Solar Biomass Hybrid Dryer

A Final Year Project Report

Presented to

SCHOOL OF MECHANICAL & MANUFACTURING ENGINEERING

Department of Mechanical Engineering

NUST

ISLAMABAD, PAKISTAN

In Partial Fulfillment
of the Requirements for the Degree of
Bachelors of Mechanical Engineering

by

Nouman Iqbal
M.Abdullah Sarfraz
Zain Haider

June 2018

EXAMINATION COMMITTEE

We hereby recommend that the final year project report prepared under our supervision by:

Nouman Iqbal	NUST201433145.
M.Abdullah Sarfraz	NUST201432238.
Zain Haider	NUST201432367.

Titled: "Solar Biomass Hybrid Dryer" be accepted in partial fulfillment of the requirements for the award of BE Mechanical Engineering degree.

Supervisor: Professor Dr Shahid Ikramullah Butt (HoD Research)	________ Dated:
Committee Member: Dr Emad Uddin (HoD) (Assistant Professor)	________ Dated:
Committee Member: Engr. M. Naweed Hassan (Lecturer)	________ Dated:

______________________ ________

(Head of Department) (Date)

COUNTERSIGNED

Dated: ________________ __________________________

(Dean / Principal)

ABSTRACT

Poor drying and processing techniques used by the farmers have led to loss of quantity and quality of agricultural crops. Drying is one of the technique for protection of fruits and vegetable but also an energy consuming process. Solar energy is one of the pillars for environment friendly energy. The downside of the solar energy is the interruption during the night and cloudy and wet weather. To counteract this and provide continuous operation an improved version of Solar Biomass Hybrid Dryer is developed. This project presents the design, construction, analysis and performance of a mixed-mode solar dryer for food preservation. The drying chamber receives hot air from two sources. One source is the solar collector and the other source is the Biomass heat exchanger. The system can be modified to cater different fruits and vegetables that require drying temperature in the range of 40 to 60 ^{o}C by varying the flowrate of the inlet atmospheric air in solar collector and heat exchanger. The main goal is to develop an efficient and cost effective dryer manufactured from locally available raw materials. The main components of the dryer are flat plate solar collector, heat exchanger, drying chamber and fans. To achieve maximum efficiency a new design of heat exchanger and drying chamber is developed. Analysis on ANSYS CFD and COMSOL is carried out to further verify the numerical results from solving equations on MTALAB. It is observed that the use of this type of dryer reduce the dehydrating time significantly and provides better product quality compared with orthodox drying method.

PREFACE

It is a great opportunity for us to have the Bachelors of Mechanical Engineering at SMME, NUST. In the accomplishment of degree, we are submitting a project report on "Solar Bio-Mass Hybrid Dryer for Food Preservation".

Crop drying is a process which consumes a large amount of energy, and yet it's a very important process because it is used for preservation purposes. It is necessary to dry crops for storage purposes because around 20% of world's grain crops (40% in Pakistan) are wasted because of inadequate post processing and poor post-harvest technology implementation. Therefore, a method is needed to preserve them for a longer period of time. The most effective method is removal of moisture from the crops, which thereby reduces its tendency of spoilage. Drying is one of the most significant preservation techniques for preservation of fruits and vegetables but also an energy consuming process. Current deficiencies of fossil fuels have encouraged the world to shift focus towards the renewable energy. Solar energy combined Bio-mass is one of the most important types of renewable energy and it is more desirable because it is abundant, infinite, and environment friendly.

To address this problem, we have developed a solar bio-mass hybrid dryer which operates on the principle of removal of moisture from food items by the application of heat and air flow. Ambient air is entered into a multi flow solar collector and bio-mass burner which heats it up to a temperature of 45 to 60 degrees. This heated air is then transferred into a drying chamber where food products are placed and it results in removal of moisture from food items.

Subject to the limitation of time, every possible attempt has been made to study the problem deeply and propose an appropriate solution. The data was acquired from experimentation and research articles, further analyzed and interpreted and finally the results were obtained.

ACKNOWLEDGMENTS

We are appreciative to our Creator ALLAH Almighty who has guided us all through this work at each progression and each new believed that struck a chord to enhance it. To be sure, we could have done nothing without HIS precious help and direction.

We are additionally grateful to our parents who were a constant source of motivation and support for us.

We might want to pay exceptional on account of our Supervisor Dr. Shahid Ikram-Ullah who has helped and guided us all through the project. We are highly thankful to him for suggesting valuable ideas and providing scientific directions to implement these ideas and taking the project to the pinnacle of success.

We are thankful to Dr. Emad Uddin and Mr. Hafiz Abdur Rehman and who guided in various aspects of projects.

Last but not the least, we would like to thank our university "National University of Sciences and Technology (NUST)", specifically the department of Mechanical and Manufacturing Engineering (SMME) for their cooperation towards the completion of our project.

ORIGINALITY REPORT

We hereby declare that no portion of the work of this project or report is a work of plagiarism and the workings and findings have been originally produced. The project has been done without any partnership and has not been a support project of any similar work serving towards a similar degree's requirement from any institute. Any reference used in the project has been clearly cited and we take responsibility if found otherwise.

Final_Report

ORIGINALITY REPORT

16%	11%	9%	9%
SIMILARITY INDEX	INTERNET SOURCES	PUBLICATIONS	STUDENT PAPERS

PRIMARY SOURCES

1	www.omicsonline.org Internet Source	2%
2	de.slideshare.net Internet Source	1%
3	energsustainsoc.springeropen.com Internet Source	1%
4	core.ac.uk Internet Source	1%
5	Submitted to Technological Institute of the Philippines Student Paper	1%
6	www.urbanprepping.com Internet Source	1%
7	Sharma, A.. "Solar-energy drying systems: A review", Renewable and Sustainable Energy Reviews, 200908/09 Publication	<1%
8	Proceedings of ISES World Congress 2007 (Vol I – Vol V), 2009.	<1%

Publication

9	uotechnology.edu.iq Internet Source	<1%
10	Submitted to iGroup Student Paper	<1%
11	Al-Kayiem, Hussain H., and Tadahmun A. Yassen. "On the natural convection heat transfer in a rectangular passage solar air heater", Solar Energy, 2015. Publication	<1%
12	Energy Systems in Electrical Engineering, 2016. Publication	<1%
13	ijrer.org Internet Source	<1%
14	Sain, Preeti, Vipin Songara, Rajan Karir, and Neha Balan. "Natural convection type solar dryer with latent heat storage", 2013 International Conference on Renewable Energy and Sustainable Energy (ICRESE), 2013. Publication	<1%
15	article.sciencepublishinggroup.com Internet Source	<1%
16	Submitted to Cranfield University Student Paper	<1%

17	Tiwari, Sumit, G.N. Tiwari, and I.M. Al-Helal. "Performance analysis of photovoltaic–thermal (PVT) mixed mode greenhouse solar dryer", Solar Energy, 2016. Publication	<1%
18	docplayer.net Internet Source	<1%
19	scholar.sun.ac.za Internet Source	<1%
20	Sharshir, S.W., M.O.A. El-Samadony, Guilong Peng, Nuo Yang, F.A. Essa, Mofreh H. Hamed, and A.E. Kabeel. "Performance enhancement of wick solar still using rejected water from Humidification-Dehumidification unit and film cooling", Applied Thermal Engineering, 2016. Publication	<1%
21	Submitted to International Islamic University Malaysia Student Paper	<1%
22	Sangamithra, A., Gabriela John Swamy, R. Sorna Prema, R. Priyavarshini, V. Chandrasekar, and S. Sasikala. "An overview of a polyhouse dryer", Renewable and Sustainable Energy Reviews, 2014. Publication	<1%
23	Submitted to University of KwaZulu-Natal	

Student Paper

<1%

24	www.arunmujumdar.com Internet Source	<1%
25	Koilraj Gnanadason, Moses; Senthil Kumar, Palanisamy; Wilson, Vincent H.; Hariharan, Gajendiran and Shenbaga Vinayagamoorthi, Navaneethakrishnan. "Design and Performance Analysis of an Innovative Single Basin Solar NanoStill", Smart Grid & Renewable Energy, 2013. Publication	<1%
26	Othman, M.Y.H.. "Development of advanced solar assisted drying systems", Renewable Energy, 200604 Publication	<1%
27	Müller, Joachim, and Werner Mühlbauer. "Solar Drying", Modern Drying Technology Volume 4 Energy Savings, 2012. Publication	<1%
28	www.interesjournals.org Internet Source	<1%
29	www.ijert.org Internet Source	<1%

Exclude quotes	Off	Exclude matches	Off
Exclude bibliography	Off		

COPYRIGHT

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

English Letters

abs	Solar Absorptivity	
A	Area	m^2
AirGap	Gap for the Air Flow Inside Solar Collector	m
B	Thermal expansion coefficient	K^{-1}
e	Emissivity	
L_c	Gap for the Air Flow Inside Solar Collector	m
L_p	Length of Plate	m
cp	Specific Heat Capacity	J/kg^oC
dt	Time Step	s
dT	Change in Temperature	^oC
g	Gravitational Acceleration	m/s^2
$h_{c_{a-g}}$	Convection Heat Transfer Coefficient Between Air and Glass in Solar Collector	W/m^2K
$h_{c_{g-sky}}$	Convective Heat Transfer Coefficient Between Glass and Sky Inside Solar Collector	W/m^2K
$h_{c_{p-a}}$	Convective Heat Transfer Coefficient Between Plate and Air Inside Solar Collector	W/m^2K
$h_{c_{p-a}}(\text{NaturalConvection})$	Natural Convective Heat Transfer Coefficient Between Plate and Air Inside Solar Collector	W/m^2K
$h_{c_{p-a}}(\text{ForcedConvection})$	Forced Convective Heat Transfer Coefficient Between	

	Plate and Air Inside Solar Collector	W/m^2K
$h_{r_{p-g}}$	Heat Transfer Coefficient Between Plate and Glass in solar collector	W/m^2K
I	Solar Intensity Directed on Glass	W/m^2
k_{air}	Thermal Conductivity of Air at 30ºC	W/mK
Nu_{g-sky}	Nusselt Number for Convection Between Glass and Ambient Air	
$Nu(NaturalConvection)$	Nusselt Number for Natural Convection in Solar Collector	
$Nu(ForcedConvection)$	Nusselt Number for Force Convection in Solar Collector	
Pr	Prandalt Number at 30ºC	
$Q_{c_{a-g}}$	Rate of Convection Between Air and Glass in Solar Collector	W
$Q_{c_{g-sky}}$	Rate of Convection Between Sky and Glass in Solar Collector	W
$Q_{c_{p-a}}$	Rate of Convection Between Plate and Air in Solar Collector	W
Q_{loss}	Rate of Conduction Between Plate and Ambient Air for Solar Collector	W
$Q_{r_{g-sky}}$	Rate of Radiation Between Sky and Glass in Solar Collector	W
$Q_{r_{p-g}}$	Rate of Radiation Between Plate and Glass	

	in Solar Collector	W
Ra_L	Raleigh Number Inside Solar Collector	
trans	Transmissivity of Glass	
T	Temperature	ºC
U_{loss}	Overall Heat Transfer Coefficient for	
	Conduction in Solar Collector	W/m^2 ºC
V	Volume	m^3
D_h	Hydraulic diameter	m
v	Velocity	m/s
Re	Reynolds's number	
R_f	Fouling Factor	m^2 ºC/W
k	Thermal Conductivity	W/m ºC
L	Plate Length	m
U	Overall Heat Coefficient	W/m^2 ºC
ΔT_{ln}	Log Mean temperature	m^2 ºC/W
Q	Heat Transfer	W
η_{fin}	Efficiency of Fins	
A_c	Cross Section Area of Duct	m^2
l	Length	m
w	Width	m
p	Perimeter	m
A_s	Surface Area of Duct	m^2
f	Friction Factor	
P	Pressure	N/m^2

$\dot{V}$	Volume Flow Rate	m^3/s
H	Head Loss	m
P_l	Pressure Loss	N/m^2
P_{fan}	Power of Fan Required	W
m_a	Mass of air	kg
M_w	Mass of Water to be Removed	kg
M_d	Mass of Bone Dry Product	kg
Q	Total Energy Required for Drying	KJ
C_p	Specific Heat of Water	KJ/kg ºC
C_d	Specific Heat of Product	KJ/kg ºC
T	Drying Time	s
ERH	Equilibrium Relative Humidity	
a_w	Water Activity	

Greek

Θ	Angle of tilt of glass from the horizontal	
ρ	Density	kg/m^3
μ	Dynamic viscosity at 30ºC	kg/ms
σ	Steffan Boltzman constant	
Δ	Change	
ε	Surface Roughness	m
ε/D	Relative Surface Roughness	
λ	Latent Heat of Vaporization	KJ/k

Subscript

a	Air
g	Glass
p	Plate
sky	Sky
in	Inside
out	Outside
h	Hot
c	Cold
pr	Product
i	Intial
f	Final
wb	Wet Basis
db	Dry Basis

CHAPTER 1- INTRODUCTION

1.1 Motivation

Crops in the untreated form are perishable being at risk to biological and physical deterioration. Conservancy is vital for all food harvests in order to preserve desired nutritious level. Lack of infrastructure for storage and selling in many countries of the Asia region results in a high amount of wastage of agricultural food, which average between 10 and 40 % [1]. Though Pakistan is a major producer of agricultural food, many Pakistanis deprive of their everyday requirement of fruits and vegetables and the Human Development Index is very low. Substantial amounts of vegetables and fruits produced in the country go to waste due to improper postharvest operations and the shortage of processing [1]. Every year considerable quantities of food crops are gone astray due to spoilage. The reasons for the wastage include unawareness about appropriate methods of preservation, non-availability of preservation services and inadequate transportation system during harvest season. Different methods for foodstuff preservation are vacuum packing, freezing, canning, food irradiation, and drying. Dehydrating is the most prevalent method of food preservation [2]. Drying to a low and safe moisture content level delays microorganism's growth within the material. Drying of farming products is an important operation under postharvest stage. The drying process is fruitful from ancient time when it was a matter of necessity to this time that properties of food, quality of product, and method of drying studied for effective drying. The zeal to learn more and contribute from our part has motivated us to pursue Solar Biomass Hybrid Dryer as our Final Year Project.

1.2 Problem Statement

Food experts have discovered that by decreasing the humidity content of foodstuff to between 10 and 20%, bacteria, mold and enzymes are prevented from spoiling it. The taste and most of the nutritious value is concentrated and conserved [3]. It is traditional to produce grain crops during a dry period by simple drying methods such as sun drying. Conversely, majority of the crop does not always correspond with a suitably dry period. The usual method widely used throughout the world is open sun drying where crops, such as fruits, vegetables, cereals, grains, etc. are spread on the ground until satisfactorily dried so that they can put in storage safely. Harvest temperature when dehydrating in the sun ranges from 10 to 20 ºC above ambient temperature. Factors that affect drying period and rate are initial humidity content, desired final humidity content, intensity of sun and season.

Hii et al. [4] have presented that crops dried under direct sunlight are cheap, but the product obtained by is of inferior quality due to contamination by insects, dust, rain and bird. In addition, loss of nutrients, vitamins and unacceptable color changes due to direct contact to ultraviolet rays, and longer time to dry. The shortcomings of open solar drying need a suitable technology that can help in improving the quality as well as the quantity of the dried products and in decreasing the wastage. This led to the application of different types of drying approaches like solar dryers, electrical dryers, and oil-burned driers. However, the high cost of power and oil and their shortage in the rural areas of most third world countries have made some of the mentioned driers very unattractive.

Solar dryers control the drying practice and protect agricultural products from insect, dust and rain. Umogbai et al. [5] made an assessment between sun and solar drying and acquired that the solar dryers create much higher temperatures, lower moisture, lower product moisture content and reduced spoilage during the drying process. Rajeshwari and Ramalingam [6] verified that dehydrating time in case of solar dryers compared to open drying decreased by about 40 % and yields improved value foods. The sun being discontinuous in supply can be enriched by other sources of energy such as biomass. Thus, Solar Biomass serves as the best way to counteract all the above-mentioned effects studied as our project.

1.3 Objectives

The aim of this study is to develop an improved hybrid dryer in which the items are dried by both solar energy and biomass fuel (wood). The current study focus on the need of the hybrid dryer, which is reliable in terms of design, financial aspect and fabrication aspect.

To achieve the above mention objectives a stepwise procedure carried as follow:

1. Literature review: Literature review highlights the importance of preservation of food, different methods used for food preservation, their merits and demerits. Different types of solar and biomass dryers are discussed in this part alongside the phenomena of drying and the factors affecting it.

2. Methodology: This phase covers the steps that we followed to choose one design over the other. All the alternatives to the chosen design are discussed.

3. Design: The SolidWorks model is discussed in detail with the 2D drawing of each part.

4. Numerical analysis: Mathematical modelling of solar collector, heat exchanger and drying chamber given in this section. These are solved in MATLAB. CFD analysis carried on each part so that a more realistic picture of the experiment is given and to verify the results which we got from MATLAB.

CHAPTER 2- LITERATURE REVIEW

Literature review highlights the importance of preservation of food using solar and biomass energy. Different types of solar dryers and biomass heaters are also discussed in this part alongside the phenomena of drying and the factors affecting it. A study of different insulation material is also carried out.

Drying of crop for moisture removal is necessary for safe storage of agricultural products for long period of time. According to research conducted by United Nations Organization approximately 20% of wheat produced is wasted because post-harvest technology has not been implemented in true letter and spirit. Agricultural products with moisture level from 18% to 40% [1] are ideal to harvest and then are subjected to drying process to a level of 7% to 11% according to the market application and safe storage for a long period of time before utilization.

With increased awareness of application of renewable energy resources, developing countries like Pakistan can get benefit for increasing the productivity of agricultural products. As a result of poor awareness to modern technology, processing, infrastructure and marketing developing countries like Pakistan suffer from wastage of at least 10% to 40% of agricultural products unable to meet the daily demand pushing the Human Development Index (HDI) to below average [1].

Solar energy is a standout amongst the most sustainable power source on the planet contrasted with non-sustainable energy sources for removing moisture of agricultural products. The idea of a dryer controlled by sun power is ending up progressively possible due to steady lessening in cost of solar cells combined with the expanding worry about air pollution caused by ordinary non-renewable energy sources utilized for drying crops. Sunlight based dryers utilized as a part of agribusiness for sustenance and yield drying for modern drying process, dryers can be turned out to be most helpful gadget from conserving energy perspective. It saves energy as well as time, involving less space, enhancing quality of the item, making the procedure more effective and environment friendly.

Solar powered dryer is utilized for removing moisture content to a desired level with efficiency and reliability as every agricultural product has unique moisture content. So necessity of solar dryer is likewise extraordinary as drying time and conditions vary according to product.

2.1 Principle Operations of Solar Dryer

The fundamental operations utilized in a solar dryer are:

Conversion of Light Energy to Heat Energy: Any darkness within solar dryer will enhance the viability of transforming light into heat energy.

Trapping Heat Energy: Confining the air inside the dryer from the air outside the dryer has an imperative effect. Utilizing a strong, similar to a plastic sack or a glass cover, will enable light to enter, however once the light is consumed and changed into heat, a plastic pack or glass cover will trap the heat inside. This makes it conceivable to achieve comparable temperatures on frosty and blustery days as on hot days.

Absorption of Heat Energy by Food: Both the natural convection dryer and the forced convection dryer utilize the convection of the warm air to move the heat to food. Drying is basically the operation of thermally expelling water substance to yield a desired product. Bound Moisture - Moisture bonded to the microstructure of the solid exerting the lesser vapor pressure than the pure liquid, and, the Unbound Moisture- excessive moisture after bound moisture, are two important factors affecting the process of drying. Drying process involves:

A. Transfer of heat energy from the environment to solid product for moisture removal.
B. Transfer of moisture to the surface and afterward the vanishing from the surface by evaporation. The rate of drying can be directed at which the two processes continues.

2.2 Types of Solar Dryer

(1) Direct type

(2) Indirect type

2.2.1 Direct Type Solar Dryer:

Crop yields are exposed to sunlight directly. The crops covered with a transparent material are exposed to solar radiations causing an increase in temperature converting the light to heat energy. For instance, the best nature of apricots (profound orange apricots) must be gotten if the all the greenish shade of them is annihilated. This green shading is fundamentally because of chlorophyll. Presentation to bright sun oriented beams can create this impact.

The primary drawback of this form of sun based dryer is that the product temperature cannot be controlled in it due to the immediate introduction. Accordingly, it is extremely hard to keep up an ideal drying rate and product quality.

Solar based dryer is basic in development and is cheaper so it is broadly mainstream among all. Its range can differ from a couple of kg's to numerous tons.

2.2.2 Indirect Type Solar Dryer:

Solar radiations not directly falling on the crops, the solar dryer is then of indirect mode. In such type of driers, a solar cell is used, which absorbs the solar radiation and converts it into heat energy. There is a usually an absorber plate in collector. Air is heated as it is passed over this absorber plate. This heated air is utilized for drying purposes and in transferred into a drying chamber containing crops.

This type of dryer is preferred for the crops which can be affected by the direct exposure to solar radiation and their final quality is reduced. High intensity of sunlight can cause case hardening and uncontrolled moisture removal which affects the nutritional value of product. Cardamom crop is perfect example of this phenomenon. When exposed to direct sunlight, its pods split resulting in destruction of chlorophyll in it. Premium quality of cardamom pods has a greenish color. Hence, an indirect solar dryer is an appropriate solution for such crops. It can easily achieve high and controllable temperatures that are optimum for drying.

The main disadvantages of this dryer are complex structure and high cost. Its range can vary from a few kg's to many tons.

2.3 Advantages of Solar Drying over Conventional Drying

A solar dryer offers six basic advantages over conventional drying system as studied by O. V. Ekechukwua and B. Norton [7]:

1. It is less time consuming. Materials can be dried in a shorter timeframe. Solar based dryers improve drying times in two ways. Firstly, the translucent, or straightforward traps heat inside the dryer, raising the temperature of the air. Also, the adaptability of growing the solar heat accumulation zone makes this process more energy efficient.
2. It is more effective. Since materials can be dried all the more rapidly, less will be lost to waste promptly after collection. This is particularly valid for items that require prompt drying, for example, naturally collected grain with high water content. Along these lines, a bigger level of items will be accessible for human utilization. Likewise, less of the product will be lost.
3. It is clean. Since materials are dried in a controlled situation, they are more averse to be contaminated, and can be put away with less probability of the development of dangerous parasites.

4. It is more advantageous. Drying materials at ideal temperatures and in a shorter measure of time empowers them to hold a greater amount of their dietary esteem, for example, vitamin C.
5. A special reward is that items will look better, which improves their attractiveness and henceforth gives better budgetary comes back to the agriculturists.
6. It is modest. Utilizing openly accessible solar energy rather than traditional energizes to dry items, or utilizing a supplementary supply of solar heat, so diminishing customary fuel request can bring about noteworthy cost investment funds.

2.4 Classification of Drying Process Based on Type of Air Flow

1. Natural Convection
2. Forced Convection

2.4.1 Natural Convection:

In this method no fan or blower is used, air flows naturally. It takes more time for drying the food, compare to active type. As the airflow is natural, there is no need of external power supply. Its drying rate is limited. It plays vital role in drying sector because of its cost competitiveness and hence more popular. However, natural convention has a limited capacity and one of the manufactured dryer of this type can be used for drying of one type of item only.

2.4.2 Forced Convection:

Fan or the blowers are the main object that are used in forced convection type solar dryer. This method is not used in many countries as it needs electricity to run the fans. It has many advantages. Forced convection solar dryers are more effective for drying food at ambient temperature. It also takes less solar assisted drying segmentation process that uses most attractive and cost effective materials or commonly used in agricultural and marine sectors. They are highly efficient. Air and water based collectors are used in this system. This method takes less time to dry the food compare to natural convection and is more versatile in nature as by varying flowrate different items can be dried.

2.5 Important Parameters Controlling Drying Process

Following are the important parameters that are concerned with solar drying system.

2.5.1 Temperature:

Each leafy food has a basic temperature above which a burned taste develops. The temperature ought to be sufficiently high to remove moisture from the product, yet not sufficiently high to cook it. Amid the drying procedure, at first the air temperature is generally high, that is, 150 degrees to 160 degrees F. (65 degrees to 70 degrees C.), with the goal that moisture can reduce rapidly from the product. Since it loses heat during drying, the air temperature can be high without expanding the temperature of the food product. In any case, when surface moisture is lost (the outside becomes dry) and the rate of moisture reduction slows down, the product heats up. The air temperature should be decreased to around 140 degrees F. (60 degrees C.) towards the finish of the drying procedure, to protect the product from getting burnt.

2.5.2 Humidity:

The higher the temperature and the lower the moisture, the more the rate of moisture removal will be. The air with greater moisture content reduces the rate of evaporation. In the event that drying happens too quick, be that as it may, "case solidifying" will happen. This implies the cells on outer surface loses the moisture more rapidly than the cells within. The surface turns out to be hard, damping the moisture loss from within. It should be ensured that the ventilation around product and the dryer is sufficient.

2.5.3 Velocity of Air:

Velocity is inversely proportional to the drying time. In the solar dryer velocity is high then drying time is less but if velocity is high then it will take less time to dry the product. In solar dryer at starting where the fan is provided, velocity of inlet air is high but after covering some distance velocity of air reduces.

2.5.4 Solar Collector Area:

Collector area depends on the product quantity. If product quantity is more then, large collector area is needed, but we can design it for maximum area of product come in contact with air.

2.5.5 Sunshine Hours:

Sunshine hours means maximum of daytime duration for a given location. On an average normal day, there are up to 8 to 6 sunshine hours, but it is different in different season. In summer sunshine hour is more than compare to monsoon and winter.

2.5.6 Pressure Drop of Air Between Inlet and Outlet:

Pressure drop is the result of frictional losses, caused by the flow resistance as the fluid flows through the tube. In case of solar dryer when air enters in the dryer it is dry and its pressure is high. After covering some distance, it becomes wet because of absorbing the moisture from the product, so the pressure of air reduces. High flow velocities result in a larger pressure drop across a section of object. Low velocity will result in lower or no pressure drop.

2.5.7 Drying Force:

Air is the main element in drying processes for transport of heat and vapor. The evaporation heat required can be transferred via heating coils or air. Water evaporated is removed by convective currents driving the colder air in and heated air out. The drying force is fundamentally the difference

between the vapor pressure noticeable of drying air and the saturation pressure at the similar temperature.

The drying force can be expressed as:

$$\Delta F = P_{sw} - P_w$$

where,

ΔF = Drying Force, (mbar)

P_w = vapor pressure (mbar)

P_{sw} = saturation vapor pressure (at the actual dry bulb temperature), (mbar)

2.5.8 Constructional Material:

Constructional material is different in different dryer but selection of material is also important because in direct type solar dryer covering material should have high transmissivity so it can transmit more amount of solar radiation and temperature air in dryer can become high as much as possible.

2.5.9 Solar Radiation:

It is radiation energy from the sun, particularly electromagnetic in nature. About half belongs to shorter wavelength part of the spectrum that is visible while the rest is mostly in the near-infrared part and a small fraction in the ultraviolet part of the spectrum.

2.5.10 Airflow Rate:

It is a measurement of the amount of air per unit of time that flows through a particular area. The volumetric or weight fraction of air can be measured but typically measured in volumetric units, but for some applications it is customary measured by mass, as its volume can vary with temperature.

$$\text{Airflow rate (Q)} = \text{Velocity} * \text{Area}$$

Erick et al. [8] through experiments demonstrated that the air mass flow rate is the key factor affecting the efficiency of overall drying process. Stagnation temperature on the collection plate results from inadequate mass flow rate of air thereby affecting the removal of heat from collection plate. As the air velocity is directly proportional to the coefficient of heat transfer so the thermal efficiency of drying is significantly affected by air flow rate.

2.5.11 Insulation:

Proper insulation is necessary to control heat losses and for effective storage of heat energy. Abdul Jabbar N Khalifa et al. [9] studied the importance of insulation on production for a thickness of up to 60mm.

2.5.12 Glass Cover Thickness:

J.S. Gawande et al. [10] studied the effect of various thicknesses of glass cover in solar drying. Lower the thickness of glass cover greater will be the production in case of solar application.

2.6 Study of Practically Implemented Solar Dryers

Research has been going on different designs of solar dryers around the globe. Several improvements and modification have been implemented and studied thoroughly. In this section, a general overview of some previous work in the world of solar dryers is presented. Different prototypes and actual models are studied in detail so that a new prototype can be built which is more efficient and cost effective. Some of the studied material is discussed below:

2.6.1 Tent Type Solar Dryer:

Arinze EA et al. [11] carried out study on tent dryer (Figure 1). Tent type solar drying units are cost efficient and easy to fabricate consists of wood shafts covered with plastic sheet. Dark plastic is utilized on the divider confronting far from the sun. The item to be dried is put on a support over the ground. It requires same time for drying of items as in outdoors drying. The fundamental reason for the dryers is to give assurance from tidy, earth, rain, wind or predators and they are normally utilized for organic product, fish, espresso or different items for which wastage is generally high. Tent dryers can likewise be brought down and put away when not being used. They have the drawback of being harmed by strong winds.

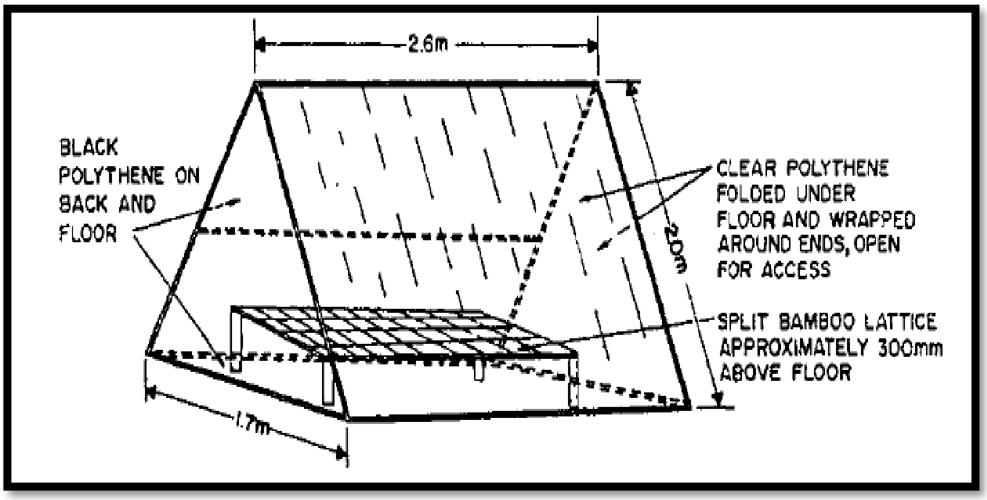


Figure 1 - Tent Dryer [11]

2.6.2 Cabinet Type Solar Dryer:

Prakash S et al. [12] studied Cabinet Dryer (Figure 2) which is an extensive wooden or metal box and the item is placed in trays or shelves inside a drying cabinet. Integral type also called the direct type is the one in which the cabinet is transparent. On the other hand, if the cabinet is opaque the dryer is indirect type or also called distributed type. Most of the time, the air is heated as it flows across the low pressure drop solar cell through the duct over the trays with the crop placed on its top.

The humid air is then released through air vents or a stack. It is ought to be insulated to limit heat losses and must be durable. It shall be constructed from metal sheets or water resistant cladding, e.g paint or tar, is suggested. Heated air streams through the pile of plates until the point that the entire product is dry. As the hot air enters through the base plate, this plate will dry first. The last plate to dry is the one at the highest point of the chamber.

The benefits of this framework are:

• Simple in development
• Low work costs
• Simple loading and unloading
• The product does not get directly exposed to the sun
• Heat storage frameworks can be applied

The drawbacks of this framework are:

• An inclination to over-dry the lower plate.

• Lower efficiency, as far as fuel utilization is concerned, in the later phases of drying when majority of the plates got dried.

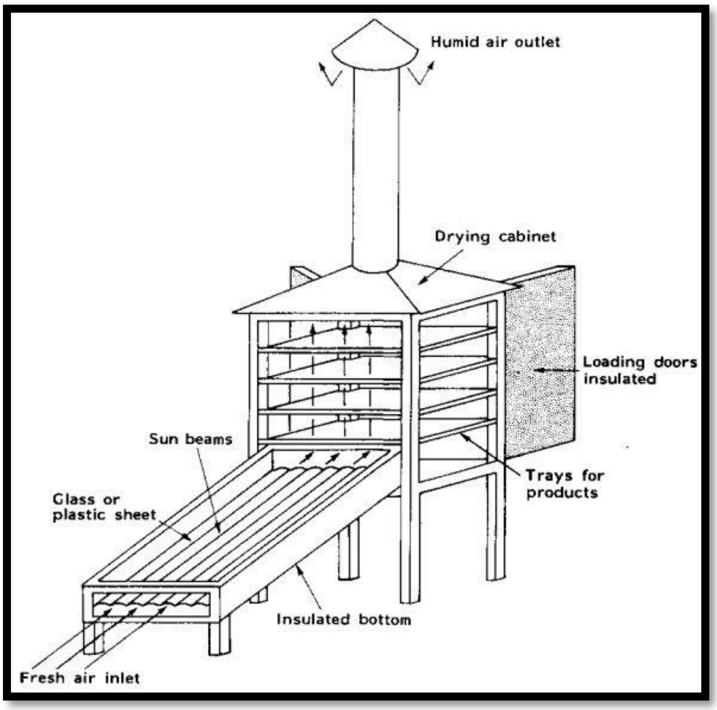


Figure 2 - Solar Cabinet Dryer [12]

2.6.3 Green House Type Solar Dryer:

Ekechukwu [13] studied a greenhouse type dryer (Figure 3) with basic phenomenon of natural convection commercially available as poly tunnel greenhouse. It consists of platform of parallel wire mesh sheets of galvanized iron placed over wooden beams also called drying platforms. A glass roof placed inclined over the drying plates that allows the radiation from sun to reach the product. A zinc sheet folded to form a cap is placed on the roof providing a vent for air exiting the chamber. The chamber normally has a diameter of 1.64m and the vertical height is of 3.0m.

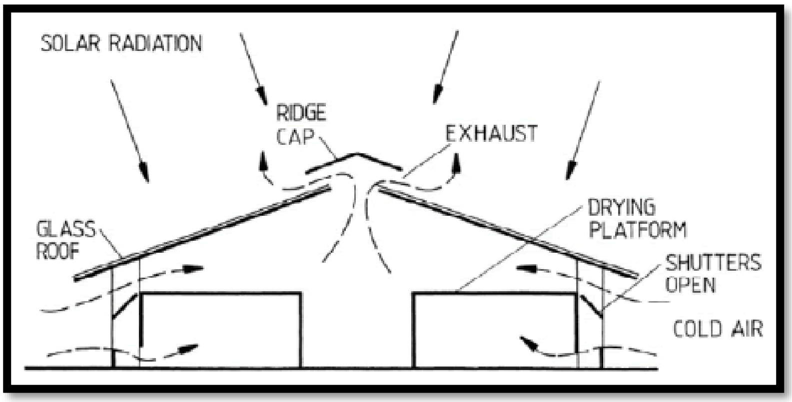


Figure 3 - Green House Solar Dryer [13]

2.6.4 Box Type Solar Dryer:

FAO [14] found that for food drying on small scale the box type solar dryer (Figure 4) is used extensively. It structure is based on wooden box with transparent lid. The inner surface is black and the product is placed on a tray with meshed surface above the base of floor. Air flows across the chamber entering holes in the front and exiting from vent holes on the top of the back wall.

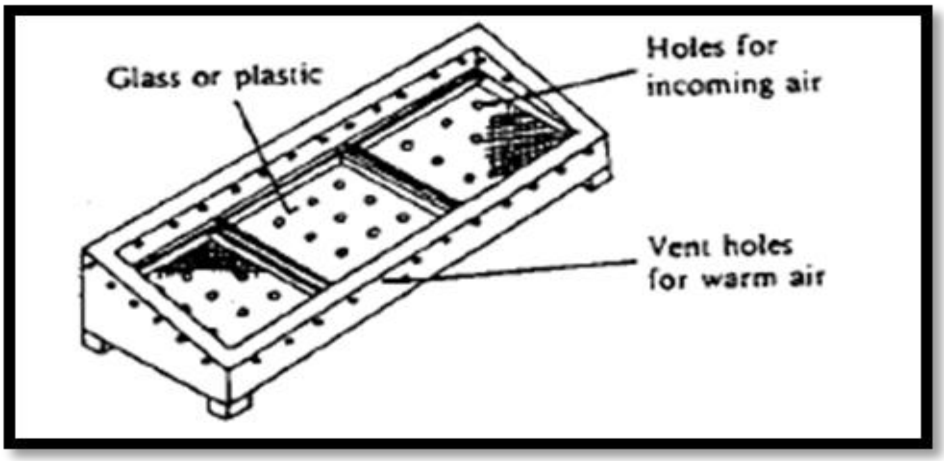


Figure 4 - Box Type Solar Dryer [14]

2.6.5 V-Groove Type Solar Dryer:

Fudholi et al. [15] has developed a V- Groove dryer (Figure 5) giving an output temperature of 50^{o}C absorbing 700 W/m^2 of solar radiations with air flow rate of 15.1 m^3/min by a V-type collection unit with an average area of 15m^2. The performance tests on this type of dryer has been conducted for chilies, noodles and herbal tea with an auxiliary source of 10 KW of heat radiations for optimum temperature control at a flow rate of 15.1 m^3/min. The type of Herbal tea produced depends on the specifications i.e. the degree of moisture content present as on the initial stage the moisture content is 87% (w/w) that has to be reduced to a level of 54% (w/w) preserving the green color of the tea.

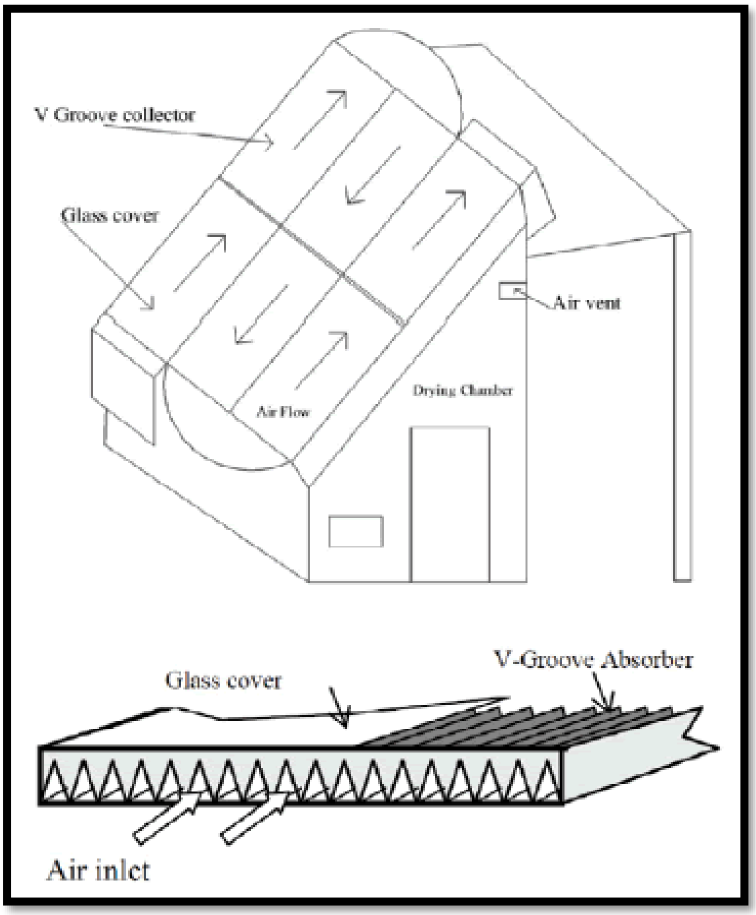


Figure 5 - Solar Dryer using V-Groove Solar Absorbing Panel [15]

2.6.6 Double Pass Solar Collector with Fins:

Sopian et al. [16] studied Double Pass Solar Collector with Fins (Figure 6). Its working principle is based on the increased heat transfer area through the addition of fins as the air firstly passes over the black surface and then the fins resulting in utilizing an increased temperature of air and then consequently the drier product. Sopian et al. [16] has reported this system for approximately 300 Kg. It comprises of a collector cell (~ 11.52 m^2) for radiations, a heater for continuous process, a blower for air and the chamber (roughly of size 4.8 x 1 x 0.6 m) in which product is placed. It utilizes an average mass flow rate of 0.05 – 12 kg/s and a temperature of upto 65 ^oC. Conventional method of open drying in sun takes 10-14 days. The use of double pass solar collector system reduces the time to 86% with a final moisture content of 10% (w/w) reduced from an initial level of 90% (w/w). Experimental results have reported that for 40 kg of product the efficiency of solar radiation collection unit is 37%, the drying system i.e. heater and blower is 27% and that of absorption of heat by product is 92% at heat flux of 544 W/m^2 and average flow rate of air 0.06 Kg/s.

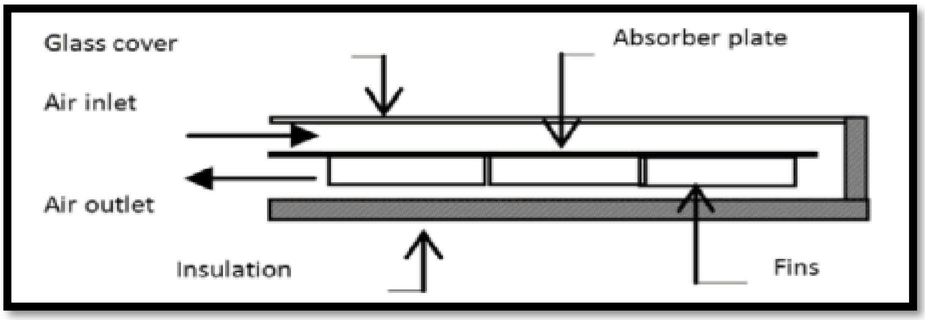


Figure 6 - Double Pass Solar Collector with Fins [16]

2.6.7 Incorporation of Concentrating Panels:

Stiling et al. [17] demonstrated that concentrating panels improves the process of drying. In their study they presented a comparison based on performance of two mixed type solar dryers with one using concentrating panels and the other one not. Temperature inside the dryer utilizing concentrating panels was 10 ºC higher than the other simple dryer. Using concentrating panels considerably increased the drying rate with almost 27 % decrease in drying time to reach dimensionless moisture content of 0.2.

2.6.8 Solar Dryer with a Biomass Stove:

Prasad and Vijay [18] have experimentally studied a solar dryer based on natural convection integral with a stove having biomass fuel. Drying of ginger and turmeric has been carried out with this system having a wire mesh base arranged as three trays (with an area of 0.991 m^2/plate) and a stove (650 x 600 x 550 mm). The efficiency of drying has been reported to be 15.59%.

2.7 Use of Biomass Fuel

The drawback of solar radiation is that it can only be utilized during day time and in rainy season or winter season its intensity lowers affecting the efficiency of drying. So, to cater for these reasons a biomass heating system can be used for continuous drying and increased efficiency. Biomass fuel has the advantages of being cheap, readily available and can be disposed of easily. Following research papers are studied in detail for in cooperation of biomass:

1. Madhlopa and Ngwalo [19] studied an indirect type dryer with natural convection having solar collection unit coupled with a biomass heating system as a backup support. The experimental setup consists of drying chamber, solar radiation collection unit and a burner with biomass fuel.
2. The efficiency of drying process was reported to be 13% with drying temperature of 41-56ºC and the fuel burning efficiency of 11%.

3. Tarigan and Tekasakul [20] also studied a dryer with solar collector as primary and biomass heater as a backup unit. Burner was reported to constructed from fire bricks and concrete to increase the
efficiency by storing heat. The drying chamber achieved a temperature of 65^{o}C by burning 50Kg of biomass.
4. A hybrid system of solar heater and biomass heater has been employed by Prasad et al. [21] for drying of turmeric. The principle of working is the utilization of solar energy during day and biomass energy during night and the efficiency was reported to be 28.57% with an average temperature of drying unit between 55^{o}C to 60^{o}C reducing the time of drying to 86% compared to conventional open drying.
5. Erick et al. [3] has studied various heating systems and mechanisms for drying. The basic three systems investigated in his study includes Solar Heating System (efficiency 24%), LPG gas Heating system (efficiency 86%) and Hybrid (Solar-Gas) Heating System (efficiency 71%). He reported a comparable efficiency of LPG gas heating System and a Hybrid heating system with 20% less fuel consumption maintaining the optimum product quality.
6. Thomas R.Miles [22] studies the effect of firing biomass in boiler. Biomass have considerable amount of alkyne which cause heavy fouling on boiler tubes. The main problem with the boiler is of deposits on its surface thus limiting the furnace volume and rising temperature of flue gases. Additives like magnesium oxides are very useful to reduction of slag.
7. MattiParikk [23] made an effort to investigate the potential of renewable energy sources to replace the fossil fuel. According to their research biomass is a great source of energy with a worldwide available biomass energy is approximately 100 EJ/a that is 30% of the energy being consumed worldwide. It is also reported that only 2/5th of the total available biomass is being utilized.
8. ChangdongSheng ,J.L.T.Azevedo [24] made effort to give an approximate equation for finding high heating valve (HHV) for biomass fuel with error of approximate 5%. This equation is based on composition of compound.

$$HHV\ (MJ/Kg) = -1.3675 + 0.3137*C + 0.7009*H + 0.0318*O$$

2.8 Solar Collector Glazing Materials

Collection units, require high radiation transmission and economic viability, are exposed to high temperatures and longer exposure times. Following are some of the properties which the glazing material for solar collector should have:

- Capability and suitability for high temperature and UV light
- Good light transmission property
- Resistance to impact
- Easy handling and lighter weight
- Reduced heat losses and IR resistant/opaque
- Economic Viability

It is not evitable to have all these properties in single material so proper selection shall be made for required application.

2.9 Comparison of Various Glazing Materials

2.9.1 Glass:

Low iron tempered glass offers good glazing properties for solar collectors in majority of applications.

Advantages: Suitable and capable of exposure to high temperature, good solar radiation absorption properties and resistant to IR radiations.

Disadvantages: Not easy to handle because of weight, easily breakable, not economical.

2.9.2 Corrugated Polycarbonate:

It is a corrugated sheet of polycarbonate normally applied for green house. As a solar collector it has excellent glazing material properties.

Advantages: good resistance to impact, suitable for application up to 270^{o}F, lighter in weight, cost competitive, reduced heat losses due to resistance to IR radiations and also easy availability.

Disadvantages: Corrugated ends must be sealed properly, availability in 2ft wide size, limited life and temperature resistant capacity as compared to glass and corrugated plates result in heat losses.

2.9.3 Rigid Sheet of Polycarbonate:

It serves as good choice for commercial applications.

Advantages: Good resistance to impact, excellent transmitter of radiations, suitable for temperature application of up to 270 ^{o}F, easy to handle, availability in wide range of sizes, resistant to IR radiations.

Disadvantages: Not cost competitive, lower life and temperature applications in contrast to glass.

2.9.4 Rigid Film Polycarbonate:

It is advantageous in comparison to corrugated polycarbonate as it does not require sealing at the ends.

Advantages: Good resistance to impact, high suitability and capability up to temperature of 270 ^{o}F, easy to handle due to lighter weight, availability in size of up to 4 ft. wide and resistant to transmission of IR radiations.

Disadvantages: Reduced life and temperature suitability in contrast to glass.

2.9.5 Coroplast (Horticultural Grade):

Horticulture grade coroplast used as glazing material in green houses offers UV inhibition and comes in natural color as parent polypropylene resin. It has 73.5% transmittance property and a life of approximately 3 years.Suitable for temperature application of up to 180 ^{o}F and has a melting temperature of 324 ^{o}F.

CHAPTER 3 - METHODOLOGY

In the previous section, different types of Solar Biomass Hybrid Dryers and their respective effects on the different types of fruits was mention. In our work, we tried to apply the concepts from the literature review and tried to manufacture an improved model. The main parts of our setup and their functions is present in this section.

3.1 Solar Collector

- Direct and Indirect type are the two most common types of SOLAR COLLECTORS. Direct type provides non-uniform heating, time consuming, spoilage due to wind and rain, bad quality output, as there are nutrient losses. As a result, indirect type was chosen.
- In Indirect type, Evacuated Tubes type and Flat Plate type are mostly used. We chose Flat Plate Solar collector as it provides constant temperature, cheap and easy to manufacture, larger heating area and most importantly, it provides the temperature in the required range of about 45^{o} C to 60^{o} C.
- A Clear glass cover with a 33.7 degrees' tilt angle is chosen and it will face south because the southern side receives the maximum amount of sunlight.
- An Aluminum plate (absorber plate) having a thickness of 5 mm placed at the bottom of the collector to absorb maximum sunlight.
- The absorber plate is painted black to maximize the absorption of thermal energy from sunlight as black surfaces are good absorbers and emitters.
- The external body of solar collector is made of wood. The main reason behind usage of wood is that it was easily available and it is a good insulator.
- Heat losses are very prominent in solar collectors. To counteract this, we have placed Jumbo Sheet (Thick thermopile sheet) inside wood frame on the sides and bottom to increase insulation.
- Two fans are place at the inlet side to maintain a constant flowrate and the flowrate can also be varied as a result it can be used to achieve different temperatures of air, thus making it suitable for variety of fruits.
- The principle of flat plate collector is very simple. Air enters through the fan and is heated

by the absorber plate through natural and forced convection. Heating the air decreases its relative humidity as a result it can absorb more moisture. This heated air is transfer to drying chamber to dry fruits.

- Obtaining the correct dimensions of the different solar collector components was the most challenging part. For this a mathematical model was made and tested in MTALAB, where we iteratively varied length of plate, air inlet velocity and other factors to obtain the desired results.
- To verify our results a detailed Analysis was carried on ANSYS CFD.

3.2 Biomass Heat Exchanger

- Shell and Tube, Spiral and Plate Type Heat Exchanger are most commonly used Heat Exchanger.
- Spiral Type Heat Exchanger are difficult to fabricate with high construction cost.
- Shell and tube heat exchanger was not used as for it we need a higher flowrate and to maintain a higher flowrate we will need a blower or compressor which is expensive.
- The type of heat exchanger which we are using is duct type. In this type of heat exchanger, cold and hot fluid flow parallel on the respective sides of metallic plate having fins attached to it for better heat transfer.
- Metal shell of heat exchanger is made in the shape of a box having length 0.6 m and width 0.3 m with a metal plate placed in center for separating cold and hot portion of heat exchanger. Metal shell is insulated by Jumbo Sheet to stop heat loss to outside environment.
- An Aluminum plate (heat transferring plate) having a thickness of 5 mm is placed at the center of metallic shell which transfers heat from hot fluid to cold.
- The heat transferring plate is painted black to maximize the absorption of thermal energy.
- For better heat transfer, we use aluminum strip (fins) having height of 38.1 mm between shell and center plate.
- For fresh air intake two fans are placed on cool side of heat exchanger, thus maintaining a constant flowrate of air. Hot air (flue gases) is coming from biomass burner.
- The principle of duct type heat exchanger is very simple. Hot gases (flue gases) heat the

metallic plate by natural and forced convection. Cold air takes heat from the metallic plate, reducing its relative humidity and increasing its moisture absorption capacity. This heated air is then transferred to drying chamber to dry fruits.

- Obtaining the correct dimensions of different components duct type heat exchanger was the most challenging part. For this a mathematical model was made, based on flow in duct and tested in MATALAB, where we iteratively varied length of plate, air inlet velocity and other factors to obtain the desired results.
- To verify our results a detailed Analysis was carried on ANSYS CFD.

3.3 Drying Chamber

- Cabinet and Green House type are the two most common types of Drying Chambers.
- In Green House Type, even heating of products is not possible, Quality of product is not good, heat losses are prominent and its operation is heavily dependent on weather conditions.
- In Cabinet Type Drying Chamber, controlled environment during drying process can be maintained, even heating of product is possible, Heat losses are negligible and it is fairly cheap to manufacture compared to other type of drying chamber. As a result, Cabinet Type Drying Chamber was chosen.
- The external body of drying chamber is made of wood. The main reason behind usage of wood is that it is easily available and it is a good insulator.
- Heat losses are very prominent in drying chamber. To counteract this, we are using wood frame on the sides and bottom to increase insulation.
- For even distribution of hot air, two inlets are provided one for each drying plate.
- The principle of drying chamber is fairly simple. Hot air enters the drying chamber, from solar collector and bio mass heat exchanger. As the hot air passes through the drying chamber, it removes moisture from products placed on drying plates by evaporation, thereby, drying the products. This humid air then moves out of the drying chamber is replaced by incoming hot air.
- To calculate overall heat energy and time required during drying process to reach the final moisture concentration, a MATLAB code was developed which is provided in the Appendix.

3.4 Design

Each component was model on SOLIDWORKS with the 2D drawings given below.

3.4.1 Solar Collector

- The solar collector is a single pass flat plate air collector.
- Two fans are install at inlet to maintain an airflow in collector.
- The dimensions of the absorber plate are 0.94 m x 0.65 m x 0.005 m.

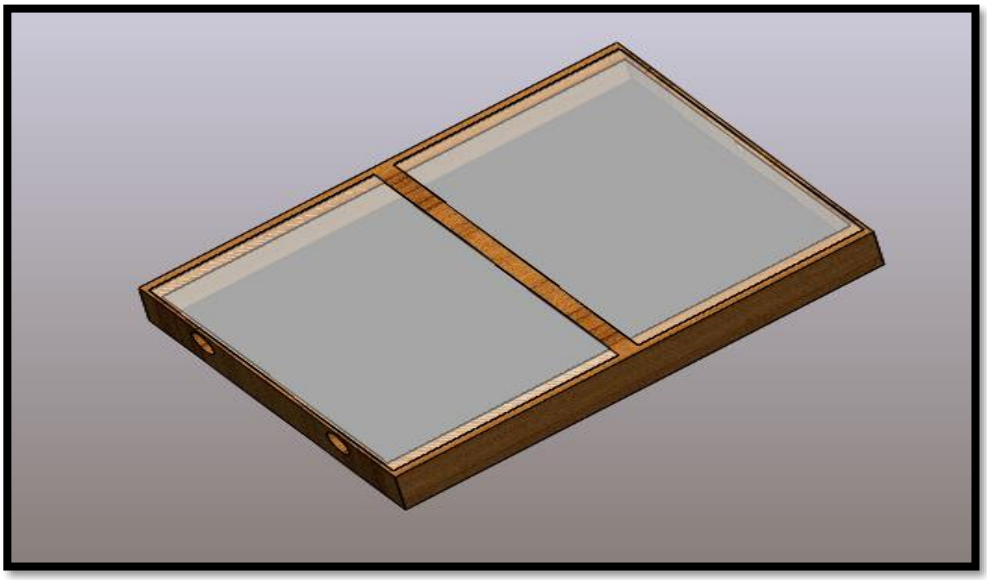


Figure 7 - 3-D Solar Collector Model

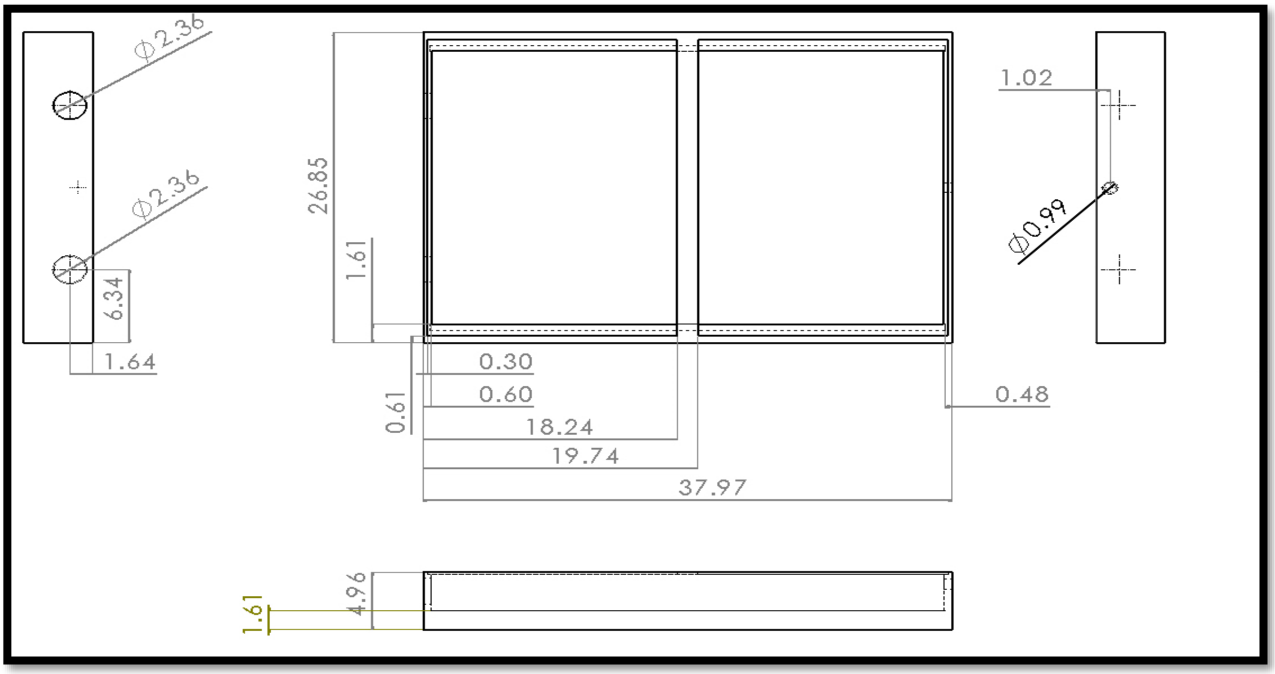


Figure 8 – 2-D Drawing Solar Collector (All dimennsion are in inches)

3.4.2 Biomass Heat Exchanger

- The Heat Exchanger is a Duct Type Heat Exchanger.

- Center plate and Fins are made of Aluminum.

- The dimensions of Centre plate are 0.6 m x 0.3m x 0.005 m.

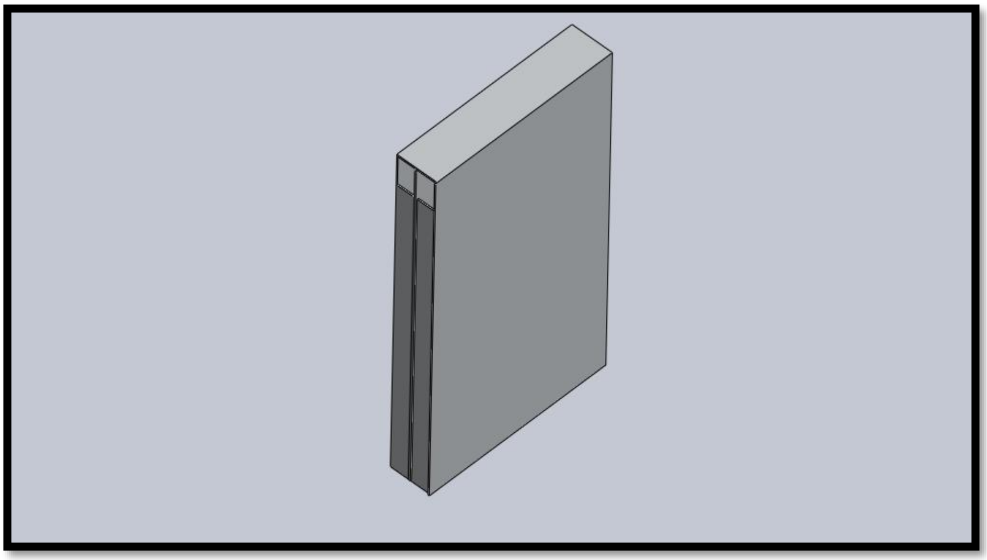


Figure 9 - 3-D Heat Exchanger Model

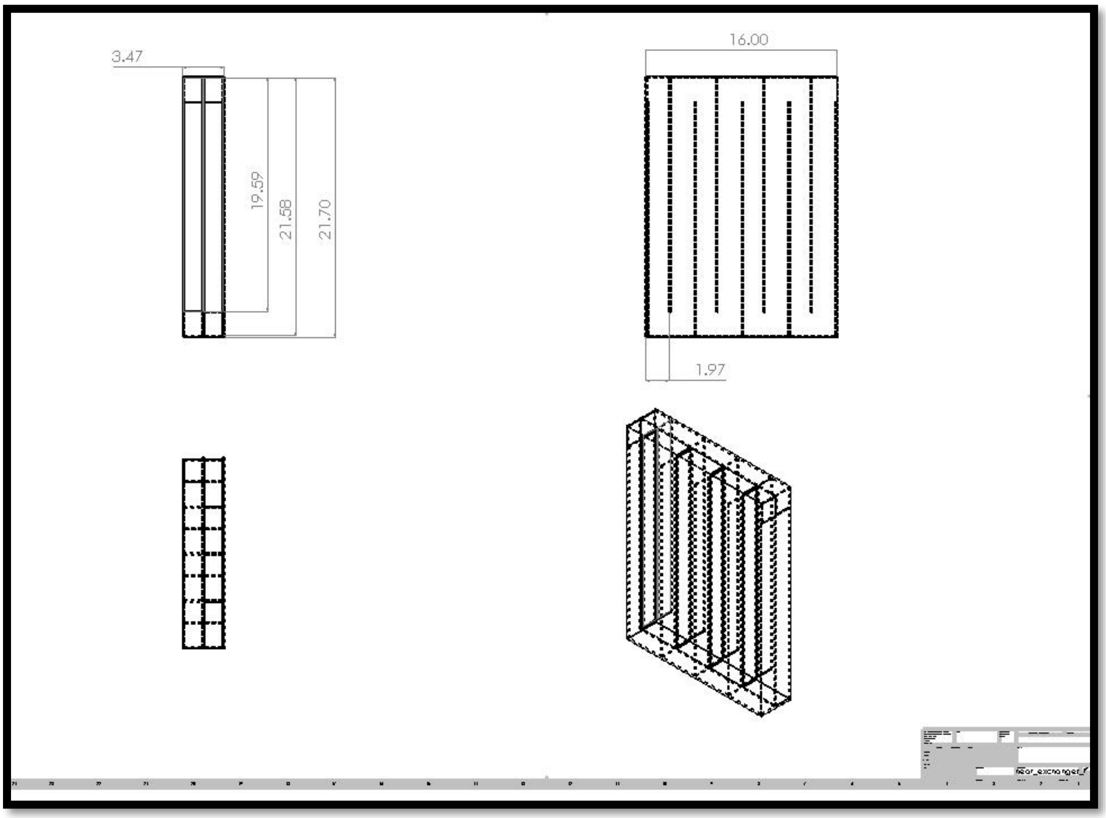


Figure 10 – 2-D Drawing Heat Exchanger

3.4.3 Drying Chamber

- The Drying Chamber is a Cabinet Type Drying Chamber.
- For even distribution of hot air separate inlets are provided for each drying plate.
- The dimensions of the drying chamber are 0.55 m x 0.44 m x 0.69 m.

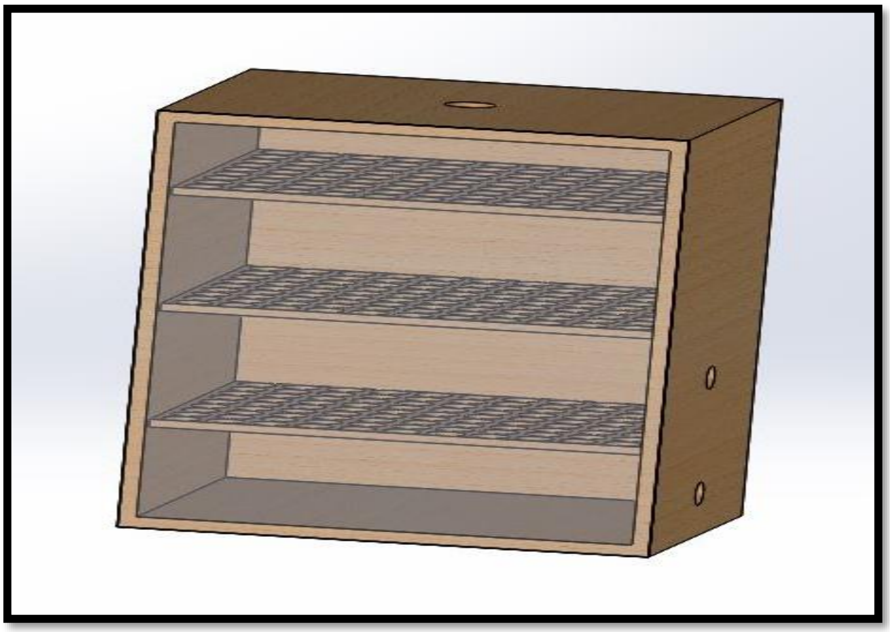


Figure 11 - 3-D Drying Chamber Model

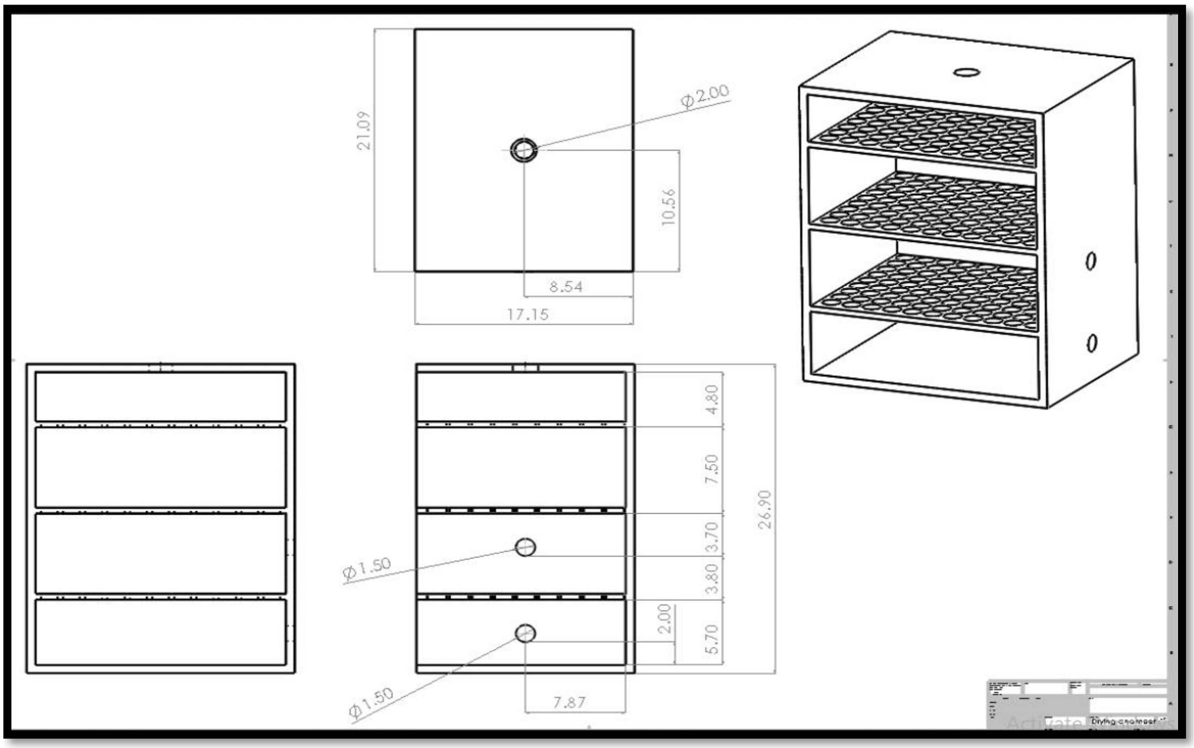


Figure 12 - 2-D Drawing Drying Chamber (All dimensions are in inches)

3.4.4 Complete Model

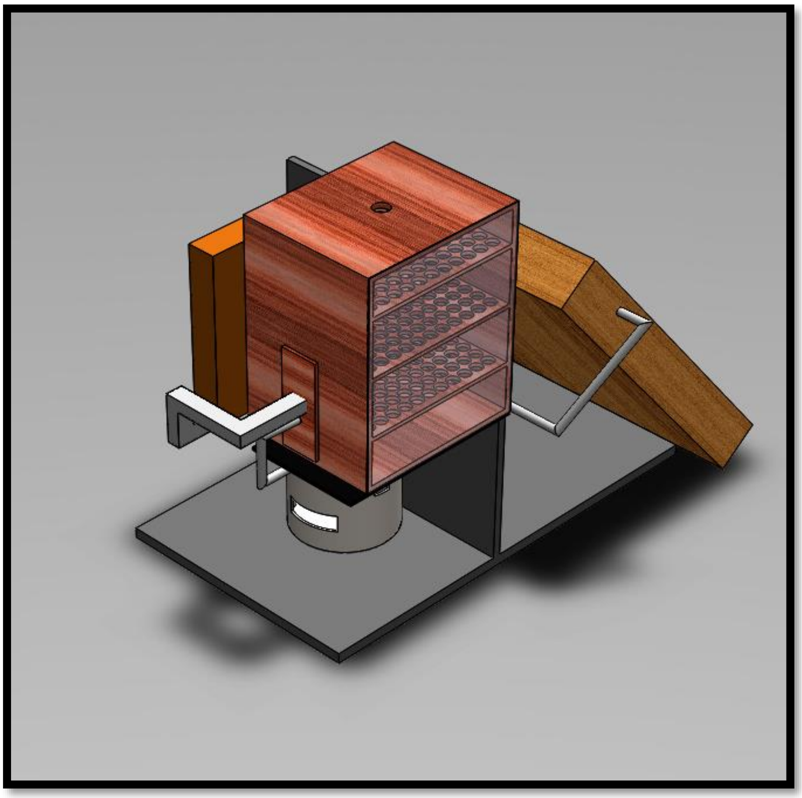


Figure 13 - Complete 3-D Model (View-1)

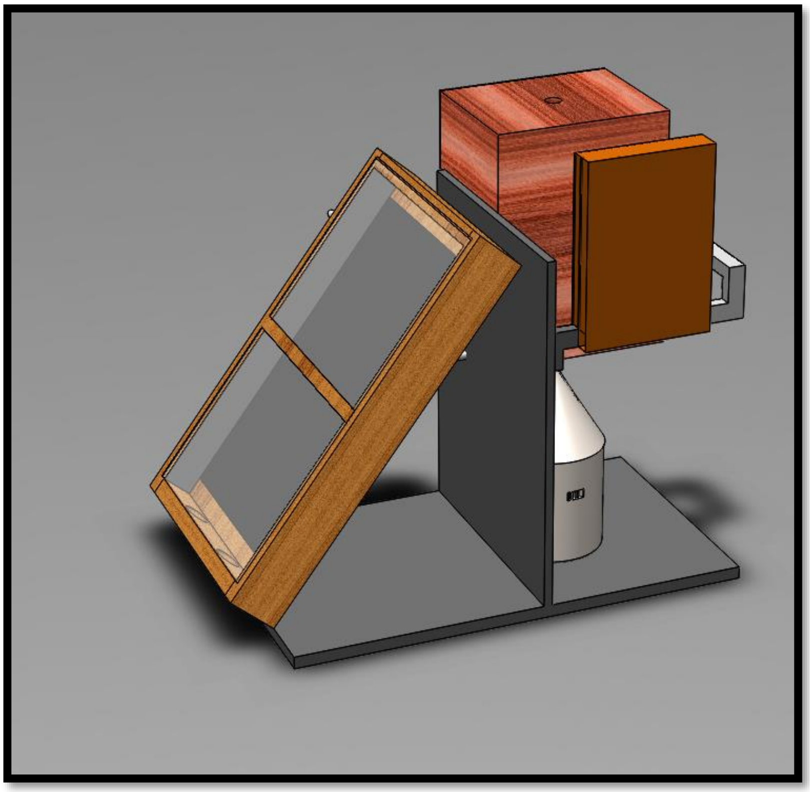


Figure 14 - Complete 3-D Model (View-2)

CHAPTER 4 - NUMERICAL ANALYSIS

The numerical analysis of Solar Collector, Heat Exchanger and Drying Chamber were carried to verify the dimensions of components to be chosen. Numerous different energy balance equations were solved using an iterative procedure in MATLAB. The various equations used for numerical analysis and the iterative procedure are mentioned in this context. The equations used in following sections are taken from book on Heat and Mass Transfer by Yungus A. Cengel and Afshin Jahanshahi [25].

4.1 Mathematical Modelling of Solar Collector

The equations used in this section are taken from Yungus A. Cengel et a1. [25] and Abdulelah Ali Al- Jumaah et al. [26].

Energy received by the black absorber is given by the equation,

$$(cp_p * \rho_p * V_p * \frac{dT_p}{dt}) = \left(I * A_p * abs_p * trans\right) - Q_{c_{p-a}} - Q_{loss} - Q_{r_{p-g}} \quad (1)$$

Energy absorbed by the air is given by the equation,

$$(cp_a * \rho_a * V_a * \frac{dT_a}{dt}) = Q_{c_{p-a}} - Q_{c_{a-g}} \quad (2)$$

Energy received by the glass is given by the equation,

$$(cp_g * \rho_g * V_g * \frac{dT_g}{dt}) = \left(I * A_g * abs_g\right) - Q_{r_{g-sky}} + Q_{r_{p-g}} + Q_{c_{a-g}} + Q_{c_{g-sky}} \quad (3)$$

Rate of energy transfer from absorber plate to air by convection is given by,

$$Q_{c_{p-a}} = h_{c_{p-a}} * A_p * (T_p - T_a) \quad (4)$$

Convective heat transfer coefficient between absorber and air is the sum of natural convective heat transfer coefficient and forced convection heat transfer coefficient.

$$h_{c_{p-a}} = h_{c_{p-a}}(NaturalConvection) + h_{c_{p-a}}(ForcedConvection)$$
(5)

Natural convection heat transfer coefficient is found from the below equation,

$$h_{c_{p-a}}(NaturalConvection) = \frac{[Nu(NaturalConvection)*k_{air}]}{AirGap} \quad (6)$$

Natural convection NUSSELT Number is found by,

$$Nu(NaturalConvection) = 1 + 1.44\left[1 - \frac{1708}{Ra_L * cos\theta}\right]^{+}\left[1 - \frac{1708*(sin1.8\theta)^{1.6}}{(Ra_L * cos\,\theta)}\right] + \left[\frac{(Ra_L * cos\theta)^{\frac{1}{3}}}{18} - 1\right]^{+}$$

(7)

$$Ra_L = (g * B * (T_1 - T_2) * L_c^3 * Pr)/(v^2) \tag{8}$$

$$B = \frac{2}{T_P + Tg}$$

(9)

Forced convection NUSSELT Number is found by,

$$Nu(ForcedConvection) = 0.664 * Re^{0.5} * Pr^{1/3} \tag{10}$$

Forced convection heat transfer coefficient is found from the below equation,

$$h_{c_{p-a}}(ForcedConvection) = \frac{Nu(ForcedConvection) * k_{air}}{Lp} \tag{11}$$

Heat loss by conduction is given by the equation,

$$Q_{loss} = U_{loss} * A * \left(T_p - T_{amb}\right)$$

(12)

For the value of U_{loss} refer to Appendix 1.

Rate of energy transfer from absorber plate to glass by radiation is given by,

$$Q_{r_{p-g}} = h_{r_{p-g}} * A_P * (T_p - T_g)$$

(13)

Radiative heat transfer coefficient between absorber plate and glass is given by,

$$h_{r_{p-g}} = \frac{\left[\sigma * \left[(T_P + 273)^2 + \left(T_g + 273\right)^2\right] * \left[(T_P + 273) + \left(T_g + 273\right)\right]\right]}{\left[\left(\frac{1}{e_p}\right) + \left(\frac{1}{e_g}\right) - 1\right]} \tag{14}$$

Rate of energy transfer between sky and glass by radiation is found using following equation given by Abdulelah Ali Al-Jumaah et al. [26],

$$Q_{r_{g-sky}} = e_g * \sigma * A_g * [\left(T_g + 273\right)^4 - \left(T_{sky} + 273\right)^4] \tag{15}$$

Rate of convective heat transfer between air and glass,

$$Q_{c_{a-g}} = h_{c_{a-g}} * A_g * (T_a - T_g)$$
(16)

Rate of convective heat transfer between sky and glass,

$$Q_{c_{g-sky}} = h_{c_{g-sky}} * A_g * (T_g - T_{sky})$$
(17)

Convective heat transfer coefficient between glass and sky,

$$h_{c_{g-sky}} = \frac{Nu_{g-sky} * k_{air}}{L_g} \quad (18)$$

In the first iteration $T_p = T_{stagnation}$ (**Refer to Appendix 2**) = 80ºC , T_a = 30ºC, T_g = 50ºC , T_{amb} = 30ºC Change in plate temperature (dT_p), change in glass temperature (dT_g) and change in air temperature (dT_a) is calculated at a time step of 0.001 s. The total time in the loop is the time the air takes to move from inlet till exit of Solar Collector which is 9 seconds, found by (L_p/v_{air}).

For the next time step the parameters are re-defined as,

$$\boldsymbol{T_p = T_p + dT_p}$$
(19)

$$\boldsymbol{T_a = T_a + dT_a}$$
(20)

$$\boldsymbol{T_g = T_g + dT_g}$$
(21)

4.2 Mathematical Modelling of Biomass Heat Exchanger

The equations used in this section are taken from Yungus A. Cengel et a1. [25] and Sandeep Kumar Patel et al. [27].

Cross section area of duct (area of air entering section)

$$A_c = l * w \quad (1)$$

Inlet Velocities of cold and hot sides

$$v_c = \frac{\dot{m}_c}{p_c A_c} \quad (2)$$

$$v_h = \frac{\dot{m}_h}{p_h A_c} \quad (3)$$

Hydraulic diameter of duct

$$D_h = \frac{4A_c}{p} = \frac{2(l*w)}{l+w} \tag{4}$$

Reynolds's number for cold and hot sides

$$Re_c = \frac{\rho_c * v_c * D_h}{\mu_c} \tag{5}$$

$$Re_h = \frac{\rho_h * v_h * D_h}{\mu_h} \tag{6}$$

Nusselt number for cold side as given by Sandeep Kumar Patel and Alkesh M. Mavani [27]

$$\mathrm{Nu}_c = 0.664 * \mathrm{Re}_c{}^{0.5} * \mathrm{Pr}_c{}^{0.3} \tag{7}$$

Nusselt number for hot side

$$\mathrm{Nu}_h = 0.664 * \mathrm{Re}_h{}^{0.5} * \mathrm{Pr}_h{}^{0.4} \tag{8}$$

Forced convection heat transfer coefficient for cold side

$$\mathrm{h}_c = \frac{\mathrm{Nu}_c * \mathrm{k}_{air}}{D_h} \tag{9}$$

Forced convection heat transfer coefficient for hot side

$$\mathrm{h}_h = \frac{\mathrm{Nu}_c * \mathrm{k}_{air}}{D_h} \tag{10}$$

Overall heat transfer coefficient

$$\frac{1}{U} = \frac{1}{h_c} + \frac{1}{h_h} \tag{11}$$

Heat transfer to get desire cold air temperature

$$\dot{\mathrm{Q}} = \dot{\mathrm{m}}_c * \mathrm{C}_{p_c} * \left(\mathrm{T}_{\mathrm{c}_{out}} - \mathrm{T}_{\mathrm{amb}}\right) \tag{12}$$

Flue gases outlet temperature

$$\mathrm{T}_{\mathrm{h}_{out}} = \mathrm{T}_{\mathrm{h}_{in}} - \frac{\dot{\mathrm{Q}}}{\dot{\mathrm{m}}_h * \mathrm{C}_{p_h}} \tag{13}$$

Log mean temperature

$$\Delta T_1 = \mathrm{T}_{\mathrm{h}_{in}} - \mathrm{T}_{\mathrm{c}_{out}} \tag{14}$$

$$\Delta T_2 = \mathrm{T}_{\mathrm{h}_{out}} - \mathrm{T}_{\mathrm{c}_{in}} \tag{15}$$

$$\Delta T_{ln} = \ln\left(\frac{\Delta T_1 - \Delta T_2}{\frac{\Delta T_1}{\Delta T_2}}\right) \tag{16}$$

Surface area of duct will need

$$A_s = \frac{\dot{Q}}{U * (\Delta T_{ln})} \tag{17}$$

Length of duct

$$Length = \frac{A_s}{2 * w + l} \tag{18}$$

For pressure loss

"f" is found by Moody chart

$$H_l = f \frac{L}{D_h} \frac{V^2}{2 * g} \tag{19}$$

Pressure loss

$$P_{loss} = \rho * g * H_l \tag{20}$$

Power requirement

$$P_{fan} = \dot{V} * P_{loss} \tag{21}$$

MATLAB code for length determination is provided in Appendix 6

4.3 Mathematical Modelling of Drying Chamber

Mass of water to be removed during drying is calculated using following Equation given by V. K. Sharma et al. [28]

$$M_w = \frac{m_{wb_i} - m_{wb_f}}{100} * m \tag{1}$$

Initial and Final Moisture content are calculated as follow:

$$m_{db_i} = \frac{m_{wb_i}}{100 - m_{wb_i}} * m_{p_r} \tag{2}$$

$$m_{db_f} = \frac{m_{wb_f}}{100 - m_{wb_f}} * m_{p_r} \tag{3}$$

Amount of energy for drying is calculated by Equation given by Juneyd F. Dadi et al. [29]

$$Q = M_d * C_d * (T_f - T_i) + M * C_p * (T_f - T_i) + M_w * \lambda \tag{4}$$

Equilibrium Relative Humidity (ERH) at which moisture concentration of surrounding air and product comes at equilibrium is given as follow

$$ERH = 100 * a_w \quad (5)$$

Where a_w is given by:

$$a_w = 1 - exp[-exp(0.914 = 0.5639 * m_{db})] \quad (6)$$

Time required for drying is then calculated by equating the total amount of energy needed to dry the product and energy of circulating hot air, which is done in following equation

$$Q = m_a * t * C_p * \left(T_f - T_i\right) \quad (7)$$

By making use of time required for drying, as calculated in above equation, we can estimate the drying rate using following equation

$$Drying\ rate = \frac{M_w}{t} \quad (8)$$

4.4 ANSYS CFD Analysis

CFD analysis was carried out on each part so that a more realistic picture of the experiment can be given and to verify the results which we got from MATLAB.

4.4.1 Solar Collector

The main problem which we faced was to choose the length, air gap and with of the solar collector and the absorber plate, because in all the research papers a full scale setup was made. So to test our prototype we varied air gap and the speed of air inside collector till we got required results.

In all the iterations the following steps were followed,

1) Creation of 3D model in SOLIDWORKS with an extrude cut instead of the plate.
2) Importing in the CFD and creating NAMED SELECTIONS.
3) Then mesh was created using a Relevance Centre =100.
4) In Solution Setup PRESSURE BASED SOLVER was used as the flow is incompressible. Steady condition for the solution is used.
5) Energy equation was on.
6) Laminar model was used as Reynold number is about 2400 (less than 500000).
7) In the Radiation Model, Discrete Ordinate Model was used. Solar Ray Tracing was incorporated so that Direct Solar Irradiation value can be set to 500W/m^2.
8) In the Materials section glass and wood was created.
9) CONVECTION as the thermal condition was used for the absorber plate and glass in which Heat Transfer Coefficient was set as 4 W/m^{2}K.
10) The velocity inlet was set at 0.05 m/s in each of the INLET of the two holes. The velocity at the inlet was set at 30^{o}C.
11) Through PATCH command the absorber plate temperature was set to 65 ^{o}C.

Some of the iterative process done in ANSYS is present below in the table.

#	Air gap (cm)	Speed of air (m/s)	Simulation result
1	15	0.35	Average of Surface Vertex Values Static Temperature (k) -------------------------------- -------------------- absorber-plate 338.68338 glass 322.09756 inlet 301.76382 insulation 313.17645 interior-solid 310.38101 outlet 309.17432 ---------------- -------------------- Net 314.08505
2	8	0.35	Average of Surface Vertex Values Static Temperature (k) -------------------------------- -------------------- absorber-plate 329.7417 glass 319.75061 inlet 302.94409 insulation 312.79913 interior-solid 313.43777 outlet 313.37683 ---------------- -------------------- Net 316.21524
3	5	0.25	Average of Surface Vertex Values Static Temperature (k) -------------------------------- -------------------- absorber-plate 333.04349 glass 322.5528 inlet 301.92123 insulation 311.51459 interior-solid 312.25671 outlet 311.85574 ---------------- --------------------
4	8	0.1	Average of Surface Vertex Values Static Temperature (k) -------------------------------- -------------------- absorber-plate 348.64743 glass 334.86563 inlet 304.33047 insulation 331.97849 interior-solid 325.72028 outlet 325.92426 ---------------- -------------------- Net 330.33533

Table 1 - Comparison of ANSYS Results with Different Speeds of Air and Air Gap

The parameter which we selected are highlighted in the red in Table 1. A more detailed analysis is provided below of the selected result.

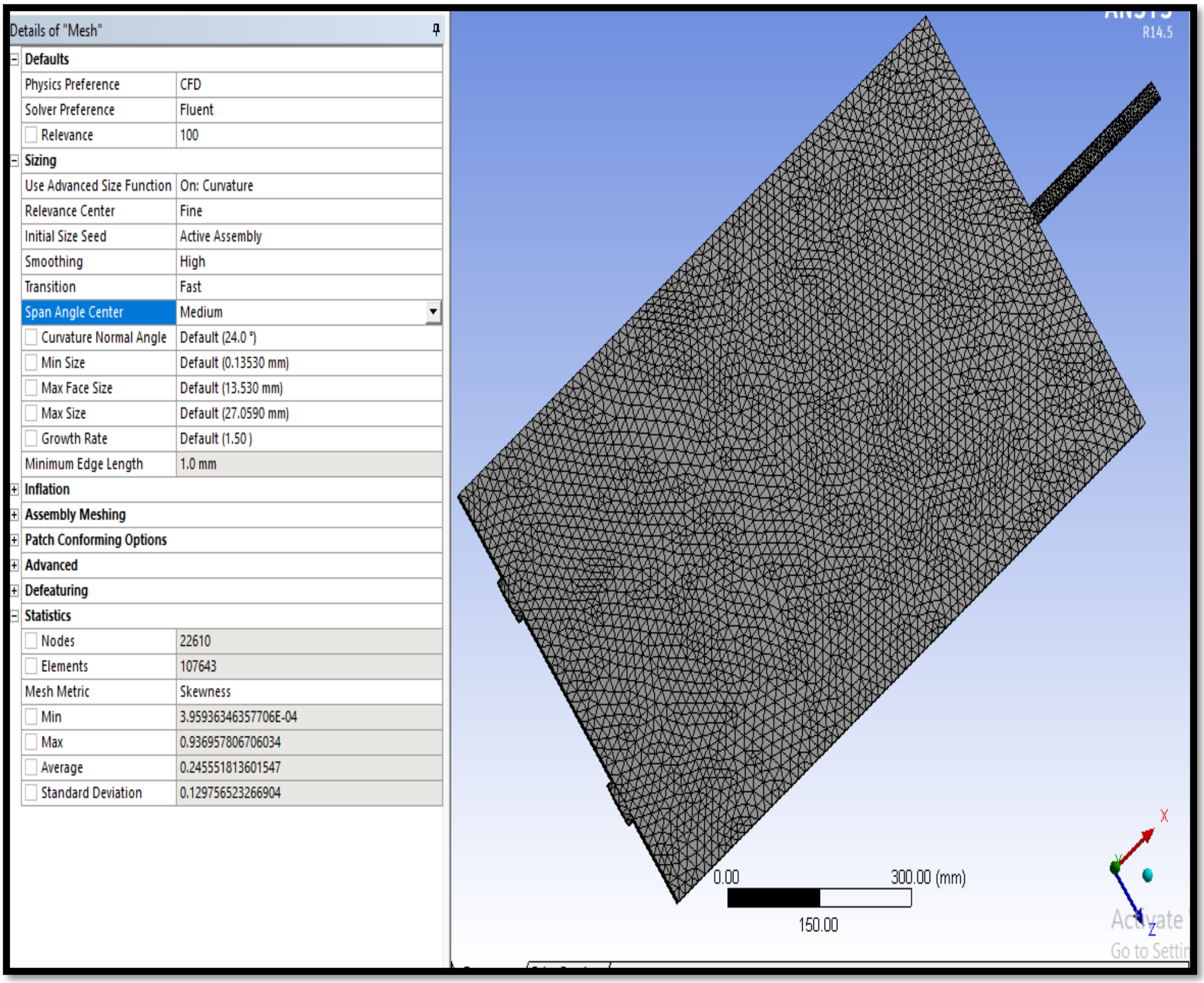
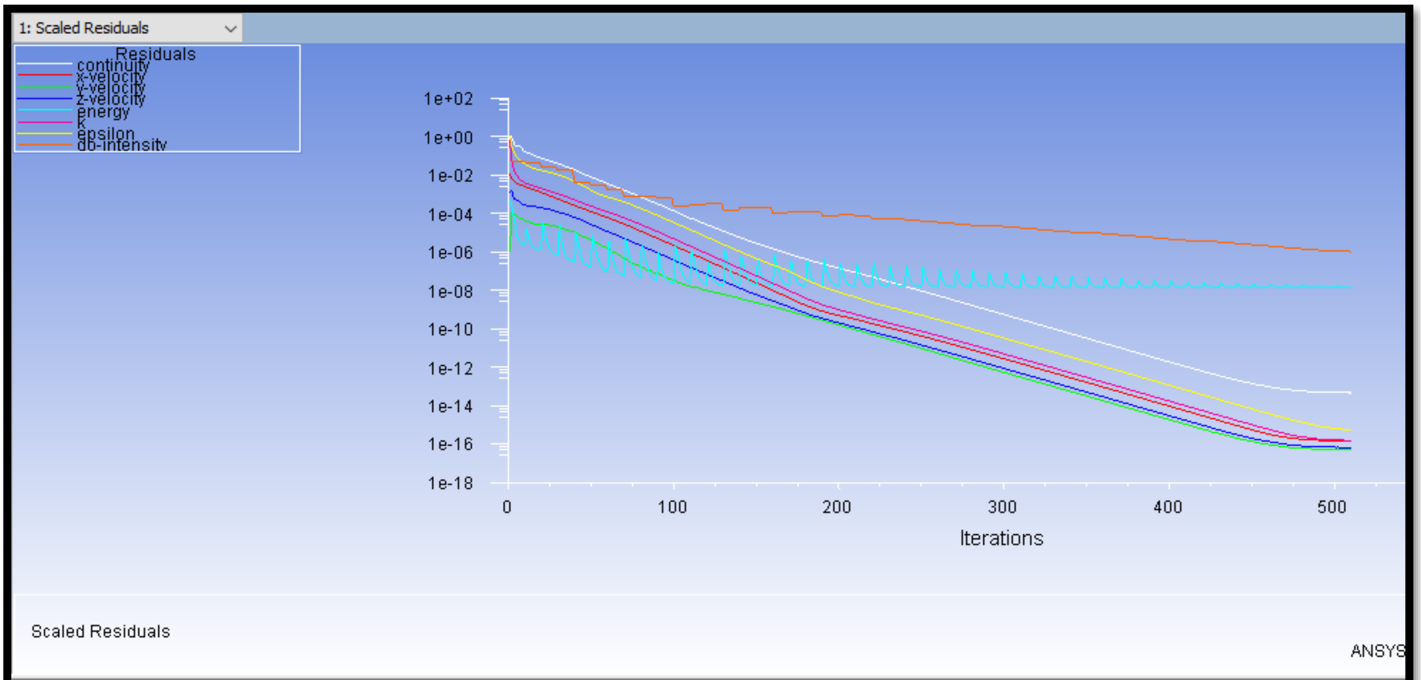


Figure 15 - Mesh of the 3-D Model with the Details of Meshing

The below figure shows the convergence which means there were no errors in the Boundary Conditions and the mesh was good as well.




```
  iter  continuity  x-velocity  y-velocity  z-velocity      energy           k     epsilon do-intensity     time/iter
   507  4.8158e-14  1.4885e-16  5.1821e-17  6.7168e-17  1.5025e-08  1.6093e-16  5.9870e-16  1.0824e-06  0:02:42  493
   508  4.8421e-14  1.4990e-16  5.1684e-17  6.7180e-17  1.4942e-08  1.5833e-16  5.8612e-16  1.0824e-06  0:02:10  492
   509  4.7948e-14  1.4978e-16  5.2610e-17  6.8736e-17  1.5000e-08  1.5937e-16  5.8211e-16  1.0824e-06  0:03:22  491
!  510 solution is converged
   510  4.8112e-14  1.4987e-16  5.3194e-17  6.9589e-17  1.4932e-08  1.5801e-16  5.8842e-16  9.3199e-07  0:02:41  490

Average of Surface Vertex Values
                 Static Temperature                  (k)
-------------------------------- --------------------
                    absorber-plate            348.64743
                             glass            334.86563
                             inlet            304.33047
                        insulation            331.97849
                    interior-solid            325.72028
                            outlet            325.92426
                ---------------- --------------------
                               Net            330.33533
```

Figure 16 - Convergence Graph and Outlet Conditions

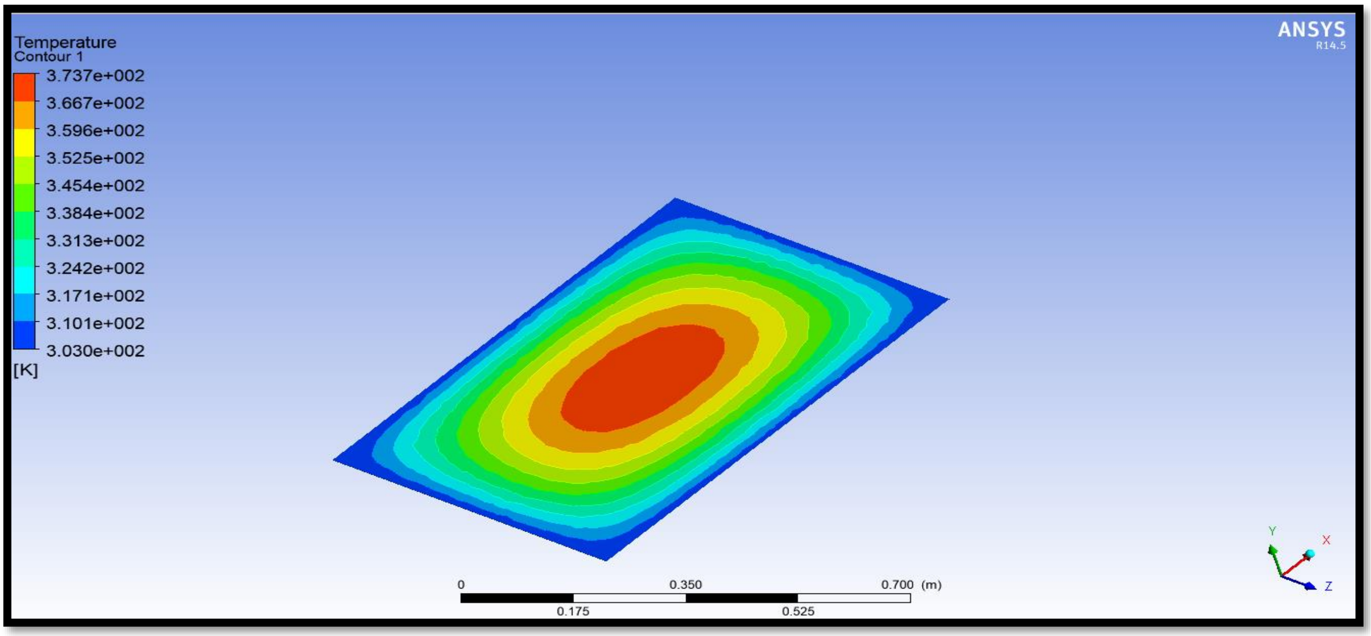


Figure 17 - Temperature Contour of the Absorber Plate

The above figure shows the distribution of the temperature at different position in the Absorber plate. At the bottom part of the plate inlet air passes by so its temperature is low. At the outlet, the temperature is low as it is in direct contact with the insulation. The maximum temperature is in the middle as it is place at 35º and so maximum sunlight is absorbed at the center position.

Figure 18 shows the velocity streamlines in the Solar Collector. The path followed is what we expected. There is no reverse flow and air is distributed all above the plate.

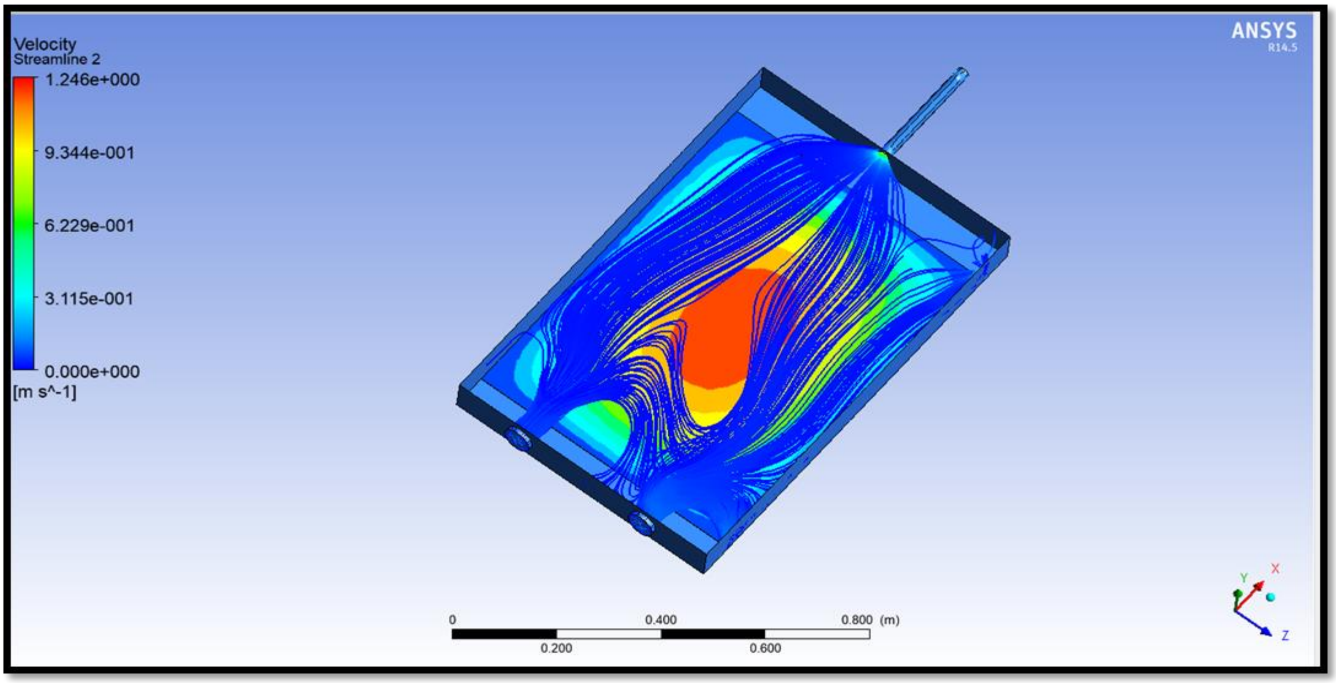


Figure 18 - Airflow Path in Solar Collector

Conclusion:

By CFD analysis we have reached to the conclusion that a temperature of 50 to 60 is reached in the solar dryer. This temperature is optimum for the drying of the product. Moreover, it is also observed that a proper air flow in established in the chamber with no eddies. Hence the CFD analysis indicated the good performance of our proposed design.

4.4.2 Biomass Heat Exchanger

Simulations were carried on the Heat Exchanger design as well. The proposed design of the heat exchanger is a new one, so CFD analysis on it was very important to get a real picture of whether the proposed design can heat the air to the required temperature or not. The steps carried for the simulation are mentioned below:

1) After importing the model from the SOLIDWORKS and creating the NAMED SELECTION, a MESH was generated as shown in Figure 19, with the mesh details as well.

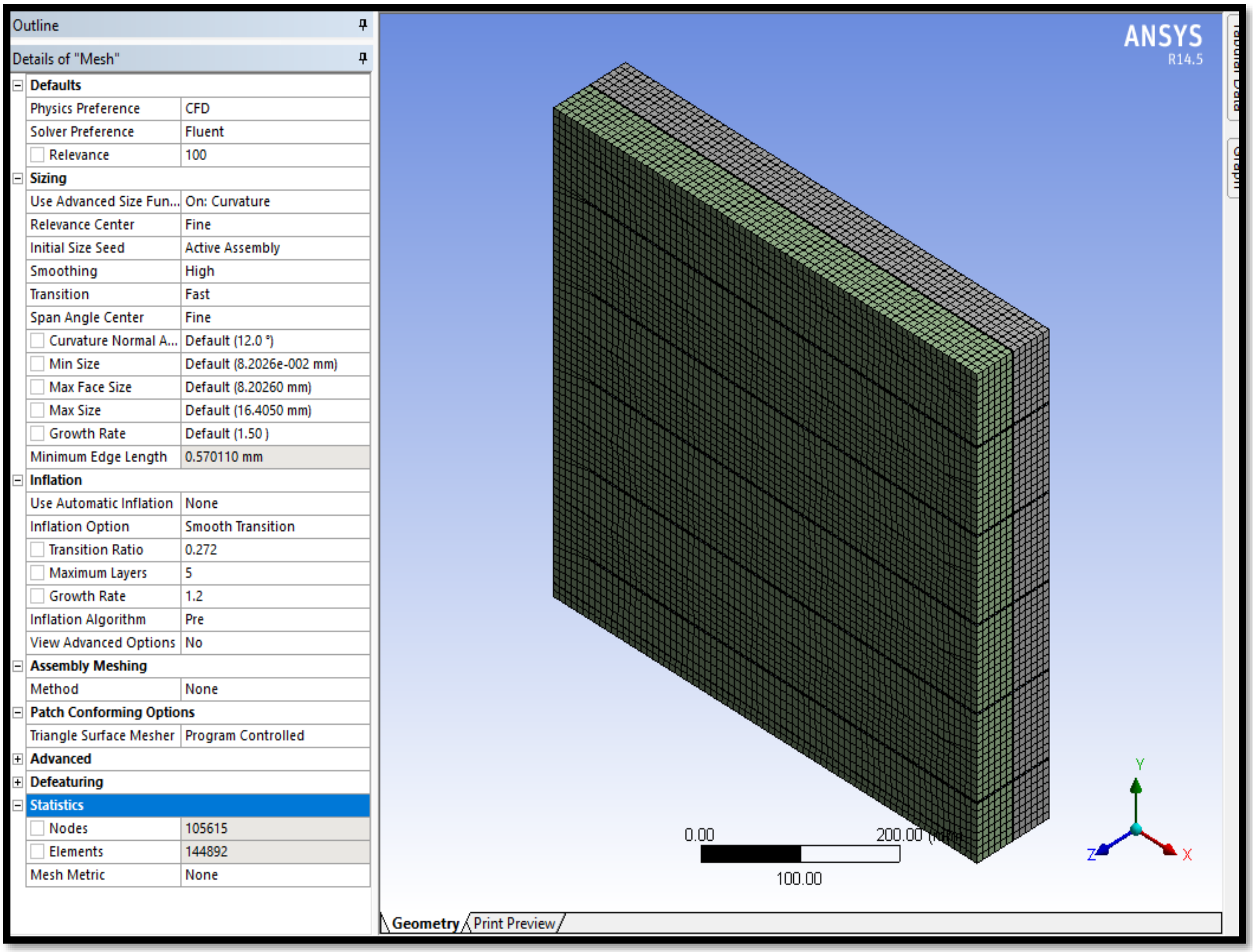


Figure 19 - Mesh of Heat Exchanger with Details

The Named Selection and their meaning is explained in the table

Name Selection	Meaning
Coldairinlet	Inlet of the ambient air
Coldairoutlet	Outlet of the ambient air after heating
Hotairinlet	Inlet of the flue gases
Hotairoutlet	Outlet of the flue gases
Fins	Fins at the flue gas side

Table 2 - Name Selection and its Meaning

2) Energy equation was turned on. k-epsilon model (Realizable + Scalable Wall Functions) Model was used as the Reynold number of air is 12000 and so turbulence occurs.

3) Thermal conditions for FINS was set to CONVECTION with Heat Transfer Coefficient as $13W/m^2K$ and Free Stream Temperature as 344.5 K (71.5°C).

4) Temperature of Hot Air Inlet = 393 K, Velocity of Hot Air Inlet = 5m/s, Temperature of Cold Air Inlet = 303 K, Velocity of Cold Air Inlet = 4m/s.

5) For solution, Fins temperature was set to 343 K using the Patch command.

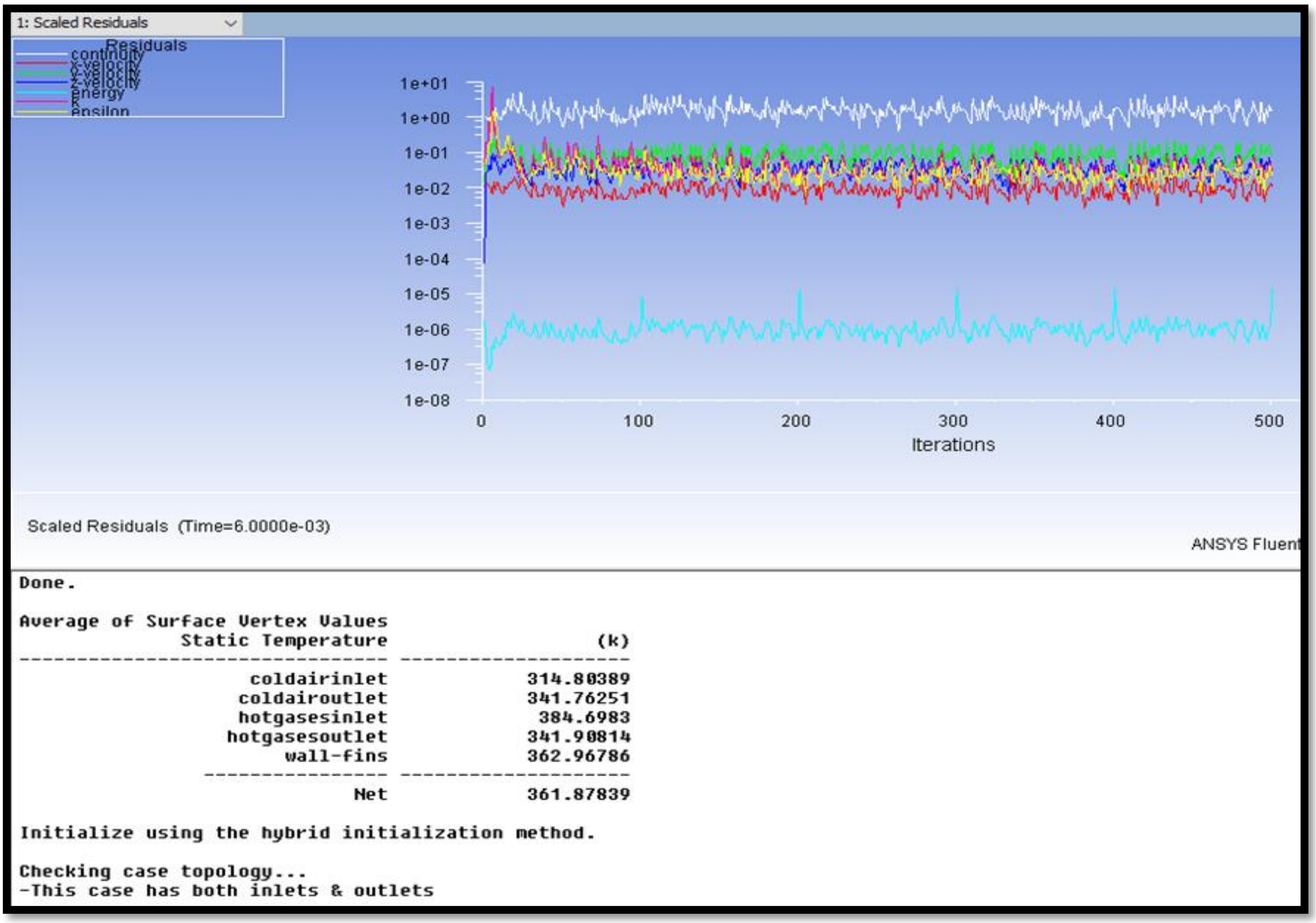


Figure 20 - Results of Simulation

The above figure shows the result after simulation, the ambient air has achieved an increase of 30ºC, thus supporting our design.

Figure 21 shows the Volume Rendering of the temperature for the flue gas side. Air enters from the top and exits from the bottom. As the air passes by the fins and the plate, its temperature decreases because it is at a higher temperature than the fins and the plate. At the inlet, temperature of the air is about 110ºC and at the exit the temperature reduces to 68ºC.

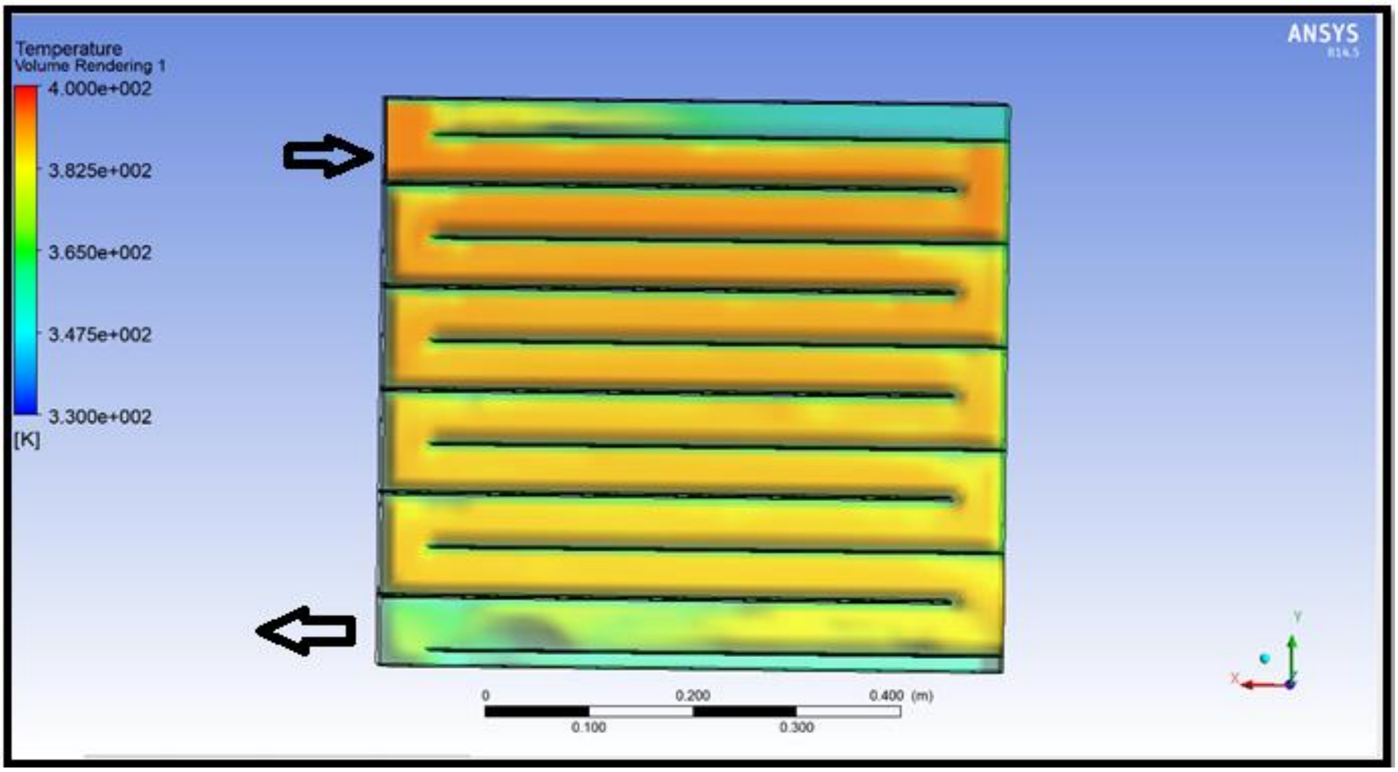


Figure 21 - Volume Rendering of Temperature of Flue Gas

Figure 22 shows the Volume Rendering of the temperature for the ambient air side. Air enters from the bottom and exits from the top. As the air passes by the fins and the plate, its temperature increases because it is at a lower temperature than the fins and the plate. Fins and plate on the cold side have a temperature of about 72ºC. At the inlet, temperature of the air is about 40ºC and at the exit the temperature increases to 65ºC.

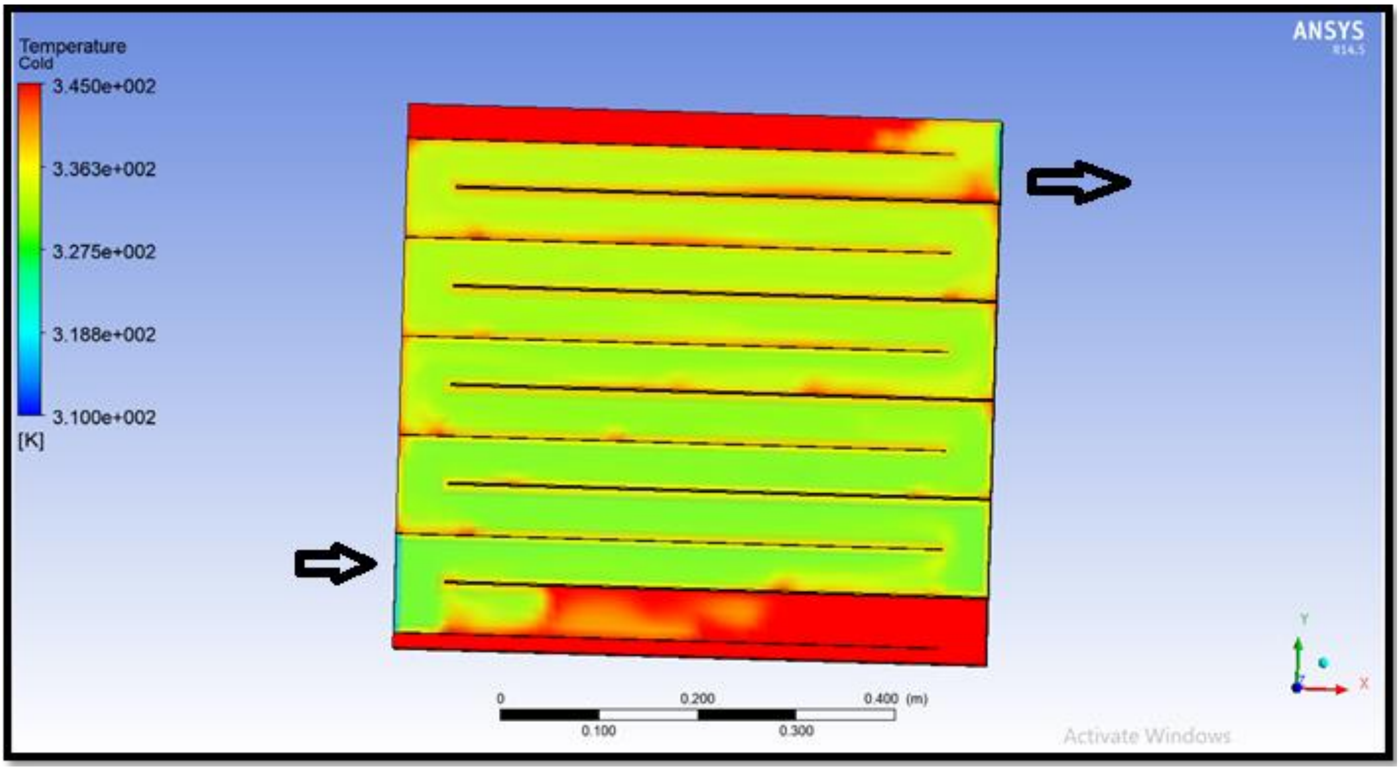


Figure 22 - Volume Rendering of Temperature of Ambient Air

4.4.3 Drying Chamber

Simulations were carried on the Drying Chamber design as well. The proposed design of the Drying Chamber is a new one, so CFD analysis on it was very important to get a real picture of whether the proposed design can heat the fruits to the required temperature or not. The proposed design was compared with the traditional one. The steps carried for the simulation are mentioned below:

1) After importing the model from the SOLIDWORKS and creating the NAMED SELECTION, a MESH was generated as shown in Figure 4.4.5, with the mesh details as well.

2) Energy equation was turned on. k-epsilon model (Realizable + Scalable Wall Functions) Model was used as the Reynold number of air is 10,224 and so turbulence occurs. Heat transfer coefficient was set to 1.84 W/m^{2}K **(Refer to Appendix 4).**

3) Temperature of the Inlet was set to 55^{o}C and the fruit temperature was set to 30^{o}C.

4) Velocity of the inlet was set to 0.1 m/s.

The results can be seen in the Figure 23 and Figure 24. The results show that the temperature variation in the fruits in all the 3 trays is very less, and this is what is required in the Drying Chamber. If there are different temperatures in different trays of fruits, then it is not a good thing as the quality of the end product varies.

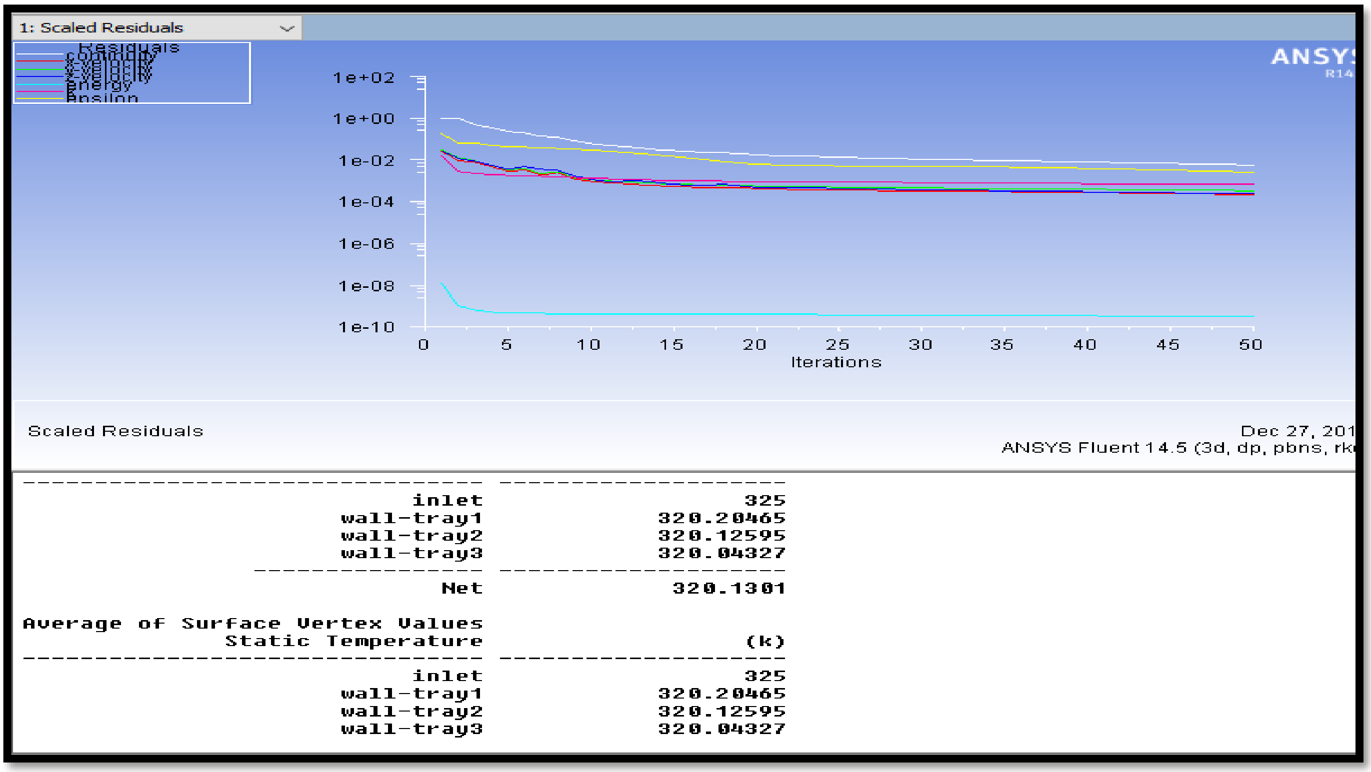


Figure 23 - Convergence Graph with Temperatures at the End of Simulation.

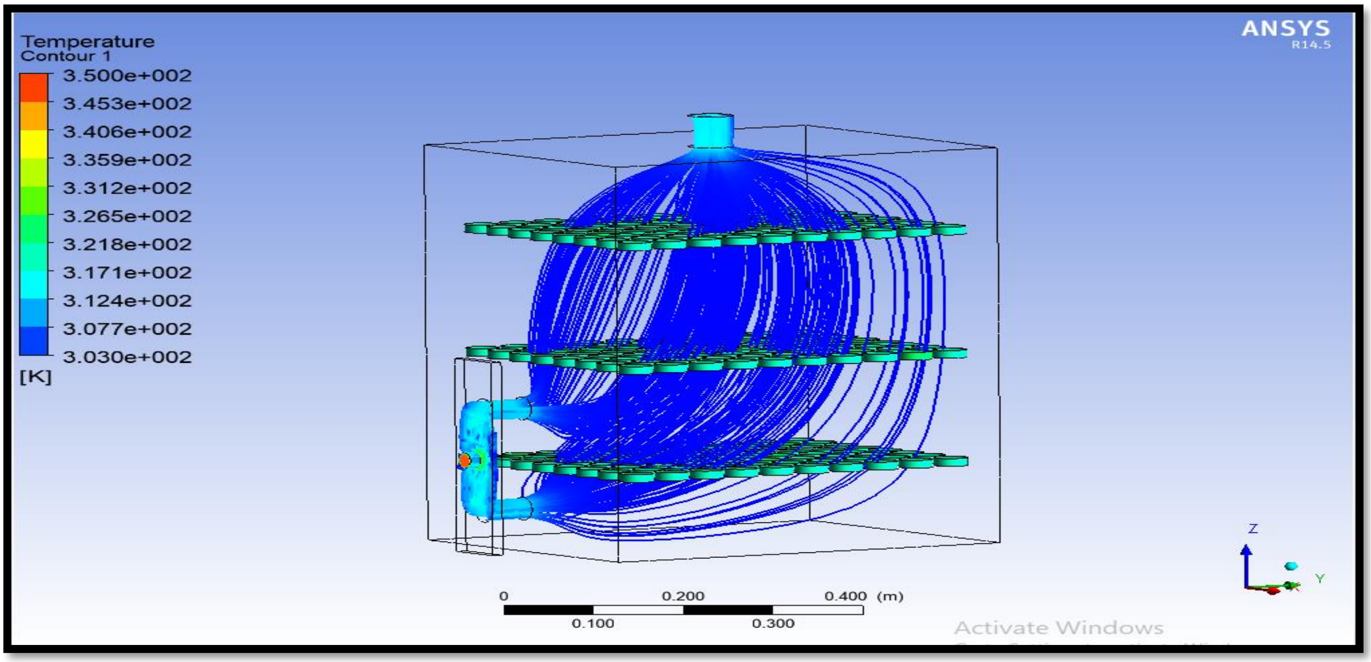


Figure 24 - Velocity Streamlines and Temperature Contours

Figure 24 shows the velocity streamlines in the Drying Chamber The path followed is what we expected. There is no reverse flow and air is distributed all the trays equally. As a result, temperature of each tray is almost equal.

As our design is an innovative and a new one we need to compare it with the traditional design. The results of which is presented in the Figure 25 and Figure 26.

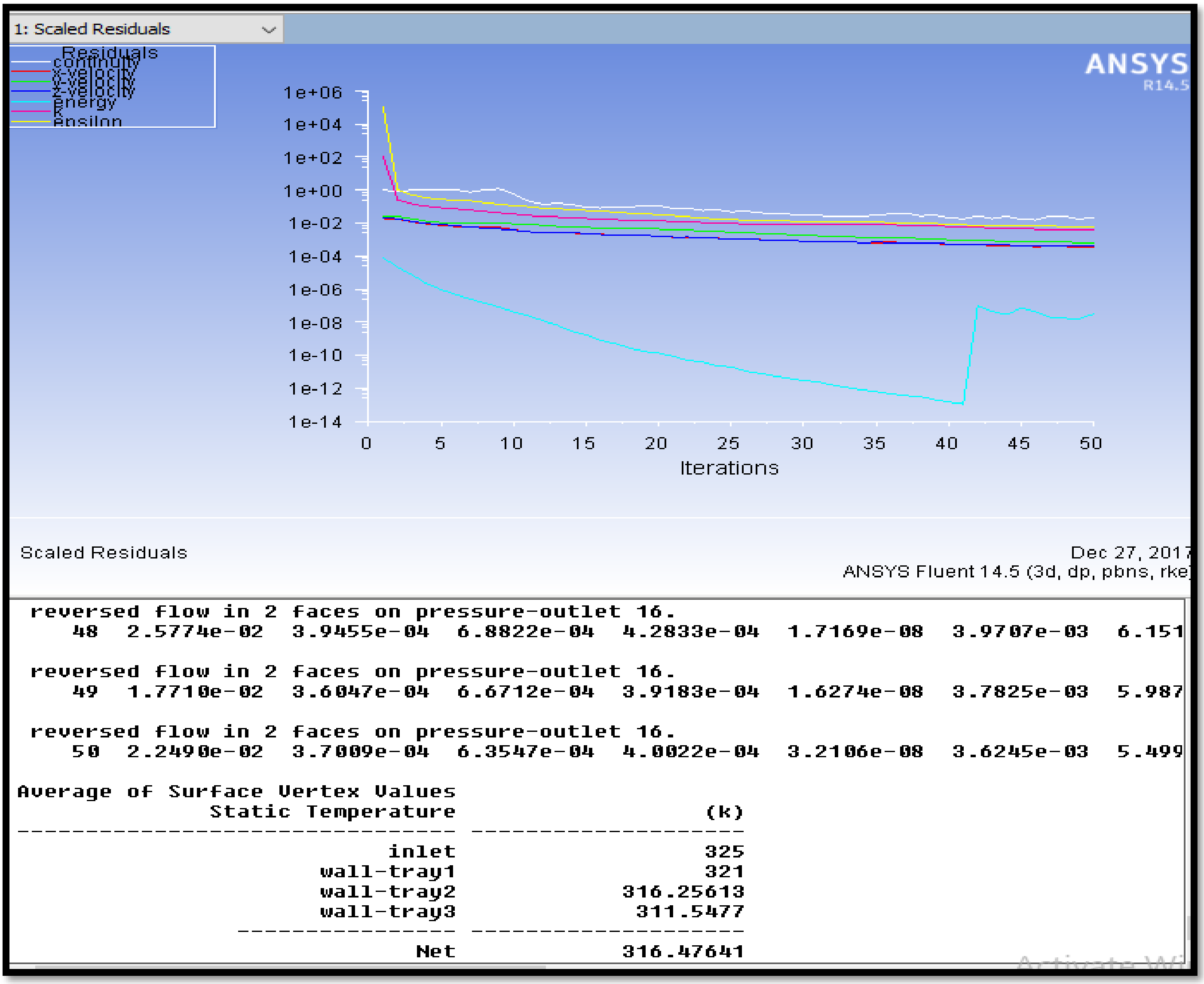


Figure 25 - Convergence Graph with Temperatures at the end of Simulation for Traditional Design

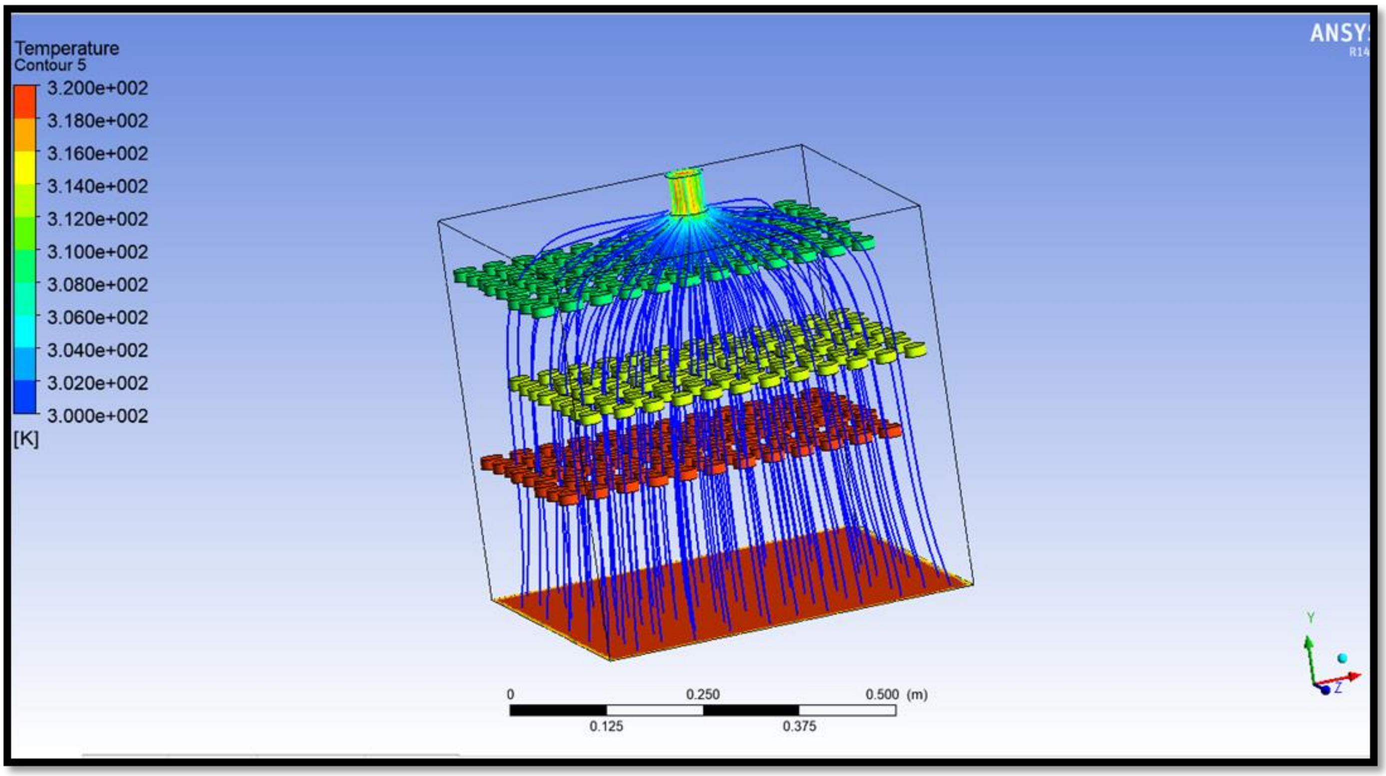


Figure 26 - Velocity Streamlines and Temperature Contour for Traditional Design

From the Figure 25 and Figure 26 it can be seen that in the traditional design there is quite a variation in drying trays temperature. The reason for this primarily due to the position and number of inlets. In the traditional design there is only one inlet from the bottom as the result the bottom tray is heated and then the remaining air heats the upper two trays. While in our proposed design there are two inlets on the sides as a result both the trays receive the air stream at a temperature of 50ºC. Thus heating both the trays evenly.

CHAPTER 5 - RESULTS

5.1 Numerical Analysis Results

Equations in Section 4 were solved in MATLAB and the result are plotted in the section below to get a more insight and understanding that how varying different parameters can lead to different outputs.

5.1.1 Solar Collector

Equations in Section 4.1.1 were solved in MATLAB and the result are plotted below. The code for the equations is written in **Appendix 5**.

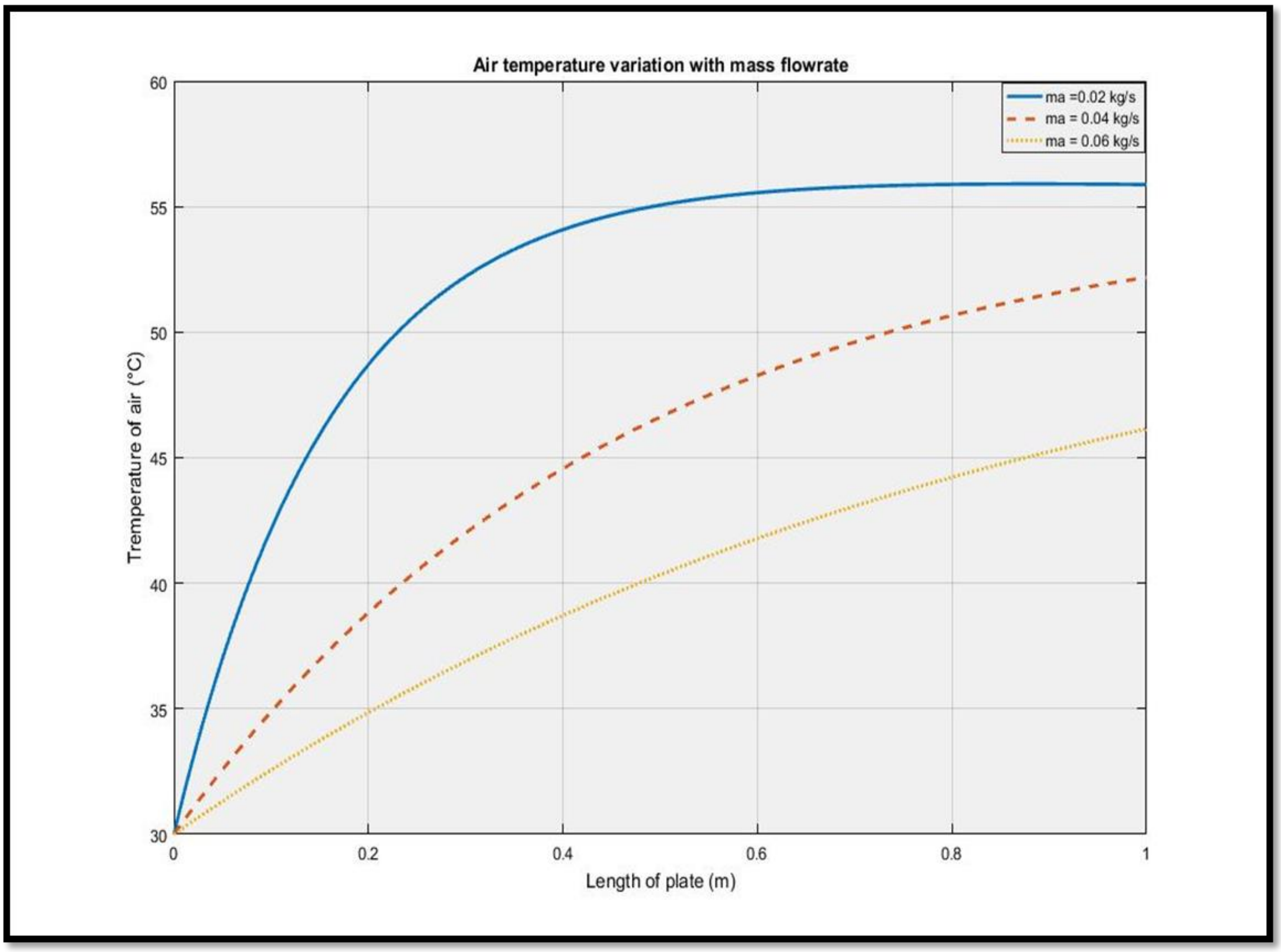


Figure 27 - Variation of Output Temperature of Solar Collector with Mass Flow Rate of Air

The above graphs suggest that decreasing the mass flowrate of air inside collector increases the outlet temperature of air, this is true theoretically as more slow the air travels it will have more time to absorb heat from the absorber plate. Also, the slope is greater at the start and decreases eventually when the air reaches at the outlet. The reason for this is that initially the temperature of air is 30ºC and the absorber plate temperature is about 80ºC but as it passes by its temperature increases to about 55ºC so change in temperature between plate and air decreases, hence decreasing heat transfer.

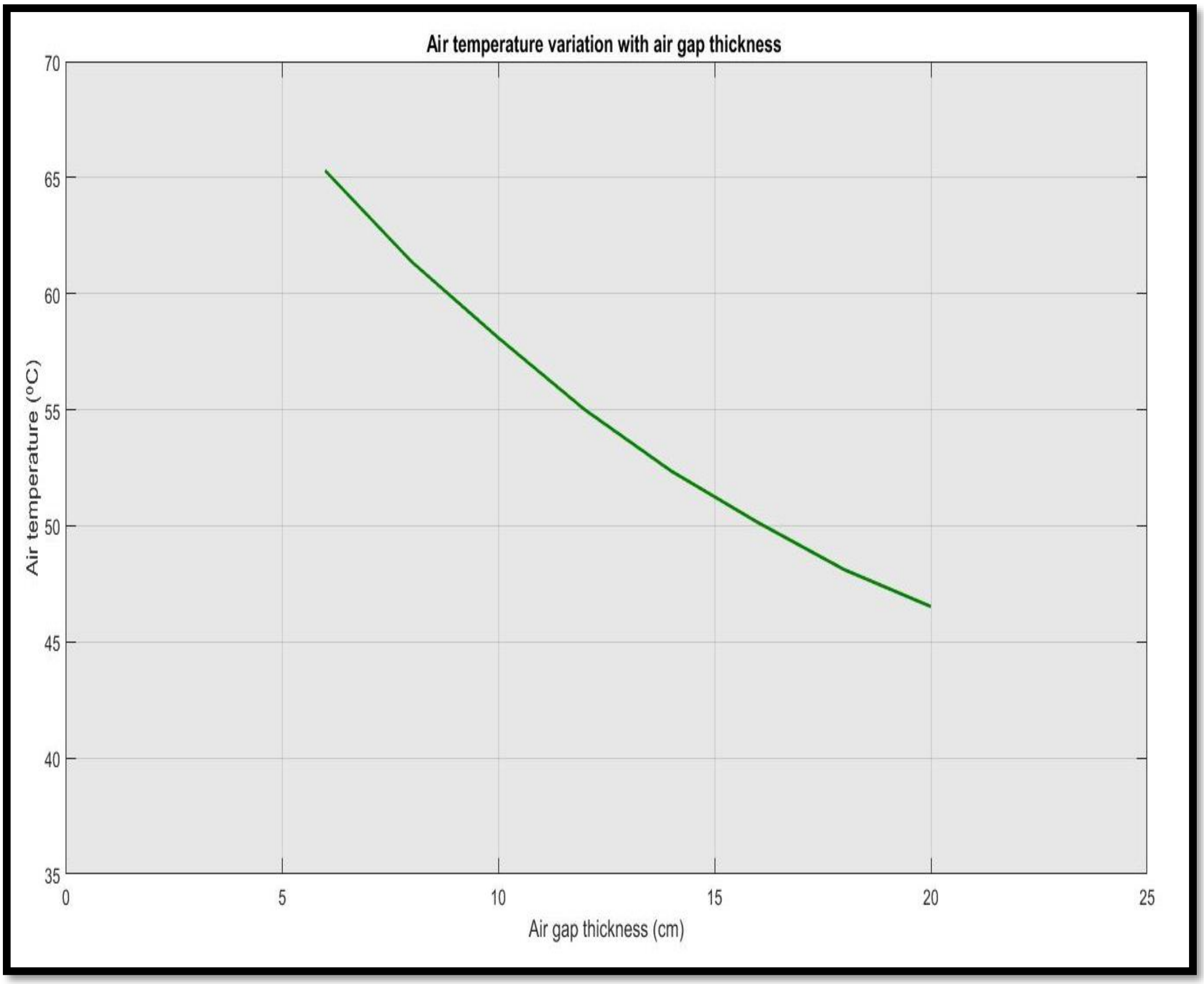


Figure 28 – Variation of Output Temperature of Air from Solar collector with Air Gap Thickness

The air gap (distance between plate and glass) is another important factor that effects the outlet temperature of air. As, it can be observed from Figure 24 that by decreasing the air gap the outlet temperature increases because the volume of air inside collector decreases and so less quantity of air can absorb more energy and increase its temperature more. However, keeping in mind the practical aspect that below 8 cm gap fabrication is tough, we did not consider any value of gap below 8 cm. It can also be seen that by increasing the gap the final temperature of air decreases, which it should as by increasing the thickness, more mass of air is present.

5.1.2 Biomass Heat Exchanger

The graph in Figure 29 shows that as we increase the temperature of flue gases, the length of heat exchanger plate required for heat transfer process decreases. The main reasons for this phenomena are:

- Heat flow rate is increasing
- Log mean temperature is increasing

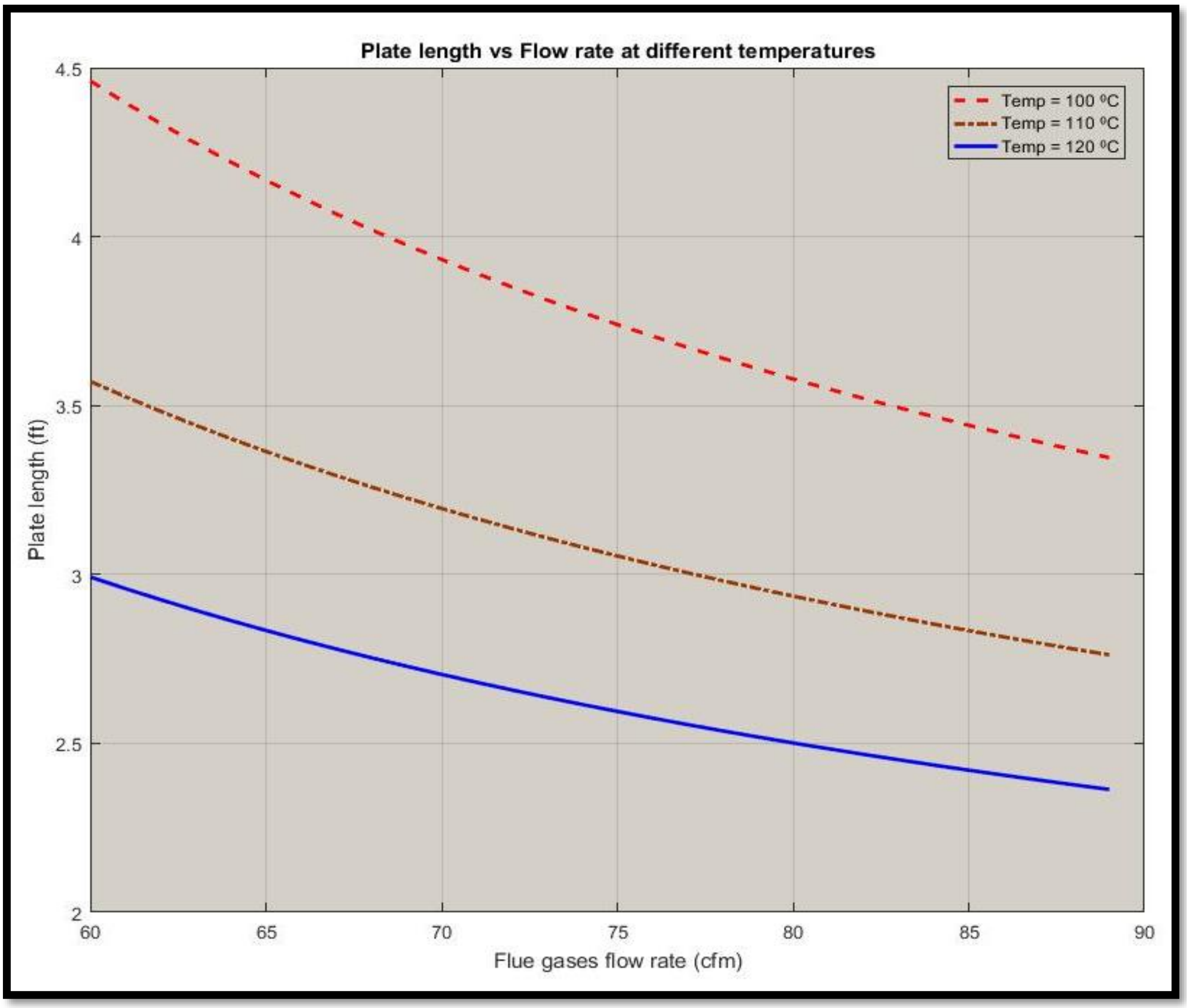
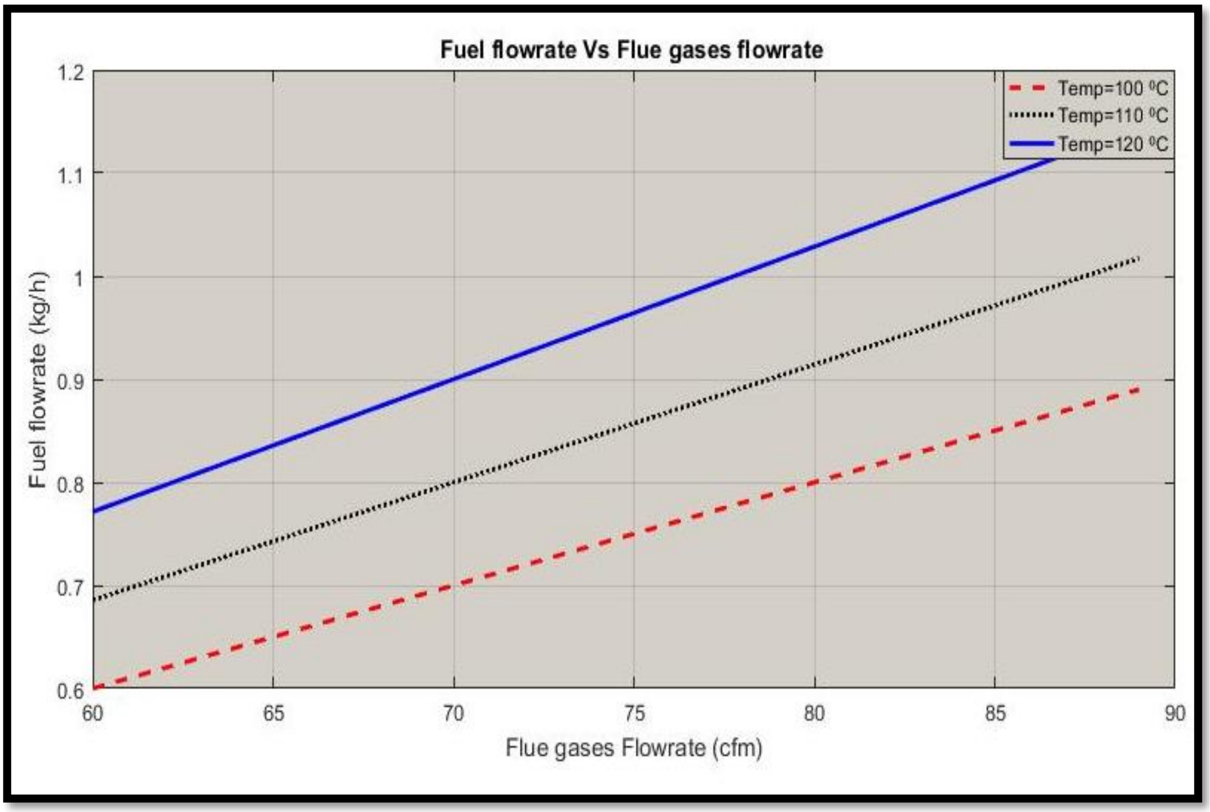


Figure 29 - Graph of Length of Heat Exchanger Plate vs Flow Rate at Different Temperatures



F

Figure 30 - Graph of Flow Rate of Fuel vs Flow Rate of Flue Gases

It is evident from Figure 29 that by increasing the flowrate of flue gases the length of plate decreases which makes heat exchanger more compatible. But it can be seen from Figure 30 that if we increase the flowrate of flue gases the inlet rate of fuel must also increase which is not desirable. So we compromise by increasing the length of plate and decreasing the flow rate of inlet fuel. The parameters chosen are listed below.

Parameters	Values
Desire temperature	60 ^{0}C
Outdoor temperature	30 ^{0}C
Flue gases temperature	110 ^{0}C
Material	Aluminum
Drying air flow rate	60 (CFM)
Flue gases flow rate	75 (CFM)

Table 3 – Design Parameters of Heat Exchanger

5.1.3 Drying Chamber

Figure 31 shows 2D axis symmetry model of the product (Grape) which is used to generate moisture and temperature graphs vs time. In this model boundaries are explicitly labelled.

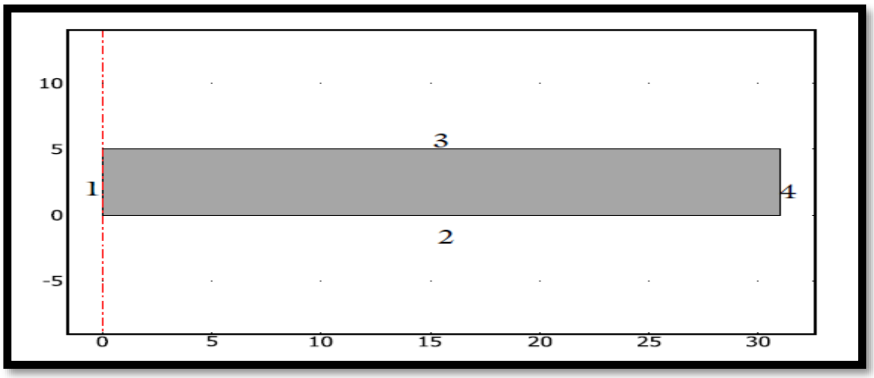


Figure 31 - Model Boundary Numbering

Below is 2-D plot of moisture concentrations vs time (Figure 32) using COMSOL for all boundaries for our model. It is evident from plot that moisture concentration decreases most rapidly through boundary no. 4 as it has largest exposed surface area to outside hot convective air which results in rapid removal of moisture through this boundary, compared to other boundaries of our model.

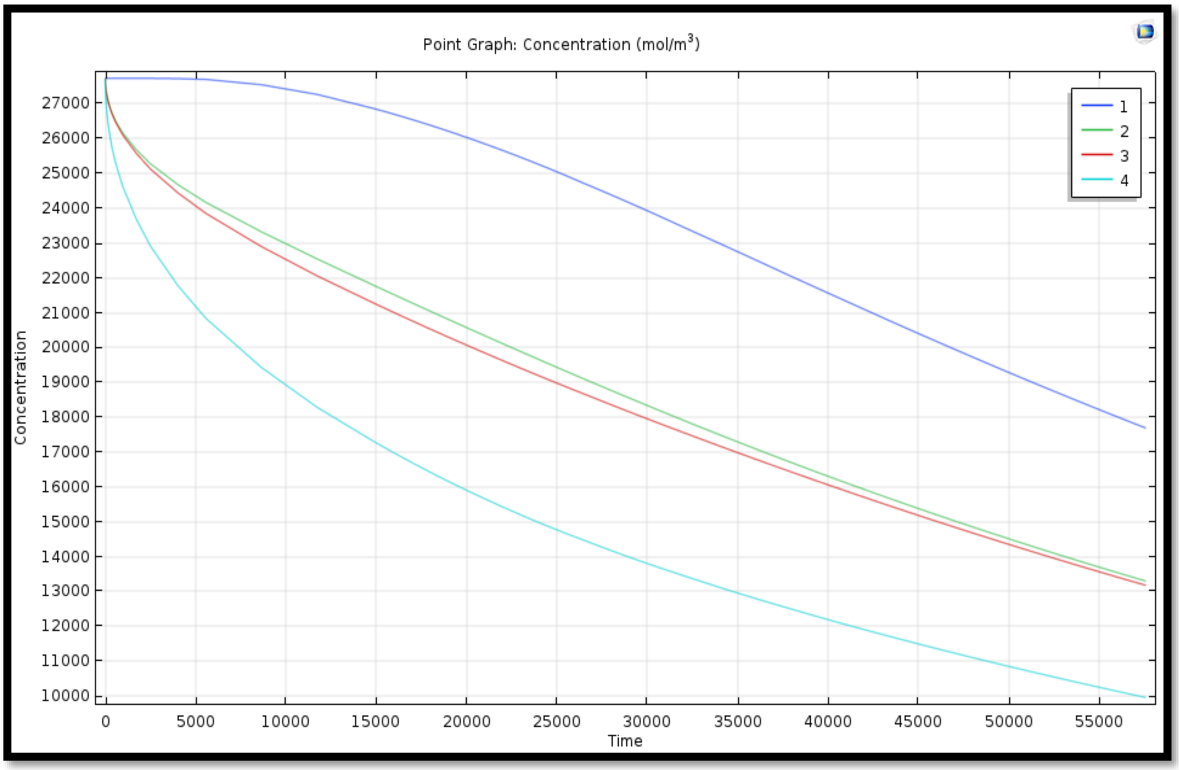


Figure 32 - Moisture Concentration vs Time

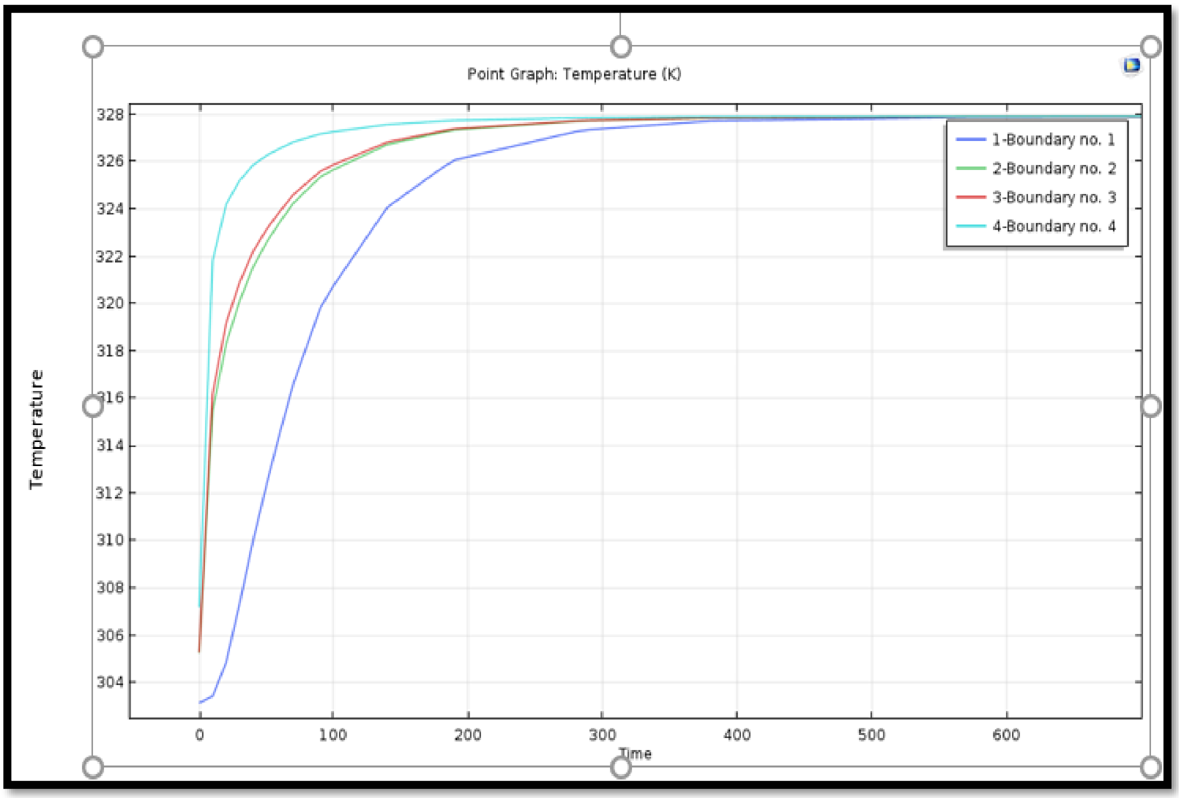


Figure 33 - Temperature vs Time

Figure 33 is a plot of temperature variation vs time across 4 boundaries of our model. It is evident from above figure that boundary 1 reaches a steady state value of temperature quickest because it is not directly exposed to outside environment and temperature across boundary 4 takes the longest to reach steady state value. It is also clear from above plot that after almost 400 seconds (6.5 minutes), all 4 boundaries reach a constant steady state value of temperature of magnitude 328 K

5.2 Experimental Analysis Results

An important part of this project was to know about the working of Solar Biomass Hybris Dryers in Pakistan. There wasn't enough research on the functioning and results of Hybrid dryers in Pakistan.

To address this issue, a prototype of dryer was manufacture for this purpose. The experiment was conducted in the month of April and May. We utilized that data and obtained readings about the temperature range in Islamabad.

It was very important to initialize the conditions so that a reasonable result is obtain. For solar collector, stagnation condition was maintained initially for 35 minutes so that the absorber plate achieved a temperature in the range of 65-75 ^{o}C. The heat exchanger was run for around 45-60 minutes to attain a suitable temperature of the heat exchanger plate. After the initial conditions were establish, the ambient air was supplied inside the solar collector and heat exchanger using blowers.

Although to get an idea of the temperature, humidity and air flowrate the setup was run many time, the result of the three experiments are given below. In the first experiment only the solar collector was run. In the second experiment, only biomass heat exchanger was used. For the third experiment a sample of ground nut (peanut) was used and compared with the open sun drying.

5.2.1 Experiment 1

In this experiment only the solar collector was tested and the temperature distribution of Ambient air (T_amb), T_plate, Drying Tray 1 (DT1) and (DT2) are shown below. The experiment was performed on 26th April from 10.00 to 17.00.

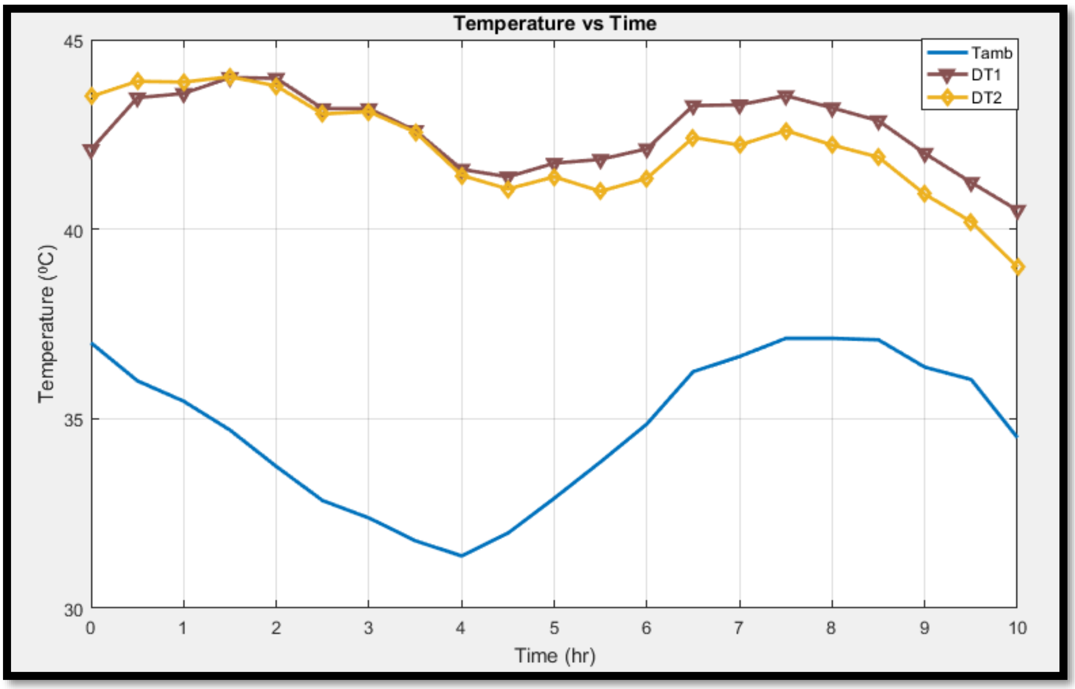


Figure 34 - Temperature vs Time (Experiment 2)

Time	Tamb (ºC)	Tplate (ºC)	DT1 (ºC)	DT2 (ºC)
10.00	26	34	30	30
10.15	23	45	32	33
10.30	22	50	34	36
10.45	27	53	39	38
11.00	27	55	39	40
11.15	28	53	40	40
11.30	29	51	42	41
11.45	30	57	43	43
12.00	36	58	42	42
12.15	37	54	43	43
12.30	38	59	40	40
12.45	39	65	44	44
13.00	36	66	45	45
13.15	37	70	46	46
13.30	35	74	44	45
13.45	34	76	46	44
14.00	33	77	46	47
14.15	32	75	45	46
14.30	30	74	46	45
14.45	31	71	44	43
15.00	32	68	43	42
15.15	29	58	42	43
15.30	24	56	39	40
15.45	25	57	38	38
16.00	21	50	36	37
16.15	26	45	35	34
16.30	29	42	40	40
16.45	24	43	32	34
17.00	22	42	32	31

Table 4 - Experiment 1 Reading

5.2.2 Experiment 2

In this experiment, only the Biomass Heat exchanger was tested and the temperature distribution of Ambient air (T_amb), Drying Tray 1 (DT1) and (DT2) are shown below.

Time(hr)	Tamb (ºC)	DT1 (ºC)	DT2 (ºC)
0.0	37	42	43
0.5	36	43	44
1.0	35	45	44
1.5	34	43	44
2.0	35	44	43
2.5	33	44	44
3.0	31	43	42
3.5	30	41	40
4.0	31	43	44
4.5	32	41	40
5.0	31	39	38
5.5	34	42	41
6.0	35	42	42
6.5	36	43	43
7.0	37	42	42
7.5	38	44	43
8.0	36	42	40
8.5	37	44	43
9.0	36	42	41
9.5	36	41	40
10.0	34	40	39

Table 5 - Experiment 2 Reading

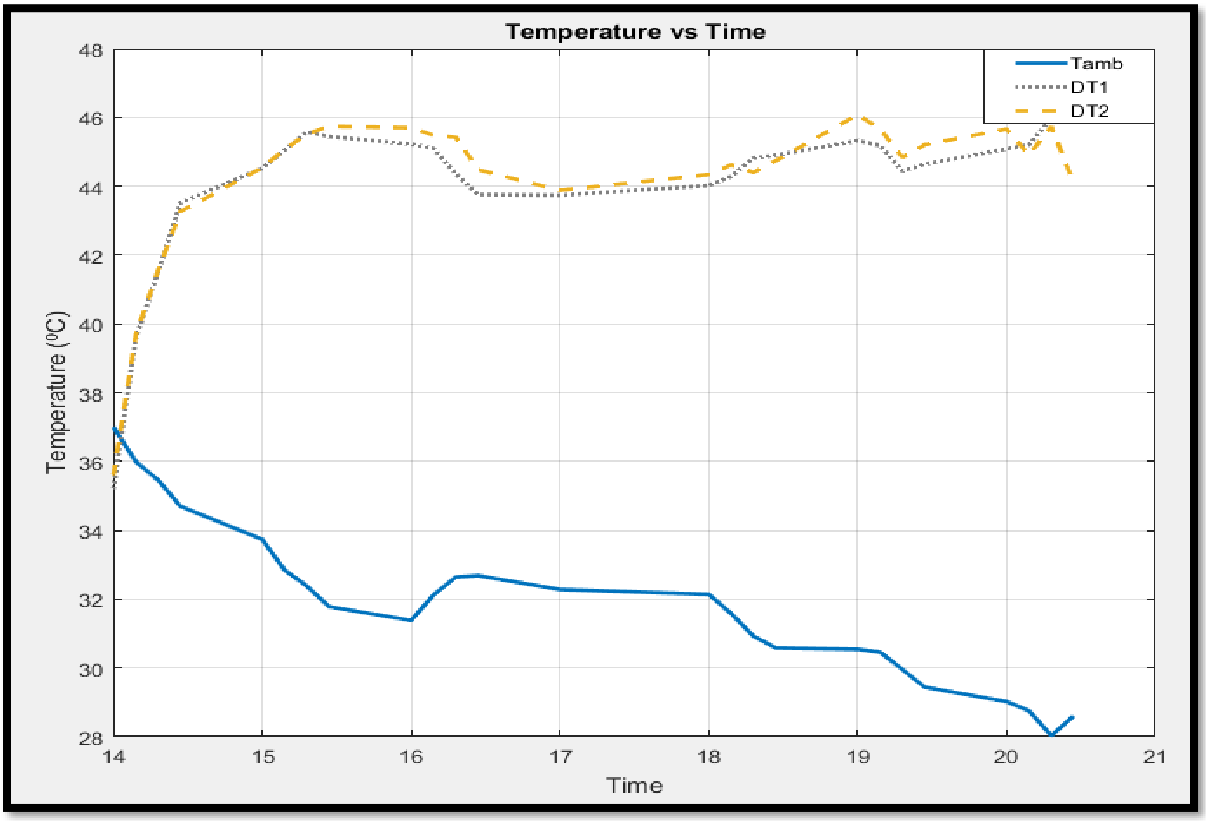


Figure 35 - Temperature vs Time (Experiment 2)

5.2.3 Experiment 3

In this experiment we used ground nut as our sample. Ground nut has a moisture content of 37%. To make it edible the moisture content should be reduced to 8-10%. We used a sample of 0.5 kg of raw ground nut and noted the time it takes to reach 0.35 kg (0.35 kg represents the moisture content of 8-10%.

mi = Intial Mass = 0.5 kg

Mi = Intial Moisture Concentration = 37 %

Mf = Final Moisture Concentration = 10 %

Mw = Amount of water removed (kg)

Mw = mi*(Mi-Mf)/(100-Mf)

Mw = 0.5*(37-10)/(100-10) = 0.15 kg

Remaining mass of product = 0.35 kg

Time (hr)	**Mass (g)**
0.0	476
1.0	436
2.0	406
3.0	380
4.0	365
5.0	350
6.0	341
7.0	330
8.0	321
9.0	310
10.0	304
11.0	296.0

Table 6 - Experiment 3 Reading

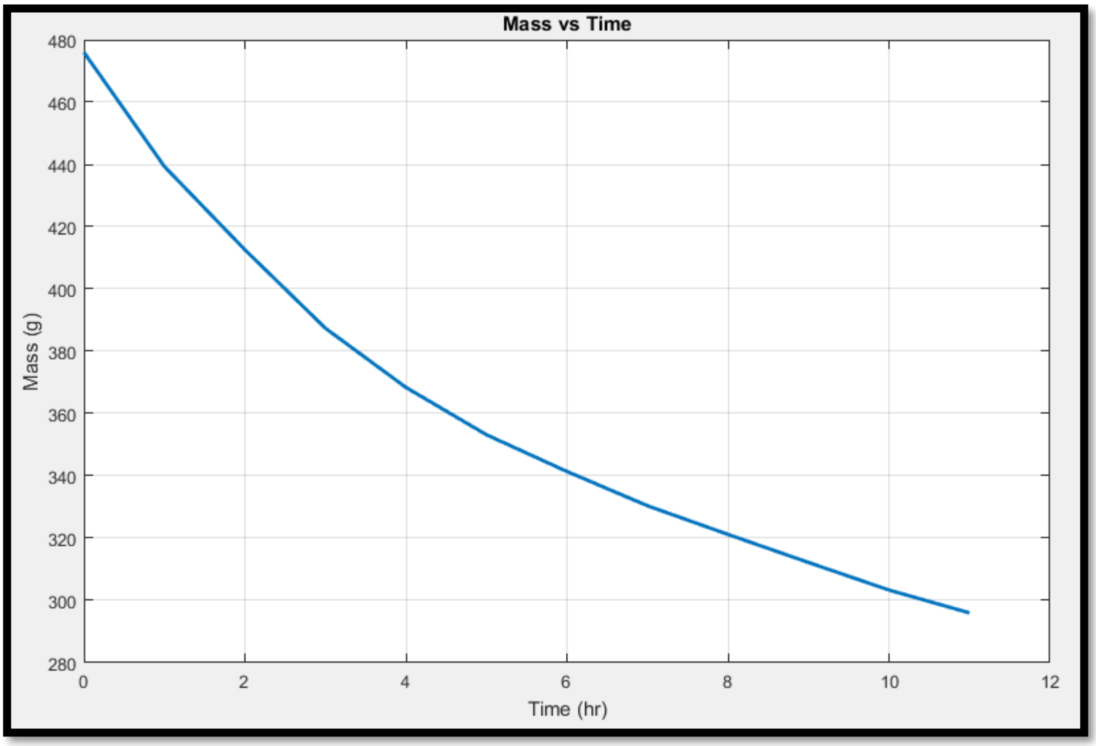


Figure 36 - Mass vs Time (Experiment 3)

The above graph shows the mass variation with time. It takes around 11 hours if we use our setup. We compared our result with a research paper written by D. Mennouche et.al [30] where it took around 12 hours.

CHAPTER 6 - COST ANALYSIS

Our Solar Biomass Hybrid Dryer setup is not entirely passive; it does require some electrical power during its operation. Complete power, airflow (associated with the blower) and cost analysis is presented in detail in this chapter.

6.1 Power Consumption

Data

Electric Power consuming devices in the Setup:

- 2 x fans
- 5 x blower

Power rating of fan 1 = 5.4 W.

Power rating of fan 2 = 5.4 W.

Combined power rating of the two fans = 10.8 W.

Power rating of one blower = 32 W

Power rating of 5 blowers = 160 W.

Combined power rating of all the devices = 10.8+160 = 170.8 W

6.1.1 Solar Collector

No. of operative hours of the Solar collector/day = 6 hours (10:00am to 4:00pm).
Food (Ground Nut) obtained via the solar collector/day = 0.25 kg/day

6.1.2 Biomass Heat Exchanger

No. of operative hours of the Biomass Heat exchanger = 6 hours (4:00 pm to 10:00 pm)

Food obtained via Heat exchanger/day = 0.25 kg/day

Total food dried from the entire setup/day = 0.25 = 0.5 kg/day

6.1.3 Drying Chamber

No. of operative hours of the Drying chamber = 12 hours (10:00 am to 10:00 pm)

$E = Px\ t$

E= 170.8 x 12=2.05 kWh

Energy consumption per kg of food collected per day = 2.05/0.5 = 4.1 kWh/kg (each day).

6.2 Cost Calculation

a) Cost of system (material and fabrication) = **Rs.55000**

Design life of the system = **10 years.**

Approximate number of days the system operates in one year = **365.**

Production of food per day = **1 kg.**

Total production of food in 10 years = 10 x 365 = **3650 kg**.

Cost of food/kg without drying = **Rs.150**

Cost of food/kg after drying = **Rs.280**

(b) Energy consumed each day = **4.1 kWh.**

Energy consumed in one month = 4.1 x 30 = **123 kWh.**

Cost of electricity per month = 123 x 6.49 = **Rs.800**

Cost of fuel for running Biomass per month = 30 x 6 x 30 = **Rs.5400**

Total cost per month = 5400 + 800 = **Rs.6200**

(c) Cost we face in one day = 6200/30 = **Rs.207**

(d) Pay back analysis

Average initial cost (without drying) = **Rs.150/kg**

Average selling cost = **Rs.280/kg.**

Selling cost per day = 280 = **Rs.280**

Payback period = investment/net earnings per day.

Net earnings per day = 280 – 207 = **Rs.73**

Payback period = 55000/73 = **753 days**.

Which is less than 2 year approximately **24 months!**

CHAPTER 7 - CONCLUSION

The maximum temperature obtained through the experimental setup is 47℃ while through analytical analysis is 55℃. The reason for this is due to heat losses through pipe although we have used insulation as well.

Solar radiation can be effectively and efficiently utilized for drying of agricultural produce in our environment if proper design is carried out. This was demonstrated and the solar dryer designed and constructed exhibited sufficient ability to dry agricultural produce most especially food items to an appreciably reduced moisture level. Locally available cheap materials were used in construction making it available and affordable to all and sundry especially peasant farmers. This will go a long way in reducing food wastage and at the same time food shortages, since it can be used extensively for majority of the agricultural food crops.

Apart from this, solar and biomass sources of energy are required for its operation which is readily available, and it is also a clean form of energy. It protects the environment and saves cost and time spent on open sun drying of agricultural produce since it dries food items faster. The food items are also protected in the dryer than in the open sun, thus minimizing the case of pest and insect attack and also contamination.

From the test carried out, the following conclusions are made:

- The solar biomass dryer can raise the ambient air temperature to a considerable high value for increasing the drying rate of agricultural crop.
- The product inside the dryer requires lesser frequent attention compare with those in the open sun drying in order to prevent attack of the product by rain or pest (both human and animals).
- There is ease in monitoring when compared to the natural sun drying technique.
- The capital cost involved in the construction of a solar biomass dryer is much lower to that of a mechanical dryer.

RECOMMENDATIONS

There are some recommendations about the improving the performance of existing solar food dryers especially in the aspect of reducing the drying time and storage of heat energy within the system.

The first recommendation is using better materials in the manufacturing of solar collector. It includes the usage of high quality aluminum as absorber plate, and better glazing materials.

Similarly, a smart temperature and humidity control system can be designed which will operate solar dryers according to the needs of the product to be dried while also taking weather forecast into account.

To improve the functionality of solar dryer, and make it efficient on overcast days, we can use phase change material like paraffin wax. Phase change material can maintain high temperature in solar collector for larger periods of time, and in this way solar dryer can also be used in night time.

REFERENCES

[1] Rosa Rolle S, "Postharvest management of fruit and vegetables in the Asia-Pacific region. Asian Productivity Organization".

[2] Jangam S.V. and Mujumdar, A.S., "Basic Concepts and Definitions in Drying of Foods, Vegetables and Fruits".

[3] M. Y H Othman, K. Sopian, B. Yatim and W. R W Daud, "Development of advanced solar assisted drying systems".

[4] C.L. Hii, S.V. Jangam, S.P. Ong and A.S. Mujumdar, "Solar drying: Fundamentals, applications and innovations".

[5] Umogbai V.I, Iorter H.A, "Design, construction and performance evaluation of a passive solar dryer for maize cobs".

[6] Rajeshwari N. and Ramalingam A., "Low cost material used to construct effective box type solar dryer".

[7] O. V. Ekechukwua* and B. Norton, "Review of solar energy drying systems".

[8] Erick Cesar Lo´pez-Vidan, Lilia L. Me´ndez-Lagunas and Juan Rodrı´guez-Ramı´rez, "Efficiency of a hybrid solar–gas dryer".

[9] Abdul Jabbar N Khalifa and Ahmad M. Hamood, "Effect of insulation thickness on the productivity of basin type solar stills: An experimental verification under local climate".

[10] J.S. Gawande and L.B. Bhuyar,"Effect of glass cover thickness on the Performance of stepped type solar still".

[11] Arinze EA, Sokhansanj S, Schoenau GJ and Trauttmansdorff FG, "Experimental evaluation, simulation and optimisation of a commercial heated air batch hay drier".

[12] Prakash S, Jha SK and Datta N, "Performance evaluation of blanched carrots dried by three different driers".

[13] Ekechukwu OV, "Review of solar-energy drying systems I: an overview of drying principles and theory".

[14] FAO, "Agricultural services bulletin No.119. Food and Agriculture Organization of the United Nations".

[15] Fudholi A, Sopian K, Othman MY, Ruslan MH and AlGhoul MA, "Heat transfer correlation for the V-Groove Solar Collector".

[16] Sopian K, Alghoul MA, Alfegi EM, Sulaiman MY and Musa EA, "Evaluation of thermal efficiency of double pass solar collector with porous nonporous media".

[17] James Stiling, Simon Li, Pieter Stroeve, Jim Thompson, Bertha Mjawa, Kurt Kornbluth and Diane M. Barrett, "Performance evaluation of an enhanced fruit solar dryer using concentrating panels".

[18] Prasad J. and Vijay V.," Experimental studies on drying of Zingier official, Curcuma longa l. and Tinospora cord folia in solar-biomass hybrid drier".

[19] Madhlopa A. and NgwaloG.," Solar dryer with thermal storage and biomass backup heater".

[20] Tarigan E. and Tekasakul P., "A mixed-mode Natural Convection Solar Dryer with Biomass Burner and Heat Storage Back-up Heater".

[21] Jaishree Prasad *, V.K. Vijay, G.N. Tiwari and V.P.S. Sorayan, "Study on performance evaluation of hybrid drier for turmeric (Curcuma longa L.) drying at village scale".

[22] Thomas R. Miles, Thomas R. Miles, Larry L. Baxter, Richard W. Bryers, Bryan M. Jenkins and Laurance L. Oden, "Boiler deposits from firing biomass fuels, Biomass and Bioenergy, Volume 10, Issue 2, 1996, Pages 125-138".

[23] Matti Parikka, "Global biomass fuel resources, Biomass and Bioenergy, Volume 27, Issue 6, 2004, Pages 613-620, ISSN 0961-9534".

[24] Changdong Sheng and J.L.T. Azevedo, "Estimating the higher heating value of biomass fuels from basic analysis data, Biomass and Bioenergy, Volume 28, Issue 5, 2005, Pages 499-507".

[25] Yungus A. Cengel and Afshin Jahanshahi Ghajar, "Heat and Mass Transfer: Fundamentals and Applications".

[26] Abdulelah Ali Al-Jumaah, Abdullah Mohamed Asiri, Mohamed Fadil Alshehri, Ahmed Mohamed Deash and Fahd Minajy Al-Hamzi, "Design and Construction of a Solar Drying System for Food Preservation".

[27] Sandeep Kumar Patel and Alkesh M. Mavani, "Shell & Tube Heat Exchanger Thermal Design with Optimization of Mass Flow Rate and Baffle Spacing".

[28] V. K. Sharma, S. Sharma, R. A. Ray and H. P. Garg, "Design and Performance Studies of a Solar Dryer Suitable for Rural Applications".

[29] Juneyd F. Dadi, Iqbal H Jujara, Jhakimohammad I. Malek, Prerak S. Patel, Pankaj S. Bhadoriya and Vatsal M. Patel, "A Review Paper on Solar Dryer".

[30] D. Mennouche, B. Bouchekima, S. Zighmi, A. Boubekri, S. Boughali and A. Matallah, "An Experimental Study on the Drying of Peanuts using a Direct Solar Dryer".

APPENDIX 1 – CALCULATION FOR OVERALL HEAT TRANSFER COEFFICIENT

U_{loss} refers to the Overall Heat Transfer Coefficient for conduction between plate and the outside ambient conditions. Mathematically,

$$U_{loss} = U_{bottom} + U_{TwoSides}$$

Refer to the resistance model in **Figure 34**. From the resistance model we obtain,

$$R_{bottom} = \left(\frac{1}{U_{bottom} * A_p}\right) = \left(\frac{R_{thermapole} * t_{thermapole}}{A_p}\right) + \left(\frac{R_{plywood} * t_{plywood}}{A_p}\right) + \frac{R_{air}}{A_p}$$

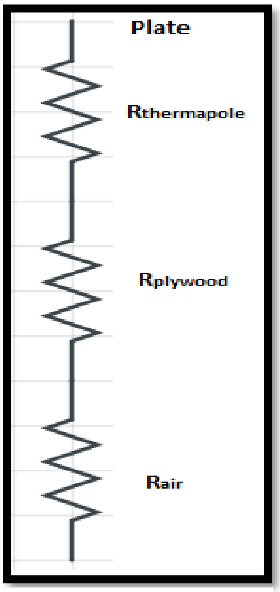


Figure 37 - Resistance Model for Bottom of Absorber Plate

$$\frac{1}{U_{bottom} * A_P} = \left[\frac{31.6 * \frac{25.4}{1000}}{0.94 * 0.65}\right] + \left[\frac{8.67 * 0.6 * \frac{25.4}{1000}}{0.94 * 0.65}\right] + \left[\frac{0.044}{0.94 * 0.65}\right]$$

U_{bottom} = 1.02 W/m^{2}K.

#	Material	R value (mK/W)
1	Plywood	8.67
2	Polystyrene (Thermapole)	31.6
3	Air	0.044 m

Table 7 - R values of the Insulation Materials (Using ASHRAE FUNDAMENTAL TABLES)

Refer to the resistance model in **Figure 35**. From the resistance model we obtain,

$$R_{TwoSides} = 2\left[\left(\frac{R_{plywood} * t_{plywood}}{A_P}\right) + \left(\frac{R_{thermapole} * t_{thermapole}}{A_P}\right) + \frac{R_{air}}{A_P}\right]$$

$$\mathbf{R_{TwoSides} = 3.2\ W/mK}$$

$U_{TwoSides} = 1/(\mathbf{R_{TwoSides}} * \boldsymbol{A_{side}})$

$U_{TwoSides} = 1/(3.2*0.94 *0.08)$

$U_{TwoSides}$ = 4.16 W/m^{2}K

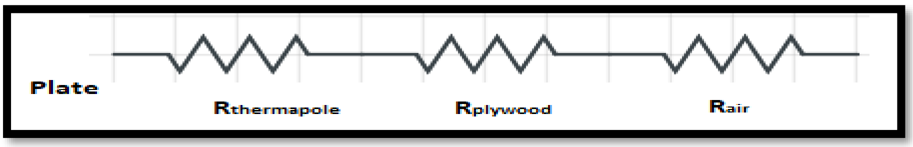


Figure 38 - Resistance Model for One side of Absorber Plate

Finally, U_{loss} = 4.16 + 1.02 = 5.18 W/m^{2}K

APPENDIX 2 – DETERMINATION OF STAGNATION TEMPERATURE

Stagnation Temperature is the maximum possible temperature of the absorber plate. Stagnation temperature can be achieved only if there is no air inside collector.

Using (1),

Energy absorbed by plate = Useful energy to air + Energy loss

$$\left(I * A_p * abs_p * trans\right) = Q_u + Q_{loss}$$

As there is no air so $Q_u = 0$

$$T_{stagnation} = T_{amb} + [(I * abs_p * trans)/U_{loss}]$$

$$T_{stagnation} = 30 + [(500*0.6*0.917)/ 5.18)]$$

$$\mathbf{T_{stagnation} = 80^{o}C.}$$

APPENDIX 3 – MATLAB CODE FOR FINDING TIME FOR PLATE TO REACH STAGNATION TEMPERATURE:

The time required for the plate to reach stagnation temperature is calculated using MATLAB by modeling all the equations in Sec 4.1 neglecting air. Time obtained from MATLAB is 1200 s =20 minutes.

MATLAB Code:

```
clc
clear all

Tg(1) =25 ;         %% Glass temperature
Tamb=30%% Ambient temperature
Tp(1) =30; %% Plate temperature
Tsky= Tamb-6;
V = 3;  %%speedof air outside collector in m/s
I = 400; %% Intensity of solar radiation
Lp = 0.8; %%Lenghth of plate
Wp = 0.6; %%Width of plate
Thickness = 0.001 ; %%Thickness of plate in m
Ap = Lp*Wp ;  %%Area of top psurface of plate
abs_p= 0.65; %% Absorptivity of aluminium plate
trans = 0.917; %%Transmitivity of glass
p_Al = 2700 ; %% Denity of aluminium
Vol = Lp*Wp*Thickness ; %%Volume of plate
mp = p_Al*Vol ; %%Volume of plate
cp_p = 877 ; %%Specific heat capacity of plate
vair =0.15; %%Speed of air inside the collector in m/s
steffan= 5.67*10^(-8) ;%% steffan boltzman constant
ep = 0.04;%%emissivity of plate
eg =0.94; %% emissivity of glass
cp_a = 1005 ; %% specific heat capacity of air
p_air = 1 ; %% density of air
Airthickness = 0.08 ;%%Thickness of air inside collector
vol_air = Airthickness*Wp*Lp; %% volume of air inside solar collector
ma= p_air* vol_air; %% mass of air inside collecor
abs_g = 0.05;%% absorptivity og glass
Ag = Lp*Wp ;%% Area of glass
hc_g_sky= 2.8+3*V; %%convective  heat transfer coefficient  between glass and sky
p_glass= 2260  ; %% Density of glass
thicknessofglass= 0.003; %%Thickness of glass
volglass= Ag*thicknessofglass; %Volume of glass
mg= p_glass*volglass;%%mass of glass
cp_g = 840 ; %%Specific heat capacity of glass
theeta =pi/6; %% Inclination angle of collector
```

```
  Re_outside = (V*Lp)/(1.963*10^(-5))  ; %% Reynold number of flowing air outside collector
Nu_g_sky = 0.664*(Re_outside^(0.5))*(0.72^(1/3)); %% Nusselt number for outside air and glass

hc_g_sky= Nu_g_sky * 0.02735/Lp; %%heat transfer coefficient between glass and outside air

i=1;
timestep=0.001;
Totaltime= 1200

for time=0:timestep:Totaltime

hr_p_g= (steffan*((Tp(i)+273)^2+(Tg(i)+273)^2)*(Tp(i)+273+Tg(i)+273))/((1/ep)+(1/eg)-1) ;%%
  radiation heat transfer coefficient between plate and glass
Tg(i+1) =timestep* ( ((I*abs_g*Ag)-(eg*steffan*((Tg(i)+273)^4 -(Tsky+273)^4) *Ag) + (hr_p_g*Ap*
  (Tp(i)-Tg(i))) -(hc_g_sky*Ag*(Tg(i)-Tsky)))   /(mg*cp_g)) +Tg(i);
Tp(i+1)=timestep*(((I*Ap*abs_p*trans)- (0.535*(Tp(i)-Tamb)))-  (hr_p_g*Ap* (Tp(i)-Tg(i+1))))
  /(mp*cp_p) +Tp(i)  ;

i=i+1;

end

 disp('Tp=:')
 Tp(i)
 disp('Tg=:')
 Tg(i)
```

APPENDIX 4 – CALCULATION OF BOUNDARY CONDITIONS:

Using continuity equation,

Outlet mass from solar collector = Inlet mass flow rate of drying chamber

$$(v_{chamber} * A_{chamber}) = (v_{Solar\ Collector} * A_{SolarCollector})$$

$$(\mathrm{v}_{\mathrm{chamber}} * 0.44 * 0.54) = (0.1 * 0.08 * 0.68)$$

$$\mathrm{v}_{\mathrm{chamber}} = 0.275 \text{ m/s.}$$

$$\mathrm{Re} = \left(\frac{\rho_{\mathrm{air}} * \mathrm{v}_{\mathrm{chamber}} * \mathrm{D}_{\mathrm{h}}}{\mu}\right) = \frac{\left(1 * 0.275 * 4 * \frac{A_{chamber}}{(2 * 0.44) + (2 * 0.54\,)}\right)}{(1.846 * 10^{-5})} = 10223.6$$

This is a fully developed Turbulent flow as Re>10,000.

$$Nu = 0.023(Re)^{0.8}(\mathrm{Pr})^{0.3}$$

$$Nu = (h_{chamber} * D_h)/k_{air}$$

$h_{chamber}$ =1.84 W/m^{2}K

APPENDIX 5 – MATLAB CODE FOR SOLAR COLLECTOR OPERATION

```
clc
clear all
Ta(1) = 30;     %% air  temperature inside collector
Tg(1) =50 ;   %% Glass temperature
Tamb=30%% Ambient temperature
Tstagnation =  Tamb+50; %% proof in copy
Tp(1) =Tamb+50; %% Plate temperature.We wait for 500 sec so that plate temperature increases
during this time no air is passed inside collector.See code
"FinalCodeUsingGuassSiedelStagnationTemperature"

Tsky= Tamb-6;
V = 3;  %%speedof air outside collector in m/s
I = 400; %% Intensity of solar radiation
Lp = 0.9456; %%Lenghth of plate
Wp = 0.6; %%Width of plate
Thickness = 0.001 ; %%Thickness of plate in m
Ap = Lp*Wp ;  %%Area of top psurface of plate
abs_p= 0.65; %% Absorptivity of aluminium plate
trans = 0.917; %%Transmitivity of glass
p_Al = 2700 ; %% Denity of aluminium
Vol = Lp*Wp*Thickness ; %%Volume of plate
mp = p_Al*Vol ; %%Volume of plate
cp_p = 877 ; %%Specific heat capacity of plate
vair =0.1/2; %%Speed of air inside the collector in m/s
steffan= 5.67*10^(-8) ;%% steffan boltzman constant
ep = 0.04;%%emissivity of plate
eg =0.94; %% emissivity of glass
cp_a = 1005 ; %% specific heat capacity of air
p_air = 1 ; %% density of air
Airthickness = 0.08 ;%%Thickness of air inside collector
vol_air = Airthickness*Wp*Lp; %% volume of air inside solar collector
ma= p_air* vol_air; %% mass of air inside collecor
abs_g = 0.05;%% absorptivity og glass
Ag = Lp*Wp ;%% Area of glass
hc_g_sky= 2.8+3*V; %%convective  heat transfer coefficient  between glass and sky
p_glass= 2260  ; %% Density of glass
thicknessofglass= 0.003; %%Thickness of glass
volglass= Ag*thicknessofglass; %Volume of glass
mg= p_glass*volglass;%%mass of glass
cp_g = 840 ; %%Specific heat capacity of glass
theeta =pi/6; %% Inclination angle of collector
Re_outside = (V*Lp)/(1.963*10^(-5))  ; %% Reynold number of flowing air outside collector
Nu_g_sky = 0.664*(Re_outside^(0.5))*(0.72^(1/3)); %% Nusselt number for outside air and glass
```

```
hc_g_sky= Nu_g_sky * 0.02735/Lp; %%heat transfer coefficient between glass and outside air
%% forced convection inside collector between air and plate
Re = (vair*Lp)/(1.963*10^(-5))  ; %% Reynold number of flowing air inside collector
NuForcedConvection = 0.664*(Re^(0.5))*(0.72^(1/3));
hc_p_aForcedConvection  = NuForcedConvection*0.02735/ (Lp);
i=1;
timestep=0.001;
Totaltime= Lp/vair;

for time=0:timestep:Totaltime
 %% natural convection inside collector between air and plate
B =   2/((Tp(i)+Tg(i)+273+273));          %%Coefficient of thermal expansion
Ra=9.81*B*(Tp(i)-Tg(i))*(Airthickness/2)^3 *0.7228 /(1.965*10^(-5))^2   ;

x= 1-(1708/(Ra*cos(theeta)))    ;
   if(x<0)
       x=0;
   end;
   y= (((Ra*cos(theeta))^(1/3))/18) - 1   ;
   if (y<0)
       y=0;
   end;
  z= 1- 1708*((sin(1.8*theeta))^(1.6))/(Ra*cos(theeta));
  NuNaturalConvection= 1+ (1.44*x*z) + y;
 hc_p_aNaturalConvection = ( NuNaturalConvection *0.02735)/(Airthickness);
 hc_p_a = hc_p_aNaturalConvection + hc_p_aForcedConvection;
hr_p_g= (steffan*((Tp(i)+273)^2+(Tg(i)+273)^2)*(Tp(i)+273+Tg(i)+273))/((1/ep)+(1/eg)-1) ;%%
radiation heat transfer coefficient between plate and glass

 %% start from here checking code
Tg(i+1) =timestep* ( ((I*abs_g*Ag)+ (hc_p_a*Ap*(Ta(i)-Tg(i)))-(eg*steffan*((Tg(i)+273)^4 -
(Tsky+273)^4) *Ag) + (hr_p_g*Ap* (Tp(i)-Tg(i))) -(hc_g_sky*Ag*(Tg(i)-Tsky)))   /(mg*cp_g))
+Tg(i);
Tp(i+1)=timestep*(((I*Ap*abs_p*trans)- (0.535*(Tp(i)-Tamb))- (hc_p_a*2*Ap*(Tp(i)-Ta(i)))-
(hr_p_g*Ap* (Tp(i)-Tg(i+1)))) /(mp*cp_p)) +Tp(i)  ;
Ta(i+1)=timestep*(((hc_p_a*Ap*2*(Tp(i+1)-Ta(i)))- (hc_p_a*Ag*(Ta(i)-Tg(i+1)))))/(ma*cp_a))
+Ta(i);

i=i+1;
end

 disp('Tp=:')
 Tp(i)
 disp('Tg=:')
 Tg(i)
disp('Ta=:')
Ta(i)
```

APPENDIX 6 – MATLAB CODE FOR LENGTH DETERMINATION OF HEAT EXCHANGER

MATLAB Code:

```
HCp=1009;%%average cp value of hot gases
Hden=0.9458;%%average density value of hot gases
HInT=120;%%average temperture value of inlet hot gases
Hlength=0.0254*1.5;%%height of cross section area of  hot gases inlet
Hwidth=0.0254*2;%%width of cross section area of  hot gases inlet
HFR=70*0.0004719474;%%flow rate of hot flue gases
HAc=Hlength*Hwidth;%%cross section area of hot flue gases side
HDh=2*Hlength*Hwidth/(Hwidth+Hlength);%%Hydraulic diameter of hot gases side
Hv=HFR/HAc;%%velocity of hot flue gases
HRe=Hv*HDh/(2.306*(10^-5));%%reynolds's number of hot gases
HNu=0.023*(HRe^0.8)*(0.711)^0.4;%%Nusselt number of hot gases
Hh=0.03095*HNu/HDh;%%heat transfer coefficient of hot gases
Hm=Hden*HFR;%% mass flow rate
LHV=18*10^6;%% low heating value
CInT=30;%%average temperture value of inlet cold gases
COutT=60;%%desire outlet temperature of air
Cden=1.109;%%average density value of cold gases
CCp=1007;%%average cp value of cold gases
Clength=0.0254*1.5;%%height of cross section area of  cold gases inlet
Cwidth=0.0254*2;%%width of cross section area of  cold gases inlet
CFR=60*0.0004719474;%%flow rate of cold  gases
CAc=Clength*Cwidth;%%cross section area of cold gases side
CDh=2*Clength*Cwidth/(Cwidth+Clength);%%Hydraulic diameter of cold gases side
Cv=CFR/CAc;%%velocity of cold gases
CRe=Cv*CDh/(1.75*(10^-5));%%reynolds's number of cold gases
CNu=0.023*(CRe^0.8)*(0.7241)^0.3;%%Nusselt number of cold gases
Ch=0.03095*CNu/CDh;%%heat transfer coefficient of cold gases
U=Ch*Hh/(Ch+Hh);%%overall heat transfer coefficient
Cdt=COutT-CInT;%% change in temperature
Cm=Cden*CFR;%% mass flow rate
q=Cm*CCp*Cdt;%% heat transfer rate
HOutT=HInT-q/(Hm*HCp); %% out let temperature of flue gases
Dtln=((HInT-COutT)-(HOutT-CInT))/(log((HInT-COutT)/(HOutT-CInT)));%% mean log temperature
As=q/(U*Dtln);%% surface area of duct
L=As/(2*Hwidth*.90+1*Hlength); %% length of duct
n=L/(23*.0254);%% number of fins
 fins=  fix(n);
fins= fins+1;
widthplate=(fins)*Hlength; %% width of centre plate
Q_flue=HFR*HCp*(HInT-30);
fm=(Q_flue/LHV)*1.5*3600;%% flow rate of flue need
totalplatesize=(widthplate+(n+1)*2*Hwidth)/(0.0254*12);
```

APPENDIX 7 – MATLAB CODE FOR PLOTTING GRAPHS OF HEAT EXCHANGER (FIGURE 29 & 30)

MATLAB Code:

```
for temp =1:1:30
   FR=60;
for fr=1:1:30
HCp=1009;%%average cp value of hot gases
Hden=0.9458;%%average density value of hot gases
HInT=(90+temp*10);%%average temperture value of inlet hot gases
Hlength=0.0254*1.5;%%height of cross section area of  hot gases inlet
Hwidth=0.0254*1.5;%%width of cross section area of  hot gases inlet

HFR=FR*0.0004719474;%%flow rate of hot flue gases
x_value_hot(fr)=(FR);
FR=60;
FR=FR+fr;
HAc=Hlength*Hwidth;%%cross section area of hot flue gases side
HDh=2*Hlength*Hwidth/(Hwidth+Hlength);%%Hydraulic diameter of hot gases side
Hv=HFR/HAc;%%velocity of hot flue gases
HRe=Hv*HDh/(2.306*(10^-5));%%reynolds's number of hot gases
HNu=0.023*(HRe^0.8)*(0.711)^0.4;%%Nusselt number of hot gases
Hh=0.03095*HNu/HDh;%%heat transfer coefficient of hot gases
Hm=Hden*HFR;%% mass flow rate
LHV=18*10^6;%% low heating value

CInT=30;%%average temperture value of inlet cold gases
COutT=60;%%desire outlet temperature of air
Cden=1.109;%%average density value of cold gases
CCp=1007;%%average cp value of cold gases
Clength=0.0254*1.5;%%height of cross section area of  cold gases inlet
Cwidth=0.0254*1.5;%%width of cross section area of  cold gases inlet
CFR=60*0.0004719474;%%flow rate of cold  gases
CAc=Clength*Cwidth;%%cross section area of cold gases side
CDh=2*Clength*Cwidth/(Cwidth+Clength);%%Hydraulic diameter of cold gases side
Cv=CFR/CAc;%%velocity of cold gases
CRe=Cv*CDh/(1.75*(10^-5));%%reynolds's number of cold gases
CNu=0.023*(CRe^0.8)*(0.7241)^0.3;%%Nusselt number of cold gases
Ch=0.03095*CNu/CDh;%%heat transfer coefficient of cold gases
U=Ch*Hh/(Ch+Hh);%%overall heat transfer coefficient
Cdt=COutT-CInT;%% change in temperature
Cm=Cden*CFR;%% mass flow rate
q=Cm*CCp*Cdt;%% heat transfer rate
HOutT=HInT-q/(Hm*HCp); %% out let temperature of flue gases
Dtln=((HInT-COutT)-(HOutT-CInT))/(log((HInT-COutT)/(HOutT-CInT)));%% mean log temperature
As=q/(U*Dtln);%% surface area of duct

L=As/(2*Hwidth*.90+1*Hlength); %% length of duct
n=L/(23*.0254);%% nuber of fins
```

```
fins=  n;
widthplate=(fins)*Hlength+fins*0.003; %% width of centre plate
Q_flue=HFR*HCp*(HInT-30);
fm=(Q_flue/LHV)*1.5*3600;%% flow rate of flue need
totalplatesize=(widthplate+(n+1)*2*Hwidth)/(0.0254*12);%%
y_value_hot1(fr,temp)=totalplatesize;
y_value_hot2(fr,temp)=fm;

end
end
for fr=1:1:30
y_value_hot1_m1(fr)=y_value_hot1(fr,1);
y_value_hot2_m1(fr)=y_value_hot2(fr,1);

y_value_hot1_m2(fr)=y_value_hot1(fr,2);
y_value_hot2_m2(fr)=y_value_hot2(fr,2);

y_value_hot1_m3(fr)=y_value_hot1(fr,3);
y_value_hot2_m3(fr)=y_value_hot2(fr,3);

end
plot (x_value_hot,y_value_hot1_m1,'--
r',x_value_hot,y_value_hot1_m2,':k',x_value_hot,y_value_hot1_m3,'-b')%%,...

figure
plot ( x_value_hot,y_value_hot2_m1,'--
r',x_value_hot,y_value_hot2_m2,':k',x_value_hot,y_value_hot2_m3,'-b')
```

APPENDIX 8 – MATLAB CODE FOR CALCULATION OF DRYING TIME

```
%% using Grapes
Mi=  78 ; %% Initial Moisture Content (Wet Basis)
Mf = 9.5 ; %% Final Moisture Content  (Wet Basis)
mi= 1 ; %% Initial Mass of Sample (Kg)
M_drybasis = Mf/(100-Mf);
Mw = mi*(Mi-Mf)/(100-Mf); %% Amount of water to be removed (Kg)
aw = 1- exp(-exp(0.914+0.5639*log(M_drybasis))); %%water activity
ERH = 100*aw ;%% Equilibrium relative humidity
M = (Mi/100)* mi   ; %% initial mass of water (Kg)
md = mi - M  ; %% Bone Dry Mass of Sample (Kg)
cp_product = 3.6*10^3  ; %% specific heat capacity of sample in J/kg
cp_water = 4.186*10^3  ; %%specific heat capacity of water
cp_air = 1020; %%specific heat capacity of air
Tf = 37.513 ; %% Outlet Temperature of Air (Calculated using Psychrometric Chart)
hf = 90.904*10^3 ; %% Entalpy of air at exit of Drying Chamber
hi = 90.074*10^3 ; %% Entalpy of air at inlet of Drying Chamber
T_ambient = 30 ;
ma = 0.028 ; %% Mass flowrate of air (Kg/s)
Ti= 55 ; %% Inlet Temperature of Air
Tpr = 35 ;  %% Product temperature
h_fg = (4.186*10^3)*(597-0.56*(Tpr)) ; %% Latent Heat of Vapporization
Q = md*cp_product*(Tpr-T_ambient) + M*cp_water*(Ti-Tf) + Mw*h_fg ;  %% Amount of Heat
required to reach Final Moisture Content
td = (Q/(ma*(hf-hi)))/3600 ; %% Dring Time (Hrs)
Drying_Rate = Mw/td ;
```