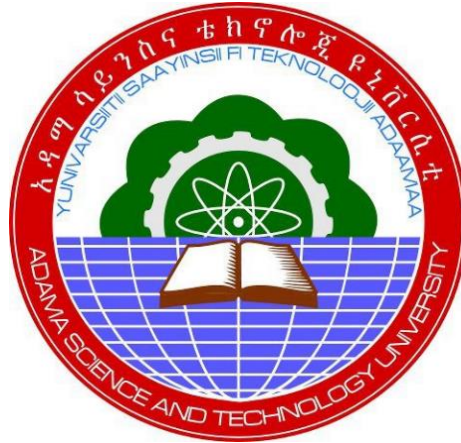


EXPERIMENTAL INVESTIGATION OF PCM INTEGRATED INDIRECT SOLAR COOKER



BY:

MEKONEN YALELET GETNET

Thesis Submitted to the Department of Thermal and Aerospace Engineering
School of Mechanical, Chemical, and Material Engineering

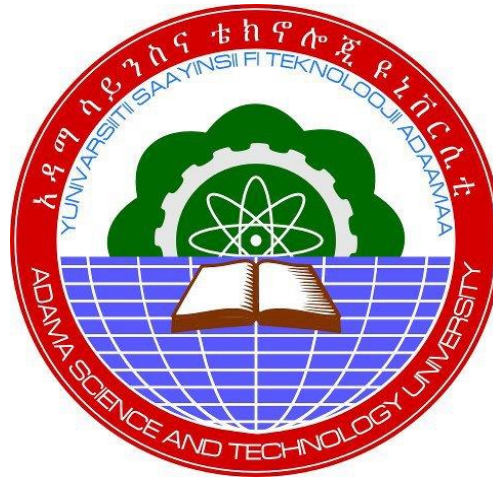
Office of Graduate Studies

Adama Science and Technology University

July 2020

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By

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ADVISOR: Dr. Dawit Gudeta

A Thesis Submitted to the Department of Thermal and Aerospace Engineering
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July, 2020

Adama, Ethiopia

DECLARATION

First, I hereby declare that this M.Sc. thesis entitled “**Experimental Investigation of PCM Integrated Indirect Solar Cooker**” is my original work and that all sources and materials used for this thesis have been duly acknowledged. This thesis has been submitted in partial fulfillment of the requirements for M.Sc. degree in Thermal Engineering at Adama Science and Technology University, School of Mechanical, Chemical and Material Engineering. I solemnly declare that this thesis is not submitted to any other institution anywhere for the award of any academic degree, diploma, or certificate. Quotation from this thesis is allowable without special permission if an accurate acknowledgement of source is made.

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Signature

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This is to certify that the above statement made by the candidate is correct to the best of my knowledge and belief. This has been submitted for examination with my approval.

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We, the undersigned members of the Board of Examiners of the final open defense by **Mekonen Yalelet Getnet** have read and evaluated his thesis entitled “**Experimental Investigation of PCM Integrated Indirect Solar Cooker**” and examined the candidate. Therefore, this is to certify that the thesis has been accepted in partial fulfillment of the requirement of the Degree of Master of science in Thermal Engineering.

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DEDICATED
TO
MY MOTHER MASTEWAL FENTE
and
MY FATHER YALELET GETNET

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ABSTRACT

Cooking is one of the most essential human activity in everyday life. The energy demand for cooking is fulfilled by conventional energy sources like bio-fuels, fossil fuels and electricity. Environmental degradation, harmful gas emissions, and the increase in prices of fuel and electricity are the main driving forces behind the need for more effective utilization of renewable energy. Among the alternative energy sources, solar energy is a clean, abundant and easily accessible form of renewable energy. However, its intermittent and dynamic nature makes it difficult to utilize it effectively. The thermal energy storage (TES) system becomes a solution to those natures and the time-mismatch between energy demand and supply.

The major objective of this thesis is to design and experimentally investigate the performance of locally constructed indirect solar cooking system using a modified flat plate collector as a heat source integrated with a phase change material (PCM) based latent heat storage (LHS) system. Numerical simulation and experimental investigation are done to study the performance of individual system components. Good agreement was obtained between the numerical and experimental results. From the analytical result and estimation of solar radiation data, daily average enhancement of 42.3% on the reflected solar radiation is found using four reflectors on each side of the FPC.

Comparative study on the performance of FPC with and without reflector are conducted using no-load experimental test. The result showed that the maximum average absorber plate temperature, maximum outlet water temperature, maximum energy and exergy efficiency of FPC without and with reflectors are found to be 364K, 344.1K, 63%, and 42% and 398.25K, 376.65K, 79%, and 48% respectively. The experimental result of LHS using stearic acid showed that time it takes and amount of sensible, latent and total energy stored/released during charging/discharging processes are found to be 141min, 1.89MJ, 3.52MJ and 5.41MJ and 153min, 1.57MJ, 3.01MJ and 4.58MJ respectively. The result of no-load test showed that maximum temperature of 372.9K and 369.1K is achieved at 13:00pm in the storage vessel and cooking pot surface respectively. After no-load test established the cooking ability of the cooker, water boiling test and different actual load test are performed.

The performance test revealed that the maximum amount of first figure of merits (F1) obtained was $0.09\text{K}^2/\text{W}$ and $0.21\text{K}^2/\text{W}$ for the cooker with and without reflector on FPC respectively. Second figure of merits(F2) for the cooker with reflector on the FPC was evaluated from water boiling test and found to be 0.28. The sensible and latent efficiency of the cooker were 29.6% and 27.3 % respectively. Finally, economic analysis was performed and the pay-back period of the cooker was found to be 3.5years.

Key words: Indirect solar cooker, Modified FPC, Stearic acid, PCM.

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ABBREVIATIONS

MoWIE	Ministry of Water, Irrigation and Energy
ENMSA	Ethiopian National Metrological Service Agency
HTF	Heat Transfer Fluid
FPC	Flat Plate Collector
PCM	Phase Change Material
LHS	Latent Heat Storage
SHS	Sensible Heat Storage
CFD	Computational Fluid Dynamics
TES	Thermal Energy Storage
PTC	Parabolic Trough Cooker
EHC	Effective Heat Capacity
CCHTP	Coupled Conjugate Heat Transfer Problems
STP	Standard Temperature and Pressure
LSC	Life cycle saving
LCC	Life cycle cost

NOMENCLATURE

G_{sc}	Solar constant (W/m^2)
n	Day of the year
G_{ext}	Extraterrestrial radiation (W/m^2)
G_o	Extraterrestrial solar radiation on a horizontal surface ($KW/m^2/day$)
H	Extraterrestrial radiation on a horizontal surface ($KW/m^2/day$)
$\bar{n}$	Daily hours of bright sunshine,
$\bar{N}$	Maximum possible daily hours of bright sunshine
a_1 and b_1	Regression coefficients
I	Global radiation (W/m^2)
H_o	Daily extraterrestrial solar radiation on a horizontal surface ($KW/m^2/day$)
H_G	Monthly average daily global solar radiation ($KW/m^2/day$)
H_d	Monthly average daily diffuse radiation on horizontal surface ($KW/m^2/day$)
H_b	Monthly average daily beam radiation (KW/m^2)
I_G	Monthly average hourly global radiation on a horizontal surface (W/m^2)
I_d	Monthly average hourly diffuse radiation on a horizontal surface (W/m^2)
I_T	Incident solar radiation on an inclined surface (W/m^2)
I_{d-sky}	Sky-diffuse radiation (W/m^2)
I_{d-gr}	Reflected radiation from the ground (W/m^2)
F_s	View factor to the sky
F_g	View factor to the ground
R_b	Beam radiation factor
T	Temperature (K)
$I_{tot-col}$	Total solar radiation on the modified FPC collector surface (W/m^2)
I_{ref}	Reflected solar radiation from all side reflectors (W/m^2)
S_{rc}	Summation of the reflectors shading factors
I_{ref-lr}	Reflected solar radiation from the left side reflector (W/m^2)
I_{ref-rr}	Reflected solar radiation from the right-side reflector (W/m^2)
I_{ref-tr}	Reflected solar radiation from the top reflector (W/m^2)
I_{ref-br}	Reflected solar radiation from the bottom reflector (W/m^2)
A_a	Absorber area (m^2)
A_p	Aperture area (m^2)

A_g	Gross area (m^2)
CR	Concentration ratio
W	Width of the collector (m)
X_t	Tube spacing (m)
n	Number of tubes
$\dot{m}$	Mass flow rate(kg/sec)
T_{in}	Inlet temperature (K)
T_{out}	Outlet temperature (K)
C_p	Heat capacity (KJ/kg.K)
$\dot{Q}_u$	Useful heat gain (KW)
$\dot{Q}_o$	Rate of heat loss (KW)
U_L	Overall heat loss coefficient ($m^2 \cdot K/W$)
T_{pm}	Mean plate temperature (K)
T_a	Ambient temperature (K)
S	Total absorbed radiation(W/m^2)
F'	Collector–efficiency factor
D	Tube diameter(mm)
T_b	Fin base temperature (K)
h_{fin}	Convective heat transfer coefficient of fin ($W/m^2 \cdot K$)
C_b	Bond conductance (W/m K)
F_R	Collector heat removal factor
h_w	Wind heat transfer coefficient ($W/m^2 \cdot K$)
U_t	Top loss coefficient ($m^2 \cdot K/W$)
U_b	Bottom loss coefficient ($m^2 \cdot K/W$)
T_{sky}	Sky temperature (K)
T_g	Glass cover temperature(K)
h_b	Bottom convective heat transfer coefficient ($W/m^2 K$)
$\hat{h}$	Specific energy flow (KJ/kg)
$\dot{Q}_k$	Rate of heat transfer (KW)
$\dot{W}$	Rate of work done (KJ/sec)
$\dot{S}_{gen}$	Entropy generation rate (W/K)
$\dot{E}x_{in,}$	Rate of inlet exergy (W)

$\dot{E}x_{out}$	Outlet exergy rate (W)
$\dot{E}x_{dest}$	Destructed exergy rate (W)
$\dot{E}x_{in.rad}$	Exergy of absorbed solar energy from sun (W)
$\dot{E}x_{in.f}$	Exergy of collector inlet heat transfer fluid (W)
$\dot{E}x_{out.f}$	Exergy of the outlet fluid (W)
V_{PCM}	Volume of PCM (m^3)
V_{tube}	Tube volume (m^3)
D_{cyl}	Diameter of PCM containment cylinder (m),
L	Length of LHS unit (m)
R_o and R_i	Outer and inner diameter of HTF tube inside the shell (m),
n	Number of tubes
T_L	Liquidus temperature (K)
T_S	Solidus temperature (K)
T_m	Melting point of PCM (K)
Cp_s and Cp_l	Specific heat capacity of PCM at solid state and liquid state (KJ/kg.K)
$L_{F.pcm}$	latent heat of fusion for the PCM (KJ/kg)
Δt_{ch}	Total charging times (hr)
Δt_{dis}	Total discharging times(hr)
P	Cooking power (KW)
Ps	Standardized cooking power (KW)
F_1	First figure of merits ($m^2 \cdot K/W$)
F_2	Second figure of merits

GREEK SYMBOL

δ_b	Thickness of bottom plate material (m)
ϵ_p	Emittance of absorber plate (%)
ϵ_g	Emittance of glass cover (%)
δ	Sheet thickness (m)
η_o	Optical efficiency of glass cover (%)
ρ_g	Reflectance of the ground (%)
γ_1, γ_2	Tilt plane angle of left and right reflectors (°)
α_1, α_2	Tilt plane angle of top and bottom reflectors (°)

σ_1, σ_2	Left and right incident angle (°)
χ_1, χ_2	Top and bottom incident angle (°)
α	Solar altitude angle (°)
β	Slope or tilt angle (°)
θ_z	Zenith angle (°)
γ	Surface azimuth angle (°)
ω	Hour angle (°)
θ	Angle of incidence (°)
ϕ	Latitude angle (°)
δ	Declination angle (°)
η_{th}	Thermal energy efficiency (%)
η_{ex}	Exergy efficiency (%)
η_o	Optical efficiency (%)
θ_f	Melt fraction (%)
η_{char}	Charging efficiency (%)
η_{dis}	Discharging efficiency (%)
$\eta_{en.overall}$	Overall energy efficiency (%)
$\psi_{exe overall}$	Overall exergy efficiency (%)
$\eta_{sensible}$	Sensible cooking efficiency (%)
η_{latent}	Latent efficiency of cooker (%)
$\eta_{overall}$	Overall system efficiency (%)

CHAPTER ONE

INTRODUCTION

1.1 Background of the Study

Ethiopia, similar to countries located in the tropical region, receives very high solar energy. The average solar energy potential of the country is about 5.2 kWh/m² per day with more than 275 sunny days in a year. As per Ministry of Water, Irrigation and Energy (MoWIE, 2017), the potential varies from season to season, with the lowest potential being 4.55kWh/m² per day, and the highest potential being 6.25kWh/m² per day (MoWIE, 2017). Despite of such potentials, the utilization of solar energy is insignificant and almost all the energy demand in the country is still fulfilled by other energy sources. There are several problems associated with the use of primary energy sources both in the world and our country. The use of bio fuels causes some serious ecological problems such as deforestation and burning fossil fuels release harmful emissions such as carbon dioxide, nitrogen oxides, aerosols, etc. which affect the local, regional and global environment (Ademe and Hameer, 2018). Several risks associated with the use of non-renewable energy source forces world public unions, international and national institutions, and other organizations to urgently need other renewable energy source and to transform present energy systems into a more sustainable basis. Exploiting other available and renewable energy resources at local level through appropriate designs of energy technologies seems a more sustainable solution. Among the alternative energy sources, solar energy is a clean, abundant and easily accessible form of renewable energy. However, its intermittent and dynamic nature makes it difficult to utilize the energy effectively. The thermal energy storage (TES) system becomes a solution to those natures and the time-mismatch between energy demand and supply.

1.2 Cooking Requirement

The daily useful energy requirement for cooking reportedly varies in the range of 1.7-2.7 MJ/person/day (Sunil and Tara, 2018). Therefore, a significant amount of fuel and electricity is required on daily basis for cooking of foods both at household as well as institutional level. The maximum internal temperature requirement of most of the foods is 80°C(Servsafe, 2012). But due to the heat losses from the cooking vessel, the temperature should reach higher than the internal temperature required by the food for effective cooking. The actual temperature range of most of food depends on the cooking method they require. The three most common methods of cooking are simmering, boiling and frying. Most of the vegetables and eggs are cooked in the simmering mode (temperature range, 85-90°C). In the case of cooking food by boiling, such as rice and lentils, the temperature of the food being cooked is about 100°C. High temperatures (140 -160°C) are required

for deep-frying. Example: Chicken, Meat, Fish are cooked in this frying range (Kumaresan, 2012). Thus, the cooking temperature range can be achieved by solar cooking systems if an effective method of solar energy utilization is applied.

During cooking, 20% of heat is spent in bringing food to boiling temperature, 35% of heat is spent in vaporization of water and 45% of heat is spent in convection losses from cooking utensils (Sedighi and Mostafa, 2014). Thus, the actual energy used for cooking is only 20% and this energy requirement for cooking can be supplied by the aid of solar cooking systems but an effective way of heat loss reduction is compulsory to get the required temperature and reduce cooking time. The convective heat losses, for example, can be easily avoided by insulating the side of the vessel and covering the vessel with a lid.

1.3 Solar Cooking Systems

The solar cooking system is one of the applications of solar thermal systems at which solar energy is used to heat, cook or pasteurize drink and other food materials. Solar cookers are heat exchangers designed to absorb solar radiation and convert it into heat for food processing (Silva *et al.*, 2005). Solar cooking technology is the solution to the world's problem of diminishing fuel-wood sources, consumption of electricity and other environmental problems associated with using wood and fuel for cooking demand (Marlett *et al.*, 2007).

There are many criterions for classification of solar cookers. One of the classifications is depending upon the heat transfer to the cooking vessel or pot (Silva *et al.*, 2005). Solar cookers can generally be divided into two: direct and indirect type cookers. The first type of solar cooker uses solar radiation directly in the cooking process. Box-type solar cookers, concentrating dish, parabolic dish and parabolic reflector are an example of direct types of solar cooker where the cooking pot is placed directly to the collector. The indirect cookers use solar radiation to heat up the heat transfer fluid (HTF), which transports this heat to the place of food processing (Silva *et al.*, 2005). In indirect type solar cookers, the pot is physically displaced from the collector, and a heat-transferring medium is required to convey the heat to the cooking pot. Flat plate collector solar cooker, evacuated tube collector cooker and parabolic reflectors solar cookers are types of indirect solar cookers (Schwarzer and Silva, 2008). Tangible electricity and fuel-saving and hence monetary savings served as an important motivator for institutions and households to use solar cooking technologies.

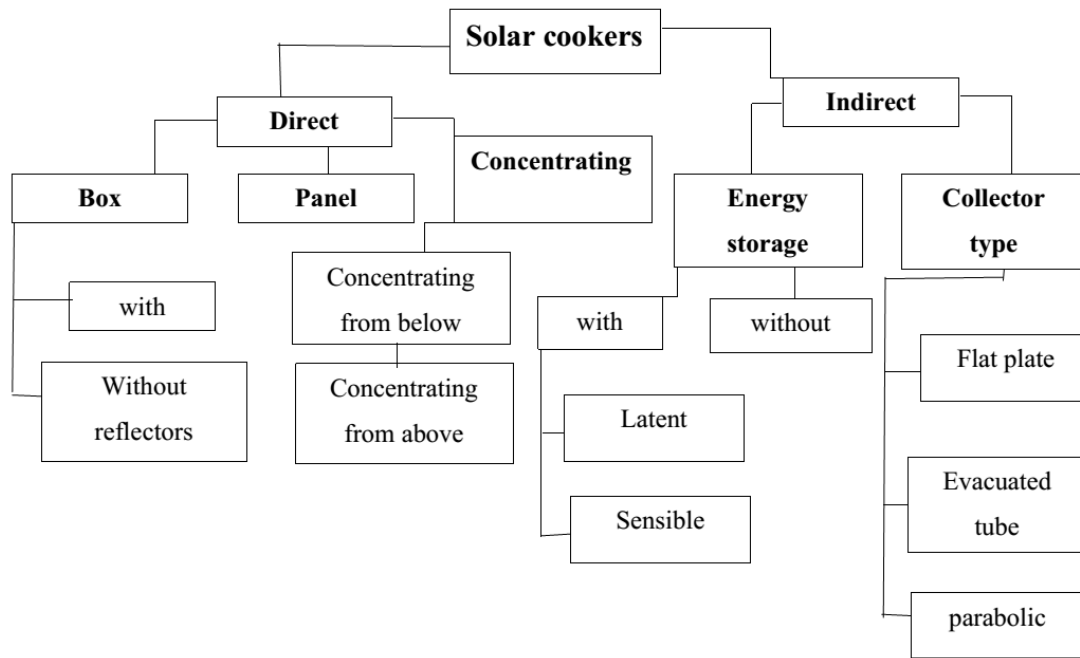


Fig.1.1 General classifications of solar cookers

1.4 Thermal Energy Storage System

Solar energy is a clean, abundant and easily accessible form of renewable energy. However, its intermittent and dynamic nature makes it difficult to utilize the energy effectively. The thermal energy storage (TES) system becomes a solution to those natures and the time-mismatch between energy demand at night and the external solar energy available during the day (Kousksou *et al.*, 2007). TES stores energy by heating, melting or vaporizing a storage medium and the stored energy is used at a later time for various applications by cooling or solidifying the storage medium. Due to the fluctuating nature of solar radiation throughout the day, a TES is required to be designed properly so as to achieve a continuous supply of heat (Dawit, 2018). Thus, TES systems are used to fully utilize solar energy and cover off-peak periods.

There are several types of TES systems such as the sensible TES through a temperature gradient, latent TES based on phase change material (PCM), and chemical TES through a chemical reaction. Among the available technologies for thermal storage systems, the latent heat storage (LHS) system can store a great amount of thermal energy in a small volume at constant temperature. Hence, this type of energy storage system is often used for different solar thermal applications.

LHS system have several advantages, which include; high storage capacity, low storage volume, and isothermal operation during the charging and discharging phases (Abokersh *et al.*, 2018). PCM is a kind of latent energy storage material that undergoes an isothermal or near isothermal phase transformation, has attracted considerable attention due to the high energy storage density and

compactness of the material (Zhangyuan *et al.*, 2015). The ideal PCM to be used for a latent heat storage system must meet the following requirements: highly sensitive, heat capacity and heat of fusion, stable composition, high density, and heat conductivity, chemical inert, non-toxic and non-inflammable, reasonable, and inexpensive (Arabacigil *et al.*, 2015).

1.5 Flat Plate Solar Cooker

The flat plate collector solar cooker often called flat plate solar cooker is one of the indirect type solar cookers at which one or more flat plate collectors with side reflectors are used to convert sun's solar energy into heat which is then transferred to the HTF to be used for heating the water or it transfers part of the sensible energy to the food in the cooking vessel. Usually, two or more side reflectors are used to provide additional solar radiation on the surface of the absorber. Such cookers usually have cooking pots installed indoors and thermal energy storage unit for later use during off sun-shine and in the night. The working fluid temperature, in good quality flat plate solar collectors, can reach a value greater than 100°C and using four flat plate reflectors concentration factor increases, leading to a significant increase in the temperature of the solar collector greater than 200°C (Zoran and Ljiljana, 2015). In that case, application of the thermal collectors is favorable which makes possible the use of those collectors in solar cooking systems (Silva *et al.*, 2002). The advantages of those cookers are: the possibility of cooking in internal rooms, high operating temperatures in a short period of time, allowing baking as well as frying, and the use of an energy storage tank to keep the food warm longer or allow nocturnal cooking (Silva *et al.*, 2005).

1.6 Statement of Problem

Bio-fuel has remained the most important source of energy in Ethiopia, which covers, about 80 per cent of the country's total energy needs. In the rural area households consume 92% of the total energy of which 99.5% constitute traditional bio-fuels and only 0.5% of the energy constitutes modern fuels. Similarly, In the rural area households consume 92% of the total energy of which 99.5% constitute traditional fuels and only 0.5% of the energy constitutes modern fuels (Woldeamanuel, 2017).

From the total energy consumption cooking applications use 80% energy supply which dominantly use traditional fuels as a source of energy (Ademe and Hameer, 2018). This has led to environmental degradation, deforestation and associated diseases due to indoor air pollution. With the rising cost of petroleum products and electricity combined with biomass energy scarcity, there is need to exploit solar energy for cooking and heating applications.

Properly designed solar cookers offer an alternative to bio-fuel stoves in areas where fuel is scarce and solar energy is available. Solar energy systems for cooking and heating have been utilized for several years but they are still out of reach mainly in developing countries of rural areas as well as

urban areas and institutions due to its high cost of installation as well as the limitation of not operating at night and during cloudy weather conditions (Sharma *et al.*, 2000; Buddhi *et al.*, 2003). Thus, in the present thesis effort has been made to properly design indirect solar cooker which utilize modified flat plate collector as a heat source and phase change material based thermal energy storage system for off-sunshine, cloudy period and late hour operation of the cooker.

1.7 Objective of the Study

This section describes the general and specific objective of the thesis.

General objective

The general objective of this thesis is to experimentally investigate the performance of indirect solar cooker using a modified flat plate collector as a heat source integrated with thermal energy storage system.

Specific Objectives

- Designing and developing prototype from locally available material for experimental investigation;
- Performing comparative study on the performance of flat plate collector with and without reflector;
- Investigating and evaluating the thermal performance of the PCM based LHS system;
- Investigating and evaluating the performance of developed indirect solar cooker.

1.8 Scope and Limitations

This study mainly focuses on experimental investigation of indirect solar cooking system by designing each cooking system components. The study also establishes the theoretical and numerical procedure for the design of system components and simulation of performance parameters. Both energy and exergy analysis for the flat plate collector and latent heat storage system is also included. Furthermore, comparison on the individual system performance with and without side reflector is done. Finally, cooking test (no-load and load test) and economic analysis for the developed prototype of the solar cooker is performed. Performing large scale economic feasibility study for the cooking system was the limitation of this thesis.

1.9 Organization of the Thesis

This thesis is composed of 8-chapters. After the previous chapter introduce the concept of solar cooking applications and system components, literature survey on the same was presented in chapter two with its summary and gap. The third chapter presents the material and methodology section of the work followed by chapter four which presents the estimation of solar potential for the selected

site. The fifth chapter formulate the theoretical design, sizing and numerical simulation of system components for individual performance investigation. After the sixth chapter describes the conceptual design working principle, construction process and the experimental setup of the prototype, Chapter seven presents the major finding and results of numerical and experimental analysis conducted with their implication. The last chapter includes the conclusion of the whole work and recommendation for further study.

CHAPTER TWO

LITERATURE REVIEW

2.1 Introduction

Many researches and developments in solar thermal energy utilization have been done in the past years to improve the efficiency and performance of such systems. There have been several innovations in the design of solar cooking systems through years of research, and still lots of researchers try to improve the efficiency and optimize the cost of solar cooking systems through new and innovative designs. In this section previous works associated with the solar cooking systems, for both domestic and institutional applications, thermal storage systems, and factors affecting the performance of flat plate collectors were reviewed. finally, the summary of all the literature and its gap is also included.

2.2 Solar Cookers

A solar cooker is a device which uses the energy of sunlight to heat, cook or pasteurize drink and other food materials. The first solar oven or solar cooker was invented in 1767 by a Swiss physicist, Horace de Saussure (Energy informative, 2019). Now a day's many sorts of solar cookers may be found in the literature. Moreover, new solar cookers designed with new improvements are continuously proposed, which requires continuous update of solar cooker classifications. Generally solar cookers can be classified as direct type and indirect type solar cookers.

2.2.1 Direct Type of Solar Cookers

Direct solar cookers may be considered the most common type available due to their ease of construction and low-cost material (Sedighi and Mostafa, 2014). There are many type and designs of direct type of solar cookers for domestic applications but commercially successful direct type cookers are box type and concentrating type cookers.

2.2.1.1 Box Type Solar Cookers

A solar box cooker is direct type of solar cooker basically consists of an insulated box with a transparent glass cover and reflective surfaces to direct sunlight into the box. The inner part of the box is usually painted black or covered with selective black painted cover in order to maximize the sunlight absorption. The glass cover provides a greenhouse effect in which it permits the passage of solar radiation but prevents it to get out from the cooking vessel and used as a cover to protect convective and irradiative heat losses. Different experiments have done on box type solar cookers in the past to measure the thermal efficiency, cooking power and time, and figure of merits. Ademe and

Hameer, (2018) conducted an experiment to evaluate the performance of double-glazed box-type solar cooker. The thermal performance was tested as per the ASAE International Test procedure and Bureau of Indian Standards (BIS) for testing the thermal performance of a box-type solar cooker. The obtained test results showed that cumulative efficiency of solar box cooker without application of wiper mechanism to be 22% and use of wiper mechanism results an increase to the cumulative efficiency by 9.4%. The standard boiling time of 1.43 kg of water was calculated to be 53.54 and 88.84 minutes for the cooker with and without the application of wiper mechanism respectively. The temperature distribution inside the box was simulated and the maximum wall temperature was found to be 139 °C where the experimental result was 161°C. The results also showed that using the vapor wiper mechanism reduces the boiling time by 35.3 minutes. Amanuel *et al.*, (2019) experimentally investigated the thermal performance of the developed solar box cooker with different reflector configurations. Two standard tests, namely stagnation test and sensible heat test were adopted in order to evaluate the thermal performance of the cooker. The highest maximum plate temperature of 148.7°C was achieved by the cooker with tracking reflector at optimal angle. The ratio of optical efficiency to heat loss factor (F1) and the product of heat exchange efficiency factor and optical efficiency (F2) were employed as a performance index. The experimental result showed that the highest computed value (F1) of 0.154°C m²/W, was obtained from the cooker with tracking reflector at optimal angle. Mulu *et al.*, (2014) performed theoretical and experimental comparison between box type solar cookers with and without internal reflector. Both the cookers were made from the same types of materials. The only difference was in the geometry of the inner metal box. The experimental result also indicated that the standard stagnation temperature of the cooker with reflector was higher than the cooker without reflector with average difference of about 22°C. Boiling tests showed that the cooker with reflector have lesser cooking time than the cooker without. Therefore, it can be concluded that the performance of box cookers can be enhanced by making appropriate side reflection.

2.2.1.2 Concentrating Type Solar Cookers

In concentrating solar cooker, food is directly cooked by absorbing the heat from the sun light without any interference of any material between the sun light and the cooking pot. It works based on the principle of solar optics in which it concentrates direct solar radiation on the bottom of the cooking pot to heat it and achieve extremely high temperatures (Herez *et al.*, 2018). Some of the previous works by applying concentrating type solar cookers are presented as follows; Panwar *et al.*, (2012) performed an experimental study to analyze the energy and exergy efficiency of domestic size parabolic solar cooker in actual use. In the experiment, the ambient air temperature, water temperatures profiles in the cooking pot, and solar radiation on a horizontal surface were monitored

at regular intervals of time. The wind speed was measured using an anemometer having accuracy of $1\% \pm 0.1$ m/s. The result showed that the maximum temperature of water was 368 K. The energy out of the cooker varied between 46.67 and 653.33 W, whereas its exergy output was in the range, 7.37-46.46 W. Maximum energy and exergy efficiencies of the cooker were experimentally evaluated, and it was about 46.82% and 32.97%, respectively. Mawire and Simeon (2014) conducted experiment to investigate the thermal performance of a cylindrical cavity receiver for parabolic dish concentrator. The thermal performance was evaluated using energy and exergy analyses. The receiver exergy rates and efficiencies were found to be appreciably smaller than the receiver energy rates and efficiencies. The exergy factor parameter was also proposed for quantifying the thermal performance. The exergy factor was found to be high under conditions of high solar radiation and under high operating temperatures. The result showed that the heat loss factor of the receiver was around 4.6 W/K and optical efficiency under high solar radiation conditions was found to be around 52% for parabolic dish system.

2.2.2 Indirect Solar Cookers

Solar cookers must meet certain social conditions besides cost and performance requirements to gain worldwide scale acceptance. It is obvious that most of users didn't prefer to cook outdoors in the hot sun. Thus, the energy from the sun needs to be transmitted to the kitchen for indoor cooking (Hussein *et al.*, 2008). Indirect solar cooker enables indoor cooking as it enables the heat required for cooking to be conveyed into a cooking unit installed inside a kitchen. It consists of a collector installed outside to gather heat. The cooking vessel is displaced from the collector in which the cooker is separated and protected from radiation as the cooking unit is installed indoor. Heat is transferred from the collector to the cooking pot by a heat transferring fluid, where a small circulating pump may be utilized to circulate the working fluid especially if the collector is kept at height above the cooking unit (Herez *et al.*, 2018). Indirect type of solar cookers may have a sensible or latent heat storage system. Fig.2.1 indicates schematic of an indirect type solar cooker with sensible heat storage unit. Many attempts have been made to design in-direct solar cooking systems incorporating a thermal storage system and analyses the performance those systems experimentally. The thermal energy storage system can store thermal energy in form of sensible or latent heat. In this section, some of flat plate collector and parabolic trough solar cookers with a thermal energy storage system are reviewed.

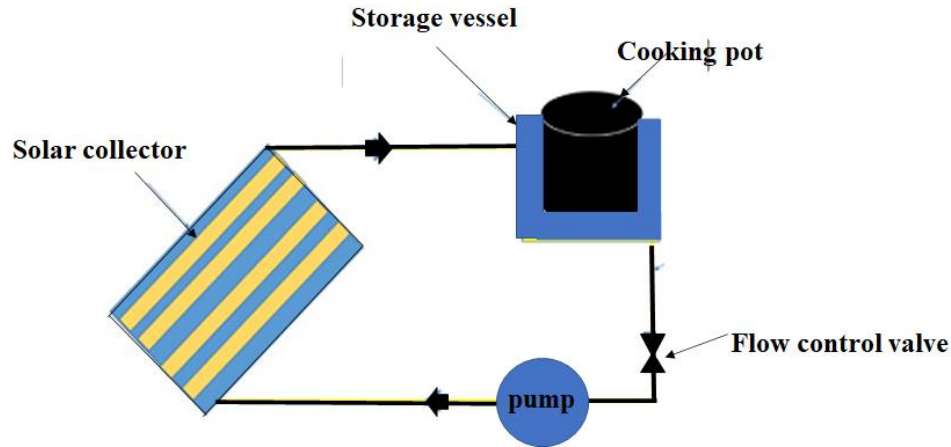


Fig.2.1 Schematic of Indirect type of solar cooker (Herez *et al.*, 2018)

2.2.2.1 Flat plate collector solar cookers

Flat plate solar cookers are indirect type cookers that uses flat plate collectors to harness solar energy. Fig.2.2 indicates the schematic of indirect type solar cooker based on flat plate collector. Those solar cooking systems may have having plane reflectors, with or without heat storage of units, have been developed and installed in different countries of the world (Hussein *et al.*, 2008). Those systems consist of one or more flat-plate collectors with a coated absorber and glass cover, cooking pots, and a storage tank. Reflectors provide additional solar radiation on the absorber surface. Oil, usually vegetable, is used as the heat transfer fluid. The oil is heated up in the collectors and moves by natural flow to the cooking unit, where it transfers part of its sensible energy to the cooking goods in double-walled pots. Manually controlled valves guide the oil flow-rate either to the pots or to the storage tank (Schwarzer and Silva, 2003).

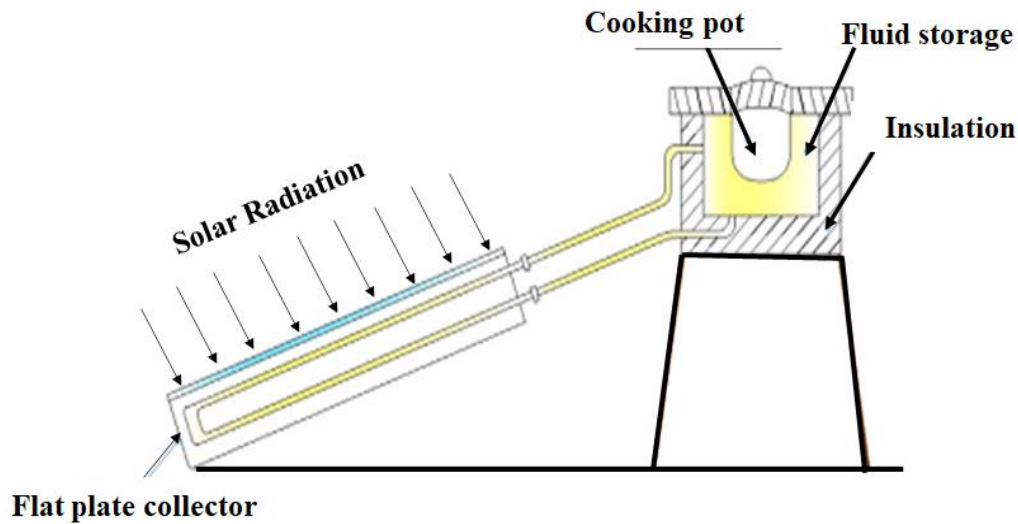


Fig.2.2 Indirect solar cooker with flat plate collector [Herez *et al.*, 2018].

The major system advantages are: the possibility of indoors cooking, the use of a thermal storage tank to keep the food warm for longer periods of time or night cooking, and the high temperatures of the working fluid reached in a short period of time, allowing fast cooking as well as frying and roasting. Cooking during early morning hours is possible for bigger systems (collector area greater than 4m^2) with storage volume greater than 100L (Schwarzer and Silva, 2008). Due to the oil natural circulation, there is no need for a pump, reducing installation and operational costs. Such system runs at ambient pressure and is self-regulated. The disadvantages are associated with the relatively high fabrication costs and the difficulty in finding all the required materials in non-industrialized countries. Experiments in many literatures showed that flat plate collector solar cookers with thermal energy storage have an improved cooking efficiency and reduced cooking time.

Ramadan *et al.*, (1988) designed a simple flat-plate solar cooker with focusing plane mirrors and energy storage system. Temperature distributions during the test periods through the absorbing plate, the glass covers, the cooking pot, the cooking fluid, the storage medium and the ambient were determined. The utilization of sensible heat storage medium such as sand was investigated thoroughly. A jacket of sand (1/2 cm thick) around the cooking pot has improved the cooker performance tremendously. The result showed that the cooker can be utilized for six hour per day of cooking time. Approximately 3 h/day of indoor cooking has been achieved. Overall energy conversion efficiency up to 28.4% has been obtained which was considered the best among other solar cookers.

Silva *et al.*, (2002) determines the experimental efficiency of a flat plate solar cooker with storage system. The system was composed of flat plate collectors, a cooking unit, control valves and a heat

storage tank. During experiment, temperatures in the cooking system and on the outside walls of the copper piping were measured by K-type thermocouples. Ambient temperature and the wind speed were measured with a PT-100 platinum resistance thermometer and an anemometer respectively. The measured data were used to determinate the heat sensible and boiling thermal efficiencies. The results showed good system performance, with the sensible efficiency varying from 0.34 to 0.38. The tests also showed that the cooker had capacity to cook large meals and keep them warm a whole day and can be used to prepare meal in institutions and schools.

To examine the testing procedures and characteristics of solar cookers, (Schwarzer and Silva, 2008) performed an experiment on flat plate solar cookers. The cooker was indirect type which allows indoors or outdoors cooking and was consists of one 1.7 m² flat plate collector, two 8L cooking pots and without thermal storage tank. For the average values during the test period: global radiation 900 W/m², direct radiation 750 W/m² and 5 kg of water per 1 m² of collector surface, the period of time to heat up water from 20⁰C to boiling(100⁰C) was approximately 60min.

Silva *et al.*, (2005) carried out an experimental testing for a comparative study between two solar cookers: Flat Plate Collector and Parabolic solar cooker. The sensible and latent efficiencies of the two cookers were determined. The result showed that the temperatures around 200⁰C were measured in the flat plate collector cooker, reaffirming the possibility of its use in cities and in the countryside. Fabrication cost for the two cookers were made and the results also showed a lower price for the parabolic cooker but the flat plate cooker had as advantages: reliability, security in operation, very low maintenance and a longer life time.

Wimmer *et al.*, (2017) reviewed solar cookers to analyze the user's acceptance on solar cooking technologies. In the review flat plate solar cooker that uses thermosiphon principle for hot oil circulation in the system was analyzed. The solar collector was set at an angle of 30–40⁰ to the ground. Four reflector panels were used to reflect more sunlight into the flat plate solar collector. Reflector films were applied on foldable polystyrene foam panels which were placed on the edges of the solar collector. The cooking area was designed by using a double-wall cooking vessel with a gap between the walls for thermal oil to run through. The result showed that the prototype can produce heat on a cooking vessel surface up to 165 ⁰C which is sufficient for simple food cooking, warming, and pasteurizing.

Hussein *et al.*, (2008) designed and experimentally tested a novel indirect solar cooker with outdoor elliptical cross section, wickless heat pipes, flat-plate solar collector and integrated indoor PCM thermal storage and cooking unit under actual meteorological conditions of Giza, Egypt. Two plane reflectors were used to enhance the insolation falling on the cooker's collector, while magnesium

nitrate hexahydrate (melting point = 89 °C, latent heat of fusion 134 kJ/kg) was used as the PCM inside the indoor cooking unit of the cooker. It was found that the average daily enhancement in the solar radiation incident on the collector surface by the south and north facing reflectors is about 24%. For the Experiment with different actual cooking loads, the cooking pots were loaded with actual meals twice a day. The first meal was comprised of 0.75 kg chicken + 1.25 kg potatoes + 0.25 kg onion + 0.5 kg tomatoes + 0.65L water and the second meal was comprised of 10 eggs (0.6 kg) + 1.5L water. The result showed that both the first and the second meals were completely cooked at 90min.

2.2.2.2 Parabolic Trough Solar Cookers

Parabolic through solar cookers are indirect type solar cookers which use a parabolic collector with a central receiver pipe to convert the reflected sunlight in to useful heat energy and then the heat will be transferred to the heat transfer fluid flowing through it. The heat transfer fluid is the conveyed to the cooking unit by an insulated connecting pipe to be utilized for cooking the food inside the cooking vessel. Usually latent heat thermal energy storage units are incorporated with such cookers to store heat energy for later use. Some of the previous works on parabolic trough solar cookers with a heat storage unit are presented below.

Haftom *et al.*, (2014) experimentally tested the performance of a solar cooker based on concentrating collector and implemented a way to increase its temperature and performance. Parabolic trough cooker (PTC) was constructed in a way allowing cooking to be done indoors, which the cooking sections were placed indoor whiles the collector parts out-door with soya bean oil conveying the energy from the absorber to the cooking stove. Ray tracing and standard stagnation tests were performed and the result showed that a 30 mm diameter copper pipe is the optimum size for the absorber. Experimental results revealed that the maximum temperatures at the mid absorber pipe was about 191°C and at the cooking stove was 119°C obtained. The efficiency of the system was also found to be 6%.

Mussard and Ole, (2013) performed two charging experiments on a solar heat storage coupled with a self-circulating solar parabolic trough filled with thermal oil (Duratherm 630). The absorber tube was not insulated in the first test, and insulated with a glass tube during the second test. The storage was mainly oil-based but contains a significant part of nitrate salts in order to store energy with latent heat (melting temperatures: (210–220°C)). The heat stored was intended for cooking purposes (boiling water and frying). The results showed that at low temperatures, the absorber without insulation is much more effective. When the storage temperature approached 200°C, the glass tube becomes an

advantage for the further heat collection at higher temperatures. But, above 200°C, it becomes difficult to collect without insulation around the receiver.

Bushnell, (1988) designed, developed and evaluates a solar energy storing heat exchanger as a step toward a solar cooking concept. Penta-erythritol (solid–solid phase change at 182°C) was used as energy storage material. The methods for describing the system performance were explained and applied to a test system containing a controllable replacement for the solar input power. The research work had a heat exchanger, which was connected to a concentrating array of CPC cylindrical troughs. Author also described the size of the solar collector area and mass of PCM needed to provide adequate energy for several family-size meals with sufficient storage to cook at night and 1 or 2 days later.

Kalogirou and Panayiotou, (2012) performed an experiment to evaluate the performance of parabolic collector having aperture area of 14.4 m² and a concentration ratio of 13.7 and which operates up to 200°C. The heat transfer fluid used in the test was water, which is pressurized to avoid boiling in the receiver. The result showed that the performance of the parabolic collector was very satisfactory and agrees with the performance curve given by the manufacturer.

2.3 Thermal Energy Storage Systems for Solar Cooking Applications

One of the major drawbacks of a solar cooker is its inability to cook during off sunshine hours or cloudy periods. In order to overcome the above constraint, solar systems should be integrated with energy storage systems. Generally, energy storage systems can be classified as mechanical, chemical, biological, magnetic and thermal energy storage systems (Kumaresan, 2012). Among those energy storage systems, thermal energy storage systems get more attention due to its ability to meet the society's needs for more efficient, environmentally friendly energy use in cooking, building heating and cooling, aerospace power, and other utility applications. From all the types of thermal energy storage systems, sensible heat storage (SHS) and latent heat storage (LHS) systems are the most common in storing solar energy for solar cooking applications. As shown in the Fig.2.3, (Ashish., 2019) classified thermal energy storage system as latent and sensible heat.

2.3.1 Sensible Heat Storage Systems (SHS)

Sensible heat storage systems have increasing heat capacity with temperature which makes them motivating options to store sensible part energy for cooking application. Due to its high specific heat and inexpensive for mid temperature range, water appears to be the most promising sensible heat storage material (Ashish, 2019). Concrete, rock, water and engine oil are also the example of sensible heat storage materials which commonly used in cooking applications. For instance (Cuce, 2018) designed a cylindrical box cooker to test bay-burt stone as a sensible heat storage material with the

aim of achieving continuous solar cooking. In this work, the presented design was compared with a normal box cooker. The result showed that while the thermal efficiency of the conventional cooker was between 14.1% and 21.2%, the designed cooker had efficiency between 21.7% and 35.3%.

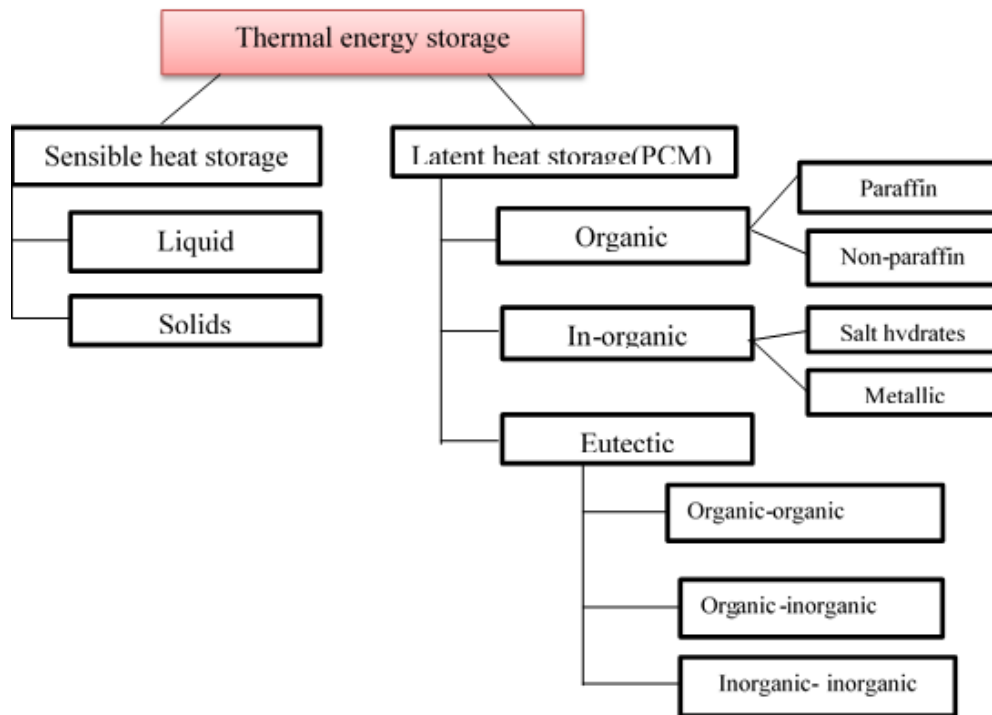


Fig.2.3 General classifications of thermal storage materials [Ashish., 2019]

2.3.2 Latent Heat Storage Systems

Latent heat storage systems also known as phase change materials store energy by using phase change material which allows storing significant amount of energy over narrow temperature range (John *et al.*, 2015). Fig.2.4 shows the comparison of stored heat with latent heat storage and sensible heat storage system. it is clear that storage of heat on the basis of latent heat is more efficient in comparison with sensible heat (Radim *et al.*, 2012). In the case of sensible heat, the heat added to the system results in temperature increasing of storage media whereas the accumulation of heat in the latent heat storage system occurs at constant temperature while the PCM is undergoing the phase change transition. The use of an LHS system using PCMs is an effective way of storing thermal energy and has the advantages of high-energy storage density and the isothermal nature of the storage process. The main advantage of using LHS is their storing capacity of heat up to 14 times more energy than SHS over relatively a small temperature interval (Sharma, *et al.*, (2009), Sarbu and Calin, (2018)). Initially, these materials act like SHS materials in that the temperature rises linearly with the system

enthalpy; however, later, heat is absorbed or release at almost constant temperature with a change in physical state.

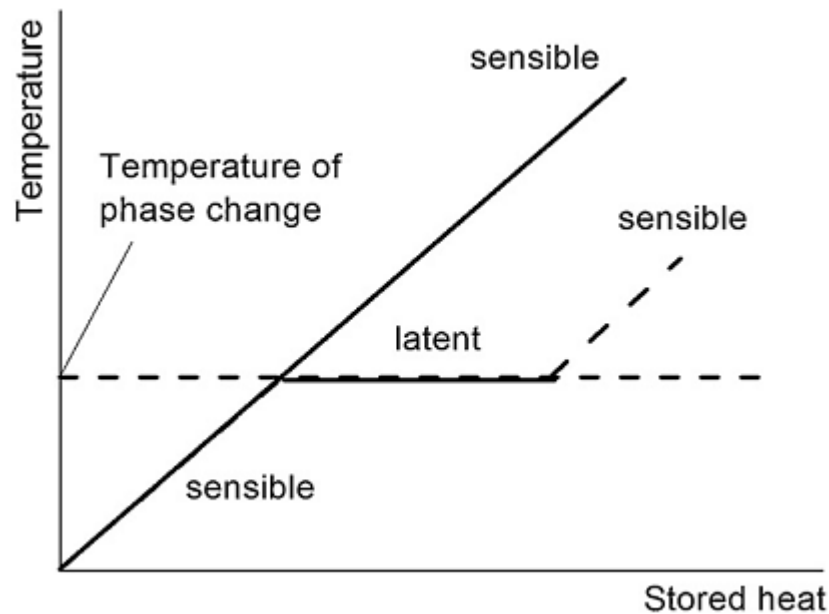


Fig.2.4 Comparison of sensible and latent heat accumulation [Radim *et al.*, 2012].

PCMs are characterized by their high thermal energy storage capacity and ability to absorb energy during heating process and release it to the environment during cooling process. During solid-liquid phase change for example, fusion absorbs energy and solidification releases it. Molten salts, paraffin waxes, eutectic salts, magnesium nitrate hexahydrate, Galactitol, magnesium chloride hexahydrate, stearic acid, acetamide, acetanilide and erythritol are some of the latent heat storage materials that can be selected for solar cooking (El-Sebaai *et al.*, 2011, Herez *et al.*, 2018. and Hussein *et al.*, 2008). Studies have been conducted with the LHS materials in solar cooker to cook the food at night and in the late evening. Domanski *et al.*, (1995) have studied the use of magnesium nitrate hexahydrate ($\text{Mg}(\text{NO}_3)_2 \cdot 6\text{H}_2\text{O}$) as a PCM for the heat storage. The result showed that solar cooker with PCM designed was able to cook the food in the late evening hours and during the non-sunshine hours. Chen *et al.*, (2008) numerically studied PCMs used as the heat storage media for solar box cookers. Magnesium nitrate hexahydrate, stearic acid, acetamide, acetanilide and erythritol were selected as PCMs. For a two-dimensional simulation model based on the enthalpy approach, calculations were made for the melt fraction with conduction only. For solar cooking systems, stearic acid and acetamide were found to be good compatibility with latent heat storage system. Sharma *et al.*, (2005) investigated thermal performance of a prototype solar cooker based on an evacuated tube solar collector with PCM storage unit. The design had separate parts for energy collection and cooking coupled by a PCM storage unit. The result revealed that noon cooking did not affect the evening cooking and evening cooking using

PCM heat storage was found to be faster than noon cooking. They also noted that the system is expensive but shows good potential for community applications. Buddhi and Sahoo (1997) designed and tested a solar cooker with latent heat storage for cooking food in the late evening. In the design, commercial grade stearic acid (melting point 55.8°C , latent heat of fusion 161 kJ/kg) was used as PCM and was filled below the absorbing plate. The experimental results demonstrate that using a phase change material as the storage medium in solar cookers is feasible for effective cooking and also it is possible to cook the food even in the evening with a solar cooker having latent heat storage. Lecuona *et al.*, (2013) used technical grade paraffin and erythritol as PCMs in a parabolic solar cooker. The PCMs were placed in the cooking vessel and divided between two pots. The result from the experiments revealed that simultaneous cooking and heat storing were possible with this method. They also used an insulated box to maintain the thermal energy of the PCMs. By applying insulation, the cooker could be used for cooking both breakfast and dinner. Mussard and Nydal, (2013) constructed an indirect parabolic solar cooker equipped with a sun tracking system and a PCM. In the experiment cooking was done by the PCM after charging. The result showed that the cooker can boil five kilogram of water in 38 min and fry meat in 20 min.

2.3.3 Stearic Acid as a PCM For Solar Cooking Application

Stearic acid ($\text{C}_{17}\text{H}_{35}\text{CO}_2\text{H}$) is a saturated organic fatty acid which can store thermal energy while it changes its phase from solid to liquid and release its heat in the reversed thermal process. Fatty acids like stearic acid have higher latent heat of fusion and melting point compared to paraffin wax. Literature showed that stearic acid can be utilized as a latent heat energy storage for solar cooking applications. Nagaraj *et al.*, (2016) developed a solar cooker for late evening cooking using stearic acid and acetanilide as PCM material. In the study solar cooker parts were developed in-house and connected to water heating system compounded with evacuated tubes solar collector and storage tank. The circumference of cooker unit was incorporated with spiral stainless-steel heat exchanger and annulus area of the pot is filled with PCM material. PCM releases heat at late evening and the cooker was able to effectively cook up to 7:30 PM. The experimental results indicated that cooker and collector efficiency was 30% and 60–65% respectively. Overall, experimental results showed satisfactory performance on the developed cooker. Buddhi and Sahoo, (1997) designed a box-type solar cooker with latent heat storage for the composite climatic conditions of India. The experimental results demonstrate the feasibility of using a phase change material as the storage medium in solar cookers. It also provides a nearly constant plate temperature in the late evening. The experimental results have also been compared with those of a conventional solar cooker. Stearic acid (melting point

is 55.1 °C and latent heat of fusion is 160 kJ/kg) was used as a PCM. The result showed that coking was possible in off-sunshine periods using stearic acid as a phase change material.

2.4 Factors Affecting the Performance of Flat Plate Collector

As the main heat input for solar based cooker is obtained from the solar collector, the performance of such systems mainly depends on the performance of solar collector which is employed in the solar cooking system. Thus, the factors which affect the performance of flat plate collector are described below;

2.4.1 Inclination Angle of Collector

Inclination or tilt angle that the solar collector makes with the horizontal is one of the important parameters that affect the performance of a solar collector. This is due to the fact that the variation in tilt angle affects the amount of solar radiation reaching the collector surface. Collector installed at an optimum tilt angle and with proper orientation resulted in maximum radiation on the surface of collector. Finding the optimum tilt angle for a solar collector is also important to avoid frequent solar tracking. Many investigations have been conducted to determine, or at least estimate, the best tilt angle for solar collectors. Tang and Tong, (2004) proposed a simple mathematical procedure for estimating the optimal tilt angle of a fixed collector based on monthly global and diffuse radiation. They made estimation of the optimal tilt angle of south-facing collectors used for the entire year (January to December) in several cities in China. Hottel, (1954) recommended that the optimum tilt angle should be the latitude of the location plus 20°. Yellott, (1973) in the other hand reported optimum tilt to be latitude plus 30°. Kern and Harris, (1975) showed it is optimum at latitude angle plus 10°. From review it is clear that there is no definite value of the tilt angle that can be recommended as it is dependent on the location where experiment is performed.

2.4.2 Wind Velocity

In the design of flat plate collector, the convective heat loss is influenced by wind speed. With increase in wind speed over the surface of the collector results in an increase in heat loss thereby decrease in the thermal performance of the solar collector. Convective heat loss in the solar collector system accounts for (20-30) % this is due to the take up of heat by the wind blowing around (Fabio, 2008). Convective losses in the flat-plate system are strong functions of configuration and geometrical parameters of the collector. This showed that minimizing the effect of wind speed by constructing a better configuration and shape of flat plate collector can enhance the performance of whole system.

2.4.3 Solar Irradiation

The total solar radiation value consists of: Direct (beam), diffuse and reflected radiation Danuta and Slawomir, (2017). Direct (beam) radiation is a short-lived radiation in which rays are spread in a straight line from the Sun to the active surface of the collector. Therefore, the wider is the angle of sunrays falling on the collector, the larger amount of energy is absorbed. It is obvious that the amount of solar irradiation incident on a solar collector affects the performance of such systems. For instance, He *et al.*, (2016) studied the effect of solar radiation on thermal efficiency of a flat plate collector. The result showed that as the solar radiation increased from 200 – 1000 W/m², the outlet water temperature from the collector increased from 32.2°C to 42.1°C resulting in an increase in the thermal efficiency from 38.1% to 64.8%.

2.4.4 Absorber Plate Coating

The collector absorber plate should have high thermal conductivity, adequate tensile and comprehensive strength, and good corrosion resistance. Those characteristics can be achieved by coating a selective material. Different researchers try to improve the thermal performance of a flat plate collector by coating different selective materials on the absorber plate. El-Sebaili and Al-Snani, (2010) investigated the effect of using absorber plates coated with various selective coating materials to improve the performance of a flat plate solar collector employed for a heater. The best performance was achieved using nickel–tin as a selective coating material with a daily average of the instantaneous efficiency of 0.46. It was also inferred that the annual average of efficiency with a nickel–tin selectively coated absorber is higher than that with a black painted absorber by 29.23%. Oliva *et al.*, (2013) proposed a low-cost material to be used as an excellent absorber for solar collectors, to increase its thermal efficiency by the high capacity to absorb solar radiation. The material, known as “smoke black” (soot) was obtained by the incomplete combustion of organic materials, such as the oxygen-acetylene, paraffin, or candles. A comparative analysis between the optical properties (reflectance, absorbance, and emissivity) were measured on three covered copper surfaces (without paint, with a commercial matte black paint, and with smoke black) shows amazing optical results for the smoke black. As Fig.2.5 indicates reflectance values of the smoke black applied over copper surfaces was improved 56 times the values obtained from commercial black paints.

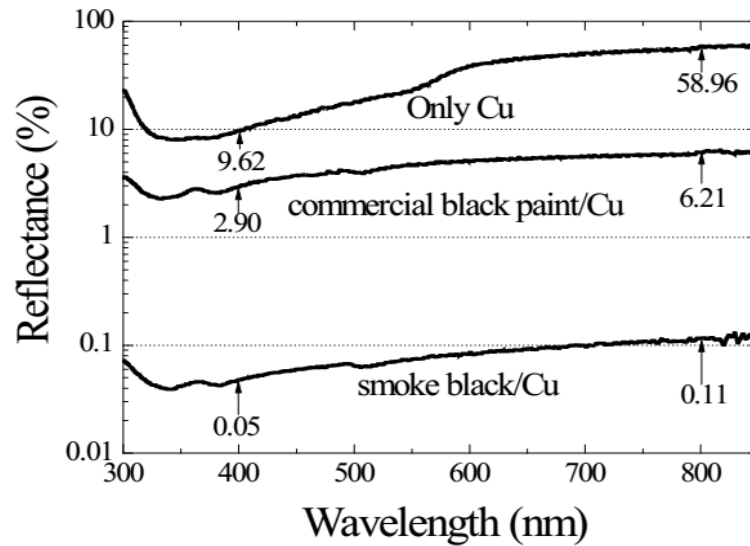


Fig.2.5 Reflectance of three different copper absorber surfaces [Oliva *et al.*, 2013].

2.4.5 Side Reflectors (Booster Mirror)

The addition of side mirrors increases the amount of radiation reaching the collector and hence increases the overall performance of the solar collector which permits high temperature operations and allows solar cooking to be possible. Many studies have been done to investigate the effect of booster side reflectors on solar thermal collectors. Zoran and Ljiljana, (2015) analyzed the impact of flat plate reflectors (bottom, top, left and right reflectors) made of Al, on total solar radiation on a solar collector during a day time over a whole year. It was found that intensity of the solar radiation on the collector increases for about 80% in the summer period (June-September) by using optimally inclined reflectors, in comparison to the collector without reflectors. Mc-DANIEL *et al.*, (1975) analyzed the amount of direct light gathered by a combination of reflector plus flat-plate collector. The result revealed that for typical winter operating conditions the enhancement in light gathering power for direct solar radiation is about a factor of 1.4-1.7. Tabet *et al.*, (2017) also performed a theoretical and experimental study on flat plate solar collector with reflectors. The result showed that an increase of 23% for solar radiation incident on the collector surface with the addition of the planar reflectors in the day of May, this increase causes an improvement of the performance of the collector and the fluid temperature increases with an average of 5%.

2.4.6 Mass Flow Rate of the HTF

Mass flow rate is also another factor for the performance of flat plate collectors. Studies showed that optimum mass flow rate is important for effective heat collection from the solar collector. Mohammad *et al.*, (2018) investigate the effect of flow rate and inlet temperature on thermal efficiency of flat plate collectors. The collector was examined for high- and low-level flow rates and the result showed

that maximum temperature increase of 62 K in the inlet temperature was achieved at a flow rate of 5×10^{-4} kg/s inside the riser pipe. Bokola, (2006) designed and experimentally analyzed the flow inside a natural circulation flat plate collector. The result showed that the performance of water heating system depends very much on both the flow rate through the collector and the incident solar radiation. In a typical day, the analysis of the system showed that collector efficiency is high especially around mid-day when solar collector receives higher energy and exhibited optimum flow rate of 0.1 Kg s.m² with a maximum collector efficiency of 68.5% and maximum water temperature obtained was 83.5°C with an ambient temperature of 34.5°C. Zhang *et al.*, (2016) investigated the performance of flat plate collectors experimentally using the results to validate a mathematical method and to calculate thermal performance of the collectors. The authors concluded that thermal efficiency, outlet water temperature and heat transfer effectiveness were affected significantly by the flow rate.

2.4.7 Insulation

It is well known from the literature that insulation is one of most crucial key points for a performance of solar collectors which are utilized for solar cooking application to provide an efficient cooking (Erdem and Pinar, 2013). For instance, Insulation in a flat plate collector should not be limited to the walls of the frame box and absorber tray since a remarkable amount of heat loss may occurs through the glazing (Saxena *et al.*, 2011).

2.4.8 Gap Between Absorber Plate and Glass

In the design of flat plate collector, spacing between the glass and the absorber plate has an effect on the amount of the convective heat losses and hence the performance. Sharma *et al.* (2014) investigated the optimum spacing (air gap) of glazing covers; of single, double and triple glazed black painted flat plate collector and optimized the combination of number of glazing and spacing of glazing covers (air gap) with minimum top heat loss. The result showed that out of the three experiments with air gap of 5mm, 10mm and 20mm of double-glazed flat plate collector, a collector set up with 10 mm air gap is the optimal combination for a black painted absorber plate.

2.5 Summary of Literature Review

In literature survey, both direct and indirect type of solar cookers has been reviewed. Most of the literatures associated with direct solar cooking showed that the cooking pot is assembled directly to the solar collector which makes actual cooking process difficult for cooker to manage. Even though direct solar cookers are easy to construct, maintain and less expensive, such cookers with bare pots suffer from convective and irradiative losses to the point of being unusable at higher wind speeds exceeding 10km/h. In addition, the disadvantages associated with the use of direct type of solar

cookers are the need of frequent solar tracking and outdoor cooking which causes repeatedly exposure of the cooker to the sun that in turn may cause skin burning and eye damage. Comparatively, literature showed that indirect solar cookers have an advantage of being able to cook indoor without suffering a problem associated with direct solar cookers.

Reviews of literatures indicated that considerable efforts have been invested in research and development of direct type solar cookers with and without storage. However, limited research works were reported on indirect type of solar cookers, particularly, flat plate type solar cookers with a thermal energy storage system. Furthermore, those limited works in literature focuses only on energy efficiency of the systems even if exergy efficiency is the true performance measure of solar-thermal system which indicate the quality of energy converted into useful form of energy. Thus, the present work aims to improve the performance indirect solar cooker by improving the performance of flat plate collectors using radiation booster side reflectors and integrating with PCM based thermal energy storage system for off-sunshine and late hour operation of the cooker.

CHAPTER THREE

MATERIALS AND METHODS

3.1 Introduction

The present study applies numerical simulations, theoretical and experimental analysis as a main approach which is used as a methodology. Theoretical approach uses governing equation for the design, sizing, selecting and formulation of the solar cooking system components. The thermal performance of modified flat plate collector (FPC) and latent heat storage (LHS) system is evaluated both numerically and experimentally. The numerical result is validated with experimental result and similar previous works. In these section research design techniques, process model, different data sources and collection methods, material and equipment used and experimental data presentation and analysis methods which will be employed for experimental investigation and performance evaluation of the present indirect solar cooking system is presented.

3.2 Methodology and Research Design

The design techniques of the research is mixed analytical and experimental research design where both qualitative and quantitative research techniques are employed. The solar cooking system is evaluated based on existing hypothesis and experiential testing of the hypothesis. The general process layout of the methodology followed in the research is presented in Fig.3.1.

3.3 Data Sources and Collection Method

The data required for the whole work is obtained from both primary and secondary sources of data.

Primary data; Data such as, ambient temperature of air, the temperature of tap and ground water is obtained by direct measuring.

Secondary data; such data are obtained from different sources in the internet such as books, documents from website, literatures local and international journal articles and data from deep literature review. Metrological and solar radiation data for the site is obtained from Adama branch of Ethiopian National Metrological Service Agency (ENMSA).

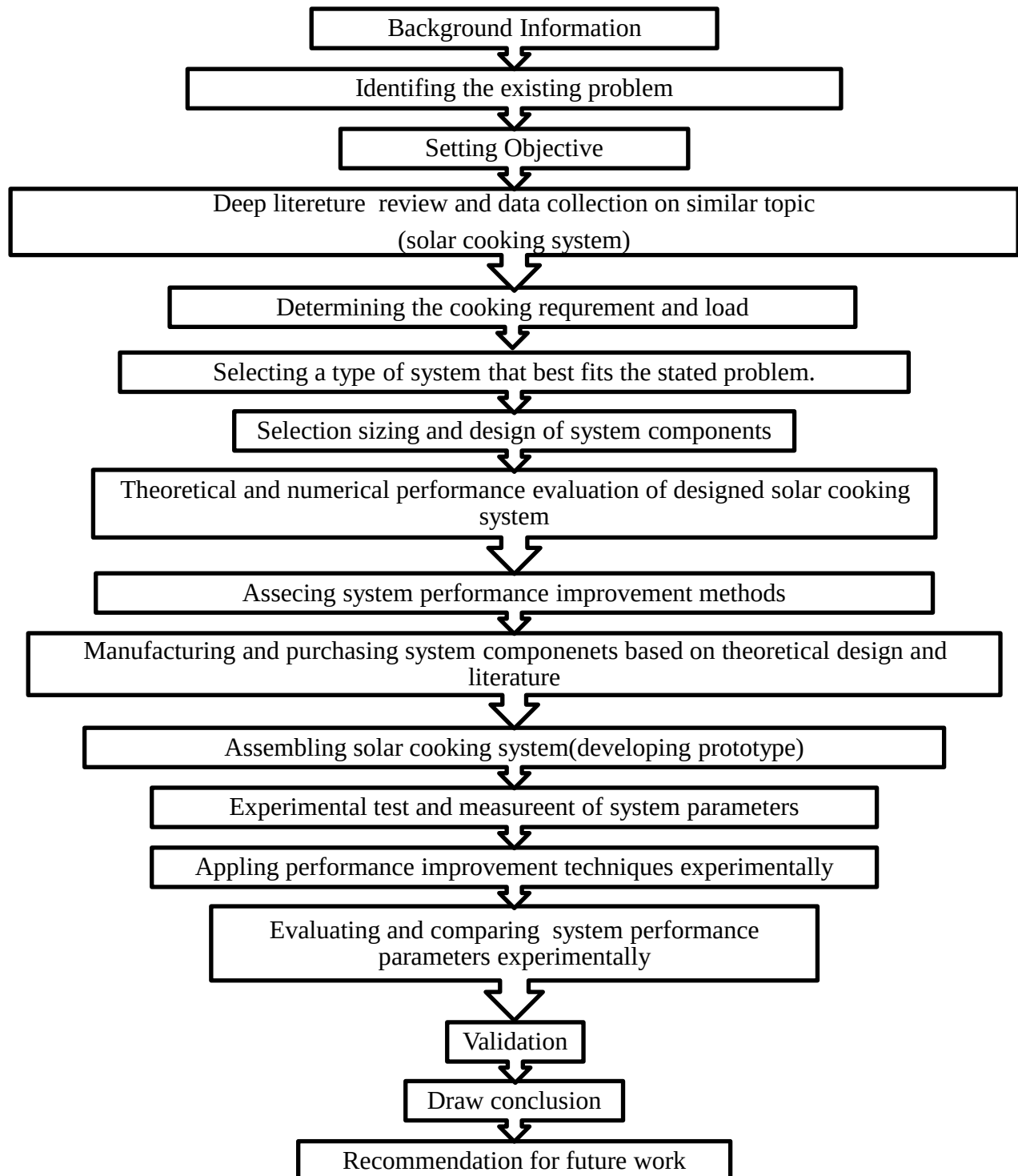


Fig.3.1 General process layout of the research

3.4 Materials and Equipment's

Materials, measuring devices and system components required for developing the experimental prototype and measuring the system parameters are obtained by purchasing from local or domestic market of Adama city (Ethiopia) and abroad.

3.5 Data Analysis and Interpretation Tools

In this thesis Excel (spreadsheet) is a basic software that is used as a data storage, estimation, analysis and interpretation tool for the solar radiation data and timely varying measured parameters during experimental test.

3.6 Model Development and Simulation Tools

SOLIDWORKS 2016 is used as component design and development software. whereas ANSYS Fluent, and COMSOL are utilized as, numerical analysis and simulation software's for the performance parameters of the developed system components.

CHAPTER FOUR

SOLAR POTENTIAL ESTIMATION

4.1 Estimation of Solar Energy Intensity for the Site

Unavailability of devices for recording the hourly values of solar and climatic data made it necessary to use the estimated values of global, diffuse and beam radiation of the specific site. Estimation of solar energy intensity is required to calculate the total amount of solar energy incident on solar collectors. The rate at which solar radiation is striking a surface per unit area of the surface is called as the total solar irradiation on the surface. Sun–earth angles are essential for estimating solar intensity throughout year for any surface at any place with a desired inclination and orientation. Those angles generally depend on location on earth, day of the year and time of the day.

4.1.1 Basic Sun - Earth Angles

The geometric relationships between a plane of any particular orientation relative to the earth at any time and the incoming solar radiation, that is, the position of the sun relative to that plane, can be described in terms of several angles.

- i. **Latitude angle (ϕ)** is the angular location of any place on earth which is north or south of the equator, north positive; $-90^\circ \leq \phi \leq 90^\circ$.
- ii. **Declination angle (δ)** is the angular position of the sun at solar noon with respect to the plane of the equator, north positive; $-23.45^\circ \leq \delta \leq 23.45^\circ$. The declination δ can be found from the approximate equation of Cooper (1969);

$$\delta = 23.45 \sin\left(\frac{360(284+n)}{365}\right) \quad 4.1$$

Where, n is day of the year (starting from January 1st).

- iii. **Slope or tilt angle (β)** is the angle between the inclined plane surface under consideration and the horizontal; $-180^\circ \leq \beta \leq 180^\circ$.
- iv. **Hour angle (ω)** is the angular displacement of the sun east or west of the local meridian due to the rotation of the earth on its axis at 15° per hour; positive in the afternoon and negative in the forenoon.

$$\omega = [\text{solar time} - 12:00] \text{ (in hours)} \times 15^\circ \quad 4.2$$

- v. **Zenith angle (θ_z)** is the angle between the vertical and the line to the sun, that is, the angle of incidence of beam radiation on a horizontal surface.

$$\cos \theta_z = \cos \delta \cos \phi \cos \omega + \sin \phi \sin \delta \quad 4.3$$

- vi. **Surface azimuth angle (γ)** is the deviation of the projection on a horizontal plane of the normal to the surface from local meridian.

$$\gamma = 0 \text{ for surfaces facing south.}$$

- vii. **Angle of incidence (θ)** is the angle between the sun's ray's incident on the plane surface and normal to that surface.

$$\cos \theta = (\cos \phi \cos \beta + \sin \phi \sin \beta \cos \gamma) \cos \delta \cos \omega + \cos \delta \sin \omega \sin \beta \sin \gamma + \sin \delta (\cos \beta \sin \phi + \cos \phi \sin \beta \cos \gamma) \quad 4.4$$

- viii. **Solar altitude angle (α)** the angle between the horizontal and the line to the sun, that is, the complement of the zenith angle.

4.1.2 Extraterrestrial Radiation

The extraterrestrial radiation on a surface normal to the sun's rays kept at the actual sun-earth distance at a specific day and time which is modified to take into account the varying distance between the sun and the earth is given by (Duffie and Beckman, 2013):

$$G_{\text{ext}} = G_{sc} [1.0 + 0.033 \cos(\frac{360(n)}{365})] \quad 4.5$$

Where, n is day of the year (counted from January 1)

$$G_{sc} \text{ is solar constant } = 1367 \text{ W/m}^2$$

The extraterrestrial solar radiation that would fall on a horizontal surface at the location is given by:

$$G_o = G_{\text{ext}} \cos \theta_z \quad 4.6$$

$$G_o = G_{sc} [1.0 + 0.033 \cos(\frac{360(n)}{365})] [\cos \delta \cos \phi \cos \omega + \sin \phi \sin \delta] \quad 4.7$$

The daily extraterrestrial solar radiation on a horizontal surface (H_o in $\text{Kw/m}^2/\text{day}$) can be obtained by integrating Eqn.4.7) from sunrise to sunset and expressed as;

$$H_o = \frac{24}{1000 \times \pi} G_{sc} [1.0 + 0.033 \cos(\frac{360(n)}{365})] [\cos \delta \cos \phi \sin \omega_{sr} + \frac{\pi \times \omega_{sr}}{180} \sin \phi \sin \delta] \quad 4.8$$

$$\text{Where, sunset hour angle } \omega_{sr} = \cos^{-1}(\tan \phi \tan \delta) \quad 4.9$$

Fig.4.1 illustrates the monthly average daily extraterrestrial radiation on horizontal surface of Adama city for Adama city.

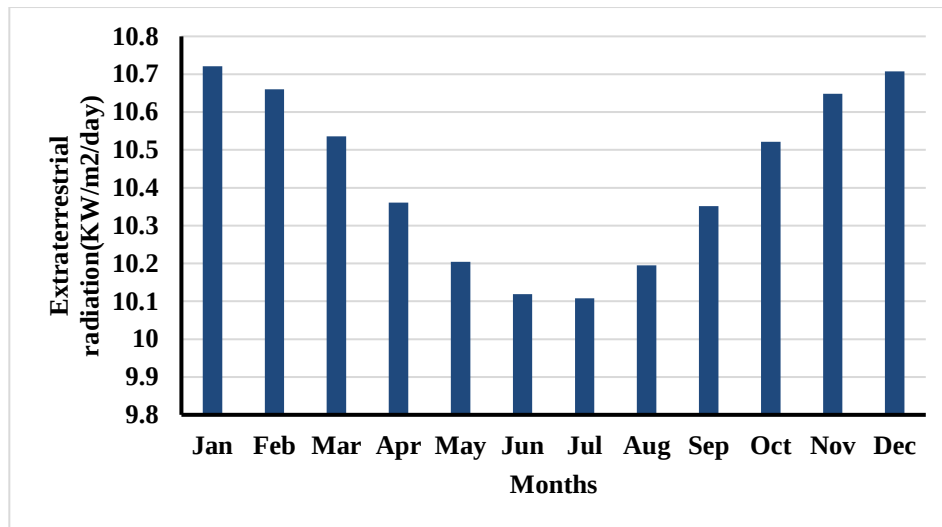


Fig 4.1 Monthly average daily extraterrestrial radiation

4.1.3 Estimation of Monthly Average Daily Global Radiations on a Horizontal Surface

To estimate the monthly average daily global radiations on a horizontal surface of selected site, it is crucial to get the data of hourly sunshine duration of the day. Thus, the daily sunshine duration (sunshine hours) data of the site (Adama city with latitude 8.54° north of the equator, longitude $=39.28^{\circ}$ E; and altitude = 1648 m above sea level) for five years was obtained from Adama branch of Ethiopian National Metrological Service Agency (ENMSA). The five-year daily sunshine hour data was then monthly averaged as shown in Fig.4.2 for the ease of estimation of monthly average daily global radiations.

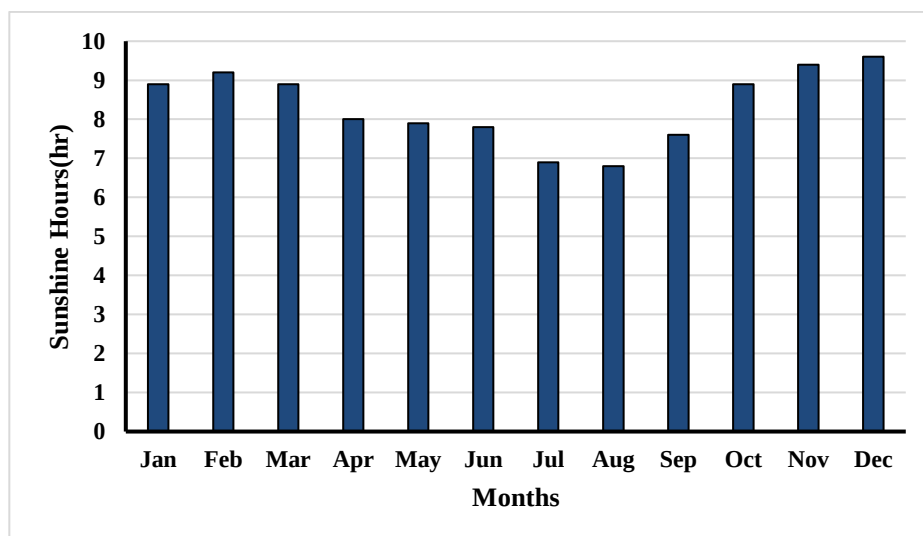


Fig 4.2 Monthly average daily sunshine hours of Adama city

The first empirical correlation for the estimation of global solar radiation was proposed by Angstrom. The correlation relates the radiation with the number of sunshine hours in a day using some empirical

constants obtained by regression analysis. The well-known Angstrom relation for estimating the monthly average daily solar radiation on a horizontal surface is given as (Tiwari et al, 2016);

$$\frac{H_G}{H_o} = a_1 + b_1 \left(\frac{\bar{n}}{\bar{N}} \right) \quad 4.10$$

Where, H_G is monthly average daily global solar radiation ($Kw/m^2/day$) in the terrestrial region,

$\bar{n}$ is the daily hours of bright sunshine,

$\bar{N}$ is the maximum possible daily hours of bright sunshine,

a_1 and b_1 are regression coefficients given by:

$$a_1 = -0.11 + 0.235 \cos \phi + 0.32 \left(\frac{\bar{n}}{\bar{N}} \right) \quad 4.11$$

$$b_1 = 1.449 - 0.533 \cos \phi - 0.694 \left(\frac{\bar{n}}{\bar{N}} \right) \quad 4.12$$

Therefore, the monthly average daily global solar radiation on horizontal surface of Adama city for five years was estimated using Eqn.4.10 as depicted in Fig.4.3.

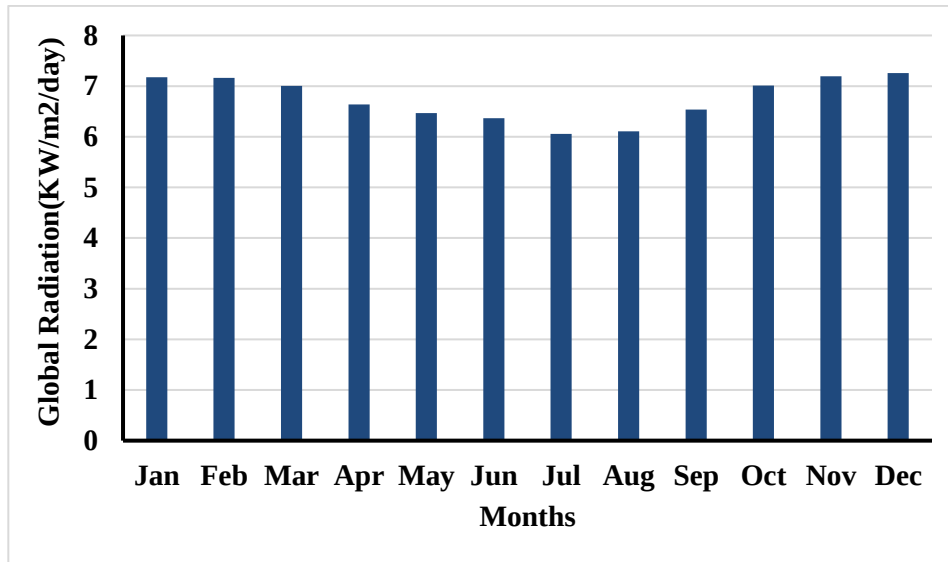


Fig.4.3 Monthly average daily global radiation on a horizontal surface

4.1.4 Estimation of Monthly Average Daily Diffuse and Beam Radiations

The monthly average daily diffuse radiation on a horizontal surface (H_d in $Kw/m^2/day$) can be determined from the monthly average daily global radiation on a horizontal surface and the number of bright sunshine hours and is given by the correlation (Tiwari et al., 2016):

$$\frac{H_d}{H_G} = 0.931 - 0.814 \left(\frac{\bar{n}}{\bar{N}} \right) \quad 4.13$$

The monthly average daily beam radiation (H_b in $Kw/m^2/day$) on a horizontal surface is the difference between the monthly average daily global and diffuse radiations.

$$H_b = H_G - H_d \quad 4.14$$

The monthly average daily global, diffuse and beam radiation on a horizontal surface of Adama city averaged for five years of climatic condition is indicated in Fig.4.4.

4.1.5 Estimation of Monthly Average Hourly Global Radiation

The monthly average hourly global radiation on a horizontal surface (I_G in Kw/m^2) is determined from the monthly average daily global radiation using the correlation (Chandraprabu *et al.*, 2019);

$$\frac{I_G}{H_G} = \frac{\pi}{24} (a_2 + b_2 \cos \omega) \left(\frac{\cos \omega - \cos \omega_{sr}}{\sin \omega_{sr} - \frac{\pi}{180} \omega_{sr} \cos \omega_{sr}} \right) \quad 4.15$$

$$\text{Where, } a_2 = 0.409 + 0.5016 \sin(\omega_{sr} - 60)$$

$$b_2 = 0.6609 - 0.4767 \sin(\omega_{sr} - 60)$$

Hourly variation of global radiation at each representative day of month for each of 24 hours in an hour interval is presented in Fig.4.5. As the graph shows global radiation intensity is detected from 6AM up to 18PM of the day (maximum at solar noon and minimum at sunset and sunrise).

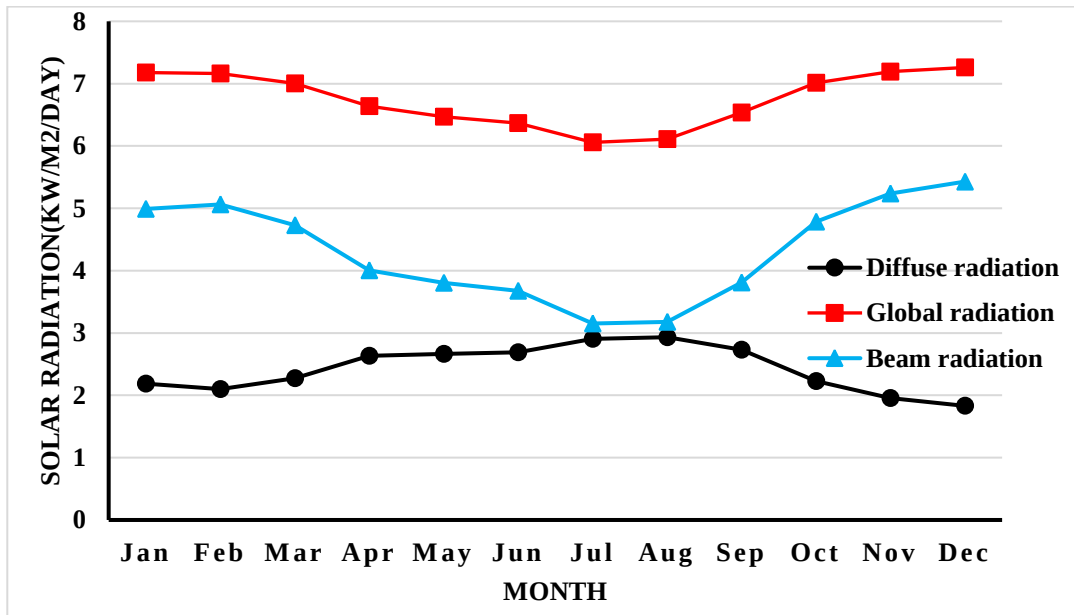


Fig.4.4 Monthly average global, diffuse and beam radiation on a horizontal surface

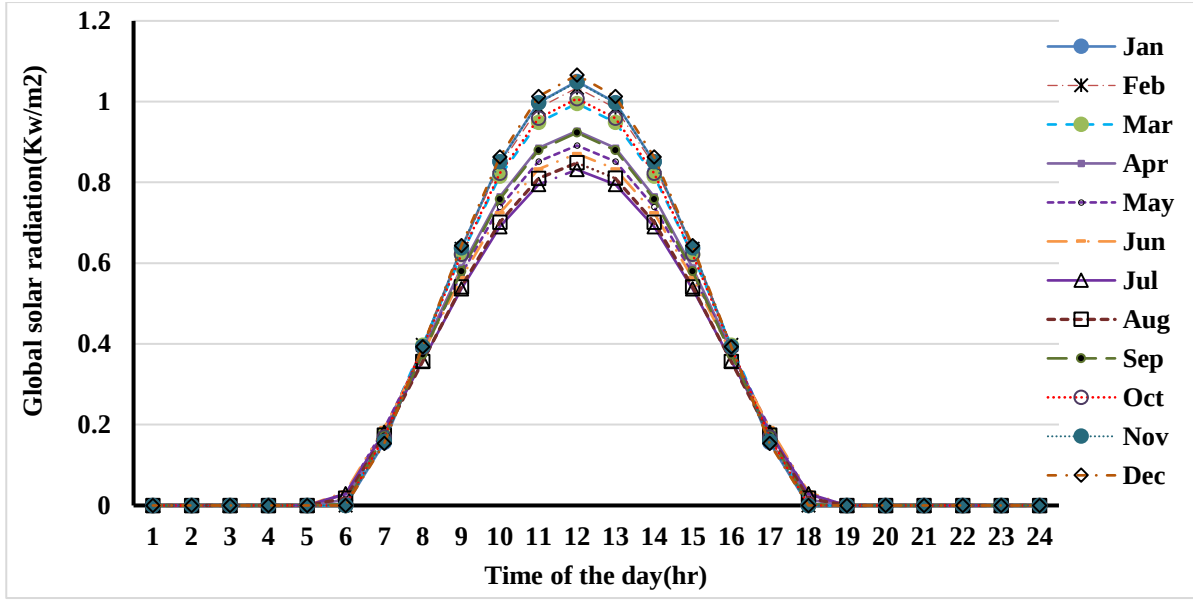


Fig.4.5 Variation of monthly average hourly global solar radiation on a horizontal surface

4.1.6 Estimation of Monthly Average Hourly Diffuse and Beam Radiations

The monthly average hourly diffuse radiation on a horizontal surface (I_d in Kw/m^2) can be determined from the monthly average daily diffuse radiation on a horizontal surface (Chandraprabu *et al.*, 2019).

$$\frac{I_d}{H_d} = \frac{\pi}{24} \left(\frac{\cos \omega - \cos \omega_{sr}}{\sin \omega_{sr} - \frac{\pi}{180} \omega_{sr} \cos \omega_{sr}} \right) \quad 4.16$$

The monthly average hourly beam radiation on a horizontal surface (I_b in Kw/m^2) is the difference between the monthly average hourly global and diffuse radiations.

$$I_b = I_G - I_d \quad 4.17$$

Hourly variation of diffuse and beam radiation at each representative day of month for each of 24 hours in an hour interval is presented in Fig.4.6 and 4.7 respectively. As the graph shows solar radiation intensity is detected from 6AM up to 18PM of the day (maximum at solar noon and minimum at sunset and sunrise).

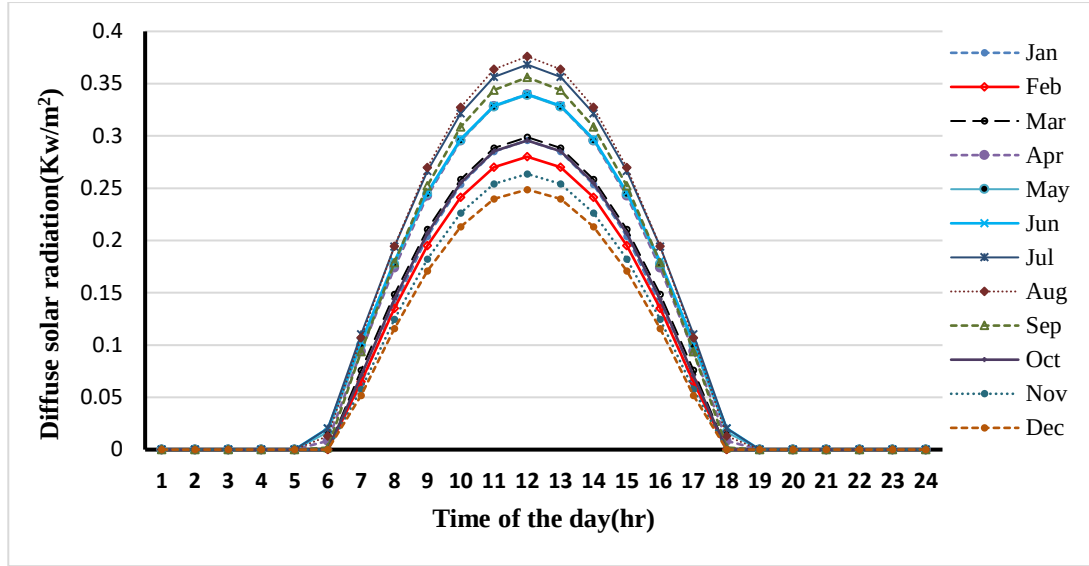


Fig 4.6 Variation of monthly average hourly diffuse solar radiation on a horizontal surface

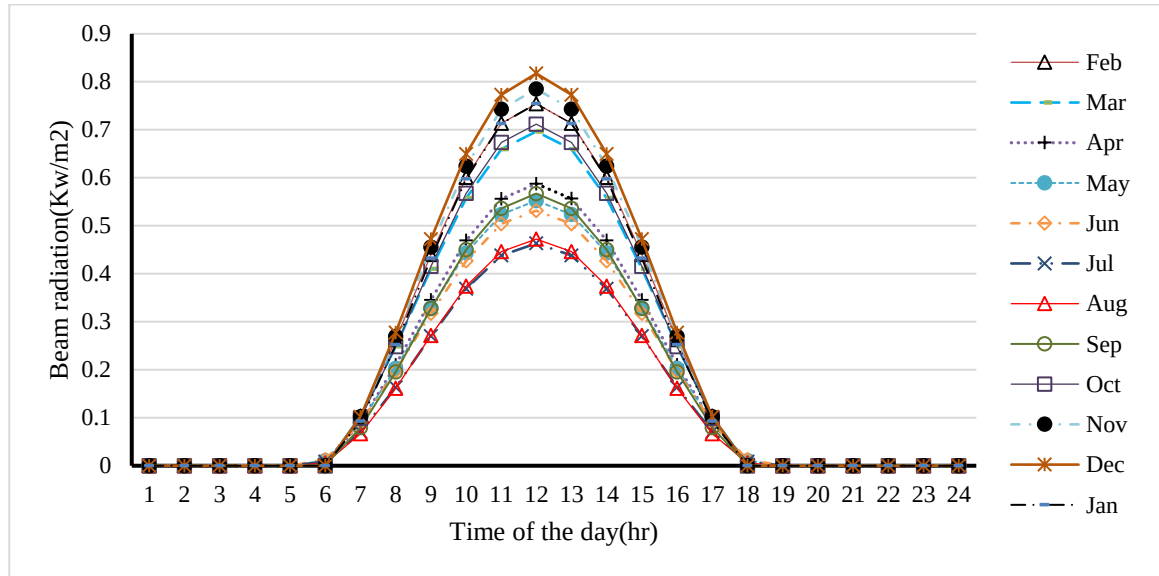


Fig 4.7 Variation of monthly average hourly beam radiation on a horizontal surface

4.1.7 Estimation of Total Incident Solar Radiation on an Inclined Surface

The incident solar radiation on an inclined surface (I_T in Kw/m^2) is the sum of beam, isotropic diffuse and reflected solar radiation from various surfaces (from a horizontal surface and surfaces surrounding it). The total diffuse radiation on the flat plate solar collector I_{d-col} is the sum of the sky-diffuse radiation I_{d-sky} and the reflected radiation from the ground I_{d-gr} .

$$I_{d-col} = I_{d-sky} + I_{d-gr} \quad 4.18$$

For a collector tilted at slope β from the horizontal, there are two factors called view factor to the sky (F_s) and view factor to the ground (F_g) which are a function of slope of the collector and expressed by the following geometrical equations;

$$F_s = \frac{(1+\cos\beta)}{2} \quad \text{and} \quad F_g = \frac{(1-\cos\beta)}{2}$$

Eqn.4.18 can be written in terms of view factors as;

$$\begin{aligned} I_{d-col} &= I_d F_s + \rho_g I_G F_g \\ &= I_d \frac{(1+\cos\beta)}{2} + \rho_g I_G \frac{(1-\cos\beta)}{2} \end{aligned} \quad 4.19$$

Where: I_G is global solar radiation on horizontal surface

I_d diffuse solar radiations on horizontal surface

ρ_g is reflectance of the ground

Therefore, the total incident solar radiation on a surface tilted at an angle β can be expressed in terms of the global, beam, diffuse and the total reflected radiation on the tilted surface (Duffie and Beckman, 2013).

$$I_T = I_b R_b + I_d \left(\frac{1+\cos\beta}{2} \right) + I_G \rho_g \left(\frac{1-\cos\beta}{2} \right) \quad 4.20$$

Where, ρ_g is reflectivity of the ordinary ground and its value is 0.2 for all months except December and March ($\rho_g = 0.4$) and January and February ($\rho_g = 0.7$)

R_b is the beam radiation factor which can be defined as the ratio of the incident beam radiation on a tilted surface to that on a horizontal surface.

$$R_b = \frac{\cos\theta}{\cos\theta_z} \quad 4.21$$

For a solar collector facing south ($\gamma=0$) beam radiation factor can be expressed as (Duffie and Beckman, 2013):

$$R_b = \frac{\cos(\beta-\phi)\cos\delta\cos\omega + \sin(\beta-\phi)\sin\delta}{\cos\beta\cos\delta\cos\omega + \sin\beta\sin\delta} \quad 4.22$$

The amount of total solar radiation incident on a flat plate collector is a function of collector tilt angle. Thus, optimum tilt angle is required to get optimum incident solar radiation on the collector. It is generally known that for a collector installed in the northern hemisphere, the optimum collector orientation is south facing ($\gamma = 0$) and the optimum tilt depends upon the latitude of the site and the day of the year. Ahmed and Tiwari, (2009) suggested optimum tilt to be latitude of the location + (10 to 15°). Thus, for a collector installed in Adama city the optimum tilt angle can be estimated be 25°.

The hourly variation of total incident solar radiation on the collector tilted at 25° is depicted in Fig.4.8. The percentage of increment in the incident radiation due to tilting can be calculated based on maximum value of solar radiation (for month February).

$$\begin{aligned} \text{\% of increase in the incident solar radiation} &= \frac{I_{T_{max}} - I_{G_{max}}}{I_{T_{max}}} \times 100 = \frac{1.174727659 - 1.066070625}{1.174727659} \times 100\% \\ &= 9.25\% \end{aligned}$$

Graphical comparison of hourly variation of total incident solar radiation on a tilted collector at 25° and global solar radiation on horizontal surface is shown in Fig.4.9.

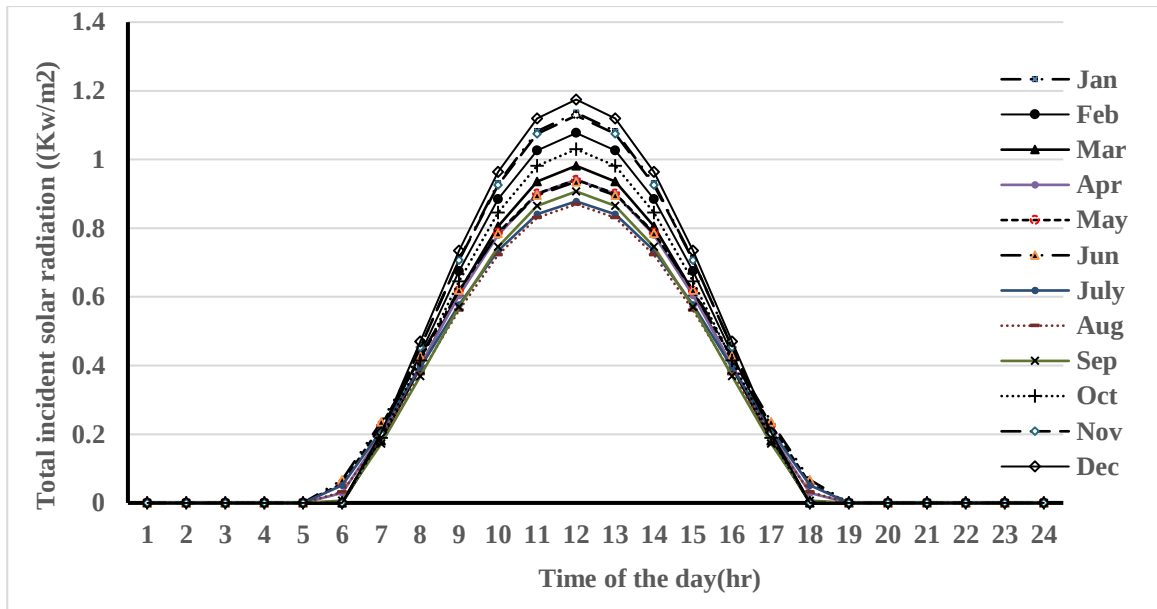


Fig 4.8 Hourly variation of total incident solar radiation on the collector tilted at 25° .

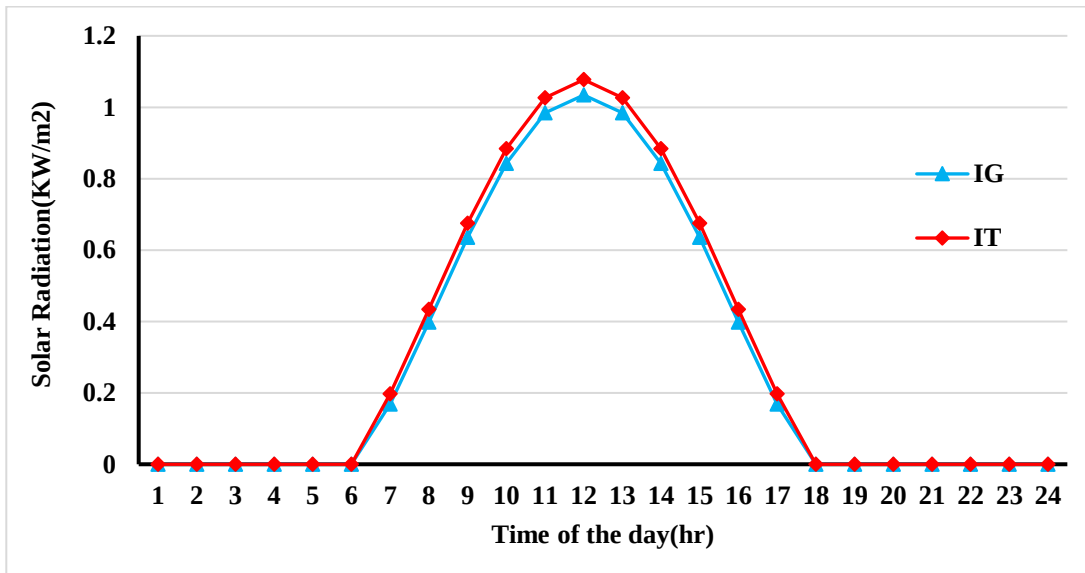


Fig 4. 9 Comparison of total incident solar radiation on a tilted collector at 25° and global solar radiation on horizontal surface.

4.1.8 Estimation of Hourly Variation of Ambient Temperature

In the analysis of thermal system, it is important to know the hourly variation of dry bulb temperature but the data available in hand is daily mean maximum and daily mean minimum temperatures of the site. Hence estimation of hourly dry-bulb temperature from the daily temperature data will give as the approximate value. Fig.4.10 shows monthly average daily maximum and minimum temperature and monthly average daily wind speed of Adama city averaged from five-year temperature data. Hourly variation of atmospheric temperature was estimated from the daily maximum and minimum temperatures. A sin-exponential model proposed by Parton and Logan (1981) was used and it describes the diurnal cycle by a sine function during daytime and a decreasing exponential function during nighttime. The major assumption of the model was the maximum temperature will occur sometime during the daylight hours before sunset and the minimum temperature will occur during the early morning hours. The mathematical description for the daytime and nighttime, respectively were expressed as (Parton and Logan, 1981):

$$T_i = (T_{max} - T_{min}) \sin\left(\frac{\pi m}{Y + 2a}\right) + T_{min} \quad 4.23$$

$$T_i = (T_{sunset} - T_{min}) \exp\left(-\frac{bn}{Z}\right) + T_{min} \quad 4.24$$

Where, T_i is temperature at the i^{th} hour,

Y and Z are the length of the day and the night in hours, a is the lag coefficient for maximum temperature = 1.8h and b is the nighttime temperature coefficient = 2.2.

m is the number of hours after the minimum temperature occurs until sunset and n is the number of hours after sunset until the time of the minimum temperature. Fig.4.11 shows the Hourly variation of temperature for Adama city.

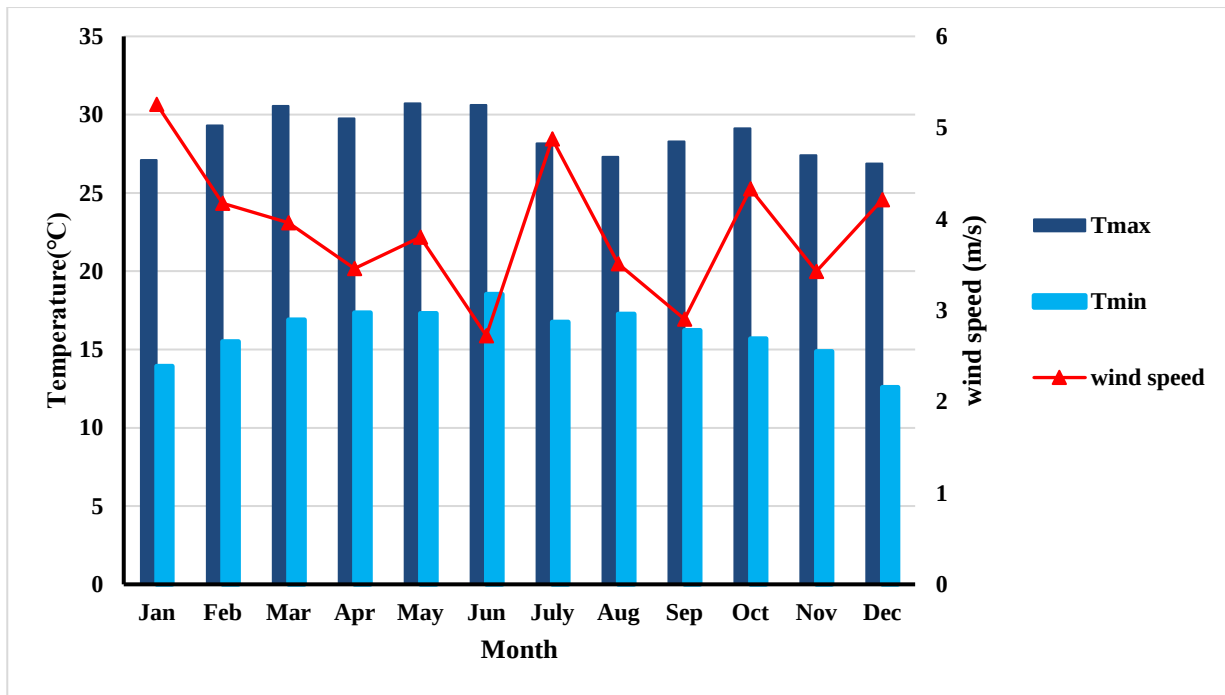


Fig.4.10 Monthly average daily maximum and minimum temperature and monthly average daily wind speed of Adama city

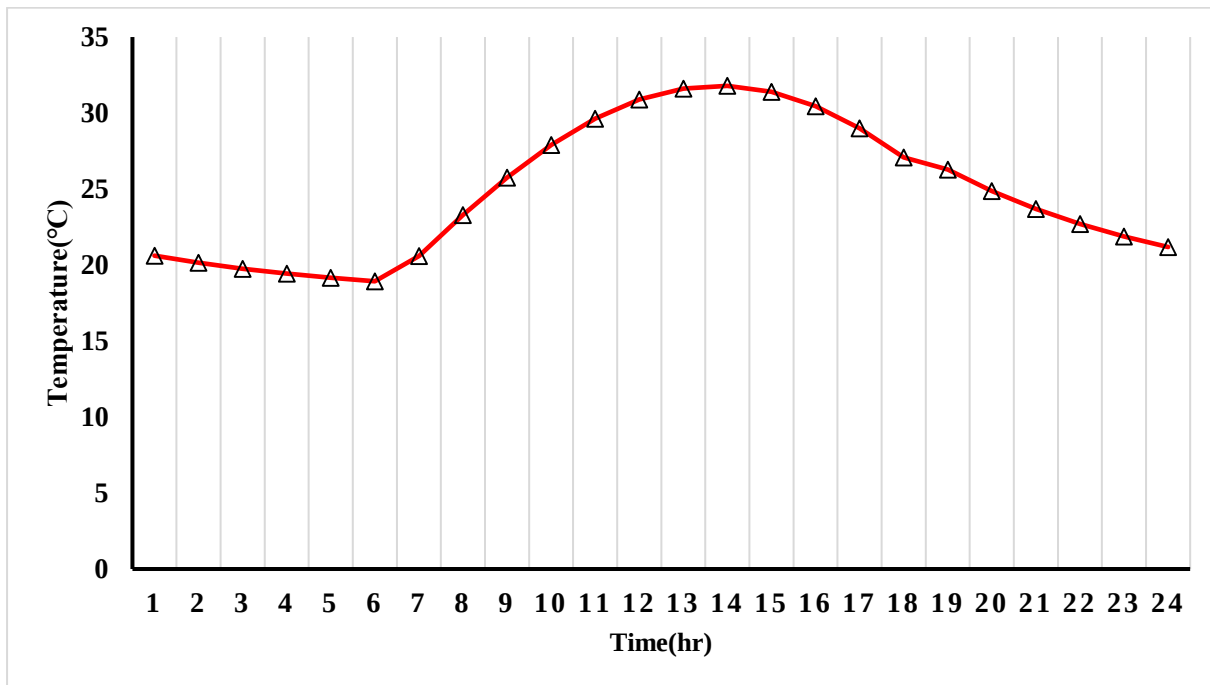


Fig.4.11 Hourly variation of daily temperature for Adama city

CHAPTER FIVE

THEORETICAL AND NUMERICAL FORMULATION OF MODIFIED FPC AND LHS UNIT

5.1 Introduction

This chapter presents the theoretical and numerical analysis of both modified flat plate collector and thermal storage unit that can be utilized for indirect type solar cooking system. The design, sizing, selection and specifications for both the modified FPC parameters and the thermal energy storage unit are presented in detail. Theoretical formulations and governing equations used for energy and exergy analysis of the FPC and LHS unit are discussed and reported. Selection of PCM and HTF and sizing of LHS unit is also addressed. In addition, the design of cooking chamber, performance parameters of the cooker, pump requirement and selection for the closed loop system is discussed. Finally, numerical analysis for FPC using ANSYS CFD (fluent) and for LHS using COMSOL software is assessed.

5.2 Design of Modified FPC

Designing of solar collector includes estimating the amount of solar radiation incident on the collector, sizing of components, material selection and specification of the system.

5.2.1 Estimation of Total Solar Radiation Absorbed by Modified FPC

The modified FPC tilted at an angle β consists of four reflectors on each side of FPC which are used to enhance the incident solar radiation on the collector. Fig. 5.1 shows the 3D Assembly of modified FPC drawn using SOLIDWORKS 2019.

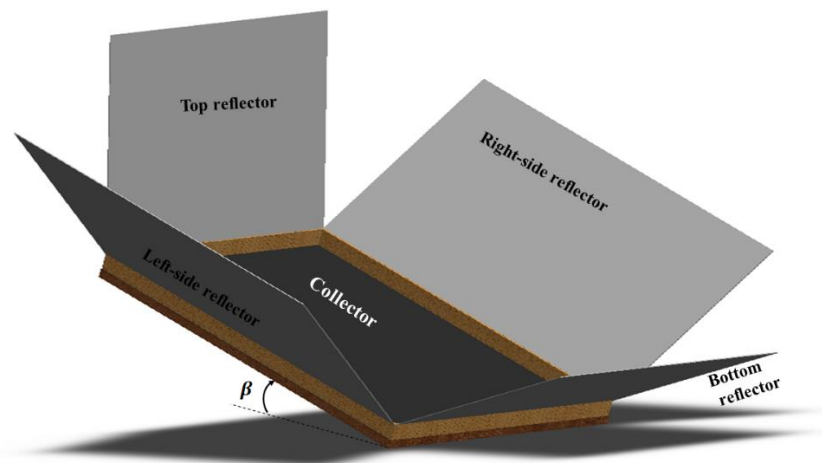


Fig.5.1 Schematic representation of modified FPC

While estimating the solar radiation absorbed by the absorber plate, the following assumptions were made (Zoran and Ljiljana., 2015):

- (i) The collector does not reflect radiation back to the reflectors;
- (ii) Shading effect of side reflectors on other reflectors is neglected
- (iii) Multiple reflection between reflectors is neglected or there are no mutual reflections between the reflectors;
- (iv) Diffuse component of the solar radiation (the sky-diffuse radiation and the reflected radiation from the ground) falls directly on the collector.

The total solar radiation on the modified FPC collector surface ($I_{tot-col}$) is the sum of the incident solar radiation on collector surface (I_T) and reflected solar radiation from all side reflectors (I_{ref}). Incident solar radiation is the sum of direct beam radiation and diffuse radiation. The reflected solar radiation is also the sum of reflected solar radiation from the right side reflector which reached the collector surface I_{ref-lr} with tilted plane angle γ_1 , the reflected solar radiation from the left side reflector which reached the collector surface I_{ref-rr} with tilted plane angle γ_2 , the reflected solar radiation from the top reflector which reached the collector surface I_{ref-tr} with tilted plane angle α_1 , and the reflected solar radiation from the bottom reflector which reached the collector surface I_{ref-br} with tilted plane angle α_2 as shown in Fig.5.2 and 5.3 (Zoran and Ljiljana., 2015).

$$I_{tot-col} = I_T + (1 - S_{rc})I_{ref} \quad 5.1$$

$$\text{Where, } I_T = I_{dir-col} + I_{d-col} = I_b R_b + I_d \left(\frac{1 + \cos \beta}{2} \right) + I_G \rho_g \left(\frac{1 - \cos \beta}{2} \right) \quad 5.2$$

$$I_{ref} = (I_{ref-lr} + I_{ref-rr} + I_{ref-tr} + I_{ref-br}) \quad 5.3$$

S_{rc} is summation of the shading factors of all the four reflectors on the collector

5.2.2 Estimation of Reflected Solar Radiation from Right and Left-Side Reflectors

The reflected solar radiation from the right-side reflector falls on the collector at an angle of σ_1 , and the reflected solar radiation from the left-side reflector falls on the collector at an angle of σ_2 . Fig.5.2 indicates the schematic diagram of right and left-side reflectors with their respective angle of inclination. These angles of incidence are founded by geometrical analysis, and can be expressed in terms of plane tilt angles of right side and left-side reflector γ_1 and γ_2 respectively as;

$$\sigma_1 = 2\gamma_1 - (\alpha + \beta) \quad 5.4$$

$$\sigma_2 = 2\gamma_2 + (\alpha + \beta) - 180 \quad 5.5$$

Incident radiations on the left-side reflector and the right-side reflector can be defined as;

$$I_{T-rr} = I_T \sin(\alpha + \beta - \gamma_1) \quad 5.6$$

$$I_{T-lr} = I_T \sin(\alpha + \beta - \gamma_2) \quad 5.7$$

Where I_T =global incident solar radiation on tilted surface

Therefore; reflected radiations from the left-side reflector and right-side reflector are defined as:

$$I_{ref-rr} = \rho_r \cdot I_T \cdot \sin\sigma_1 \cdot \sin(\alpha + \beta - \gamma_1) = \rho_r \cdot I_{T-rr} \cdot \sin\sigma_1 \quad 5.8$$

$$I_{ref-lr} = \rho_r \cdot I_T \cdot \sin\sigma_2 \cdot \sin(\alpha + \beta + \gamma_2) = \rho_r \cdot I_{T-lr} \cdot \sin\sigma_2 \quad 5.9$$

Where; ρ_r is the reflectance of reflector (for an aluminum reflector $\rho_r = 0.89$) (Ekechukwu and Ugwuoke, 2013).

5.2.3 Estimation of Reflected Solar Radiation from Top and Bottom Reflectors

Using similar procedure of estimation, the reflected solar radiation from the top reflector which falls on the collector at an angle of χ_1 , and the reflected solar radiation from the bottom reflector which falls on the collector at an angle of χ_2 is expressed in terms of plane tilt angles of top and bottom reflector α_1 and α_2 respectively as indicated in Fig.5.3.

$$\chi_1 = \beta - \alpha + 2\alpha_1 \quad 5.10$$

$$\chi_2 = \alpha - \beta + 2\alpha_2 \quad 5.11$$

Incident radiations on the top and bottom reflector can be defined as;

$$I_{T-tr} = I_T \sin(\alpha - \alpha_1) \quad 5.12$$

$$I_{T-br} = I_T \cos(\alpha + \alpha_2) \quad 5.13$$

Thus; reflected radiations from top and bottom reflector are defined as:

$$I_{ref-tr} = \rho_r \cdot I_T \cdot \sin\chi_1 \cdot \sin(\alpha - \alpha_1) = \rho_r \cdot I_{T-tr} \cdot \sin\chi_1 \quad 5.14$$

$$I_{ref-br} = \rho_r \cdot I_T \cdot \sin\chi_2 \cdot \sin(\alpha + \alpha_2) = \rho_r \cdot I_{T-br} \cdot \sin\chi_2 \quad 5.15$$

Therefore; Eqn. 4.1 can be written as;

$$I_{tot-col} = \left[I_b R_b + I_d \frac{(1+\cos\beta)}{2} + \rho_g I_G \frac{(1-\cos\beta)}{2} + \rho_r (1 - S_{rc}) (I_{T-rr} \cdot \sin\sigma_1 + I_{T-lr} \cdot \sin\sigma_2 + I_{T-tr} \cdot \sin\chi_1 + I_{T-br} \cdot \sin\chi_2) \right] \quad 5.16$$

$$I_{tot-col} = \left[I_T + \rho_r (1 - S_{rc}) (I_{T-rr} \cdot \sin\sigma_1 + I_{T-lr} \cdot \sin\sigma_2 + I_{T-tr} \cdot \sin\chi_1 + I_{T-br} \cdot \sin\chi_2) \right] \quad 5.17$$

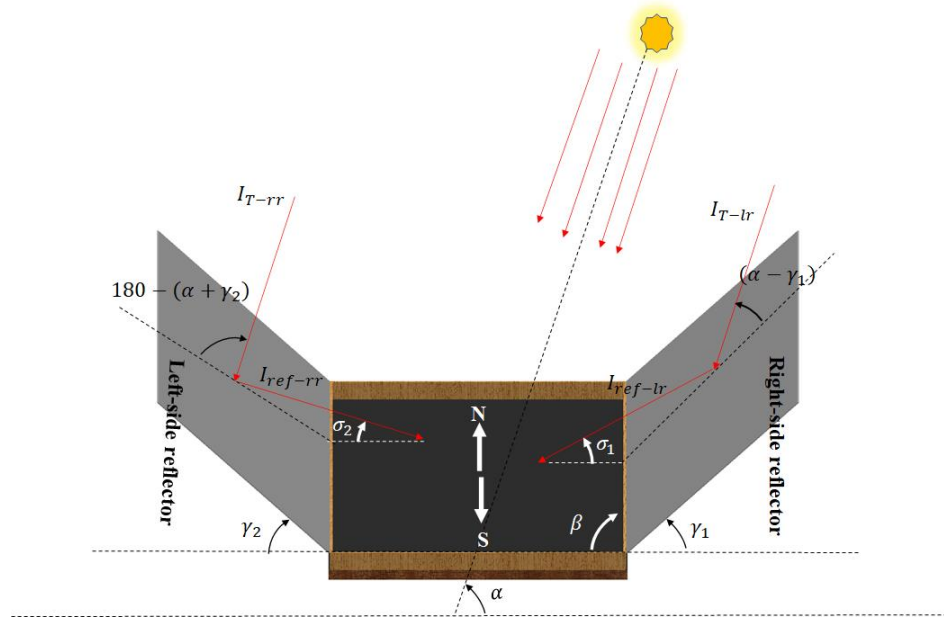


Fig.5.2 Schematic diagram of geometrical angles and solar radiation incident on right and left-side reflectors of modified FPC

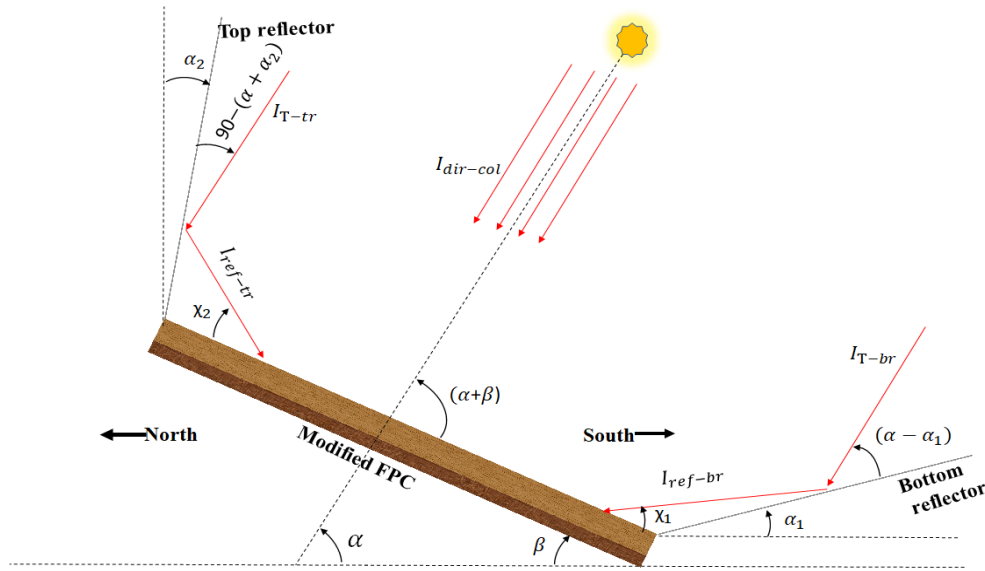


Fig 5.3 Schematic diagram of geometrical angles and solar radiation incident on top and bottom reflectors of modified FPC

Equation 5.17 shows that the total amount of solar radiation incident on the collector is a function of incident solar radiation intensity, incident angle of reflected radiation, optical properties of reflecting materials, the geometrical angle and orientation of both the collector and reflectors. The incident angle of reflected radiation from right-side, left-side, top and bottom of the reflectors $\sigma_1, \sigma_2, \chi_1$ and χ_2 respectively, depends on the tilt angle of collector (β), solar altitude angle (α) and

reflector inclination angles $\gamma_1, \gamma_2, \alpha_1$ and α_2 . To collect maximum amount of solar radiation, optimization of those angles is required.

The optimum values of both the collector and reflector inclination angle depends on the day number of the year (i.e. n starting from January 1st) at which the experiment is taken place. In the present case, the experiment is planned to be conducted in the range of days between June 1st (n =168) up to July 10th (n=208). The optimal values of collector tilt angle and all the reflector angles are estimated using the result found by Zoran and Ljiljana., (2015). The authors presented the variation of those angles with day of the year. Accordingly, the optimal values of required angles in relation with the day at which the present experiment takes place is presented in table 5.1 and used an input to determine the total amount of solar radiation absorbed ($I_{tot-col}$) by the collector.

Table 5.1 Optimum values of selected optical and geometrical angles of collector and reflectors of modified FPC

Angles	Optimal value
β	25°
α_1	40°
α_2	2.5°
γ_1	60°
γ_2	60°

Using values in table 5.1, the variation of the incident angle of reflected radiation from left-side, right-side, top and bottom of the reflectors $\sigma_1, \sigma_2, \chi_1$ and χ_2 with solar altitude angle(α) is estimated using the following co-relations obtained in the above sections.

For right and left-side reflectors $\sigma_1 = 2\gamma_1 - (\alpha + \beta) = 95 - \alpha$

$$\sigma_2 = 2\gamma_2 + (\alpha + \beta) - 180 = \alpha - 35$$

For top and bottom reflectors $\begin{cases} \chi_1 = \beta - \alpha + 2\alpha_1 = 105 - \alpha \\ \chi_2 = \alpha - \beta + 2\alpha_2 = \alpha - 20 \end{cases}$

Solar altitude angle can be found using the expression given by (Tiwari *et al.*, 2016);

$$\sin \alpha = \cos \phi . \cos \delta . \cos \omega + \sin \phi . \sin \delta \quad 5.18$$

substituting the above optimized values in Eqn.5.17 can be written as a function of solar altitude angle which varies from sunshine to sunset.

$$I_{tot-col} = I_T + [(1 - S_{rc})\rho_r((\sin(95 - \alpha).\sin(\alpha - 35) + \sin(\alpha - 35).\sin(\alpha + 85) + \sin(105 - \alpha).\sin(\alpha - 40) + \sin(\alpha - 20).\sin(\alpha + 2.5)))] \quad 5.19$$

5.2.4 Sizing of Modified FPC

In the sizing of FPC, the following terms are frequently used to define the area of flat plate solar collectors.

- ii. Absorber area (A_a) is defined as the dimensions of the absorber which absorbs the incident solar radiation.
- iii. Aperture area (A_p) is the area over which the solar radiation enters the collector
- iv. Gross area (A_g) is defined as the area based on the outer dimensions of the collectors (e.g. including the frame, glazing etc.).

Most of cooking application including the present study uses boiling type of food preparation. Thus, the main intention is to boil the water first and cooking the food required in it. When measurements are to be made, it is suggested that the cooking-pots should be filled with 5 kg of water per 1 m² of absorber area, and that the average solar radiation should be higher than 800 W/m² (Schwarzer and Silva., 2008). The area of the collector in the design of FPC depends on the heat gain by the working fluid and solar radiation incident on the collector. The collector area was obtained from the relationship given by (Schwarzer and Silva, 2003);

$$A = \frac{\dot{m}C_p\Delta T}{I_t} \quad 5.20$$

Where I_t is the average of total incident solar radiation on the tilted collector;

ΔT is the temperature difference of HTF at the outlet and inlet of the collector.

Study on mass flow rate ($\dot{m}$) through the collector by Kalogirou (2004) reported 0.02 Kg/s as minimum flow rate for testing FPC. In the present case an absorber area is calculated to be 2.18m² with maximum temperature difference between inlet and outlet HTF of 25K which is approximated into the commercial standard of 2m² with a length of 2m and width of 1m. As a flat plate collector alone can't achieve the required operating temperature for boiling type of food preparation, side reflectors are used to boost the amount of radiation falling on the absorber so as the outlet temperature and efficiency of the collector. In such cases the aperture area of the collector is the sum of absorber area and area covered by parallel sun rays on the collector. Fig.5.4 (a)and (b) indicates the geometrical analysis which is used to determine the aperture and gross area of modified flat plate collector.

Calculation of aperture area is done for solar noon where the collector receives maximum solar radiation with in a day and at solar noon solar altitude angle is at right angle. From Fig.5.4 (a) and (b) aperture area can be expressed as;

$$A_p = \text{projected absorber area} + \text{sum of projected area of reflectors facing the beam}$$

$$= L \times W + L (a_1 + a_2) + W (b_1 + b_2) \quad 5.21$$

Where: a_1 and a_2 are projected width of right and left-side reflectors respectively

b_1 and b_2 are projected length of top and bottom reflectors respectively

The expression for a_1 , a_2 , b_1 and b_2 was found in terms of actual length of reflectors using geometrical relation as follows;

$$a_1 = x_1 \sin(90 - \gamma_1) \quad a_2 = x_2 \sin(90 - \gamma_2)$$

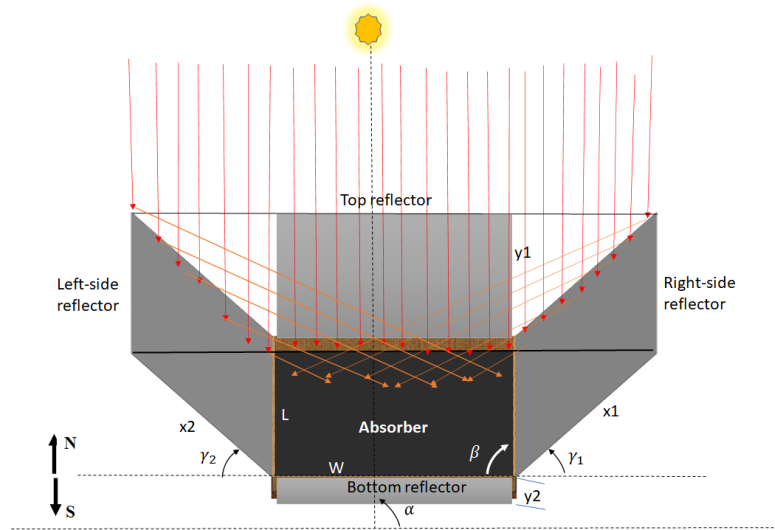
$$b_1 = y_1 \cos(\alpha_1) \quad b_2 = y_2 \sin(\alpha_2)$$

$$\text{Thus, } A_p = L \times W + L (x_1 \sin(90 - \gamma_1) + x_2 \sin(90 - \gamma_2)) + W (y_1 \cos(\alpha_1) + y_2 \sin(\alpha_2)) \quad 5.22$$

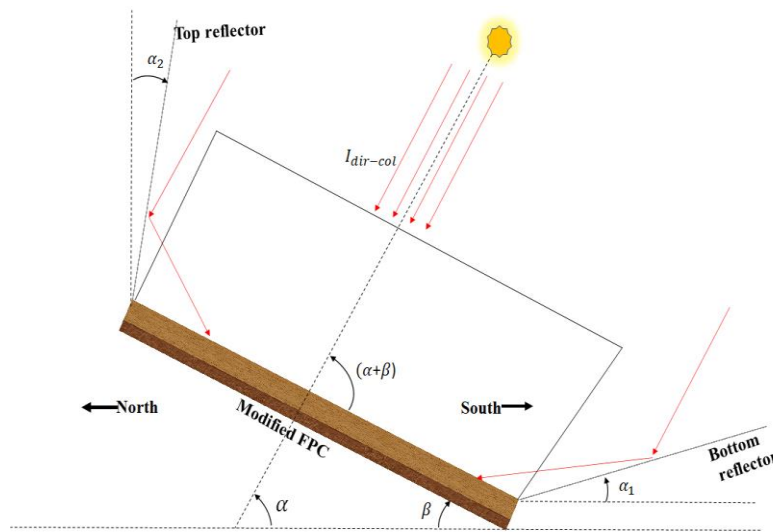
From Eqn.5.21 it can be seen that aperture area is a function of length and width of absorber and angle of inclinations of all the reflectors. Note that at solar noon α_2 should be adjusted so that solar radiation will be reflected from top reflector. The values of both aperture area and gross area are determined using optimized values of the inclination angles of all the reflectors from Table 5.1.

$$A_p = 2 \times 1 + 2(1 \sin(90 - 60) + 1 \sin(90 - 60)) + 1 (1 \cos(40) + 1 \sin(2.5)) = 6.81 \text{m}^2$$

The number of tubes in the design of flat plate collector is determined using the flow condition of heat transfer fluid in-side the tubes. The air gap between the absorber plate and the glass cover has its own effect on the convective heat loss from the absorber and shading effect. The natural convection coefficient decreases and the shading due to side walls increases with increase in gap size. Therefore, optimal value of air gap should be selected. Buchberg *et al.*, (1976) reported that the spacing between a tilted hot solar absorber and successive glass covers should be in the range of 3 to 8 cm in order to assure minimum conductance. Thus, for present case an air gap of 3cm is selected to minimize the convective heat loss as much as possible. Back and side insulations are required to decrease the conductive heat loss from the absorber plate. Thus, side and back of the collector was insulated by 30 mm thick thermo foam insulator. Table 5.2 summarizes the size and specification of the FPC and its components used in the development of the prototype.



a) Elevated top view



b) Side-view

Fig 5.4 Geometrical configuration of modified FPC

5.2.5 Mass Flow Rate of the System

Flat plate collector systems can be operated with different flow rates. At the same collector output, whenever the flow rate is higher, a lower temperature will be obtained and higher efficiency in the collector circuit. On the other hand, a lower flow rate results in a higher temperature output but lower efficiency. The mass flow rate of the system can be estimated from the efficiency of the collector and required outlet temperature of HTF using Eqn.5.22. With a given efficiency of collector the range of mass flow rate for flat plate collector varies from 0.009kg/sec to 0.03kg/sec (Dagim, 2016). Outlet of temperature of HTF from the collector can be obtained by;

$$T_o = \frac{\eta \cdot I_{tot-col} \cdot A_p}{\dot{m} \cdot C_p} + T_i \quad 5.22$$

The required outlet temprature from the collector is 373K and the average tap water temprature of adama city is 303K. if the collector effcency is assumed to be 0.7, the mass flow rate of the system can be determined from the maximum solar radiation of $1280\text{W}/\text{m}^2$ collected with in the given apprature area is found to be **0.02087kg/sec**. From the mass flow rate; volume flow rate of water can be determined as;

$$\dot{Q} = \frac{\dot{m}}{\rho} = 2.084 \times 10^{-5} \text{m}^3/\text{sec}$$

Table 5.2 Summary of Size and specification of modified FPC

Specification	Value
Absorber area	2 m ² (std)
Aperture area	6.81 m ²
Gross area	6.81 m ²
Width of absorber plate	1m
Length of absorber plate	2m
Width and Hight of left reflector	2m×1m
Width and Hight of right reflector	2m×1m
Width and Hight of top reflector	1m×1m
Width and Hight of bottom reflector	1m×1m
Air-gap b/n plate and covering glass	0.05m
Number of tubes	7
Thickness of the glass	0.005
Diameter of the riser tube	0.01m
Diameter of the header tube	0.022m
Thickness of side insulation	0.025m
Thickness of the header tube	0.0015m
Thickness of the riser tube	0.0015m
Bottom insulation thickness	0.03m

5.2.6 Material Selection for Modified FPC

i. Absorber plate

The primary function of the absorber plate is to absorb as much as possible of the radiation reaching through the glazing, to loss as little heat as possible upward to the atmosphere and downward through the back of the FPC, and to transfer the retained heat to the HTF. Factors that determine the choice of absorber material are its thermal conductivity, corrosion resistance, withstands the stagnation temperature its durability and ease of handling, its availability and cost. Keeping this in mind and of the commonly available materials, cooper is promising candidate. Thus, copper was selected as an absorber plate for the present FPC.

ii. Covering glass

Covering glass is used to admit as much solar radiation as possible and to reduce the upward loss of heat to the lowest attainable value. From the standpoint of the utilization of solar energy, the important characteristics are reflectivity (ρ), absorptivity (α), and transmissivity (τ). As a selection criterion for the glass reflectivity, absorption should be as low as possible and the transmissivity should be as high as possible for maximum efficiency. In the present case, plain glass (5mm thickness) was selected as the principal material used to glaze FPC. Because it has relatively high transmittance as much as 90% of the incoming short-wave radiation and is also easily available in local market as compared to other glazing materials.

iii. Header tube and Riser tube

Header tubes are used to admit and discharge the heat transfer fluid to the connecting tubes whereas riser tubes are used for directing and conducting the heat transfer fluid from the inlet header to the outlet. While designing, the factors such as choice of materials, inner and outer diameter, flow rate, inlet pressure, number of tubes, spacings of tubes and the thermal conductivity of bond between tube and absorber sheet must be exercised carefully. In the present case, copper tubes were selected as a riser tube to take the advantage of higher thermal conductivity whereas header tubes were circular steel tube as they are easily available and coat effective.

iv. Insulation

Flat plate collectors should be insulated to reduce conduction and convection losses through the back and sides of the collector box. As the collector may have to operate at temperature as high as 200°C the insulation must not deteriorate, outgas, expand or contract at temperatures between 30°C and

200°C. It must also have structural stability and must not become compact or settle with time and should not attract moisture with being fire resistant and light in weight. Among the possible insulating materials thermo-foam insulation is available and was selected for bottom and side wall insulation.

v. Casing and collector housing

To reduce contact heat loss from the absorber, wood was selected as a casing material as it is easily available and easy to construction and has lower thermal conductivity and used as an additional side insulation between absorber and from metal frame. The collector housing is provided to protect the insulation and absorber plate from the environment and also to minimize heat loss to the surrounding. Materials like aluminum, galvanized steel, fiber glass, wood etc. are used for making the collector housing. It is advisable to use the same material for collector support structure and housing to avoid contact corrosion. Thus, steel frame and wood are selected as a housing for present study.

vi. Frame and support

It is advisable to use the same material for collector support structure and housing to avoid contact corrosion. Mild steel angle metal frame and square pipe stand was selected as a support as it is suitable to withstand the weight of collector and has longer service life.

Table 5.3 Summary of material and property used for the design of flat plate collector

Part name	Material and property
Absorber plate	Sheet metal
Flow cross section	Circular
Glass cover	Plain glass
Insulation	Thermo-foam
Casing and cladding	Wood
Header tubes	Steel
Riser tubes	Copper
Frame and support	Steel
Thermal conductivity of absorber	385 W/m. K
thermal conductivity of Insulation	0.044 W/m. K
Transmittance of glass cover	0.91
Reflectivity of Aluminum foil	0.86
Heat transfer fluid (HTF)	Water

5.2.7 Optical Efficiency of Glass Cover

The optical efficiency (η_o) indicates the ability of glass cover to transmit the incident solar radiation into the absorber plate. It is usually expressed as a product of transmissivity of the glass (τ) and absorptivity of the absorber plate (α) (Dawit, 2018).

$$\eta_o = \tau\alpha \quad 5.23$$

For solar collectors having single glass cover, of the radiation passing through the cover system and incident on the plate, some is reflected back to the cover system. From the incident solar radiation on the glass cover, a fraction of $\tau\alpha$ is absorbed by the absorber plate and $(1 - \alpha)\tau$ is reflected back to the cover system.

5.3 Thermal Analysis of Modified FPC

In the thermal analysis of FPC's, the following assumptions has to be taken in to consideration (Duffie and Backman, 2013).

- Energy analysis, exergy analysis and their correlations are under steady state condition.
- The headers cover a small area of collector and can be neglected and it provides uniform flow to tubes.
- Absorption of solar energy by a cover is neglected
- heat flow is one dimensional through the cover and is opaque to infrared radiation with negligible temperature drop though it.
- The sky can be considered as a blackbody for long-wavelength radiation at an equivalent sky temperature.
- Temperature gradients around tubes can be neglected.
- The temperature gradients in the direction of flow and between the tubes can be treated independently.
- Material properties are independent of temperature.
- Loss through front and back are to the same ambient temperature.
- For new design dust and dirt on the collector can be negligible.

Thermal analysis of flat plate collector involves both energy and exergy analysis.

5.3.1 Energy Analysis

i. Useful heat gain

Generally, the useful heat gain from collector absorber to the working fluid for a flat plate solar collector can be expressed as:

$$\dot{Q}_u = \dot{m}C_p(T_{out} - T_{in}) \quad 5.24$$

Where $\dot{m}$, T_{in} , T_{out} and C_p are mass flow rate, outlet temperature, inlet temperature and heat capacity of working fluid respectively.

The rate of heat loss ($\dot{Q}_o$) in the form of overall heat loss coefficient (U_L), mean plate temperature (T_{pm}) and ambient temperature (T_a) was determined as;

$$\dot{Q}_o = A_p U_L (T_{pm} - T_a) \quad 5.25$$

For a steady state condition, the useful energy output of a flat plate collector with an aperture area (A_p) is the difference between the absorbed solar radiation and the thermal loss ($\dot{Q}_o$) (Farzad and Emad, 2012);

$$\dot{Q}_u = A_p [S - U_L (T_{pm} - T_a)] \quad 5.26$$

Where; T_{pm} is the mean absorber plate temperature in K

T_a is the ambient temperature in K

S is total absorbed radiation which can be expressed as S (W/m^2)

$$S = \eta_o I_{tot-col} \quad 5.27$$

η_o is optical efficiency of glass cover

$I_{tot-col}$ is total solar radiation incident on the collector (W/m^2)

ii. Collector-efficiency factor (F')

Flat plate collector efficiency factor is defined as the ratio of the actual rate of useful heat gain ($\dot{Q}_u$) to the rate of useful heat collection when the collector's absorbing plate temperature (T_p) is at the local fluid temperature (T_f) (Tiwari *et al.*, 2016).

$$F' = \frac{\dot{Q}_u}{\dot{Q}_u @ T_p = T_f} = \frac{\dot{Q}_u}{A_p [S - U_L (T_f - T_a)]} \quad 5.28$$

It can be calculated by considering the temperature distribution between two pipes of the collector absorber and by assuming that the temperature gradient in the flow direction is negligible. This analysis can be performed by considering the sheet tube configuration in Fig.5.5 where the distance between the tubes is W , the tube diameter is D , and the sheet thickness is δ .

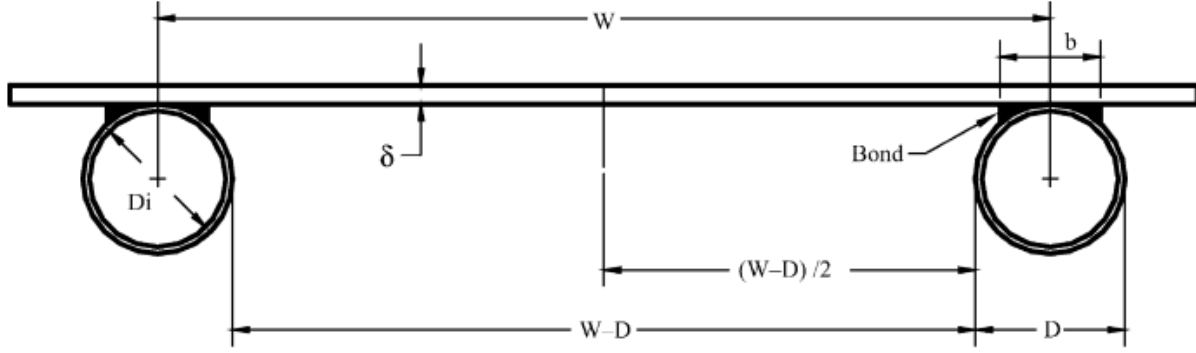


Fig 5.5 Absorber sheet-tube configuration of typical flat plate collector (Kalogirou, 2004)

By considering the sheet tube configuration as a classical fin problem, the useful heat gain by the fluid, can be expressed in terms of two resistances (i.e. fin convective resistance and sheet-tube bond resistance) (Kalogirou, 2004).

$$\dot{q}_u = \frac{T_p - T_f}{\frac{1}{h_{fin}\pi D_i} + \frac{1}{C_b}} \quad 5.29$$

Where: T_b is fin base temperature

h_{fin} is convective heat transfer coefficient of fin

C_b is bond conductance which can be estimated from knowledge of the bond thermal conductivity K_b ; the average bond thickness b_t ; and the bond width b . The bond conductance on a per unit length basis is given by;

$$C_b = \frac{K_b b}{b_t} \quad 5.30$$

At $T_p = T_f$ expression for useful heat gain by the fluid is;

$$\dot{q}_u = WF'[S - U_L(T_f - T_a)] \quad 5.31$$

Where: F' is collector-efficiency factor can be expressed as;

$$F' = \frac{\frac{1}{U_L}}{W \left[\frac{1}{U_L[D + (W-D)F]} + \frac{1}{C_b} + \frac{1}{h_{fin}\pi D_i} \right]} \quad 5.32$$

iii. Collector heat removal factor (F_R)

Collector heat removal factor (F_R) is quantity that relates the actual useful energy gain of a collector to the useful gain if the whole collector surface were at the fluid inlet temperature and can be expressed as (Tiwari *et al.*, 2016);

$$F_R = \frac{\dot{m} C_p}{A_p U_L} \left[1 - \exp \left(- \frac{F' U_L A_p}{\dot{m} C_p} \right) \right] \quad 5.33$$

Thus, equation 5.36 can be written in terms of heat removal factor to show the effects of collector's ability to transfer the absorbed heat to the heat transfer fluid, total heat loss coefficient (U_L) and optical efficiency of collector on thermal efficiency and can also be expressed as.

$$\dot{Q}_u = A_p F_R [S - U_L (T_{f in} - T_a)] \quad 5.34$$

Where: $T_{f in}$ = is inlet temperature of working fluid

F_R is Heat removal factor.

iv. Overall heat loss coefficient of the collector

In in a flat plate solar collector a part of absorbed solar energy S is thermally lost through the top bottom and side of the collector. Thus, the total heat loss coefficient (U_L) of a flat plate solar collector consists of top heat loss, bottom heat loss and side heat loss. Since the value of side loss is very small compared with other two losses, this term is neglected. The total heat loss coefficient (U_L) was obtained by the expression (Farzad and Emad, 2012).

$$U_L = U_t + U_b \quad 5.35$$

The energy loss through the top is the result of convection and radiation between glass cover and absorber plates. An empirical equation for the heat loss from the top of the absorber plate (U_t) for single glazing was developed by Klein, (1979) as cited by (Duffie and Beckman, 2013).

$$U_t = \left(\frac{N}{\frac{C}{T_{pm}} \left(\frac{T_{pm} - T_a}{(N+f)} \right)^e + \frac{1}{h_w}} \right)^{-1} + \frac{\sigma (T_{pm} + T_a) (T_{pm}^2 + T_a^2)}{\frac{1}{\epsilon_p + 0.00591 N h_w} + \frac{2N + f - 1 + 0.133 \epsilon_p}{\epsilon_g} - N} \quad 5.36$$

Where; N = number of glass covers

$$f = (1 + 0.089 h_w - 0.1166 h_w \epsilon_p) (1 + 0.07866 N)$$

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

$$e = 0.430 (1 - 100/T_{pm})$$

ϵ_p = emittance of absorber plate

ϵ_g = emittance of glass cover

h_w = wind heat transfer coefficient ($W/m^2 \text{ } ^\circ C$)

The top heat loss coefficient can also be derived as a function of sky temperature (T_{sky}) and glass cover temperature (T_g) from the expression of heat loss rate from the top of the collector ($Q_{t l}$);

$$Q_{tl} = h_w(T_g - T_a) + \sigma \epsilon_g(T_g^4 - T_{sky}^4) \quad 5.37$$

$$U_t = \frac{Q_{tl}}{T_{pm} - T_a}$$

Therefore:

$$U_t = \frac{h_w(T_g - T_a) + \sigma \epsilon_g(T_g^4 - T_{sky}^4)}{T_{pm} - T_a} \quad 5.38$$

Where; $T_{sky} = T_a - 6$

Heat loss coefficient from the bottom of absorber plate is calculated from the following equation;

$$U_b = \frac{k_b}{\delta_b} \quad 5.39$$

Where; k_b is the thermal conductivity of bottom material

δ_b is the thickness of bottom plate material

The convective heat loss from bottom and side wall ($h_b = h_w$) of the collector as a function of wind speed can be expressed as (Dawit, 2018).

$$h_b = h_w = 2.8 + 3V_w \quad 5.40$$

v. Energy efficiency (η_{th})

The thermal efficiency (η_{th}) of the solar collector is defined as the ratio of the output useful energy to the absorbed solar energy. After calculating net heat transfer rate between the absorber and heat transfer fluid, the performance of the flat plate collector in terms of energy efficiency (η_{th}) can be obtained from (Farzad and Emad, 2012).

$$\eta_{th} = \frac{\dot{Q}_u}{A_p I_{tot-col}} \quad 5.41$$

Substituting equation 5.24 in to equation 5.41, leads to the following expression for thermal energy efficiency;

$$\eta_{th} = \frac{\dot{m} C_p (T_{out} - T_{in})}{A_p I_{tot-col}} \quad 5.42$$

The temperature distribution on the surface of the solar collector along the direction of the fluid flow can be determined by the following correlation given by (Tiwari *et al.*, 2016).

$$\frac{T_{out} - T_a - S/U_L}{T_{in} - T_a - S/U_L} = \exp\left(-\frac{F' U_L A_p}{\dot{m} C_p}\right) \quad 5.43$$

Therefore, using Eqn.5.42 and Eqn.5.43 the correlation of energy efficiency of the flat plate collector can be rephrased into the following form:

$$\eta_{th} = \frac{\dot{m}c_p \left[(T_{in} - T_a - S/U_L) \left(\exp\left(-\frac{F' U_L A_p}{\dot{m}c_p} \right) - 1 \right) \right]}{A_p I_{tot-col}} \quad 5.44$$

5.3.2 Exergy Analysis

Unlike first law of thermodynamics, second law of thermodynamics quantifies the quality of energy. The quality of energy is gauged from the state of system in relation to the surrounding conditions. This quality or usefulness of energy is termed as “exergy” (Jegadheeswaran *et al.*, 2010). It can also be defined as the maximum amount of work which can be produced by the system when it is in equilibrium with the surrounding (Dawit, 2018). Exergy analysis is the utilization of 2nd law of thermodynamics which provides a useful measure of energy quality. Considering the thermodynamics system as a control volume and starting from first law of thermodynamics, the rate of energy storage can be written as follows:

$$\frac{dE}{dt} = \sum \dot{Q}_k - \dot{W} + \sum \dot{m}_i \hat{h}_i - \sum \dot{m}_e \hat{h}_e \quad 5.45$$

Where: $\hat{h}$ is specific energy flow

$$\hat{h} = \sum \text{specific entalphy}(h) + k.e + p.e \quad 5.46$$

Second law of thermodynamics for such control volume can also be written as;

$$\frac{ds}{dt} = \sum \frac{\dot{Q}_k}{T_k} + \sum \dot{m}_i s_i - \sum \dot{m}_e s_e + \dot{S}_{gen} \quad 5.47$$

Eliminating the term of heat transfer to surrounding (Q_o) which conducts in the ambient temperature T_o and re-arranging equation 5.45 and 5.47, the following equation will be found;

$$\dot{W} = -\frac{d}{dt} (E - T_o S_x) + \sum_{k=1}^n \left(1 - \frac{T_o}{T_k} \right) \dot{Q}_k + \sum_i \dot{m}(\hat{h} - T_o S_x) - \sum_e \dot{m}(\hat{h} - T_o S_x) - T_o \dot{S}_{gen} \quad 5.48$$

Where: $\dot{S}_{gen}$ is generated entropy by system irreversibility.

In reversible process, entropy will not be generated. Thus, equation 4.48 can be written as

$$\dot{W}_{rev} = -\frac{d}{dt} (E - T_o S_x) + \sum_{k=1}^n \left(1 - \frac{T_o}{T_k} \right) \dot{Q}_k + \sum_i \dot{m}(\hat{h} - T_o S_x) - \sum_e \dot{m}(\hat{h} - T_o S_x) \quad 5.49$$

Where: $\dot{W}_{rev}$ is the maximum work which a system in a control volume can produce and is known as exergy of a system.

The following assumptions were taken for exergy analysis:

- (i) Steady flow condition.
- (ii) kinetic and potential energies effect were neglected.

- (iii) The exergy destructed during transfer of incident radiation from reflectors to the collector is neglected.

Under steady state condition, the equation of exergy balance takes the following form;

$$\sum \dot{Ex}_{in} - \sum \dot{Ex}_{out} = \sum \dot{Ex}_{dest} \quad 5.50$$

Where: $\dot{Ex}_{in}$, $\dot{Ex}_{out}$ and $\dot{Ex}_{dest}$ are the inlet, outlet and destructed exergy rate, respectively.

Inlet exergy in the analysis of flat plate solar collector consists of exergy of absorbed solar energy from sun ($\dot{Ex}_{in.rad}$) and Exergy of collector inlet heat transfer fluid ($\dot{Ex}_{in.f}$). Whereas outlet exergy of the collector is the same as the exergy of the outlet fluid ($\dot{Ex}_{out.f}$). Thus, the amount of exergy loss in the collector is the difference of the two components.

The exergy rate from working fluid is calculated through the following equations;

$$\dot{Ex}_f = \dot{m}C_p \left[(T_f - T_0) + T_0 \ln\left(\frac{T_f}{T_0}\right) \right] \quad 5.51$$

The increase in exergy flow can be represented by the difference in the inlet and outlet exergies as;

$$\dot{Ex}_{out.f} - \dot{Ex}_{in.f} = \dot{m}C_p \left[(T_{f.out} - T_{f.in}) - T_a \left(\ln \frac{T_{f.out}}{T_{f.in}} \right) \right] \quad 5.52$$

The exergy content of an exergy source with temperature T , heat transfer rate $\dot{Q}$ and environment temperature T_0 can be obtained as;

$$\dot{Ex}_{heat} = \dot{Q} \left(1 - \frac{T_0}{T} \right) \quad 5.53$$

The major part of inlet exergy to the system is that solar radiation which is absorbed by the absorption plates and it is possible to calculate the amount of solar radiated exergy using eq.5.53:

$$\dot{Ex}_{in.rad} = A_p I_{tot-col} \left[1 - \left(\frac{T_a}{T_s} \right) \right] \quad 5.54$$

Where: T_s is the apparent sun temperature as exergy source which is approximately 3/4 of the blackbody temperature of the sun. It is assumed to be 4500 K (Farzad and Emad, 2012).

The amount of absorbed exergy by the flat plate collector's absorber plate can be calculated from Eqn.5.51 as:

$$\dot{Ex}_{in.abs} = \eta_o A_p I_{tot-col} \left[1 - \left(\frac{T_0}{T_p} \right) \right] \quad 5.55$$

In the present system, heat transfer from sun to the working fluid consists of two main parts, absorbing the solar radiation by absorber plate and heat transfer from absorber plate to working fluid. Thus, exergy destruction mainly occurs during such heat transfer process other than flow exergy destruction parts. Neglecting the exergy destructed during radiation heat transfer from reflectors to

the absorber plate, exergy destruction in absorbing process is due to temperature difference between collector's absorber plate and temperature of solar radiation source.

$$\begin{aligned}\dot{E}x_{dest.s-p} &= \dot{E}x_{in.rad} - \dot{E}x_{in.abs} \\ \dot{E}x_{dest.s-p} &= A_p I_{tot-col} \left[1 - \eta_o + \eta_o \left(\frac{T_o}{T_p} \right) - \frac{T_o}{T_s} \right]\end{aligned}\quad 5.56$$

The second part of exergy destructions is due to heat leakage from absorber plate to surroundings. $Q_{leakage}$ from absorber plate is occurred due is temperature difference between absorber plate and its surrounding and can be obtained from:

$$Q_{leakage} = U_L A (T_p - T_o) \quad 5.57$$

Thus, exergy destruction due to collector's heat losses is calculated from Eqn. 5.51 and Eqn.5.57:

$$\dot{E}x_{dest.leakage} = U_L A (T_p - T_o) \left(1 - \frac{T_o}{T} \right) \quad 5.58$$

The other part of exergy losses is due to heat transfer process from absorber plate to the working fluid. The difference between transferred exergy from the plate to fluid causes the working fluid's exergy to increase. Thus, the transferred exergy from absorber plate to the working fluid is obtained from;

$$\dot{E}x_{p-f} = \dot{m} C_p (T_{f.out} - T_{f.in}) \left(1 - \frac{T_o}{T_p} \right) \quad 5.59$$

The rate of exergy destruction in the heat transfer process from absorber plate to working fluid can be obtained by subtracting Eqn.5.52 from Eqn. 5.59:

$$\dot{E}x_{dest. p-f} = \dot{m}_f C_p T_o \left[\ln \left(\frac{T_{f.out}}{T_{f.in}} \right) - \frac{T_{f.out} - T_{f.in}}{T_p} \right] \quad 5.60$$

Therefore, exergy efficiency (η_{ex}) of a flat plate collector is calculated by dividing the increase in working fluid flow exergy as a total outlet exergy with the total inlet exergy from solar radiation.

$$\eta_{ex} = \frac{\dot{m}_f C_p \left[(T_{f.out} - T_{f.in}) - T_o \left(\ln \frac{T_{f.out}}{T_{f.in}} \right) \right]}{A_p I_{tot-col} \left[1 - \left(\frac{T_o}{T_s} \right) \right]} \quad 5.61$$

The equation of exergy efficiency can be rephrased by substituting Eqn.5.43 in to Eqn.5.61 as:

$$\eta_{ex} = \frac{\dot{m}_f C_p \left[\left(T_{f.in} - T_a - \frac{S}{U_L} \right) \left(\exp \left(-\frac{U_L A_p F'}{\dot{m} C_p} \right) - 1 \right) \right] - \dot{m} C_p \left[T_a \frac{\left(\exp \left(-\frac{U_L A_p F'}{\dot{m} C_p} \right) - 1 \right)}{T_{f.in}} \left(\left(T_{f.in} - T_a - \frac{S}{U_L} \right) + 1 \right) \right]}{A_p I_{tot-col} \left[1 - \left(\frac{T_o}{T_s} \right) \right]} \quad 5.62$$

5.4 Design of Latent Heat Storage System

The design of latent heat storage system involves the selection of a storage material, design of geometrical configuration of containment, and heat exchanger type.

5.4.1 Selection of Phase Change Material

To select an appropriate PCM for a specific solar application, the first criterion to be considered is the operational temperature range required by the cooking system and then its availability and cost. The operating temperature must match the transition temperature of the storage material. The PCM to be used in the design of thermal storage systems should accomplish the following desirable thermophysical properties (Medved *et al.*, 2010).

- i. Melting temperature in the desired operating temperature range.
- ii. High latent heat of fusion per unit volume so that the required volume of the container to store a given amount of energy is lesser.
- iii. High specific heat to provide for additional significant sensible heat storage.
- iv. High thermal conductivity of both solid and liquid phases to assist the charging and discharging of energy of the storage systems.
- v. Small volume changes on phase transformation and small vapor pressure at operating temperatures to reduce the containment problem.
- vi. Congruent melting of the PCM for a constant storage capacity of the material with each freezing/melting cycle

Assuming that a temperature difference of about 10°C is used to charge the storage unit, the melting point of the PCM should be about 10°C below charging temperature (Kantole, 2012). Since the lowest temperature required for cooking most kinds of foods is about 75°C (Domanski *et al.*, 1995), in literature there is no appropriate PCM within the required temperature range as some of them are above the required and some below. PCM selection for the present case is highly depend on availability in local market and cost. Among the available PCM, industrial grade stearic acid ($C_{17}H_{35}CO_2H$) with melting point ($T_m = 70-72^\circ C$), latent heat of fusion 192.1 kJ/kg is selected to be used as the PCM in the indoor cooking unit of the present indirect solar cooking system. Stearic acid is available in Ethiopian local market and was purchased from Wehib Chemical Trading found in Merkato, Addis Ababa (Ethiopia). Table 5.4 shows the physical property of stearic acid.

5.4.2 Selection of Heat Transfer Fluid

Criteria to be considered when selecting the HTF (Joseph, 2012) are: Coefficient of expansion, viscosity, thermal capacity, freezing point, boiling point, flash point; it should not exhibit a high level of the toxicity behavior. From the most commonly used HTF, water was selected for the present study as it is non-toxic, inexpensive, high specific heat, low viscosity, and easy to be circulated. Table 5.5 shows the property of water at standard temperature and pressure (STP).

Table 5.4 Thermo-physical properties of stearic acid (C₁₇H₃₅CO₂H)

Property	Value (Nagaraj <i>et al.</i> , 2016, Dominski <i>et al.</i> , 1994, Sherma <i>et al.</i> , 2009, Ali <i>et al.</i> , 2007)
Melting point(K)	343-345
Latent heat of fusion (KJ/kg)	192.1
Thermal conductivity(W/m.K)	0.173(liquid 343K), 0.3(solid 298K)
Specific heat capacity (KJ/kg.K)	1.6(solid), 2.2(Liquid)
Density (kg/m ³)	941(solid 298K), 847(liquid 353K)
Thermal expansion coefficient(1/k)	0.000872
Dynamic viscosity (Pa.s)	0.00779

Table 5.5 Properties of water at STP

Property	Value (Sherma <i>et al.</i> , 2009, Cabeza <i>et al.</i> , 2011, Murat and khamid, 2006)
Boiling point(°C)	100
Dynamic viscosity (kg/m.s)	0.000798
Thermal conductivity(W/m.K)	0.6
Specific heat capacity (kJ/kg.K)	4.178
Density (kg/m ³)	995.71

5.4.3 Sizing of Latent Heat Storage Unit

The size of latent heat storage unit was determined based on the amount of PCM in kilogram required to heat for late hour operation of the solar cooker with a load on it. The average daily useful energy requirement for cooking reportedly varies in the range of 1.7 –2.7 MJ/person/day (Sunil and Tera, 2018). Assuming 2.1MJ/day is required for an average person who needs a meal 3 times a day, 0.7 MJ/person/meal is required. The present solar cooker is intended to prepare about 80% of daily meal (i.e. other meals are considered as frying based meals out of most Ethiopian foods) for 10 persons. Thus 5.6MJ heat capacity is considered to determine the mass and storage volume of LHS unit. The storage volume of LHS unit can be determined from the mass of PCM by the expression (Joseph, 2012);

$$V_{storage} = 1.5\left(\frac{\pi D_{cyl}^2}{4}\right)L \quad 5.63$$

To find the volume of storage required, the tube volume (V_{tube}) should be determined as follows;

$$V_L = \pi L n R_o^2 \quad 5.64$$

Where D_{cyl} diameter of containment cylinder, L is the length of LHS unit, R_o and R_i are outer and inner diameter of HTF tube inside the shell, n is the number of tubes, and Allowance factor 1.5 is induced accounting the thermal expansion of the PCM whenever phase change occurs.

The mass of the PCM m_{PCM} is determined from the amount of heat stored expressed as;

$$Q = m_{pcm} [Cp_{pcm}(T_m - T_{ini}) + L_{F.pcm}] \quad 5.65$$

$$V_{PCM} = \frac{m_{PCM}}{\rho_{PCM}} \quad 5.66$$

$$V_{storage} = V_{PCM} + V_{tube} \quad 5.67$$

5.4.4 Determination of LHS Shell and Tube Design Parameters

Using the above correlations, the storage volume of the LHS can be determined from determining the mass of PCM required to store the required heat. Using Eqn.5.65 the mass of PCM required was found to be 20kg. Using Eqn.5.66 volume of the PCM is determined to be 0.02046m³. To store 20 kg of PCM inside the shell and tube storage unit and its dimensional parameters were determined based on the procedure in **APPENDIX A**. Table 5.6 summarizes the design parameters and specification of shell and tube LHS unit.

Table 5.6 Design parameters of LHS unit

Parameter	Value
Length of LHS(m)	0.8m
Diameter of shell (D_{cyl})	0.2m
Outer radius of tube (R_o)	0.0172m
Inner radius of tube (R_i)	0.016m
Number of tube(n)	5
Mass of PCM (m_{PCM})	20kg
Volume of PCM (V_{PCM})	0.02046 m ³
Tube volume (V_{tube})	0.003717m ³
Volume of LHS unit ($V_{storage}$)	0.0241m ³

5.5 Thermal Analysis of Thermal Energy Storage System

In the analysis of thermal energy storage system, it is worthy to mention some of the following the critical terms.

i. Charging and charging time

Charging is the process of storing heat energy in the latent heat storage material while it changes its phase from solid state to liquid state. Charging time implies the total time it takes for the LHS material or PCM to completely melt into liquid state. The LHS is said to be fully charged when the entire PCM is completely melted, i.e. when the melt fraction reaches a value of one.

ii. Discharging and discharging time

Discharging is the process of releasing heat energy in the latent heat storage material while it changes its phase from liquid state to solid state. Discharging time for LHS is defined with respect to the temperature decrease of the PCM. The LHS is said to be fully discharged when the entire PCM is solidified, i.e. the melt fraction becomes zero.

iii. Melt fraction

Melt fraction (θ_f) is generally referred as the liquid fraction of the PCM when it exists in a 2-phase form. The highest temperature below which the material is completely in the solid state is called solidus temperature (T_S) and the lowest temperature above which the material is in the liquid state is called liquidus temperature (T_L). Melt fraction (θ_f) of the PCM was obtained from the relationship (Dawit, 2018).

$$\theta_f = \frac{T - T_S}{T_L - T_S} = \frac{T - T_m + \Delta T_m}{2\Delta T_m} = \begin{cases} 0 & \text{for } T < T_S \\ 0 - 1 & \text{for } T_S \leq T \leq T_L \\ 1 & \text{for } T > T_S \end{cases} \quad 5.68$$

Where T_m is the melting point of PCM

5.5.1 Energy Analysis of Latent Heat Storage System

Energy analysis of latent heat storage system applies first law of thermodynamics for both charging and discharging modes of the PCM.

5.5.1.1 Energy Stored During Charging

Energy stored can be expressed in terms of stored heat flux during charging process of the latent heat storage which consists of sensible heat and latent heat flux and is obtained from the expression given by (Salwa *et al.*, 2014):

$$\dot{Q}_{ch} = \frac{m_{pcm}[Cp_s(T_m - \bar{T}_{ini-ch.pcm}) + Cp_l(\bar{T}_{fin-ch.pcm} - T_m) + L_{F.pcm}\theta_f]}{\Delta t_{ch}} \quad 5.69$$

Where: m_{pcm} is mass of PCM

θ_f is melt fraction of PCM material

Cp_s and Cp_l are specific heat capacity of PCM at solid state and liquid state respectively.

$L_{F.pcm}$ is latent heat of fusion for the PCM

Δt_{ch} is the total charging time

T_m is melting temperature

$\bar{T}_{ini-ch.pcm}$ and $\bar{T}_{fin-ch.pcm}$ are average temperature of PCM at initial and final charging process respectively.

5.5.1.2 Energy Released During Discharging

Energy released from the LHS PCM material to the HTF can also be expressed in terms of released heat flux during discharging process (Niyas, 2017).

$$\dot{Q}_{dis} = \frac{m_{pcm}[Cp_l(T_m - \bar{T}_{fin-dis.pcm}) + Cp_s(\bar{T}_{in-dis.pcm} - T_m) + L_{F.pcm}(1 - \theta_f)]}{\Delta t_{dis}} \quad 5.70$$

Where: Δt_{dis} is the total discharging time

$\bar{T}_{ini-dis.pcm}$ and $\bar{T}_{fin-dis.pcm}$ are average temperature of PCM at initial and final discharging process respectively.

5.5.1.3 Energy Efficiency of Latent Heat Storage System.

The energy efficiency of LHS system can be defined by charging and discharge efficiencies separately as:

$$\eta_{char} = \left[\frac{\text{Total energy stored in the PCM}}{\text{Total energy supplied by the HTF}} \right] \quad 5.71$$

$$\eta_{dis} = \left[\frac{\text{Total energy absorbed by the HTF}}{\text{Total energy stored in the PCM}} \right] \quad 5.72$$

Thus, the overall energy efficiency is expressed as;

$$\eta_{en.overall} = \eta_{char}\eta_{dis} = \left[\frac{\text{Total energy absorbed by the HTF}}{\text{Total energy supplied by the HTF}} \right] \quad 5.73$$

5.5.2 Exergy Analysis of Latent Heat Storage System

Exergy analysis takes into account the temperature at which the exergy is available for storing as it uses the difference between the inlet temperature of HTF and environment temperature (T_a) rather than the difference between the inlet and outlet temperatures of HTF.

5.5.2.1 Exergy Stored During Charging

The rate of exergy supplied by HTF during charging period of the LHS system is given by (Jegadheeswaran *et al.*, 2014):

$$\dot{E}x_{in,HTF} = \dot{m}_f C_p \left[(T_{f,out} - T_a) - T_a \left(\ln \frac{T_{f,out}}{T_a} \right) \right] \quad 5.74$$

During the charging mode of operation, the exergy stored at any time instant is computed from the instantaneous heat transfer rate expressed as:

$$\dot{E}x_{stored} = Q \left[1 - \frac{T_a}{T_{fin-ch.PCM}} \right] - Q_{loss} \left[1 - \left(\frac{T_a}{T_{fin.ch.PCM}} \right) \right] \quad 5.75$$

Therefore, the total exergy stored in PCM during charging can be readily obtained by performing time integration of Eq. (5.75) from initial charging time to final. However, it requires the computation of heat loss to the surrounding. An alternate form, simpler than the time integrated form of Eq. (5.75) can be found in (Jegadheeswaran *et al.*, 2014):

$$\begin{aligned} \dot{E}x_{stored.ch} = & m L_f \left[1 - \frac{T_a}{T_m} \right] + m C_{p,s} \left[(T_m - \bar{T}_{ini-ch.pcm}) - T_o \left(\ln \frac{T_m}{\bar{T}_{ini-ch.pcm}} \right) \right] + m C_{p,l} \left[\bar{T}_{fin-ch.pcm} - \right. \\ & \left. T_m \right) - T_o \left(\ln \frac{\bar{T}_{fin-ch.pcm}}{T_m} \right) \right] \end{aligned} \quad 5.76$$

5.5.2.2 Exergy Released During Discharging

Expressions for exergy stored during discharging process can be obtained in the similar way corresponds to the charging process as (Jegadheeswaran *et al.*, 2014);

$$\begin{aligned} \dot{E}x_{released.dis} = & m L_f \left[1 - \frac{T_a}{T_m} \right] + m C_{p,s} \left[(\bar{T}_{ini-ch.pcm} - T_m) - T_o \left(\ln \frac{\bar{T}_{ini-ch.pcm}}{T_m} \right) \right] + m C_{p,l} \left[T_m - \right. \\ & \left. \bar{T}_{fin-ch.pcm} \right) - T_o \left(\ln \frac{T_m}{\bar{T}_{fin-ch.pcm}} \right) \right] \end{aligned} \quad 5.77$$

5.5.2.3 Exergy Efficiency

The overall exergy efficiency of latent heat storage is expressed as;

$$\psi_{exe\ overall} = \frac{\text{Exergy extracted from the PCM by the HTF during discharging}}{\text{Exergy input to the PCM during charging}} \quad 5.78$$

$$\psi_{exe\ overall} = \frac{Ex_{released\ dis}}{Ex_{stored.ch}} \quad 5.79$$

5.6 Design of Cooking Chamber

The cooking chamber consists of storage vessel and cooking pot were designed based on the amount of water required to be stored in the vessel and utilized for cooking in the cooking pot. Amount of water were determined by considering the capacity of pot to perform an experimental test using the given standard and to prepare a meal for the required number of people. The amount of water required to prepare a meal for persons is estimated to be 5l and storage tank volume were estimated to be 10L. Thus, volume of pot (V_{pot}) is found to be 0.005m^3 whereas the volume of storage vessel (V_{vessel}) is 0.01m^3 .

5.7 Performance Parameters of the Cooker

The performance of solar cookers has generally been rated previously by test procedures that are dependent not on unique solar cooker properties but on climatic parameters, namely solar insolation, ambient temperature and local wind speeds and thus it is difficult to follow a unique standard for testing their performance (Ekechukwu and Ugwuoke, 2013). Different test standards have been proposed for evaluating the performance of solar cookers. Those are the American Society of Agricultural Engineers Standard ASAE S580, the European Committee on Solar Cooking Research (ECSCR) and the Bureau of Indian Standard for Testing (BIST). ASAE S580 uses single figure of merit which is the cooking Power(P) to measure the performance of the solar cooker. Although ASAE S580 provides a simple, yet meaningful and objective measure of cooker performance, it does not address issues other than strictly thermal performance of the cooker and it leaves out any direct measurement of heat losses (Shaw, 2002). The ECSCR provides a more subjective measure of cooker performance as it deals with observation of safety factors, ease of cooking pot access and estimated durability but it includes no means of normalization and is faced to errors and inaccuracies as it depends on measurement of some condition which depend on time (Shaw, 2002). BIST is mostly used test method which is first proposed by Mullick *et al.*, (1987) to evaluate the performance of solar cookers and provides a more technical framework using two figures of merit, calculated so as to be as independent of environmental conditions as possible. Thus, the present study uses both the ASAE S580 test standard and the two figures of merits of BIST as performance parameter.

5.7.1 Cooking Power

The cooking power(P) in watt of a solar cooker is found by using ASAE S580 test standard as follows;

$$P = m_w C_{p.water} \frac{T_t - \bar{T}_{initial}}{\Delta t} \quad 5.80$$

Where: m_w is the mass of water in kg

C_p = specific heat of water = 4.2 kJ/kg.K

$\bar{T}_{initial}$ is the average initial temperature of tap water in Adama city

T_t is the water temperature after time t

Δt is the time it takes for $\bar{T}_{initial}$ to reach T_t

The standardized form of cooking power (Ps) is also given by Shaw, (2002).

$$P_s = P \left(\frac{700}{I_{tot.col}} \right) \quad 5.81$$

Where: $I_{tot.col}$ = average of total insolation on the collector with in the test interval (W/m²)

5.7.2 Thermal Figures of Merits

The two figures of merits based on BIST can serve as a performance criterion for a solar cooker.

- i. **The first figure of merits (F_1)** is the ratio of optical efficiency of the glass cover (η_0) and total heat loss coefficient (U_L)

$$F_1 = \frac{\eta_0}{U_L} = \frac{(T_{mp} - T_a)}{H_s} \quad 5.82$$

Where H_s is solar insolation on horizontal surface taken at stagnation time

- ii. **The second figure of merits;** derived from the sensible heating of water is a measure of time required to raise the temperature of a known mass of water to its boiling temperature and is given as Mullick *et al.*, (1987);

$$F_2 = \frac{F_1(m_w C_{pw})}{A_p \cdot t} \ln \left[\frac{1 - \frac{1}{F_1} \frac{(T_{initial} - T_a)}{H}}{1 - \frac{1}{F_1} \frac{(T_{final} - T_a)}{H}} \right] \quad 5.83$$

Where: $T_{initial}$ = initial tap water temperature

T_{final} = final water temperature (100°C) after time (t)

A_p = aperture area pf the collector

H = average horizontal insolation

5.8 Required Pumping Power

In order circulate the HTF through the system, it must be equipped with a pumping system that can overcome the pressure drop throughout the whole system which is specified by ΔP . pressure drop in the system was determined by the relationship (Dawit, 2018).

$$\Delta P = f \frac{\rho V^2 \Delta L}{2d} + K \frac{\rho V^2}{2} \quad 5.84$$

Where V is the mean flow velocity of fluid, d is tube diameter, ρ is density, f is friction factor, Δl is length of the tube and K is head loss coefficient.

Frictional factor f which caused by the viscosity of fluid during laminar flow inside tube was determined by the relation;

$$f = \frac{64}{Re} \quad 5.85$$

where Re is the Reynolds number and is expressed as;

$$Re = \frac{\rho V d}{\mu} \quad 5.86$$

Mean velocity of fluid (V) can be determined by;

$$V = \frac{4\dot{m}}{\rho \pi d^2} \quad 5.87$$

Therefore; the pumping power can be determined by;

$$P_{pump} = \frac{\dot{m} \Delta P}{\rho} \quad 5.88$$

Thus; the required pumping power can be determined as follows;

5.8.1 Pressure Drop in the Collector

Pressure drop in the collector is the sum of pressure drop in the riser, header, fittings and bending's.

Fig.5.6 indicates the flow configuration of inside the riser and header of the collector.

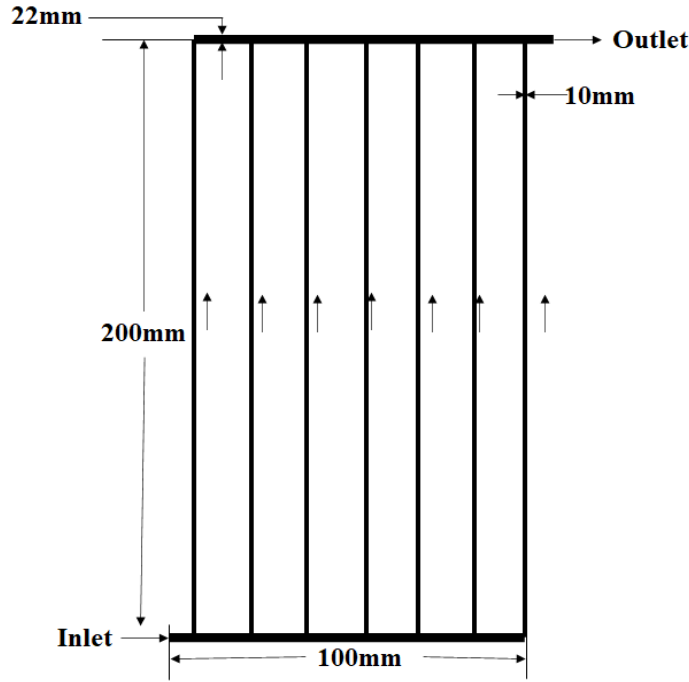


Fig 5.6 Flow configuration in the collector tubes

i. Riser pressure drop

Assuming uniform flow inside each tube volume flow rate in each tube can be determined by;

$$\dot{Q}_i = \frac{\dot{Q}}{7} = 2.977 * 10^{-6} m^3/sec$$

$$\text{Average velocity in the riser } V_{avr} = \frac{\dot{Q}_i}{A_r} = 0.037 m/sec$$

$$Re = \frac{\rho \cdot V_{avr} \cdot d}{\mu} = 461.38 < 2300 \text{ implies laminar flow}$$

Thus, for laminar flow $\Delta P = f \frac{\rho V_{avr}^2 L}{2d}$ and $f = \frac{64}{Re}$

$$\Delta P_r = 11.2 pa$$

ii. header pressure drop

Volume flow rate in side header tube $\dot{Q}_o = \frac{\dot{Q}}{2} = 1.042 * 10^{-5} m^3/sec$

$$\text{Average velocity in the header } V_{avh} = \frac{\dot{Q}_o}{A_h} = 0.027 m/sec$$

$$Re = \frac{\rho \cdot V_{avh} \cdot D}{\mu} = 740.7 < 2300 \text{ implies laminar flow}$$

Thus, for laminar flow $\Delta P_h = f \frac{\rho V_{avh}^2 L}{2D}$ and $f = \frac{64}{Re}$

$$\Delta P_h = 1.424pa$$

For the two headers

$$\Delta P_H = 2.85pa$$

- iii. pressure drop inside the bending and fittings

Fig.5.6 shows that the collector tubes have 14 bending's, pressure loss for fittings can be determined using;

$$\Delta P_f = K_f \frac{\rho V_{av}^2}{2} \quad 5.89$$

Where K_f is pressure loss factor

V_{av} is the average velocity in the tube upstream fittings

For right angle bend, laminar flow $K_f = 0.9$

$$\Delta P_f = 6.418pa$$

- iv. pressure drop due to the inclination of collector

The longest leg of the collector exceeds the shortest leg by 0.4 m. Thus, $\Delta P_{inc} = \rho gh = 3876.95pa$

$$\begin{aligned} \text{Therefore, } \Delta P_c &= \Delta P_r + \Delta P_H + \Delta P_f + \Delta P_{inc} \\ &= 3897.41pa \end{aligned} \quad 5.90$$

5.8.2 Pressure Drop in LHS System

The LHS system had 5 HTF tubes with 800mm length, 16mm diameter inside a 1000mm length and 200mm shell. Volume flow rate inside the LHS can be determined by assuming uniform flow inside the HTF tubes.

$$\dot{Q}_{LHS} = \frac{\dot{Q}}{5} = 4.168 \times 10^{-6} m^3/sec$$

$$V_{avLHS} = \frac{\dot{Q}_{LHS}}{A_t} = 0.02m/sec$$

$Re = \frac{\rho V_{avLHS} \cdot d}{\mu} = 412.8 < 2300$ laminar flow and the pressure drop developed can be estimated as;

$$\Delta P_{LHS} = f \frac{\rho V_{avLHS}^2 L}{2D} \quad 5.91$$

$$\Delta P_{LHS} = 15.4pa$$

5.8.3 Pressure Drop in Connections Fitting and Bending's in the Whole System

The length of total connecting tubes is 8meter 25mm diameter with 12 right angle bending's. Thus, the pressure drop developed can be estimated by assuming uniform flow inside the pipes;

$$\Delta P_{con} = f \frac{\rho V_{av}^2 L}{2D} + K_f \frac{\rho V_{av}^2}{2} \quad 5.92$$

$$\Delta P_{con} = 12.1\text{pa} + 4.83\text{pa} = 16.83\text{pa}$$

$$\text{Hence, the total pressure drop of the system } \Delta P_{tot} = \Delta P_c + \Delta P_{LHS} + \Delta P_{con} = 3929.64\text{pa}$$

$$\text{Allowance factor of 1.5 is given for reliability of the pump } \Delta P_{tot} = 5894.46\text{pa}$$

$$\text{Total specific work } Y = \frac{\Delta P_{tot}}{\rho} = 5.96\text{m}^2/\text{s}^2$$

$$\text{Total dynamic head TDH} = \frac{Y}{g} = 0.61\text{ m}$$

$$\text{Total volume flow rate } \dot{Q} = 2.084 \times 10^{-5} \text{m}^3/\text{sec} = 0.075\text{m}^3/\text{hr}$$

Therefore, a proper pump with requirement of head more than 0.61m and flow rate 0.075m³/hr can perform the required pumping effect. In the market the least power of pump(centrifugal) available is 0.5hp. But it was costly with fixed motor speed. For the present indirect solar cooking system, the available electrically driven drum pump with the following specification were selected.

Table 5.7 Specification of selected pump

Pump specification		Motor specification	
Normal speed	1000rpm (adjustable)	Frequency	50hz
Flow rate	100L/min	Voltage	220V
Shaft power	700w	Speed	1000rpm
Lift distance	8m	Rated power	0.8kw
Efficiency	50%	Electric current	4A

5.9 Numerical Modeling of Modified FPC

In order to analyze the performance of the collector, computational fluid dynamics (CFD) simulation analysis has been carried out using ANSYS FLUENT software for the modified flat plate under similar geometrical and flow conditions used in the experimental investigation. As the analysis of solar flat plate collectors depends on the flow distribution in the riser tube, the numerical model considers single riser tube attached on the center of sectioned absorber plate.

5.9.1 Model Description and Meshing

The 3D model consisting of absorber plate and circular riser tube with inlet and outlet at the center of the absorber plate was developed using SOLIDWORK as shown in Fig.5.7. The Absorber plate of 2m length, 0.125m width and 0.0012m thickness was used. A circular riser tube made of copper with 10 mm internal diameter and 1mm thickness of is attached to the center of the absorber plate. Meshing of the flow model was carried out using ANSYS CFD software. The meshed view of the model is shown in Fig.5.8. The generated mesh consists of 490809 nodes and 406448 of elements.

5.9.2 Grid Independency Test

Grid independency test was carried out for the developed model to decide the mesh size and study its effect on the outlet temperature of HTF. Fig.5.9 shows the variation of HTF temperature for different mesh counts. The test was performed using fine, medium and coarse mesh sizes.

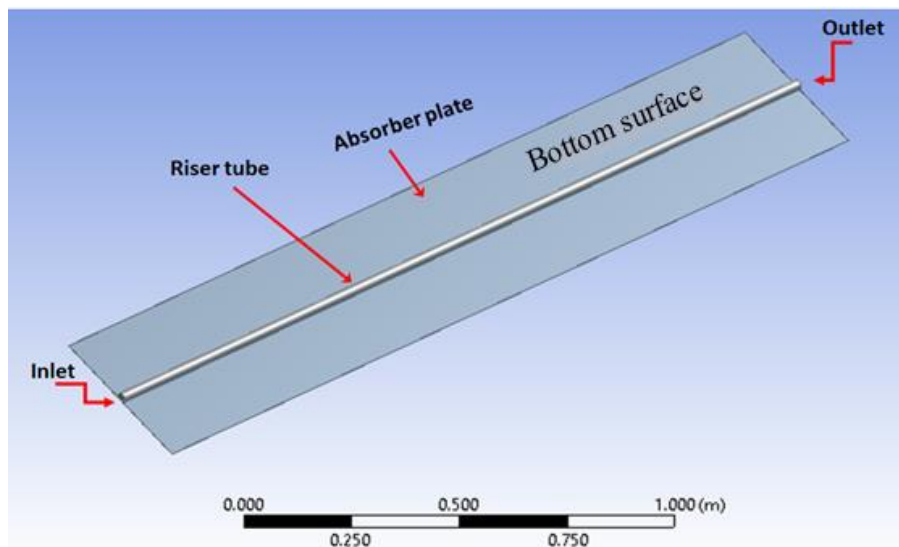


Fig 5.7 Model geometry of absorber and riser tube

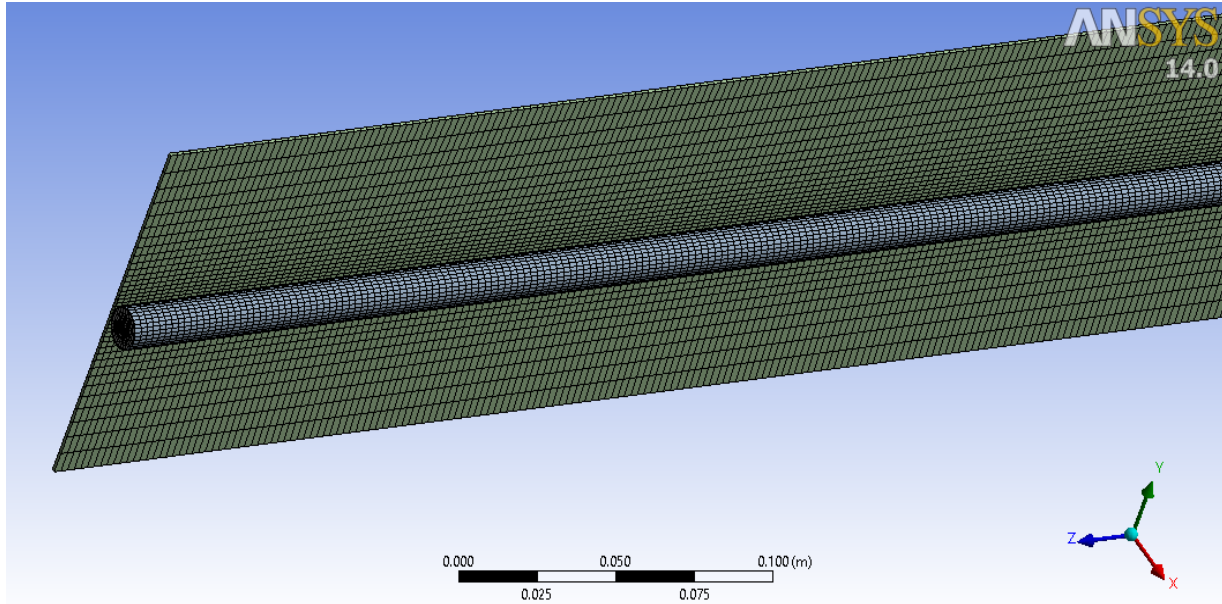


Fig 5.8 Model geometry of absorber and riser tube

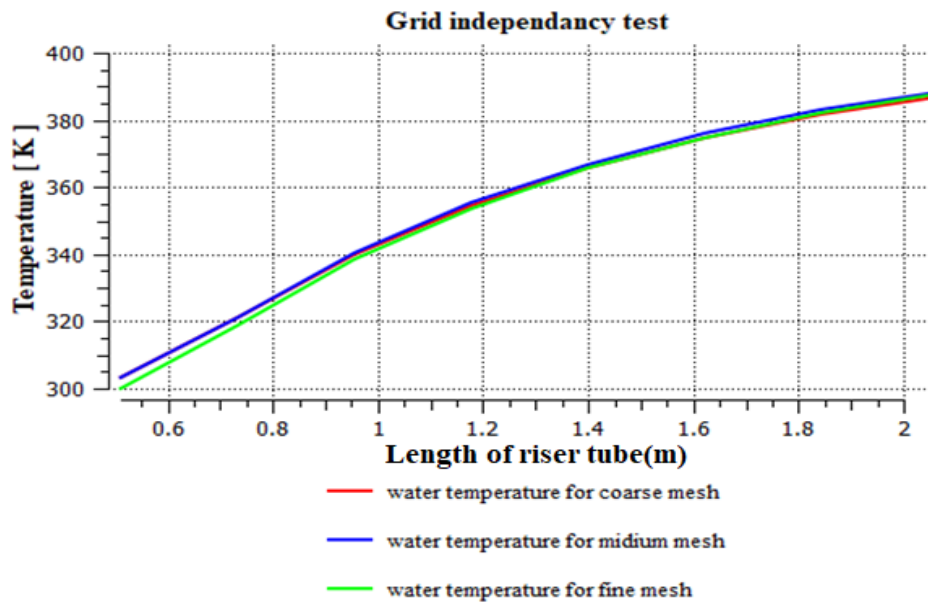


Fig 5.9 Grid independency test

5.9.3 Governing Equations

The following assumptions were taken while formulating the governing systems of the modified flat plate solar collector system.

- (i) The density, specific heat and thermal conductivity are considered to be constant for the HTF.
- (ii) The flow is natural steady flow of HTF under laminar flow condition.
- (iii) The fluid is a continuous medium and incompressible.

The mathematical equations that governs the physical phenomenon of the incompressible flow through the annular of riser tube are given below:

- i. Continuity equation; the general continuity equation is given as;

$$\nabla \cdot \rho \vec{V} + \frac{\partial \rho}{\partial t} = 0 \quad 5.93$$

For one directional incompressible laminar flow of HTF inside the riser tube, the equation can be simplified as;

$$\frac{\partial(\rho_{HTF} \cdot u_j)}{\partial x_j} = 0 \quad 5.94$$

Where the i and j represents direction unit vector along the x (longitudinal) and y direction respectively.

- ii. Momentum (Navier-Stoke's) equation; the general momentum equation for incompressible flow can be written as;

$$\nabla \cdot (\rho \vec{V} \vec{V}) = -\nabla p + \nabla \cdot (\tau) \quad 5.95$$

Using the above assumptions equation 5.86 can be simplified in component-wise (y and z components) as;

$$-\frac{1}{\rho} \frac{\partial p}{\partial y} = 0 \quad \text{and} \quad -\frac{1}{\rho} \frac{\partial p}{\partial z} = 0$$

Thus, the x-component (fluid flow direction) of the Navier-Stoke's equation becomes

$$\rho_{HTF} \frac{\partial(u_i u_j)}{\partial x} = -\frac{\partial p}{\partial x} + \frac{\partial(\mu_{HTF} \frac{\partial u_i}{\partial x_j})}{\partial x_j} \quad 5.96$$

- iii. Energy equation is derived using boundary layer approximation that temperature gradient along the flow direction is relatively small compared to its perpendicular direction i.e.

$$(\frac{\partial T}{\partial x_j} \gg \frac{\partial T}{\partial x_i}). \text{and } \frac{\partial u_j T}{\partial x_j} = \alpha_{HTF} (\frac{\partial^2 T}{\partial x_j^2}) \quad 5.97$$

5.9.4 Computational Procedure

- i. **Boundary conditions**

Various boundary conditions used in modified flat plate collector configuration are given as;

- HTF Inlet conditions

HTF inlet Velocity inlet - 0.03m/sec

Temperature - 303K

- HTF outlet conditions

Pressure outlet - P_{atm}

➤ Boundary surfaces

Absorber top surface; subjected to heat flux = average $I_{tot-col}$ of the day

Absorber bottom surface and side walls; wall boundary condition with specified convective heat transfer coefficient and freestream temperature

➤ Interface boundary condition the interface between the absorber and riser tube is treated like a wall having coupled heat transfer condition.

ii. Numerical treatment

ANSYS CFD software was used to solve the governing equations using implicit pressure-based solver celled fluent 7. Those equations were solved under laminar steady flow condition with SIMPLE pressure-velocity coupling algorithm. Spatial discretization was carried out using Green-Gauss cell-based gradient and second order upwind scheme for energy and momentum. The simulation uses standard solution initialization method. Residual factor of 10^{-6} is given for the solution to converge.

5.10 Numerical Modeling of Latent Heat Storage System

Before designing of an experimental prototype numerical modeling is usually applied for the analysis of complex heat transfer characteristics of both the HTF and PCM. This section presents the model description, meshing, governing equation and computational procedure used for the numerical analysis of LHS.

5.10.1 Model Discription and Meshing

The sectional view of the 3D shell and tube type storage system filled with heat transfer medium in the shell and water in the HTF tube as shown in Fig.5.10. The developed model consists of five steel tubes inside the cylindrical shell containing the PCM. As the model is symmetric about x-axis, only one fourth of the model was taken for the investigation. The LHS shell has a diameter of 200mm, and 800mm high and the HTF tube has 16mm diameter, 800mm length, and 1mm thickness. Commercial grade stearic acid was used as a PCM. The meshing of LHS model were performed using COMSOL Multiphysics software. Physics controlled free tetrahedral and triangular meshes are used for meshing the domains and boundaries as shown in the Fig.5.11. The mesh has 51441 number of elements.

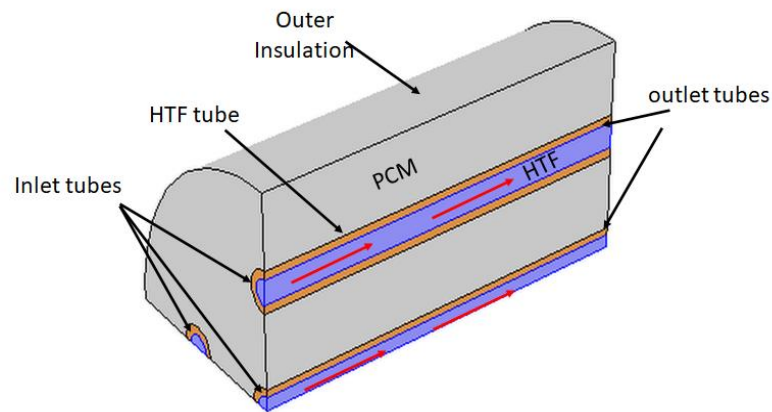


Fig 5.10 Numerical LHS model

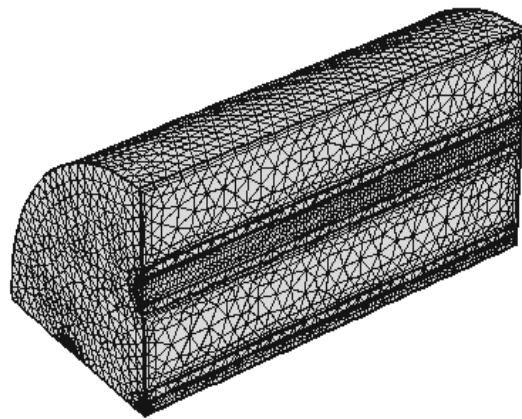


Fig 5.11 Physics controlled meshed part of LHS model

5.10.2 Governing Equations

While simulating the physical model of the latent heat storage system, the following assumptions were made;

- HTF flow is laminar and negligible viscous dissipation,
- HTF is incompressible and Newtonian,
- Phase change during melting and solidification occur only within melting range
- The initial temperature of the PCM is uniform,
- PCM is homogenous and isotropic.
- The heat losses through the outer wall are negligible

The model developed using COMSOL Multiphysics software was solved by the coupled conjugate heat transfer problems (CCHTP) which simultaneously solve both the flow behavior of the HTF and PCM. The problem associated with the incorporation of latent heat in the sensible heat-based model

was solved by implementing the equivalent heat capacity (EHC) method, which comprises both the specific heat and latent heat of the PCM in a single term called EHC.

EHC method was used for the first time for solving this kind of numerical problem by Bonacina *et al.*, (1973). The modified heat capacity of the PCM and the effective heat capacity are obtained from the expression (Dawit., 2018).

$$C_p = \begin{cases} C_{p.S} & \text{for } T < T_S \\ C_{p.EFF} & \text{for } T_S \leq T \leq T_L \\ C_{p.L} & \text{for } T > T_L \end{cases} \quad 5.98$$

$$C_{p.EFF} = \frac{C_{p.S} + C_{p.L}}{2} + \frac{L_F}{T_L - T_S} \quad 5.99$$

where subscript S and L stands for solid and liquid, respectively.

In a thermal storage system using LHS the phase change occurs within a temperature interval not at specific point. Problems of this type often referred mushy region problems. Such discontinuous problems of heat capacities will be solved by implementing COMSOL Multiphysics software using smoothed Heaviside function with a continuous second derivatives. In the analysis of LHS model, general continuity, momentum and energy equations are applied with a modification of momentum equation to include the effect of buoyancy ($\vec{F}$) and Darcy law source term ($\vec{S}$). Such modifications are helpful to simplify the complexity of Navier-stokes equations and are called Boussinesq-approximation which was obtained using the expression given by (Niyas., 2017);

$$\vec{F} = \rho \vec{g} \beta (T - T_m) \quad 5.100$$

$$\vec{S} = \frac{(1-\theta)^2}{\theta^3 + \varepsilon} A_{mushy} \vec{V} \quad 5.101$$

Where β is the thermal expansion of the PCM,

A_{mushy} is mushy zone constant is defined as the transition of velocity in the mushy region.

The mostly recommends value for the mushy zone constant is in the range $10^3 - 10^7$.

ε is constant value of 0.01

$\vec{V}$ is the velocity of the PCM

5.10.3 Boundary Conditions

(i) During charging mode

$$@t = 0 \begin{cases} V_{inlet} = 0; \\ \text{All domains (HTF, PCM, HTF - tube) are at constant lower temperature } T_{initial} \end{cases}$$

$$t > 0 \begin{cases} V_{inlet} = constant \\ HTF \text{ at higher temprature } T_{inlet} \end{cases}$$

(ii) During discharging mode

$$@t = 0 \begin{cases} V_{inlet} = 0; \\ All \text{ domains } (HTF, PCM, HTF - tube) \text{ are at constant higher temprature } T_{initial} \end{cases}$$

$$t > 0 \begin{cases} V_{inlet} = constant \\ HTF \text{ at lower temprature } T_{inlet} \end{cases}$$

(iii) Adiabatic condition for all the outer surface to avoid heat loss

(iv) No-slip boundary condition for the tube wall to avoid zero velocity at the walls

5.10.4 Computational Method

COMSOL Multiphysics 4.3a was used to solve the governing equations using Galerkin finite element discretization method. The EHC method made the system of equations to be solved nonlinear. A time-dependent solver, PARDISO (Niyas, 2017) was used to solve the equations. For both charging and discharging simulations, the time stepping was done based on the backward differentiation formula (BDF). In BDF, there is an initial minimum time step and a maximum time step. The time stepping is dynamic and it varies based on the error value of the previous iteration. Due to the highly nonlinear nature of the problem, the solution of the numerical model diverges for the convergence criterion of any value smaller than 10^{-3} . Hence, the convergence criterion for temperature and velocities is set to 10^{-3} .

CHAPTER SIX

EXPERIMENTAL SETUP AND PROCEDURE

6.1 Introduction

This section presents the conceptual model, working principle, manufacturing process and the whole experimental setup and procedure followed while testing the prototype of indirect solar cooking system. The conceptual model of the system was developed using SOLIDWORKS 2016. The working principle of the system is elaborated in detail. The manufacturing process used for preparing the prototype is presented with descriptive photos of the whole process. The experimental setup and the measuring and testing devices used are also presented. Finally, procedures followed while measuring, testing and investigating of the various useful parameters of the system are discussed.

6.2 Conceptual Model of Indirect Solar Cooking System

Fig.6.1 illustrates the schematic diagram of indirect type modified flat plate collector based solar cooker drawn using SOLIDWORKS 2016. The system consists of a flat plate solar collector with four side reflector, LHS unit, cooking chamber, frame and support and closed loop piping network as connection for the whole system.

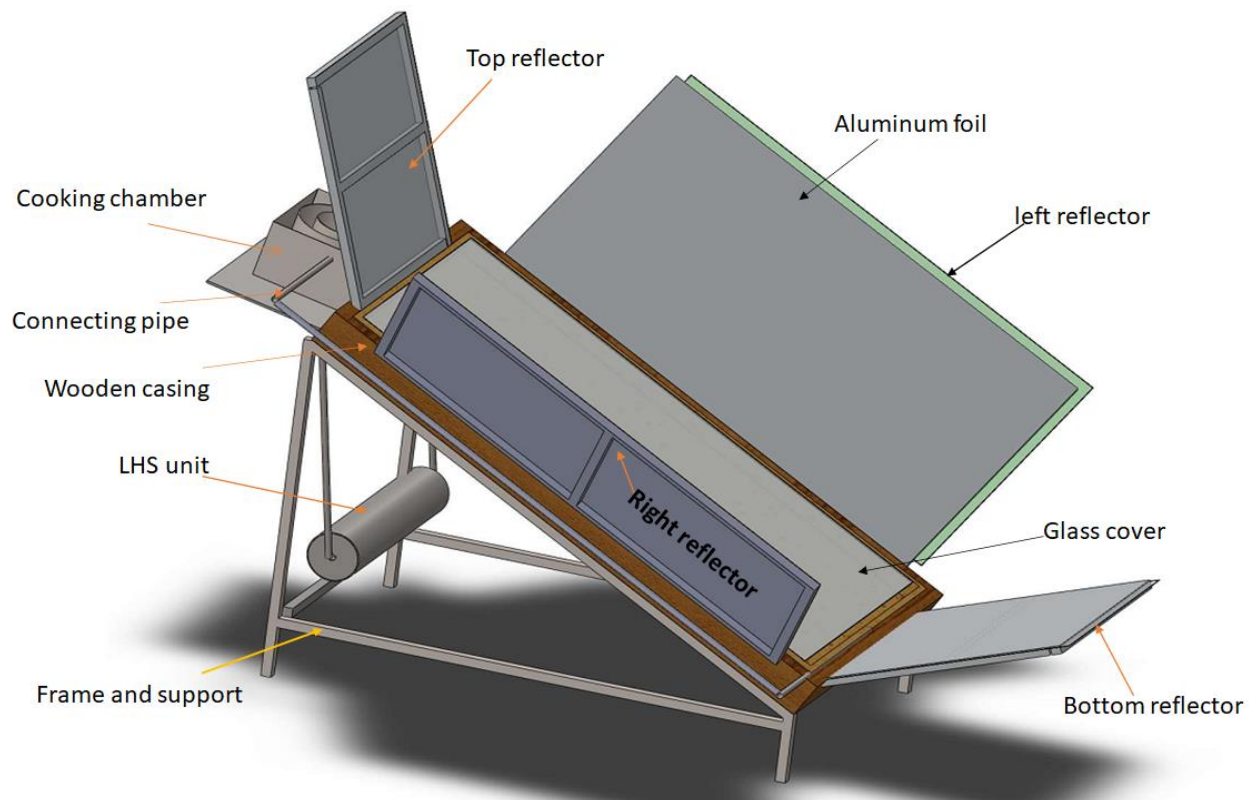


Fig 6.1 Conceptual schematic of indirect solar cooker

6.3 Working Principle of the Present Indirect Solar Cooking System

The main concept behind solar cooking application is being able to change the sun's solar radiation into a useful form of energy. There are three main principles of operation for all solar cookers: those are concentrating sunlight to some area of interest, converting the light to useful heat energy and trapping the heat by isolating the system from surrounding. The present system uses an indirect type of solar cooking system which consists of a modified FPC, PCM based shell and tube LHS unit and well insulated cooking chamber with storage vessel and cooking pot, pump, and closed loop piping network. In this system, the modified FPC collector will absorb and harness the sun's radiant energy and transfer it to the riser tubes through which the HTF flows. The riser tube transfers the HTF to the header tube and is conveyed with the connecting tubes to the shell and tube latent heat storage unit to melt the PCM in it for late off-sunshine operation.

Figure 6.2 describes the working principle of the whole indirect solar cooking system. During charging mode of the PCM, hot HTF from the collector outlet enters into LHS unit and pass through it into the cooking chamber. As the HTF from the outlet of the collector pass through the LHS unit it will release its heat to charge and store heat in the PCM. As the PCM changes its phase from solid to liquid latent heat will be stored. Whenever there is a requirement of hot water in the chamber as fast as possible it will pass through the bypass pipe and flows into the storage vessel. The storage vessel will store hot water and transfer its heat to the cooking pot at which the food is placed to be cooked. HTF from the outlet of cooking chamber will return to the collector using a pump installed in the connection during day time. When discharging process starts during off-sunshine, night time and late evening, the HTF from the storage vessel will flow through the LHS unit absorbing the latent heat stored and conveyed it back to the cooking chamber. As the PCM changes its phase back to solid state it will release latent heat to the HTF.

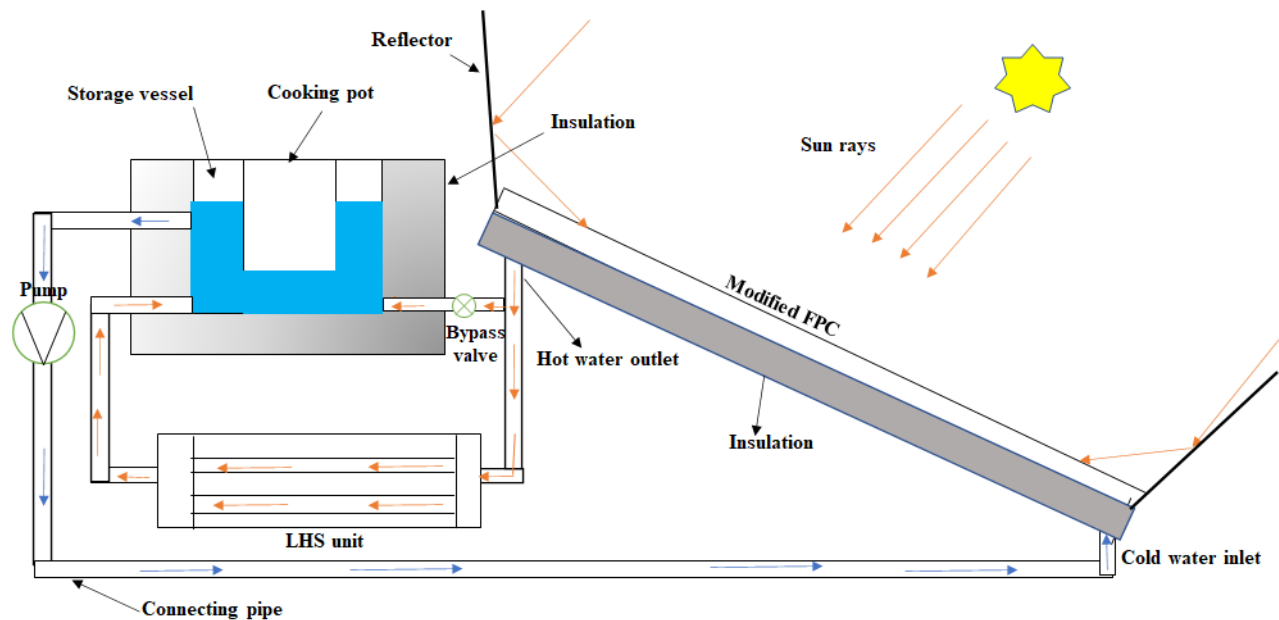


Fig 6.2 Schematic for the working principle of indirect solar cooker

6.4 Manufacturing Processes

This section describes the manufacturing process conducted while preparing the experimental prototype of the system.

6.4.1 Frame and Support

In order to have a rigid, movable structure and better service life, solar collectors should be mounted on a strong support system. The supporting structure were prepared from square metal pipe by joining end to end using welding process to hold the wooden casing of the collector as shown in Fig.6.3. The support structure is at an angle equal with the collector tilt angle of 25° . To do so, steel legs were welded on each corner of the rectangular frame with the two front legs shorter than the others so as to hold the collector inclined at the tilt angle. Angle frame made of steel were welded on the top of the supporting structure which is used as a casing for the wooden frame. To make the whole system movable, four wheels were welded at each legs of the supporting structure.



Fig 6.3 Supporting frame and structure of the prototype

6.4.2 Modified FPC

During the manufacturing of the solar collector, 7 commercial copper tubes with 10mm nominal diameter was cut using a special tube cutter in to 2m length each. To facilitate the flow of HTF through the collector, those tubes were arranged vertically parallel to each other as a riser tube with spacing of 0.125m between them. The riser tubes were brazed with the bottom and top end of two header tubes arranged horizontally and made of 22mm nominal diameter steel tube using drilling and using oxy-acetylene welding as shown in the Fig.6.4.

For ensuring better heat transfer between the absorber and HTF, the copper absorber plate of 1 mm thickness, 1m width and 2 m length many attempts were made to attach the riser tubes with the bottom surface of the absorber plate. The first trial was to braze riser tube with the absorber copper plate using oxy-acetylene welding and bronze as a filler material. In doing so the plate starts to shrink as it gets heated and riser tubes were unable to bond with the copper sheet. Although the filler metal was changed with cooper this trail was failed again. The second method was trying to use soldering but it was also unable to work as expected and quitted after one of the riser tubes is scarified beside it was so costly. The last chance left was to connect the riser tubes with the sheet using stainless wire and a magnificent result was obtained as the riser tubes were fitted and touched well with the absorber tube with no gap between. Actually, there is an expense of absorber area in the process of connecting the sheet with hole passage in the surface but there will be no bond resistance exist as the tube and absorber are joined directly with no intermediate interference.

After the dust is removed, anti-rust is painted on both sides of the absorber and its top surface was the painted with chalkboard black paint having high heat absorptivity property. To add more absorptivity the surface of the absorber was covered with a carbon smoke “soot” from incomplete combustion of diesel fuel.

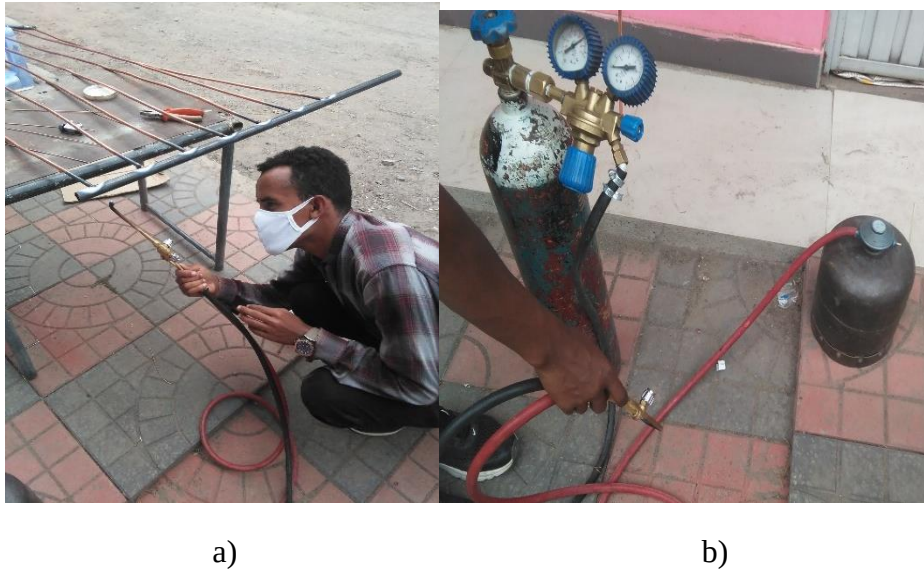


Fig 6.4 Oxy-acetylene welding of copper riser tubes with header tubes a) welding operation b) oxygen and acetylene cylinders

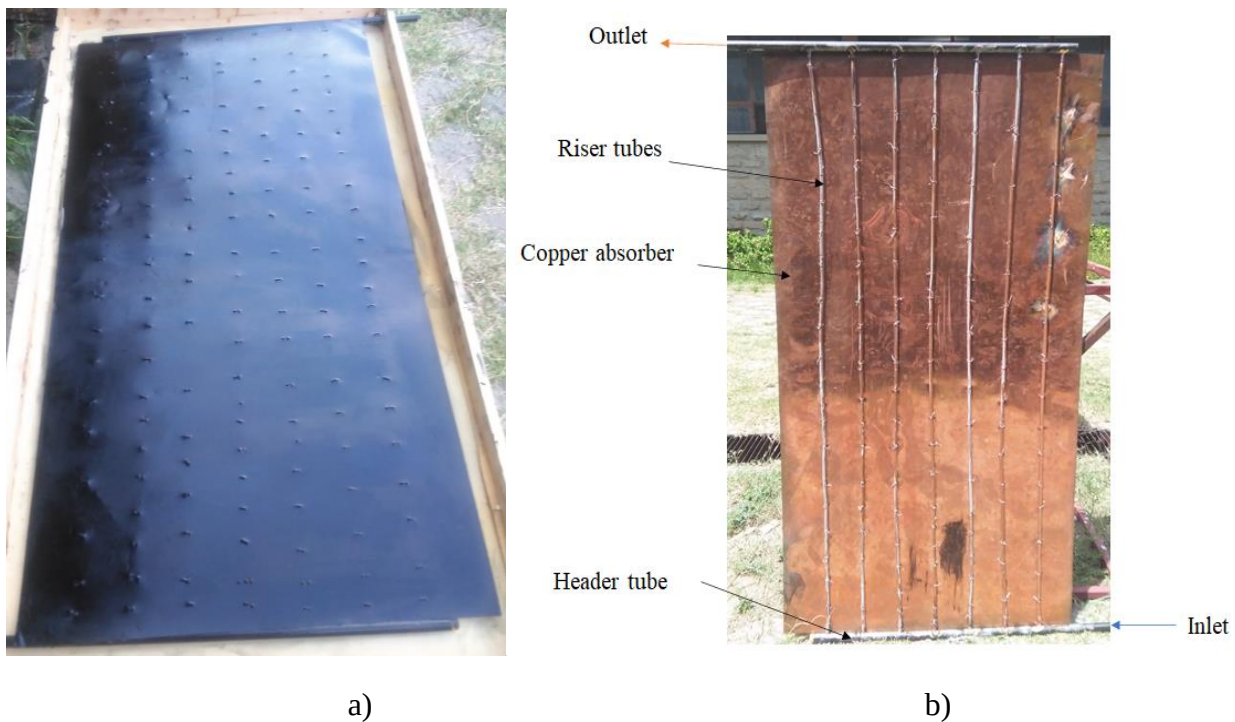


Fig 6.5 Absorber-riser assembly of the prototype a) black painted top surface b) bottom surface with riser and header tube

The absorber assembly was then incorporated in to a wooden casing having 2.02mx1.02m cross-section as shown in Fig.6.6. To reduce conductive and convective heat loss from the absorber, the bottom and back side of the wooden casing were insulated with 2,5cm and 1.5cm thermo-foam insulation respectively.



Fig 6.6 Photograph of collector a) wooden casing with back thermo-foam insulation and b) FPC with cover glass

Fig.6.6 (b) indicates the setup FPC covered with 5mm thick ordinary glass. Although tempered glass is far more suitable for use as a glazing, ordinary glass was used because it was easily available and less expensive. To reduce the convective heat loss from the plate surface optimum air gap of 30mm were maintained between the absorber and the glass cover. To enhance the amount of solar insolation falling on the solar collector, four plane reflectors were mounted on each side of the collector. Plane reflectors were made from a wooden frame and wooden sheet. Fig.6.7 indicates the reflector-collector arrangement of the prototype. To increase the reflectivity of solar radiation, the top surface of all the plane reflector was covered with an aluminum foil. For optimal collection of solar radiation, manual adjusting mechanism is attached with the back of the reflectors and hinged with the supporting frame as indicated in Fig.6.8.



Fig 6.7 Collector with four reflectors in all the sides



Fig 6.8 Manual adjusting mechanism for the reflectors

6.4.3 The Latent Heat Storage Unit

The shell of the LHS unit was made of 628mmx1000mm mild steel sheet metal after it is rolled into cylindrical shape of 200mm diameter using rolling machine as shown in Fig.6.9 (a). Five HTF tubes with 0.8m length were made of steel tube. The two ends of the HTF tubes were welded with the circular plate of 200mm diameter. The HTF tubes were inserted into the shell and the two ends of the shell were welded and closed with circular plate of 200mm diameter as shown in in Fig.6.9(b). HTF

inlet and outlet channel were formed and a hole is drilled on the outer surface of the shell to fill it with PCM. In order to prevent leakage polyester putty mixed with hardener is filled on the possible opening that may exist of shell and both sides of the tube inlet and outlet connections as indicated in Fig.6.10.

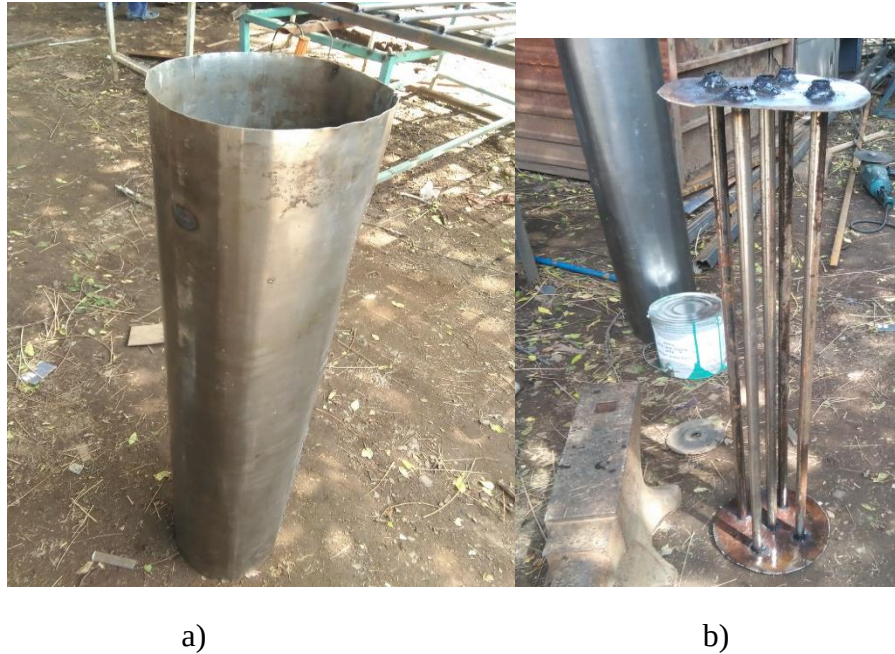


Fig 6.9 Manufacturing of shell and tube LHS a) shell b) HTF tubes



Fig 6.10 Filling the leakage opening using polyester putty

6.4.4 The Cooking Chamber

The cooking chamber consists of indoor cooking unit having rectangular sheet metal box made of mild steel with base dimension of 50cm length, 34cm width and 25cm. To facilitate the flow of HTF,

two holes were drilled on the side and bottom side of the box. Side hole were used to connect the outlet pipe of the collector with the inlet pipe of the cooking unit whereas the bottom hole used as a return pipe connection. The box was fitted with a cover in which circular hole is formed for the cooking and storage unit. Two concentric cylindrical vessels with 350 and 200mm diameter were fitted inside the hole of the box. The inner vessel is used as a cooking pot whereas the outer vessel used as a storage for storing and a passage through which the heat transfer fluid passes. The cooking pot was equipped with a cleaning pipe and cleaning valve. To reduce heat loss from the cooking unit, the remaining space between the concentric vessels and inner box were filled with insulating material. Fig.6.11 shows the cooking chamber assembly with storage vessel, cooking pot, outer box casing and insulation.

6.4.5 Connection and Other Fitting Units

Connecting tubes which is used to connect the closed circuit were made of 25mm Raktherm polyester tubes. Those tubes are selected as they are easy to connect, available, corrosion resistant, and can withstand pressure up to 80bar and temperature up to 120⁰c. Fig.6.12 indicates the connecting tubes insulated with thermo-foam to reduce heat loss.

6.4.6 Preparation Shell and Tube LHS System

In order to fill the LHS storage with PCM, stearic acid was melted using an electric stove. Its melting point was measured using metallic tip digital thermometer as depicted in Fig.6.13. When fully melted, it was then poured in to the shell of LHS though the PCM filling hole as shown in Fig.6.14.

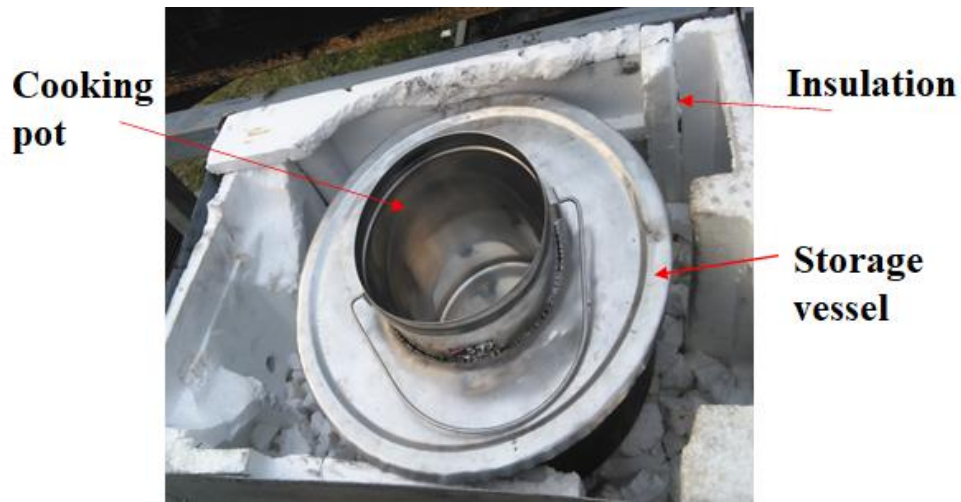


Fig 6.11 Preparation of cooking chamber



Fig 6.12 Connecting tubes of closed the loop



Fig 6.13 Melting and pouring process of stearic acid in to shell and tube storage unit.

The photograph of the complete setup for LHS unit with its components and thermocouple positions is depicted in Fig.6.14. The inlet and outlet water temperature from the LHS and central and top

surface temperature of the stearic acid were measured with four T-type thermocouples (T1, T2, T3, and T4) fixed at the inlet, outlet, center and at equ-distance from center to shell.



Fig 6.14 Setup of LHS unit and positions of thermocouples

6.5 Experimental Setup

For experimentally investigating the present indirect solar cooking system, the setup was installed in the test field of ASTU, mechanical engineering department, Adama Ethiopia. The complete experimental setup of indirect solar cooker consists of modified FPC tilted at 25° , shell and tube LHS unit, cooking chamber which consists of the cooking vessel and storage vessel and a closed loop pipe network having HTF (water) used as a connection for the whole system components. Fig.6.15 shows the photograph for the complete setup of indirect solar cooker. In the setup the heat source for the cooker is obtained from the modified FPC using water as HTF which is circulated with the help of drum pump connected between the collector and the LHS unit.

The pump provides enough pressure to overcome the pressure drop in the circuit. To get variable flow rate in the system the speed of the pump rotor is controlled using speed controller of the pump. The measuring devices used in the experiment includes well calibrated thermocouple thermometer based on the procedure described in **APPENDIX B**, 10 K-type thermocouples fixed at different locations

of the prototype to measure temperature at the collector, LHS, and cooking chamber. Fig.6.16 shows the photograph of thermometer and thermocouple wires used for temperature measurement.

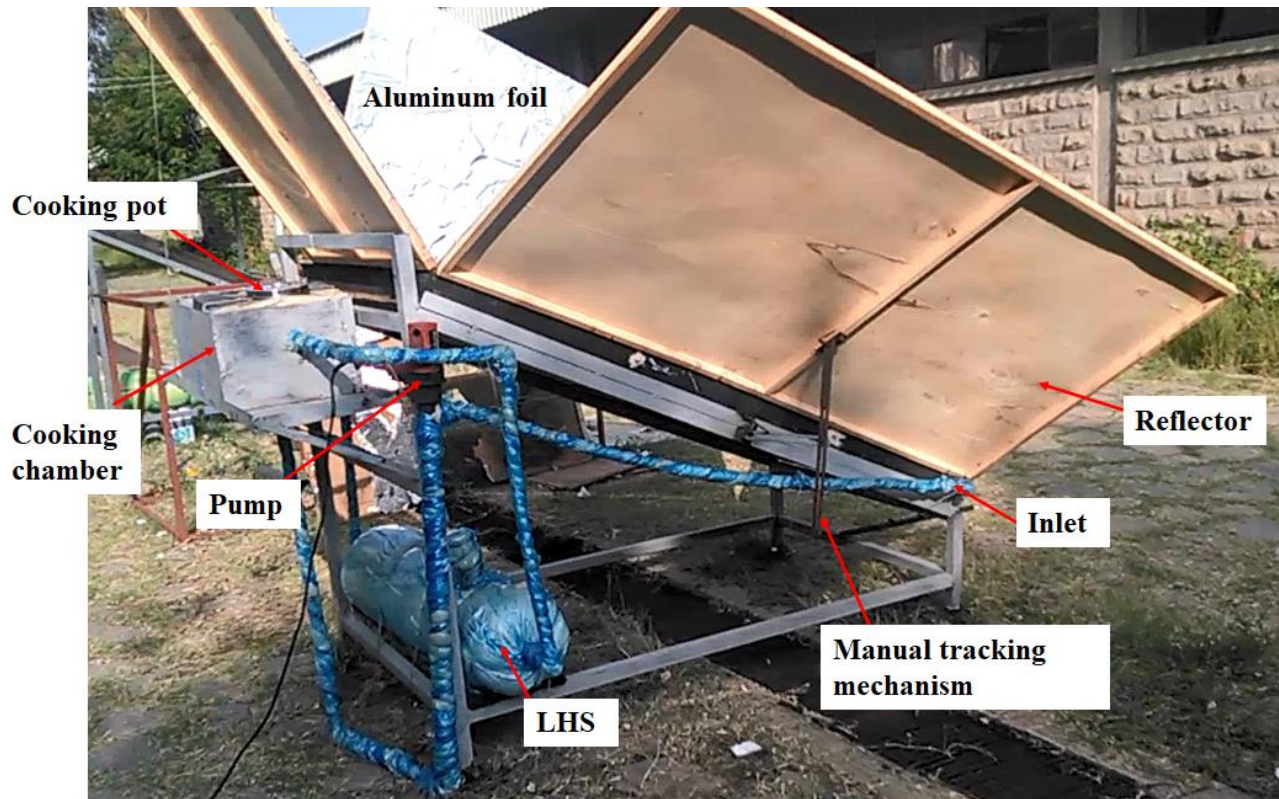
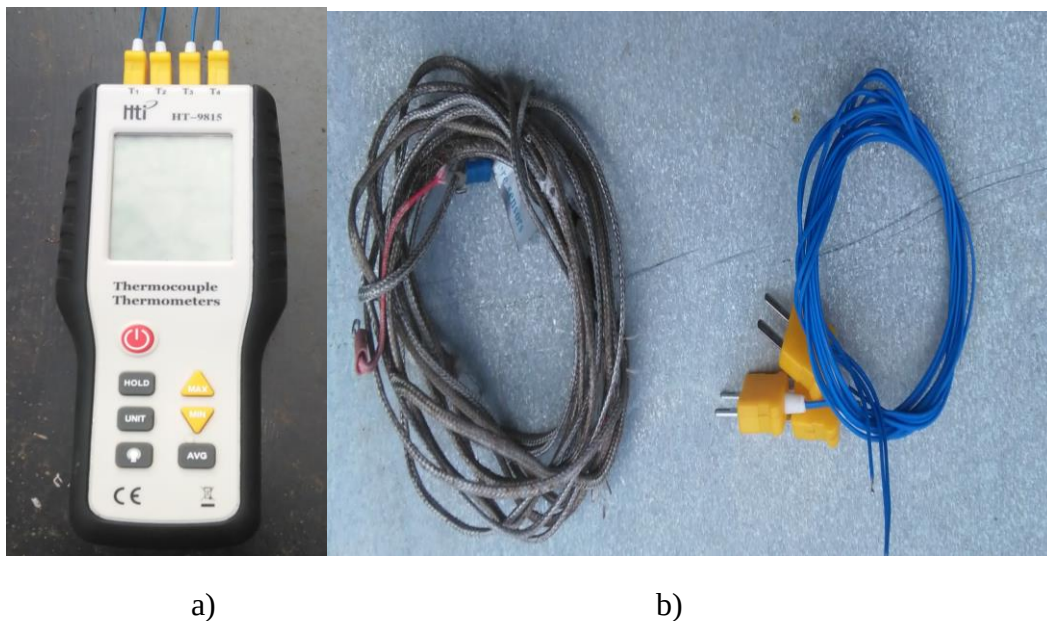


Fig 6.15 Complete experimental setup of indirect solar cooker



a)

b)

Fig 6.16 Temperature measuring devices a) Thermometer b) Thermocouple wires

The experimental study using indirect solar cooking system was carried out for different setups.

- i. Experiment performed with a solar cooker without reflector

ii. Experiment performed with a solar cooker having side reflector

Under those two categories, different experimental tests were performed for the solar collector system, thermal storage system and cooking chamber. Before filling the system with water leakage test in each component and forced pipe cleaning were performed. In the experiment, temperature at different components were measured and recorded. To measure the inlet and outlet temperature of water from the FPC, two K-type thermocouples were fixed at the inlet and outlet of the collector. The temperature of absorber plate was measured using 2-k-type thermocouples at different location of the plate and averaged to get the mean plate temperature. Other than four thermocouples fixed at different locations to measure the temperature of LHS another two thermocouples were fixed at the storage vessel and cooking pot.

CHAPTER SEVEN

RESULTS AND DISCUSSIONS

7.1 Introduction

This chapter presents the results obtained from the numerical and experimental analysis of the modified FPC, LHS unit and cooking unit under different conditions. The first section presents the results of numerical simulation made on the FPC using CFD ANSYS fluent and numerical investigation of LHS unit using finite element based COMSOL Multiphysics software. The next section presents the experimental investigations made on each components of the indirect solar cooker and its validated with the numerical one. The results of no-load, boiling, cooking and performance tests performed on the cooking chamber is also presented and discussed. Finally, the economic analysis of the solar cooker is presented.

7.2 Numerical Results of Modified FPC

The input for the numerical simulation of modified flat plate collector were obtained from the estimation of daily and hourly solar radiation data, geometrical analysis of collector-reflector relation, the temperature of tap water as an inlet HTF temperature and inlet velocity of HTF. This data was imposed to the ANSYS fluent software and excel spread sheet to obtain the following results.

7.2.1 Effect of Reflectors on Total Solar Radiation Incident on a Tilted Collector

Fig.7.1 indicates the effect of adding reflectors on the solar radiation incident on the collector for month January. The predicted result shows adding reflectors had significant change on the amount of solar radiation incident on the absorber surface. The daily average enhancement in the incident solar radiation on the collector with addition of reflector was found to be 42.3%. Hussein *et al.*, (2008) reports an enhancement of 36% with only two reflectors on top and bottom of the collector. The effect of reflectors is also shown in the next successive months of the years. Fig.7.2 illustrates the effect of adding a reflector in the amount of solar radiation incident on the collector for January, February and march.

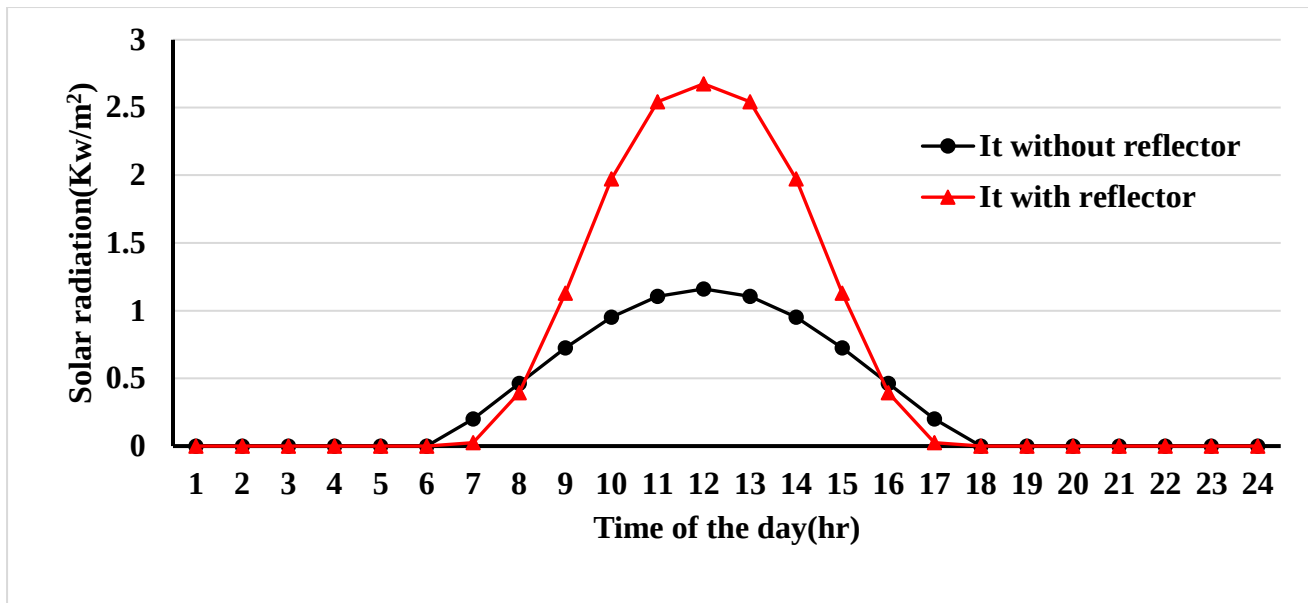


Fig 7.1 Effect of reflectors on solar radiation

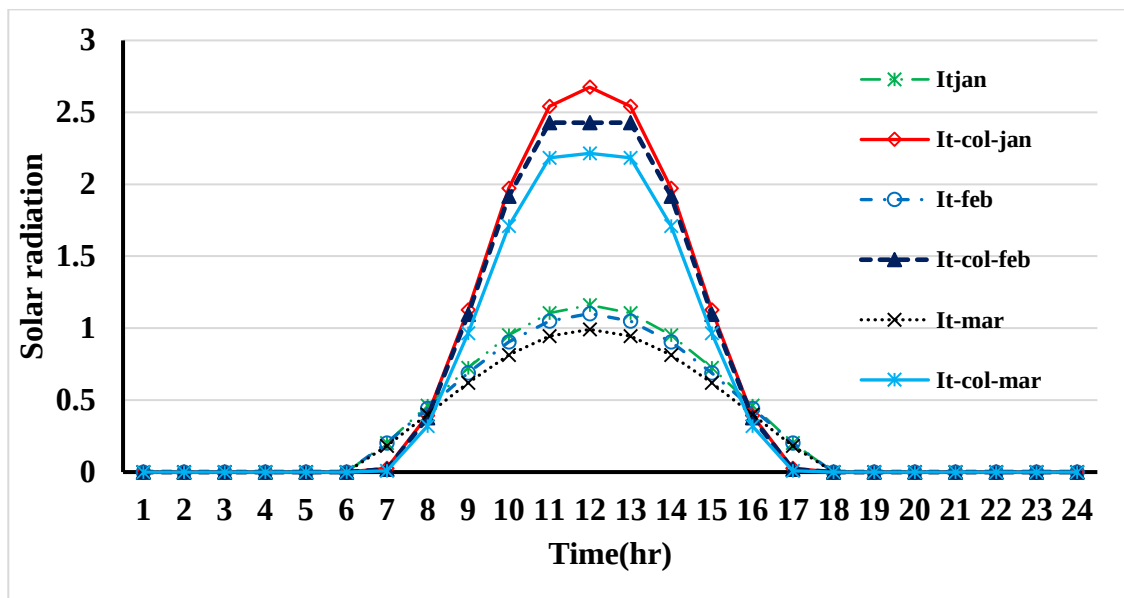


Fig 7.2 Effect of adding a reflector in the collector for 3 successive months of a year

The contribution of each reflectors in the total incident solar radiation on the collector for June month is depicted in Fig.7.3. From the predicted result, it was found that the effect of the bottom reflector was nearly twice larger than from the top one. Contribution of the right-side reflector to the solar radiation is identical to that of the left side reflector because of their assumption of symmetrical positions in respect to the collector. Almost the same result was reported by Zoran and Ljiljana, (2015) using 4 reflectors covered with aluminum foil.

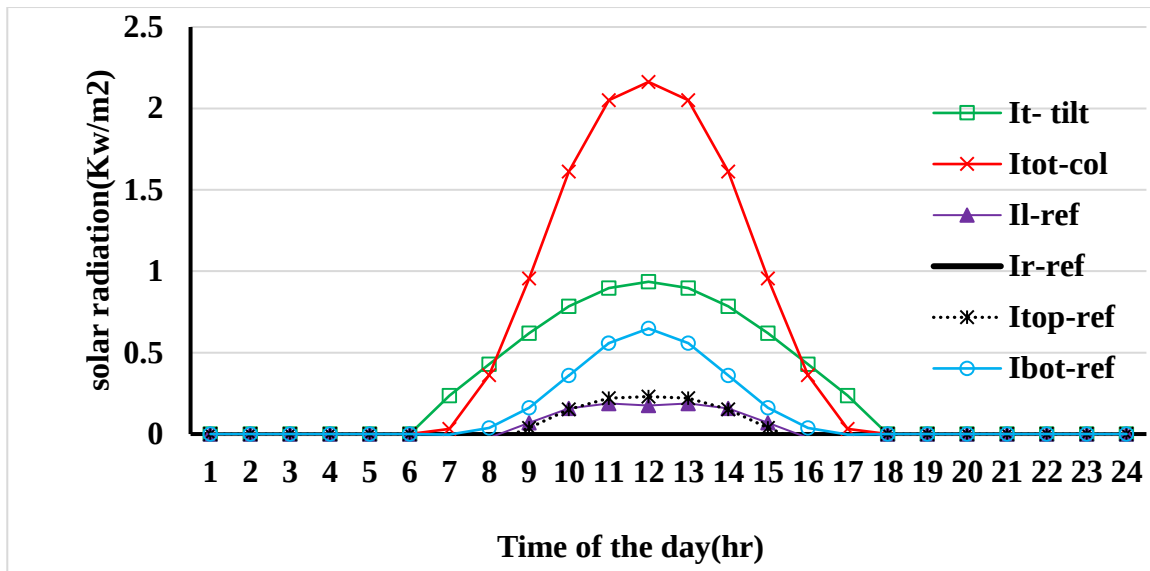


Fig 7.3 Contribution of each reflectors on the incident solar radiation

7.2.2 Effect of Reflectors on the Instantaneous Variation of Outlet Temperature of Water

The main aim of adding a reflector is to increase the operating temperature of the system as it is intended for solar cooking application. Thus, to get a higher operating temperature at the cooker section the outlet temperature of the collector should be higher. Fig.7.4 depicts the effect of adding a reflector on the outlet temperature of water from the collector section. Maximum outlet temperature of 388K is observed at solar noon as the solar insolation is very high.

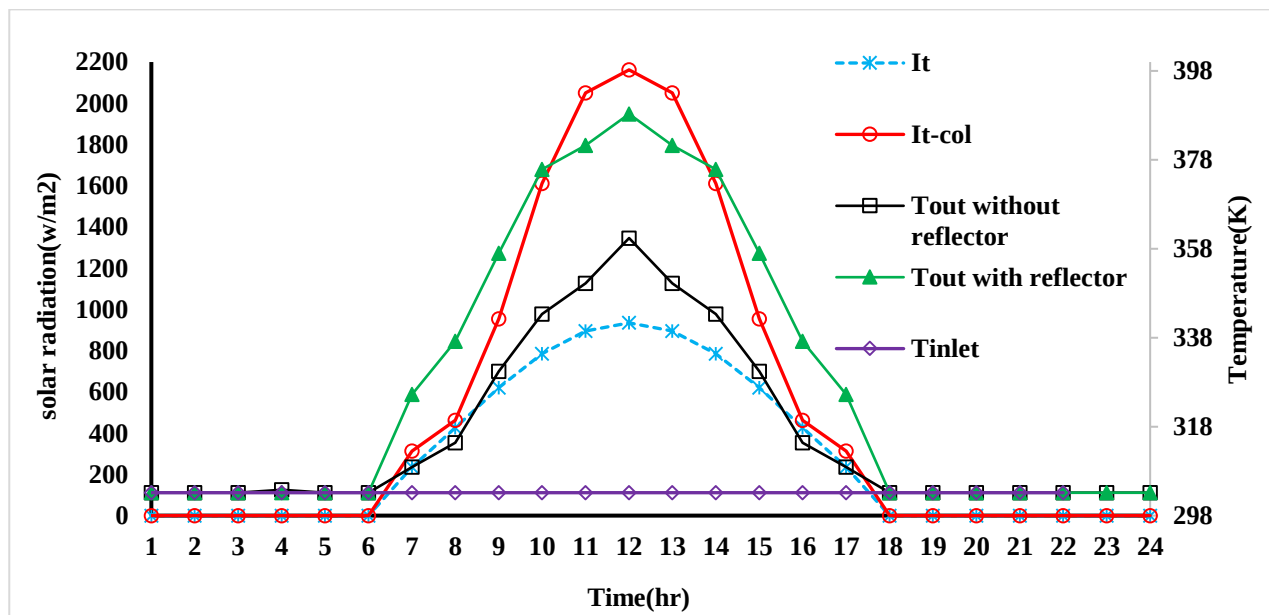
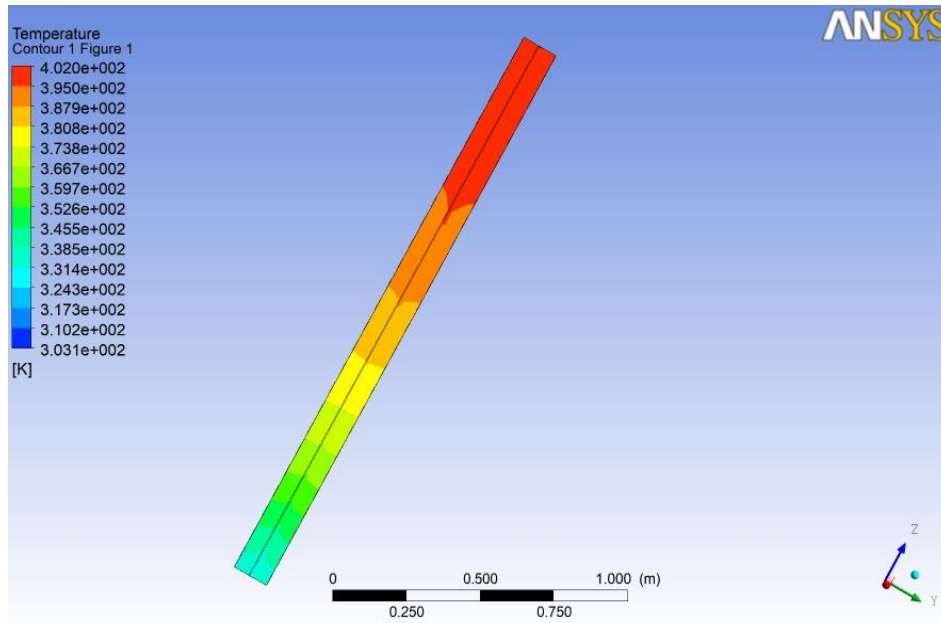


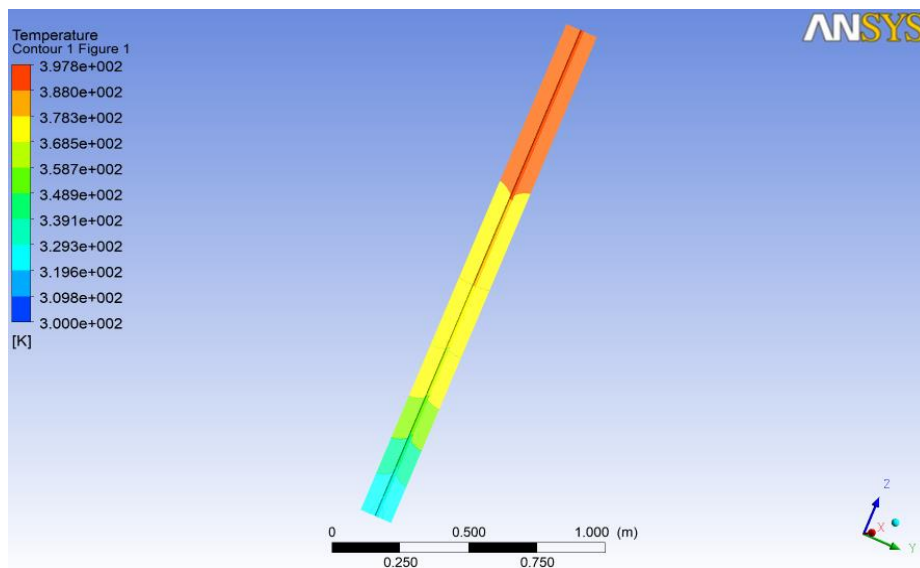
Fig 7.4 Effect of reflectors on the instantaneous outlet temperature of water

7.2.3 Contours of Absorber Temperature at Different Inlet Velocity of HTF

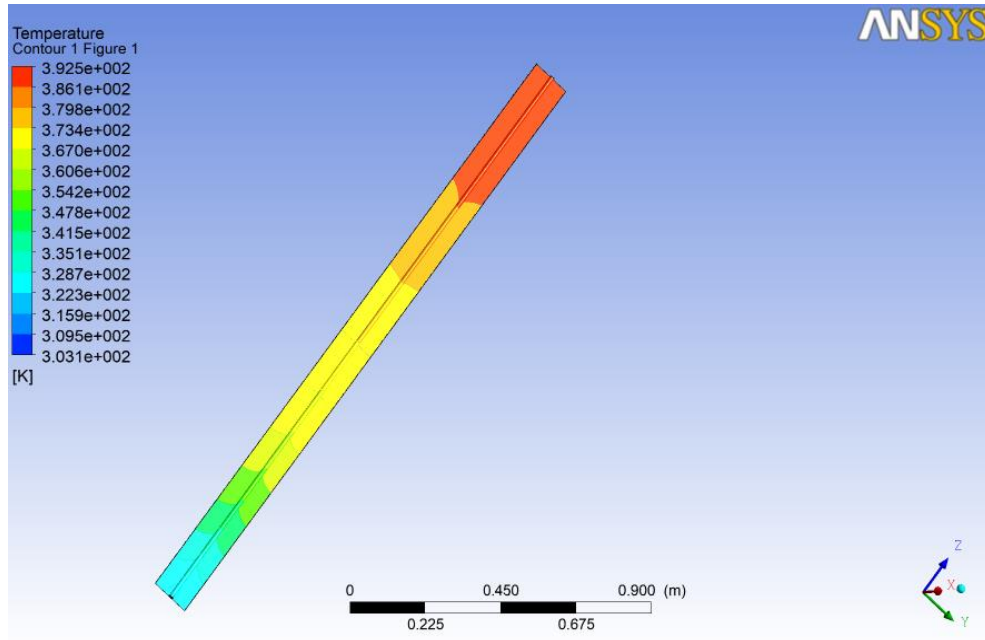
The results of ANSYS fluent for different inlet velocity of HTF and the variation of outlet water temperature along the riser tube, at the outlet pipe and absorber plate at average solar radiation of the day (1200 W/m^2) are presented in Fig.7.5 (a)-(d). In literature, matching result was reported by Vasudeva *et al.*, (2017) and Karanth *et al.*, (2011).



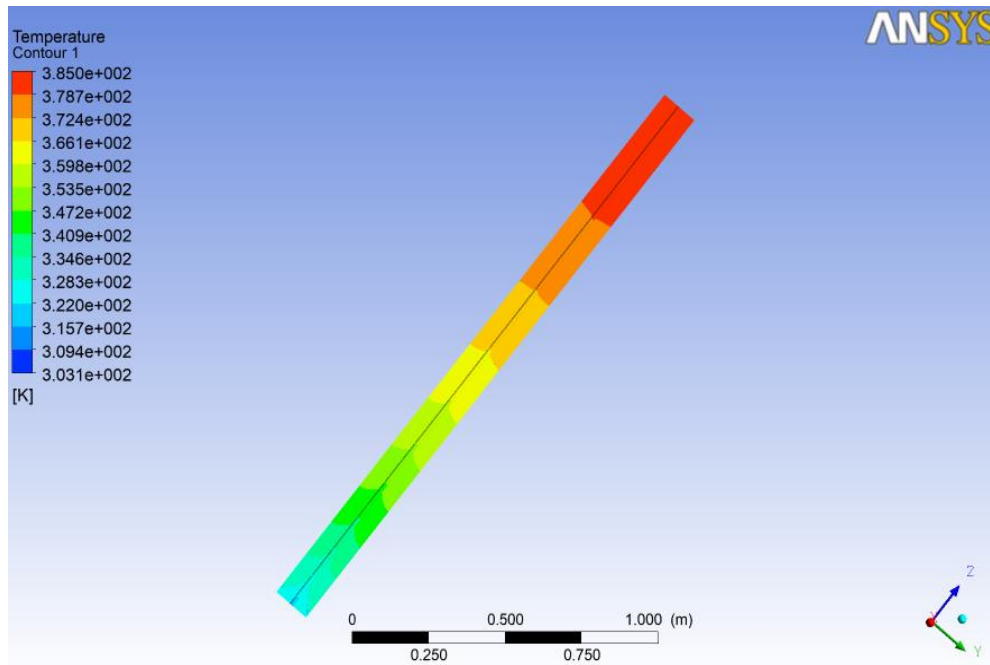
a)



b)



c)



d)

Fig 7.5 Contours of absorber plate temperature for different inlet velocity of water

a) $V = 0.02\text{m/s}$ b) $V = 0.025\text{m/s}$ c) 0.03m/s d) 0.04m/s

7.2.4 Effect of Inlet Velocity on the Collector Outlet Temperature of Water

Fig.7.6 depicts the longitudinal variation of water outlet temperature for different inlet velocity of HTF. As the predicted result shows, when the inlet velocity increases the outlet temperature of water

decreases which is resulted from the decrement of time it takes for the water to absorb the heat from the absorber. Whenever the inlet velocity decreases it gets enough time to absorb the heat from the absorber as it passes though the riser tubes.

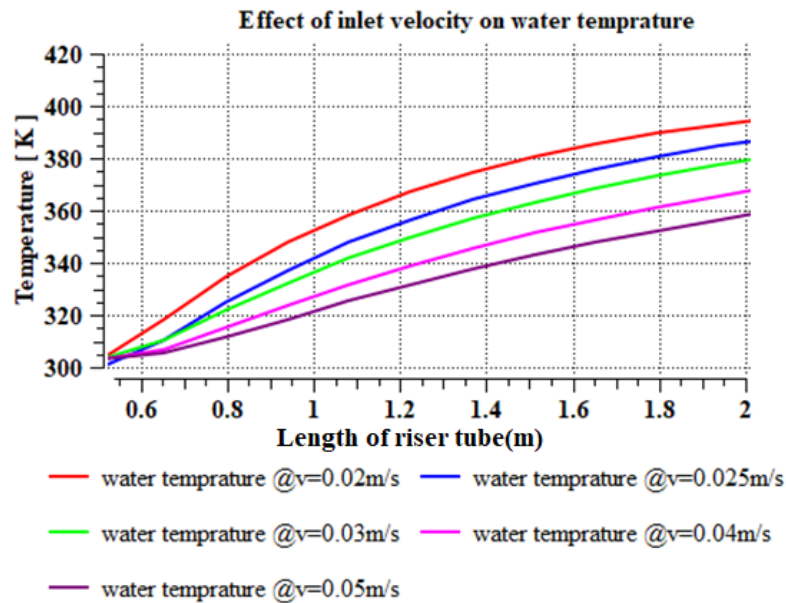


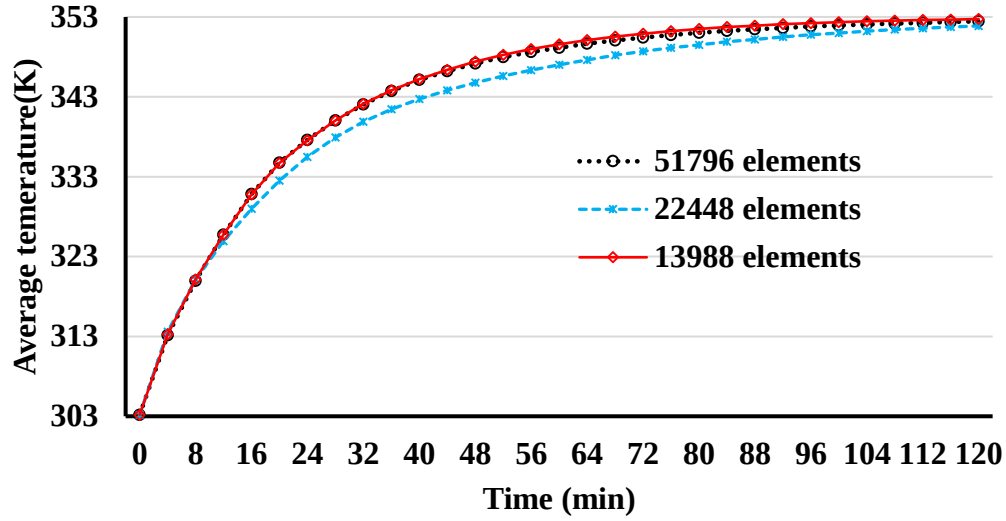
Fig 7.6 Effect of inlet velocity of water on outlet temperature of HTF

7.3 Numerical Results of LHS System

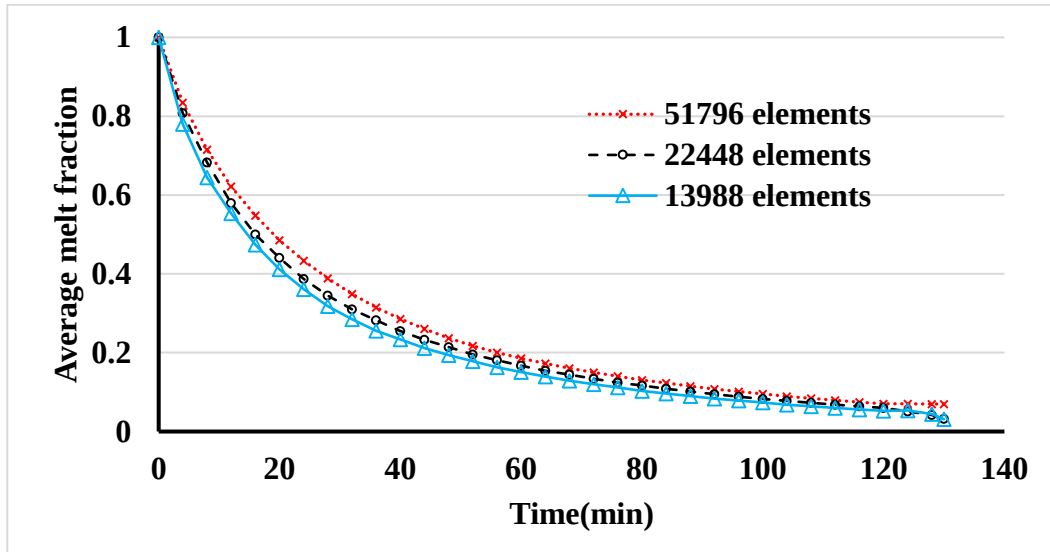
This section presents the results of numerical simulation on the thermal performance of stearic acid filled LHS unit. Grid independency test of the model, the variation of average temperature, average melt fraction, energy absorbed/released during charging/discharging process of the LHS have been presented and discussed.

7.3.1 Grid Independency Test

The variation of mesh element sizes has a major control in the numerical results of all models. To test the grid independency of the numerical results, simulations with different mesh element sizes were run for both charging and discharging processes. Fig.7.7 a) indicates the grid independency of the average temperature during charging processes for 13988, 22448 and 51796 number of elements. Fig.7.7 b) depicts grid independency test of average melt fraction for 13988, 22448 and 51796 number of elements during discharging process.



a)



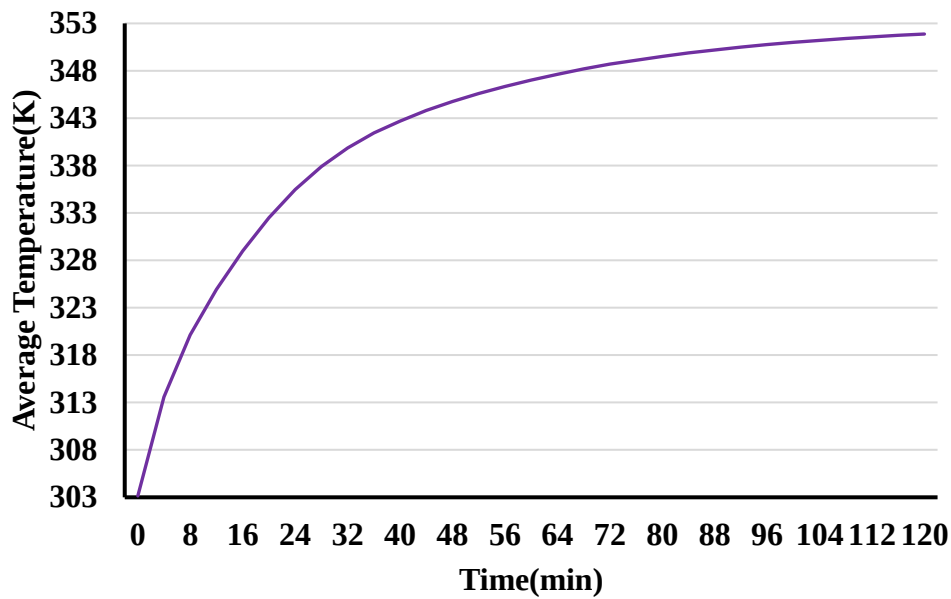
b)

Fig 7.7 Grid independency test with a) average temperature for charging and b) average melt fraction for discharging

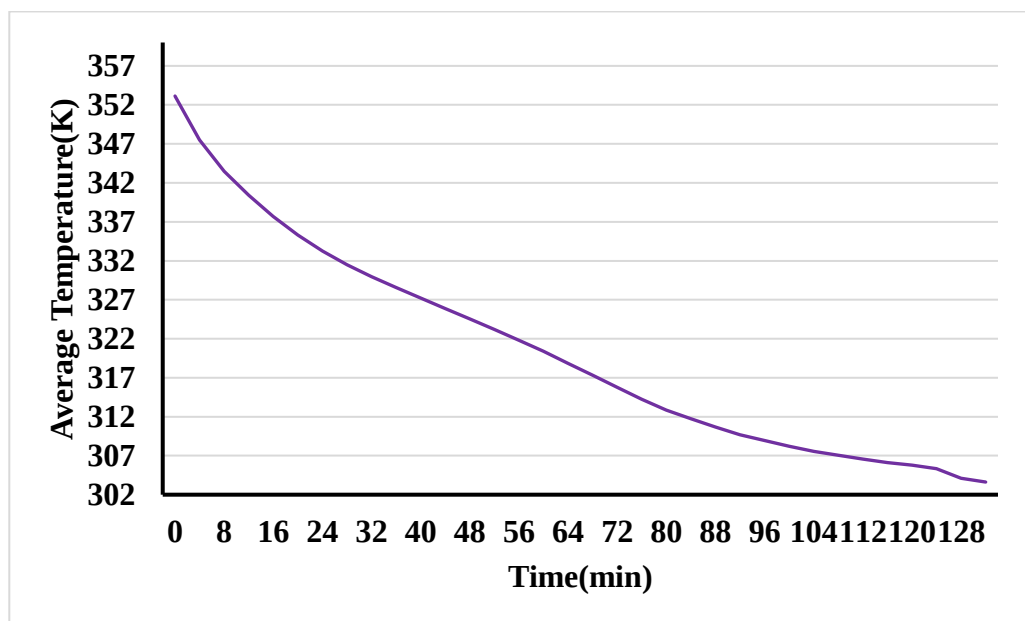
7.3.2 Temperature Variation of Stearic Acid During Charging and Discharging.

The average temperature variation of the PCM during charging and discharging period is depicted in the Fig.7.8 (a) and (b). The average temperature is the volumetric average temperature of all the mesh elements of the numerical model. When water from the collector at 363K passes through the HTF tubes the temperature of PCM increases as it melts during charging process as shown in Fig.7.8 (a).

Discharging process starts when off-sun shine or cloudy period and during night time as water at 303K passes through the HTF tubes, it absorbs heat and the temperature of the PCM decreases until equilibrium occurs as it is depicted in Fig.7.8 (b).



a)



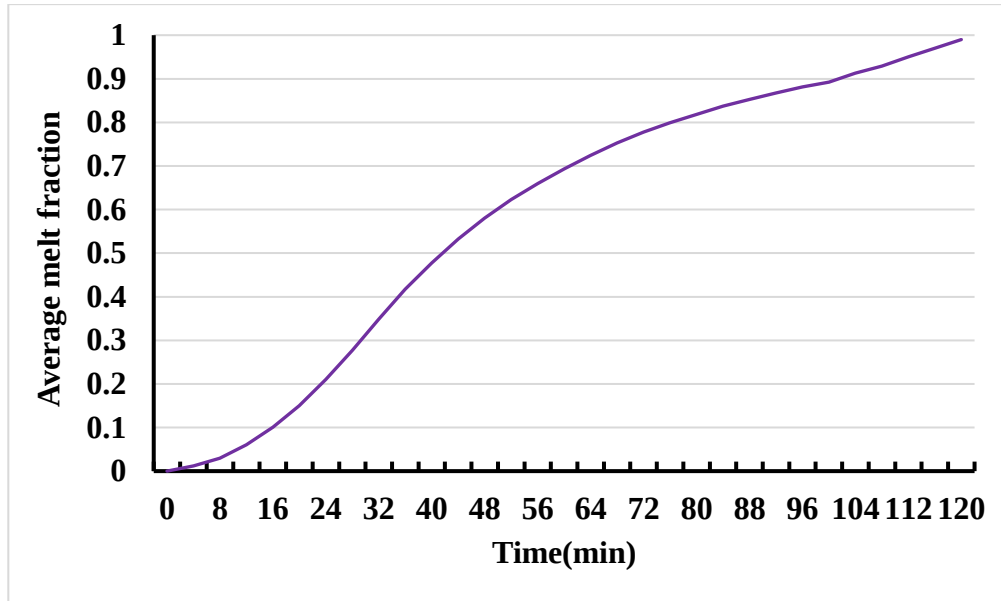
b)

Fig 7.8 Average temperature variation of shell and tube LHS system. a) Charging b) Discharging

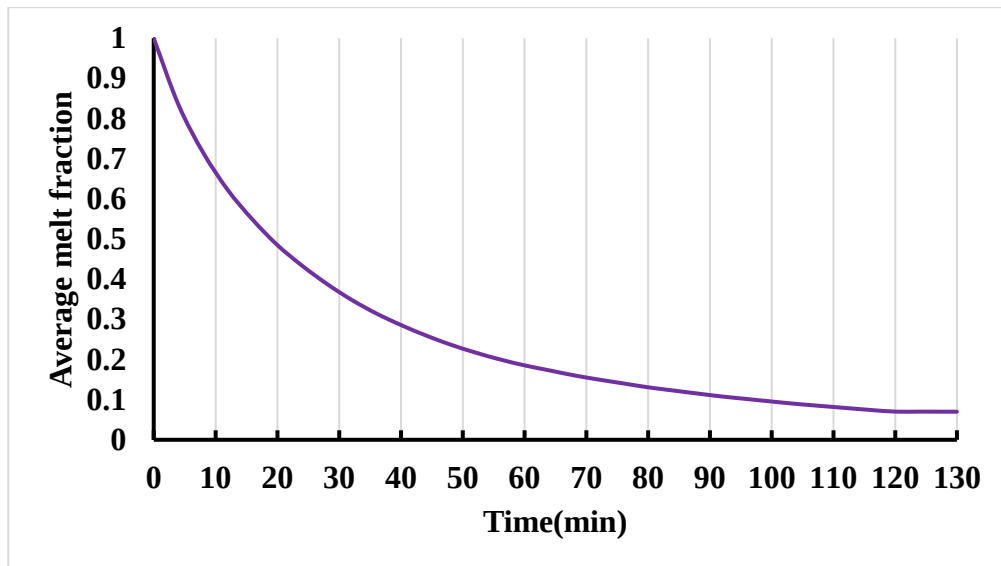
7.3.3 Charging / Discharging Time

Melt fraction is a critical performance parameter, which indicate only the latent heat storage/discharge characteristics during the charging/discharging process. Fig.7.9 (a) and (b) indicates the variation of

average melt fraction of the PCM during charging and discharging period. Average melt fraction is the volumetric average melt fraction of all the mesh elements of the numerical model.



a) Charging process



b) Discharging process

Fig 7.9 Charging and discharging time of stearic acid filled LHS model

7.3.4 Contours of Average Melt Fraction

The contour plots of average melt fraction of the stearic acid versus time during charging/discharging process is depicted in Fig.7.10 (a) and (b). It can be observed that at time $t = 0$ min, the average melt fraction of the latent heat storage varied from 0 to 1 for charging process and 1 to 0 for discharging.

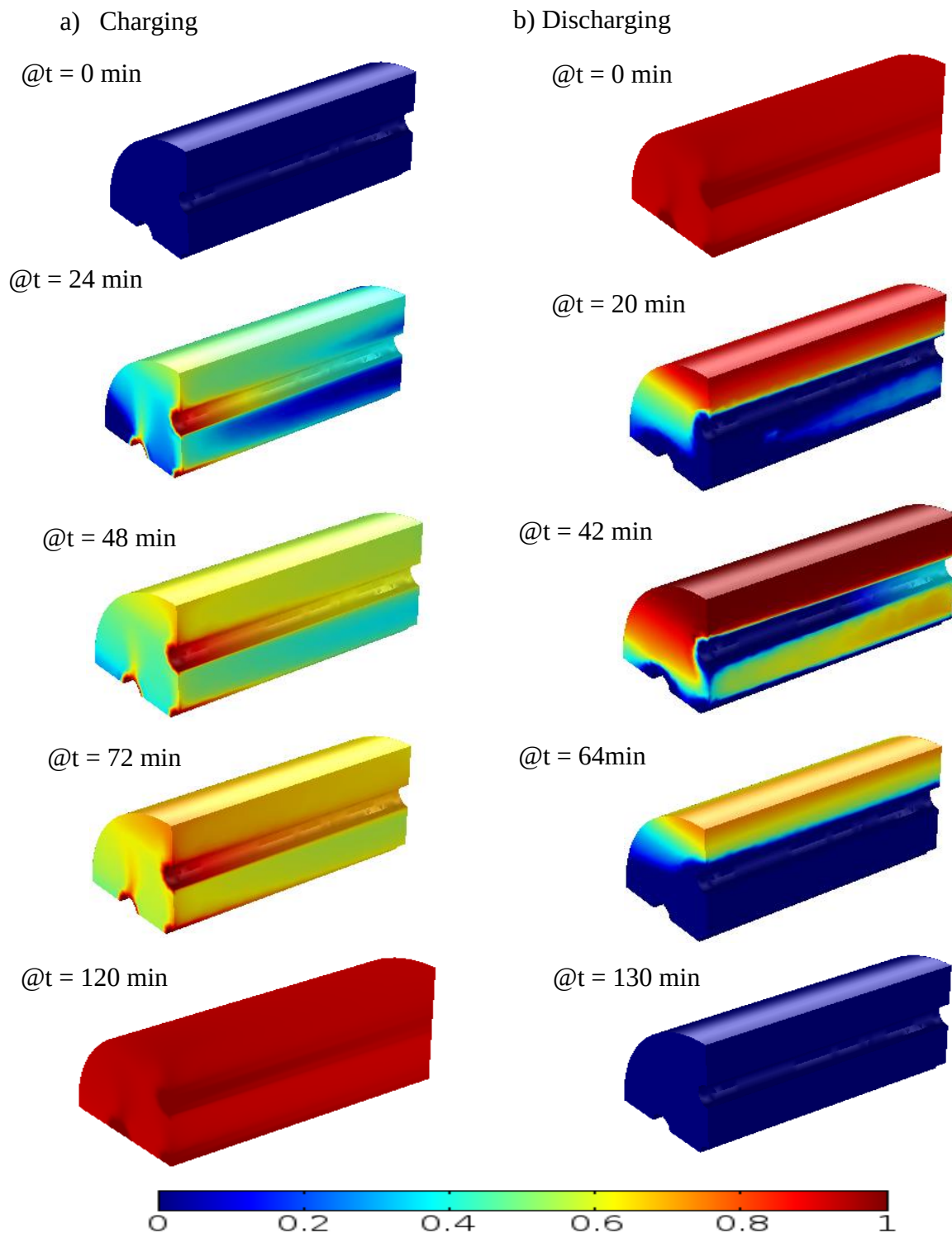


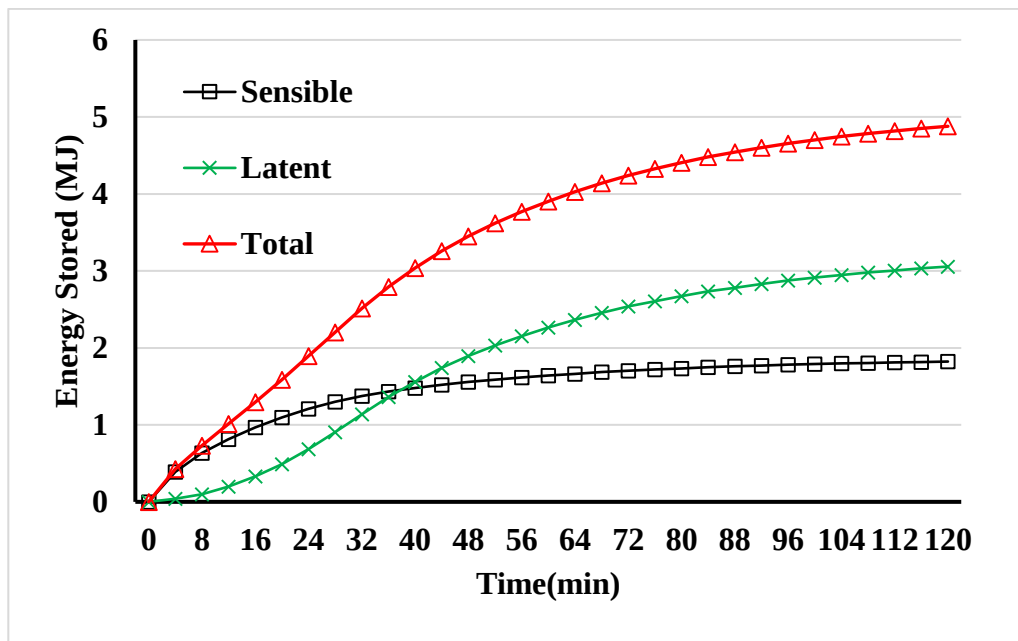
Fig 7.10 Contour plots of melt fraction of stearic acid for a) Charging b) Discharging processes

From Fig.7.10 (a) it is shown that the LHS is fully charged at ($t = 120\text{min}$) whereas Fig.7.10 (b) indicated that the LHS is not fully discharged within 2hr. This indicates that discharging process takes

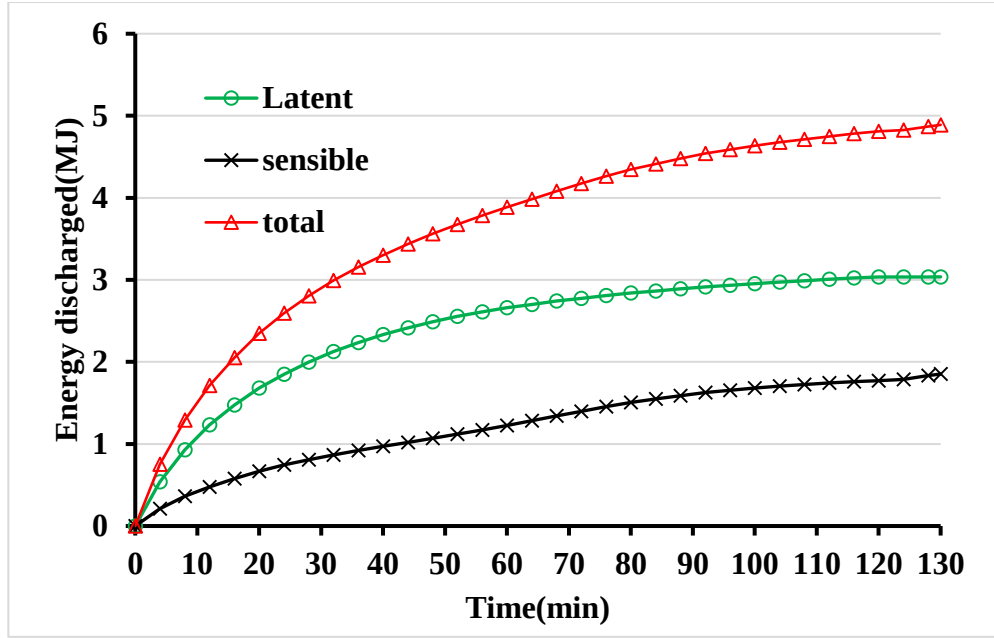
more time than charging which is caused by the presence of additional natural convection heat transfer other than conduction heat transfer for charging process. As conduction heat transfer dominated for discharging process, it takes more time to release its heat and become fully solidified with in 2hr. Thus, it takes 130min for the stearic acid to fully solidified whereas it is fully charged within 120min.

7.3.5 Energy Stored/Discharged

Fig.7.11 depicts the variation of energy storage /discharging rate from the LHS during both charging and discharging process. When the water at 353K circulated through the HTF tubs of shell and tube storage unit in charging cycle, heat transfer occurs from the water in the tube to the stearic acid stored in the shell and during discharging process, water at 303K circulated through the tube results in the release of stored energy from the PCM to the water. At the beginning of both processes, the heat gets stored or discharged in the form of sensible heat only. But, when the stearic acid reaches its phase change temperature, heat starts to be stored or released in the form of latent heat.



a)



b)

Fig 7.11 Energy storage/discharging rate of LHS model during a) charging b) discharging

7.4 Experimental Results of the FPC Collector: Without Reflector

To investigate the performance of FPC without side reflectors experimental tests were performed on June 29, 2020. The closed loop test was started after performing the experimental procedure described in chapter 6. The various results obtained are presented and discussed in the following sub section.

7.4.1 Comparison and Validation of Numerical Absorber Plate Temperature With Experimental Result.

Plots of experimental absorber plate temperature and simulated result versus time for the given flow rate is shown in Fig.7.12. The Figure indicated that for the same flow rate of water through the riser tubes of the absorber, the experimental values of plate temperature is higher than the predicted one. This difference is resulted from the assumption of perfect contact between the absorber plate and the tube, better heat transfer occurs between the plate and the tube compared to the actual case. Hence, the simulated absorber plate temperature values are lower than that of the experimental values with maximum error deviation of not more than 1.89%.

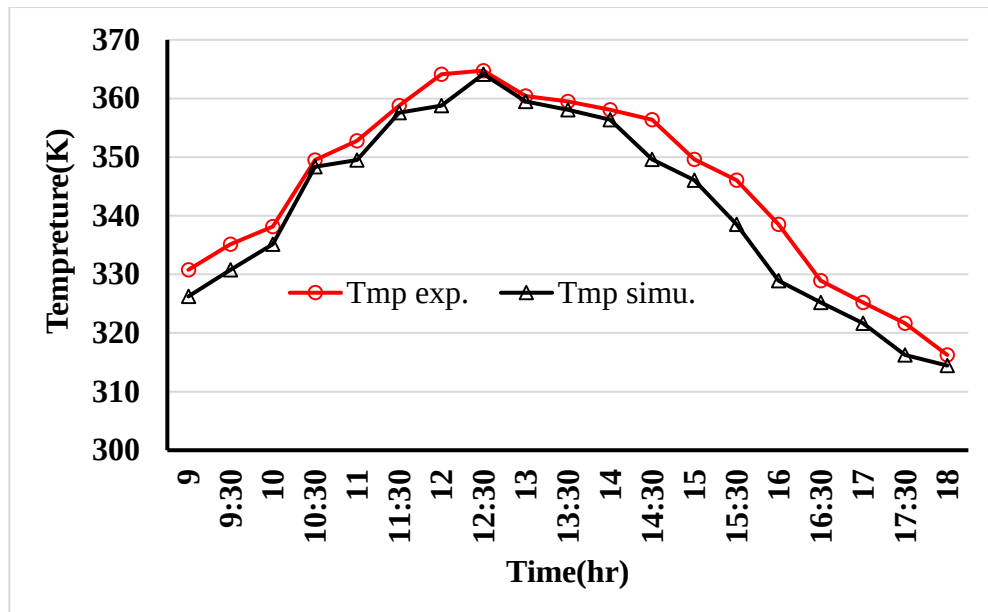


Fig 7. 12 Variation of simulated and experimental results of mean plate temperature versus time

7.4.2 Validation of Collector Outlet Temperature

The numerically simulated results of collector outlet temperature of water were validated with the experimental results of the same. The variation of water temperature as a function of time with the variation of solar insolation and ambient temperature throughout the day is depicted in Fig.7.13. As the Figure illustrates, water temperature of simulated result exceeds the experimental one. The difference in outlet water temperature is resulted from the assumption of perfect bonding contact between the absorber and the riser tubes through which the water flows. The maximum error deviation between the two results were not more than 3.2%.

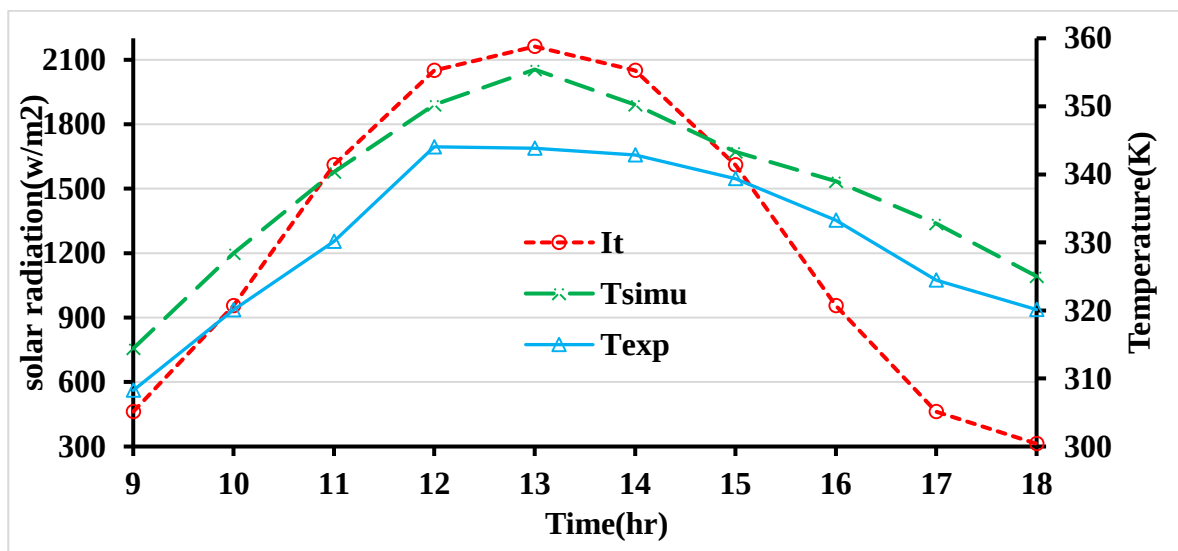


Fig 7.13 Comparison of simulated and experimental results of water outlet temperature

7.4.3 Variation of Different Collector Parameters for Closed Loop System

Fig.7.14 depicts the variation of average plate temperature, collector outlet temperature, collector inlet temperature, solar insolation and ambient temperature with time of the day. The absorber plate reaches its maximum temperature at the mid-day (12:30pm) where solar insolation is maximum. The temperature of water from the outlet of collector is also proportional with the variation of solar radiation falling on the collector. Whenever the solar radiation reduces outlet temperature of water exceeds the absorber plate temperature. This is resulted from the heat capacity of water and the reduction of ambient air temperature which in turn results the convective heat loss for the absorber to increase.

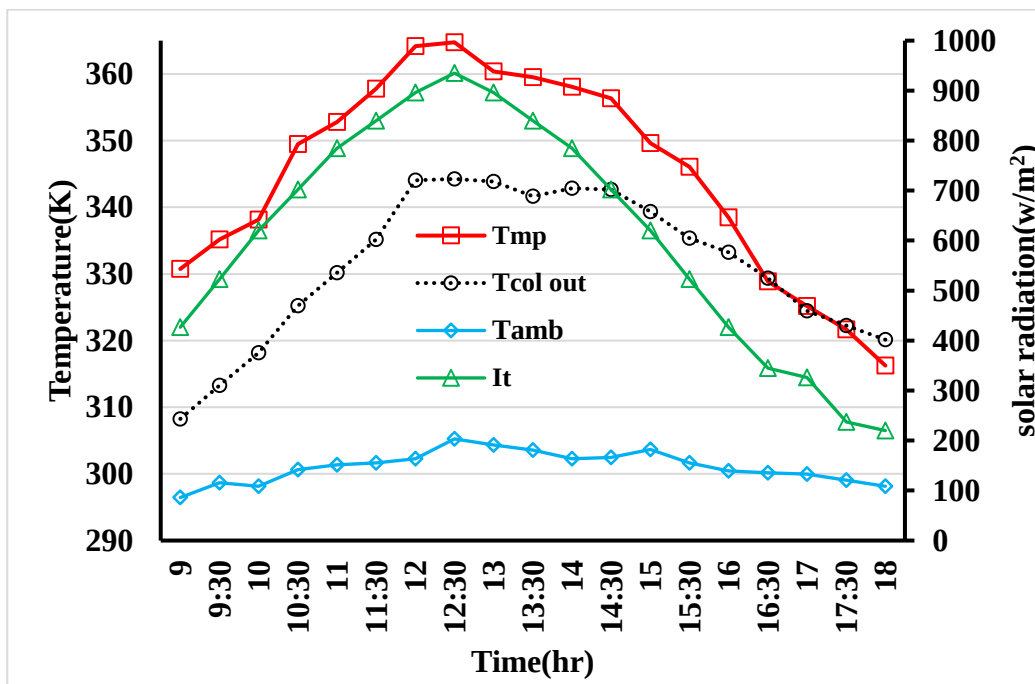


Fig 7.14 Average absorber plate temperature as a function of time of the day.

7.4.4 Thermal Efficiency of the Collector

Fig.7.15 elaborates the variation of instantaneous thermal efficiency of FPC under the given measured input parameters of mass flow rate, ambient air temperature and solar radiation of the day. The plot indicates that the thermal efficiency of the collector increases and reaches maximum (63%) at mid-day (12:30pm) where solar radiation falling on the collector is maximum. Whenever the difference in the inlet and outlet temperature of water from the collector decreases the thermal efficiency also decreases and vice-versa. As it can be seen from the graph, the inlet and outlet temperature of water were 303.15K and 308.15K in the morning where the experiment started (9:00am) respectively and it gradually increases until it reaches maximum temperature of 326K (at 12:00pm) for the inlet and 344.15K (at 12:30) for the outlet temperature of water.

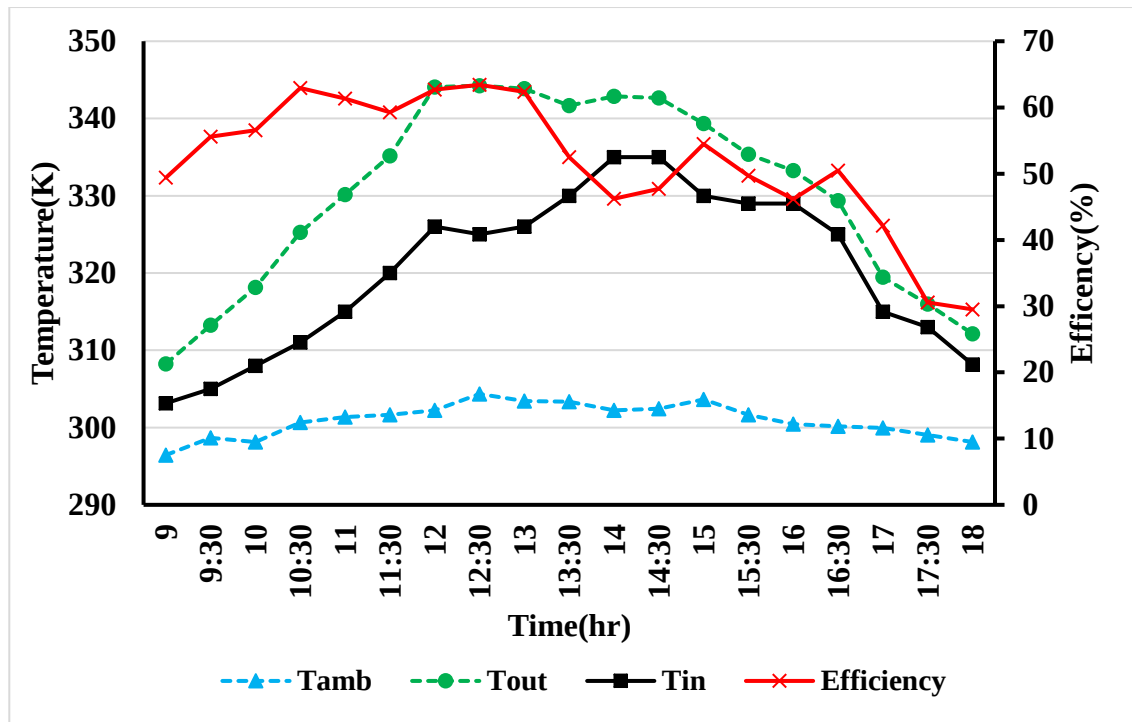


Fig 7.15 Instantaneous thermal efficiency of closed loop FPC system

7.4.5 Exergy Efficiency of the Collector

Fig.7.16 depicts the variation of exergy efficiency as a function of time for a constant mass flow rate of 0.002087kg/sec. exergy efficiency indicates the net amount of useful heat which is utilized for heating the water in the riser tubes. The maximum exergy efficiency obtained was 42% at 12:30pm.

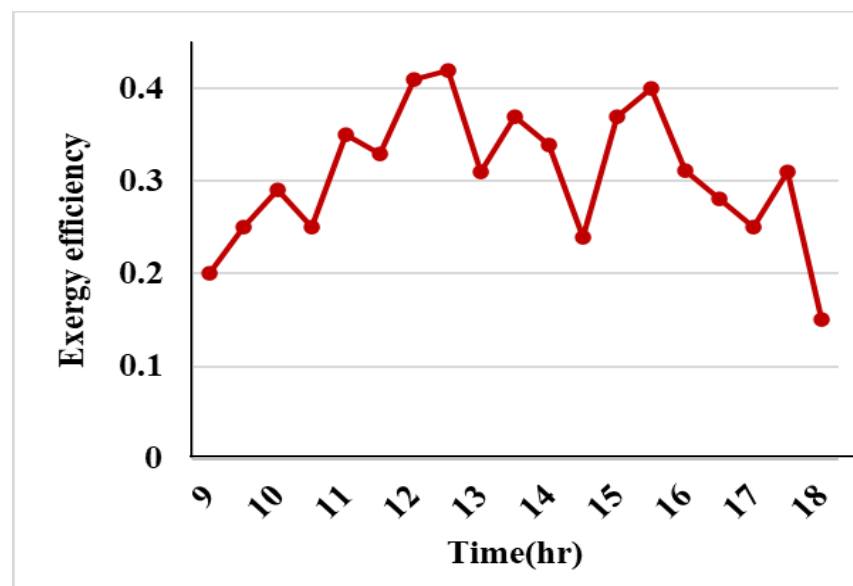


Fig 7.16 Exergy efficiency of FPC as a function of time

7.5 Experimental Results of Modified FPC (With Reflector)

In order to investigate the effect of reflectors on the performance and operating temperature of the FP solar collector, closed loop experiment was performed with a modified FPC having 4 reflectors in each side. Measurement of different parameters were performed using thermocouples fixed at different locations of the system after adjusting the reflectors using manual tracking mechanism to their optimum angle which will give maximum reflection of solar radiation to the collectors. The angle of reflectors was adjusted in the interval of 2hr for optimal reflection of solar radiation. Validation of experimental results were performed by comparing the result with the numerical simulation result and previous works. Estimated value of hourly solar radiation, calculated value of mass flow rate and measured ambient air temperature were taken as an input parameter to generate the following graphical results.

7.5.1 Validation Collector Outlet Temperature

The numerical result of water outlet temperature from the collector is validated using the experimental result. Fig.7.17 shows the variation of collector outlet temperature of water as a function of time. As the graphical plot depicts the temperature of water at 9:00 morning was 325.15K and it oscillates due to the effect of wind speed and cloud with increasing trend until mid-day reaching maximum temperature of 376.65K at 13:00pm. Comparatively the plot of simulated result is smooth as the wind speed and cloud variations are omitted and also causes the difference in the two results with maximum deviation of 4.25%.

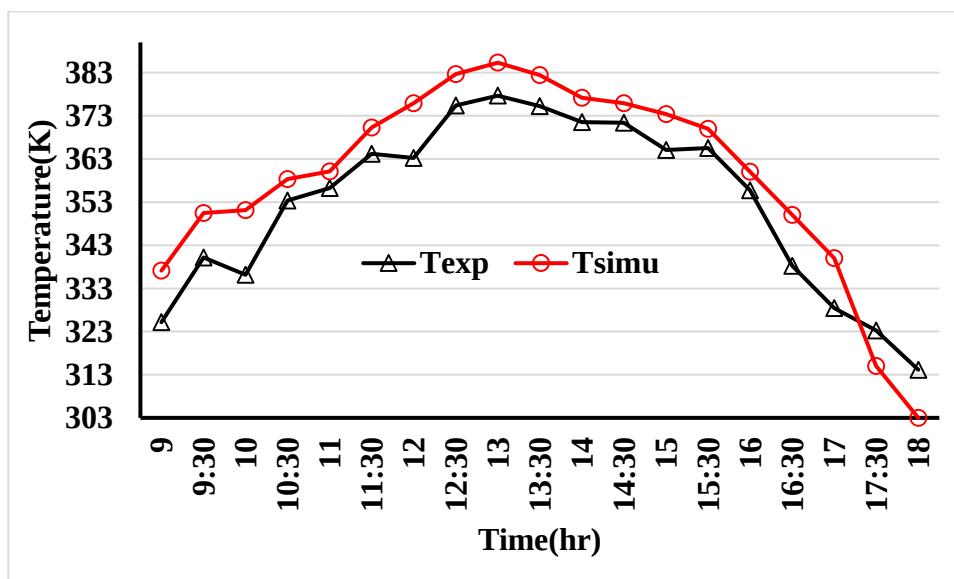


Fig 7. 17 Comparison of experimental and numerically simulated result

7.5.2 Validation of Average Absorber Plate Temperature

In the experiment the temperature of the plate was measured using thermocouples fixed at different locations of the absorber plate and the recorded figure were averaged to get mean plate temperature (T_{mp}). In the no-load test of closed loop experiment, the absorber plate temperature gradually increases until mid-day and reaches a maximum temperature of 400.75K at 13:00pm as indicated in Fig.7.18. As the solar intensity and coverage of reflection decreases the absorber plate temperature also decreases. Comparing with the simulated result the actual case is higher because of the assumption of perfect heat transfer between the absorber and riser tubes during simulation resulted in taking conducting the absorbed heat to the fluid flowing at a mass flow rate of 0.02087kg/sec through riser tubes. The error deviation between the two results were not more than 4.7%.

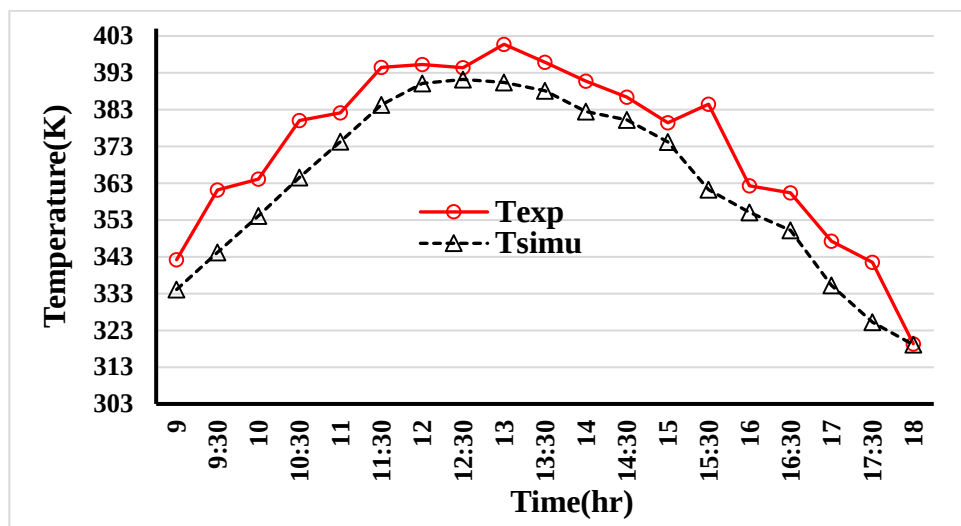


Fig 7.18 Comparison of experimental and simulated results of absorber plate temperature

7.5.3 Useful Heat Gain and Thermal Efficiency of Modified FPC

The performance of solar collector is measured with its effectiveness to increase the temperature of inlet fluid flowing at specific flow rate desired outlet temperature with a given aperture area and solar insolation. The instantaneous useful heat gain and thermal efficiency of modified FPC is evaluated from estimated solar radiation, measured inlet and outlet temperature of water flowing at a rate 0.02087kg/sec using Eqn.5.24 and 5.42 respectively. Fig.7.19 illustrates the instantaneous useful heat gain of the collector with respect to the time of the day. It is seen from the figure that the instantaneous useful heat gain increases with the increase in the intensity of incident and reflected solar radiation until 12:00pm. This is due to the fact that useful heat gain is strongly influenced by the incident and reflected solar radiation into the FPC.

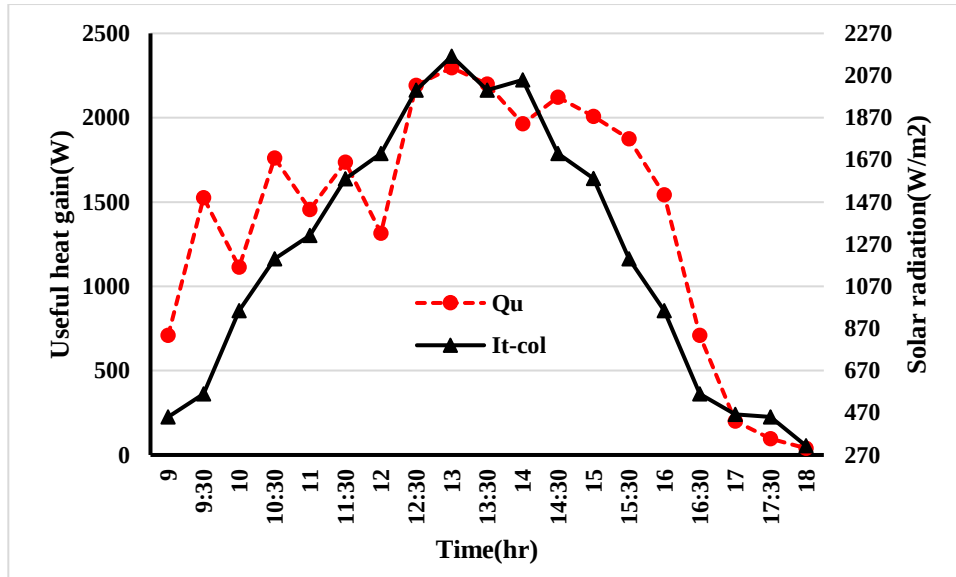


Fig 7.19 Useful heat gain of modified FPC as a function of time

The variation of instantaneous thermal efficiency of the modified FPC as a function of time is plotted in Fig.7.20. The graph depicted that the instantaneous efficiency of the modified FPC increases from 42% to 67.8 % during the time interval between 9.00 and 11.30am. after that, further increase is at a slower rate until noon, and then it starts decreasing. The peak efficiency attained around noon (12:30pm) is 79.1 %. The initial increase in the efficiency is due to the higher heat gain by the water at the lower temperature range, that prevailed in the time interval between 9:00 and 11:00 am. Further, the increase in the solar insolation rate is the highest during the time interval between 9:00 and 11:00 am. The pattern of efficiency is related with the difference between inlet and outlet of water from the collector.

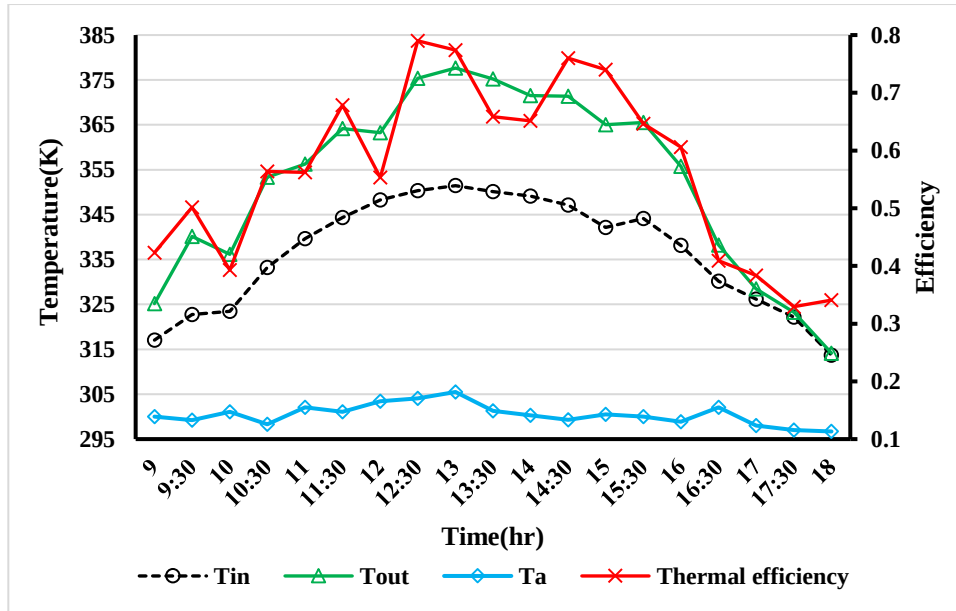


Fig 7.20 Thermal efficiency of modified FPC as a function of time

7.5.4 Exergy Efficiency of the Modified FPC

Exergy efficiency is a true measure of the performance of solar collector which indicate the amount of heat converted into useful work which causes the increase in working fluid flow exergy. The Exergy efficiency of modified FPC was evaluated using Eqn.4.61 with reference temperature of 303.15K and sky temperature of 4500K. Fig.7.21 illustrates the variation of exergy efficiency with time.

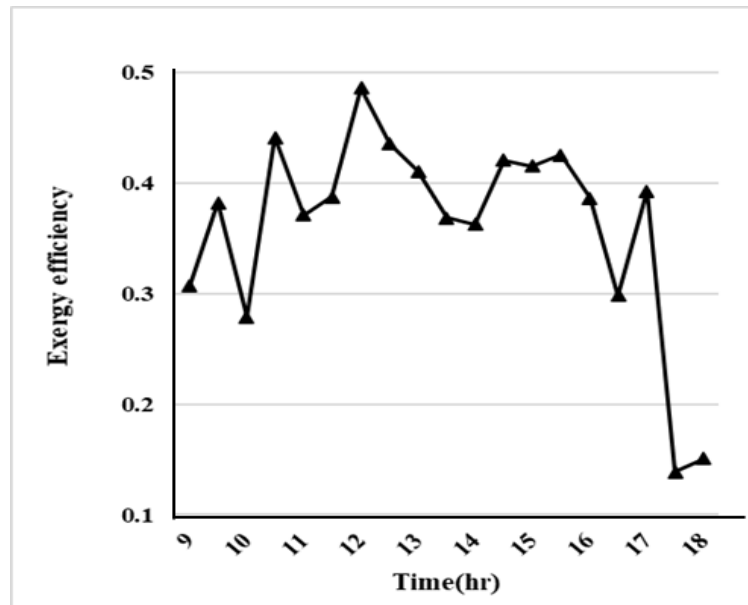


Fig 7.21 Exergy efficiency of modified FPC

7.6 Comparison of Various Performance Parameters for FPC With and Without Reflectors

In this sub section, effect of adding a side reflector on the performance of FPC is investigated and discussed. The comparison is performed using different measured performance parameters in the closed loop experiment.

7.6.1 Comparison of Absorber Plate Temperature With and Without Reflectors

Fig.7.22 indicates the effect of adding booster side reflector on the enhancement of collected solar radiation and hence the absorber plate temperature. Because of the addition of side reflectors, the average temperature of absorber plate with a reflector exceeds the absorber without by 25.1⁰C at 12:30pm.

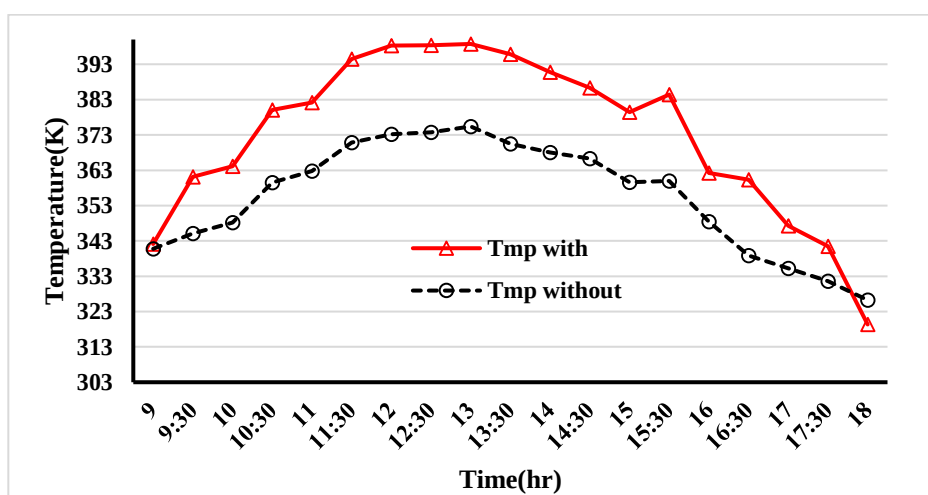


Fig 7.22 Effect of reflectors on the average plate temperature

7.6.2 Comparison of Outlet Temperature of Water from FPC With and Without reflector

The main aim of adding side booster reflectors was to increase the operating temperature range of FPC to be used as an indirect heat source for the solar cooking application. Fig.7.23 depicts the results of adding reflector on the enhancement of collector outlet temperature. Maximum outlet temperature difference of 20⁰ C was observed between a collector when it has reflectors and hasn't.

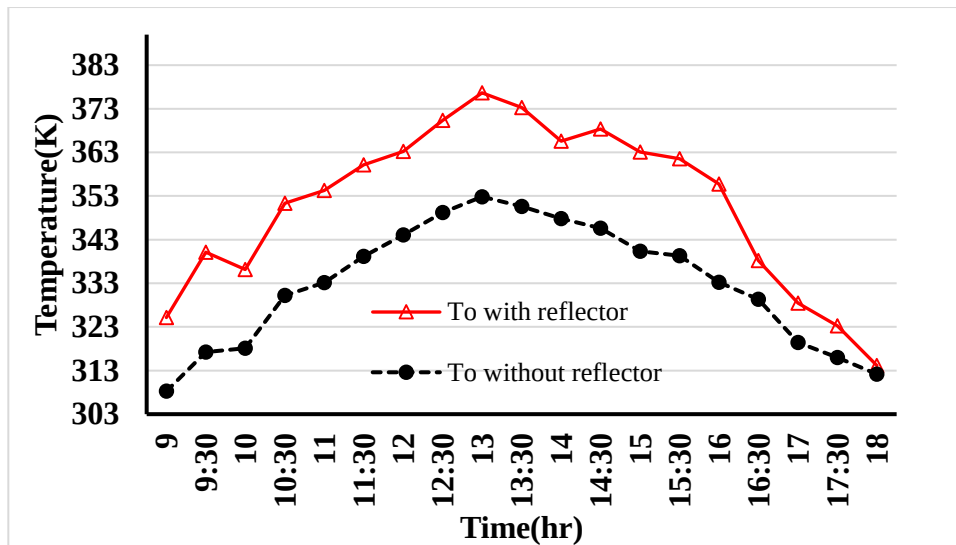


Fig 7.23 Effect of reflectors on collector outlet temperature of water

7.6.3 Comparison of Efficiency of FPC With and Without Reflector

Fig.7.24 illustrates the effect of adding side reflector in the performance of FPC. As it can be seen the collector with reflector had better performance than without. In late morning and sun set the efficiency of FPC without reflector better as the difference in inlet and outlet temperature are relatively higher and the collector with reflector faces shadow problem which reduce the performance of the collector.

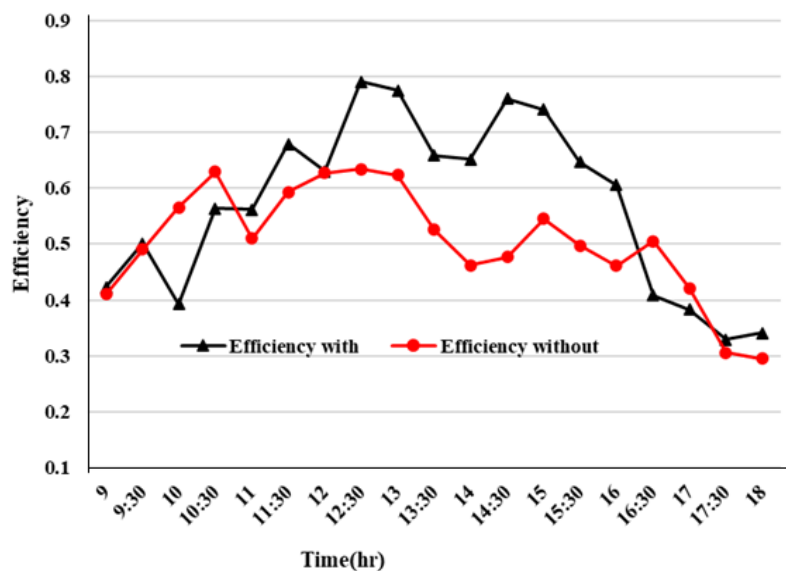


Fig 7. 24 Comparison of efficiency of the FPC with and without reflector

7.6.4 Comparison of Exergy Efficiency of the Collector With and Without Reflector

Fig.7.25 indicates the increment in exergy efficiency of the FPC as a result of addition of side reflectors. For both cases the exergy efficiency is dependent on the intensity of solar radiation falling on the collector but at peak hour (12:00pm) maximum exergy efficiency of 48% was achieved with modified FPC whereas an exergy efficiency of only 42% is achieved with FPC having no reflector.

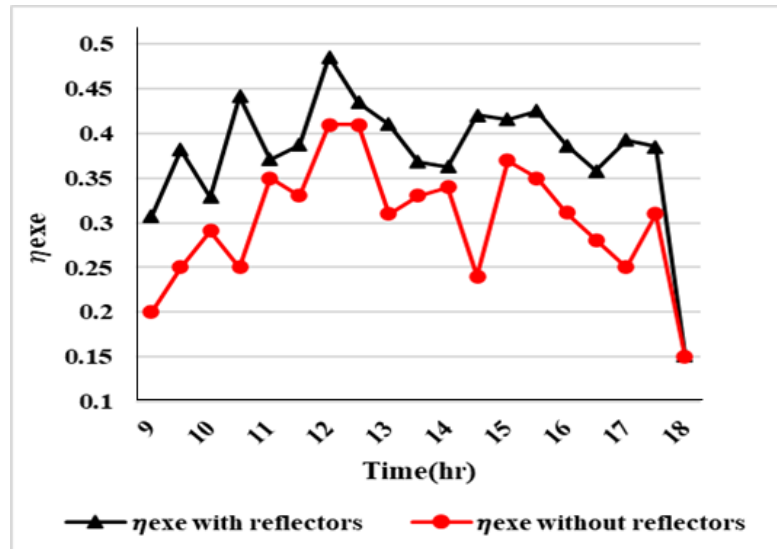


Fig 7.25 Comparison of exergy efficiency of FPC with and without reflectors

7.7 Experimental Results of LHS System

In this section the results and their implication obtained from experimental investigation of LHS unit during charging/discharging processes are discussed and presented in detail. The results were obtained from closed loop experiment with no-load on the pot.

7.7.1 Temperature Variation of Stearic Acid During Charging/Discharging Process

During charging test hot water from the collector outlet flows to the LHS and the variation of temperature at LHS inlet/outlet and at center and at upper surface of the PCM is measured and presented. From Fig.7.26 it is evident that the inlet temperature of LHS increases with time until solar noon and it starts to decrease as the solar radiation intensity decreases. Likewise, the outlet temperature of the LHS increases gradually with time until the PCM melts. Stearic acid starts melting after water temperature reaches 343.15K (melting point of stearic acid). As the plot shows the PCM fully melts within 2 hrs after the water temperature reaches 343.15K. As it can be seen inlet water temperature reaches 343.15K at a time range between 10:00-10:30 and stearic acid fully melts when PCM temperature at the center and middle of the shell become equal in the time range between 11:30 and 12:00.

The temperature of the stearic acid becomes constant as it stores latent heat until 13:00. Afterward, any temperature increase in the inlet temperature of water after the PCM melts results only sensible heat to be stored which is available until the inlet temperature starts to drop below the melting point of the PCM. When the inlet temperature of water drops below 343K around 16:30 the PCM temperature suddenly drops to its melting point. Afterward the temperature of the PCM becomes nearly constant as it takes time to release the stored latent heat. Comparing the LHS inlet and outlet temperature of water after 16:30 it is observed that the inlet temperature gradually decreases. However, the LHS outlet temperature of water was higher than the inlet temperature. That was because the outlet temperature takes heat from the PCM and maintain its temperature with the same fashion as the PCM.

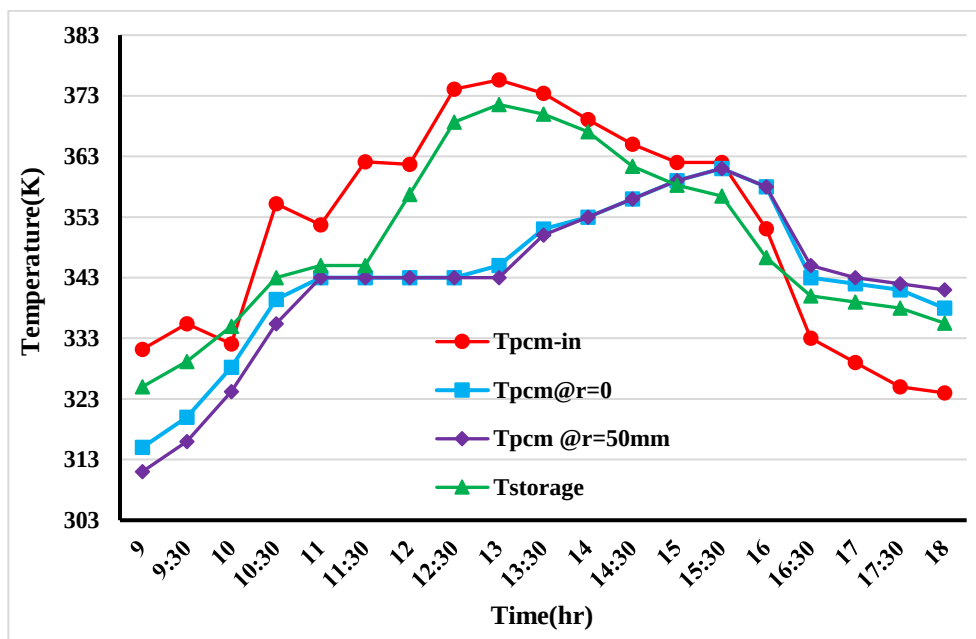


Fig 7.26 Temperature variation of stearic acid with time during charging process

Discharging process came into play whenever the solar radiation is unavailable and heat is required to be conveyed from the LHS to the cooking unit. Fig.7.27 indicates the temperature variation of stearic acid during discharging process. It showed that, after peak hour of the day, the inlet temperature of water decreases gradually so as the outlet temperature of the water. When water at flows though the LHS tubes at a temperature below the melting point of PCM, the stearic acid starts to release the stored energy until equilibrium reaches.

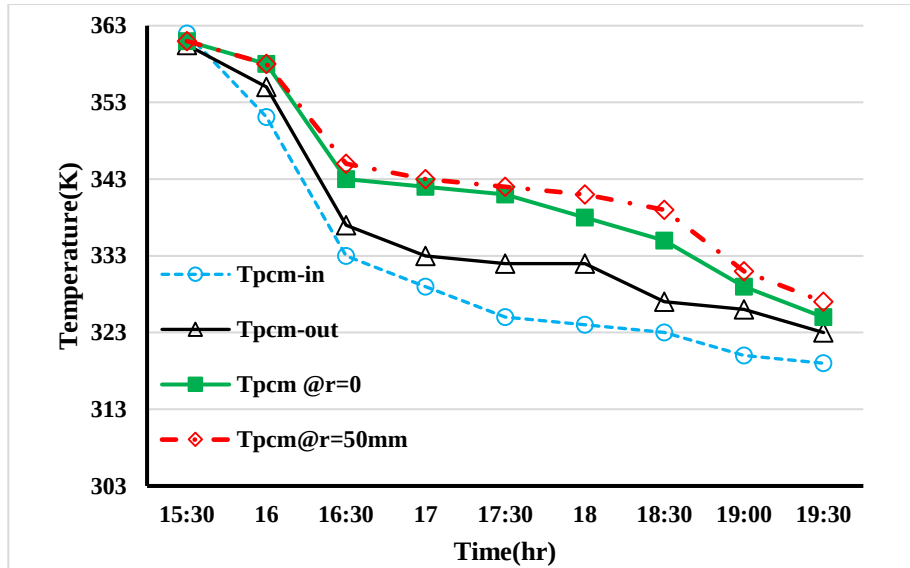


Fig 7.27 Temperature variation of stearic acid during discharging process

7.7.2 Variation of Temperature of Water in Storage Vessel During Charging/Discharging Process

No-load test is performed to see the effect of LHS on the temperature of water in the storage vessel. Fig.7.28 illustrates the effect of the PCM in the temperature variation of water in the storage. From the plot it is evident that the storage temperature increases with time during charging process. When the temperature drops resulting from decreasing the solar intensity at 16:30 the temperature of water drops suddenly until it reaches the melting point temperature (343.15K) of the PCM. After 16:30 the source of sensible heat for the storage vessel is the latent heat stored in the PCM.

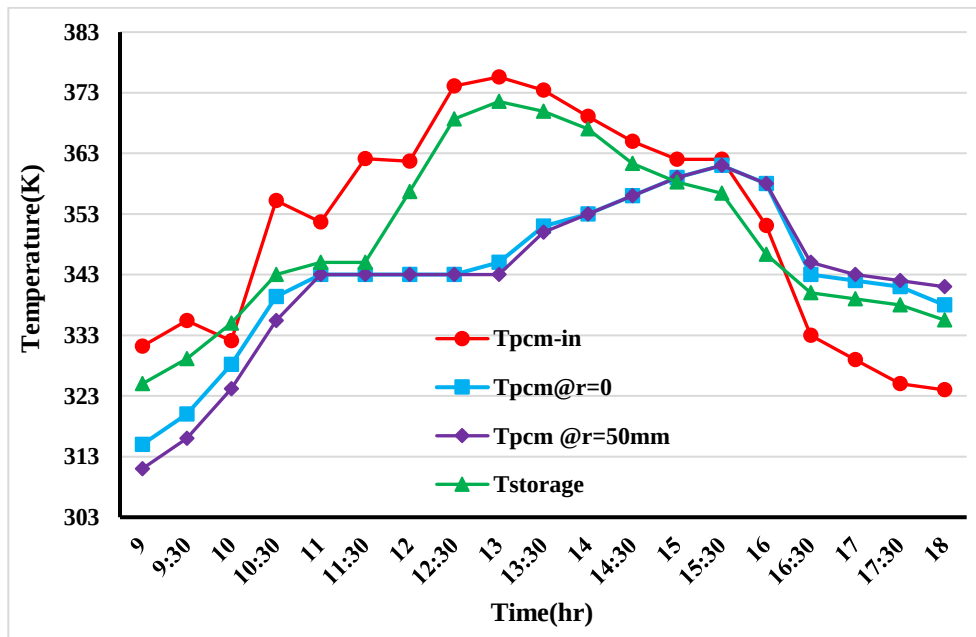


Fig 7.28 Variation of water temperature in the storage vessel during charging/discharging process

7.7.3 Energy Stored/Released During Charging/Discharging Processes

Initially, when the PCM is in the solid-state during charging, heat gets stored in the form of sensible heat only. Once the PCM reaches near the phase change temperature (343.7K), heat gets stored in the form of latent heat. Similarly, after melting of the PCM, further increment in temperature of PCM results heat to be stored in the form of sensible heat again. When the stearic acid reaches to its melting point during charging process with in 141min (after the fluid temperature reaches 343.7k) the amount of sensible, latent and total energy stored in the PCM were 1.89MJ, 3.52MJ and 5.41MJ respectively. During discharging process on the other hand, the PCM is initially in liquid state (361.5K) then heat gets released inform of sensible heat until it reaches to its melting point 343.7K. then the PCM starts to release its latent heat as the phase change occurs. The amount of sensible, latent and total energy released during discharging process with in 153min were 1.57MJ, 3.01MJ and 4.58MJ respectively.

7.8 Cooking Tests

To cheek the performance of the cooker under various conditions, different types of test were performed on the cooker. (i) Experiment without loading the cooking pot of the solar cooker. This set of experiment was conducted to investigate the effectiveness of the cooker and to investigate its thermal capability of the integrated PCM storage unit for cooking. (ii) Experiment with water as a cooking load this set of experiment were performed to conform the results of the first experiment with a load of high heat capacity and to calculate the cooking power of the cooker ate different cooking time. (iii) Experiment with different actual cooking loads. This experiment is also performed to investigate the cooker capability to cook actual food items based on the result of the second experiment.

7.8.1 No-Load Test

The behavior of HTF, charging and discharging processes as well as the general analysis of the system when it was not performing any cooking, were determined using a no-load experiment in which the water was just heated, with no cooking load added to the system. This test was used to predict the time it takes the system to cook various food items. In the present study no-load test were performed for 3 consecutive days (From June 30-July 2) using full setup of the experiment described in section 6.3. To start the experiment, the system pipe lines were filled with clean water as HTF. The reflectors were manually adjusted to their optimal angle based on the solar altitude angle and number of days of the year that the experiment is carried out. Fig.7.29 shows the variation of average stagnation plate temperature (T_{mp}), collector outlet temperature of water from the LHS ($T_{PCM-out}$), the average water temperature at the storage vessel($T_{storage}$), ambient air temperature (T_{amb}), and temperature at the surface of cooking pot(T_{pot}) as a function of time from 9:00am-18:00pm. From the plot it is noticed

that absorber plate temperature in the stagnation test increases up to 403K with time until solar noon. It then decreases as the solar intensity in the afternoon decreases. Water temperature from the LHS system, water temperature at storage vessel and temperature of the cooking pot surface have the same trend with each other during morning and afternoon. At 13:00pm maximum temperature of 372.9K and 369.1K was achieved in the storage vessel and cooking pot surface was respectively. The result of no-load test implies that load test can be performed as the temperature archived is enough to boil water at peak hour of the day (12:00-14:00pm) in the selected site (Adama latitude =8.54° longitude =39.28°E altitude = 1648 m). Similar no-load test was performed on 2th June 2020 to see the effect of daily solar radiation and ambient air temperature and repeatability of the result. As Fig.7.30 reveals almost similar fashion was obtained in the two different test days. Ekechukwu and Ugwuoke, (2002) reported almost the same result with reflector augmented solar cooker.

Fig.7.31 (a) and (b) shows the comparison of variation in water temperature of the storage vessel and cooking pot of the cooker with and without reflector during no-load test. From the Fig. 7.29 (a) it is evident that addition of reflector improves the average stagnation temperature of the water in the storage from 347 K to 370.77 K. As Fig. 7.31 (b) indicates the comparison of stagnation cooking pot surface temperature of the cooker with and without reflectors. On average an improvement of 18°C as possible by adding booster reflectors on the FPC.

To see the effect of LHS system, no-load test was performed after the inlet temperature of LHS drops below melting point of the PCM. Fig.7.32 shows the temperature variation of water in the storage vessel and cooking pot surface during no-load discharging process. From the graph it is evident that the temperature of water in the storage vessel

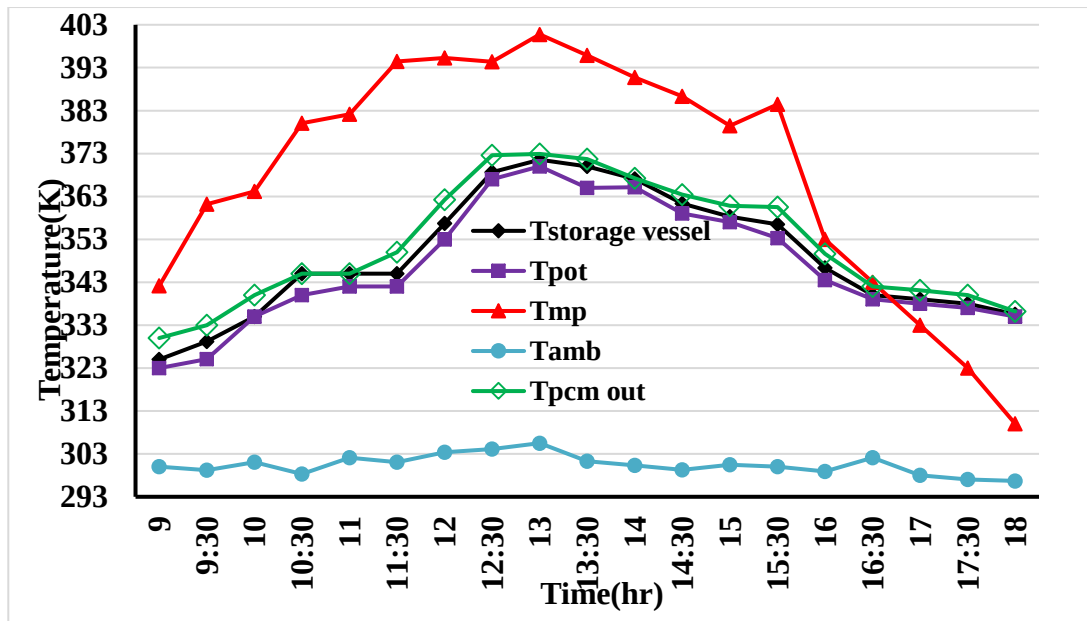


Fig 7.29 Variation of no-load test parameters with time (June 30th, 2020)

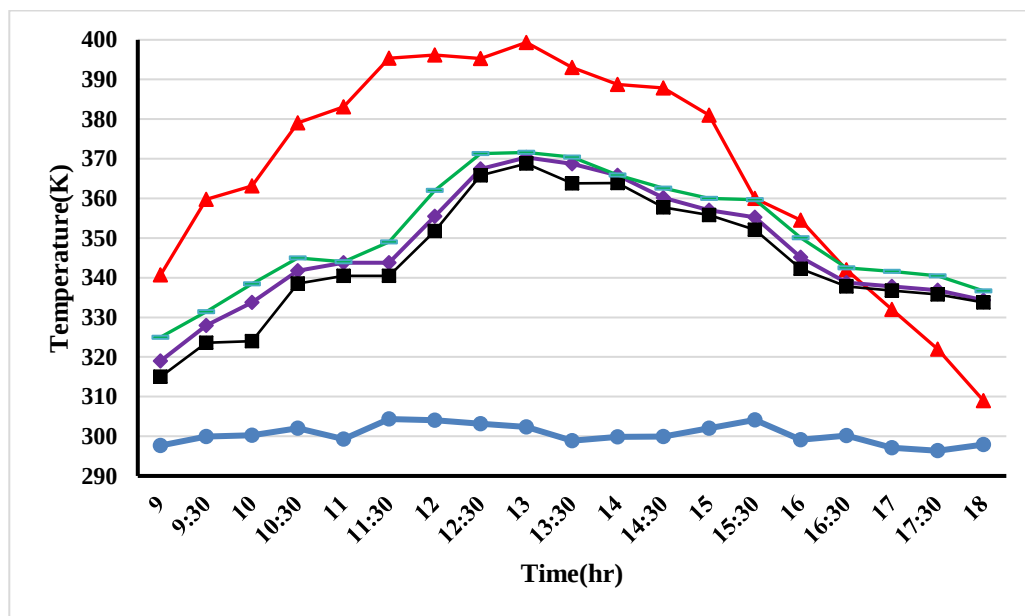
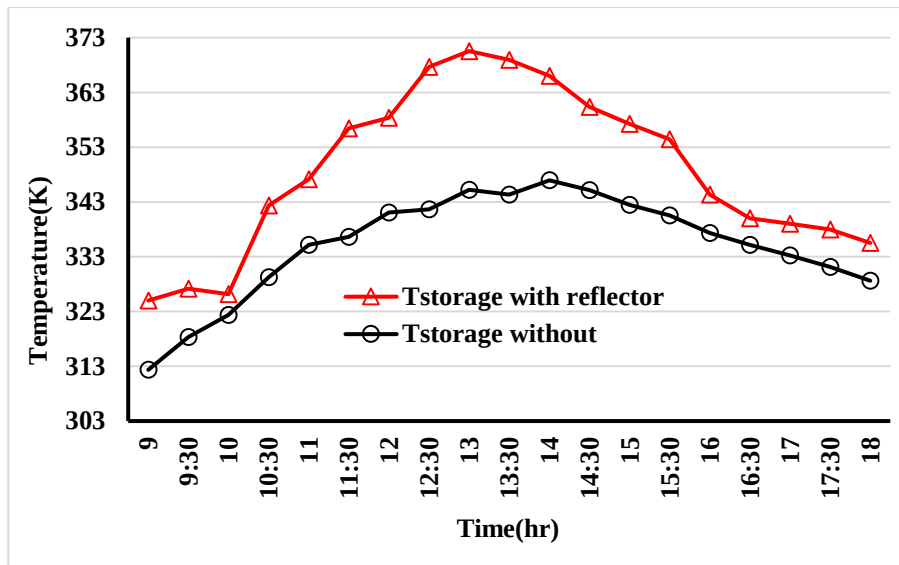
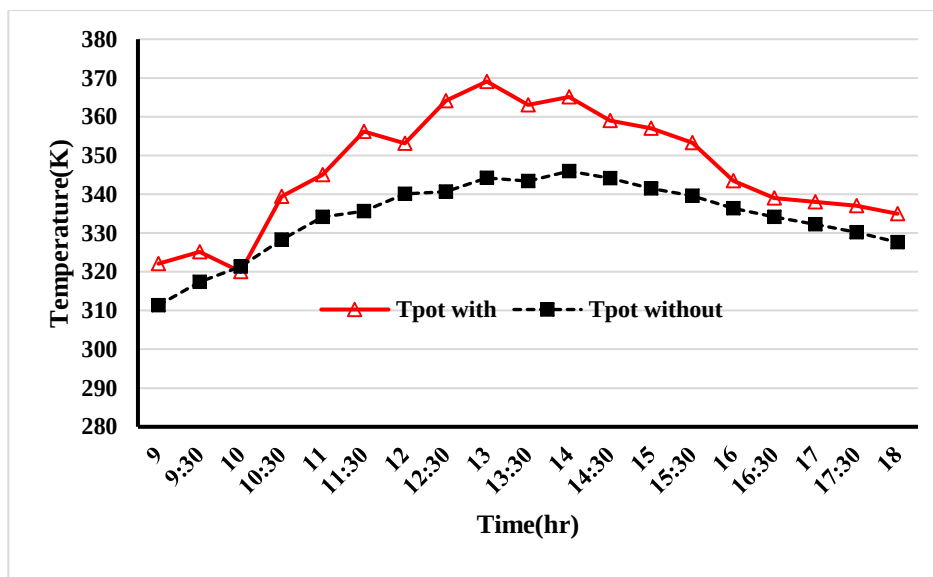


Fig 7.30 Variation of no-load test parameters (2th July 2020)



a)



b)

Fig 7.31 Comparison of no-load tests with and without reflectors

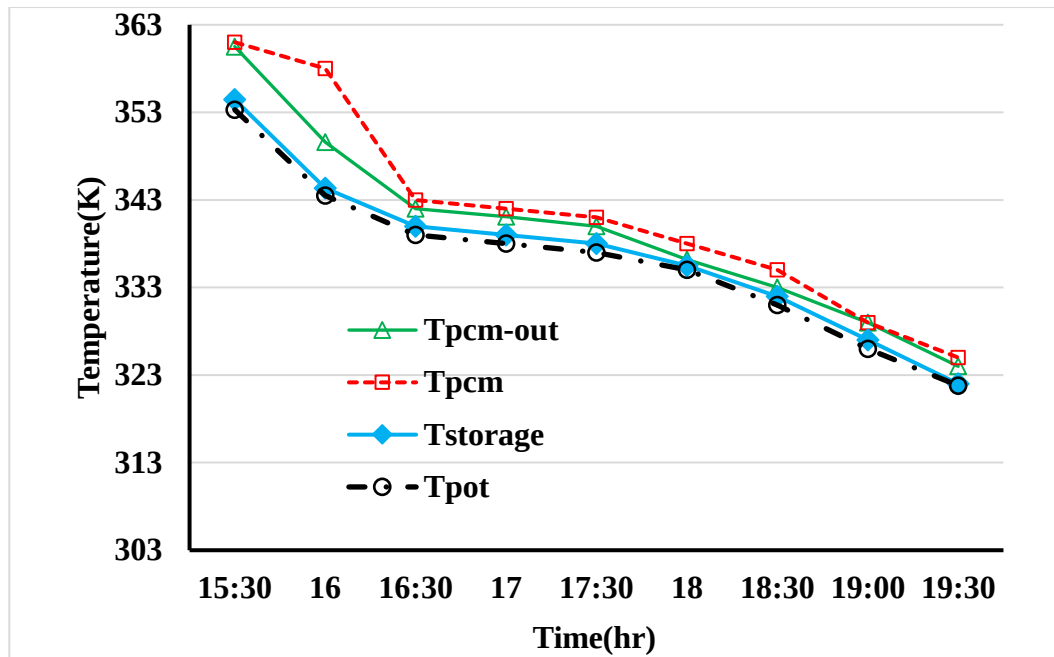


Fig 7.32 Effect of LHS during no-load discharging test of the cooker

7.8.2 Boiling Test

The experimental results from boiling water with solar cookers are often the basis on which the cookers' performance can be measured. Since almost all food items involve heating water, it is of the utmost importance that the behavior of this solar cooker be checked during water boiling. A serious of water boiling experiment was performed on 4th July 2020 from 11:30am - 6:30pm. Those boiling experiment were done using 3kg of water. Highly conductive removable cooking pot (Aluminum pot) filled with water is bathed in the hot water of the storage vessel. Indirect conduction heat transfer occurring between hot water and the cooking pot makes the water in the pot to rise its temperature. Boiling test was started after manually adjusting all the reflectors to their optimum angle at 12:00am when water temperature inside the storage vessel reaches 90.1°C (>80 °C higher than the lowest temperature required for cooking most boiling type foods (Domanski *et al.*, (1995))). The initial temperature of loaded water was 30°C. Fig.7.33 indicates the temperature variation of water in the pot in each test. In the first test water starts to form bubble with in 10min after it had been placed in the cooker and starts boiling after 30min at 97.5°C (caused by the Altitude of Adama city) and the storage water temperature was around 98.7°C. After water in the first experiment is poured out from the pot, the second boiling test starts. Starting from 30°C, the water in the second test took 36min to reach 90°C. The last experiment was performed after unloading the water in from the second experiment. As the plot indicates it takes about 42 min for the water in the third test to achieve the temperature of 65°C. From the boiling test it can be seen that for the test period from 12:30 to 15:30hr the cooker can be used for cooking any kind of boiling type meals. After 15:30hr the cooker can

maintain a temperature about 70°C that can be used to cook meals which require low heating power like beans until 16:30. After 17:00 the cooker will reach a temperature below the melting point of the stearic acid (71°C). The stored energy in the PCM can be used for heating or keeping the meals hot at night and early morning. Schwarzer and Silva, (2003) reported almost the same result using indirect type of solar cooker with 3L of water as a load.

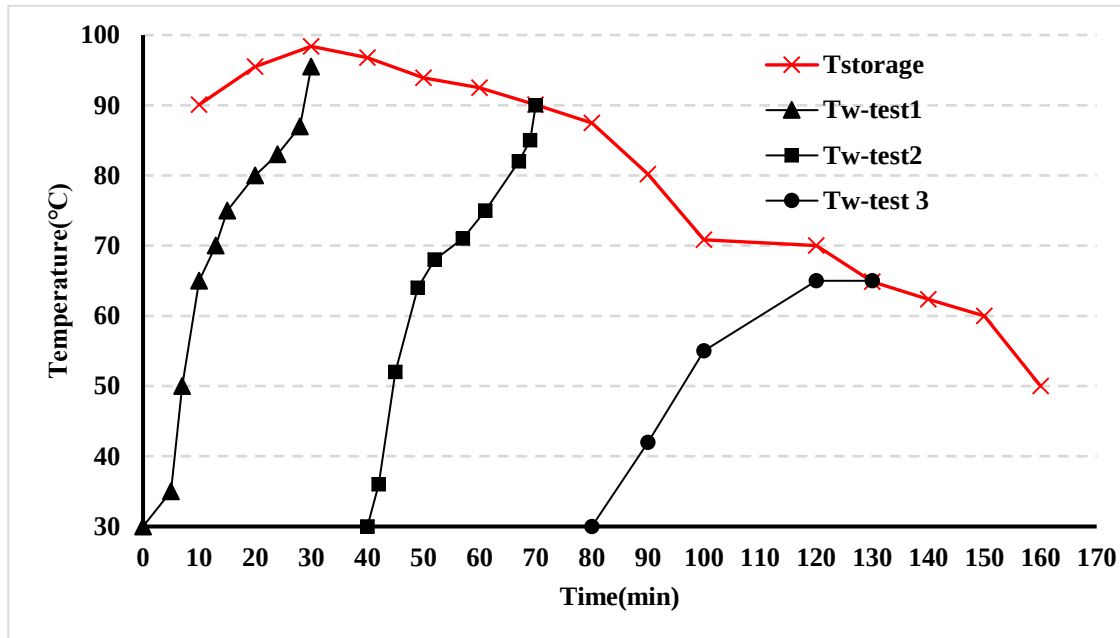


Fig.7.17 Sensible heating of water during boiling test

7.8.3 Load Test with Different Food Items

Based on the results of the boiling test, a final set of experiments has been conducted on the cooker with different actual cooking loads on 5th July, 2020. The first load was 1kg of pasta using 1.5 kg of water and the second load was 1kg of rice using the same amount of water. Those loads were loaded at different batch (evening, noon, sun-set). The first batch was started at 12:30pm with 1kg of pasta with 1.5kg of water. The temperature of water in the storage vessel and cooking pot surface during loading was 95.5 °C and 93.4 °C respectively. The status of pasta in the pot was checked every 30min. The pasta in the pot were taken out at 2:20pm and it was found that the pasta was well cooked. It takes more time than expected as the day was with fluctuating sunlight and partially cloudy. The second batch of cooking started at 2:30pm with 1kg of rice and 1.5kg of water. During the time of loading the temperature of water in storage vessel and cooking surface was 92.4°C and 90.5 respectively. The rice was loaded 30min later than the water as it highly requires boiling type of cooking. Checking the status of the rice after 3:35 pm it was found that the rice was well cooked faster than the pasta does. This was because the improvement of solar radiation in the cooking period. To see the effect of LHS storage in heating and warming of food for specific period of time, third batch

of cooking test was started at 4:30pm after the solar intensity is low enough for the PCM to operate. The average temperature of stearic acid (PCM), water at storage vessel and cooking pot surface was 69.2°C, 71.3°C and 68.6°C respectively. For the last batch the cooker was loaded with 1kg of pasta and 1kg of water. Maintaining the pasta in the pot for 2hr it was found that the cooker was able to cook it even if it takes more time. Different photographs taken during load test is presented **APPENDIX D**.

7.9 Performance Tests of the Cooker

The result of water boiling test establishes the cooking ability of the present indirect solar cooker in hand. But performance measurement should be done to evaluate the cooker performance with given standard and to perform comparative study with other type of cookers.

7.9.1 Sensible and Latent Cooking Power of the Cooker.

The cooking power during boiling test of the cooker was determined using Eqn.5.81. The cooking power obtained was dependent on how fast the temperature of water to increases during the test. Fig.7.34 illustrates the sensible cooking power of the cooker. As it can be seen, in the first water boiling test, the cooker used 458W to boil 3kg of water in 30min. To boil the same mass of water in 36min it took 378W and to sensibly heat the same mass of water from 30°C to 65°C with in 42min it takes about 175W.

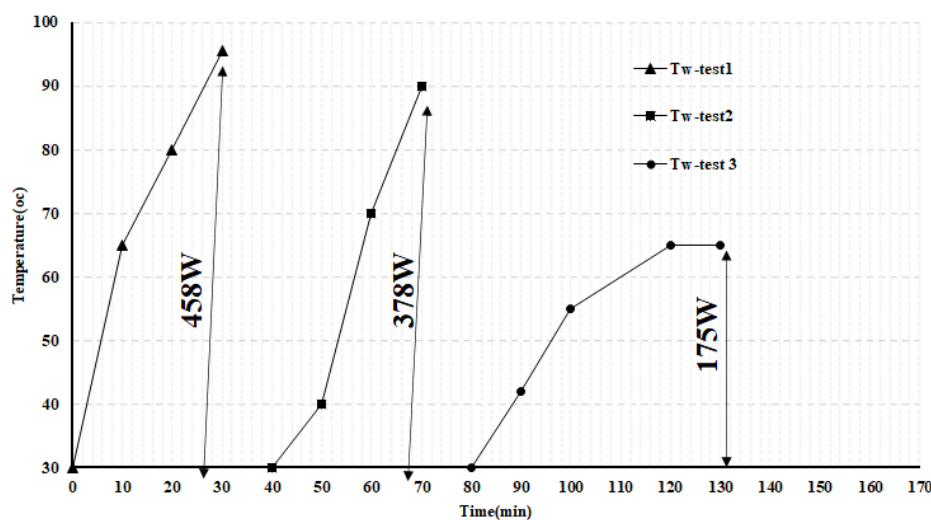


Fig 7.33 Sensible cooking power for the boiling tests

7.9.2 Thermal Figures of Merit

First figure of merits for the cooker with and without reflectors on FPC were determined from the stagnation no-load test of results of mean plate temperature, ambient air temperature and average solar insolation using Eqn.5.83. Maximum amount of F1 obtained was 0.09Km²/W and 0.21Km²/W

for the cooker with and without reflector on FPC respectively. The minimum possible value of F1 to cook any meal using solar cooker is $0.1\text{Km}^2/\text{W}$ (Ekechukwu and Ugwuoke, 2002). Second figure of merits was evaluated from water boiling test only for the cooker with reflector and was found to be 0.28.

7.9.3 Sensible and Latent Efficiency of the Cooker

Sensible efficiency of the cooker which is the ratio of sensible heat required to rise a kg of water from ambient to boiling temperature to the global solar radiation incident on the tilted collector with given time can be expressed as (Silva *et al.*, 2003);

$$\eta_{sensible} = \frac{m_w C p_w (T_b - T_i)}{I_{ave-tot-col} A_c \Delta t} \quad 7.1$$

$$\eta_{latent} = \frac{m_w h_{fg}}{I_{ave-tot-col} A_c \Delta t} \quad 7.2$$

Sensible efficiency for the cooker with reflector on the FPC were determined from the boiling experiment using Eqn.7.1. The daily average value of solar insolation falling in the 2m^2 collector area with an additional reflector enhancement for month July was estimated to be $800\text{W}/\text{m}^2$ and from boiling test the time it takes to change the temperature of water from 30°C to 97.7°C was 30min. Thus, the average value of sensible efficiency for the cooker with reflector on the FPC was found to be 29.6%. Latent efficiency of the cooker can be determined using Eqn.7.2 and was found to be 27.3%. Almost similar values of thermal sensible and latent efficiency were reported in literature with the same indirect solar cookers (Silva *et al.*, 2003).

7.9.4 Overall System Efficiency

Overall system efficiency is the measure of how much energy is saved by the utilization of the developed solar cooker.

$$\eta_{overall} = \frac{\text{output energy}}{\text{input energy}} \quad 7.3$$

To determine the overall efficiency of the cooker, the amount of daily wattage used to boil water was taken as output heat used by the cooker and the sum of daily average solar radiation falling on the collector area and the power taken by the pump as an input.

Thus, the overall system efficiency of the indirect solar cooker was found to be 30.1%.

7.10 Economic Analysis of the Indirect Solar Cooking System

In the utilization of solar thermal application, the main thing that arose in customers mind is its economical point of view. Generally, solar processes are characterized by high first cost and low operating costs. Thus, the basic economic problem is comparing an initial known investment with

estimated future operating costs. Among the economical evaluation methods, life cycle saving method is mostly used evaluation tool and thus, it was selected for the present study.

7.10.1 Total Cost of the Present Indirect Solar Cooker

The total initial investment cost of the present cooker includes the cost of the collector, storage system, cooking units and all other supporting and connecting units of the system. **APPENDIX C** summarizes the details of all the costs associated with the present solar cooker.

7.10.2 Investment Cost

Investment cost is related with the buying and installing costs of the solar system which includes all the cost of collector, thermal storage cooking unit and other connecting and related costs. Installed cost of a solar system is the sum of cost related with collector area and cost independent of collector area (Dagim, 2016).

$$CI = \text{cost of collector} + \text{cost components other than collector} \quad 7.4$$

Thus, $CI = 6509 \text{ birr} + 10,851.8 \text{ birr} = \mathbf{17360.8 \text{ birr}}$

7.10.3 Life Cycle Cost (LCC)

LCC is the sum of all costs associated with an energy delivery system including all of the items of hardware and labor that are involved in installing the equipment plus the operating expenses over a selected period of analysis or over its useful operating lifetime. LCC of a solar system can be determined from the investment cost and present value of maintenance and pumping cost over its life time using a relation given by (Addisu, 2007);

$$LCC = CI + C_m \frac{1}{i} \left(1 - \left(\frac{1}{(1+i)^n} \right) \right) \quad 7.5$$

Where; CI is investment cost

C_m is annual maintenance and pumping cost

n is operating life time in years

in the present economic analysis, interest rate, inflation rate, salvage rate and discount rate were assumed to be 10% and the life cycle cost of the solar cooker for 7 years of operation can be determined as;

$$LCC = 17360.8 + 1736.8 \frac{1}{0.1} \left(1 - \left(\frac{1}{(1+0.1)^7} \right) \right) = \mathbf{25816.27 \text{ birr}}$$

7.10.4 Life Cycle Saving of Energy

The life cycle saving (LCS) of solar cooker is the total energy cost saved due to replacement of electricity by alternative solar energy in the present worth (Duffie and Beckman, 1991).

LCS of energy (KWh) can be determined by;

$$LCS = P_e E_s \left(\frac{1+i_e}{1-i_e} \right) \left[1 - \left(\frac{1+i_e}{1+i} \right)^n \right] \quad 7.6$$

Where; P_e unit price of solar energy

E_s annual energy saving for a solar system

i - is the annual market discount rate.

i_e - is the annual market rate of energy escalation is assumed to be 15% for 2019G.C

The unit price of solar energy system can be determined when the life cycle cost of the system is equal to the life cycle energy savings of the solar cooking system during its life time.

When $LCC = LCS$

$$CI + C_m \frac{1}{i} \left(1 - \left(\frac{1}{(1+i)^n} \right) \right) = P_e E_s \left(\frac{1+i_e}{1-i_e} \right) \left[1 - \left(\frac{1+i_e}{1+i} \right)^n \right]$$
$$P_e = \frac{CI + C_m \frac{1}{i} \left(1 - \left(\frac{1}{(1+i)^n} \right) \right)}{E_s \left(\frac{1+i_e}{1-i_e} \right) \left[1 - \left(\frac{1+i_e}{1+i} \right)^n \right]} \quad 7.7$$

For the present case annual energy saving (E_s) by the solar cooker was estimated from the average daily useful energy requirement of cooking which reportedly varies in the range of 1.7 –2.7 MJ/person/day (Sunil and Tera, 2018). Thus, the average annual energy saving by the present solar cooker which is intended to prepare 80% of daily meal (i.e. other meals are considered as frying based foods for most of Ethiopian foods) for 10 persons can be estimated as follows;

Daily energy requirement for average person is $2.2\text{MJ} \times 80\% / \text{day} \cong 1.76\text{MJ}/\text{day}$

Daily energy requirement for 10 persons is $22\text{MJ}/\text{day} \cong 14.7\text{kW}$

Annual energy requirement for 10 persons is $8030\text{MJ}/\text{year} = 5365.5\text{kW}$

Therefore; annual energy saving by the solar cooker $E_s = 4960.2\text{birr}$

7.10.5 Payback Period of the Indirect Solar Cooking System

Payback period is defined as the time needed for the cumulative energy savings to equal the total initial investment, that is, how long it takes to get an investment back by savings in electricity or fuel.

Neglecting the maintenance cost, the payback period of solar cooker is evaluated when;

$$LCCS = CI @ n = Np;$$

$$C_f E_s \left(\frac{1 + i_e}{1 - i_e} \right) \left[1 - \left(\frac{1 + i_e}{1 + i} \right)^n \right] = CI$$

$$Np = \frac{\ln \left(1 + \frac{CI \left(\frac{i_e - i}{1 + i_e} \right)}{C_f E_s} \right)}{\ln \left(\frac{1 + i_e}{1 + i} \right)} \quad 7.8$$

Where LCCS is the life cycle saving of electric cost [Birr]

C_f is the unit cost of electricity for the first year of analysis (Birr/KWh)

Therefore; the payback of the present indirect solar cooker is **3.5 years**.

CHAPTER SEVEN

CONCLUSION AND RECOMMENDATION

8.1 Conclusion

In this study experimental investigation and performance evaluation of indirect solar cooking system using modified FPC as a heat source and integrated PCM based thermal energy storage system has been carried out. The components of the cooking system were designed, sized and fabricated based on per-determined capacity of the system. Numerical simulation was conducted for the modified FPC using ANSYS CFD software and simulation for the LHS unit was done using COMSOL Multiphysics 4.3a. The numerical model for the FPC considers single riser tube in the bottom of absorber plate cross section to predict the mean plate temperature and water outlet temperature. Charging and discharging simulation was conducted on the stearic acid based LHS system to capture variation of temperature at the inlet, outlet, middle and surface of the shell and tube storage and melt fraction of the PCM at varies melting and solidification time. Different experimental investigation and tests were conducted to evaluate performance of system components. Closed loop experiment was conducted on the developed prototype of FPC to investigate various performance parameters of the same with and without reflectors. Good agreement was obtained between the numerical and experimental results. The comparative study of energy and exergy efficiencies was investigated from the experimental results of FPC with and without reflector. Investigation on the enhancement of absorber plate temperature, storage water temperature, and cooking pot surface with the addition of reflectors were studied using stagnation no-load test. The effect of PCM based LHS on the system performance was investigated using charging and discharging experiment. Moreover, boiling, actual load and performance tests were conducted with the developed prototype of the indirect solar cooker. Finally, economic analysis is addressed for the cooking system. The results obtained from the numerical and experimental investigation of indirect solar cooking system components is presented in the following section.

i. Results from the numerical simulation and experimental analysis of FPC

- The daily average enhancement of solar radiation falling on FPC using four reflectors were estimated to be 42.3 % more than the one without reflectors.
- The optimal water velocity through the riser tubes were found to be 0.03m/sec.
- The numerical simulation of riser absorber assembly was independent of grid size.
- The error deviation between the estimated and experimental results of outlet temperature of water and mean plate temperature of FPC without reflectors were not

more than 3.2 % and 1.89% respectively. Whereas for the modified FPC the error deviation was 4.25% and 4.7% with the same mass flow rate of 0.02087kg/sec.

- The maximum average absorber plate temperature, maximum outlet water temperature, maximum energy and exergy efficiency of FPC without and with reflectors were found to be 364K, 344.1K, 63%, and 42% and 398.25K, 376.65K, 79%, and 48% respectively.
- From no load test at 13:00hr maximum stagnation temperature of absorber plate was 403K and a temperature of 372.9K and 369.1K was achieved in the storage vessel and cooking pot surface was respectively.
- The experimental results of no-load test establish the ability of the cooker for cooking.

ii. Result from the numerical and experimental analysis of LHS unit

- The simulated charging/discharging time of stearic acid was 120/130min respectively.
- When the stearic acid reaches to its melting point during charging process with in 141min (after the fluid temperature reaches 343.7k) the amount of sensible, latent and total energy stored in the PCM were 1.89MJ, 3.52MJ and 5.41MJ respectively.
- The amount of sensible, latent and total energy released during discharging process with in 153min were 1.57MJ, 3.01MJ and 4.58MJ respectively.
- Using stearic acid as a PCM, cooking time extends for 1 and half hour.

iii. Result from the experimental and cooking test on the developed and assembled prototype

- During water boiling test, the cooker used 458W to boil 3kg of water in 30min during the first boiling test. To boil the same mass of water in 36min it took 378W and to sensibly heat the same mass of water from 30°C to 65°C with in 42min it takes about 175W.
- Three batch cooking was possible until 19:00hr using different load of 1kg of pasta with 1.5kg water and 1kg of rice with 1.5kg of water.
- In the performance test the maximum amount of $F1$ was $0.09 \text{ K.m}^2/\text{W}$ and $0.21 \text{ K.m}^2/\text{W}$ for the cooker without and with reflector on FPC respectively. The second figure of merits of cooker with reflector was 0.28. the sensible, latent efficiency of the cooker was 29.6% and 27.1% respectively. The overall system efficiency was found to be 30.1%

The total investment cost of the cooker was 17360.8birr with annual electricity saving of 4960.2birr with pay-back period of 3.5yers.

8.2 Recommendation

Solar cooking system provides a solution to reduce consumption of electricity and bio-fuels. I recommend that institutions should train to develop new solar thermal technologies to minimize their need of high energy demand. For sure operation of solar cooker in large scale application I recommend auxiliary heating source to be included in the system in addition to thermal energy storage system to control the heating process in manageable manner. The present study addressed the experimental investigation and performance evaluation of modified FPC based indirect solar cooker integrated with PCM based thermal energy storage system. Some of future scopes and suggestions other than the present one for further study are;

- Analysis, design and economic feasibility study of an array of indirect solar cooker for large scale application can be addressed.
- Indirect solar cooking system using other type collector can be conducted.
- Flow in the system is considered to be laminar, the turbulent analysis of flow in the system can be performed to capture the full history of fluid flow.

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APPENDIX A

Design Parameters of LHS Unit

In the design of latent heat storage systems, effectiveness(ε) and number of transfer units (NTU) are principal design parameters. Effectiveness is the measure of how efficient the LHSS. According to **Joseph (2010)**, NTU of LHSS is defined only using the fluid side of the heat transfer coefficient(h_i), rather than the usual overall heat transfer coefficient in a conventional heat exchanger (HX). For a total length of tube L and number of tubes n in the HX, the equation for the NTU and the definition of the effectiveness applied to the LHSS is given by (**Shamsundar *et al.*, (1992)**);

$$NTU = \frac{\pi d_i h_i L n}{m_f C_p} \quad \text{eq A1}$$

$$\varepsilon = \frac{T_f - T_i}{T_m - T_i} \quad \text{eq A2}$$

Where d_i is inside diameter of tube

h_i is inside heat transfer coefficient

L is length of tube

n is number of tubes

m_f and C_p are the mass flow rate and specific heat capacity of HTF

T_f and T_i is outlet and inlet temperature of HTF

T_m is melting point of PCM

In sizing latent heat storage unit, the following dimensionless parameters should be well defined.

$$\text{Parameter } (\beta) = \frac{h_i R_i}{K_{pcm}} \quad \text{eq A3}$$

$$\text{Biot number (Bi)} = \frac{h_i R_o}{K_{pcm}} \quad \text{eq A4}$$

$$\text{Nusselt number Nu} = \frac{h_i R_o}{K_f} \quad \text{eq A5}$$

where; K_{pcm} and K_f are thermal conductivity of the PCM and the HTF respectively;

r_i and r_o are the inner and outer radius of HTF tube.

To estimate the parameter β , the Nusselt number Nu is estimated first depending on the type of flow regime in the tube being either laminar or turbulent.

For laminar flow inside HTF tubes, the value of $Nu = 1.835$

$$\text{For turbulent flow } Nu = 0.023 Re^{0.8} Pr^{0.4} \quad \text{eq A6}$$

$$\text{Thus, } \beta = \text{Nu} \left(\frac{K_f}{K_{PCM}} \right) \left(\frac{R_i}{R_o} \right) \quad \text{eq A7}$$

The outside radius R_o of the tube heat exchanger can be determined by (Joseph, 2010);

$$R_o = \left(\frac{K_{PCM}(T_m - T_{init}) \cdot t}{\tau \rho_{PCM} L_f} \right)^{1/2} \quad \text{eqA8}$$

Where t = storage period;

ρ_{PCM} = density of the PCM;

L_f = heat of fusion of the PCM;

K_{PCM} = thermal conductivity of the PCM;

T_m = melting point of the PCM;

T_{init} inlet temperature of HTF;

τ is a non-dimensional time variable which depend on the parameter β and initial PCM solidification fraction (Fo). In the process of sizing the LHS unit, the inner diameter of tube, length of the tube and diameter of shell were simply chosen depending on the volume of PCM which is required to obtain the value of required storage volume of LHS unit.

APPENDIX B

Calibration of the Thermocouple Thermo-Meter

Thermocouples are sensors used to measure temperature by forming a junction between two wires of separate metals. One end of the junction is called the measurement “hot” junction and the other end is called the reference “cold” junction. When there is a temperature variation in the joint, voltage is created and this voltage can be converted into temperature with a thermocouple reference table or data logger. Before taking any measurement in the process of testing the experiments, calibration of measuring device is compulsory. In the present work the thermocouple was calibrated using boiling water experiment. Thermocouples can be calibrated using a reference thermometer in which a temperature calibration test was conducted by comparing both measurement values of a reference thermometer and k-type thermometers. The thermocouple used in this study was a K-type four channel thermocouple and a constant high temperature water bath was used for the calibration.



Fig.B.1 Thermometers a) Reference thermometer (WL-362) b) K-type 4-channel thermocouple thermometer

K-type thermocouple specifications

Serial number: HT-9815

Temperature range: 0-800°C

Power supply: 9V

Weight: approx. 250g

Table B.1 Data from thermocouples during calibration.

Time	T _{w1} (WL-362) (°C)	T _{w2} (K-type thermocouple) (°C)	Uncertainty (%) $\frac{(T_{w1}-T_{w2})}{T_{w1}} * 100$
1	30.1	29.8	0.99668
3	35.6	35.4	0.5618
5	47.3	47.1	0.42283
7	57.1	56.8	0.52539
9	66.3	65.9	0.60332
11	72.6	72.1	0.68871
13	76.8	76.4	0.52083
15	79.8	79.5	0.37594
17	82.6	82.1	0.60533
19	84.3	83.9	0.4745
20	86.6	86.2	0.46189
22	88.1	87.8	0.34052
24	90.5	90.1	0.44199
26	91.3	90.9	0.43812
Average uncertainty			0.53

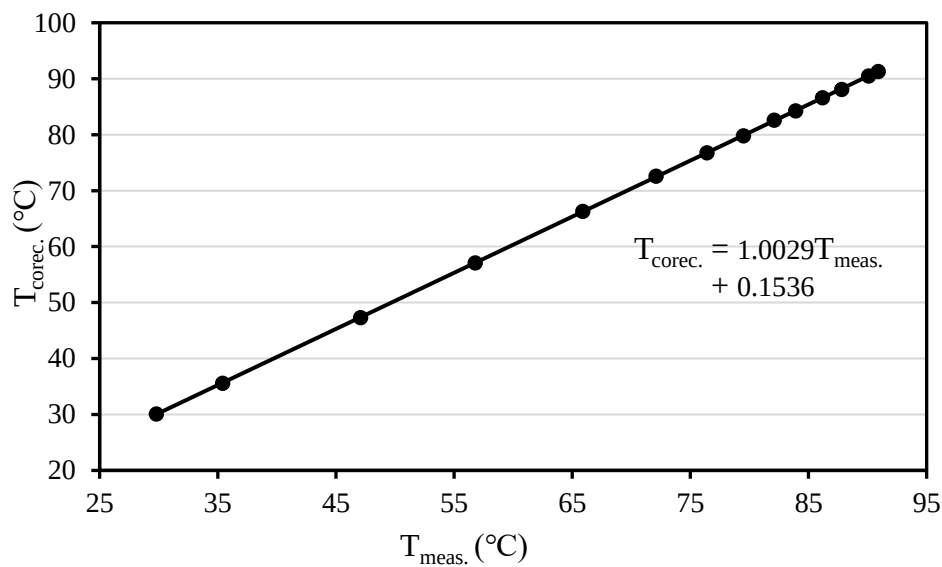


Fig.B.2 Plots of corrected vs measured result

APPENDIX C

Total Cost of Indirect Solar Cooking System

Table C.1 List of components costs

No.	Item	Specification	Qty	Unit price [Birr]	Total price[birr]
1	Flat plate collector				
1.1	Absorber plate	(2mx1m) copper plate, 0.5mm thickness	1	2000	2000
1.2	Covering Glass	(2mx1m) plain window glass 5mm thickness	1	600	600
1.3	Casing	2.1mx1.1m wooden casing	1	500	500
1.4	Insulating material	5mx2m rubber foam 2.5mm thick	1	280	280
1.5	Header tubes	1m length 22mm dia. 1.25mm thick steel tube	2	80	160
1.6	Riser tubes (copper)	15.75m coiled tube 10mm dia. 0.5mm thick	1	2000	2000
1.7	Reflective aluminum foil	2m x 45cm($\rho=0.86$)	5	60	300
1.8	spray paint	Black	1	100	100
	selective black coating	chalk board black paint	1	150	150
1.9	silicon sealant	GSM brand	1	110	110
1.10	Frame and support	Square pipe steel	1	500	500
					Sub-total = 6509
2	Cooking unit				
2.1	Cooking pot	15cm length 20cm dia. Aluminum vessel	1	300	300
2.2	Storage vessel	12cm length 32cm diameter Aluminum vessel	1	300	300

2.3	Box casing	Sheet metal	-	100	100 Sub-total =700
3	Thermal storage system (Shell and tube) LHS (PCM)				
3.1	Stearic acid	25 Kg	1	2587	2587
3.2	Insulating thermo foam	(5x2) m ²	1	300	300
3.3	Cylindrical shell	0.8m length galvanized sheet metal 1.2mm thick	1	300	300
3.4	Steel Tubes	0.8m length 16mm dia.	6	25	150 Sub-total=3337
4	Other connecting and miscellaneous costs				
4.1	Connecting polyester pipes	3m	2	130	260
4.2	Elbows	¼ inch Polyester elbow Steel elbow 1 ¼ inch	8 1	25 92	492
4.3	Reducer	1 ¼ inch to ¼ inch steel	1	90	90
4.5	Pump	Electric driven 0.5hp centrifugal pump	-	-	2500 sub-total=3342
					total=13,888
5	Manufacturing, labour and assembly cost	25% material cost			3472
6	Operating(power), maintenance and replacement cost	10% of investment cost			1736.8
					Total cost = 18,748.8(birr)

APPENDIX D

Photos from experimental setup during cooking test



Fig.D. 1 Photographs from cooking test