SOLAR SPACE HEATING SYSTEM INTEGRATED WITH THERMAL ENERGY STORAGE

A Final Year Project Report

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In Partial Fulfillment

of the Requirements for the Degree of

Bachelors of Mechanical Engineering

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ABSTRACT

Pakistan is a country with diverse environmental conditions and seasons. Based on its geography, Pakistan has a severe winter season that lasts for 3 to 4 months. The northern areas of Pakistan suffer from extreme weather conditions for even longer. Normally, during the winter season, population of Pakistan relies on gas and electric heaters to provide them with comfortable environment to live. But the natural gas supply is not consistent rather scarce and the electricity is quite expensive as Pakistan is facing electricity deficit. In northern and under-developed areas, people still do not have access to natural gas and electricity and they are compelled to use fossil fuels like wood for such domestic purposes. However, there is a huge potential of renewable energy available which is an attractive alternative in order to overcome this difficulty.

Our project aims for the effective utilization of one of the inexhaustible energy sources available in nature, i.e. Sunlight, for domestic space heating purpose during winters. Our project addresses the issue of energy deficit mentioned above and provides a solution by utilizing solar energy to heat the space during day time. Although, there is sufficient solar radiation available at day time, peak demand for heating occurs during night hours. Therefore, proper storage mechanism is essential to extend the feasibility and utility.

To make system sustainable and useful at night, when there is no sun, we have incorporated thermal energy storage system to compensate for the absence of sun and keep the system running for a specific period of time. This is achieved by incorporating a Phase Change Material which absorbs heat energy while being charged during the day and stores it in the form of latent heat and during night it discharges and thus converts back into the initial state while releasing the initially stored latent energy. The results achieved from experimentations depict that for each hour of heating requirement of the model space at night, approximately 1 kilogram of phase change material is required. The detailed calculations have been mentioned in the results later.

ACKNOWLEDGMENT

This task couldn't have been completed without the cooperation of our respected supervisor. We wish to express indebted gratitude and special thanks to "Lecturer Muhammad Umer, SMME, NUST" who in spite of being extraordinarily busy with his duties, took time out to hear and guide us throughout the project.

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For their valuable guidance and for providing us this valuable experience

Regards…

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ABBREVIATIONS

PCM Phase Change Material

CFM Cubic Feet per Minute

ACH Air Change per Hour

NOMENCLATURE

M	Mass of pcm
H	Enthalpy
Q	Heat stored in pcm
ṁ	Mass flow rate of air(kg/s)
C_p	Specific heat of Air(kJ/kg)
L_f	Latent heat of fusion(kJ/kg)
K	Thermal conductivity(W/m.k)
A_{pcm}	Heat transfer area of pcm
R	Thermal resistance
T	Temperature
H	Convective heat transfer Coefficient(W/m^2-K)
U	Overall heat transfer Coefficient (W/m^2-K)
ΔT	Change in temperature
T	Time(second)
h 1,2	Forced convection heat transfer coefficients (W/m^2-K)
H	Incident global radiation (W/m^2)
A	Flow cross sectional area (m^2)
L	Length of collector(m)

CHAPTER 1

INTRODUCTION

INTRODUCTION

1.1 BACKGROUND:

In today's modernized era with technology development at its peak, energy sector holds the key rank among a country's economic status. The energy consumption of the world is a derivate of the total energy consumed by the entire population. It is usually measured on an annual basis and involves all the energy coming from different renewable and non-renewable energy sources. In the present era, the developing countries are keen on having a sustainable economic system.

The world has seen an enormous increase in its population over the past few decades giving a continuous rise to the energy demand across the globe. There are numerous energy sources present to counter this energy demand which can be grouped in different ways. The consumption and production of energy resources are one of the key components for the global economy.

The energy sources can be classified as primary or secondary sources along with another useful classification being the renewable and non-renewable sources.

The classification based on the nature of the sources is given as follows:

1) Primary energy sources: The sources that can be used directly to get energy are classified as the primary ones. They exist naturally in the environment. Some of the examples include Coal, Oil, Natural Gas, Sun, Wind, etc.
2) Secondary energy sources: The sources that derive energy from the transformation of the primary energy sources are classified as the secondary ones. Some of the examples include Petrol, Electric Energy, etc.

The classification based on the availability of the sources is given as follows:

1) Renewable sources: The sources that give out energy that is chemically or physically transformed repeatedly and is regenerated are known as renewable sources. The renewable or non-exhaustible sources are unlimited and include the sun, wind, water, tides, etc.
2) Non-renewable sources: The sources that are limited and are characterized by long regeneration times and are considered depleted after exploitation are known as the non-

renewable sources. The non-renewable or exhaustible sources are the various types of fossil fuels i.e. Oil, Coal, Natural Gas, etc [1].

The fossil fuels depletion has caused the need for innovation in and consumption of renewable energy sources. The renewable energy industry has a key focus on using new and appropriate renewable energy technologies. Although the renewable energy technologies have not replaced the non-renewable fuels but they can be a useful alternative in the near future.

The global energy consumption fractions can be studied from the below mentioned statistics. It can be seen that the renewable resources constitute 10.4% of total energy consumption of the world and among renewable resources, Hydel power contributes the greatest share while solar power contributes only 7.1%. The motive of this project is to increase the utilization of solar energy and subsequently the share of renewable resources in global energy consumption.

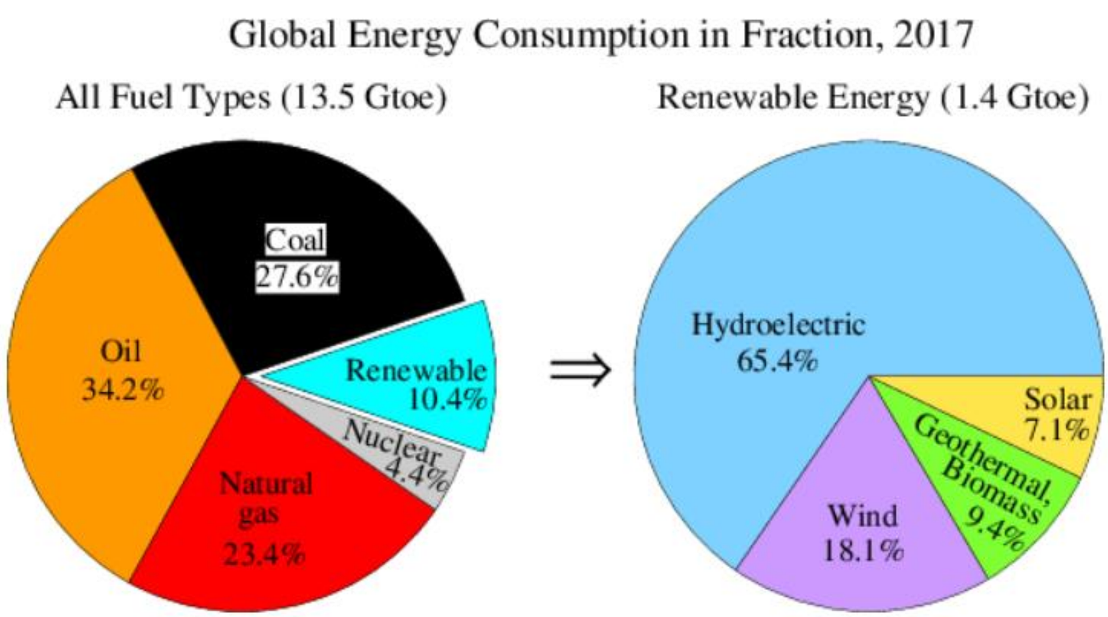


Figure 1: Total Global Consumption of renewable and non-renewable resources *[2]*

1.2 PROBLEM STATEMENT:

The aim of this project is the efficient utilization of solar energy for the purpose of space heating with air as a solar fluid to tackle the ongoing energy crisis while keeping the system and utility feasible, sustainable, viable and economical. For shade or night hours a PCM based energy storage device is used to increase utility.

The immense solar potential available in Pakistan and the criminal utilization of it gave rise to the need for this project. Some other key decisive factors have been mentioned in the motivation section below.

1.3 OBJECTIVES:

The core objectives and the key deliverables of the project include:

1) Investigation of the Phase Change Materials for thermal energy storage system.
2) Thermal Analysis of Solar Space Heating System for required system.
3) Techno-economic analysis of the Solar Space Heating System.
4) Development and Demonstration of prototype Solar Space Heating System.

1.4 MOTIVATION:

The core motivation for our project comes from one of the key issues that we face in our daily lives i.e. the energy crisis which has an impact on millions of individuals. Pakistan, being a developing country is highly dependent on non-renewable energy resources like fossil fuels for power generation and energy. Pakistan imports crude oil worth 12 billion USD annually that heavily influences the economy. 70% of this crude oil is used for power generation which is then used as a domestic and industrial utility, for our concern in space and water heating applications [2]. In the remote and off-grid areas, coal and wood are being used for heating applications.

The issues for the motivation for this project can be summarized from the key factors mentioned below:

- Underutilization of the renewable energy resources available in Pakistan and severe energy crisis affecting millions of lives all across the country.
- Conservation and effective use if non-renewable energy resources i.e. fossil fuels and reducing dependence on them for space heating.

- Confrontation of issues including depletion of fossil fuels, green-house gases emission leading to global warming, environmental pollution and minimizing dependence on imported fuels that heavily influence the economy.

Pakistan has also been blessed with an immense potential of inexhaustible energy resources like solar as well as wind energy which is currently being underutilized [3]. So, we aim at effectively utilizing the available Solar Potential i.e. up to 100,000 MW with our Solar Space Heating System for space heating purpose [1]. By doing so, we not only aim at reducing the dependence on fossil fuels but also counter issues like deforestation, depletion of fossil fuel reserves and environmental problems.

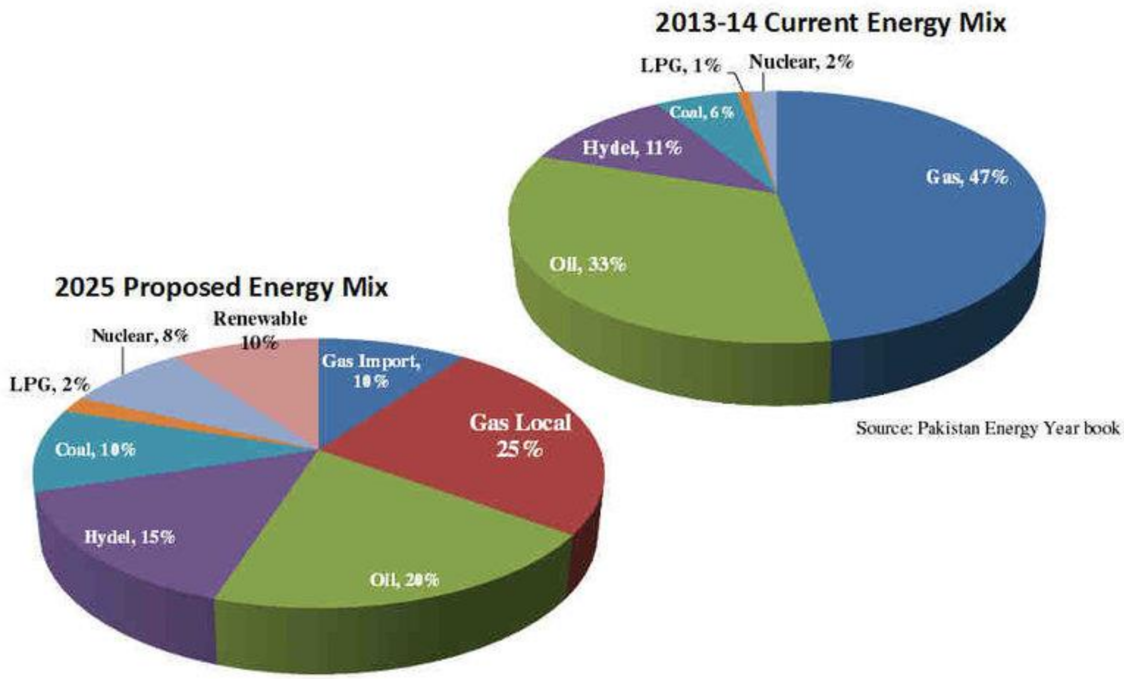


Figure 2: Total Energy Consumption of Pakistan *[4]*

For our literature review, we consulted different books and relevant materials and developed our understanding of the functionalities of the individual components like the solar collectors, different types of phase change materials, piping system and the thermal energy storage tank as well as the system as a whole. The core subjects whose knowledge would be incorporated in this project include Heat and Mass Transfer, HVAC, Fluid Mechanics and Materials Engineering.

CHAPTER 2

LITERATURE REVIEW

LITERATURE REVIEW

2.1 Solar Energy:

2.1.1 Definition:

Solar energy is the energy gained from the light and heat energy from the Sun. This energy is furnished by techniques such as photovoltaic technology, Solar Heating or cooling, solar architecture etc.

2.1.2 Explanation:

Solar energy is the most important form of renewable energy since it is available across the globe while its intensity varying depending on the location and climate. It has an indefinite lifetime with zero adverse effects on the environment as there are no harmful by-products. Solar energy is in two basic energy forms which can be converted into subsequent forms utilizing the appropriate apparatus.

2.2 Solar Collectors:

2.2.1 Definition:

"Solar Collectors are devices that utilize the solar radiation to heat a fluid which can be used for different purposes."

2.2.2 Types:

Solar collectors can be classified into multiple types according to different characteristics. These types are explained below:

Nature of Fluid:

Solar Water Collectors:

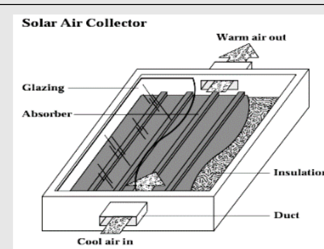
These are the collectors that heat water which is further utilized for different purposes for example in space heating and in geysers. These collectors have complex structure or network of pipes through which water flows as compared to the other type.

Solar Air Collectors:

These collectors use air as their working fluid which is heated by passing it through a gap between the absorber plate and the upper glass cover which serves an air flow channel. Since there is no need for piping for the air collectors hence they are easy to manufacture and cheaper as well.

A comparison of both types of collectors is shown in the table below [5]:

Table 1: Comparison of Solar air and water Collectors

Characteristic	Water Solar Collector	Air Solar Collector
Regulation Device	Pump	Fan
Passage	Tubes	Air Channel
Power Consumption	55 Watt/Pump	5-10 Watt/Fan
Cost	Expensive	Economical
Temperature Elevation	High	Low
Materials	Metal Frame	Wooden Frame
	Copper Tubes	No Tubes
	Aluminum Absorber Sheet	Aluminum Absorber Sheet
	Glass Cover	Glass Cover
		

Glass Cover:

Unglazed Solar Collectors:

These are the type of solar collectors which do not have a top glass cover. They are also known as transpired solar collectors [6]. The outer panel of a transpired solar collector contains numerous small perforations that permit the heat to uniformly get absorbed into the air cavity behind the upper panel. This heated air is drawn into the building's ventilation system by a negative pressure.

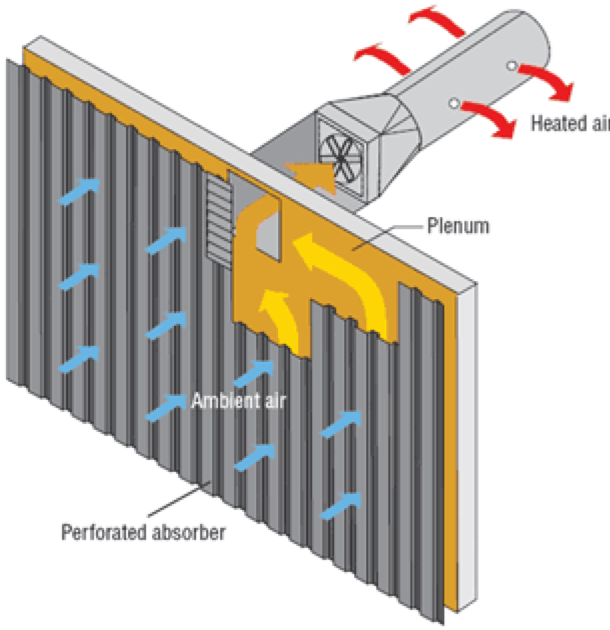


Figure 3: Unglazed Solar Air Collector *[7]*

These type of solar collectors are used primarily for to heat ambient air commercially, industrially and other process applications.

Glazed Solar Collectors:

These solar collectors have a glass cover above the absorber plate and the gap between the two serves as an air channel through which the air flows and is heated. Air enters into the cavity from one side, as it passes it is heated by radiation and convection and then exits from the other side. There is an insulation present at the back of the absorber plate and the walls are usually made of wood which serves as an insulation as well [8].

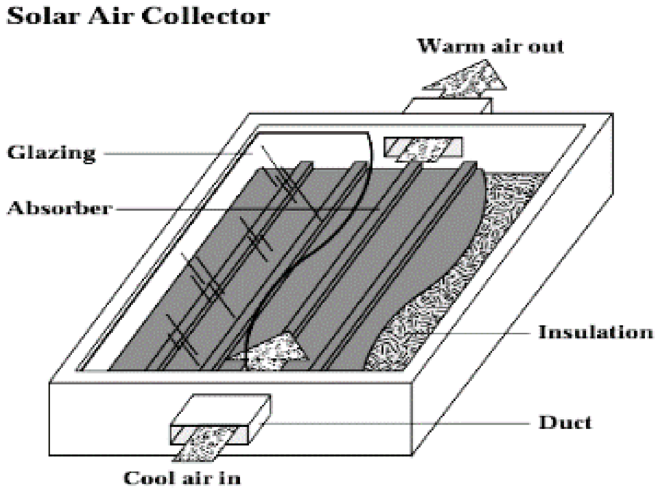


Figure 4: Glazed Solar Air Collector

2.3 Thermal Energy Storage:

A solar system can only work during the day time when the sun shines. Hence in order to make the system work during night as well it is required to store the excess thermal energy during the day which can then be reused during night. This process of storing excess thermal energy present in a material is known as thermal energy storage.

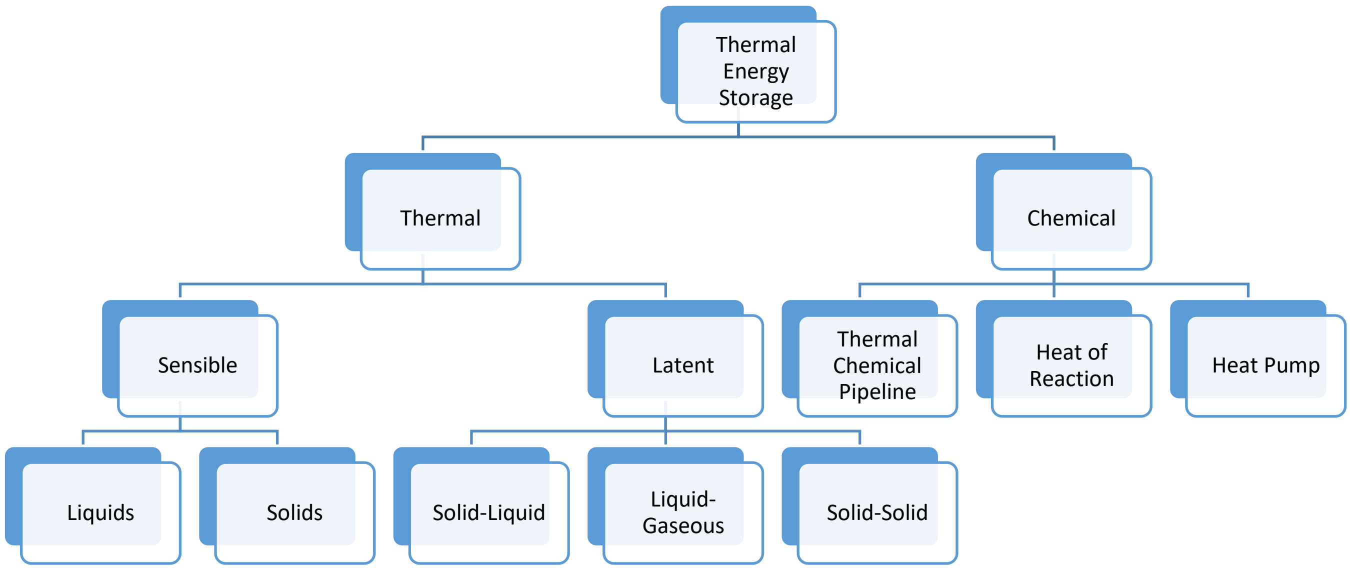


Figure 5: Chart of Various Methods of Thermal Energy Storage *[9]*

Sensible Heat Storage (SHS):

In SHS systems the energy is stored by heating a substance to a particular temperature without any change in its phase [9]. This energy stored (kinetic and potential energy of atoms of the substance) during charging and can be extracted when the ambient temperature falls below the temperature of the material and hence the material starts to radiate the stored energy during discharging. Water has high specific heat and is inexpensive and hence serves as the best SHS medium but can only be used below 100°C. Rock bed type storage materials are preferred for air heating [10].

Latent Heat Storage (LHS):

Latent Heat Storage (LHS) as the name suggests absorbs or releases heat as a material changes its phase form [9].

Phase change of material occurs at a constant temperature, also this method can provide high energy storage density and hence is quite an attractive method of thermal energy storage. The transition phases used for latent energy storage are: Solid to Solid, Solid to Liquid, Solid to Gas, Liquid to Gas and vice versa

2.4 Heat Exchanger:

2.4.1 Definition:

"A heat transfer device that efficiently transfers or exchanges heat between two substances."

2.4.2 Explanation:

There are multiple types of heat exchangers present and used worldwide. Some common examples of heat exchangers used include radiator of a car, boiler, condenser etc. Heat exchangers are used for cooling and heating purposes both.

2.4.3 Types:

Heat exchangers present around the world can be divided into a number of categories depending on the type of fluid used, direction of flow and contact between fluids

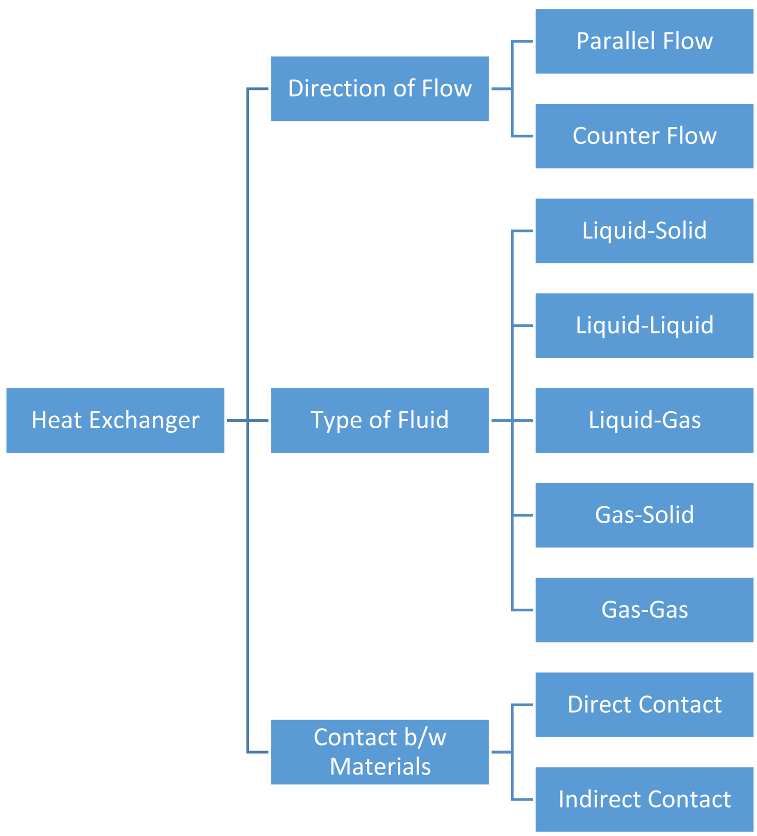


Figure 6: Categories of Heat Exchangers

The heat exchanger consisting of a solid material has three particular arrangements

1) Cylindrical casing with a central fluid passage and solid material placed on shelves around the central passage. The disadvantage of this design is that it provides a limited area for heat transfer. It is preferred for Non-organic PCMs since it can be rotated about a central axis and hence can remove incongruent melting and phase separation.

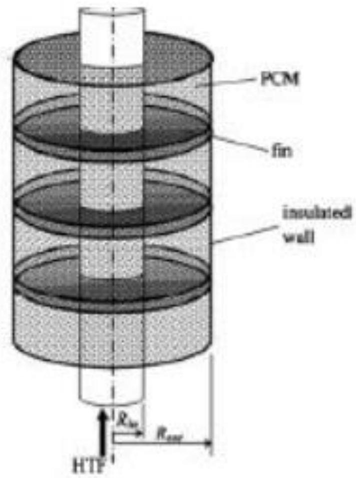


Figure 7: Cylindrical Heat Exchanger

2) Plate type heat exchanger with solid PCM placed on the shelves and the fluid flows over these shelves in a serpentine flow and heat transfer occurs. Its advantage is that in this type of heat exchanger we get a large surface area for heat transfer and hence better efficiency.

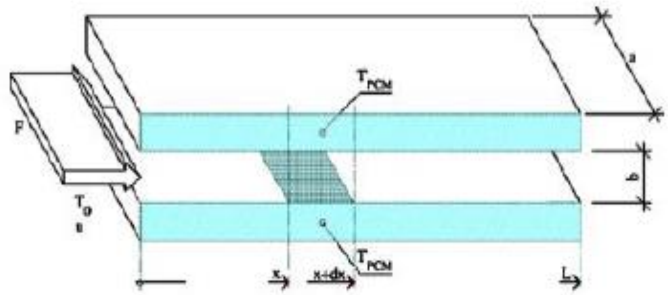


Figure 8: Plate Type Heat Exchanger Description

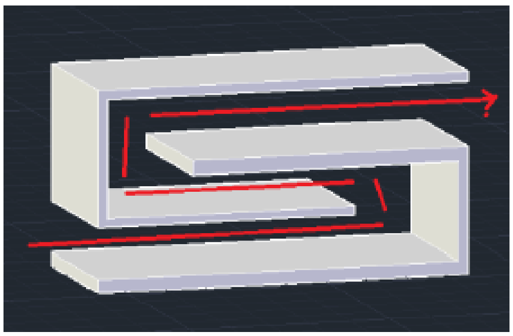


Figure 9: Flow Description for Flat Plate Heat Exchanger

3) Shell and tube heat exchanger with PCM placed on the outer side of the tubes inside the shells and the fluid flows through the tubes surrounded by the PCM in single-pass or multi-pass configuration. This type of heat exchanger are most widely used and are easily available. They also provide large heat transfer surface with uniform heat transfer from or to the fluid inside the tubes. They have low maintenance and maintenance cost. Boilers and condensers are usually of this type [11].

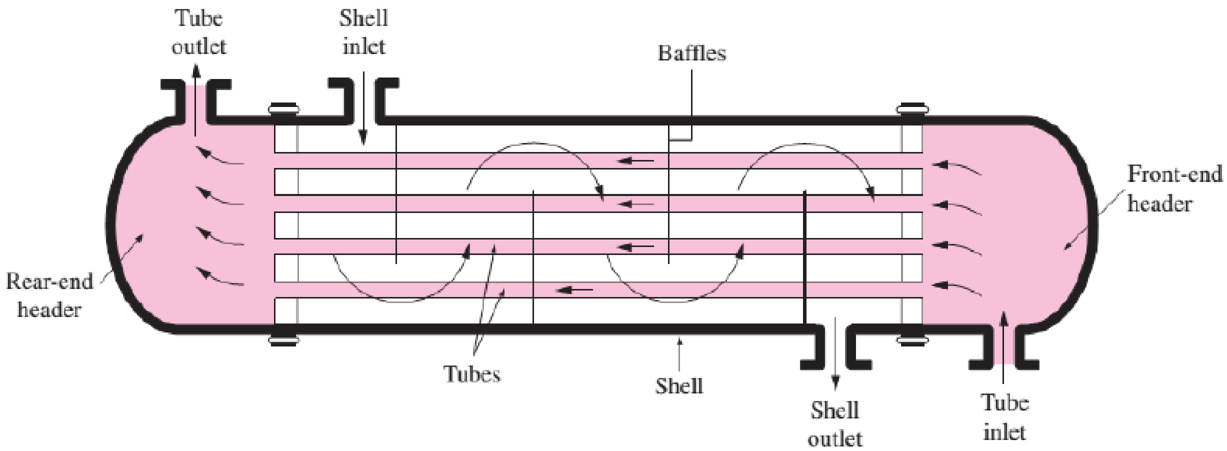


Figure 10: Shell and Tube Heat Exchanger

2.5 Phase Change Material:

2.5.1 Definition:

A Phase Change Material (PCM) is a kind of material having high heat of fusion (latent heat) .Their properties include melting and solidifying at a certain temperature and can store and release large amounts of energy.

2.5.2 Explanation:

In a phase change material heat is absorbed or released in the form of latent heat to break the bonds of its molecule while it changes phase [12]. This heat can then be recovered when the material cools down to ambient temperature and hence releases this stored latent heat while it changes back to solid phase [13].

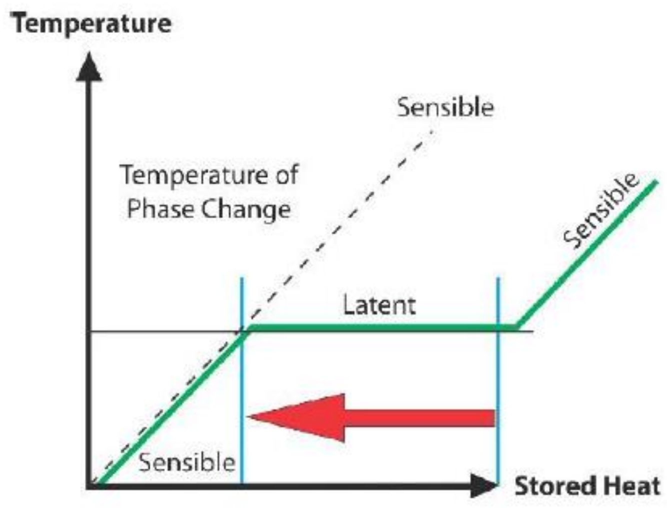


Figure 11: Energy Storage Graph of PCM

2.5.3 Types:

There are a number of PCMs used worldwide depending on the application temperatures and operating conditions. The PCMs can be divided into three basic types on the basis of their chemical characteristics. The following chart depicts the types of PCMs present [9]:

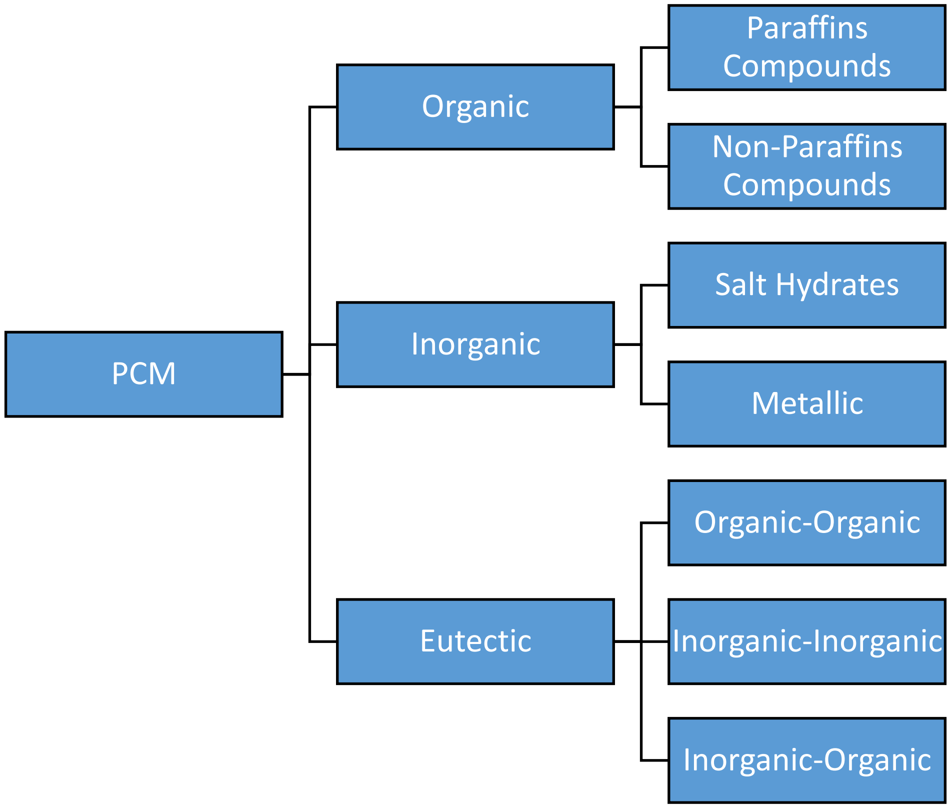


Figure 12: Types of PCMs

Organic PCMs:

Organic PCMs are paraffin compounds and non-paraffin compounds. Congruent melting (part of organic compounds), meaning they can freeze, melt and crystallize with minimal supercooling behavior and are generally non-corrosive.

Table 2: Comparison of Organic PCMs

Organic PCMs	
Paraffin Compounds	**Non-Paraffin Compounds**
Cheap	Expensive
Moderately Flammable	Inflammable
Non-Corrosive	Mildly corrosive

Chemically Inert and stable below 500°C	Instable at high temperatures
Congruent Melting	Low Flash Point
Low thermal conductivity	Low thermal Conductivity
Good Nucleating	Variable Toxicity
Long Freeze-Melt Cycle	High Heat of Fusion

Some of the Organic PCMs are mentioned below in the following table [9]:

Table 3: List of some Paraffins along with their properties

Paraffins		
No of Carbon Atom	**Melting Point °C**	**Latent Heat of Fusion (KJ/kg)**
14	5.5	228
15	10.0	205
16	16.7	237.1
17	21.7	213
18	28.0	244
19	32.0	222
20	36.7	246

Table 4: List of some Non-paraffins along with their properties *[9]*

Non-Paraffin		
Material	**Melting Point °C**	**Latent Heat of Fusion (KJ/Kg)**
Formic Acid	7.8	247
Acetic Acid	16.7	187
Glycerin	17.9	198.7
D-Lactic Acid	26	184
Methyl Palmitate	29	205
Phenol	41	120
Acrylic Acid	68	115
Polyethylene Glycol-600	22	146
Benzoic Acid	121.7	142.8
Cyanamide	44	209
Caprylone	40	259
4-Heptadacanone	41	197
Docasyl Bromide	40	201

Inorganic PCMs:

Inorganic PCMs are further classified as salt hydrate and metallic. They do not supercool appreciably and their heats of fusion do not degrade with cycling.

Table 5: Comparison of Inorganic PCMs

Inorganic PCMs	
Salt Hydrate	**Metallic Compound**
High latent heat of fusion per unit volume	High heat of fusion per unit volume

Latent heat of fusion does not degrade	Low heat of fusion per unit weight
Salt segregation	Low specific heat
Super cooling	Low vapor pressure
Incongruent melting	High Thermal Conductivity
Inexpensive	
Mild Toxic	
Mild Corrosive	
Poor nucleating	

Table 6: List of some Metallic PCMs along with their properties

Metallic PCMs		
Material	**Melting Point**	**Latent Heat (KJ/Kg)**
Gallium-Gallium antimony eutectic	29.8	-
Gallium	30	80.3
Cerrolow eutectic	58	90.9
Bi-Cd-In eutectic	61	25
Cerrobend eutectic	70	32.6
Bi-Pb-In eutectic	70	29

Eutectic PCMs:

A eutectic is basically a composition of two or more than two components in which each component melts and freezes congruently and hence producing a mixture of crystals of components during crystallization. Usually there is no segregation during melting and freezing as they freeze forming a complex mixture of crystals which does not give enough time for the components to separate.

CHAPTER 3

DESIGN METHODOLOGY

DESIGN METHODOLOGY

The design phase was divided into three main steps. Each step had a different approach but all the steps were dependent on the first one which was the calculations and design of Solar Collector. The methodology followed to design the system can be depicted by the following diagram:

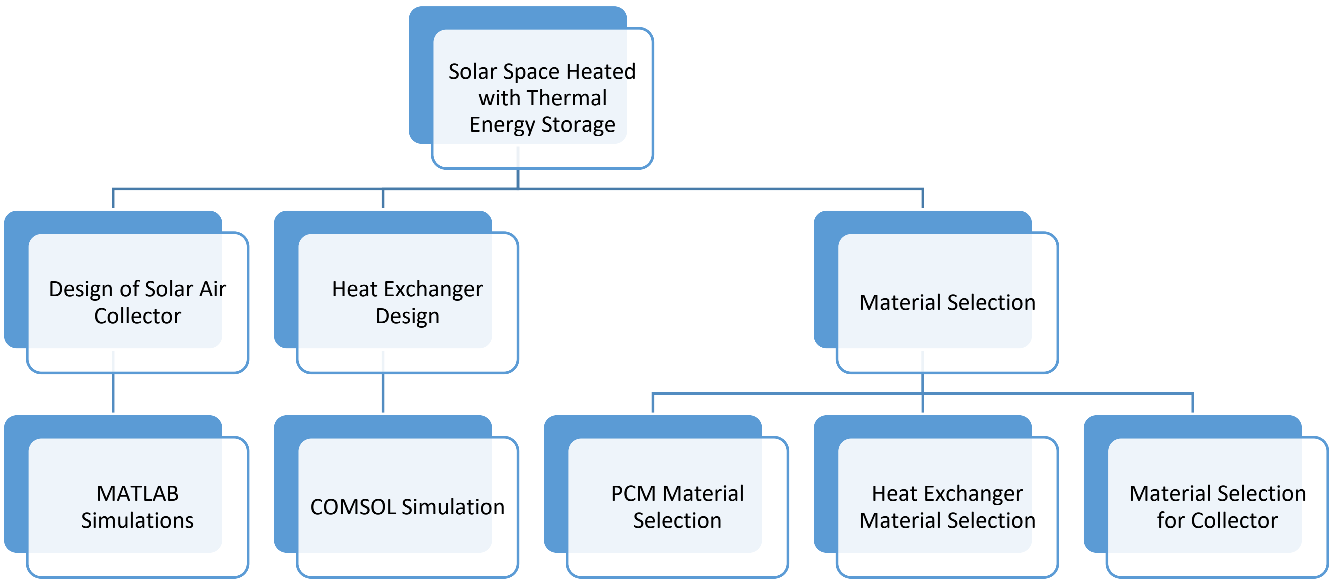


Figure 13: Design Methodology

3.1 Solar Collector:

Different types of solar collectors discussed in the literature review were studied and analyzed. On the basis of their characteristics, operating conditions, outputs and cost it was decided to go for the air type solar collector.

The next step was to select the appropriate type of solar air collector and for this following four models were studied [14]:

Type 1:

Glazed Solar Air Collector with a single channel design having only one channel for flow of air present between absorber plate and glass cover.

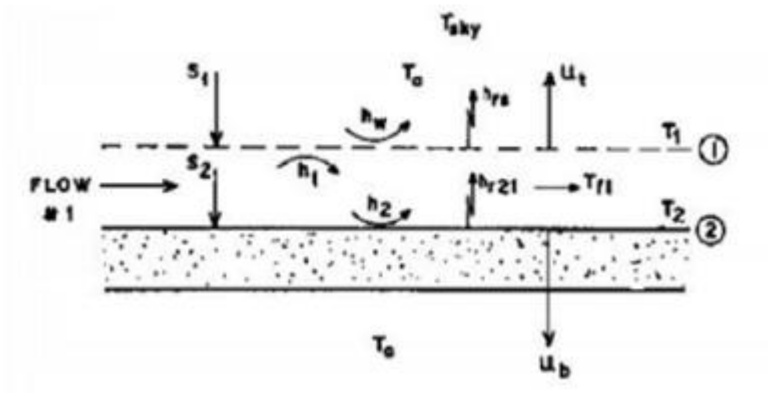


Figure 14: Type 1 Solar Air Collector

Type 2:

Unglazed Solar Air Collector with a single channel design having only one channel for flow of air present b/w absorber plate, the bottom plate. It doesn't incorporate a glass cover [15].

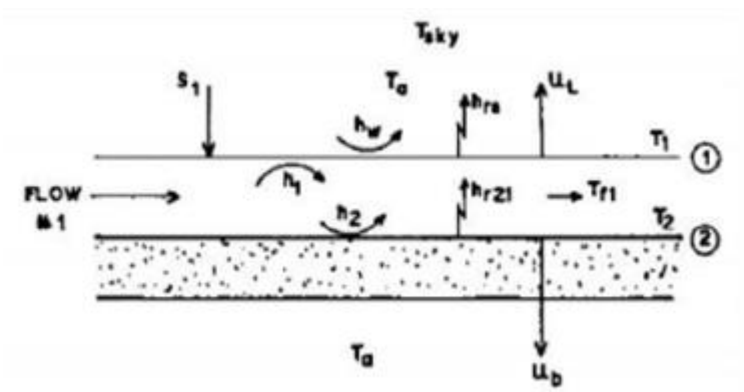


Figure 15: Type 2 Solar Air Collector

Type 3:

Glazed Solar Air Collector with a double channel design having a single air flow passage between the absorber plate and the bottom plate.

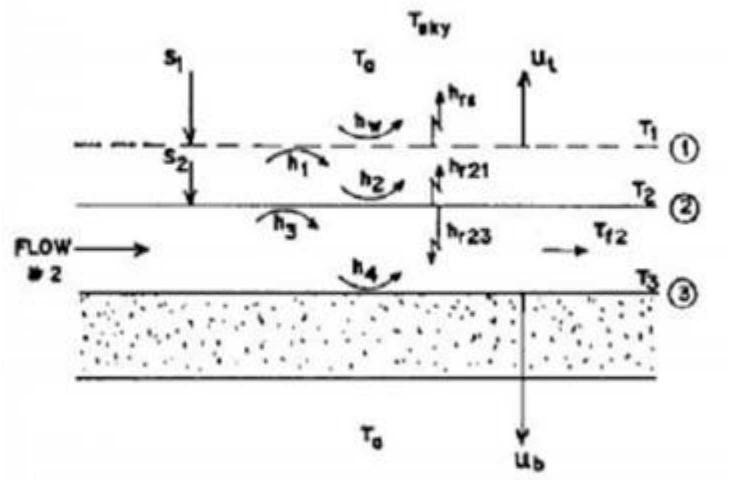


Figure 16: Type 3 Solar Air Collector

Type 4:

Glazed Solar Air Collector with a double channel design and subsequently two air flow passages between the top glass and absorber plate and between the absorber plate and the bottom plate.

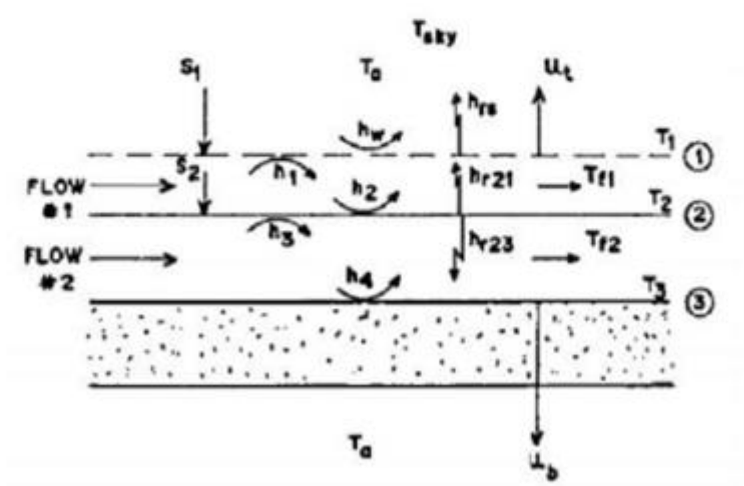


Figure 17: Type 4 Solar Air Collector

Selection:

The solar collector that was selected for our project was type 1 due to the following two reasons:

1. The increment in the output temperature for this type is sharper as compared to the others [16].
2. It is simpler, cheaper and also the construction will be easier as compared to type 3 and 4. Type 2 has lower output temperature for the given collector length as compared to type 1 hence despite of being cheaper and simpler than type 1 it was not selected.

3.1.1 Mathematical model and MATLAB simulation:

A Flat plate Solar Collector of type 1 is composed of three major components which are listed below:

1. Glass Cover (Glazing):

The top most layer of our solar air collector consists of a transparent glass cover which ensure maximum transmission of solar radiations. The cover material should be of high transmissivity, high temperature stability, low absorptivity and low reflection. The most common type of material for glazing is of solar collector is tempered glass having low iron content. Transmissivity value for this is greatest with a value of (0.85-0.90). It allows the incoming short wave radiations to pass but does not allow or have zero value of transmissivity for long wave radiations which are emitted by the absorber plate [5].

2. Absorber Plate:

This plate is present below the glass cover and is usually made up of a metal with high absorptivity. The primary function of this plate is to absorb incoming incident short have radiations from the sun and transfer this heat energy to the air flowing above it with minimum loss [5]. Metals that are commonly used for this purpose are corrugated sheets of aluminum and copper. Apart from the characteristics of the materials these plates are painted with some dark colour preferably black in order to increase its absorptivity.

3. Insulation:

In order to minimize the loss of heat energy from the collector to the surroundings, insulation is used. Most of the insulation is present behind the absorber plate so that it does not radiate any heat from the lower side [17]. Apart from that the frame of the collector is also made of such a material that the heat losses from the walls is also reduced. For this purpose the outer frame of the collector is made of wood while the insulation attached at the lower side of the absorber plate can be of a number of materials such as rock wool, glass wool, mineral wool etc [18]. We have selected glass wool due to its easy availability.

A general model of a flat plate solar collector can be seen from the following figure:

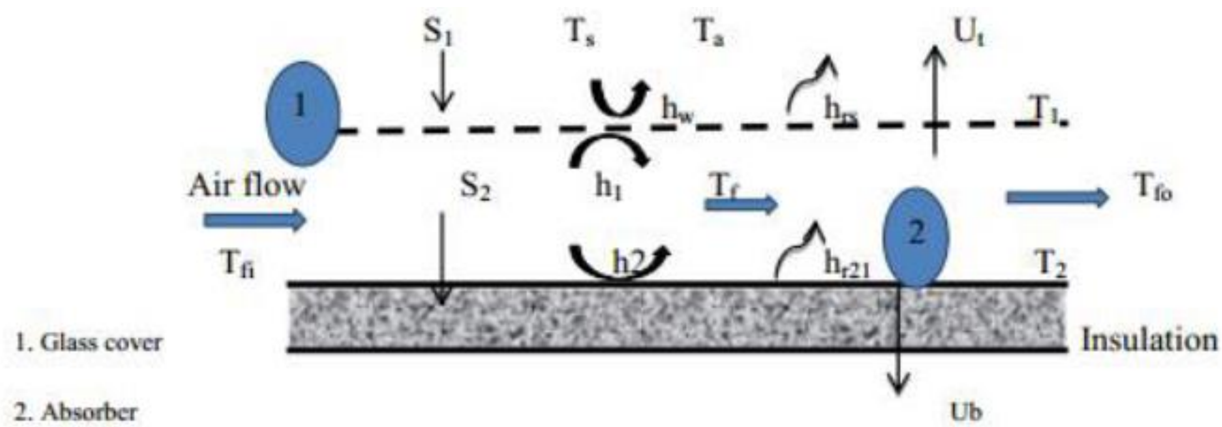


Figure 18: Component diagram of a flat plate solar air collector

3.1.1.1 Numerical Calculations:

In order to calculate the temperatures of the collector components for the entire day calculations were done on MATLAB. The formulas and equations used for this purpose are explained below in the form of a summary.

1) Solar Radiations:

The solar radiations absorbed by the top glass cover can be given by

$$S_1 = \alpha_1 * H$$

Similarly the solar radiations absorbed by the absorber plate is given by

$$S_2 = \tau * \alpha_2 * H$$

2) Flow Regime and convective heat transfer coefficient in the air gap:

There are 3 flow regimes in total, laminar, turbulent and transitional, the flow of air through the air gap can be of any of these three types depending upon the air velocity and subsequently mass flowrate. The Reynolds number for the flow is given by the formula

$$Re = (\dot{m}*D_h)/(A*\mu)$$

Laminar flow:

If Re<2300 then the flow is said to be laminar and the formula used to calculate Nusselt number is given by

$$Nu = Nu_{infinity} + (Nu_1)/(Nu_2)$$

Where,

$$Nu_1 = a*[Re*Pr*(D_h/L)]^m$$

$$Nu_2 = 1+b*[Re*Pr*(D_h/L)]^n$$

While the constants are

$$a = 0.0019;\ b = 0.00563;\ m = 1.71;\ n = 1.17;\ Nu_{infinity} = 5.4 \text{ (for } Pr = 0.7)$$

Turbulent flow:

If Re>6000 then the flow is said to be turbulent. In this case the Nusselt number can be calculated by the following formulas

$$Nu = 0.023*Re^{0.8}*Pr^{0.4}$$

Transition flow:

If 2300<Re<6000 then the flow is said to be transitional and this is the phase between the laminar and turbulent phase. In this case the Nusselt number can be calculated by

$$Nu = 0.116*(Re^{0.68}-125)*(Pr^{0.33})*(1+D_h/L)^{0.68}$$

Now using the Nusselt number calculated from the above mentioned methods we can calculate convective heat transfer coefficient from the following formula

$$h = Nu*k/D_h$$

Where D_h is the hydraulic diameter of the passage.

The convective heat transfers form the cover to the air (h_1) and from the absorber plate to the air stream (h_2) are both equal

$$h = h_1 = h_2$$

3) Convective Heat transfer due to wind:

The convective heat transfer coefficient from the top cover to the wind outside of the air collector is given by

$$h_w = 5.7+3.8*V$$

Where V is the wind speed at that hour.

4) Radiative Heat transfer:

The radiative heat transfer from the top glass to the sky is given by

$$h_{rs} = \sigma*\varepsilon_1*(T_1+T_s)*(T_1^2+T_s^2)*(T_1-T_s)/(T_1-T_a)$$

When $T_a=T_s$ the above equation becomes

$$h_{rs} = \sigma*\varepsilon_1*(T_1+T_s)*(T_1^2+T_s^2)$$

The radiative heat transfer coefficient between the cover and absorber is

$$h_{r21} = (\sigma*(T_1+T_2)*(T_1^2+T_2^2))/(1/\varepsilon_1+1/\varepsilon_2-1)$$

5) Overall Loss Coefficient:

The overall heat loss coefficient from the top surface is

$$U_t = h_w+h_{rs}$$

While that from the bottom of the collector is

$$U_b = 1/[(x_{bi}/k_{bi})+(1/h_w)]$$

3.1.1.2 MATLAB Simulations and Iteration procedure:

Iterations were done in order to calculate the output temperature of the collector for an entire day. These iterations were carried out using the algorithm mentioned in the following chart

[19]. The MATLAB code that was used to do these iterations is attached in the appendix of the report.

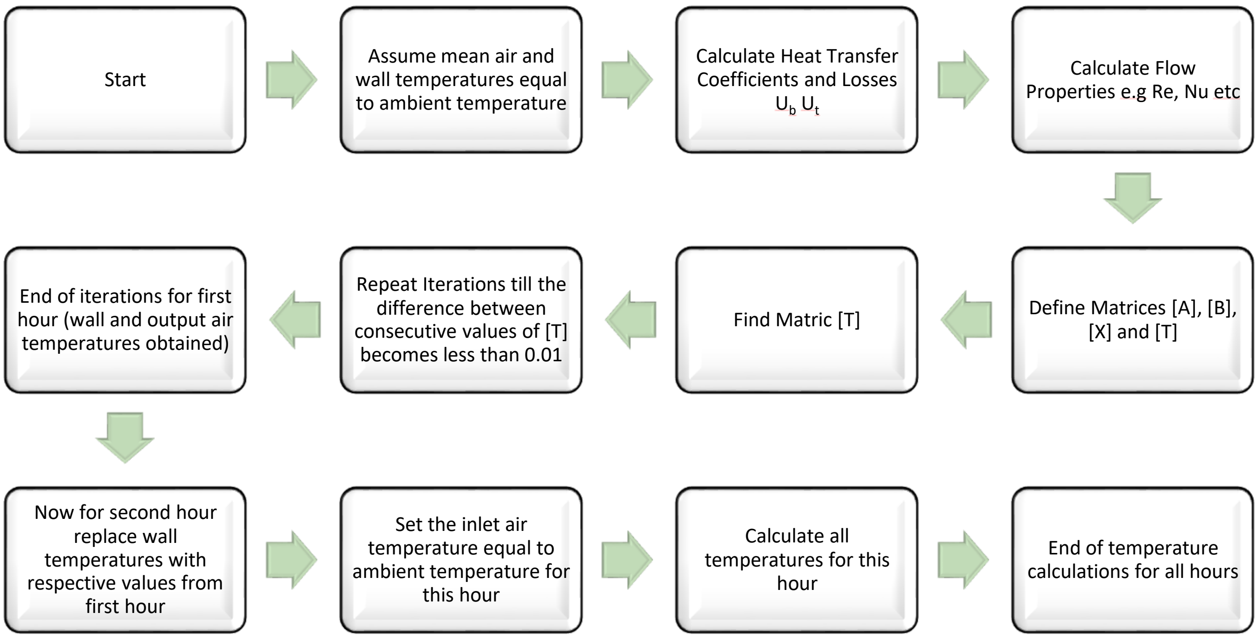


Figure 19: MATLAB Code Algorithm Steps *[14]*

3.1.2 Results:

The calculations done were according to the environmental condition of Islamabad, Pakistan for the date of 15th January that were obtained from the Meteonorm global meteorological database [20]. But the code developed in the MATLAB is not area specific and can be used to calculate the results for any place, however the base input parameters will be changed accordingly. After the iterations and calculations the results obtained are as follows

The temperature curves obtained for these parameters are shown below:

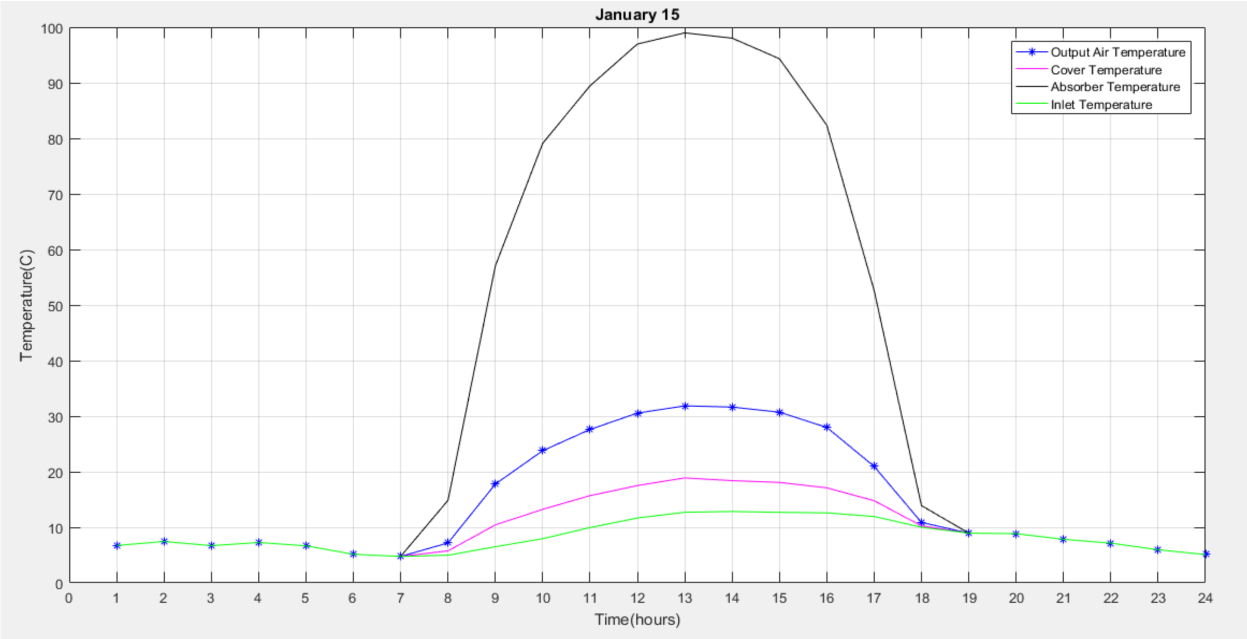


Figure 20: Output temperature curves of the selected Collector Design

3.1.2.1 Environmental Factors affecting the output results:

There are three basic environmental factors affecting the output results of our simulations which are as follows

1) Global incident radiations
2) Ambient temperature
3) Wind speed

Their graphs depicting the variation of these parameters for 15th January are shown in the graphs below:

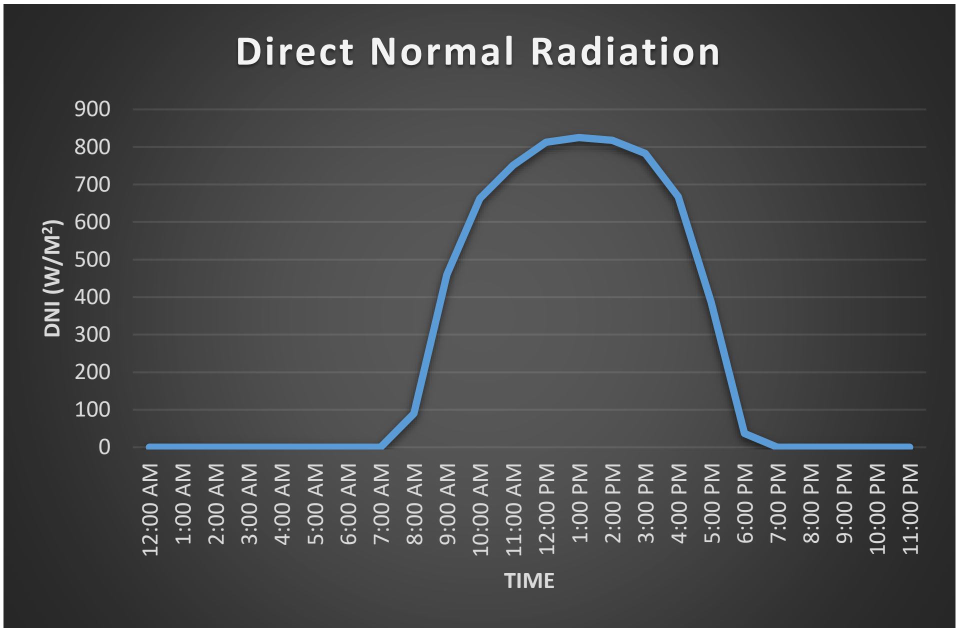


Figure 21: Direct Normal Radiations on 15th Jan

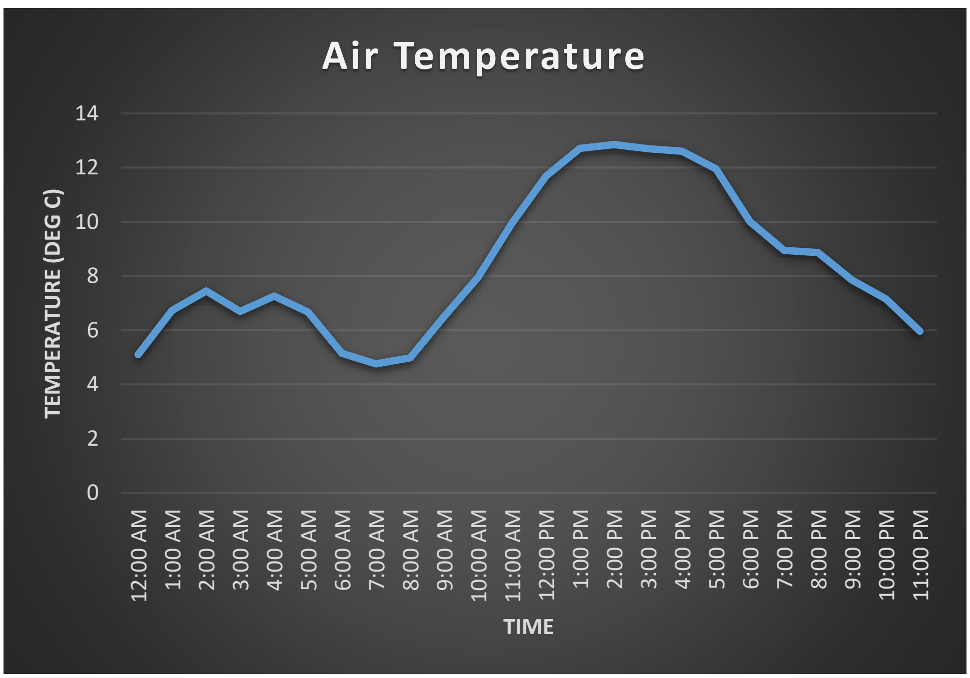


Figure 22: Ambient Air Temperature on 15th Jan

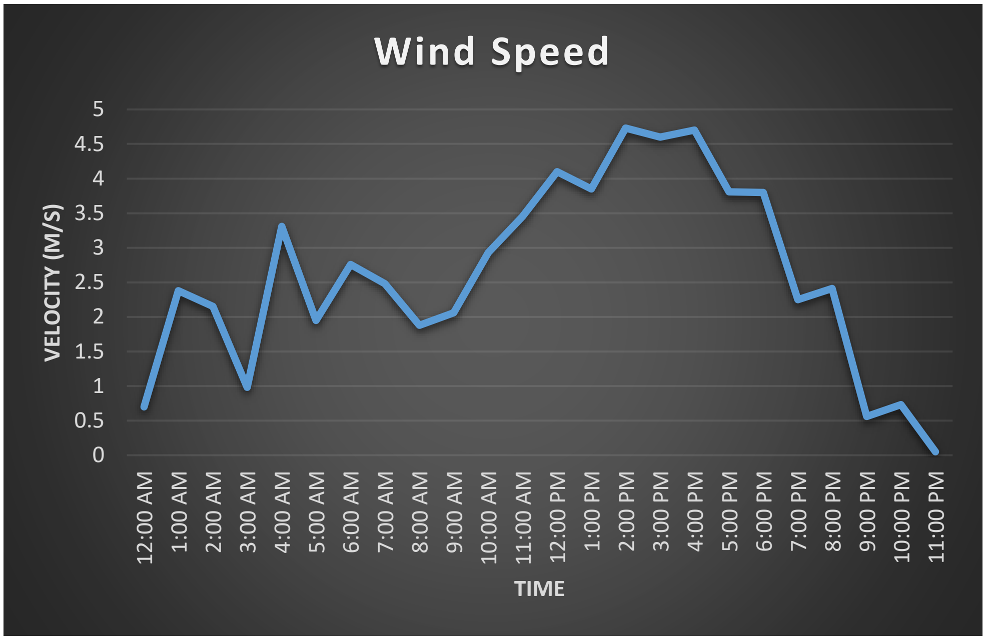


Figure 23: Wind Speed on 15th Jan

3.1.2.2 Effect of Different mass flowrates:

When we keep all the other parameters of the collector as constant and vary the mass flowrate only the affect was as shown in the graph below:

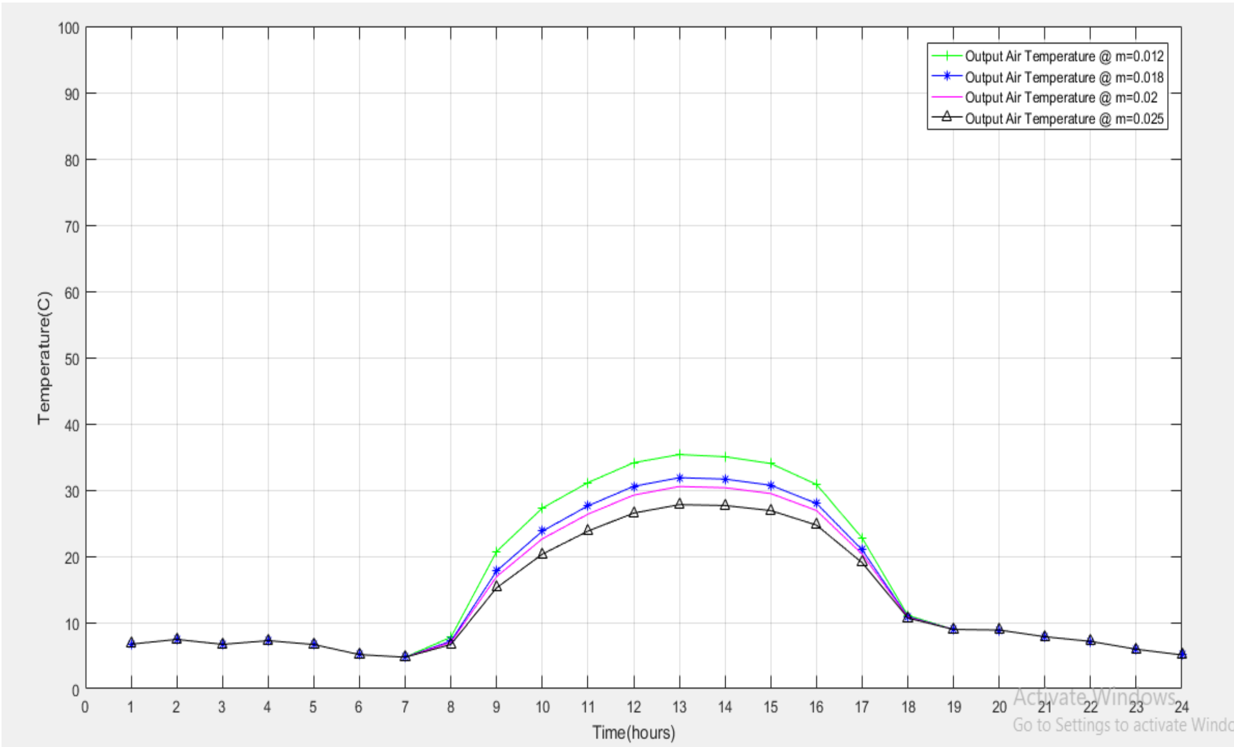


Figure 24: Effect of Variable mass flowrate on Collector output

3.1.2.3 Effect of Different Length of Collector:

When mass flowrate and other parameters are kept constant and length of the collector is varied then the results obtained are as follows:

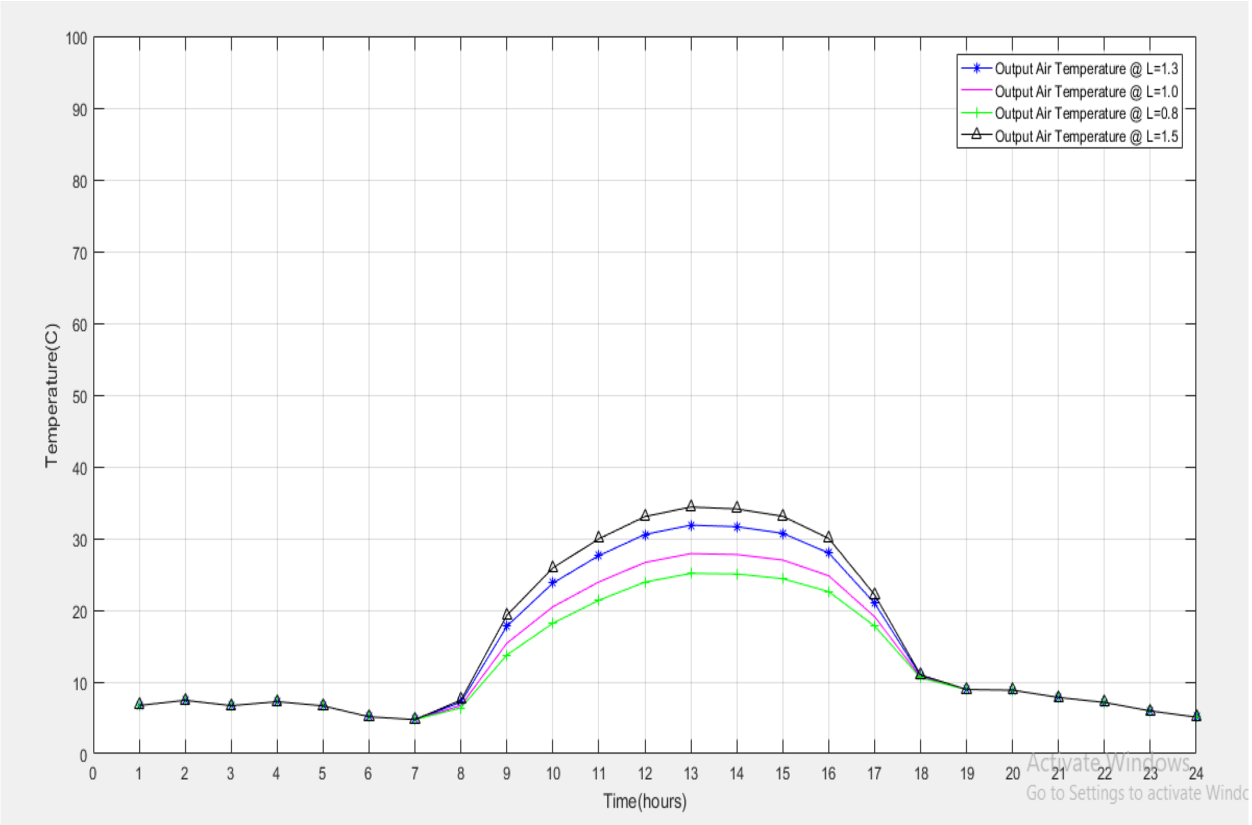


Figure 25: Effect of Variable Length on output of Collector

3.1.2.4 Effect of Different Width of Collector:

When the mass flowrate and length of the collector is kept constant and width is varied then the results obtained are as follows:

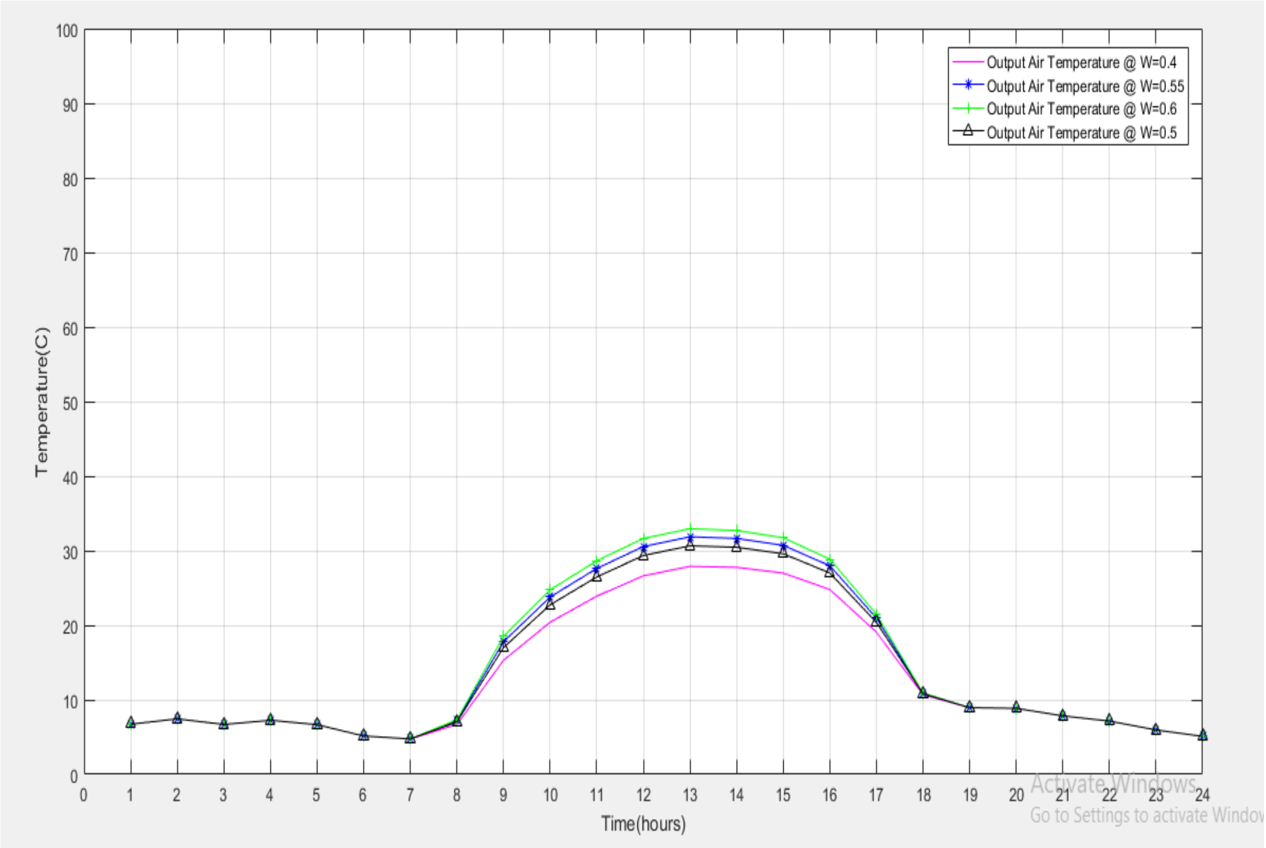


Figure 26: Effect of Variable Width on Collector Output

3.1.3 Design:

The parameters of the Collector selected after comparison shown in the following graphs are mentioned in the table below:

Table 7: Collector Parameters

Parameters	Values (m)
Length	1.3
Width	0.55
Air Gap	0.03

Insulation thickness	0.035
Sheet Thickness	0.002
Frame Thickness	0.02
Inlet/Outlet hole dia	0.025

The final design of our Collector is shown in the following diagrams:

Figure 27: Collector with Tempered Glass Cover

Figure 28: Collector with hidden Glass cover and corrugated Aluminum Sheet

3.2 PCMs:

Following characteristics need to be kept in mind while selecting a suitable PCM [21].

1. Respectable Melting Point
2. High Specific Heat
3. High Enthalpy per volume(kJ/m^3)
4. High Density(this causes a low change in volume which ensures compactness)
5. High Conductivity(Thermal)
6. Non-corrosive
7. Non-Flammable

The Phase change materials that were shortlisted for Thermal Storage were:

1. Paraffin Wax(C_{17}, C_{16})
2. Polyethylene Glycol 600
3. Glycerin

Following table is formulated to provide a better comparison

3.2.1 Paraffin:

Following two paraffins were considered as their melting temperatures lied was less than the temperature provided by our warm air. This makes sure that the selected material behaves as a phase change material

The following table compares their properties

Table 8: PCM comparison A

PCM	C_{16}	C_{17}
Melting Point(C)	18.2	22
Density	774	778
Thermal Conductivity(W/mk)	0.21^s	0.21^s
Latent Heat of fusion (kJ/kg)	238	215

Some advantages and disadvantages of using paraffins are listed below:

3.2.1.1 Advantages:

- No supercooling.
- Chemically available.
- High heats of fusion.
- Paraffins are safe.
- Non-corrosive [9].

3.2.1.2 Disadvantages:

- Thermal conductivity is very low. (0.21 in our case)
- Commercial PCMs are not compact(No sharp melting points)
- Technical grade PCMs are cheaper but properties are not as good as pure paraffins.
- They have an added disadvantage of flammability [9].

3.2.2 Non-paraffins:

Unlike paraffins, non-paraffins donot have a generalized set of properties. This means that their properties vary from material to material. Each material has different physical and chemical properties. In our case, the non-paraffins that were studied were

1. Glycerin
2. Polyethylene glycol 600

Table 9: PCM Comparison B

PCM	Glycerin	Polyethylene glycol 600
Melting Point(C)	17.9	22
Thermal Conductivity(W/mk)	0.285	0.23
Density(kg/m3)	1260	1126
Latent Heat of fusion (kJ/kg)	198.7	146
Category	I	I

I=Most promising,II = Promising, III= least promising

Both of these PCMs come under the category of fatty acids. Fatty acids have comparable characteristics of the paraffins. Fatty acids have relatively higher values of latent heat than paraffins. They do not supercool even when they are subjected to a respectable number of freezing or melting cycles (they only have one component; no separation of phase)

For our project, we selected Glycerin due to its following characteristics [12]

- High density which ensures compactness.
- Suitable melting point according to our application.
- High latent heat of fusion
- Greater thermal conductivity
- Market availability

- Affordable yet efficient

3.3 Heat Exchanger:

In order to ensure simplicity and ease of fabrication, a suitable heat exchanger design had to be selected. This thermal energy storage system stores energy in the PCM during the daytime and it discharged heat into the heat transfer fluid during night .Our thermal energy storage system effectively acts as a heat-exchanger. As we are using air as the heat transfer fluid which has relatively low thermal conductivity, we selected a simple heat exchanger design which consists of a rectangular shell made of aluminum along with a single copper tube passing through the inside of the shell in a serpentine single pass configuration.

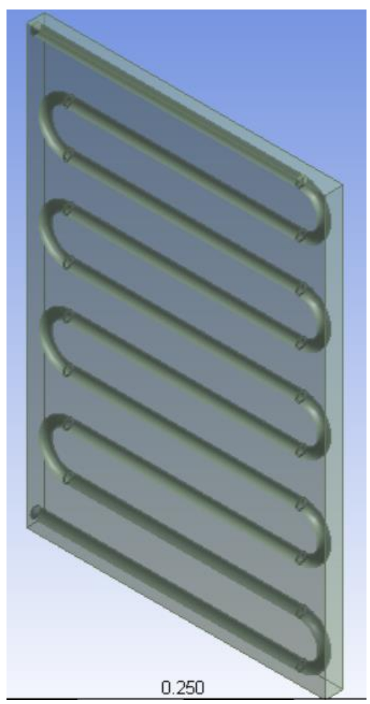


Figure 29: Cross-sectional view of Heat Exchanger

PCM would be incorporated inside the shell, surrounding the copper tubes. Air will flow through the copper tube passage in a serpentine flow. Heat transfer will occur through the walls of the tube. Copper was selected for the tubing material since air has low while copper has high thermal conductivity coefficient so to achieve efficient heat transfer. During day time the hot air flowing through the tubes would transfer its heat to the solid PCM, which would melt inside the shell and will be converted into liquid state. During night, PCM would solidify transferring its heat to the incoming air passing through the tubes. There would be a single inlet and single outlet for air while a port will be provided for the filling of PCM inside the shell.

To select a suitable thermal energy storage size and the amount of PCM to be stored for a suitable discharge time, the following procedure was selected to model the heat exchanger.

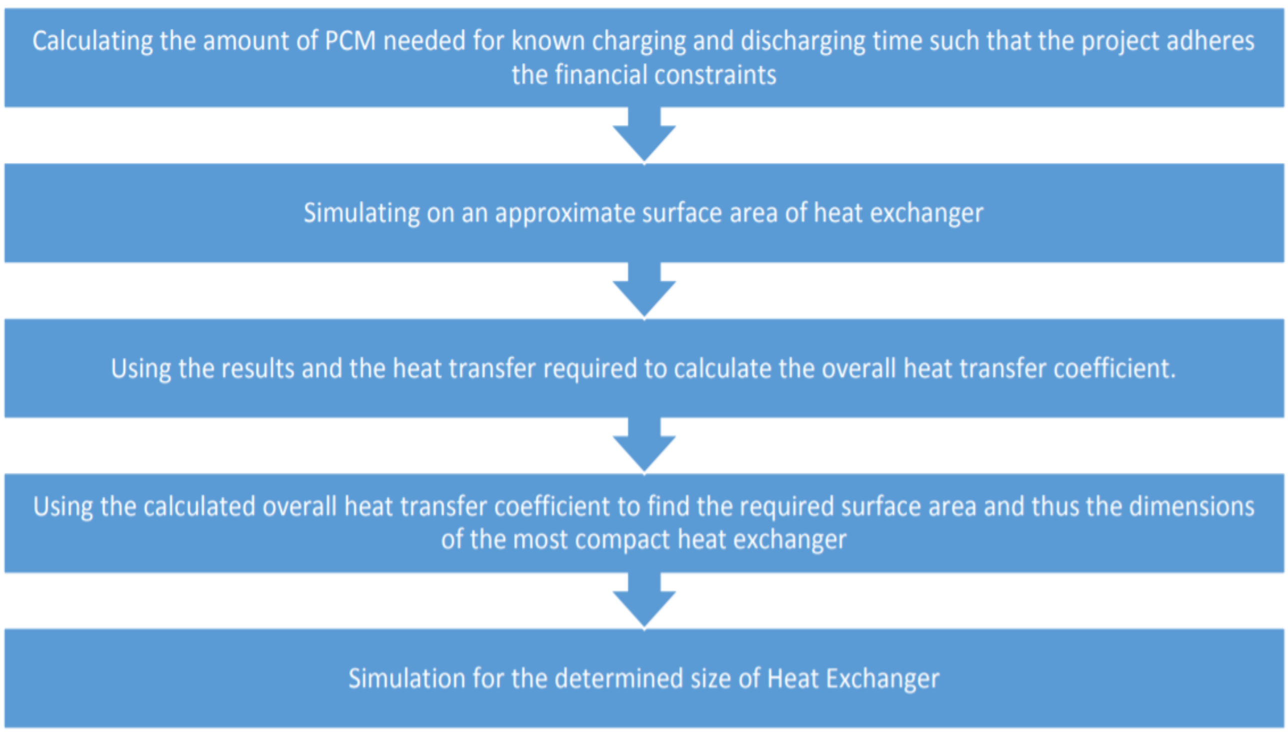


Figure 30: Approach for heat exchanger calculations

The first step is to define a suitable discharging time and then to calculate the amount of PCM that can discharge the heat energy according to the set discharge time. The methodology to do that is explained below:

3.3.1 Amount of PCM required:

As we are interested in the latent heat that is released due to solid-liquid change of phase. The advantage of latent heat storage lies in the fact that it can store large amounts of heat as compared to the sensible storage. The PCM stores energy while undergoing a phase change from solid to liquid.

$$Q=m.\ \Delta H$$

Where Q=heat stored in PCM.

m= mass of PCM and ΔH is the change of enthalpy between solid and liquid phases.

This equation can also be written as:

$$Q= m.\ l_f$$

As air flows through the heat exchanger it dissipates energy that is absorbed by the PCM.

$$Q= \dot{m}.C_p.\ \Delta T.t$$

Here t is the charge/discharge time of the PCM.

Applying energy balance we get:

$$m.\ l_f= \dot{m}.C_p.\ \Delta T.t$$

For a given discharge time, latent heat of PCM and mass flow rate of air we can calculate the mass of PCM required for the discharge.

3.3.2 Calculating Heat Transfer Coefficient of PCM:

First we need to calculate the heat absorbed by the PCM which is given by the formula:

$$q=k_{PCM}*A*(T_s-T_{pcm})$$

Heat transfer coefficient for the PCM is calculated as:

$$h_{PCM}=\frac{q}{A*(Tair-Tpcm)}$$

3.3.3 Calculating Heat transfer Coefficient U (total):

Total U depends upon the following parameters:

- R_{air}
- R_{metal}
- R_{PCM}

Overall heat transfer coefficient can be calculated by using the resistance model used in heat and mass transfer studies. This resistance model is used to calculate U_{tot}.

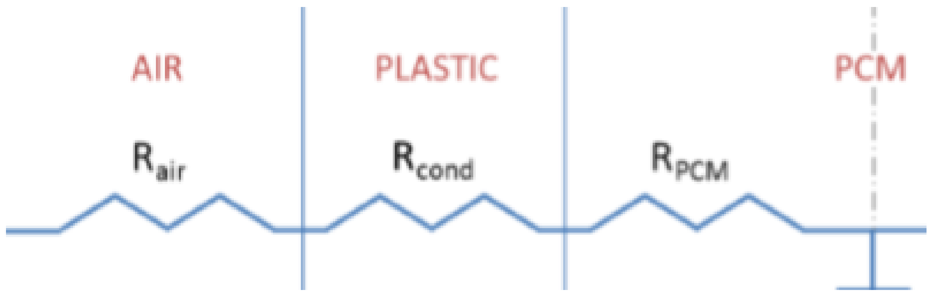


Figure 31: Thermal Resistence Model

Total resistance:

$$R_{tot} = R_{air}+R_{cond}+R_{pcm}$$

Where R_{cond} is the conduction resistance of the copper tube surrounded by the PCM

The PCM resistance is given by

$$R_{PCM} = (t/2.k_{corr}.A_s)$$

Here

$$k_{corr} = k_{PCM}\text{-}c_{nat,conv}$$

Conduction resistance for air is given by

$$R_{air} = \frac{1}{h.As}$$

Where h is the convective heat transfer coefficient of the air.

U_{tot} is given by

$$U_{tot} = \frac{1}{A*Rtot}$$

If different flowrates are desired during day time (charging) and night time (discharging), we can calculate the heat transfer coefficient for both charging and discharging.

3.3.4 Use of U_{tot} to calculate the most compact Surface area:

Now that we have calculated U_{tot} from the network showed above, we can use it to calculate the most compact surface area of the thermal storage

We have the formula:

$$q=UA\Delta T$$

Here ΔT is the logarithmic mean temperature difference as evaluated in the study of heat exchanger [22].

$$q= \dot{m}\, C_p\Delta T$$

So for already established mass flow rate, LMTD that can be evaluated and Q as evaluated in the formula mentioned above, we can calculate the most compact area for the thermal energy storage. In addition to that, the thickness and quantity can also be simply calculated once we know the dimensions (L * W) of the rectangular heat exchanger.

Once the mass of PCM has been determined and set then the volume of PCM was calculated and this is equal to the empty volume inside the shell required to hold the PCM.

$$V_{shell} = V_{PCM} + V_{Tubes}$$

The number of tubes required to achieve the necessary heat transfer is then obtained from the ANSYS simulations. Their volume is calculated by the formula:

$$V_{Tubes} = \pi r^2 l$$

3.3.5 Heat Load Calculation:

To define air flow requirements, first the size of the space to be heated needs to be determined which directly effects the space heating requirements. Other than that, internal factors that contribute to heating or cooling load are

- Occupancy
- Electrical Equipment
- Lighting

In addition to these other losses occurring are

3.3.6 Transmission loss:

Heat losses from walls, ceiling and other surfaces.

$$Q=AU(T_i-T_o)$$

3.3.7 Heat loss through infiltration and ventilation:

These kind of losses can further be subdivide into the following two categories

- Sensible Heat Loss
- Latent Heat Loss

Generally, we calculate heating requirement during the night time in case of winters. This usually results in us neglecting the internal loads like occupancy, equipment, motors .Hence the heating load calculated is for the worst case scenario and the load calculated is greater than the load requirement.

3.3.8 Air Flow Requirements:

Before we can establish the required air flow rate, first we need to define the requisite size of the heating space. The greater the size of the room, the greater is going to be the flow rate requirement in the room in order to maintain a constant set temperature.

The size of the room is also restricted by:

- The flow rate of the fan that is placed for the pumping system.
- Heat provided by the solar heating system

In order to make sure that the air in the system remains fresh and doesn't become stale, a factor known as air changes per hour (ACH) is introduced. It means the number of times air enters or leaves the space to be heated or cooled. The value of ACH for given room sizes and different activities can be estimated from different tables mentioned in manuals (ASHRAE etc).

Its formula is given as:

$$ACH=\frac{CFM\ x\ 60\ min}{Volume\ of\ room}$$

Typical Air Changes Per Hour Table

Residential	
Basements	3-4
Bedrooms	5-6
Bathrooms	6-7
Family Living Rooms	6-8
Kitchens	7-8
Laundry	8-9
Light Commercial	
Offices	
Business Offices	6-8
Lunch Break Rooms	7-8
Conference Rooms	8-12
Medical Procedure Offices	9-10
Copy Rooms	10-12
Main Computer Rooms	10-14
Smoking Area	13-15
Restaurants	
Dining Area	8-10
Food Staging	10-12
Kitchens	14-18
Bars	15-20
Public Buildings	
Hallways	6-8
Retail Stores	6-10
Foyers	8-10
Churches	8-12
Restrooms	10-12
Auditoriums	12-14
Smoking Rooms	15-20

Figure 32: Air Changes per Hour *[23]*

3.3.9 Results:

3.3.9.1 Calculation of Outlet Temperature of Heat Exchanger:

The outlet temperature of the PCM was determined using ANSYS Fluent. Using the transient model and by applying the input conditions, materials of the tubes (copper) and the outer body(aluminum), properties of the PCM (Glycerin), the temperature at the heat exchanger outlet for different times was calculated.

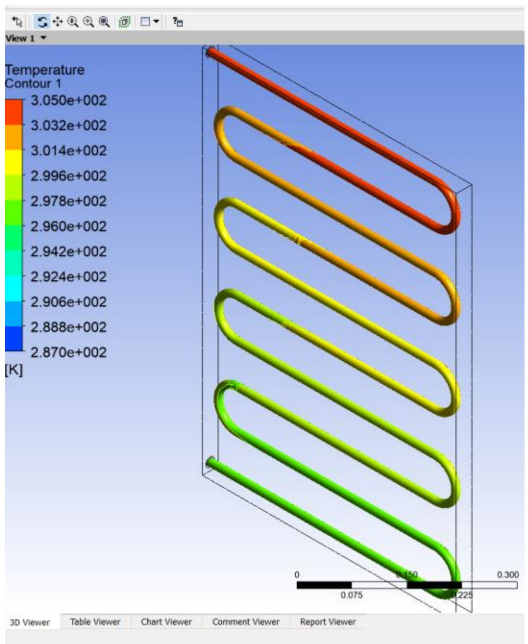


Figure 33: Temperature of the air for a fluid flow of 1.5 hours

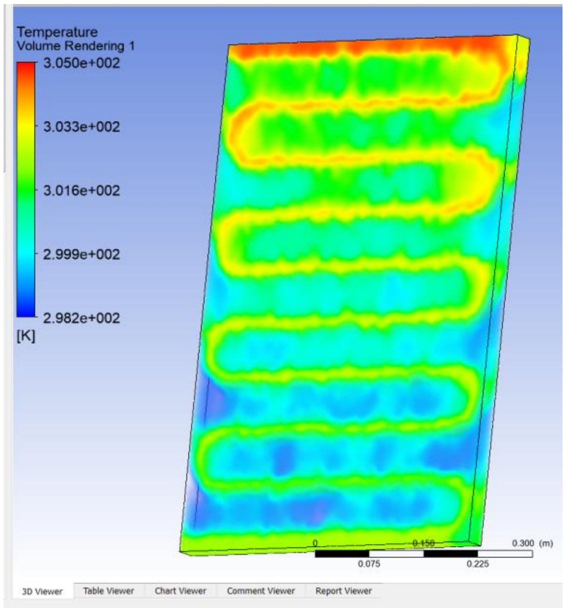


Figure 34: Temperature of the heat exchanger as a whole for a fluid flow of 1.5 hours

3.3.9.2 Calculation of PCM amount:

The amount of PCM required according to the discharge time and in accordance with the formulas method is given by

Table 10: PCM Mass Calculation with varying mass flowrate:

M (kg/s)	**C_p**	**T_1 (K)**	**T_2 (K)**	**ΔT (K)**	**P (W)**
0.002	1005	283	293	10	20.1
0.004	1005	283	293	10	40.2
0.006	1005	283	293	10	60.3
0.008	1005	283	293	10	80.4
0.01	1005	283	293	10	100.5
0.012	1005	283	293	10	120.6
0.014	1005	283	293	10	140.7

0.016	1005	283	293	10	160.8
0.018	1005	283	293	10	180.9
0.02	1005	283	293	10	201

DC Time (2 hours)	**Q (J)**	**DC Time (3 hours)**	**Q (J)**
7200	144720	10800	217080
7200	289440	10800	434160
7200	434160	10800	651240
7200	578880	10800	868320
7200	723600	10800	1085400
7200	868320	10800	1302480
7200	1013040	10800	1519560
7200	1157760	10800	1736640
7200	1302480	10800	1953720
7200	1447200	10800	2170800

Mass of PCM required for 2 hour discharge (kg)	**Mass of PCM required for 3 hour discharge (kg)**
0.99	1.49
1.98	2.97
2.97	4.46

3.96	5.95
4.96	7.43
5.95	8.92
6.94	10.41
7.93	11.89
8.92	13.38
9.91	14.87

A graph showing the relation between variation of mass flow rate and mass of PCM for both 2 hours and 3 hours discharge time shows that increasing mass flow rate would cause us to increase the amount of PCM needed for a fixed discharged time. Due to financial constraints, we cannot increase it beyond a certain limit even though it serves to increase the discharge time which is a favorable characteristic. Hence we selected the mass of PCM to be 3 kg.

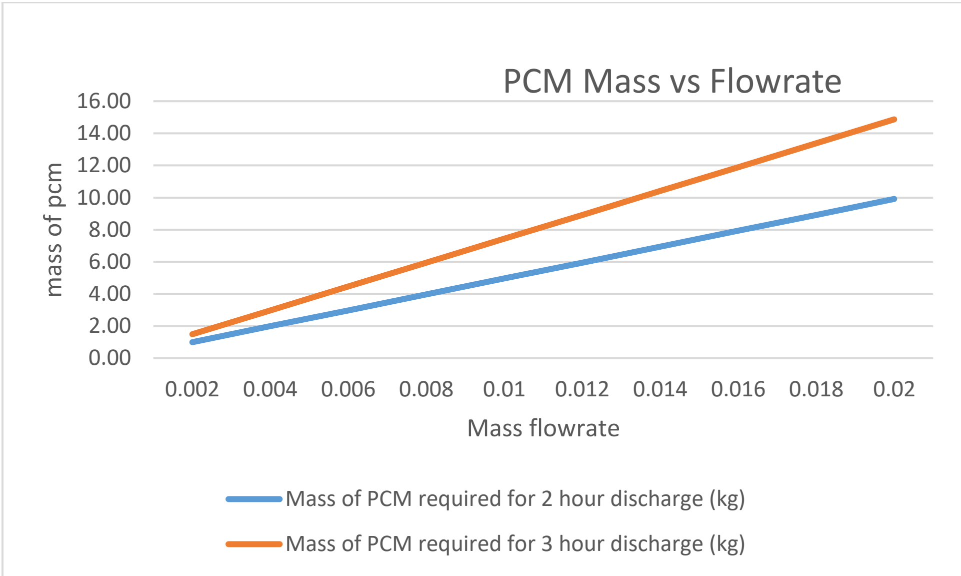


Figure 35: Flowrate with Graph

3.4 Space Heating Calculations:

The following calculations were performed to get a detailed overview about space heating requirements and considerations.

3.4.1 Airflow Requirements:

Mass flowrate of air supplied to space=0.018 kg/s or 31.78 CFM.

ACH= 5 (From ACH table)

A graph was plotted which illustrates the relationship between the volume in CFM provided by the fan to be installed in the system relative to the air requirement as the volume of the heating space changes. It was observed that for a given ACH increasing the volume of the room hence requires greater flowrate of air that enters the room to maintain the pre-set ACH which in our case is 5

Table 11: CFM requirement for space

Mass Flow Rate of air (kg/s)	Volume Flow Rate (m^3/s)	Volume in CFM	ACH
0.018	0.015	31.78	5
0.018	0.015	31.78	5
0.018	0.015	31.78	5
0.018	0.015	31.78	5
0.018	0.015	31.78	5

Table 12: CFM requirement for space

Fan (CFM)	Space Volume	Air to enter in an hour for ventilation	Air to enter per min for ventilation
31.78	27	135	2.25
31.78	64	320	5.333333333
31.78	125	625	10.41666667
31.78	216	1080	18
31.78	343	1715	28.58333333

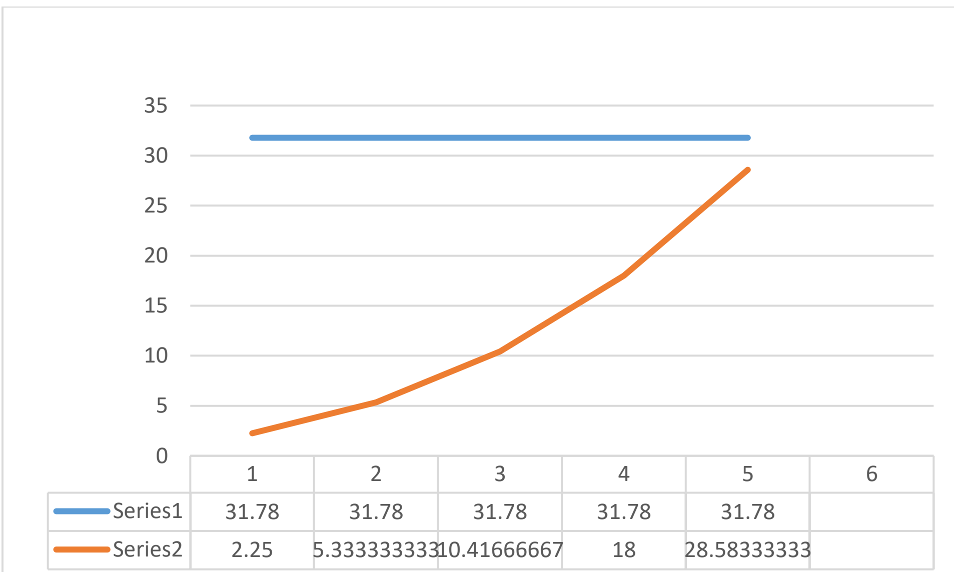


Figure 36: CFM requirement vs CFM supplied by fan:

3.4.2 Heating Load Calculations:

A simplified approach to calculate heating load is used in this section. Rather than computing sensible and latent ventilation, infiltration loads and transmission loads, a simple rule of thumb can be used which states that:

12000(BTU/hr) of heating/Air conditioning is enough for 400 ft^2. Simplifying it for 1 ft^2 we can use it to find the heating requirement for a specified area (ft^2) under consideration.

For 1 ft^2 heating load required is

$$\frac{12000}{400} = 30 \text{ Btu/hr}$$

Following table gives us a detailed overview of the sizing of the room relative to the heating requirements

Table 13: Heating Load requirement

Length	Width	Area of Room	Heating Factor	Requirement (BTU/hr)
3	3	9	30	270
4	4	16	30	480
3.5	3.5	12.25	30	367.5

Table 14: Continued-Heating Load Requirement

Q_{req} (W)	mass flow rate (kg/s)	C_p (kj/kg.K)	ΔT	$Q_{supplied}$ (W)
79	0.018	1005	6	108.54
140.6	0.018	1005	8	144.72
107.7	0.018	1005	10	180.9

This table also shows the Q supplied by our system to the space to be heated. It can be clearly seen that the heat supplied by the system is greater than the heating load requirement of the system. This heat supplied was calculated at different temperatures according to changing ambient conditions of the atmosphere.

3.4.3 Space Size Selected:

It can be observed from the aforementioned calculations that the airflow requirements and the heating load requirements were evaluated for different space volumes. Due to the compactness desired and the constraints faced (primarily budget) we had to select a suitable space size .Hence we selected the space size of 3.5*3.5*3.5 ft^3 as it was the most ideal for us.

CHAPTER 4

MANUFACTURING AND EXPERIMENTAL RESULTS

MANUFACTURING

Based on the literature review and theoretical calculations, we proceeded to the development of a prototype of the “Solar Space Heating System integrated with Thermal Energy Storage”. The prototype consisted of three major components:

- Solar Collector
- Heat Exchanger
- Piping Network and Connections

We proceeded with the development of our prototype after conducting a detailed analysis of the manufacturing processes that had to be undertaken.

4.1 Solar Collector:

The first and the most important part of this project is the Solar Collector that absorbs the radiations coming from the sun and in return heats the solar fluid i.e. air in this case passing through the passage.

For the manufacturing of solar collector, we initiated by acquiring 1 mm thick aluminum sheet and corrugated it as per the required dimensions to increase the surface area for convection process. In the next step, we manufactured a wooden outer box for the corrugated solar collector sheet having space for the thermal insulation i.e. glass wool beneath the corrugated aluminum sheet. The glass wool was inserted in the wooden box and the corrugated aluminum sheet was bolted above it with the sideward outer wooden box. Inlet and outlet holes were drilled into the wooden box as well.

In the next step, we coated the corrugated aluminum sheet with Matt Black paint to increase its absorptivity. To increase turbulence and enhance air’s heat transfer, we introduced 6 aluminum meshes at uniform distances to avoid any uncontacted air and having an enhanced heat transfer.

The solar collector was finished by equipping it with a 5 mm thick tempered glass as the unglazed glass cover to reduce losses and enhance heat transfer. An angle iron stand was welded together to place the Solar collector unit.

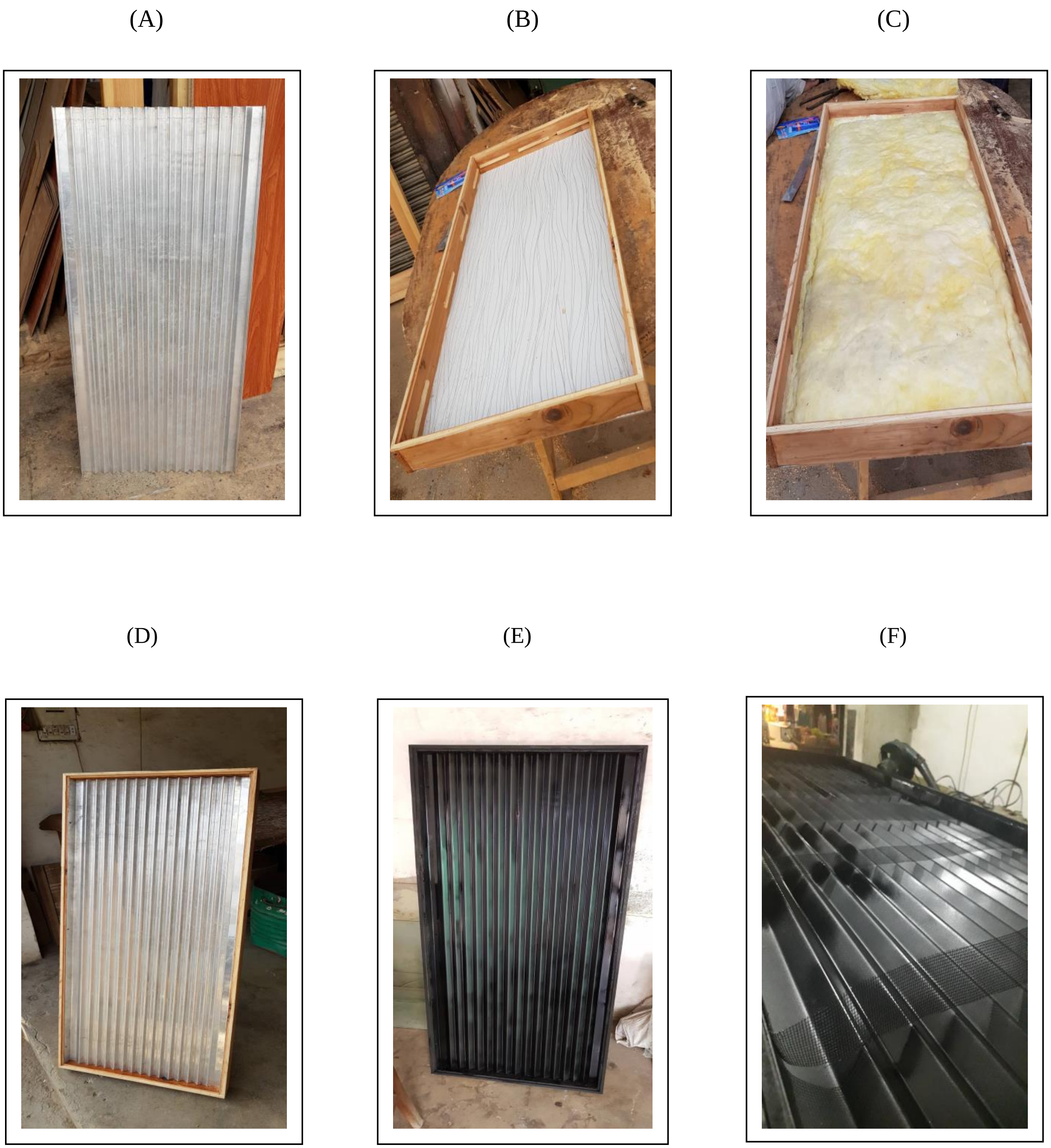


Figure 37: Steps of Solar Collector Manufacturing

4.2 Heat Exchanger:

The second component of this project is the Heat Exchanger that serves as the thermal energy storage unit. The heat exchanger had been filled with the Phase Change Material and the process of charging and discharging of the PCM takes place inside the heat exchanger so it is more like a Solar battery.

For the manufacturing of heat exchanger, we initiated by using the already acquired 1 mm thick aluminum sheet and bended it to form an outer aluminum box. Inside the box, an S-shaped copper tubing was introduced for the air to flow through it. The copper tubes were bent to form a compact network. To enhance the heat transfer further, copper stripes were used as fins and were welded with the copper tubing.

In the next step, this whole tubing network was riveted with the base and sides of aluminum box and the rest of the sides of aluminum box were welded. The box was then filled with the phase change material i.e. Glycerin.

The heat exchanger was finished by equipping it with glass wool all around as thermal insulation and wrapped in aluminum foil for further insulation and neatness.

(A) (B)

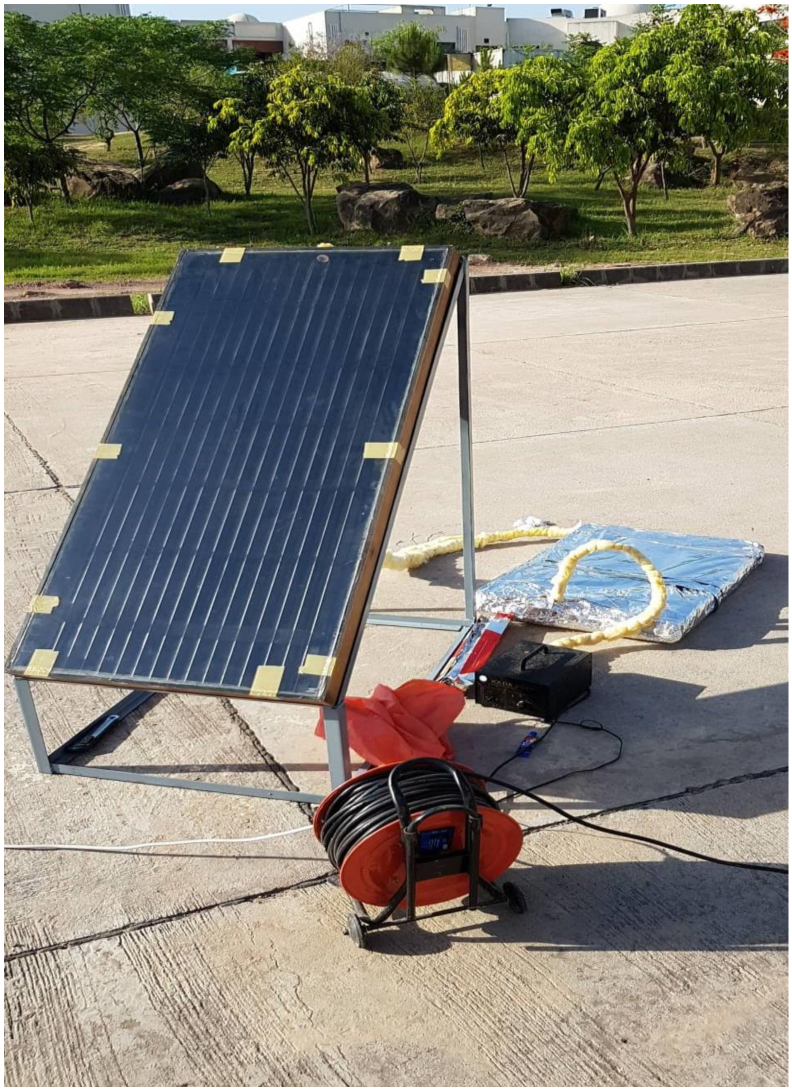
Figure 38: Steps of Heat Exchanger Manufacturing

4.3 Piping Network and Connections:

The supplementary components of this project are the piping network and connections that serve as the flow path of the solar fluid i.e. air through the system.

The air enters the system through the circular inlet in the solar collector and passes through the wide air passage before entering the piping network again and passing through the copper tubing across the heat exchanger. The pipes and the connections had been clamped for reduction in losses. After flowing through the copper tubing, the air goes into the model space and all this movement of air is supported by a small scale 20 W gas compressor that sucks the air and creates an induced draught. This facilitates the air flow through the entire system.

For reduction in losses, the entire piping network was covered by glass wool as thermal insulation and wrapped up further using aluminum foil for enhanced thermal insulation and neatness.



EXPERIMENTAL ANALYSIS

After the fabrication of the heat exchanger, the next phase was to subject the Space heating system to actual conditions. For this purpose the following sensors and equipment were used.

- 1x DS18B20 Thermocouple Sensors
- 2 x DHT11 sensors

- Anemometer

The ambient temperature of air was measured through the anemometer. One DHT11 was placed inside the outlet pipe of the solar collector. The second DHT11 sensor was placed on the inlet of the heat exchanger while DS18B20 sensor was placed inside the space to check the temperature inside the room. Flowrate inside the space was also measured through the anemometer.

The sensors were programmed using Arduino and the result was displayed on the computer screen. The following experiment was carried out for 4-5 hours during the day to get the best results.

4.4 Collector Outlet temperature:

The temperature of air at the collector outlet kept on rising till it reached 63.56 degrees. This was the recorded collector temperature. It is important to note that during the subsequent testing of the collector, its outlet air temperature rose as high as 70 Degrees. Here the worst case situation was picked and hence selected for calculations.

4.5 Collector Power:

The power provided by the Solar collector is given by the following formula.

$$q = \dot{m}\, C_p \Delta T$$

Applying it gives us a power of approximately 1.8 kW.

4.6 Heat Exchanger Inlet temperature:

The temperature of the heat exchanger was also measured as the system was kept on. The temperature at the inlet of the exchanger when the collector outlet air temperature was recorded as 63.75 C was recorded to be 58 degrees.

4.7 Heat Exchanger Storage Capacity:

The heat exchanger storage capacity is measured by the following formula:

$$q = \dot{m}\, C_p \Delta T$$

This comes out to be 600 W.

4.8 Space Inlet temperature:

After passing through the heat exchanger, the air finally enters into the space. Here a DS18B20 sensor was placed to note the temperature .The space inlet temperature came out to be 37.5 degrees.

4.9 Space Inlet Flowrate:

The space inlet flowrate was measured by placing the probe of the anemometer in front of the incoming air. The flow rate was measured to be 0.04 kg/s.

CHAPTER 5

DISCUSSION

Based on the theoretical analyses and the experiments carried out, we were able to accomplish the desired objectives of our project. The experimental results showed favorable compliance with the theoretical results, with this we conclude our project "Solar Space Heating Integrated with Thermal Energy Storage System". A brief conclusion of the project along with future recommendations have been discussed below.

CONCLUSION

- Viable, economical, feasible and sustainable utility incorporating solar energy as the power generation source.
- PCM has been incorporated and is to be used for sustainability to provide the required outlet temperature during night time when there is no solar radiation intensity.
- Theoretical calculations and mathematical modelling of solar collector provides an optimized design both with performance and cost point of view.
- The finalized dimensions of 1.3*0.55 m with a 0.03 m air passage and glass wool as thermal insulation. Apart from solar collector, another important factor is the Phase Change Material (PCM) selection. Through rigorous inspection and studies, Polyethylene glycol 600 has been selected as our PCM.
- The next step was the selection and analysis of an optimized design of heat exchanger for the storage and effective heat transfer. For this purpose, through studies and simulations, a rectangular heat exchanger with a zig-zag passage has been selected which contains small aluminium covered plates that contains PCM slabs. The whole system is to be well insulated with glass wool and rubber piping is to be incorporated in it for the air flow. For the forced convection, a small induced draft fan with our required flowrates is to be incorporated which would be powered by a small solar panel to make the utility independent.
- This covers the major aspects of the project and the theoretical calculations show the desired output conditions with the ambient conditions of Islamabad during winters as the real life data for calculation and simulation purposes.

RECOMMENDATIONS

1) More PCM mass can be incorporated to increase the capacity of the system during night time. We had to keep our system economically feasbile and compact so we plan on using approximately 3 kgs of PCM initially.
2) The system can be automated by integrating with control systems in order to cater for output variations.
3) Automation can be done by using variable flowrate valves and fans etc. to adjust to variable input conditions to give specific required outlet conditions.
4) An auxiliary system can be integrated as a backup system to account for the days when the environmental conditions (radiation intensity, wind speed etc.) are not suitable.

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APPENDIX 1: MATLAB CODE

```
%--------------------Fluid properties at 20 Degrees Celsius--------------

clear all
m_a = 0.018;  %mass flowrate in kg/s
k = 0.026341;  %thermal conductivity in W/m.K
u =1.868*10^-5;         %dynamic viscosity in kg/m.s
rho =1.1649;       %density in kg/m^3
Cp =1006.5;        %specific heat of air in J/kg.K
Pr =0.71375;        %Prandtl number

%---------------------Atmospheric Working conditions---------------

t_atm = [6.73 7.45 6.7 7.26 6.67 5.15 4.76 4.98 6.5 7.96 9.95 11.68 12.71
12.85 12.7 12.6 11.95 10.01 8.95 8.86 7.85 7.15 5.96 5.1]; %Hourly
atmospheric temperature in degrees celsius
v_wind = [2.38 2.15 0.98 3.31 1.95 2.76 2.48 1.88 2.06 2.93 3.45 4.1 3.85
4.73 4.6 4.7 3.81 3.8 2.25 2.41 0.56 0.73 0.05 0.7];  %Hourly wind speed in
m/s
G_Rad = [0 0 0 0 0 0 0 89.45 460 662.8 750.85 812.26 825.06 817.83 782.86
667.87 385.98 36.42 0 0 0 0 0 0];  %Hourly normal global radiations in W/m^2

%---------------------Properties of collector-----------------------

L =1.3;         %Length of Collector in meters
W =0.55;         %width of Collector in meters
Gap =0.03;       %Air Gap between glass and absorber in meters
A_xsec = W*Gap;    %Area of cross section for passage of air in m^2
A_top = L*W;       %Area of the absorber plate and glass cover from top in
m^2
Dh = (2*A_xsec)/(W+Gap); %Hydraulic Diameter
E_1 =0.88;       %Emissivity of glass
E_2 =0.12;       %Emissivity of plate
Ab_1 =0.06;       %Absorptivity of glass
Ab_2 =0.95;       %Absorptivity of plate
```

```
Tao =0.84;        %Transmisivity of glass
sigma = 5.67*10^-8;     %boltzmann constant
k_ins =0.04;        %thermal conductivity of insulation
x_ins =0.035;        %thickness of insulation

%-------------------Initial Hour---------------------


T_amb = t_atm(1)+273;
T_g = T_amb;         %Glass cover temperature
T_ap = T_amb;        %absorber plate temperature
T_f = T_amb;         %Mean fluid temperature
T_i = T_amb;         %Inlet fluid temperature
T_sky = 0.0522*T_amb^1.5        %sky temperature
V = v_wind(1);
Rad = G_Rad(1);
H_wind = 5.7+(3.8*V);        %Heat Transfer Coefficient of wind
H_gs = sigma*E_1*(T_g+T_sky)*(T_g^2+T_sky^2);    %Heat Transfer from glass
surface to the sky
H_gap = (sigma*(T_g^2+T_ap^2)*(T_g+T_ap))*((1/E_1)+(1/E_2)-1)^-1;     %Heat
Transfer in the air gap
Z = (2*m_a*Cp)/A_top;     %Coefficient

%CALCULATIONS FOR FLOW PROPERTIES


Re = (m_a*Dh)/(A_xsec*u)        %Reynolds number
a = 0.0019;
b = 0.00563;
m = 1.71;
n = 1.17;
Nu_infinity = 5.4;

if Re<2300
    disp('Laminar Flow')
    Nu_1 = a*(Re*Pr*(Dh/L))^m;
    Nu_2 = 1+b*(Re*Pr*(Dh/L))^n;
    Nu = Nu_infinity+(Nu_1)/(Nu_2);
elseif Re>6000
```

```
    disp('Turbulent')
    Nu = 0.023*Re^0.8*(Pr^0.4);
else
    disp('Transition Flow')
    Nu = 0.116*((Re^0.68)-125)*(Pr^0.33)*(1+(Dh/L)^0.68);
end

h = (Nu*k)/Dh;      %heat transfer coefficient for forced convection
h_g = h;
h_ap = h;
U_t = H_wind+H_gs;  %top loss
U_b = 1/((k_ins/x_ins)+(1/H_wind));    %bottom loss
SR_g = Ab_1*Rad;        %Solar Radiation absorbed by top cover
SR_ap = Tao*Ab_2*Rad;   %Solar Radiation absorbed by absorber
A = [(h_g+H_gap+U_t) -h_g -H_gap; h_g -(h_g+h_ap+Z) h_ap; -H_gap -h_ap
(h_ap+H_gap+U_b)];
T = [T_g; T_f; T_ap];
B = [(U_t*T_amb)+SR_g; -(Z*T_i); SR_ap+(U_b*T_amb)];
X = A\B;

while(X-T>0.01)
    T_g = X(1);
    T_f = X(2);
    T_ap = X(3);
    T_i = T_amb;
    H_wind = 5.7+(3.8*V);       %Heat Transfer Coefficient of wind
    H_gs = sigma*E_1*(T_g+T_sky)*(T_g^2+T_sky^2);    %Heat Transfer from
glass surface to the sky
    H_gap = (sigma*(T_g^2+T_ap^2)*(T_g+T_ap))*((1/E_1)+(1/E_2)-1)^-1;
%Heat Transfer in the air gap
    Z = (2*m_a*Cp)/A_top;     %Coefficient
    h = (Nu*k)/Dh;      %heat transfer coefficient for forced convection
    h_g = h;
    h_ap = h;
    U_t = H_wind+H_gs;  %top loss
    U_b = 1/((k_ins/x_ins)+(1/H_wind));    %bottom loss
    SR_g = Ab_1*Rad;        %Solar Radiation absorbed by top cover
    SR_ap = Tao*Ab_2*Rad;   %Solar Radiation absorbed by absorber
```

```
    A = [(h_g+H_gap+U_t) -h_g -H_gap; h_g -(h_g+h_ap+Z) h_ap; -H_gap -h_ap
(h_ap+H_gap+U_b)];
    T = [T_g; T_f; T_ap];
    B = [(U_t*T_amb)+SR_g; -(Z*T_i); SR_ap+(U_b*T_amb)];
    X = A\B;
end

i = 1;
hour = i;
X;
glass_temp(1) = X(1);
absorber_temp(1) = X(3);
Fluid_temp(1) = (2*X(2))-T_i;      %this is the exit temperature of the fluid

%---------------------For Next 23 Hours--------------------

for(i=2:1:24)
    T_amb = t_atm(i)+273;
    T_g = X(1);         %Glass cover temperature
    T_ap = X(3);        %absorber plate temperature
    T_f = X(2);         %Mean fluid temperature
    T_i = T_amb;         %Inlet fluid temperature
    T_sky = 0.0522*T_amb^1.5;        %sky temperature
    V = v_wind(i);
    Rad = G_Rad(i);
    H_wind = 5.7+(3.8*V);        %Heat Transfer Coefficient of wind
    H_gs = sigma*E_1*(T_g+T_sky)*(T_g^2+T_sky^2);    %Heat Transfer from
glass surface to the sky
    H_gap = (sigma*(T_g^2+T_ap^2)*(T_g+T_ap))*((1/E_1)+(1/E_2)-1)^-1;
%Heat Transfer in the air gap
    Z = (2*m_a*Cp)/A_top;     %Coefficient
    h = (Nu*k)/Dh;       %heat transfer coefficient for forced convection
    h_g = h;
    h_ap = h;
    U_t = H_wind+H_gs;  %top loss
    U_b = 1/((k_ins/x_ins)+(1/H_wind));    %bottom loss
    SR_g = Ab_1*Rad;         %Solar Radiation absorbed by top cover
```

```
    SR_ap = Tao*Ab_2*Rad;   %Solar Radiation absorbed by absorber
    A = [(h_g+H_gap+U_t) -h_g -H_gap; h_g -(h_g+h_ap+Z) h_ap; -H_gap -h_ap
(h_ap+H_gap+U_b)];
    T = [T_g; T_f; T_ap];
    B = [(U_t*T_amb)+SR_g; -(Z*T_i); SR_ap+(U_b*T_amb)];
    X = A\B;
    hour = i;
    glass_temp(i) = X(1);
    absorber_temp(i) = X(3);
    Fluid_temp(i) = (2*X(2))-T_i;
end

glass_temp()
absorber_temp()
Fluid_temp()

%--------------------------Graph Plotting Function---------------

time = 1:24;
subplot(1,1,1)
plot(time,Fluid_temp-273,'b*-');
hold on
plot(time,glass_temp-273,'m-');
plot(time,absorber_temp-273,'k-');
plot(time,t_atm,'g-')
legend('Output Air Temperature','Cover Temperature','Absorber
Temperature','Inlet Temperature')
xlabel('Time(hours)')
ylabel('Temperature(C)')
title('January 15')
set(gca,'Xtick',0:1:24)
set(gca,'Ytick',0:10:600)
axis([0,24,0,100])
grid on
```