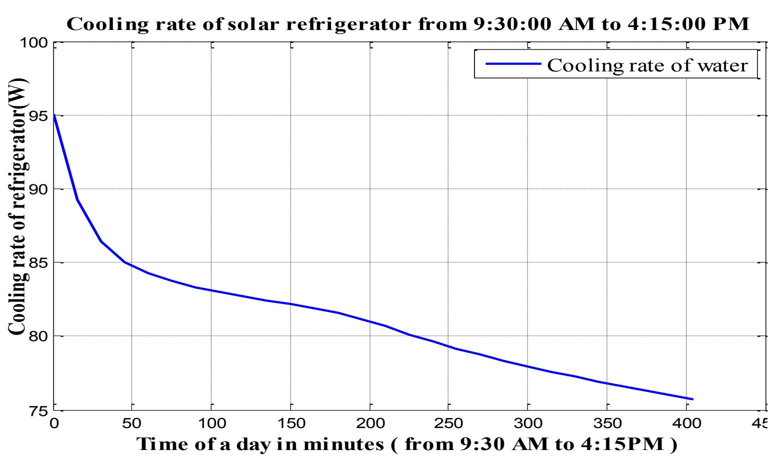


Figure 5.12 Cooling rate of water

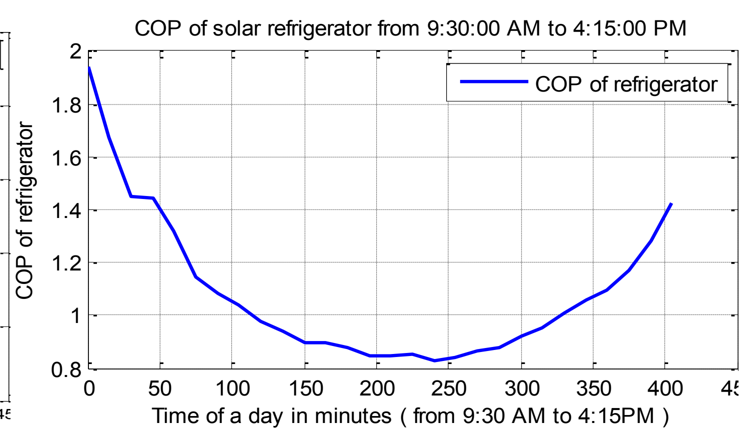


Figure 5.13 COP of the system

CHAPTER SIX

7. Conclusion and Recommendation

7.1 Conclusion

A two-dimensional axisymmetric model of evaporator section which consisting of water-ice and milk chamber of solar refrigerator and using water as the ice as a thermal storage for directly milk cooling at night time was simulated in ANSYS Fluent. The solidification and melting process of ice were conducted by specifying a constant average temperature of -12.5^{0}C on the copper tube wall according to the evaporation temperature of refrigerant (R134a).

For the selected compressor and PV panel size the amount of ice formation and Average water – ice (PCM) temperature was investigated for solidification processes. It was found that water-ice temperature of 271.78K (-1.37^{0}C) and 1 kg of ice was formed with 7 hours compressor running time. Also it was observed that during the discharging process the milk temperature was reduced from 310K to 279.6K within four hours and further reduced to 274.007K after 12 hours; however it should be 277.15K within 4 hours duration as per FAO standard. This can be achieved by steering the milk as it is cooled. 1kg of ice at average 271.78K is sufficient to preserves 5 kg of milk for 12 hours and more.

Solar PV variable dc compressor refrigerator model was produced as shown figure 5-5.The PV system was virtually simulated. According to the site solar irradiance the output PV panel was analyzed and dc equivalent of this amount supplied dc variable speed refrigerator compressor. It was observed in experiment that the average water-ice temperature was -12.5^{o}C. In the discharging process the Milk temperature was reduced 310K to 274K within 12 hours. And it was observed that the milk temperature 0℃ to 4℃ within time period of 4hours.

7.2 Recommendation

- Estimate and enhance operating efficiency while the system is running PV panel
- Based on the experimental result, after cooling the milk for 12 hours the ice in water-chamber almost melted to liquid water. But in the simulation, liquid fraction at the end simulation was 0.715. So, it is important to improve the efficiency of the thermal insulation.
- Perform 3D model CFD modeling of water –ice further work should have be done to run the refrigerator directly using PV panel power source (on field experiment).
- Temperature contours of milk cooling, it is observed that there is temperature gradient. Milk should be periodically steered so that it could have uniform temperature to avoid uneven cooling.

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Appendix A: Table 1: four years average daily solar radiation for a month of August (Adama)

Table 1: four years average daily solar radiation for a month of August

Hours of A day	2015		2014		2013		2012		average	average
	w/m2	temp	w/m2	temp	w/m2	temp	w/m2	temp	w/m2	temp
12:00 AM	0.58	19.46	0.55	18.83	0.45	17.69	0.51	19.14	0.52	18.78
12:15 AM	0.58	19.35	0.54	18.68	0.43	17.30	0.51	19.07	0.51	18.60
12:30 AM	0.58	19.23	0.54	18.49	0.38	16.76	0.51	18.99	0.50	18.37
12:45 AM	0.58	19.22	0.55	18.41	0.44	17.12	0.50	18.85	0.52	18.40
1:00 AM	0.57	19.05	0.54	18.35	0.45	17.16	0.51	18.72	0.52	18.32
1:15 AM	0.59	18.89	0.54	18.23	0.45	17.05	0.50	18.70	0.52	18.22
1:30 AM	0.58	18.81	0.54	18.17	0.43	16.95	0.50	18.52	0.51	18.11
1:45 AM	0.57	18.71	0.53	18.15	0.44	16.80	0.50	18.43	0.51	18.02
2:00 AM	0.57	18.61	0.54	18.10	0.42	16.36	0.50	18.46	0.51	17.88
2:15 AM	0.56	18.46	0.54	18.02	0.38	16.15	0.50	18.47	0.50	17.78
2:30 AM	0.58	18.32	0.55	17.88	0.44	16.74	0.50	18.44	0.52	17.85
2:45 AM	0.57	18.24	0.54	17.78	0.45	16.68	0.50	18.41	0.52	17.78
3:00 AM	0.56	18.16	0.55	17.77	0.44	16.63	0.51	18.28	0.51	17.71
3:15 AM	0.57	18.06	0.52	17.80	0.43	16.58	0.50	18.16	0.51	17.65
3:30 AM	0.57	17.99	0.54	17.78	0.43	16.56	0.50	18.05	0.51	17.60
3:45 AM	0.58	17.91	0.55	17.71	0.42	16.13	0.50	18.01	0.51	17.44
4:00 AM	0.57	17.84	0.54	17.64	0.38	16.12	0.51	17.94	0.50	17.38
4:15 AM	0.57	17.85	0.53	17.54	0.45	16.61	0.50	17.81	0.51	17.45
4:30 AM	0.56	17.80	0.53	17.54	0.45	16.53	0.50	17.62	0.51	17.37
4:45 AM	0.57	17.81	0.54	17.48	0.44	16.60	0.51	17.58	0.51	17.37
5:00 AM	0.56	17.84	0.54	17.39	0.44	16.51	0.50	17.46	0.51	17.30
5:15 AM	0.58	17.81	0.53	17.33	0.58	16.42	0.50	17.35	0.55	17.23
5:30 AM	0.57	17.73	0.53	17.20	1.80	15.97	0.51	17.24	0.85	17.04
5:45 AM	0.57	17.59	0.54	17.15	8.87	16.18	0.50	17.17	2.62	17.02
6:00 AM	0.60	17.47	0.57	17.17	26.85	16.64	0.53	17.11	7.14	17.10
6:15 AM	1.61	17.33	1.46	17.17	50.11	16.59	1.30	16.98	13.62	17.02
6:30 AM	6.85	17.27	5.69	17.11	81.50	16.99	5.34	16.89	24.84	17.06
6:45 AM	19.75	17.31	16.11	17.12	109.48	17.29	14.46	16.79	39.95	17.13
7:00 AM	39.95	17.57	30.20	17.20	128.51	17.25	28.58	16.93	56.81	17.24
7:15 AM	63.98	17.90	51.95	17.35	179.35	17.44	46.09	17.07	85.34	17.44
7:30 AM	93.40	18.23	77.13	17.57	212.13	18.06	56.20	17.28	109.72	17.78
7:45 AM	124.96	18.59	107.08	17.84	239.73	18.54	72.64	17.52	136.10	18.12
8:00 AM	163.80	18.95	149.28	18.13	240.44	18.23	93.09	17.57	161.65	18.22
8:15 AM	205.47	19.29	182.64	18.35	271.79	18.92	112.49	17.67	193.10	18.56
8:30 AM	260.85	19.66	210.56	18.49	331.75	18.86	134.54	17.80	234.43	18.70

8:45 AM	314.76	20.00	240.66	18.69	392.40	19.57	146.81	17.98	273.66	19.06
9:00 AM	370.28	20.35	269.26	18.97	460.92	19.87	144.70	18.20	311.29	19.35
9:15 AM	397.05	20.62	338.77	19.32	471.58	19.68	154.40	18.34	340.45	19.49
9:30 AM	452.05	20.95	399.48	19.70	526.40	20.63	176.78	18.66	388.68	19.98
9:45 AM	467.60	21.20	427.10	19.96	568.81	21.03	231.44	18.87	423.74	20.27
10:00 AM	502.16	21.50	479.42	20.29	542.97	21.34	369.98	19.20	473.63	20.58
10:15 AM	549.04	21.86	524.15	20.57	369.14	21.64	425.81	19.40	467.04	20.87
10:30 AM	599.26	22.20	541.17	20.84	412.79	21.72	471.84	19.70	506.26	21.11
10:45 AM	649.46	22.56	576.12	21.07	552.16	21.98	542.96	20.03	580.17	21.41
11:00 AM	657.36	22.77	612.46	21.43	634.91	22.10	557.98	20.30	609.27	21.50
11:15 AM	696.22	23.12	622.02	21.67	659.12	22.40	581.72	20.58	633.32	21.79
11:30 AM	731.90	23.52	661.98	21.91	696.94	22.72	616.18	20.84	670.02	22.09
11:45 AM	766.76	23.87	702.65	22.30	734.71	23.09	624.56	21.05	697.99	22.41
12:00 PM	803.18	24.13	721.96	22.56	762.57	23.35	658.22	21.31	727.79	22.67
12:15 PM	763.98	24.29	739.51	22.65	751.75	23.47	666.45	21.59	723.31	22.84
12:30 PM	746.32	24.48	752.11	22.80	749.21	23.64	718.03	21.89	738.82	23.06
12:45 PM	758.50	24.72	770.05	23.04	764.28	23.88	762.16	22.23	763.57	23.33
1:00 PM	752.37	24.97	787.90	23.40	770.14	24.19	727.44	22.38	755.91	23.58
1:15 PM	742.01	25.13	757.81	23.57	749.91	24.35	732.55	22.61	744.13	23.77
1:30 PM	765.08	25.44	758.99	23.78	762.04	24.61	772.77	22.89	765.62	24.04
1:45 PM	727.85	25.56	756.79	24.03	742.32	24.80	765.74	23.13	750.12	24.24
2:00 PM	711.80	25.75	721.17	24.20	716.49	24.97	737.97	23.25	723.65	24.40
2:15 PM	709.55	25.97	698.56	24.38	704.05	25.18	719.10	23.41	709.07	24.59
2:30 PM	671.00	26.11	657.34	24.46	664.17	25.29	680.84	23.54	669.73	24.70
2:45 PM	640.12	26.24	621.93	24.58	631.03	25.41	674.01	23.64	645.35	24.82
3:00 PM	586.58	26.29	590.81	24.67	588.70	25.48	643.65	23.77	607.02	24.91
3:15 PM	566.85	26.34	565.45	24.74	566.15	25.54	595.79	23.79	576.03	24.96
3:30 PM	536.22	26.54	556.54	24.89	546.38	25.71	571.25	23.87	554.67	25.10
3:45 PM	521.45	26.63	515.28	24.95	518.36	25.79	513.51	23.84	516.75	25.14
4:00 PM	483.52	26.74	455.58	24.72	469.55	25.73	472.59	23.87	470.57	25.11
4:15 PM	421.35	26.65	429.34	24.68	425.35	25.66	412.42	23.76	421.04	25.03
4:30 PM	368.73	26.57	371.04	24.64	369.89	25.61	378.98	23.75	372.92	24.99
4:45 PM	306.83	26.39	303.52	24.44	305.17	25.41	325.05	23.67	311.80	24.83
5:00 PM	263.57	26.28	252.16	24.29	257.87	25.28	276.76	23.57	264.17	24.71
5:15 PM	210.63	25.97	199.78	24.07	205.21	25.02	221.00	23.43	210.47	24.49
5:30 PM	153.71	25.48	158.00	23.84	155.85	24.66	178.10	23.25	163.27	24.19
5:45 PM	109.83	24.99	113.76	23.53	111.79	24.26	128.93	22.93	117.51	23.82
6:00 PM	68.19	24.55	70.91	23.17	69.55	23.86	81.50	22.55	73.53	23.43
6:15 PM	38.36	24.03	36.86	22.65	37.61	23.34	41.81	22.06	39.01	22.92
6:30 PM	17.15	23.45	14.88	22.17	16.02	22.81	15.13	21.55	15.72	22.39
6:45 PM	3.83	22.70	3.76	21.69	3.80	22.20	3.71	21.08	3.77	21.83

7:00 PM	0.85	22.09	0.83	21.30	0.84	21.70	0.76	20.63	0.81	21.34
7:15 PM	0.59	21.86	0.56	20.97	0.58	21.42	0.50	20.35	0.55	21.06
7:30 PM	0.59	21.69	0.56	20.66	0.57	21.18	0.50	20.20	0.55	20.85
7:45 PM	0.60	21.43	0.54	20.29	0.57	20.86	0.51	20.09	0.55	20.60
8:00 PM	0.58	21.23	0.55	20.11	0.56	20.67	0.50	19.95	0.54	20.43
8:15 PM	0.59	21.06	0.53	20.02	0.56	20.54	0.50	19.87	0.54	20.32
8:30 PM	0.58	20.86	0.54	19.83	0.56	20.35	0.50	19.71	0.54	20.14
8:45 PM	0.58	20.76	0.54	19.75	0.56	20.25	0.50	19.61	0.54	20.04
9:00 PM	0.59	20.57	0.54	19.75	0.57	20.16	0.50	19.58	0.54	19.97
9:15 PM	0.58	20.35	0.56	19.68	0.57	20.02	0.50	19.44	0.55	19.82
9:30 PM	0.59	20.31	0.54	19.70	0.57	20.00	0.50	19.45	0.54	19.82
9:45 PM	0.59	20.11	0.54	19.60	0.56	19.85	0.50	19.38	0.54	19.70
10:00 PM	0.58	19.94	0.54	19.48	0.56	19.71	0.50	19.36	0.54	19.59
10:15 PM	0.59	19.85	0.54	19.43	0.56	19.64	0.50	19.26	0.54	19.51
10:30 PM	0.58	19.77	0.54	19.39	0.56	19.58	0.50	19.17	0.54	19.45
10:45 PM	0.59	19.71	0.55	19.26	0.57	19.48	0.50	19.16	0.55	19.38
11:00 PM	0.58	19.56	0.54	19.16	0.56	19.36	0.50	19.15	0.54	19.29
11:15 PM	0.59	19.49	0.55	19.04	0.57	19.27	0.50	19.15	0.55	19.23
11:30 PM	0.59	19.38	0.54	19.01	0.56	19.20	0.51	19.14	0.55	19.18
11:45 PM	0.59	19.43	0.53	18.98	0.56	19.20	0.50	19.15	0.54	19.19

Appendix B: Electronic control unit wiring diagram for MT35DC compressor

Controller

This compressor motor runs on DC brushless motor. When you use this controller, you must be properly connected with compressor. Connection as below：

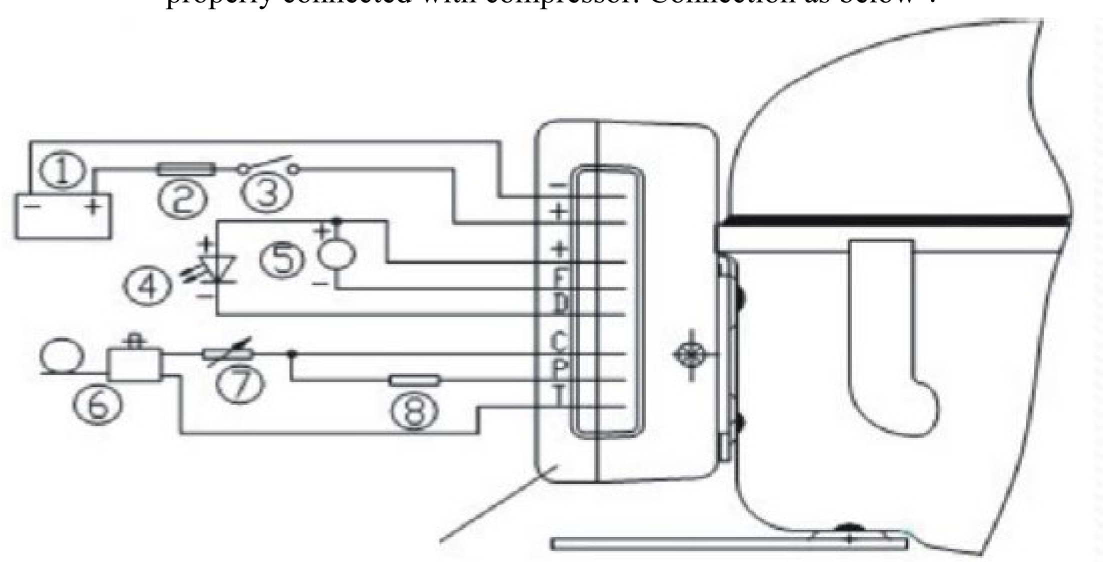

Figure 3.7: Electronic control unit wiring diagram for MT35DC compressor

1. Controller must be directly connected with DC power supply ①, connect the plug to + and minus to **-**, otherwise the electronic unit won't work.
2. For protection of installation, fuse② must be connected with positive wire and close to power supply .15A fuse for 12VDC, 7.5A fuse for 24VDC are recommended.
3. If a main switch ③ is used, the rated current shall be min 20A.
4. In order to avoid excessive voltage change caused by the controller to protect, don't add other circuit in the circuit.
5. A 10mA light emitting diode④ can be connected between the terminals + and D. The number of flashed depends on what kind of operational error was recorded. Each flash will last 1/5second.If the launch failure each interval 60 seconds to try to start again, till launch successfully.

Table 3.4: Compressor controller fault indicator

Flash time	fault type
5	controller thermal protection
4	minimum speed errors
3	compressor launch failure
2	an over current protection
1	Battery voltage protection

6. Radiating fan ⑤ is connected between + and F. Since the output voltage is always regulated to 12V, a 12V fan must be used for both 12V and 24V power supply systems. Fan current protection is 1A, it is recommended to use no more than 5W fan.
7. Thermostat switch ⑥ is connected between C and T. It control compressor start and stop.
8. Speed regulation resistor ⑦ is connected between C and T. It adjusts resistor value to change the speed of compressor. Compressor speed range is 2000rpm～3500rpm, the relation of speed regulation resistor ⑦ and speed as below:

Appendix C: Compressor speed adjusting resistance value and control unit power protection resistor values

Table 3.5: Compressor speed adjusting resistance value

Speed, rpm	RESISTANCE (Ohm)	Current, (milli Ampere)
2000	0	5
2100	51	4.8
2200	100	4.6
2300	150	4.4
2400	200	4.2
2500	277	4
2600	330	3.8
3000	692	3
3100	816	2.8
3200	963	2.6
3300	1137	2.4
3400	1331	2.2
3500	1523	2
Stop		

Power protection resistor ⑧ is connected between C and P. Different resistor value change battery voltage protection setting. The relation of battery voltage protection resistor setting and power protection resistor ⑧ is as below:

Table 3.6: Compressor control unit power protection resistor value

(K Ω) power Protection resistor	12V protection	12V minimum work voltage	12V ma□imum work voltage	24V protection voltage	24V minimum work voltage	24V maximum work voltage
0	9.6	10.9	17	21.3	22.7	31.5
1.6	9.7	11.0	17	21.5	22.9	31.5
2.4	9.9	11.1	17	21.8	23.2	31.5
3.6	10	11.3	17	22	23.4	31.5
4.7	10.1	11.4	17	22.3	23.7	31.5
6.2	10.2	11.5	17	22.5	23.9	31.5
8.2	10.4	11.7	17	22.8	24.2	31.5
11	10.5	11.8	17	23	24.5	31.5

14	10.6	11.9	17	23.3	24.7	31.5
18	10.8	12	17	23.6	25	31.5
24	10.9	12.2	17	23.8	25.2	31.5
33	11	12.3	17	24.1	25.5	31.5
47	11.1	12.4	17	24.3	25.7	31.5
82	11.3	12.5	17	24.6	26	31.5
220	9.6	10.9				31.5

Controller automatically adjusts to the applied voltage. If power voltage is less than 17V, the controller will work in 12VDC system. If power voltage is more than 17V, the controller will work in 24VDC system. If a 220KΩ resistor is connected between C and P, compressor work voltage range will be extended to 9.6V～31.5V.

Appendix D: V-I and P-V characteristics the selected PV panel

```
% V-I and P-V characteristics the selected PV panel for different solar radiation and at 25degree cent%%
 clear,clc
Iscn=8.1;%nominal short circuit current
Voc_n=22.1;%nominal open circuit voltage
KI=0.055;%Isc current temprature coeficient
KV=-.35;% Voc temprature coeficient
K=1.3806503e-23;%Boltzman constant
q=1.60217646e-19;%charge of electron
a=1.3;% diod constant
Tn=273+25; %nominal temprature
Gn=1000; % nominal solar radiation
Ns=36;% number of cells in serous
Eg=1.12;% energy band gap of semiconductor
Vmmp=17.6;
Immp=7.95;
Pmax=Vmmp*Immp;
```

```
P1=Pmax+2;% a value to limit the x axis of the graph
Vtn=Ns*K*Tn/q;%thermal voltage of array
Rs=0;%intialization of iteration;
Iomn=Iscn/(exp(Voc_n/(a*Vtn))-1);% diod nominal saturation current
Rp=(Vmmp+Rs*Immp)/(Iscn-Iomn*(exp((Vmmp+Rs*Immp)/(Vtn*a))-1)-Immp);
In=1;% to initiate the iteration since the I(v)function is implicit
I=1;
format long e
for i=1:20000;
   Rs=Rs+0.0001;
   Ipvn=(Rp+Rs)*Iscn/Rp;
   Rp=(Vmmp+Rs*Immp)/(Ipvn-Iomn*(exp((Vmmp+Rs*Immp)/(Vtn*a))-1)-Immp);
   for j=1:500;
      In=Ipvn-Iomn*(exp((Vmmp+(In*Rs))/(Vtn*a))-1)-((Vmmp+(Rs*In))/Rp);
   end
   Pn=In*Vmmp;
   error=abs(Pn-Pmax);
   V=0:.01:Voc_n;
   I=Ipvn-Iomn*(exp((V+(I*Rs))/(Vtn*a))-1)-((V+(Rs*I))/Rp);
   P=V.*I;
   Pmaxx=max(P);
   err=abs(Pmaxx-Vmmp*In);
   if error<.001 && err<.002
      break
   end
end
G=0;
T=Tn;
Ipvn=(Rp+Rs)*Iscn/Rp;
for l=1:10
   G=G+100;
```

```
    Ipv=(Ipvn+KI*(T-Tn))*(G/Gn);
    V=0:.01:Voc_n;
    for ii=1:1500;
        I=Ipv-Iomn*(exp((V+(I*Rs))/(Vtn*a))-1)-((V+(Rs*I))/Rp);
    end
    P=V.*I;
    figure(1)
    plot(V,I)
    title('V-I characteristics at 25 degree cent')
    xlabel('Voltage')
    ylabel('current')
    grid on
    hold on
    axis([0 Voc_n 0 9])
    figure(2)
    plot(V,P)
    hold on
    axis([0 Voc_n 0 P1])
    title('V-P characteristics at 25 degree cent')
    grid on
    xlabel('Voltage')
    ylabel('power')
end
```

Appendix E: Refrigerator compressor operating point at compressor speed for selected PV Panel

```
%% Refrigerator compressor operating point at compressor speed 2000RPM,2500 RPM, 3000 RPM, 3500 RPM for the selected PV panel and site %%
clear,clc
Iscn=8.1; %nominal short circuit current
Voc_n=22.1;%nominal open circuit voltage
KI=0.055; %Isc current temperature coefficient
KV=-.35; % Voc temperature coefficient
K=1.3806503e-23;%Boltzman constant
q=1.60217646e-19;%charge of electron
a=1.3;% diode constant
Tn=273+25; %nominal temperature
```

```
Gn=1000; % nominal solar radiation

Ns=36;% number of cells in serous

Eg=1.12;% energy band gap of semiconductor

Vmmp=17.6;

Immp=7.95;

Pmax=Vmmp*Immp;

P1=Pmax+2;% a value to limit the x axis of the graph

Vtn=Ns*K*Tn/q;%thermal voltage of array

Rs=0;%initialization of iteration;

Iomn=Iscn/(exp(Voc_n/(a*Vtn))-1);% diode nominal saturation current

Rp=(Vmmp+Rs*Immp)/(Iscn-Iomn*(exp((Vmmp+Rs*Immp)/(Vtn*a))-1)-Immp);

In=1;% to initiate the iteration since the I(v)function is implicit

I=1;
format long e
for i=1:20000;
   Rs=Rs+0.0001;
   Ipvn=(Rp+Rs)*Iscn/Rp;
   Rp=(Vmmp+Rs*Immp)/(Ipvn-Iomn*(exp((Vmmp+Rs*Immp)/(Vtn*a))-1)-Immp);
   for j=1:500;
      In=Ipvn-Iomn*(exp((Vmmp+(In*Rs))/(Vtn*a))-1)-((Vmmp+(Rs*In))/Rp);
   end
   Pn=In*Vmmp;
   error=abs(Pn-Pmax);
   V=0:.01:Voc_n;
   I=Ipvn-Iomn*(exp((V+(I*Rs))/(Vtn*a))-1)-((V+(Rs*I))/Rp);
   P=V.*I;
   Pmaxx=max(P);
   err=abs(Pmaxx-Vmmp*In);
   if error<.001 && err<.002
      break
   end
end
T=Tn;
Ipvn=(Rp+Rs)*Iscn/Rp;
       G=315;
        Ipv=(Ipvn+KI*(T-Tn))*(G/Gn);
   V=0:.01:Voc_n;
```

```
for ii=1:1500;
  I=Ipv-Iomn*(exp((V+(I*Rs))/(Vtn*a))-1)-((V+(Rs*I))/Rp);
end
Pa=V.*I;
 plot(V,I)
hold on
     G=383;
      Ipv=(Ipvn+KI*(T-Tn))*(G/Gn);
    for ii=1:1500;
    I=Ipv-Iomn*(exp((V+(I*Rs))/(Vtn*a))-1)-((V+(Rs*I))/Rp);
    end
Pb=V.*I;
plot(V,I)
hold on
     G=460;
      Ipv=(Ipvn+KI*(T-Tn))*(G/Gn);
    for ii=1:1500;
   I=Ipv-Iomn*(exp((V+(I*Rs))/(Vtn*a))-1)-((V+(Rs*I))/Rp);
    end
     Pc=V.*I;
plot(V,I)
hold on
     G=542;
      Ipv=(Ipvn+KI*(T-Tn))*(G/Gn);
   for ii=1:1500;
    I=Ipv-Iomn*(exp((V+(I*Rs))/(Vtn*a))-1)-((V+(Rs*I))/Rp);
   end
Pd=V.*I;
plot(V,I)
title('V-I characteristics at 25 degree cent')
xlabel('Voltage')
ylabel('current')
grid on
hold on
Iex1=47.69./V;
 plot(V,Iex1)
hold on
axis([0 Voc_n 0 9])
hold on
Iex2=57.69./V;
```

```
   plot(V,Iex2)
 hold on
   axis([0 Voc_n 0 9])
   hold on
   Iex3=67.69./V;
   plot(V,Iex3)
 hold on
   axis([0 Voc_n 0 9])
   hold on
   Iex4=78.46./V;
   plot(V,Iex4)
 hold on
   axis([0 Voc_n 0 9]
```

Appendix F: Mat Lab Code for COP, Cooling Rate, Compressor in Put Power, as Solar Refrigerator Solidifies Water

```
%% mat lab code for COP, Cooling rate, compressor in put power As solar
refrigerator solidifies water %%

Te =273.15-12.5;%% average evaporator temperature
Tw= [274.6299495    273.7824087 273.3666753 273.1580349 273.0419337
272.9666435
    272.9075571 272.8576335 272.8141202 272.7731616 272.7346686 272.6985529
    272.6518033 272.589268  272.5164207 272.4361234 272.3623451 272.2950559
    272.2310489 272.1726268 272.1171977 272.0634189 272.0128152 271.9647052
    271.9174943 271.871771  271.8282713 271.7856961];%% water-ice
temperature
Ae=0.0527;%% effective heat transfer area of evaporator
Uo=129;%% overall heat transfer coeficient b/n water and copper tube
Qref=Uo*Ae*(Tw-Te);%% Qref cooling rate of refrigerator
Wc=[48.97335484 53.39119258 59.67786774 58.84644644 63.78922742 73.10187422
    76.76773548 79.79837419 84.42257419 87.94691613 91.70103871 91.13756129
    93.09110323 96.21007742 95.24407742 93.75985161 96.46749677 94.51571613
    91.17956129 89.34294194 84.3855871  81.31457419 76.48389677 72.58006452
    69.88867742 65.11016129 59.29126452 53.05087742];%% power supply to
compressor
COP=Qref./Wc; %% COP coeficient of performance of refrigerator
T=(0:15:405);%% T refrigerator running time in minutes
figure(1)
plot(T,COP)
ylabel('COP of refrigerator')
xlabel('Time of a day in minutes ( from 9:30 AM to 4:15PM )')
title('COP of solar refrigerator from 9:30:00 AM to 4:15:00 PM')
legend('COP of refrigerator')
grid on
figure(2)
plot(T,Qref)
ylabel('Cooling rate of refrigerator')
```

```
xlabel('Time of a day in minutes ( from 9:30 AM to 4:15PM )')
title('Cooling rate of solar refrigerator from 9:30:00 AM to 4:15:00 PM')
legend('Cooling rate ')
grid on
figure(3)
plot(T,Wc)
ylabel('Power Supply to compressor')
xlabel('Time of a day in minutes ( from 9:30 AM to 4:15PM )')
title('work input to compressor refrigerator from 9:30:00 AM to 4:15:00 PM')
legend('power supply to compressor')
grid on
```