



Jimma University

Jimma University Institute of Technology

FACULTY OF MECHANICAL ENGINEERING

Sustainable Energy Engineering

**Design and performance Evaluation of solar thermal baking system using
Scheffler reflector integrated with basalt rocks as energy storage for
household**

A thesis submitted to the School of Graduate Studies of Jimma University in
partial fulfillment of the requirements for award of Degree of Masters in
Sustainable Energy Engineering.

By; Abduselam Aliyi

Jun, 2024
Ethiopia, Jimma

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Jun, 2024
Jimma, Ethiopia

DECLARATION

I, the under signed, declare that this thesis entitled by “Design and performance Evaluation of solar thermal baking system using Scheffler reflector integrated with basalt rocks as energy storage for household” is my original work, and has not been presented by any other person for an award of a degree in this or any other University, and all sources of materials used for the thesis have been duly acknowledged.

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Abstract

*This study investigates the use of a solar thermal baking system with a Scheffler reflector and basalt rocks as an energy storage medium to meet the increasing energy demands for baking while minimizing environmental impact. The research includes design, manufacture, experimental performance evaluation, and simulation of the system. A Scheffler reflector with 6.8 m² surface area and 1.4 m focal length was designed and manufactured for baking 2.6 kg of dough in one cycle. The installation of a Scheffler reflector in a specific location requires consideration of several factors based on findings from COMSOL simulations. These factors include true north, polar axis, declination angle, and hour angle, and they all have a significant impact on performance. According to the baking capacity requirement, a 95*70*70 cm³ oven working with natural convection was designed and manufactured. The findings from ANSYS simulations have revealed an important aspect to consider when placing bread in the oven. It is not recommended to place the same amount of bread on both the top and bottom trays of the oven, as this can lead to uneven baking due to differences in heat distribution. Furthermore, it is advised to not place both trays in the same direction inside the oven, as this can also result in a lack of uniformity in baking.*

The highest temperature recorded on the receiver surface during the unloaded test was 303°C at 1:15 PM, with an irradiance of 855 W/m². The bread baking test showed that the oven has an overall efficiency of 42% at an average solar irradiance of 838 W/m². Based on the design, using the Scheffler reflector for bread baking, it is estimated to save up to 3.5 tons of CO₂ emissions and 7564 ETB in electricity costs annually. The Scheffler reflector is an environmentally friendly and cost-effective solution that can help to reduce carbon emissions and save money on energy consumption.

Keywords: solar energy, Scheffler reflector, baking, Thermal energy storage

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Table of Contents

DECLARATION	i
Abstract	i
Acknowledgment	ii
List of Figures	vi
List of Tables	x
Nomenclature, Abbreviations and Symbols	x
CHAPTER ONE	13
1. Introduction.....	13
1.1 Background of Study.....	13
1.2 Problem statement.	15
1.3 Objective	16
1.3.1 General Objective	16
1.3.2 Specific Objectives	16
1.4 Scope of the research.....	16
1.5 Significance of the Study	16
CHAPTER TWO	17
2. Literature Review.....	17
2.1 Working of Scheffler reflector:	17
2.2 Working of tracking mechanisms	18
2.3 Design and Construction	19
2.4 Performance Evaluation	21
2.5 Application	23
2.5.1 cooking.....	23
2.5.2 water heating	24

2.5.3	Industrial process heat.....	25
2.6	Research Gaps	28
CHAPTER THREE		29
3.	Research Methodology	29
3.1	Materials.....	29
3.1.1	Materials for construction	29
3.1.2	Instruments for Data collection.....	30
3.1.3	Dough preparation	30
3.2	Methods.....	31
3.2.1	Research Study Area.....	31
3.2.2	Estimation of solar radiation data	31
3.2.3	Research Design.....	31
3.2.4	Estimation of Energy Required for Bread Baking.....	31
3.2.5	Scheffler concentrator design	34
3.2.6	Calculation of cross-bar	47
3.2.7	Design of stand	52
3.2.8	Design of the Tracking system	55
3.2.9	Design of Thermal Energy Storage.....	58
3.2.10	Design of the receiver	60
3.2.11	Design of the oven	61
3.2.12	Ray tracing.....	64
3.3	ANSYS Fluent Simulation of Receiver	70
3.3.1	Prediction of temperature and heat flux values on the receiver surface	70
3.3.2	Prediction of temperature, Velocity and Pressure distribution inside the oven.....	71
3.4	Experimental study.....	74

3.4.1	Determination of Receiver Energy Losses	75
3.5	Environmental Benefits of Solar Bread Baking System	78
CHAPTER FOUR.....		79
4.	Result and Discussion	79
4.1	Result of Ray trace Simulation for the optical performance of Scheffler reflectors.....	79
4.1.1	Reflector at equinox at noon at 0^0 polar axis	79
4.1.2	Reflector at equinox at noon at 7.6^0 polar axis	80
4.1.3	The sun's position at summer solstice, reflector at equinox without adjusting and reflector at summer solstice with adjusting declination angle at noon at 7.6^0	80
4.1.4	The sun's position at the Winter solstice and reflector at the equinox without adjusting and reflector at winter solstice with adjusting declination angle at noon at 7.6^0 ..	81
4.1.5	Reflector rotates on polar axis with adjusting hour angle at -60^0 (8:47 AM) and without adjusting hour angle at 7.6^0 polar axis.	85
4.1.6	Reflector rotates on the polar axis without adjusting the hour angle at -15^0 (11:47 AM) at 7.6^0 polar axis.	86
4.1.7	Reflector rotates on 0^0 polar axis but installed in 7.6^0 location at equinox with adjusting hour angle at -60^0 (8:47 AM) at 7.6^0 polar axis.....	87
4.1.8	Reflector deviates 10^0 from true north to east at equinox at noon at 7.6^0 polar axis	88
4.1.9	Reflector deviate 10^0 from true north at 11:45 am at 7.6^0 polar axis.....	88
4.1.10	Reflector deviate 10^0 from true north at 1:45 pm at 7.6^0 polar axis.....	89
4.2	Result of ANSYS Fluent Simulation	90
4.2.1	Prediction of temperature and heat flux values on the receiver surface	90
4.2.2	Prediction of temperature, Velocity, and Pressure distribution inside the oven.....	91
4.3	Experimental Results.....	100
4.3.1	Temperature distribution inside the oven without load and without storage.....	100

4.3.2	Temperature distribution inside the oven without load and with storage	101
4.3.3	Temperature distribution inside the oven with load and with storage	102
	Conclusion	109
	Recommendations.....	109
	Reference	110
	Appendix.....	114

List of Figures

Fig. 1.	Scheffler Reflector [15].	18
Fig. 2.	Tracking system.....	19
Fig. 3.	Schematic and solid model of Scheffler reflector for design charts.....	20
Fig. 4.	Section of Scheffler reflector in a paraboloid.....	20
Fig. 5.	Scheffler Type Dish Collector and Solar Receiver	21
Fig. 6.	A 16 m ² Scheffler technology tested in Ludhiana (Latitude: 30.9°N; Longitude: 75.85°E), India.	22
Fig. 7.	Schematic view of the experimental device with instrumental set-up.	23
Fig. 8.	Solar bakery unit developed system	24
Fig. 9.	Experimental Setup with instrumentation	25
Fig. 10.	Photographic view of experimental setup	25
Fig. 11.	Steam for dry cleaning and laundry for Gajaraj Cleaners, in Ahmadnagar, Maharashtra, India.	26
Fig. 12.	System component of Scheffler reflector solar sesame oil distiller	26
Fig. 13.	Training of the technical staff to solve any upcoming problems	27
Fig. 14.	Solar steam generators with multistage evaporators	27
Fig. 15.	Digital multimeter (A), K-type thermocouple probe(B), Solar irradiance meter (C).....	30
Fig. 16.	Section of Scheffler reflector in a paraboloid [29]	35
Fig. 17.	Effect of the seasonal change in the aperture area[42].	37
Fig. 18.	Basic parabola with vertex at the origin and f 140cm.	39
Fig. 19.	The parabola is revolved around the y-axis to form a paraboloid of revolution	40

Fig. 20. Aperture area and collector area when cutting by cutting circle.	41
Fig. 21. The same aperture area but d/t radius cutting circle centered	42
Fig. 22. Scheffler reflector model formed when cutting by cutting a circle.	43
Fig. 23. Description of the parabola of a Scheffler reflector	44
Fig. 24. A desired parabola cut out from a large parabola at point A and B.	45
Fig. 25. Description of the central bar and depth of crossbars.	46
Fig. 26. During Manufacturing central bar.	47
Fig. 27. Bending Radius of the crossbar.	48
Fig. 28. Intersection points of seven crossbars for the elliptical frame of Scheffler dish.	49
Fig. 29. Radius of bending and length of ellipse	49
Fig. 30. Estimation of arc Length	50
Fig. 31. Manufactured cross bar.	51
Fig. 32. Manufactured Scheffler reflector frame.	52
Fig. 33. Changing of focal point with seasonal change.	53
Fig. 34. End of the lower point of Scheffler at winter solstice	54
Fig. 35. End of the lower point of Scheffler at winter solstice with latitude of location.	54
Fig. 36. Design of stand by considering latitude	55
Fig. 37. Design of the mechanical tracking	56
Fig. 38. Design of tracking channel	57
Fig. 39. Assembled parts and attaching mirror in the Scheffler frame	58
Fig. 40. Design of storage	60
Fig. 41. Design of the receiver	61
Fig. 42. Distance between Trays	62
Fig. 43. Design of the oven	62
Fig. 44. Assembled all parts	63
Fig. 45. Manufactured Scheffler Reflector.	63
Fig. 46. Manufactured Scheffler Reflector	64
Fig. 47. geotimedate.org plat form	67
Fig. 48. Model of the Scheffler reflector	68
Fig. 49. Meshing of Scheffler reflector model	68
Fig. 50. Ray tracing methods for evaluating concentrators flow chart	69

Fig. 51. Model of the Receiver	71
Fig. 52. Meshing of the Receiver model.....	71
Fig. 53. Model of the Oven.....	73
Fig. 54. Meshing of the oven model	74
Fig. 55. Radiation input and heat loss from a receiver	75
Fig. 56. Heat transfer network and Resistance network[7].....	76
Fig. 57. Simple collector network.....	76
Fig. 58. Oven prototype	78
Fig. 59. The reflector at equinox at noon at 0° polar axis	79
Fig. 60. The reflector at equinox at noon at 7.6° polar axis.....	80
Fig. 61. The sun's position at summer solstice (21, June), reflector at equinox without adjusting to declination angle.	81
Fig. 62. The sun's position at the summer solstice (21 June), the reflector adjusted to declination angle (38.35°).	81
Fig. 63. The sun's position at winter solstice (21, December), reflector at equinox without adjusting to declination angle.	82
Fig. 64. The sun's position at the winter solstice (21 December), the reflector adjusted to declination angle (61.8°).	82
Fig. 65. Effects caused by not adjusting the declination angle.....	83
Fig. 66. Solar Declination vs Number of days.....	84
Fig. 67. Telescopic bar.....	84
Fig. 68. The sun's position at equinox (21, march) at -60 hour angle without adjusting hour angle of Scheffler reflector to hour angle (reflector at noon).	85
Fig. 69. The sun's position at equinox (21, march) at -60 hour angle with adjusting hour angle of Scheffler reflector to hour angle -60° (8:47 AM)	85
Fig. 70. The sun's position at the equinox (21, march) at -15° hour angle without adjusting the hour angle (reflector at noon).	86
Fig. 71. Daily tracking	86
Fig. 72. The sun's position at equinox (21, march) at -60 hour angle (8:47 AM), reflector rotates on 0° polar axis but in installed in 7.6° location with adjusting hour angle.	87
Fig. 73. Level measurement tool.....	87

Fig. 74. The sun's position at equinox (21, march) at noon, reflector deviates 10^0 from true north to east.	88
Fig. 75. The sun's position at equinox (21, march) at 11:45 pm (15^0 before noon) reflector deviates 10^0 from true north to east.	88
Fig. 76. The sun's position at equinox (21, march) at 1:45 pm (15^0 after noon) reflector deviates 10^0 from true north to east.	89
Fig. 77. Digital Compass	89
Fig. 78. Temperature Variation at 3500 w/m^2 heat flux.	90
Fig. 79. Temperature Variation at 2500 w/m^2 heat flux.	91
Fig. 80. Temperature and Velocity Distribution inside the oven when both trays are closed to the left	92
Fig. 81. Temperature and Velocity Distribution inside the oven when both trays are closed to the right	92
Fig. 82. Temperature and velocity Distribution inside the oven when top tray closed air flow...	93
Fig. 83. Temperature, velocity, and Pressure distribution inside the oven when top tray is closed to left and bottom tray is closed to the right	93
Fig. 84. Temperature and Velocity Distribution inside the oven when both trays are closed to the left	96
Fig. 85. Temperature and Velocity Distribution inside the oven when both Trays are closed to the right	96
Fig. 86. Temperature and velocity Distribution inside the oven when top tray closed airflow	97
Fig. 87. Temperature, velocity, and Pressure distribution inside the oven when top tray is closed to the left and bottom tray is closed the right.....	97
Fig. 88. Variation of temperature with time for no-load and no-storage test for 6 February 2024	100
Fig. 89. Variation of temperature with time for no-load and storage test for 7 February 2024.	101
Fig. 90. Variation of temperature with time for load and storage test for 8 February 2024	102
Fig. 91. Efficiency and moisture losses	102
Fig. 92. Moisture losses (a), Temperature inside the Bread (b).....	103
Fig. 93. Variation of Receiver temperature and Heat losses.....	105

Fig. 94. Comparison experimental (a) and simulation result (b) of Temperature Distribution inside the oven.....	107
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List of Tables

Table 1.Materials used for construction.....	29
Table 2.Instruments for Data collection.....	30
Table 3. Description of study areas.....	31
Table 4. Power for heating.....	33
Table 5. Points of parabola with vertex at the origin and f 140 cm.	39
Table 6. Key points on parabola with $mp = 0.001786$	45
Table 7.Estimation of arc Length.....	51

Nomenclature, Abbreviations and Symbols

Nomenclature

k	thermal conductivity (W/m.K)
h	heat transfer coefficient (W/(m ² .K))
C _p	specific heat capacity (J/kg.K)
h_{fg}	Latent Heat of Vaporization of Water (J/kg)
Q	heat quantity (J)
P	power

m	mass (kg)
V	volume (m^3)
T	temperature (K)
Ac	area of collector
Ar	area of receiver
A_p	area of aperture
r	radius
DNI	Direct Normal Irradiance
EES	Engineering Equation Solver

Greek Symbols

ε	Emissivity
α	Absorptivity
τ	Transitivity
η	Efficiency
π	Pi
μ	Dynamic Viscosity
ν	Kinematic Viscosity
σ	Steven Boltzmann Constant

Units

$^{\circ}\text{C}$	Degree Celsius
h	Hour
J	Joule
K	Kelvin
K_g	Kilogram
KJ	Kilo Joule
KWh	Kilo Watt Hour
m	Meter
min	Minute
cm	Centimeter

<i>MWh</i>	Mega Watt Hour
<i>Pa</i>	Pascal
<i>s</i>	Second
<i>W</i>	Watt

CHAPTER ONE

1. Introduction

1.1 Background of Study

Baking is a method of cooking using dry heat and is an important way of preparing various food items such as breads, biscuits, cakes, pastries, pies, cookies, and crackers[1]. Bread is a staple food that is closely related to people's daily lives and is a significant source of carbohydrates in the food pyramid. Baking plays a major role in bread making and accounts for about 75% of the total energy input [2]. The concern for energy and food is particularly relevant in developing countries, as the food production chain consumes a substantial amount of energy [3]. Improving energy generation, estimation, and conservation is crucial for the development of industries and sustainable food and energy practices. To address environmental issues, it is important to develop new eco-friendly heating, baking, and cooking technologies. The baking industry has the chance to decrease its reliance on primary energy sources and reduce the energy cost per production unit by utilizing solar thermal energy. Various solar thermal collectors are used globally to produce low to medium temperature solar thermal energy [4]. These collectors absorb solar energy as heat and efficiently transfer it to a heat transport fluid, which can then be used for other applications.[5].

There are primarily two kinds of solar collectors: non-concentrating and concentrating types. In the non-concentrating type, the solar absorbing surface is flat and does not concentrate incoming solar radiation [6]. It can absorb both direct and diffuse radiations from the sun, does not require tracking systems, and is mechanically strong. It also has lower maintenance costs but cannot achieve high temperatures. Examples of non-concentrating type solar collectors include flat plate solar collectors and flat plate solar air heaters. On the other hand, concentrating type solar collectors use reflectors or optical means to concentrate solar radiation onto a small absorbing area, allowing them to attain higher temperatures due to the concentration of radiation. They generally collect direct solar radiation and minimal diffuse radiation[7]. The concentrating type of solar collectors includes various designs such as parabolic trough collector, compound parabolic collector, mirror strip reflector, Fresnel lens collector, paraboloid dish type collector, and Scheffler reflector. The Scheffler reflector, conceptualized by German scientist Wolfgang Scheffler, is a commercial product designed to harness solar energy. The basic idea that lead to the development of the Scheffler Reflectors was to make solar cooking as comfortable as possible.

To make cooking simple and comfortable the cooking place should not have to be moved, even better it should be inside the house and the concentrating reflector outside in the sun. It is a small section of a paraboloid that concentrates the sun's radiation over a fixed focus by tracking the movement of the sun. This unique design, with an inclined cut, produces the typical elliptical shape of the Scheffler reflector [8]. If the reflector were a rigid paraboloid, the focus would move as the dish turns. To prevent this, the reflector is designed to be flexible and bends as it rotates in order to keep the focus in one place. Ideally, the reflector should maintain a perfect paraboloid shape at all times. The concentrated light produces heat that can be used for various purposes such as heating, generating steam, cooking, baking bread, and heating water. The effectiveness of a solar concentrator relies heavily on its focal alignment, so in stationary systems, there can be a significant loss in energy availability [9]. Ray tracing software is a valuable tool that enables users to estimate the quantity and distribution of focused solar energy that a receiver can transmit at any given time, while also defining the geometry and construction materials. Comsol, a new optical modeling software, is currently being utilized for modeling solar power optical systems and analyzing their performance.

The integration of a thermal energy storage (TES) system that includes sensible, latent heat, and thermochemical energy storage is required to increase the performance of solar cooking systems, due to the intermittency of solar energy input. Sensible and latent thermal energy storage has become increasingly important in energy management, for effective use and reuse of waste heat and solar energy in manufacturing, buildings, and cooking [10]. Sensible thermal storage materials are the most commonly used for low and medium temperature applications, such as water, organic oils, rocks, ceramics, and molten salts. Based on cost, sensible thermal storage materials are cheaper and more readily available for people in rural and some urban areas of developing countries.[11].

A thermal energy storage system, using a packed bed of rocks with air as a heat transfer fluid, has emerged as a promising and cost-effective alternative for storage applications in concentrated solar power. Basalt rocks have been identified as suitable filler material for energy storage applications in high temperature concentrated solar power plants (ranging from 20 to 700 °C). These rocks are widely available at a low cost and have a low environmental impact. Additionally, they exhibit high thermo-physical properties and good thermal stability in this temperature range.

1.2 Problem statement.

Our country primarily relies on biomass, fossil fuels, and electricity for bread baking. This has a significant impact on environmental sustainability and economic viability. The use of biomass, especially wood logs, contributes to deforestation and worsens global warming. Power fluctuations and failures also affect the quality of bread, leading to undesirable outcomes for both bakers and consumers. Additionally, the increasing costs of electricity can inflate operational expenses and reduce profit margins for bakeries and food production facilities, making food products less accessible for consumers. These circumstances may necessitate price adjustments for baked goods, potentially influencing consumer purchasing behavior and overall market demand. To address the rising cost and environmental impact of conventional fuels, it is crucial to develop eco-friendly baking systems. The implementation and adoption of renewable energy sources, such as solar energy, can help mitigate environmental impact and reduce energy costs. Balancing economic considerations and environmental sustainability will be critical in addressing these challenges and ensuring a more sustainable future for bread production. Since bread production requires medium temperatures, this is achieved by concentrating solar energy. Concentrating Solar Thermal (CST) systems rely on precise alignment between the collector and the sun to achieve high energy concentration and efficient heat transfer. However, maintaining this optimal alignment poses a significant challenge due to the dynamic nature of the Sun-Earth relationship. The Earth's rotation and revolution around the sun create a constantly shifting solar position in the sky. This necessitates continuous and precise tracking of the sun's trajectory throughout the day and year to maintain optimal concentration at the receiver. Inaccurate tracking results in reduced energy collection, lower operating temperatures, and diminished system efficiency. Therefore, the aim of this study is to design and develop a solar thermal baking system using a Scheffler reflector integrated with basalt rocks as energy storage. This system can be a viable solution to the challenges associated with non-renewable energy resources and the rising cost of conventional energy sources.

1.3 Objective

1.3.1 General Objective

- The main objective of this study is to design, develop, carry out simulation and experimental testing of a solar thermal bread baking system integrated with basalt rock as energy storage by using a Scheffler reflector system.

1.3.2 Specific Objectives

- Integrate basalt rock as an energy storage mechanism
- Carry out simulations of the integrated system
- Conduct experimental testing of the bread baking system
- Compare experimental data with simulation results

1.4 Scope of the research

This study mainly focuses on detailed design and selection of appropriate solar collector type (Scheffler Reflector), suitable thermal energy storage, mechanical tracking mechanism, best fit heat transfer fluid, determining heat loss mechanisms associated in the absorber section, fabrication of prototype and the testing of bread baking system numerically and experimentally.

1.5 Significance of the Study

Bread baking is a profitable business for private and small-scale enterprises in Ethiopia. They sell their bread products to various markets, including households, hotels, cafeterias, universities, wholesalers, and others where bread is in demand. It is important to note that the solar thermal Scheffler reflectors bread baking system has the potential to be a promising and commercially viable alternative to non-renewable energy sources.

The main significances of this thesis work were:

- Minimize energy cost as solar energy is freely available.
- Creates viable option for fossil fuel dependent bread bakeries.
- Zero GHG emission thus, environmentally friendly.
- Used in both indoor and outdoor baking.
- Reduce deforestation because, large amount of biomass is used for traditional bakeries.
- Health benefits, since while using biomass-based bakeries there is huge amount of smoke which is harmful to human health.
- Due to the rise of electric energy cost tariff, using solar thermal energy for bread baking process is best options.
- Best opportunity for entrepreneurs, privates and small-scale enterprises.

CHAPTER TWO

2. Literature Review

One of the primary household tasks that require energy for human survival is cooking. Across the world, people use various types of cooking fuels such as fossil fuels, firewood, and charcoal to source energy for cooking. Due to the rising cost of cooking gas and fossil fuels, as well as the unreliable power supply in developing countries, the majority of people in rural and urban areas still rely on firewood, charcoal, and residue for cooking.[12]. One possible reason for deforestation and the emission of greenhouse gases is the high consumption of fuel wood. As a result, there is a growing interest in harnessing the abundant potential of the clean, safe, and readily available renewable energy source - the Sun - for cooking. Solar cookers are generally divided into two categories: box-type and concentrating type. The box-type allows solar radiation to pass directly through the glass window for cooking, while the concentrating type uses parabolic or spherical, linear Fresnel reflectors (LFRs are solar thermal collectors that use long, narrow mirrors to focus sunlight onto a receiver tube and can achieve a concentration ratio 10-40), Scheffler reflector and a central receiver tower to focus the Sun's rays onto the cooking pot at the focal point [13] [7]. Scheffler reflectors are lateral section of parabolic dish solar concentrators that have gained significant attention due to their potential for harnessing solar energy for various applications. This literature review aims to provide a comprehensive overview of the current state of knowledge about Scheffler reflectors, including their design, performance, and applications.

2.1 Working of Scheffler reflector:

An axis that parallels earth by a mechanical system turns to 15 degrees per day to make up for earth rotation and utilizes a big ideally paraboloid reflector. The axis moves through the middle of mass reflector so that the reflector is easy to turn. The receiver is placed on the rotary axis so that the mirror focuses on it all day long. Therefore, the reflector would not stay stationary when the reflector is tilted, if it is a rigid paraboloid. The reflector shape must differ to maintain the focus fixed. The length and another parameter are different as the center of the paraboloid is changed. The Scheffler reflector is therefore elastic and not sensitive to its form. It is often made up of many glass mirrors combined by flexible plastic. A reflector support structure involves a system for tilting and bending the reflector properly. The mirror is never precisely paraboloid but is always sufficiently near to radius [14].

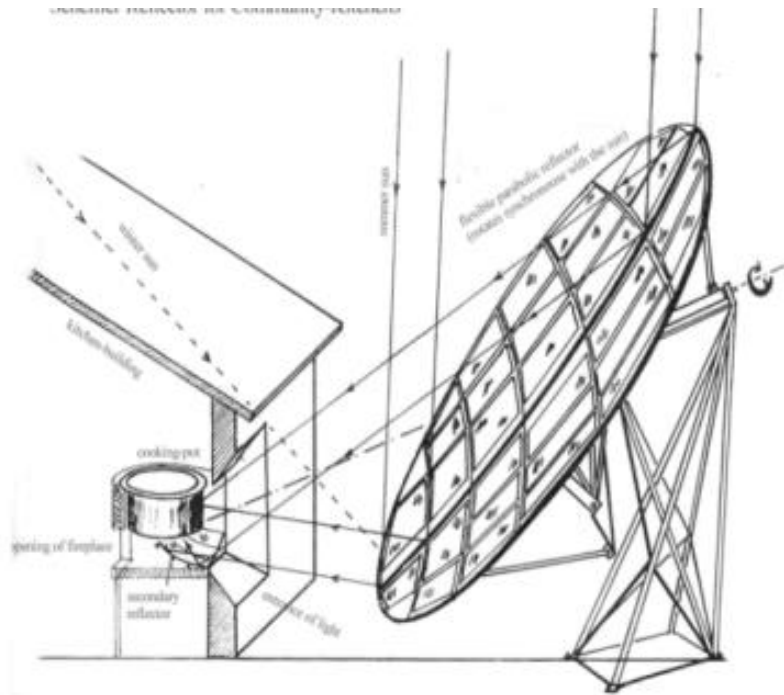
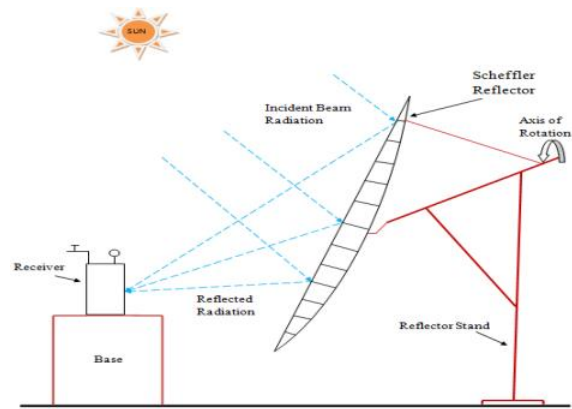


Fig. 1.Scheffler Reflector [15].

2.2 Working of tracking mechanisms

Tracking mechanism is used to track the direction of sun on the concentrator. There are two type Tracking mechanism is used to focus the sunlight onto the collector. On the back side there is the rotational axis, the tilt of this axis is very important. It has to be equal to that place's latitude so that's parallel to earth axis. The daily movement of the reflector goes around this axis. During the day clockwork makes it slowly drift following the sun. The force for rotating the reflector is given by a weight which is applied to the axis of rotation. The weight is pulled up manually when the reflector is rotated into its morning position. Now the weight wants to fall down with the effect of rotating the reflector into its evening position very fast. In order to keep the focus in a fixed position the rotational movement of the reflector has to be slowed down the weight has to fall with the same speed the sun travels across the sky. A mechanical

clockwork can be used as a breaking system. The pendulum will give way to the fall of the weight at intervals (period of the pendulum) that allow the reflector to rotate simultaneously with the sun. For the seasonal movement because during the year sun moves south and then back to north the whole thing has to move. Originally this was made with two telescopic bars one in the lower end and other one in the upper end those has to be adjusted independently both manually. System can be moved on a single axis. Scheffler should be moved about Centre of gravity to assure minimum power requirement.

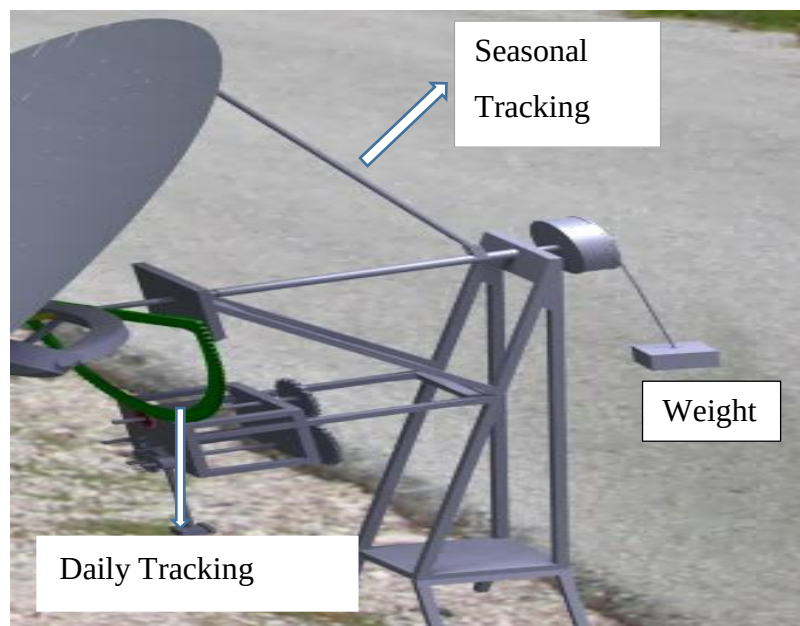


Fig. 2. Tracking system

2.3 Design and Construction

Parimal S.et. Explained the design procedure for a 16 m² standing type Scheffler concentrator along with its dome-type receiver. Depends on the declination angle variation the aperture area calculated for the Scheffler reflector varies from 9.23 m² to 13.7 m². Based on the image analysis at the receiver of the and thermal loss analysis, the diameter and the volume capacity for the stainless-steel receiver are calculated as 0.5 m and 15 liters respectively.

Desireddy Reddy et. al. presented parametric design charts for the Scheffler reflector, which has distinctive advantages such as flexible surface curvature, fixed focal area and shadow less concentration over other solar concentrators. According to authors aperture area and focal length are input parameters required for development of design charts. For working design of the Scheffler reflector, inclination angle can be varied in a very small window of 42–44.9°.

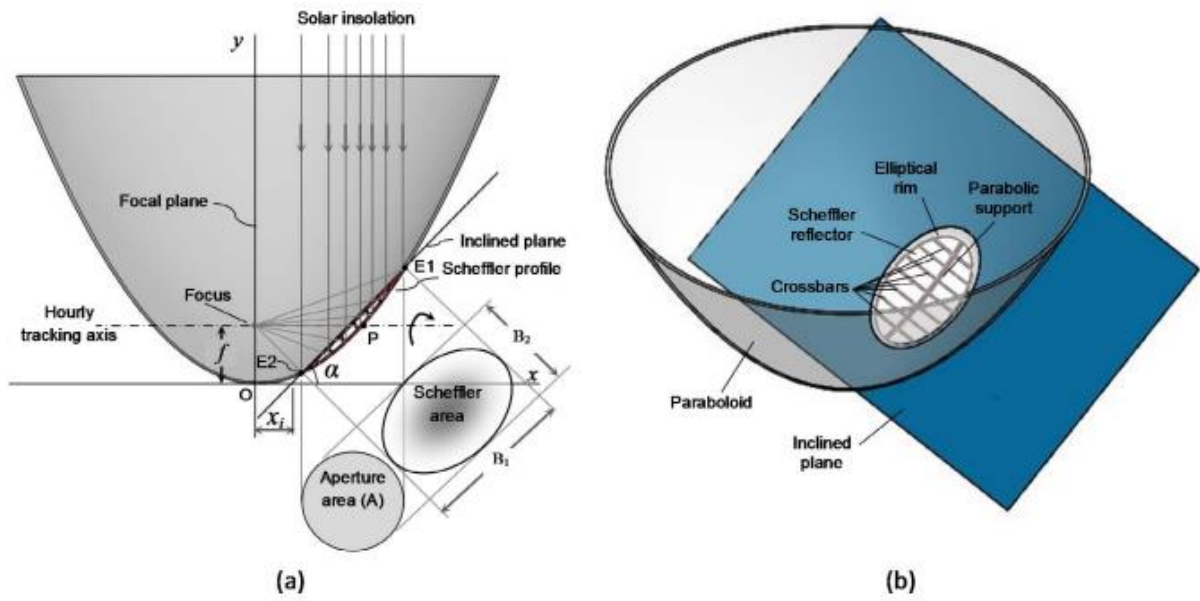


Fig. 3. Schematic and solid model of Scheffler reflector for design charts.

Munir et al. described about the design principle and construction details of an 8 m² surface area Scheffler concentrator. The mathematical calculations to design the reflector parabola curve and reflector elliptical frame with respect to equinox by selecting a specific lateral part of a paraboloid are tabulated. Crossbar equations and their ellipses, arc lengths and their radii are also calculated to form the required lateral section of the paraboloid.

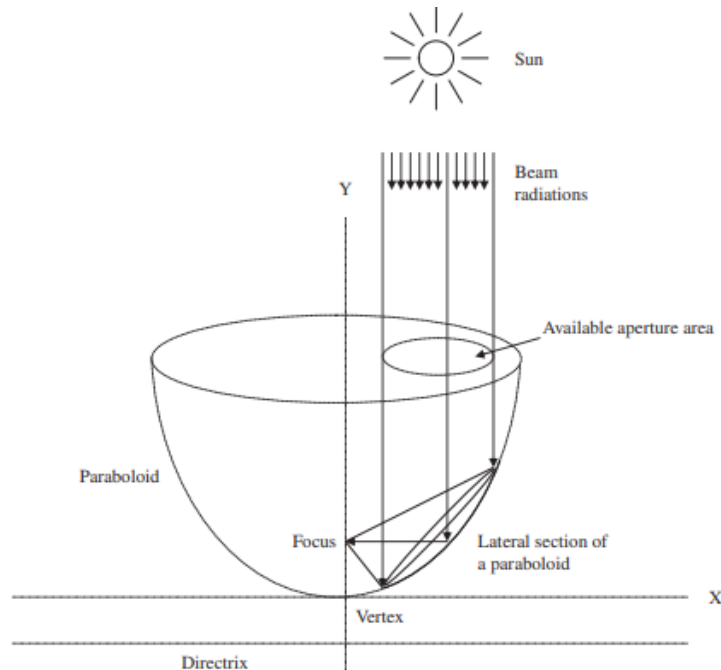


Fig. 4. Section of Scheffler reflector in a paraboloid

2.4 Performance Evaluation

Several experimental studies have investigated on the performance evaluation of Scheffler reflectors.

Ramalingam Senthil, et al. studied the effect of various parameters on the thermal performance and heat losses from the concentrated solar receiver. Increase in wind velocity increases the forced convection heat losses and the increase in the ambient temperature reduces the thermal output of the system were investigated. The beam solar radiation increased from 600 W/m^2 to 800 W/m^2 produces the receiver surface temperature of $360\text{--}400^\circ\text{C}$. Emissivity of the receiver coating plays a major role in the reduction of the radiation heat losses from the receiver. A reduction of emissivity from 0.9 to 0.2 decreases the radiation heat loss by 75% is recorded [15].

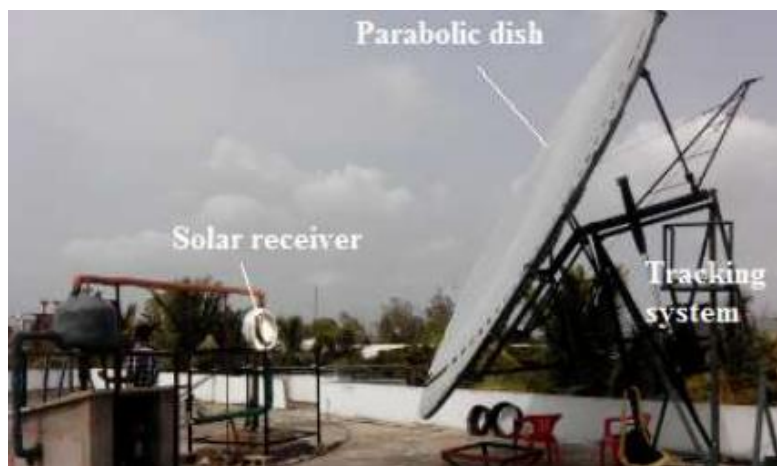


Fig. 5. Scheffler Type Dish Collector and Solar Receiver

S. kumar .T ,et al. investigated 16 square meter Scheffler concentrator system with automatic polar axis tracking. The optical efficiency of the system is achieved up to 59% and the convection heat losses ($\text{W/m}^2\text{K}$), radiation heat losses coefficient ($\text{W/m}^2\text{K}^2$) are 2.2 and 0.001 respectively. The performance of the system was also studied based on various reflectivity (i.e. 84 % and 95 %) of the solar-grade mirror and various selective coating (i.e. Silicon based high-temperature coating and black paint). The performance characteristic of the system is also investigated on a seasonal basis i.e. equinox, winter and summer solstice.

It is concluded that in the near future the Scheffler concentrator system will constitute one of the competitive options of various end use process heat applications from solar energy [16].

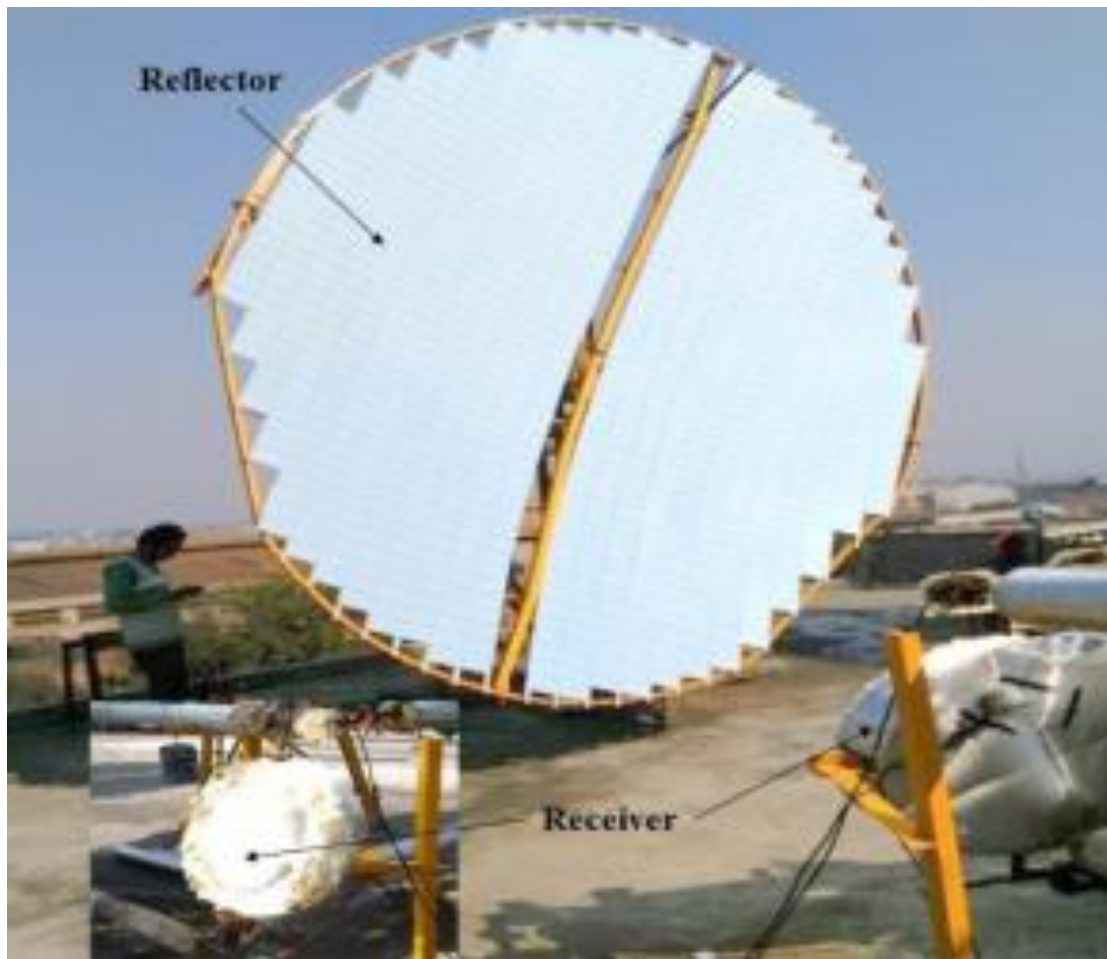


Fig. 6. A 16 m² Scheffler technology tested in Ludhiana (Latitude: 30.9°N; Longitude: 75.85°E), India.

Dnyaneshwar M. et al. Studied performance analysis of an improved receiver for Scheffler solar concentrator experimentally. The tube coil is fixed on the cavity's inner wall and water circulated through the coil to harvest the maximum energy. The average intensity of solar beam radiation was 638 W/m² during the experimentation time of 9:00 to 15:00 h. The stagnation test result estimated that 109 W/m²K overall heat loss coefficient of the receiver. The receiver's energy efficiency with the first, second, and third arrangements was obtained to be 57.71%, 60.93%, and 68.03%, respectively, under the same operating conditions. In all receiver arrangements, the third arrangement has been found attractive for enhancing the parabolic dish concentrator's efficiency[17].

Rupesh. J et. al. Studied the performance of Scheffler reflector. In this system, a single drum of 20 liter capacity was installed at focal point to serve the dual purpose as absorber tube and storage tank. The performance analysis tests have revealed that the average power and efficiency in terms of water boiling to be 1.30 kilowatts and 21.61 % respectively against an

average value of beam radiations of 742 W/m^2 . The maximum water temperature in the storage tank to 98°C was achieved on a clear day operation and ambient temperature between 28°C to 31°C [18].

Anil kumar described the investigation of the four Scheffler reflectors with 16 m^2 surface area each, for the medium temperature applications experimentally. The variation in solar beam radiation from 840 W/m^2 to 1278 W/m^2 over the entire day was observed. $116\text{--}195^\circ\text{C}$ the absorber plate temperature was recorded. 59.28% energy efficiency has been achieved [19].

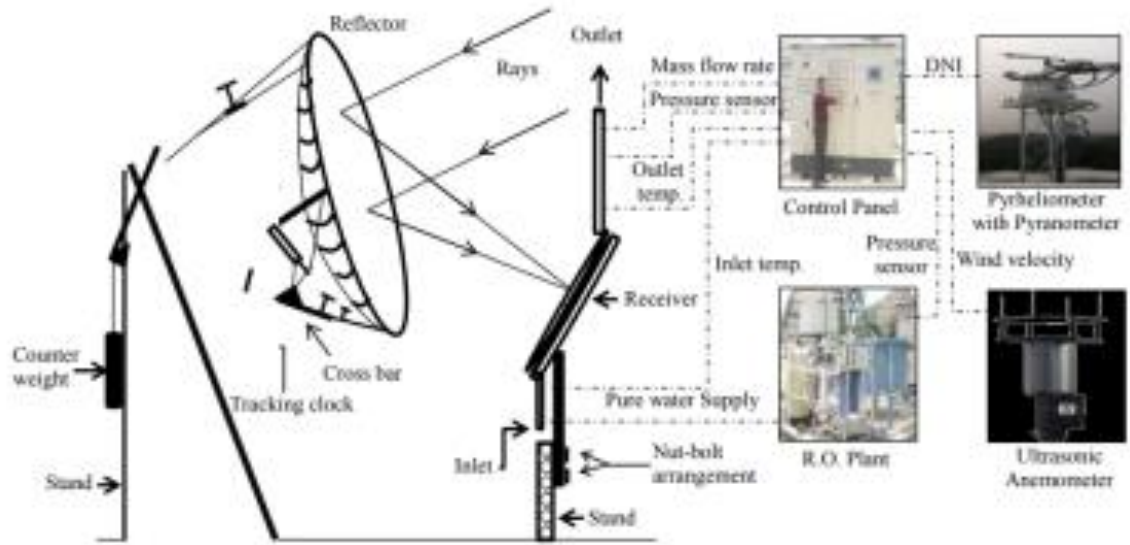


Fig. 7. Schematic view of the experimental device with instrumental set-up.

2.5 Application

Several studies have investigated the application of Scheffler reflectors in various areas, including cooking, oil extraction, water heating, industrial process heat, and power generation.

2.5.1 cooking

Iqra Ayub, et al. developed the Solar Thermal Application for Decentralized Food Baking Using 10 m^2 Scheffler Reflector Technology. Experimental and CFD-based simulation were performed to get appropriate design configurations, which could provide uniform airflow in the baking chamber. 300 and 400°C temperature was obtained at receiver. $200\text{--}230^\circ\text{C}$ temperature at the inlet of baking chamber was obtained.. A series of experiments were conducted for the baking of cakes and total energy available in baking chamber was about 3.29 kW and cake utilized 0.201 kW energy to be baked [20].



Fig. 8. Solar bakery unit developed system

Melkamu Yayeh, Investigated on Scheffler reflector integrated with thermal storage device for indoor Injera baking. The maximum temperature obtained by simulation on the baking pan surface was seen to be about 313°C during charging and 233°C after cooling which can allow Injera baking for the average size of a family in one session of baking[21] .

Christoph Müller, designed and developed five solar community bakeries in the Argentinean Altiplano. They developed the solar bakery oven using a fixed focus 8 m^2 Scheffler concentrator. The oven had a volume of about 200 l with an input power of about 3 kW, and a temperature up to 350°C . As the oven was available for over 6 h each day, up to 60 kg of bread could be baked [22].

2.5.2 water heating

C V Sudhir, et al. investigated potential of 16 m^2 solar Scheffler reflectors for thermal applications experimentally. The pressure and temperature rise in steam header connected to the receivers of the Scheffler reflectors were conducted. 40 kg of distilled water at 29°C of

temperature was used. The temperature at the steam header rise to 140 to 180 °C with beam radiation range of 700 to 850 W/m² on horizontal surface [23] .



Fig. 9.Experimental Setup with instrumentation

Kamboj .M. et al. investigated the coffee making using 1.54 m² Scheffler reflector experimentally.120 °C steam temperature was generated. This steam is directly mixed with coffee in the coffee container. The temperature of 500ml of coffee increased from 18.9°C to 95°C [24].

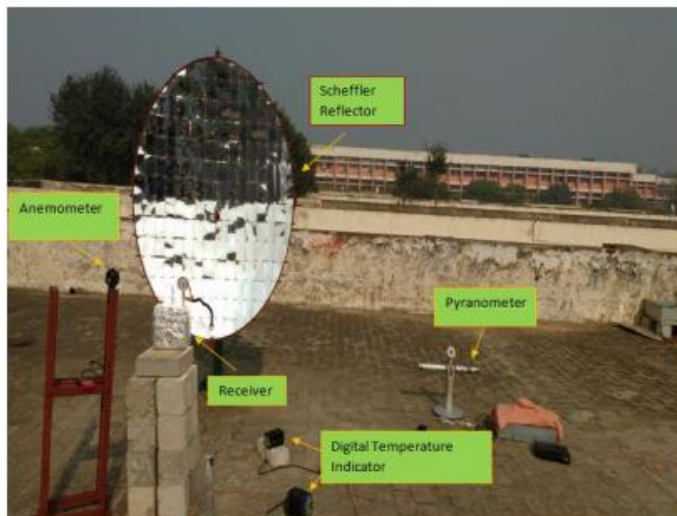


Fig. 10.Photographic view of experimental setup

2.5.3 Industrial process heat

Jayasimha et. Al. Investigated application of Scheffler reflectors for process industry. 16 m² Scheffler reflector were used for commercial laundry, dry cleaning and second application was steam generation for washing compressor parts[25].



Fig. 11. Steam for dry cleaning and laundry for Gajaraj Cleaners, in Ahmadnagar, Maharashtra, India.

Melaku Tarekegne. Designed and evaluate Scheffler reflector for Sesame Seed Oil Extraction. 2.96m² Scheffler reflector system for sesame oil extraction was manufactured. From 2 kg of sesame 1.1 kg of oil was extracted in 2:30 h. The theoretical calculated heat energy and thermal efficiency are 1.55kwh and 42%, respectively [26].



Fig. 12. System component of Scheffler reflector solar sesame oil distiller

Tyroller et. al. Developed solar steam sterilizer for rural hospitals. 10 m² surface area Scheffler Reflector for solar steam sterilizer unit for rural hospitals was used[27].



Fig. 13. Training of the technical staff to solve any upcoming problems

Chandakat al. Designed multistage Evaporators for Integrating with Scheffler Solar Concentrators for Food Processing Applications. Two 16 square meter reflectors were used, and a more significant pressure of 8 bars was obtained from the reflectors [28].

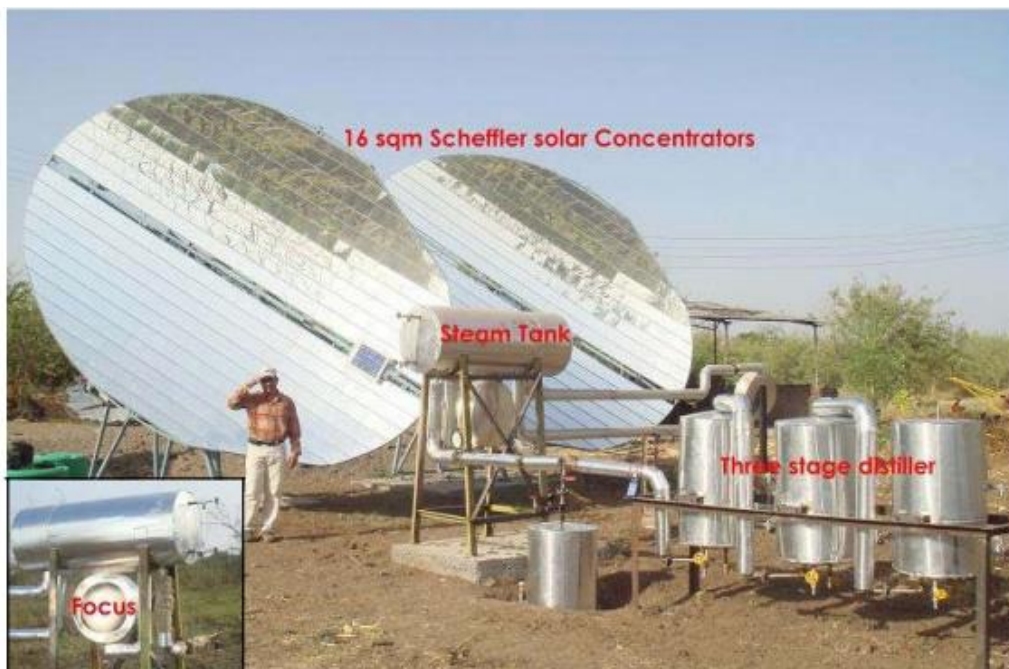


Fig. 14. Solar steam generators with multistage evaporators

2.6 Research Gaps

Scheffler reflectors are a promising solar energy collection technology for areas with an abundant amount of sunlight. Efficient installation methods are crucial for optimal performance and functionality. Existing studies focus on design optimization, thermal efficiency, and cost-effectiveness[29], [30], [31]. However, there is a lack of comprehensive studies that quantify the impact of installation errors on reflector performance. Proper alignment and assembly are crucial for optimal energy capture, but there is a shortage of empirical data on how specific installation errors affect the system's efficiency. Variations in the polar axis, focal length, true north, hourly tracking and seasonal change during installation could potentially lead to inadequate sunlight concentration and reduced thermal energy output. However, there is a limited understanding of the extent to which these deviations impact the overall performance of Scheffler reflectors in real-world settings. Addressing this research gap through systematic Numerical studies that simulate different installation scenarios and measure their impact on reflector performance is crucial for developing evidence-based guidelines and strategies for improving the installation process. Filling these gaps will facilitate wider adoption of Scheffler reflector technology and contribute to the advancement of solar energy systems.

Solar baking is an eco-friendly and sustainable way of preparing food, using natural convection ovens. These ovens rely on the natural flow of air generated by buoyancy to distribute heat within the cooking chamber. However, achieving a uniform temperature distribution inside the oven is still a challenge. While existing studies have focused on the overall performance and efficiency of solar ovens in terms of energy conversion and baking outcomes, there is a lack of detailed analysis on how natural convection affects temperature uniformity and variations within the baking chamber[22].

Specifically, there is a need for research that investigates the impact of factors such as, chimney position, air circulation patterns, storage position and tray positions on temperature distribution during solar baking. Understanding how these variables influence heat transfer and convection currents inside the oven is crucial for optimizing baking conditions and ensuring consistent results across different recipes and food types.

Addressing this research gap through computational modeling can provide valuable insights into the complex dynamics of natural convection in solar baking ovens. Filling these gaps will contribute to the development of more effective oven designs, promoting sustainable and energy-efficient food preparation methods.

CHAPTER THREE

3. Research Methodology

Materials and Method

3.1 Materials

Materials and apparatus used in this experiment for construction and evaluation are:

3.1.1 Materials for construction

The design consideration of solar thermal bread baking system by using Scheffler reflectors is based on its: Cost effective, locally available materials, environment friendly and ease in operation.

Table 1. Materials used for construction

Material Type	specification	Quantity	Used to construct
Mild steel sheet metal	1m*2m*1.5mm	1.2	Oven body (outer)
Galvanized iron	1m*2m*0.8mm	1	oven body (inner)
Round bar	Ø=0.8 cm L=6m	2.4	Cross bar
Square pipe	4cm*3mm L=6m	4	Central bar and stand
Square pipe	3cm*2.5mm L=6m	2	Elliptical frame
Flat iron	4cm*3mm L=6m	1	Tracking channel
Silver coated mirror	25cm*20cm*3mm	110	Reflector
Sprocket	48 teeth	1	Tracking
	10 teeth	1	
Gear	96 teeth	1	Tracking
	6 teeth	1	
chain	Øpitch=1.27 cm	235 chain link	Tracking
Aluminum sheet metal	1m*2m*0.0015	0.725	Receiver and tray
shaft	Ø=3 cm L=1m	1	For daily rotation
Basalt rock	26 kg	26	Thermal storage
Aluminum reinforced fiber glass	1m*2m*10cm	1.2	Insulator
Galvanized iron	1m*2m*0.8mm	2	Supporting mirror

Ordinary window	glass	40cm*40cm*3.8mm	1	Receiver protection
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3.1.2 Instruments for Data collection

Table 2. Instruments for Data collection

Instruments	Reading Capacity	Used to measure
Multimeter mestech MY64	-20 ⁰ C~1000 ⁰ C	Temperature
Solar Irradiance Meter	0~2000 W/m ²	Solar Radiation
K-type thermocouple probe	-200 ⁰ C~1250 ⁰ C	Temperature
Oven dry DHG 9055	10 ⁰ C~300 ⁰ C	Moisture content
Hygrometer	-50 ⁰ C~70 ⁰ C	Ambient Temperature
Stop watch		Time



(A)



(B)



(C)

Fig. 15. Digital multimeter (A), K-type thermocouple probe(B), Solar irradiance meter (C)

3.1.3 Dough preparation

The bread dough was prepared by mixing wheat flour, salt, dry yeast, water and sugar. The dough was mixed manual by finger infringement into mixing bowl or vessel for 10 to15 min until consistent dough character was achieved. Then the dough was placed on flat table, divided into required weight, and kneaded into a ball shape. After the kneading process, the molded dough was transferred and properly arranged on the baking tray then loaded inside oven. The temperature was recorded at an interval of 15 min interval for bread chamber in order to see

effective baking time average temperature required to bake the dough and avoid over cooking which leads to bread burning.

3.2 Methods

3.2.1 Research Study Area

The solar thermal baking system was manufactured in Jimma Agricultural Engineering Research Center (JAERC) workshop, Oromia Agricultural Research institute, Ethiopia.

Table 3. Description of study areas

Place	Jimma Agricultural Engineering Research Center, Oromia Agricultural Research Institute, Ethiopia
Altitude	1730 msl
Geo-reference data	7.6 N latitude and 38.5 E longitude
Ambient temperature	25°C
Average Annual Rain fall	1450-1806 mm per year
Local water boiling test	96 °C
Rainy season	July-September

3.2.2 Estimation of solar radiation data

Solar radiation data was collected from solar irradiance meter instrument.

3.2.3 Research Design

The research design for this study was conducted by using the quantitative research method. The data was collected by doing an experiment in the Jimma Agricultural Engineering Research Center (JAERC) workshop.

3.2.4 Estimation of Energy Required for Bread Baking

Bread baking is indeed a complex thermal process that involves the heating of the dough, boiling of water and the evaporation of moisture[32], [33]. The total energy required for baking bread broadly falls into two main categories the sensible heat (to raise the dough's temperature to a particular temperature and to raise water temperature to boiling point) and the latent heat required for moisture evaporation[34].

The energy used to heat the dough (sensible heat) can be calculated using specific heat capacities and the mass and desired temperature change of the dough and the energy used to boil the water (sensible heat) can be calculated using specific heat capacity of water and the mass and desired temperature change of the water. Sensible heat is essential for achieving the structural and textural characteristics of the bread[35].

For evaporation, the heat required depends mostly on the latent heat necessary to turn the moisture into vapor. This part of the energy is considerable given that a significant amount of

water must be evaporated from the dough during baking. The mass of moisture evaporated is highly dependent on the product and baking process[34].

Thus, to find the energy required in bread baking, the initial mass of dough, and the mass of bread after baking was determined.

The mass of water vapor can be the difference between the mass of the bread product and the initial mass of dough and the sensible heating period of goods or dough (which directly relates to the total dough weight). Therefore, for this study, the oven was designed to meet the following criteria.

- The total mass of dough is 2.6 kg. (for the design requirement)
- The power must keep the oven at a minimum and maximum temperature range of 175 °C & 225 °C respectively.
- The baking process requires 45 minutes for baking,
- Average ambient temperature of a site is 25 °C and
- Average wind speed is 0.6 m/s [36].
- System fluid flow is considered as a laminar flow

moisture content analysis

The weight loss in the food samples (bread dough) was calculated by subtracting the weight of the bread dough sample before baking from the final weight of the food sample after baking (heating). The percentage of moisture loss was obtained using Equation follows;

$$\text{Weightloss} = (\text{Initialweight}) - (\text{Finalweight}) \dots\dots\dots(1)$$

$$\dots\dots\dots(2)$$

The power required to bake the given amount of dough (2.6 kg) is the amount of heat energy used to (need to) turn the lump of dough into a loaf of bread with the required time by evaporating a certain amount of water from a dough. During the baking process, the specific heat capacity (C_p , dough) varies with temperature [37]. Therefore C_p , dough at ambient Temperature was be taken.

$$Q_{\text{baking}} = M_{\text{dough}} * C_{pb} * (T_{br} - T_{amb}) + C_{pw} * M_w * (T_b - T_{amb}) + (M_{\text{dough}} - M_{\text{bread}}) * h_{fg} \dots\dots\dots(3)$$

$$C_{pb} = 0.077T^2 - 68.94T + 16646 \dots\dots\dots(4)$$

Where:

C_{pb} , the specific heating capacity of bread=2.8 kJ/kg°C,

C_{pw} , the specific heating capacity of water=4.18 kJ/kg°C,

T_b , the boiling temperature of water = 96°C,

T_{br} , is the final temperature of the bread=93°C, A bread is considered to be baked when internal temperature reaches 92 °C - 95 °C [38].

M_w , the mass of water

T_{amb} , the ambient temperature = 25°C and

h_{fg} , the latent heat of vaporization of water= 2260 kJ/kg

For this study, assume that the dough 37.5% by mass is water, Then at the end of baking from the total dough moisture 10% will evaporate (for design).

$$M_{bread} = M_{dough} - 10\% M_{water}$$

Where total dough mass (M_{dough}) is 2.6 kg and the mass of bread (M_{bread}) is 2.5 kg

$$Q_{baking} = 2.6 \cdot 2.8 \cdot (93 - 25) + 4.18 \cdot 0.975 \cdot (96 - 25) + 0.1 \cdot 2260$$

$$Q_{baking} = 1002 \text{ kJ}$$

The power needed for 45 minutes

$$P_{baking} = (Q_{baking}) / \Delta t = 1002 \text{ kJ} / (45 \cdot 60) = 0.371 \text{ kW}$$

Initially, the oven space will have air at ambient conditions, preheating the oven usually takes 45 minutes. At this time hot air flows into the oven by buoyancy, replacing the cold air and raising the temperature of the oven until the whole space is at 200°C. The oven materials and dimensions are known and the power required to heat the oven in 45 minutes can be determined. The power required to heat the oven was calculated using equations (5) and (6).

$$Q_{heat} = m \cdot c_p \cdot \Delta T \dots \dots \dots (5)$$

The specific heat of these materials is known, Q is the energy required to heat this material from room temperature 25°C up to 200°C.

This energy is transferred within 45 minutes, and this time change can be used to determine the power input.

Power is given by the equation:

$$P_{heat} = Q_{heat} / \Delta t \dots \dots \dots (6)$$

The power required for heating different components and replacing cold air with hot air is calculated and recorded in Table 1

Power for heating

Table 4. Power for heating

	Volume (m ³)	Density (kg/m ³)	Mass (kg)	Specific heat (kJ/kg °C)	Energy (kJ)	Power (kW)
Air in the oven	0.75*0.5*0.5	1.178	0.1619	1.005	38.96	0.0141
Aluminum Tray	0.4*0.5*0.0015*2	2699	1.62	0.9008	255.3	0.0963
Aluminum Receiver	0.35*3*0.0015	2699	4.373	0.9008	689.4	0.2553
Inside wall (galvanized mild steel)	(0.5*4)*0.0008	7500	12.32	0.47	894.8	0.3314
Total					1878.5	0.657

The minimum required heat energy in the oven to bake 2.6 kg of dough within 45 minutes is the sum of all the above considering heat energy demand and the thermal loss from the oven to the surroundings through the glass cover and the oven door when opening to place/check (assume there is no heat loss through the insulator). Therefore, for this paper, by including the heat loss through the door, glass cover, and the utilized energy in the oven, the overall baking efficiency of the oven (η_{oven}) is assumed at 45%, and the required total power supplied to the oven is determined.

$$P_{required} = P_{baking} + P_{heating} \text{ ----- (7)}$$

$$P_{required} = 0.371 + 0.657 = 1.028 \text{ kW}$$

$$P_{total} \text{ (for design)} = P_{required} / \eta_{oven} \text{ ----- (8)}$$

$$P_{total} = 1.028 \text{ kW} / 0.45 = 2.28 \text{ kW}$$

Thermal efficiency of the oven

Thermal efficiency of the system was calculated by using the ratio of power required for baking to available power from solar energy

$$\eta_{thermal} = \frac{P_{required}}{P_{available}} \text{ (9)}$$

$$P_{available} = \eta_{optical} \text{ DNI} * A_p \text{ (10)}$$

Where, $\eta_{optical}$ optical efficiency (receiver efficiency and reflectivity of mirror)

DNI, direct normal Irradiance (W/m²)

A_p , aperture area

3.2.5 Scheffler concentrator design

The two parameters that are required to begin the design of a Scheffler reflector are

- (a). The aperture area and
- (b). The focal length of the parabola [31].

Aperture area A_p is the projected area of the Scheffler concentrator and focal length f is the distance of focus of the paraboloid from the vertex as shown in Fig.3.

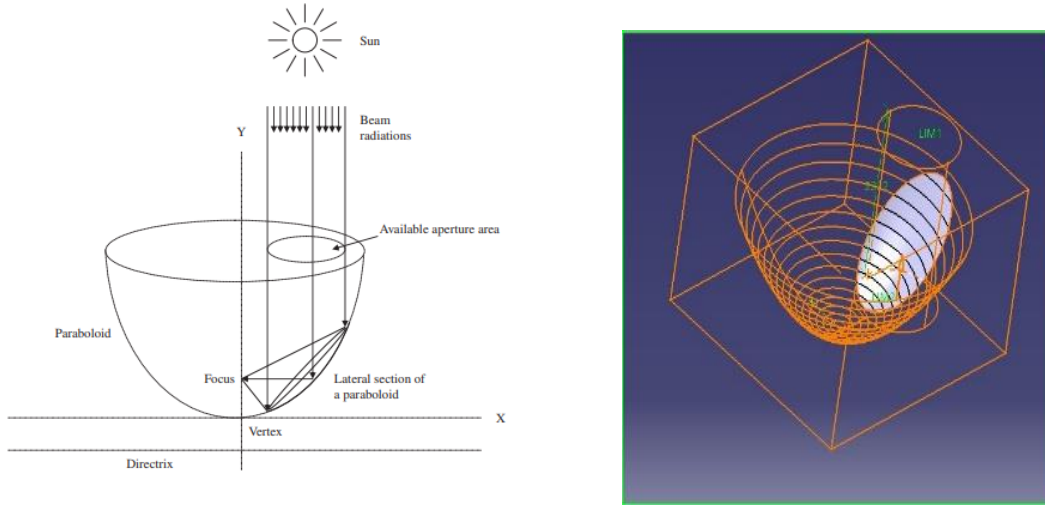


Fig. 16. Section of Scheffler reflector in a paraboloid [29]

Unlike a conventional paraboloid concentrator, the Scheffler fixed focus a concentrator is a lateral part of a paraboloid, as shown in the figure above. While designing a parabola curve for the Scheffler reflector, all calculations are done concerning the equinox (zero solar declination) [32]. But for the construction of the Scheffler concentrator, the latitude of the location must be considered because the location affects the designed focal length as the axis of rotation of the reflector and polar axis are parallel to each other. To calculate the equation of the parabola curve, the calculations are made by considering the side view of the paraboloid. In this way, the paraboloid and reflector frames are drawn in the form of a parabola curve and a straight line, respectively.

Estimation of the aperture area of Scheffler reflectors

The aperture area of a Scheffler reflector is a critical parameter in its design and directly impacts its ability to concentrate solar energy. The aperture area A_p represents the projected area of the Scheffler concentrator. In other words, it's the size of the area through which the solar rays pass before being reflected and concentrated at the focal point.

To estimate the aperture area of a Scheffler reflector, it begins with the energy requirements for intended application.

The area of the aperture area should fulfill the power needed for the oven by considering 45% oven efficiency.

$$P_{\text{solar}} = P_{\text{total}} \text{-----} (11)$$

$$P_{\text{solar}} = P_{\text{available}} = \eta_{\text{optical}} A_p * \text{DNI} \text{-----} (12)$$

Where, η_{optical} = efficiency of the receiver *efficiency of reflective materials

Assume 0.9 for reflectivity [39] and

0.8 for receiver efficiency [40].

DNI= direct normal irradiation assumes 750 W/m² (for design)

From the above formula and assumptions, The Aperture area needed to satisfy the oven power was calculated

The power needed for the oven by considering 0.45% oven efficiency is 2.28kw

$$P_{\text{solar}} = P_{\text{total}} = 2.28 \text{ kw}$$

$$P_{\text{solar}} = \eta_{\text{optical}} * A_p * \text{DNI} = 0.9 * 0.8 * 750 * A_p \text{.....(13)}$$

From above equation

$$A_p = P_{\text{solar}} / (\eta_{\text{optical}} * \text{DNI}) = 2.26 \text{ kw} / (0.9 * 0.8 * 750) = 4.2 \text{ m}^2$$

The aperture area of Scheffler solar concentrator varies with solar declination angle, which can be calculated from the following equation

$$A_p = A_c * \cos(42.47 \pm \delta/2) \text{.....(14)}$$

Where, A_c - collectors area

δ , declination Angle [41]

$$\delta = 23.45 * \sin((360/365) * (284 + N)) \text{(15)}$$

From the above equation, collector's area at the equinox is equal to

$$A_c = A_p / (\cos(42.47 \pm \delta/2)) = 4.2 / 0.7286 = 5.785 \text{ (for design)}$$

From eqn 14 the aperture area at constant collector varies from 3.37 to 5.1 at summer solstice ($N = 172$, Jun 21) and At winter solstice ($N = 354$, Dec 21) respectively.

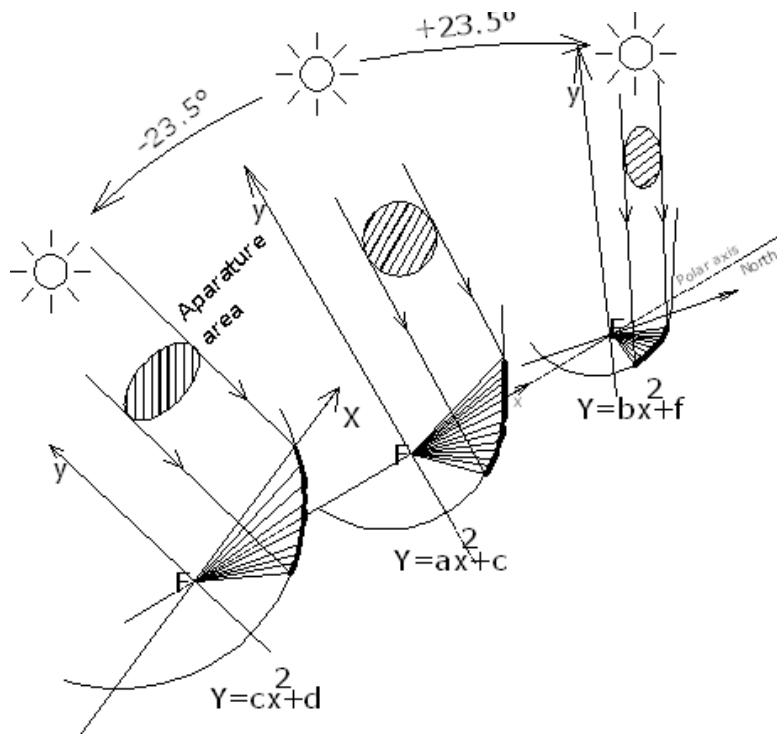


Fig. 17. Effect of the seasonal change in the aperture area[42].

The difference between the Aperture area at the equinox and summer solstice is 0.8m^2

To decrease these effects, we add 0.8m^2 to the actual aperture area

From the above suggestion, the area needed for design is

$$4.2 + 0.8 = 5$$

A 5m^2 aperture area are needed for our design

Simultaneously collector area also increases with increasing aperture area

A new collector area from equation 14, is 6.8m^2

From these aperture areas, we decided our focal length by considering our latitude

$$A_p = \pi * r^2 \text{ -----(16)}$$

Radius needed for a aperture area $r = \sqrt{A_p / \pi}$

$$r = \sqrt{5 / \pi} = 1.25\text{ m}$$

Estimation of the length focal point

The focal point is the location where concentrated sunlight converges after being reflected by reflectors.

The length of x coordinate from the center of the focal axis to the focal point must be greater than the radius of the aperture area to avoid shading that usually exists around the focal point[29].

Let's assume takes 1.4 m (for design purposes). All of our design is based on the focal length. After the focal length is decided, equation of the parabola is formulated.

The equation of a parabola whose axis is parallel to the y-axis having focal length f and its vertex at origin is given by

$$Y = m_p X^2 + c_p \text{-----(17)}$$

Where m_p is the slope of the parabola and C_p is the y-intercept of the parabola.

$$m_p = 1/4 * f \text{-----(18)}$$

$$Y = X^2/4 * f \text{----- (19)}$$

By using Eqn.18 the values of m_p and C_p are calculated as 0.001786 and 0 respectively.

The parabola equation for the equinox is given as:

$$Y = 0.001786 X^2 \text{----- (20)}$$

These equations inserted into EES to get points used for parabolic drawing.

Table 5. Points of parabola with vertex at the origin and f 140 cm.

$\{y=x^2/(4*f)$ The equation of parabola whose axis is parallel to
having focal length f and its vertex at origin}
 $f=140$ {focal length}
 $\{x=280\}$
 $y=x^2/(4*f)$

Parametric Table: Table 1

	x	y	f
Run 1	0	0.00	140
Run 2	20	0.71	140
Run 3	40	2.86	140
Run 4	60	6.43	140
Run 5	80	11.43	140
Run 6	100	17.86	140
Run 7	120	25.71	140
Run 8	140	35.00	140
Run 9	160	45.71	140
Run 10	180	57.86	140
Run 11	200	71.43	140
Run 12	220	86.43	140
Run 13	240	102.86	140
Run 14	260	120.71	140
Run 15	280	140.00	140
Run 16	300	160.71	140
Run 17	320	182.86	140
Run 18	340	206.43	140
Run 19	360	231.43	140
Run 20	380	257.86	140
Run 21	400	285.71	140
Run 22	420	315.00	140
Run 23	440	345.71	140
Run 24	460	377.86	140

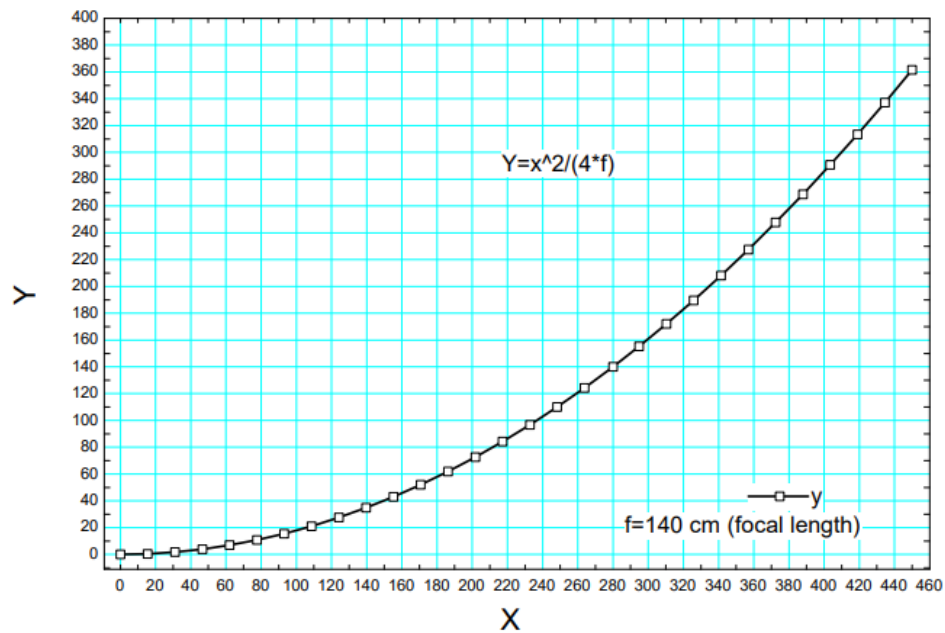


Fig. 18. Basic parabola with vertex at the origin and f 140cm.

The points in EES are input into the Catia software to calculate the central and cross bars for the Scheffler reflector design.

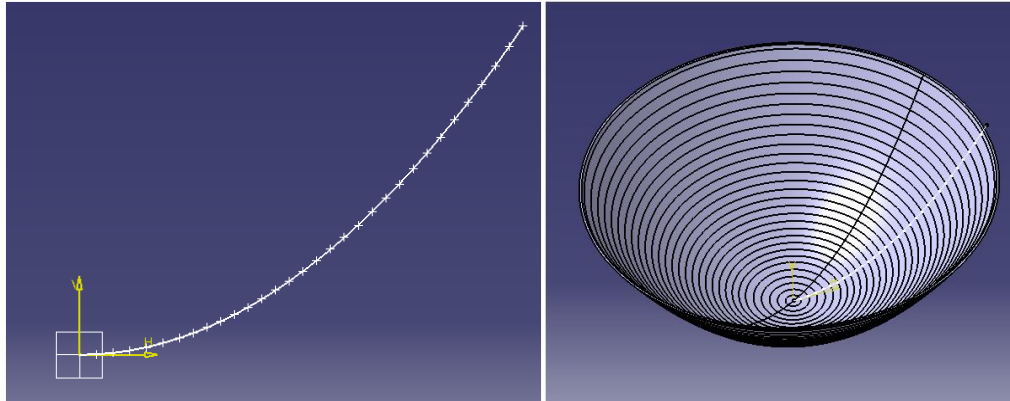


Fig. 19. The parabola is revolved around the y-axis to form a paraboloid of revolution

The Scheffler reflector rotates on its focal axis, the surface area below the center of the focal axis and the surface area above the center of the focal axis must be balanced to minimize the force used for rotating. To get these, the center of the circle used for cutting the elliptical is less than the center of the focal axis. As the surface area above center of the focal axis is greater than that of below center of the focal axis. From the equation, the center of the focal axis is found at $x=280$ and $y=140$ ($x=2*f$). Let's assume the center of the circle for cutting is at $(x=250.5\text{cm}, y=140\text{cm})$ and the radius of the circle is the radius used for aperture area $r = 125\text{cm}$.

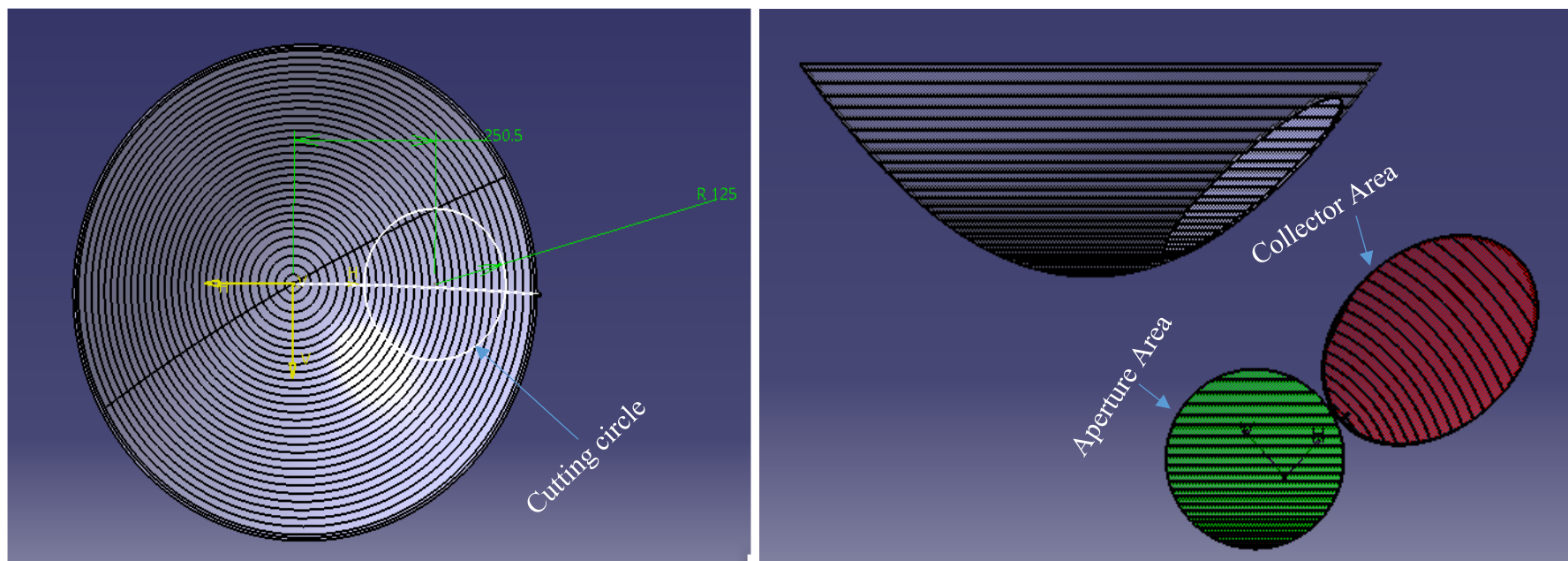


Fig. 20. Aperture area and collector area when cutting by cutting circle.

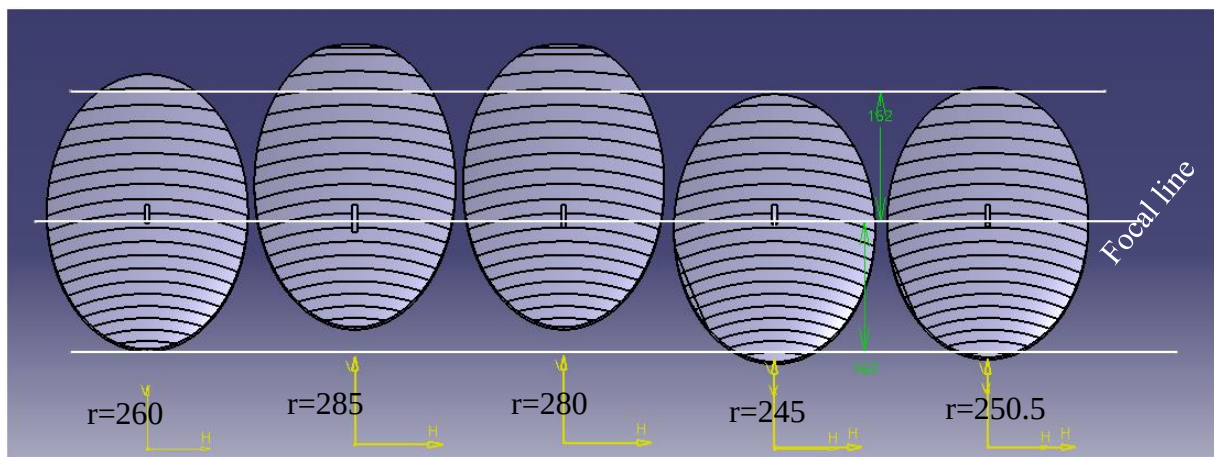
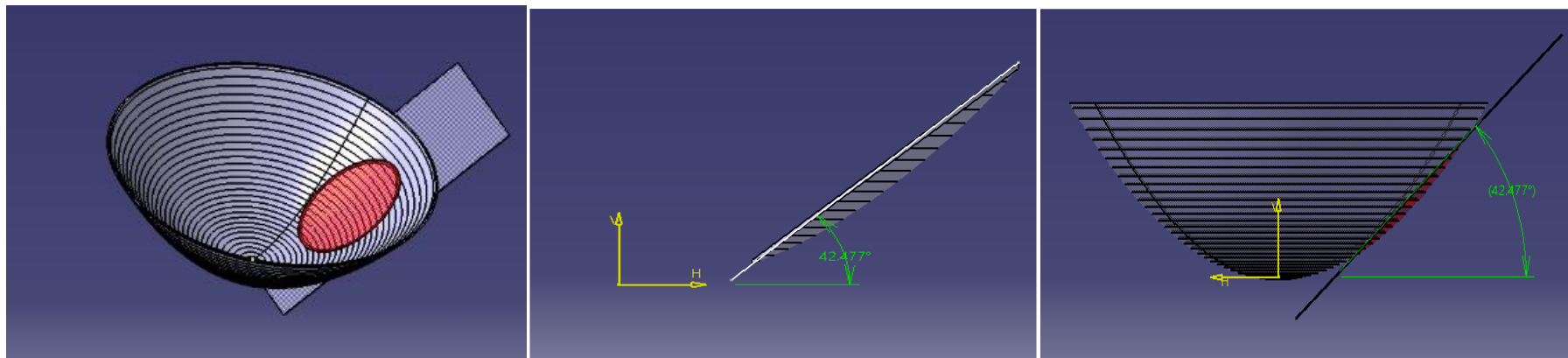


Fig. 21. The same aperture area but d/t radius cutting circle centered

From the above picture, the ellipse cutting at the circle of radius 250.5 cm is best for balancing the surface area above and below the focal line. This is one used for the design of Scheffler reflectors



According to Reddy for the working design of the Scheffler reflector, the inclination angle can be varied in a very small window of 42–44.9° [30].

From these designs 42.477° was obtained the selection of a cutting circle 250.5cm radius is acceptable.

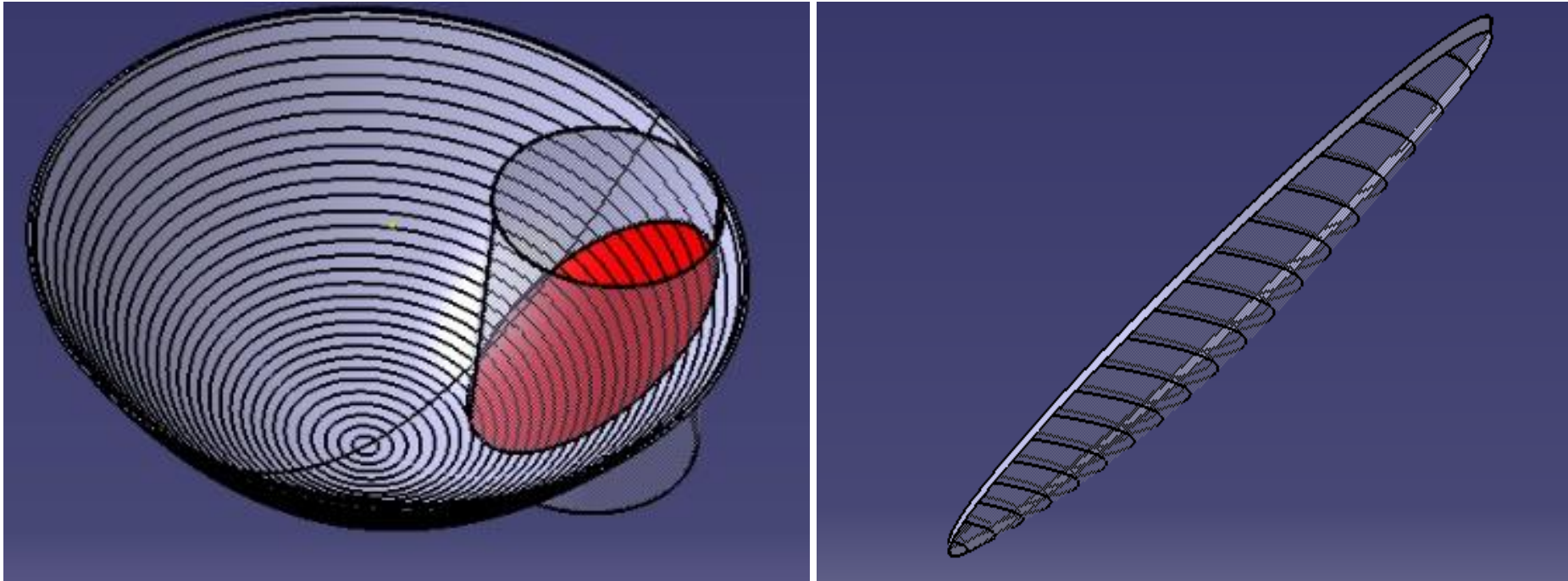


Fig. 22. Scheffler reflector model formed when cutting by cutting a circle.

After the center of the circle used for cutting is estimated, the next step is to cut out the parts used for the design of central bar. For an aperture area of 5m^2 , the point C is defined on the parabola as the point with a slope of 45 degrees. the x-coordinate of point C is taken as 280cm. This coordinate defines the distance of this point to the focal point.

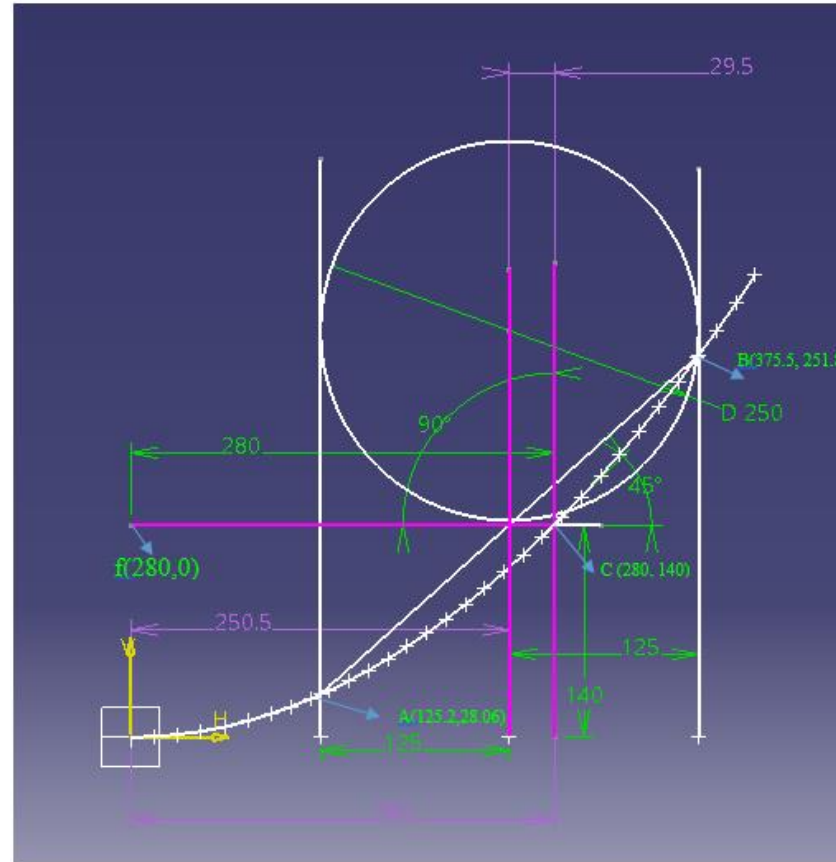


Fig. 23. Description of the parabola of a Scheffler reflector

Table 6. Key points on parabola with $mp = 0.001786$

	X (cm)	Y(cm)
f-focus	0	140
C- mid point	280	140
A Bottom point	125.5	28.06
B Top point	375.5	251.8

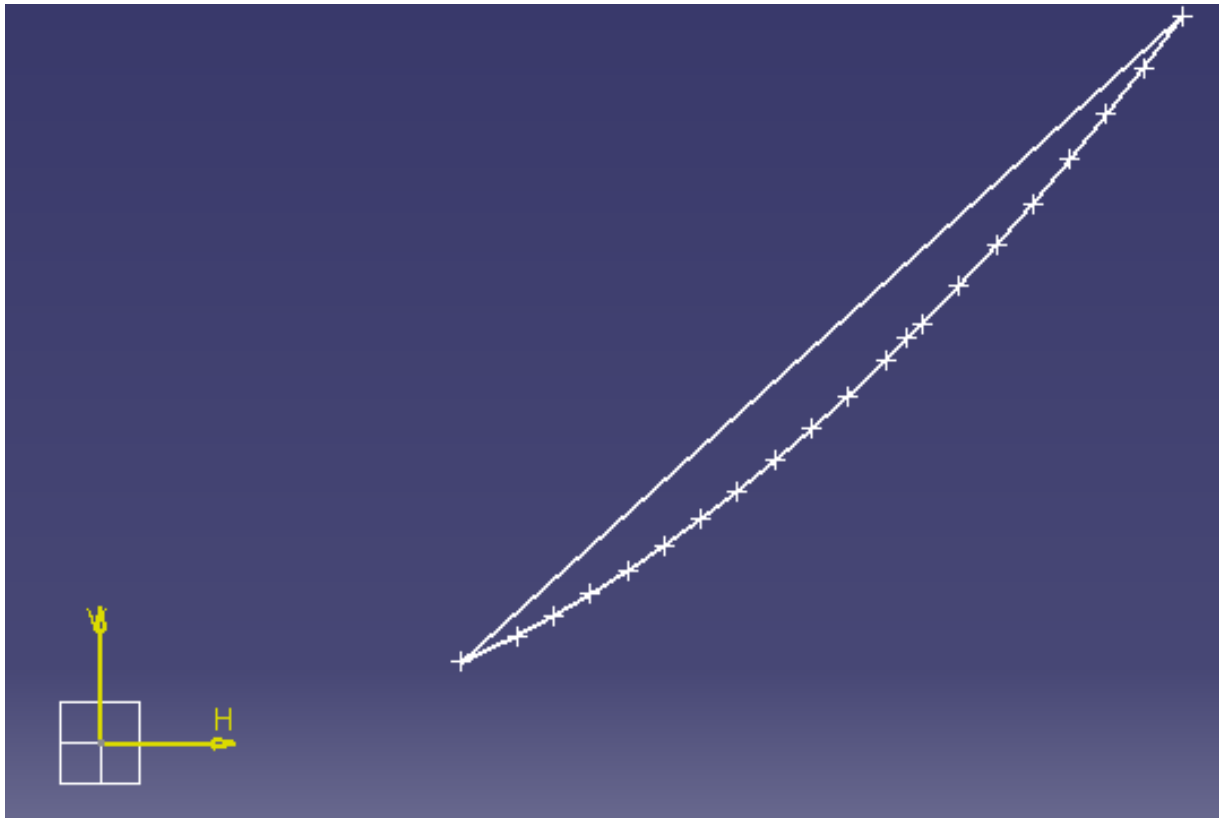


Fig. 24. A desired parabola cut out from a large parabola at point A and B.

After cutting out the desired parabola from a large parabola, the distance between the drawing point and the straight line is measured, by giving 16 cm between points on a straight line. This measurement gives the depth length of the crossbar and the shape of the central bar.

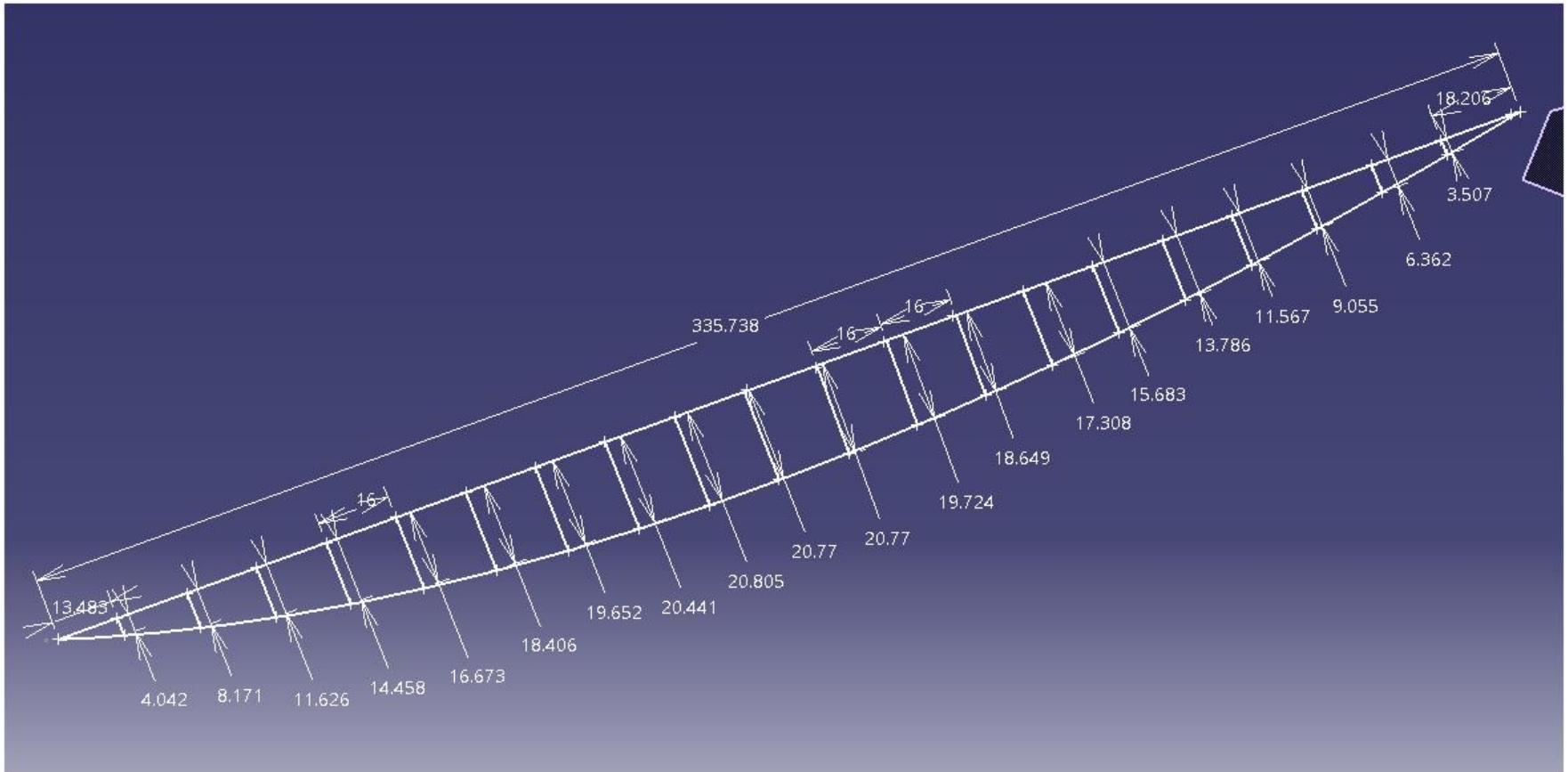


Fig. 25. Description of the central bar and depth of crossbars.

The above-measured dimension is used to construct a central bar on the ground by using a tape meter and try square. Central bar to be made with Mild steel square pipe 4cm*3mm. As the square pipe is not suitable for direct bending, use a nail on the straight log to measure the dip and construct on the log. After these jigs are prepared, cut the square pipe by strip and bend on the prepared jigs. After bending on the jigs, weld the cutting strips properly.



Fig. 26. During Manufacturing central bar.

3.2.6 Calculation of cross-bar

For the construction of the Scheffler reflector, it is necessary to know the exact position of the crossbars on the reflector frame. Several points can be taken but not more than seven to nine crossbars are sufficient to make the required section of the paraboloid for a 5m² Scheffler reflector. Let's assume the distance between one crossbar to the other crossbar is 48cm. For the calculation of the crossbar, the radius for bending must be considered.

It is evident from Fig. 16 that the line perpendicular to the slope line (straight line) to the axis of revolve (y-axis) is considered as the bending radius of the crossbar.

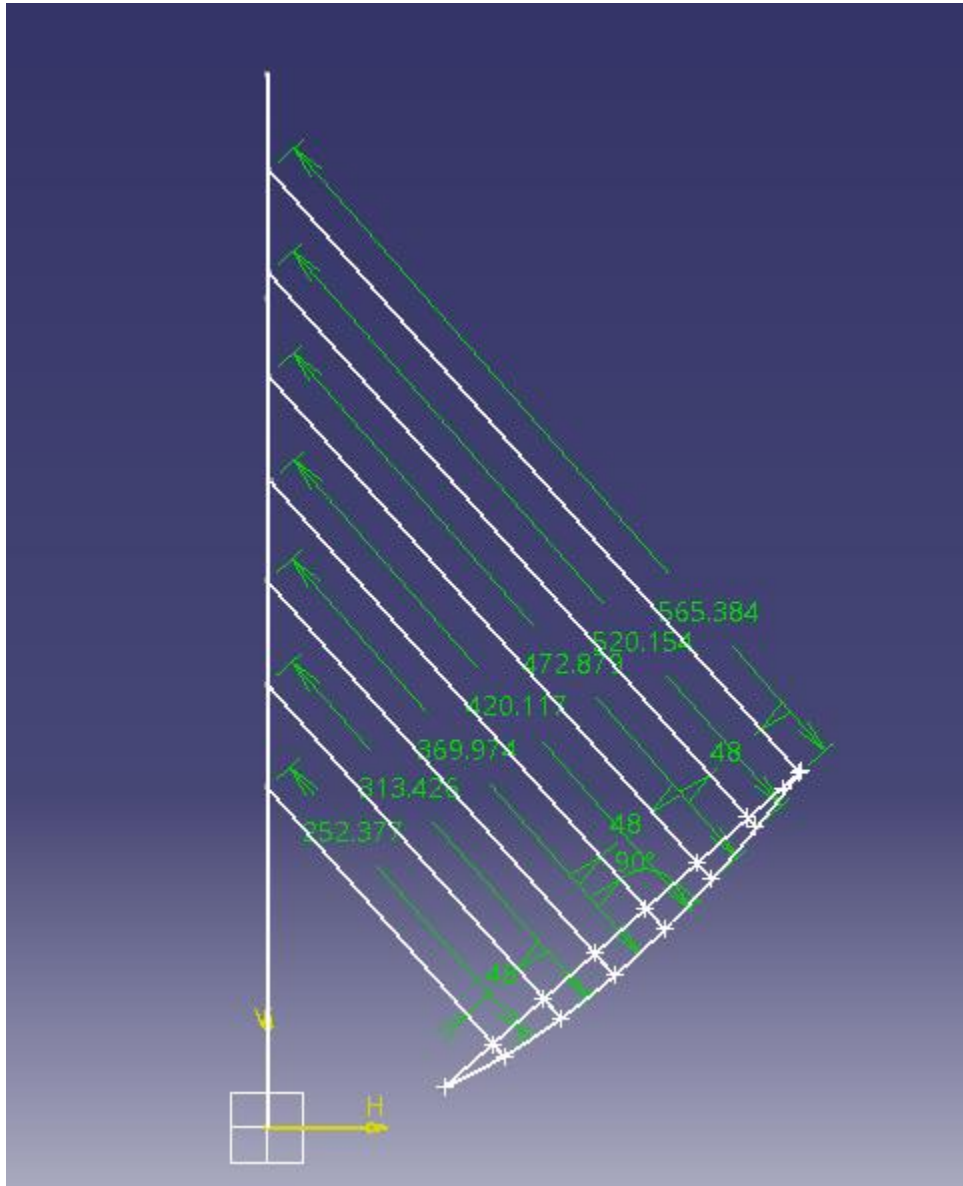


Fig. 27. Bending Radius of the crossbar.

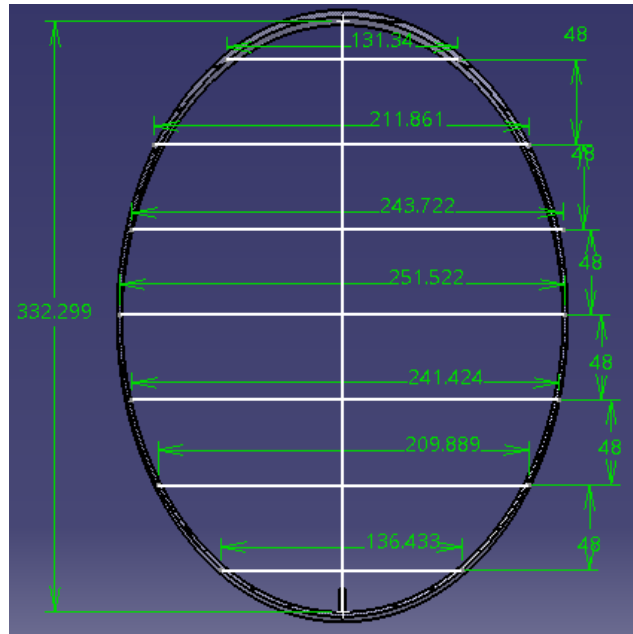


Fig. 28. Intersection points of seven crossbars for the elliptical frame of Scheffler dish.

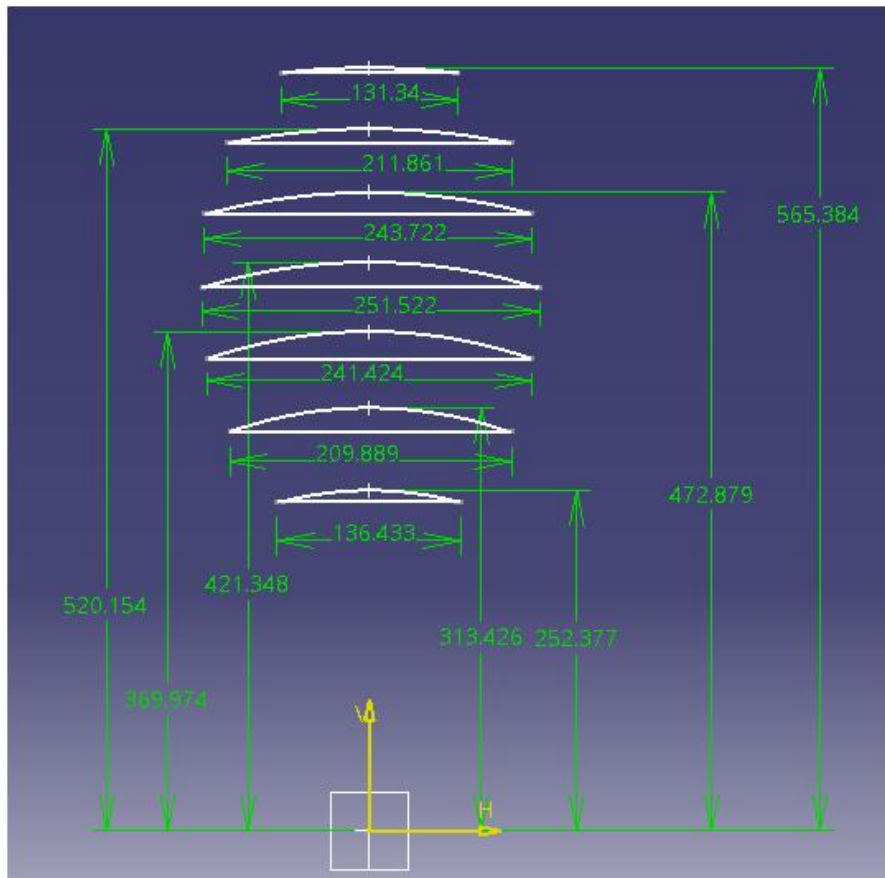


Fig. 29. Radius of bending and length of ellipse

For the construction of a Scheffler reflector, it is necessary to know the exact measurement of the arc length of the crossbars. The arc length of the crossbar is calculated by using the radius of bending, the length of the ellipse, and the angle formed between the length of the ellipse and the radius of bending.

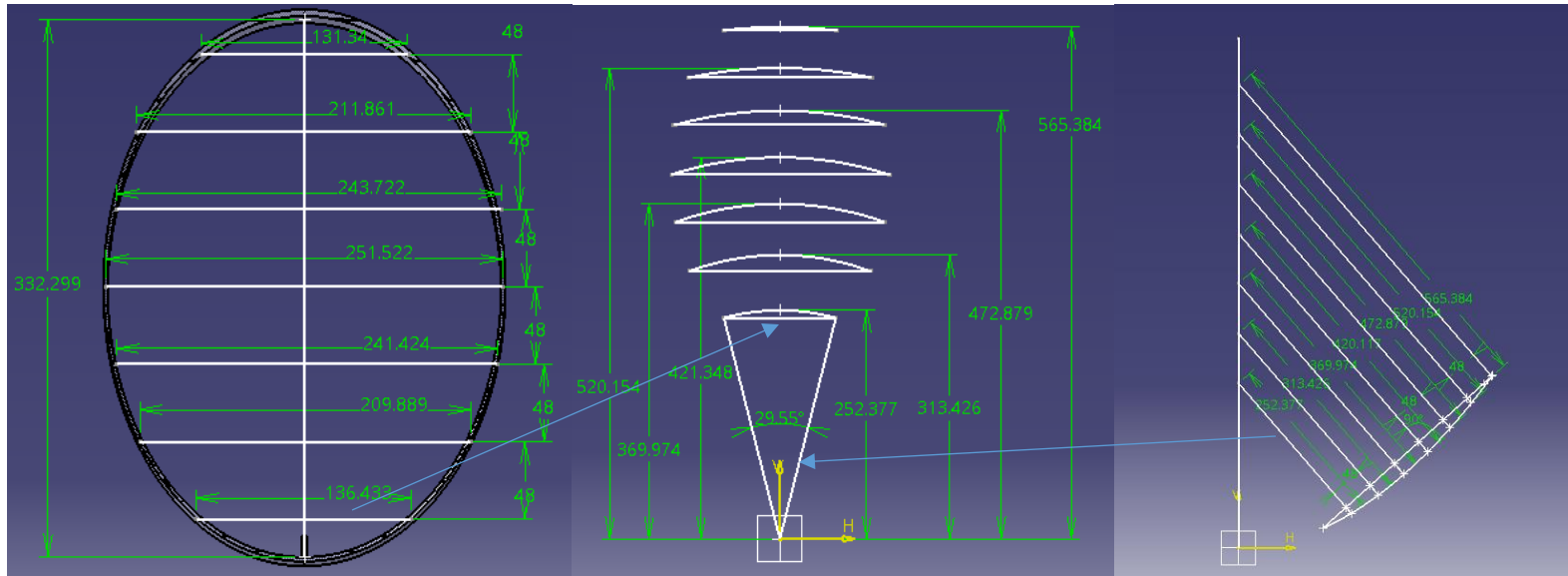


Fig. 30. Estimation of arc Length

Table 7. Estimation of arc Length

{arc length of cross bar}
 {teta=29.546}
 beta=2*pi*teta/360 {change degree into radians}
 {r=252.377} {radius of bending}
 s=beta*r {arc length of cross bar}

Parametric Table: Table 2

	r [cm]	teta [-]	β [-]	s [cm]
Run 1	252.4	29.55	0.5157	130.1
Run 2	313.4	38.74	0.6761	211.9
Run 3	370	38.02	0.6636	245.5
Run 4	420.1	34.33	0.5992	251.7
Run 5	472.9	29.32	0.5117	242
Run 6	520.2	23.03	0.4019	209.1
Run 7	565.4	12.91	0.2253	127.4

The circles were drawn by using a compass on a flat surface and use this as a jig. Bend the arc length with the bending radius. After that, check if the bent pieces are twisted or not. 8mm diameter round bar was used for cross bar manufacturing.



Fig. 31. Manufactured cross bar.

After cross bar was checked the crossbar and ellipse are welded on central bar by concerning their position.

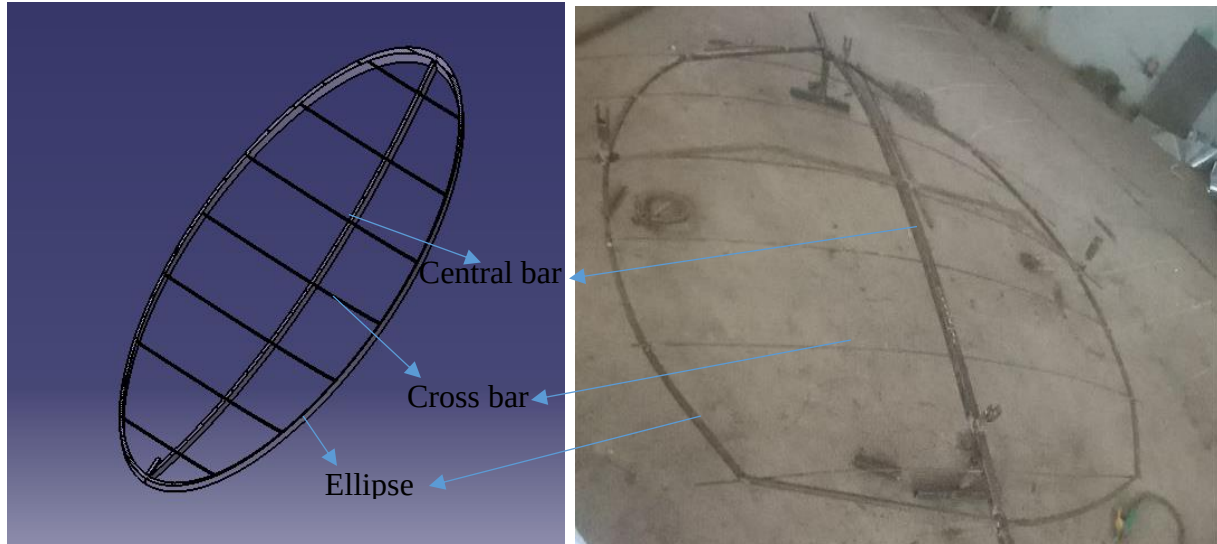


Fig. 32. Manufactured Scheffler reflector frame.

3.2.7 Design of stand

Stand was to be made with Mild steel square pipe 4cm*3mm.

For the design of the stand, we must consider the length of the central bar at winter solstice with the latitude of the location. As the angle of incidence is the same as the angle of reflection, we must change the angle of the Scheffler with the declination angle to get our receiver at a fixed point.

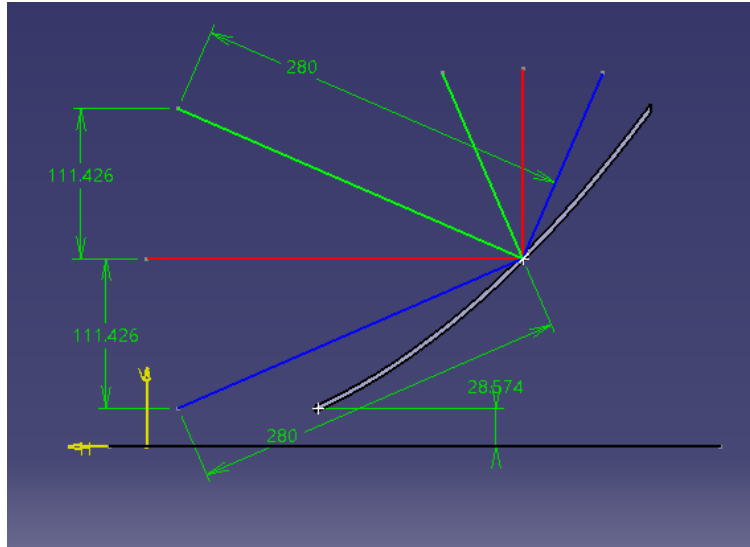


Fig. 33. Changing of focal point with seasonal change.

Without changing the angle of the Scheffler from the equinox (red line) the distance from the expected focal to the new focal (blue and green) is 111cm, and the distance from the ground to the lower end of Scheffler is 28 cm at the equinox but it also changes with the declination angle. To change these distances to the expected focal point the angle of the Scheffler must be adjusted to the declination angle.

The graph below shows when Scheffler reflector adjusted to the winter solstice without considering latitude, the location at the end of the lower point of Scheffler is 3.6 cm below the ground

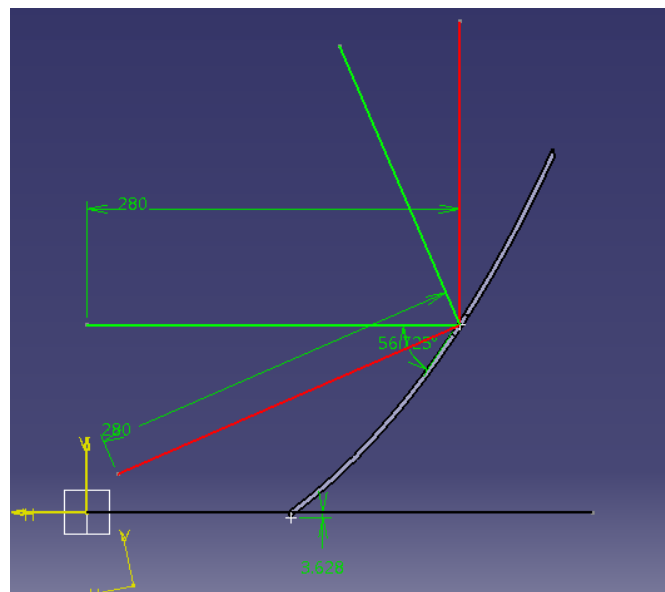


Fig. 34. End of the lower point of Scheffler at winter solstice

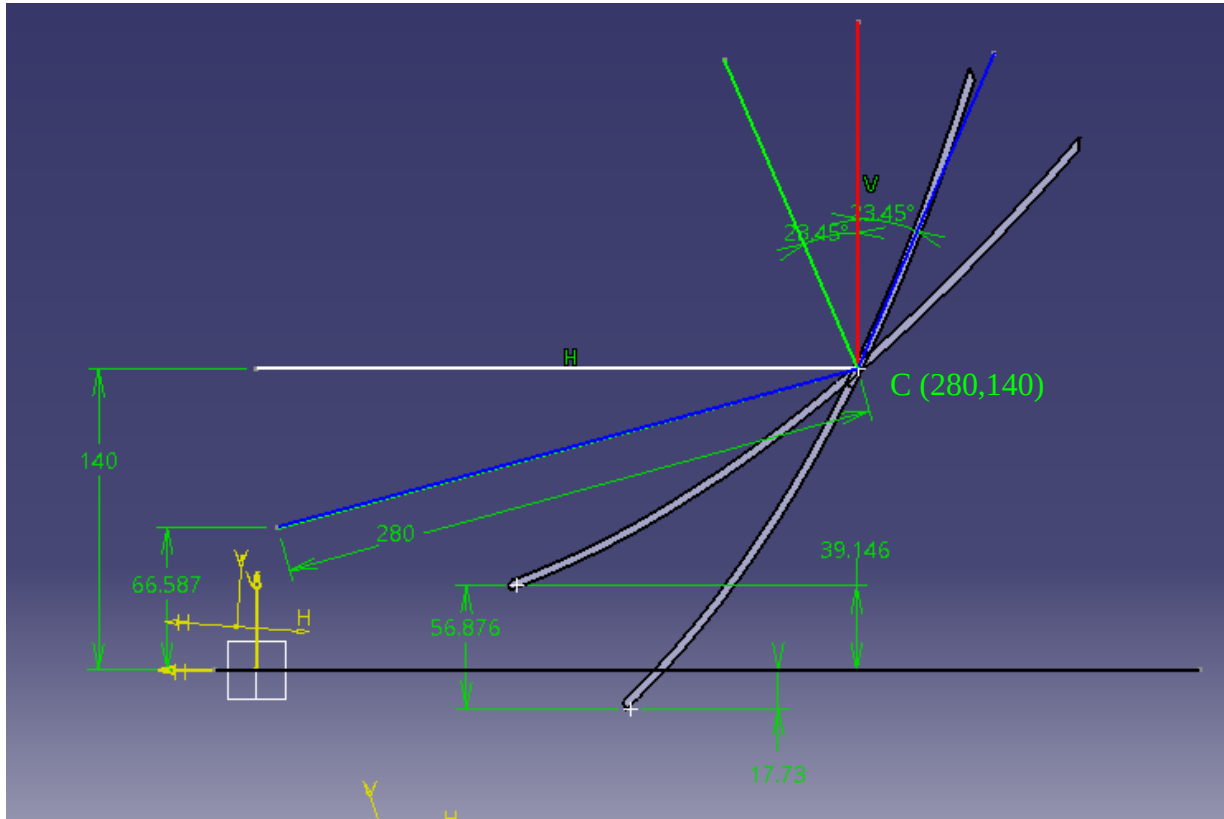


Fig. 35. End of the lower point of Scheffler at winter solstice with latitude of location

But when considering latitude 7.6° (Jimma) the end of the lower point at winter solstice is 17 cm below the ground and also change the receiver position from 140 cm to 66.587 cm.

For the design of the stand, the lower end point at winter solstice at least 15cm from the ground for safety purposes.

By considering this, point C increased from $x=140$ cm to $x=172$ cm simultaneously the focal point also increased from $x=66.587$ cm to 128.7cm

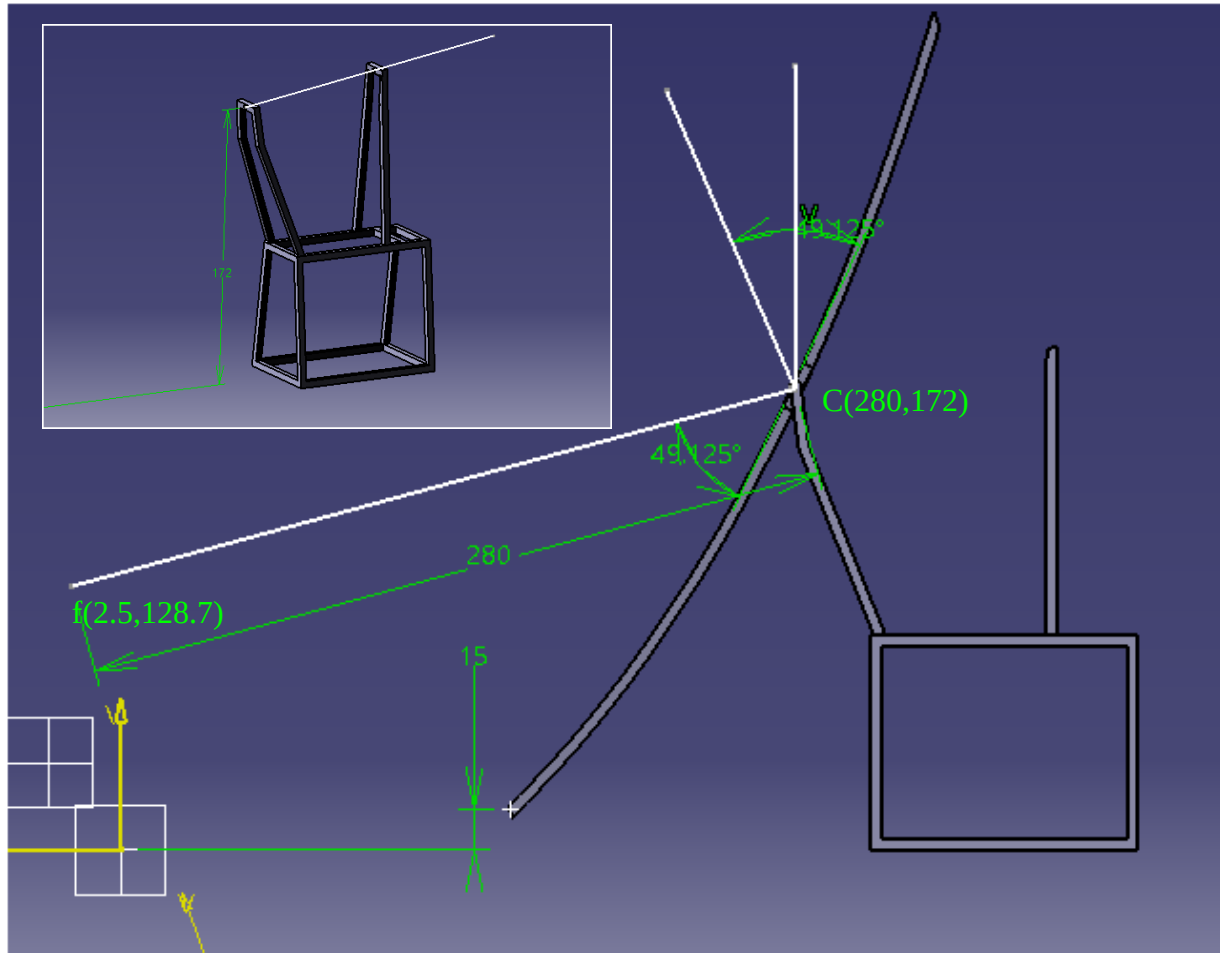


Fig. 36. Design of stand by considering latitude

3.2.8 Design of the Tracking system

Mechanical tracking system was used. The force for rotating the reflector was given by a weight, which is applied to the axis of rotation. The weight was pulled up manually when the reflector is rotated into its morning position. A mechanical clockwork was used as a breaking system. The pendulum was used to give way to the fall of the weight at intervals (period of the pendulum) that allow the reflector to rotate simultaneously with the sun. For the seasonal movement telescopic bars, one in the lower end and other one in the upper end was used.

Materials used for tracking

- ✓ 48 teeth bicycle sprockets
- ✓ 6 teeth gear
- ✓ 10 teeth sprocket

Design of tracking system starts from time of escapement

According to Matthias Leuenberger periods of the pendulum for 31.5 cm length of the pendulum is 2.35 sec [43].

The time taken to finish one revolution of 48 teeth sprocket is

$$48 \times 2.35 = 115 \text{ seconds}$$

48 teeth directly coupled with 6 teeth gear

6 teeth driven by 96 gear from these gear ratio

$$x/115 = 96/6$$

$x = 1840 \text{ sec}$, 96 gear to finish one revolution 1840 sec is needed

96 gear directly coupled with 10 teeth sprocket

10 teeth sprocket is driven by chain found in the channel of tracking

12 hours is needed for channel to finish half revolution

$$12 \times 60 \times 60 = 43200 \text{ sec is needed}$$

From these relationships, the radius of channel tracking is calculated

$$y/43200 = 10/1840 \quad y = 230 \text{ (diameter) cm}$$

The radius of tracking channel is 115 cm

10 teeth sprocket need the chain to rotate, the radius tracking channel changes into a chain

The pitch of one chain is 1.27 cm

$$\text{Total of chain links} = (115 \times \pi) / 1.27 = 235$$

235-chain link is needed.

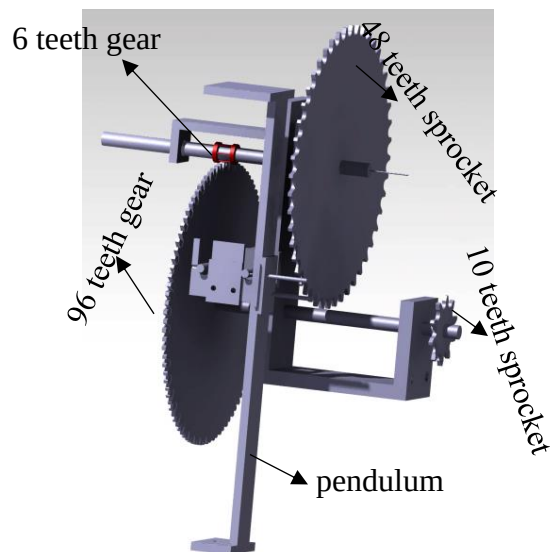


Fig. 37. Design of the mechanical tracking

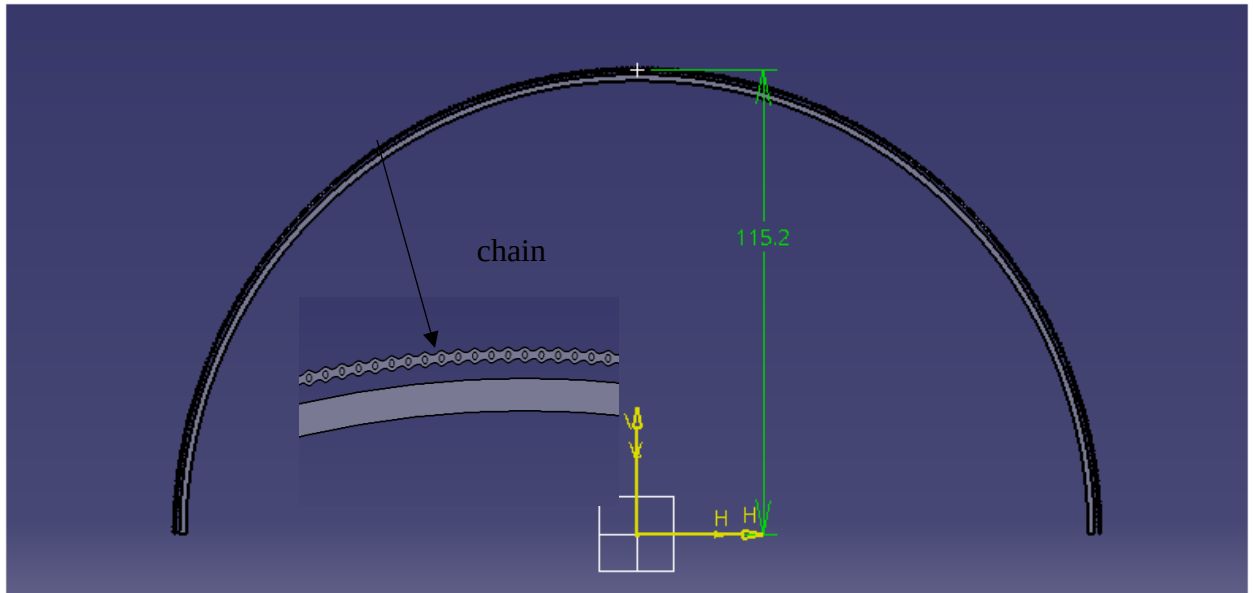


Fig. 38. Design of tracking channel

After designing, manufacturing and assembling Scheffler reflector parts, the silver-coated 3mm thick glass mirror was cut into 25*20 cm and attached to the crossbar using galvanized iron and 1mm diameter copper wire.

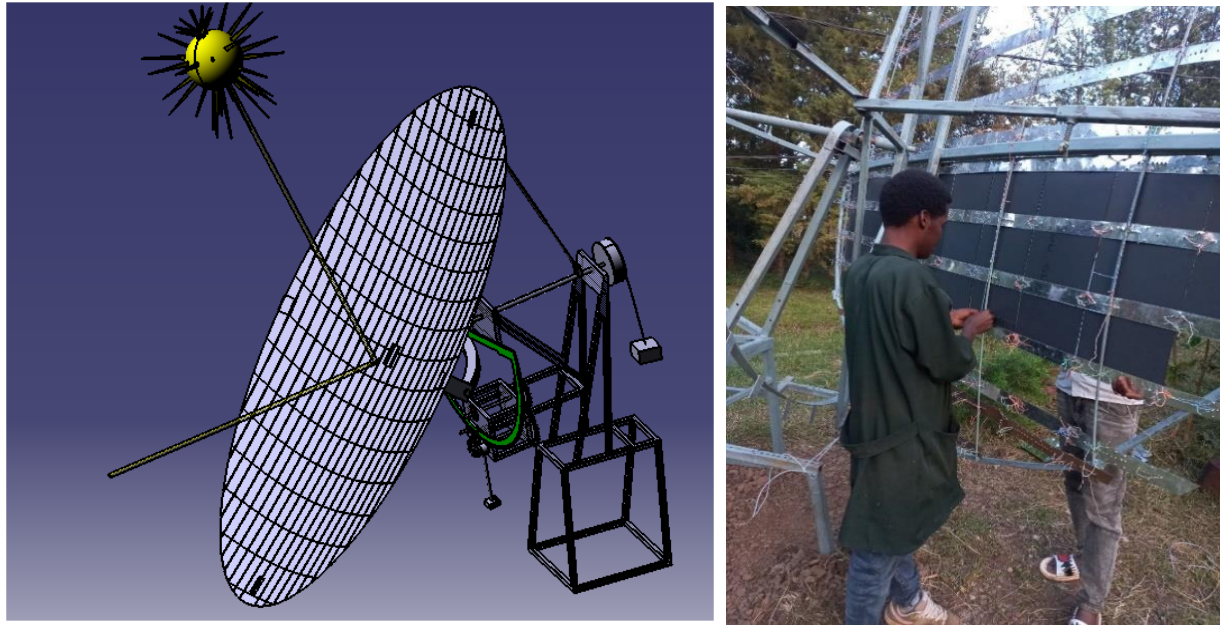


Fig. 39. Assembled parts and attaching mirror in the Scheffler frame

3.2.9 Design of Thermal Energy Storage

Sensible heat storage

A sensible thermal storage system stores the thermal energy by increasing the temperature of the storage medium, without causing any phase transition in the storage material [44].

System Material selection

When selecting thermal energy storage different criteria should be considered. Therefore, TES must have a high energy density (per-unit mass or per-unit volume) in the storage material, good thermal properties, mechanical and chemical stability of storage material, compatibility between storage material, complete reversibility for a large number of charging/discharging cycles or prolonged use, minimum thermal losses, ease to control and minimal cost.

Determination of the total mass of the Thermal storage medium

The total amount of thermal storage required to generate vital energy for the oven for two baking cycles can be calculated by using the following equation

$$Q_s = \int_{T_1}^{T_2} m_s * C_s dT = \int_{T_1}^{T_2} \rho * V * C_s * dT \dots \dots \dots (21)$$

where dT is the storage temperature increase over a specified period, Cs is the specific heat of the storage, ms is the mass of the storage materials

The energy needed to bake 2.6 kg of dough is 1002 for 45 minutes for one cycle
 for two cycles $1002*2= 2004$ KJ is needed
 from this mass of storage is calculated

$$Q_s = m_s * C_s * \Delta T \dots\dots\dots(22)$$

$$m_s = Q_s / C_s * (\Delta T)$$

To satisfy the baking condition the storage temperature must be greater than 200°C
 let's assume 220°C

the ambient temperature is 25°C

Due to their availability and cost-effectiveness, basalt rocks are selected for sensible thermal storage. According to Douglas W. Waples, the density and specific heat capacity of basalt rocks is 2870 kg/m^3 and 898 J/Kg K [45].

$$m_s = 2004 / (220 - 25) * 0.898$$

$$m_s = 21.64 \text{ kg is needed}$$

Assume the efficiency of storage is 80%

$$\text{Mass needed} = 1.2 * 21.64 = 25.97 \text{ kg}$$

The volume of storage medium

The volume needed is calculated from the mass storage and density of materials

Volume = mass storage/density of storage

$$V_s = m_s / \rho_s \dots\dots\dots(23)$$

$$V_s = 25.97 / 2870 = 0.00905 \text{ m}^3$$

$$V = 0.00905 * 1000000 \text{ cm}^3 = 9050 \text{ cm}^3$$

It is important to consider the volume of the air inside the storage because air is the medium for heat transfer, during charging and discharging. Assume 25% of the storage volume is air.

$$\text{Total volume needed} = 1.25 * 9050 = 11312 \text{ cm}^3 = \text{approx.} = 10\text{cm} * 30\text{cm} * 40\text{cm}$$

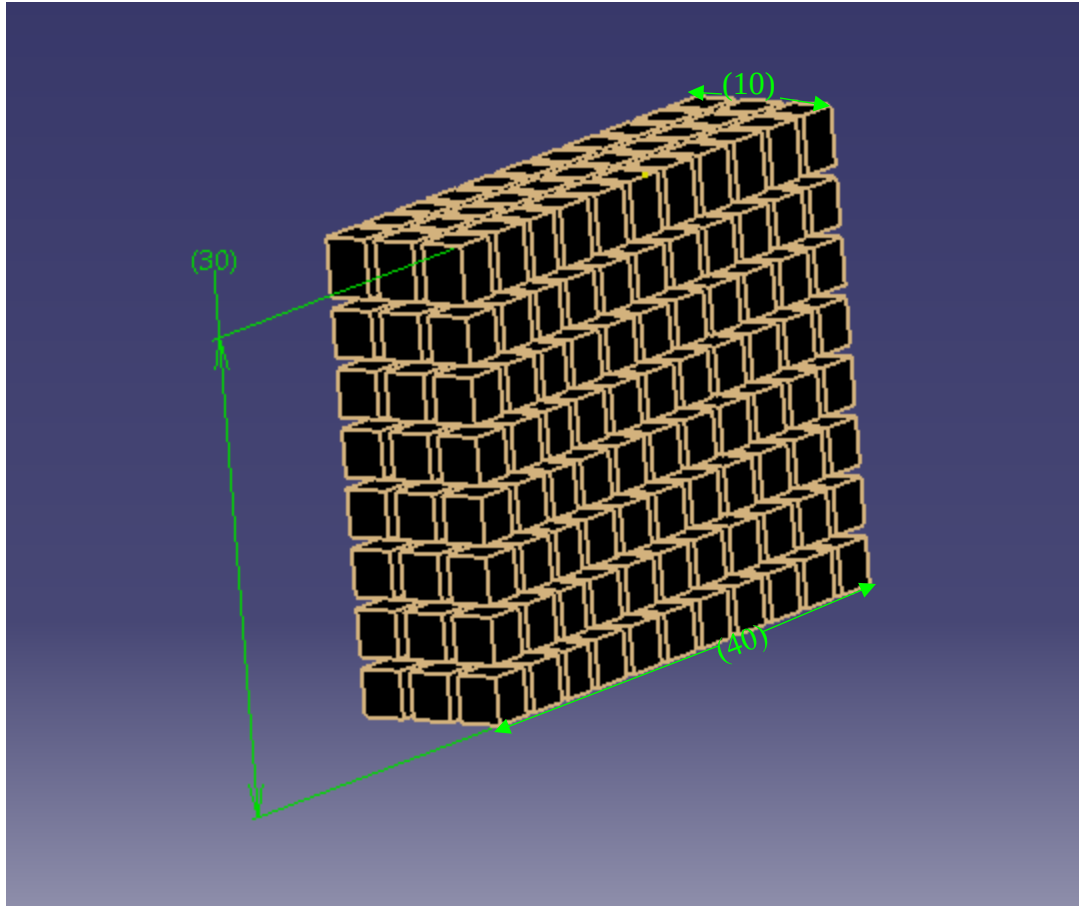


Fig. 40. Design of storage

3.2.10 Design of the receiver

The receiver is the key component of the solar thermal system where the solar irradiation reflected by optical concentrators is absorbed and converted to high-temperature thermal energy. The receiver is constructed from 1.5 mm thick aluminum sheet, which is bent in a zig-zag profile. According to Muller ZigZag improves efficiency and minimizes radiation losses [22]. The area of the receiver is greater than or equal to the area of a single mirror used for reflector construction. From this $35 \times 30 \times 30 \text{ cm}^3$ was constructed. The zigzag receiver was protected by a $35 \times 35 \text{ cm}^2$ 3.8 mm thick ordinary glass window and it was cut into strips to avoid the breaking of glass due to heat expansion.

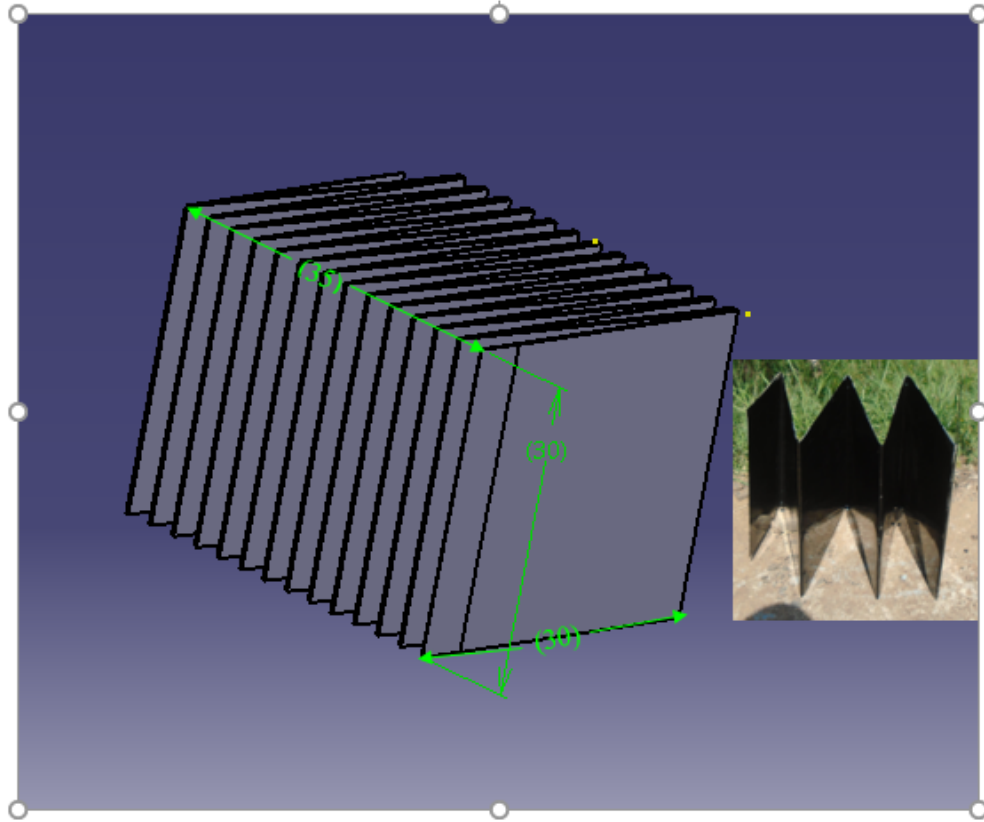


Fig. 41. Design of the receiver

3.2.11 Design of the oven

The oven must satisfy the energy needed and the thermal storage capacity.

For the design of the oven, the following criteria must be considered

- 1) Number of tray
- 2) Distance between tray
- 3) Volume of the receiver
- 4) Volume of the thermal storage
- 5) Thickness of insulators

Assume two trays.

Assume the distance between trays is 15 cm, considering adequate airflow between the trays.

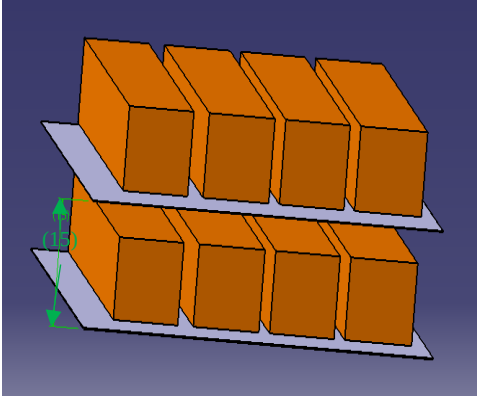


Fig. 42. Distance between Trays

Estimation of critical insulation thickness

According to Muller, the insulation thickness for rectangular surface solar baking ovens operating at 250°C is 10cm is optimum[46]. From these 10 cm 0.04 W/m. K thermal conductivity fiberglass was selected for oven surface insulation.

By considering all measurements and assumption $95*70*70$, cm³ oven was manufactured.

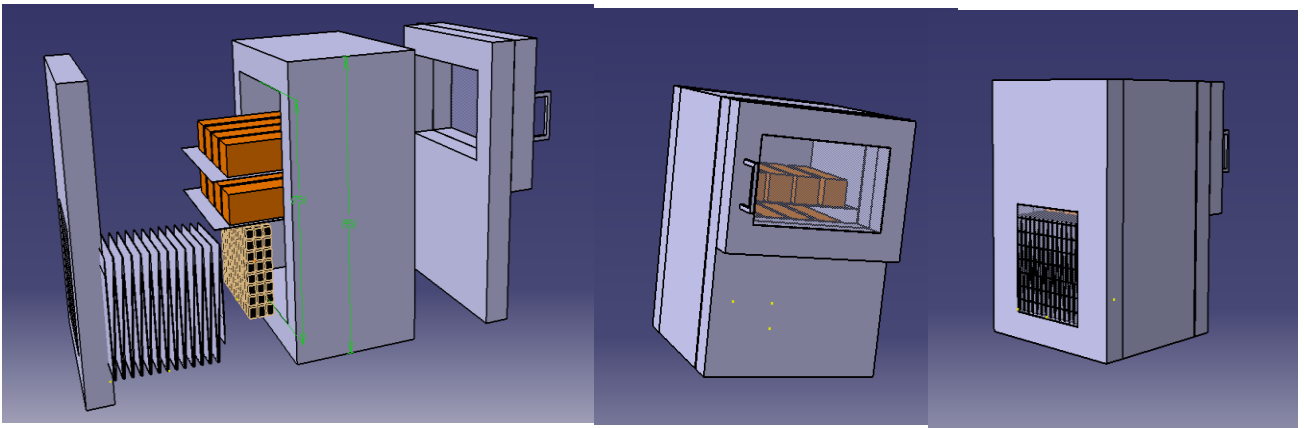


Fig. 43. Design of the oven

Assembly

The entire components of Scheffler reflectors were mounted on the stand after all parts are manufactured by using welding, bolt and nut.

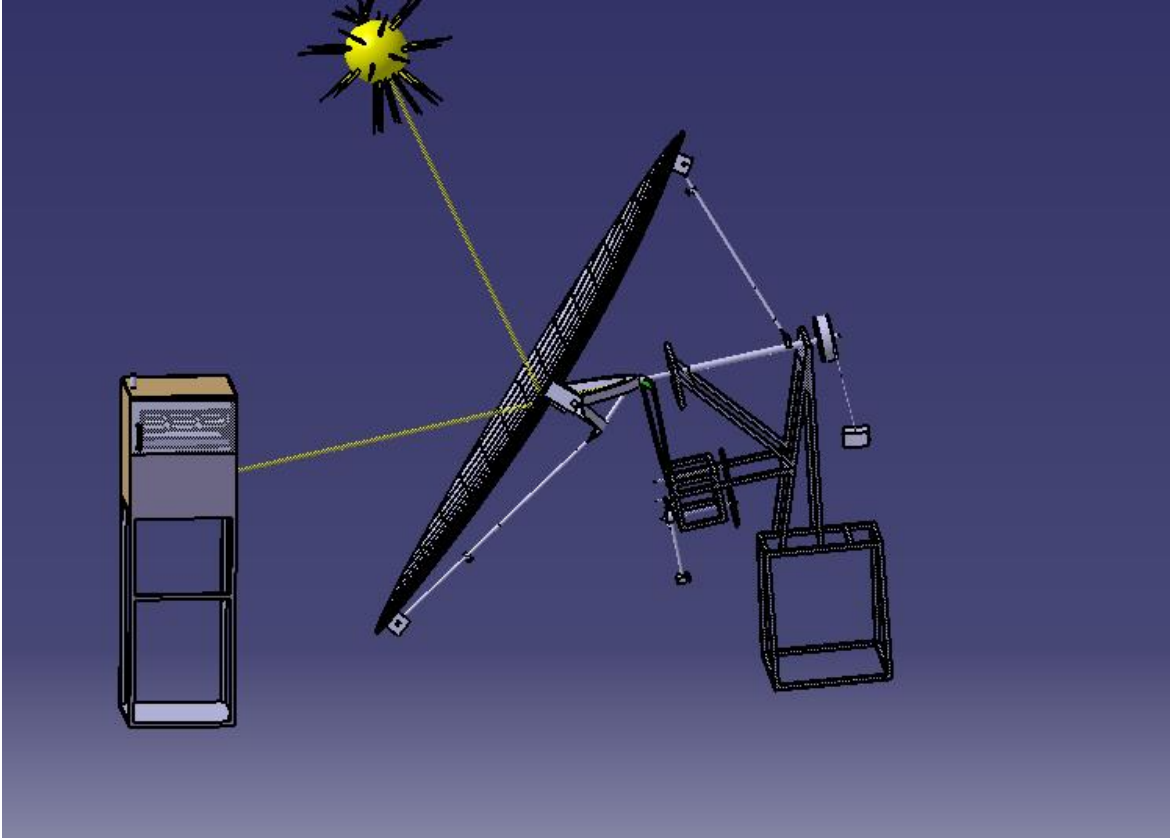


Fig. 44. Assembled all parts



Fig. 45. Manufactured Scheffler Reflector.



Fig. 46. Manufactured Scheffler Reflector

3.2.12 Ray tracing

Ray tracing methods for evaluating concentrators

Analysis of concentrating collectors is commonly done by Ray-trace methods [47]. This is the process of following the path of large rays of incident radiation through the optical system to determine the distribution of processed rays on the surface of interest[47]. Ray-trace methods were done by using Geometrical Optics COMSOL Multiphysics. The Geometrical Optics interface can be used for ray optics simulation of cameras, telescopes, spectrometers, solar collectors, laser systems, and room lighting [48]. It solves for the position and wave vector of individual rays. It also allows them to interact with boundaries that intersect their paths.

Scheffler reflectors and receiver models were developed using Catia v2021 and imported to Comsol Multiphysics to model and analyze the optical performance of Scheffler reflectors with

the surface of the receiver. 6.8m² collector area, 1.4 m focal length and 1m*3m area of receiver were used for analysis. A dedicated boundary condition called Illuminated Surface is used to release rays directly from the surface of the dish, initializing their directions as if they were reflected from a distant plane wave source. The direction at which the rays are released from the surface of the dish depends on the incoming ray direction vector n_i and the outward surface normal n_s , according to the formula[48].

$$n_r = n_i - 2(n_i \cdot n_s)n_s \dots \dots \dots (24)$$

The intensity of each ray can be computed along its trajectory; the evolution of ray intensity depends heavily on the curvature of the dish. Each ray released is also assigned a fixed power, which is assigned an appropriate value based on the Source power setting for the Illuminated Surface feature. When the rays reach the surface of the solar collector, the Wall feature stops them. The Deposited Ray Power sub feature computes the incident heat flux in the focal plane. Some of the incoming radiation is absorbed by the dish itself. In this model, the absorption coefficient is set to 0.1, meaning that 90% of the incoming radiation is reflected. An additional correction is included due to the finite size of the sun. Not all incident rays will be parallel; instead, the incident rays are sampled from a narrow cone with maximum angle, ψ_m , of 4.65 mrad. For these analysis, rays are released from 100,000 distinct points. For better accuracy, fine meshing was used. Deposited ray power was enabled to compute the total incident energy flux on a surface based on the power of incident rays.

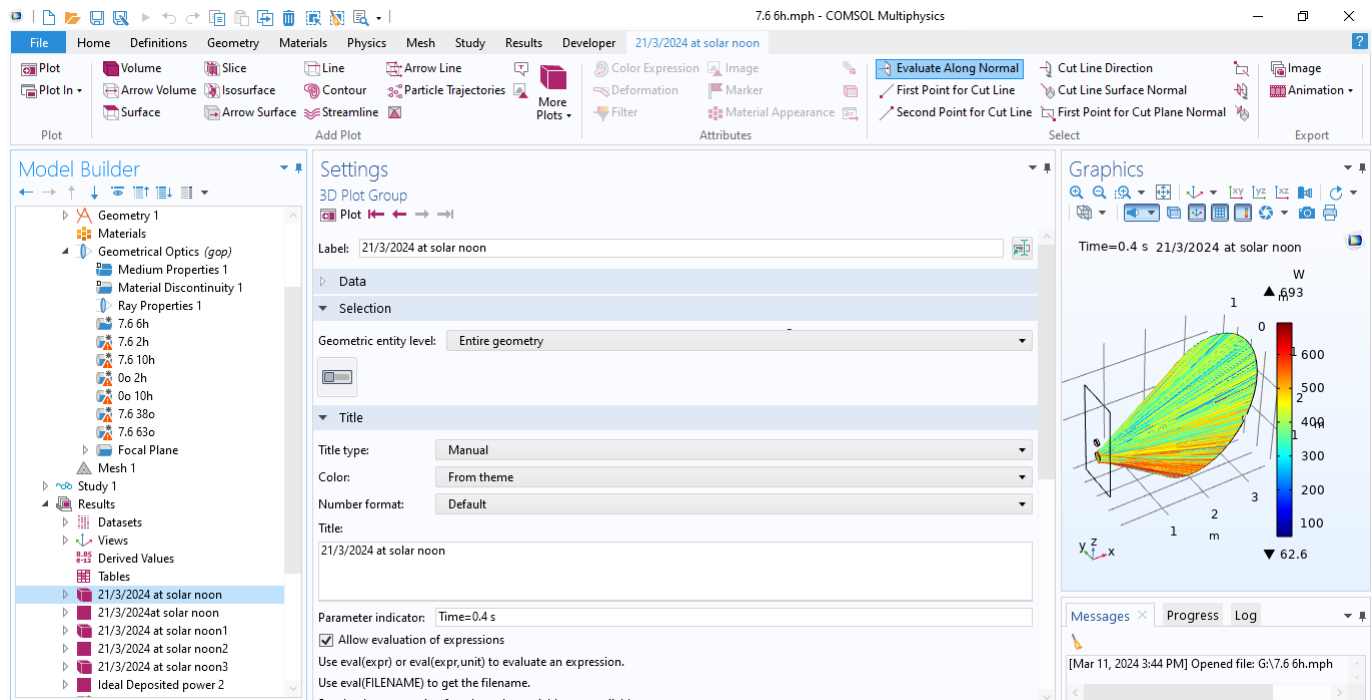
$$P = \alpha * I \dots \dots \dots (25)$$

Where: α , is the absorption coefficient of the material. This parameter describes how strongly the material absorbs the incident radiation. I is the intensity of the incident radiation (unit: W/m²). This represents the power per unit area carried by the rays.

Solar radiation was used under ray release feature to determine the ray direction vector in real time. Under solar radiation ray release feature knowing solar noon time is the most important parameter that helps estimate the position of the sun's given location. Solar noon is the moment when the sun reaches its highest point in the sky for a given location. It occurs when the sun crosses the observer's meridian, which is an imaginary line that passes through the North and South Poles[41]. At this time, maximum sunlight exposure can be achieved, resulting in optimal energy generation. It is also used for determining the true north.

geotimedate.org is a free web tool that calculates sunrise, sunset, solar noon, and day length [49]. The solar noon is taken from geotimedate.org at a given latitude and longitude in real-time (Figure 34). To compute the result defining sun direction is important.

The sun direction is determined the y-axis of the global coordinate system points due north, the z-axis points towards zenith and the x-axis points due west. 750 W/m^2 solar intensity was used.



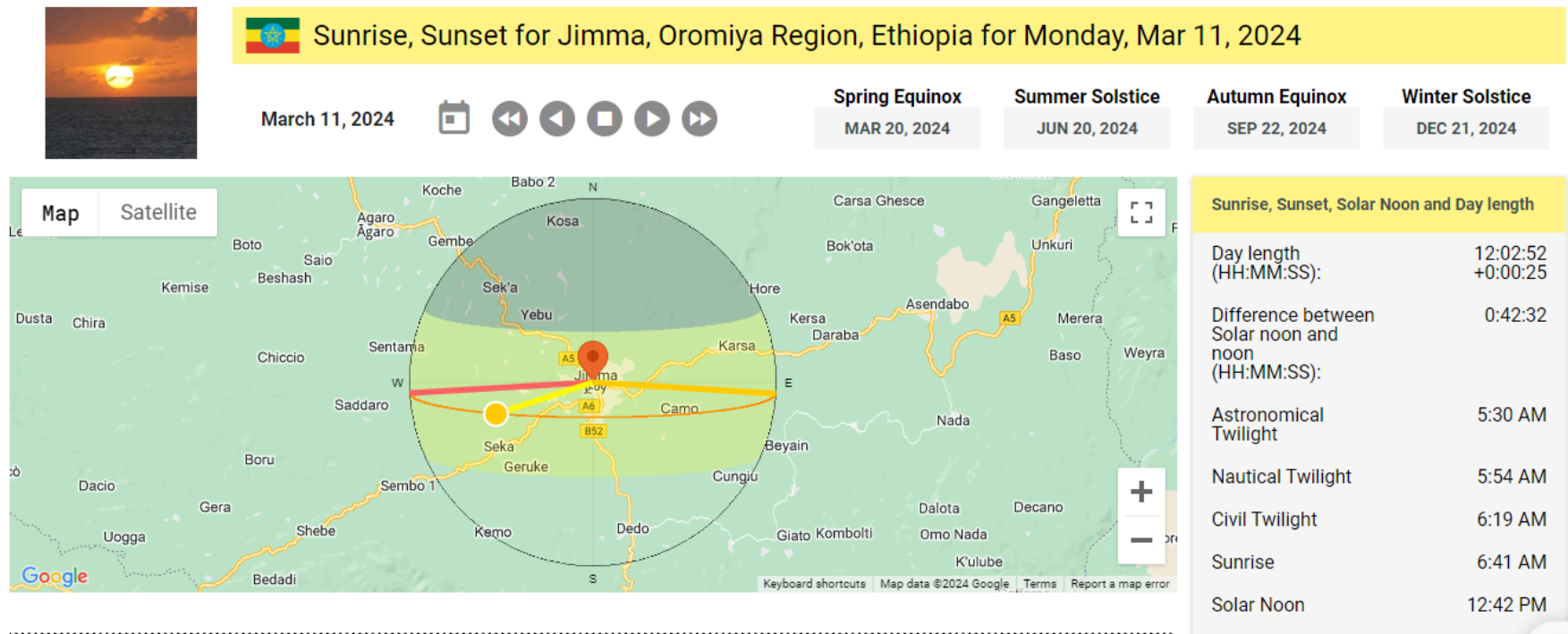


Fig. 47. geotimedate.org plat form

Rays to release:

Reflected

Incident ray direction vector:

Solar radiation

Location defined by:

Coordinates

Location:

Latitude (+ to N), in °	Longitude (+ to E), in °	Time zone (+UTC)
7.6	36.8	+3

Date:

Day	Month	Year
21	3	2024

Local time:

Hour	Minute	Second
12	44	47

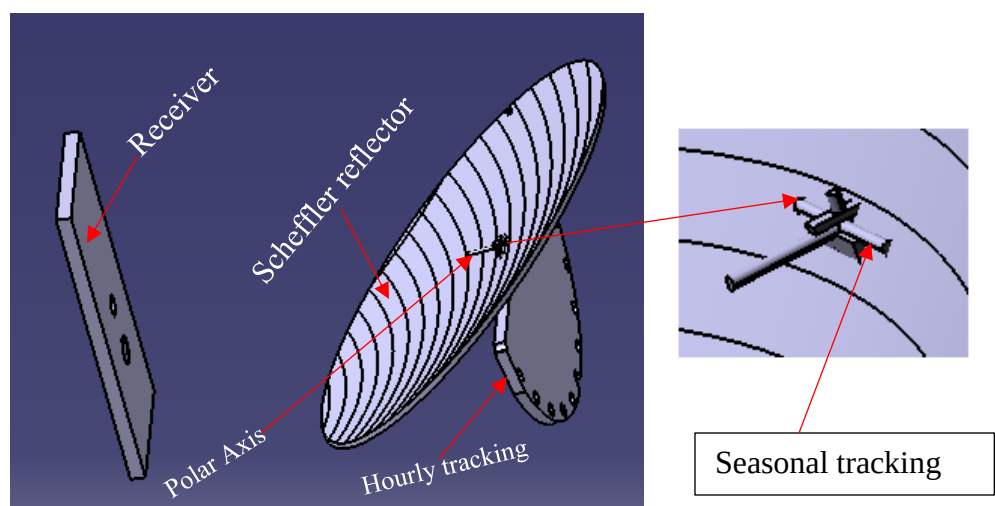


Fig. 48. Model of the Scheffler reflector

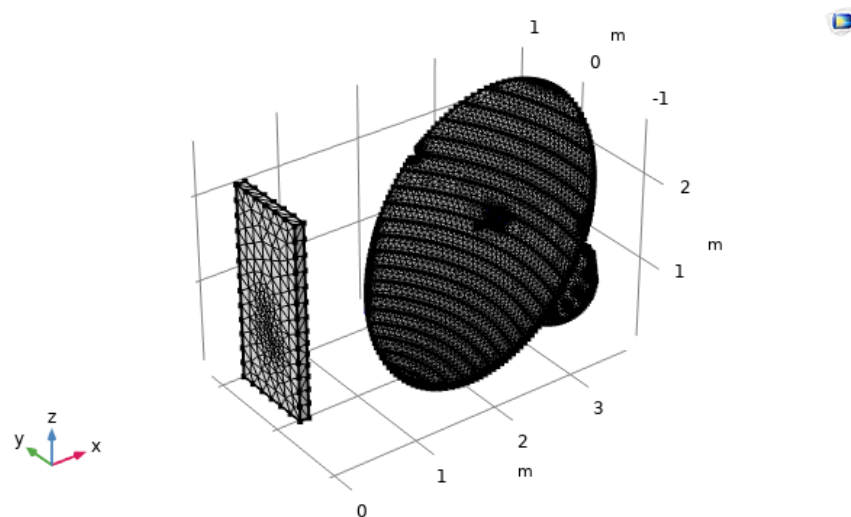


Fig. 49. Meshing of Scheffler reflector model

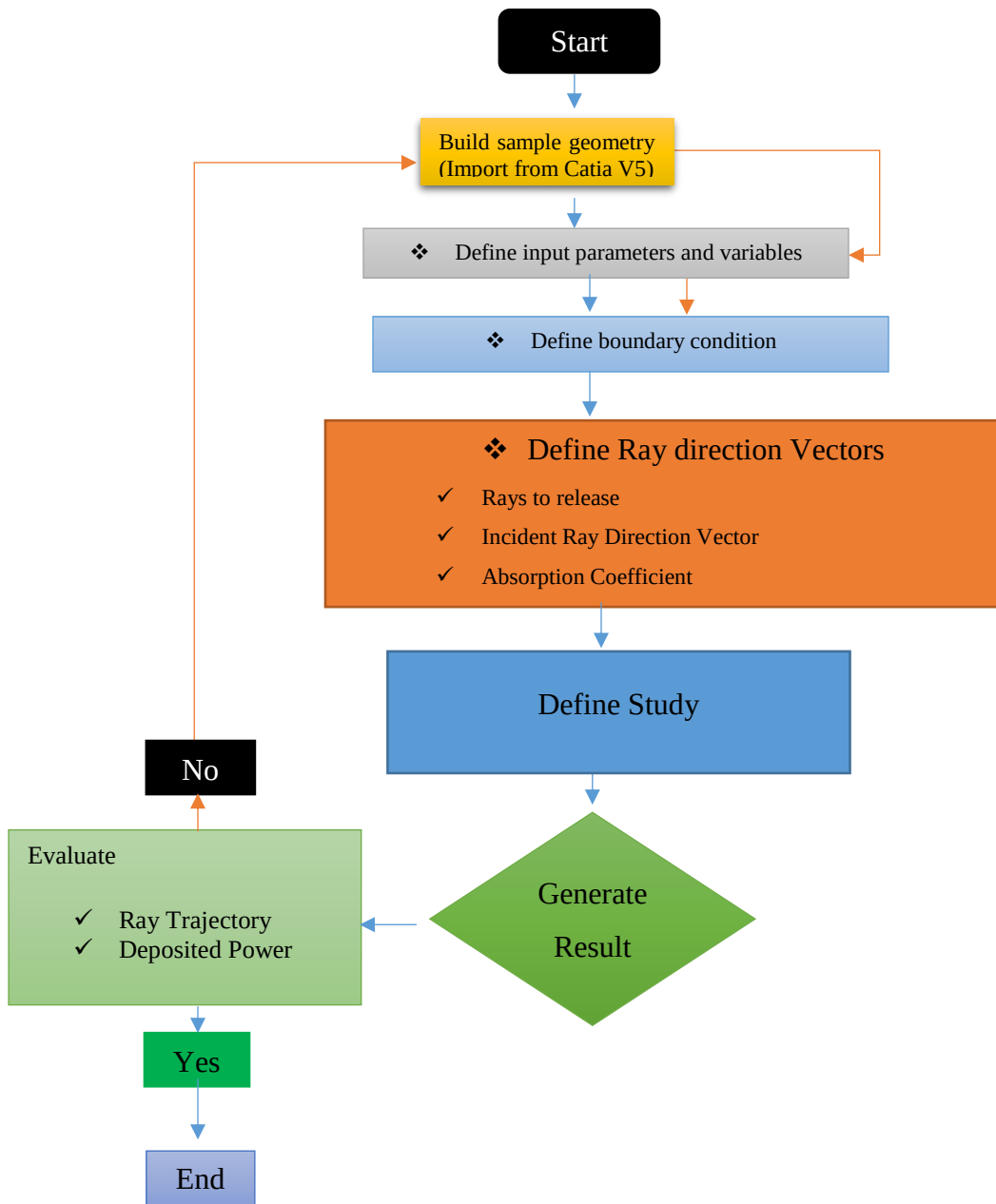


Fig. 50. Ray tracing methods for evaluating concentrators flow chart

3.3 ANSYS Fluent Simulation of Receiver

3.3.1 Prediction of temperature and heat flux values on the receiver surface

Heat flux values and Temperature on the receiver surface were simulated in the Solar Load Cell Model.

ANSYS FLUENT R19.2 provides a solar load model that can be used to calculate radiation effects from the sun's rays that enter a computational domain. Under solar load cell discrete ordinates, (DO) irradiation was used. Discrete ordinates (DO) model is used to calculate radiation effects within the domain, an option is available to supply outside beam direction and intensity parameters directly to the DO model.

The receiver model was developed using Catia v2021 and imported to ANSYS Fluent to simulate temperature and heat flux around the surface of the receiver. For better accuracy, fine meshing is used. Energy equations, laminar model equations, and Discrete ordinates (DO) equations are considered, Convergence criteria is set for scaled residuals of energy. For discretization of momentum and energy, First-order upwind and Third-order Monotone Upstream centered Schemes for Conservation Laws (MUSCL) are applied

The DO Model Equations

$$\frac{\partial I}{\partial x_i} + (a + \sigma_s)I(r, s) = an^2 \frac{\sigma T^4}{\pi} + \frac{\sigma_s}{4\pi} \int_0^{4\pi} I(r, s') \Phi(s, s') \Omega' \dots\dots\dots(30)$$

r= position vector

s=direction vector

S' = scattering vector

S=path length

α =absorbition coefficient

n = refractive index

σ_s =scattering coefficient

σ =staefan boltsman constant = 5.669×10^{-8} W/m²K⁴

I=radation intensity

T=local temperature

Ω' =face function

Φ =solid angle

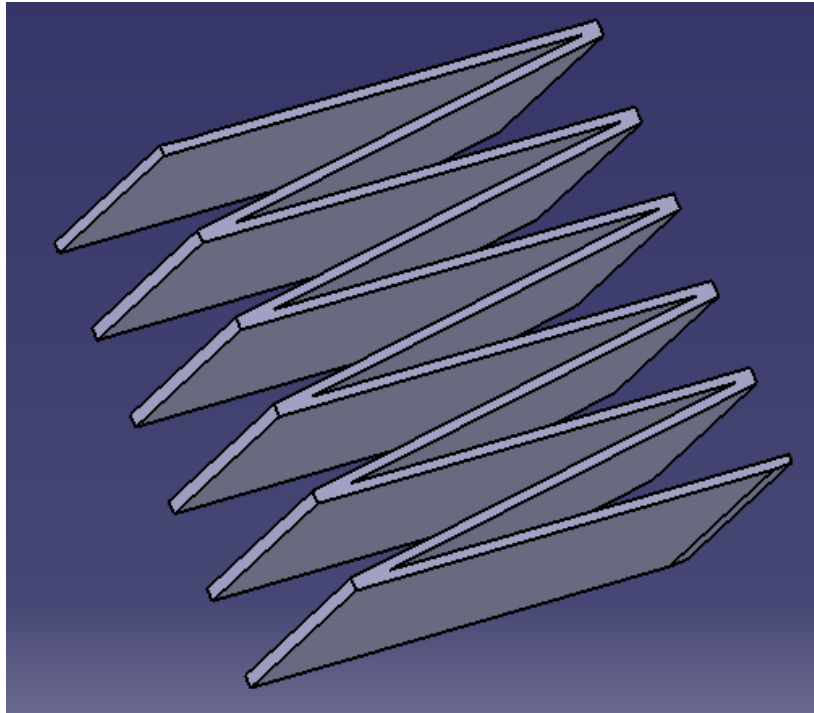


Fig. 51. Model of the Receiver

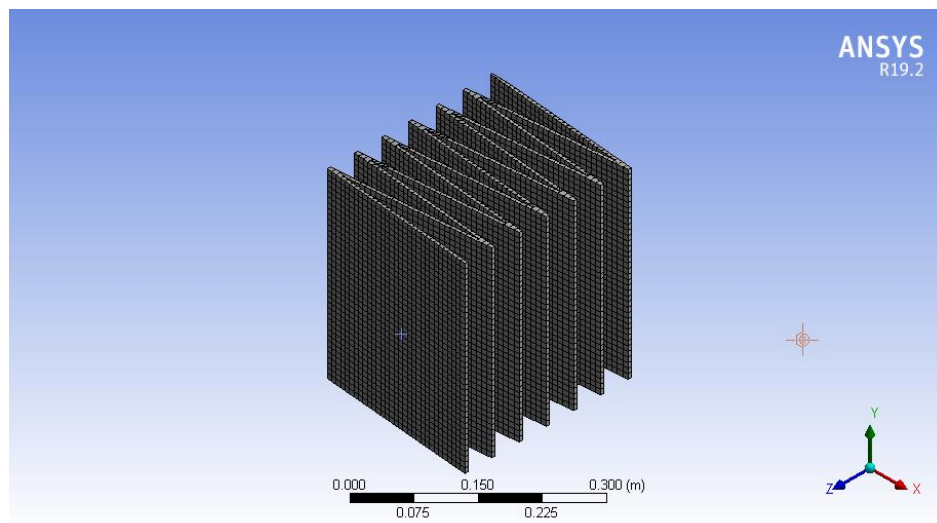


Fig. 52. Meshing of the Receiver model

3.3.2 Prediction of temperature, Velocity and Pressure distribution inside the oven

The computational fluid dynamics analysis of the solar oven for the baking system was done using ANSYS Fluent R19.2. 2D simulations were performed to study the behavior of the air domain in

terms of temperature, velocity, and pressure distribution inside the oven. The simulation result of the temperature from the surface receiver was used as input parameters.

A solar oven can be considered as a closed system during the baking process. The solar oven operates with natural convection. When heat is added to a fluid and the fluid density varies with temperature, a flow can be induced due to the force of gravity acting on the density variations. Such buoyancy-driven flows are termed natural-convection. A convective airflow is accomplished by a chimney, which leads the airflow into the top of the baking chamber. From here the air passes through the baking product transfers parts of its thermal energy and leaves the chamber at the bottom. A basalt rock storage is situated in the bottom, which helps to store the thermal energy of the circulating air.

The model was developed using Ansys design modular to simulate temperature, velocity, and pressure distribution inside the enclosed oven. The model is 0.5m width and 0.70m height.

The fluid flow characteristics are numerically studied during natural convection heat transfer for air in the rectangular enclosures. The geometry consists of a rectangular enclosure with a receiver's hot surface and chimney, trays for baking bread, and a stone cold surface, as shown in fig below. Insulated properties are implied to the other four surfaces of the enclosure. During numerical calculations, air was considered as heat transfer medium. For better accuracy, fine meshing is used near the edges of the rectangular geometry. The Boussinesq approximation is used, which implies that the fluid movement is mainly due to buoyancy force. The steady state simulations are performed with the acceleration of gravity in the y-direction (negative). Pressure based solver is applied along with the absolute velocity formulation. Energy equation and laminar model equations are considered. For buoyancy calculations, the Body force weighted method is implied for pressure-based solutions. Convergence criteria is set for scaled residuals of energy, velocity, and continuity models. For discretization of momentum and energy, First-order upwind and Third-order Monotone Upstream centered Schemes for Conservation Laws (MUSCL) are applied.

$$(\rho - \rho_0)g \approx -\rho_0\beta(T - T_0)g \dots \dots \dots (27)$$

where ρ_0 is the (constant) density of the flow, T_0 is the operating temperature, and β is the thermal expansion coefficient

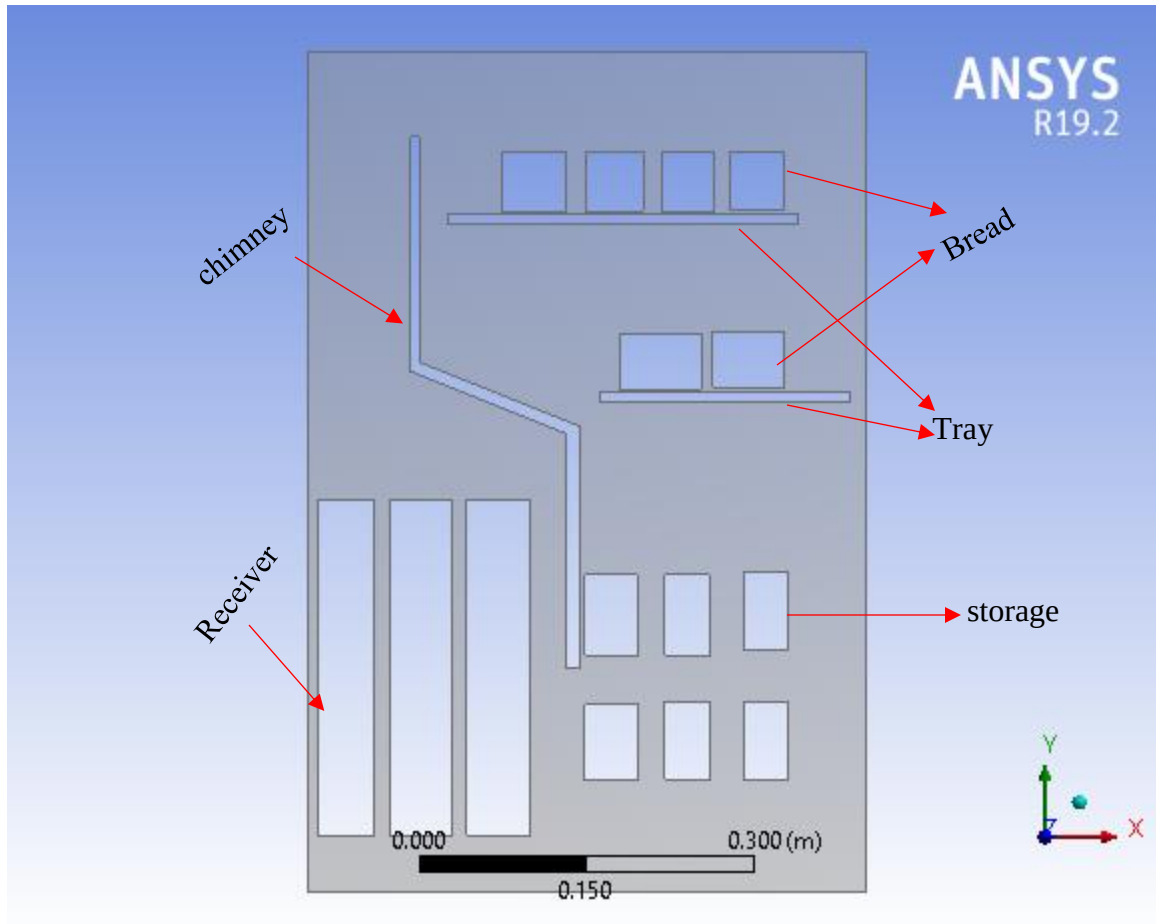


Fig. 53. Model of the Oven

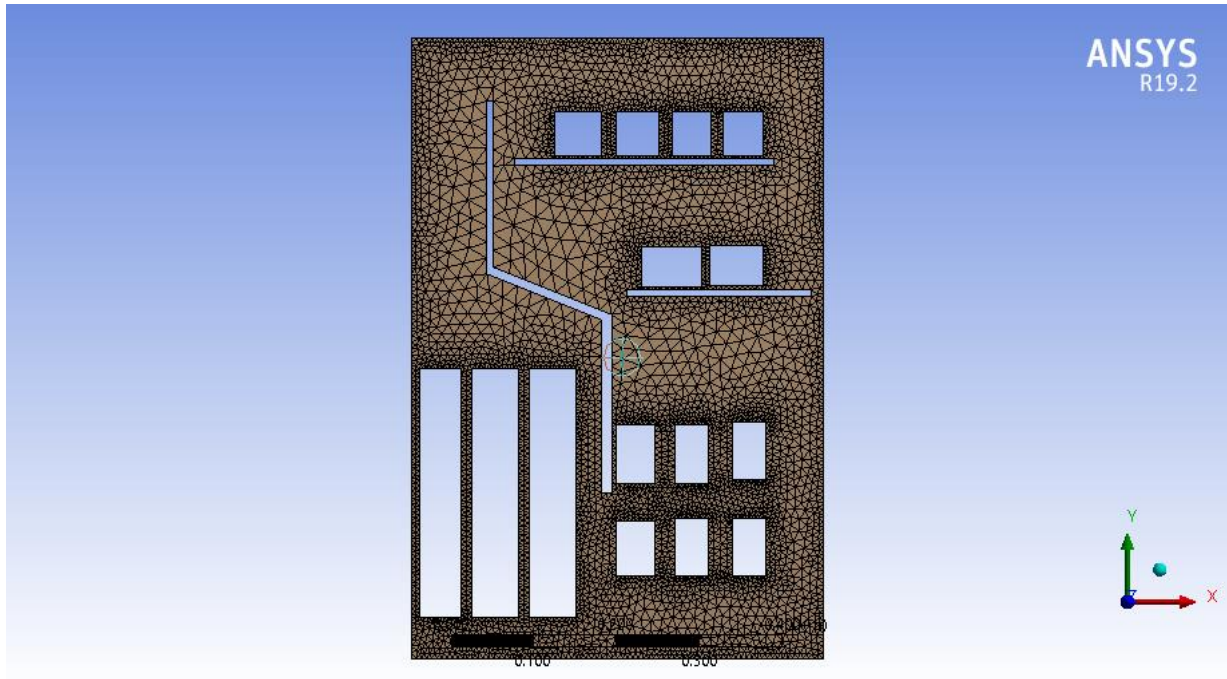


Fig. 54. Meshing of the oven model

3.4 Experimental study

After system design, manufacturing, and assembling were completed, the prototype was evaluated at Jimma Agricultural Engineering Research Center. The experiment was conducted under the area's normal weather conditions for six days, on (5-10, February 2024). The measurement data were taken for seven hours and forty-five minutes for loading and unloading. Begun from 09:15 am to 5:00 pm in the 15-minute time intervals. The K-type thermocouple probe was used to measure the receiver temperature, air temperature at the top tray, air temperature at the bottom tray, temperature inside the dough, air temperature at the inlet of storage, and temperature of storage.

3.4.1 Determination of Receiver Energy Losses

When a specific amount of concentrated solar radiation strikes the surface of a receiver, the majority of it is absorbed and given to the transport fluid, where it is taken away as usable energy. However, like in all thermal systems, heat losses to the environment through various routes of heat transfer are unavoidable.[7]. Determining thermal losses from receivers is difficult due to the wide variety of forms and designs, high temperatures, and non-uniformity of radiation flow, leading to significant temperature gradients across energy-absorbing surfaces. High temperatures mean that thermal radiation is important, leading to the loss coefficient being temperature-dependent. The thermal network for a glass cover Scheffler reflectors in terms of convection, and radiation and in terms of the resistance between plates are shown in Figure 44. The temperature of the receiver plate is T_r , the basalt rock temperature is T_b , and the absorbed solar radiation is S . In a simplified way, the various thermal losses from the collector can be combined into a simple resistance, RL , as shown in Figure 44,

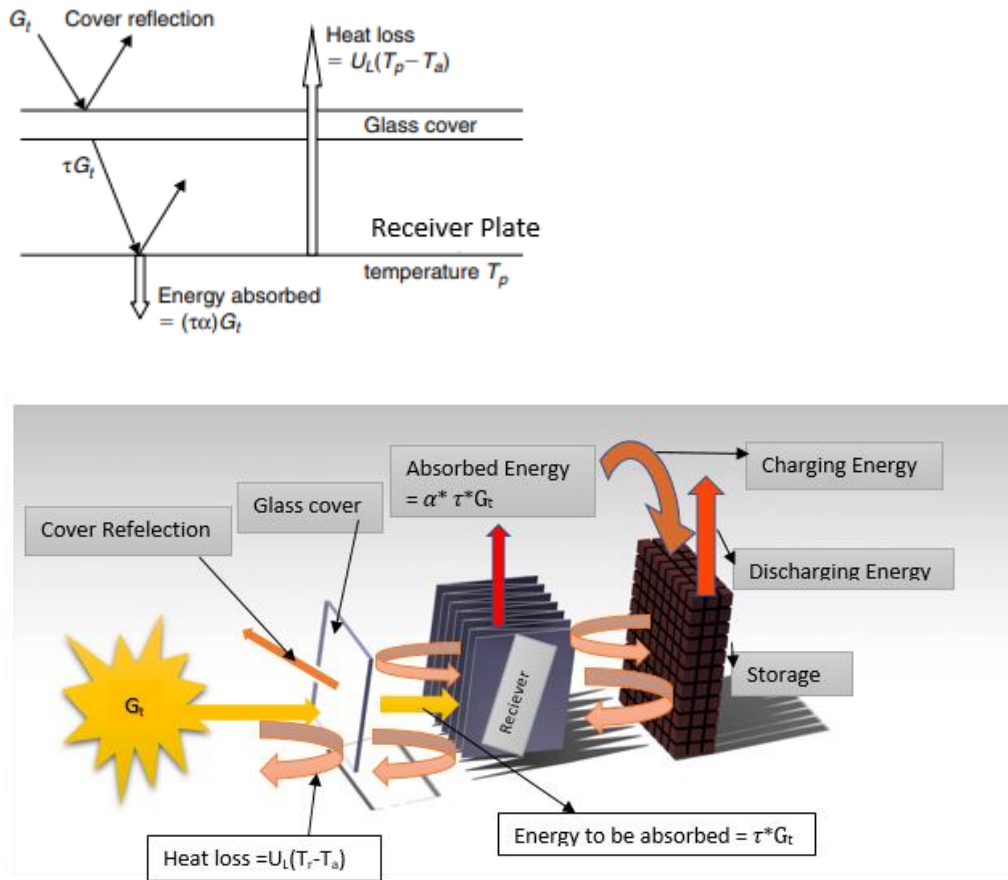


Fig. 55. Radiation input and heat loss from a receiver

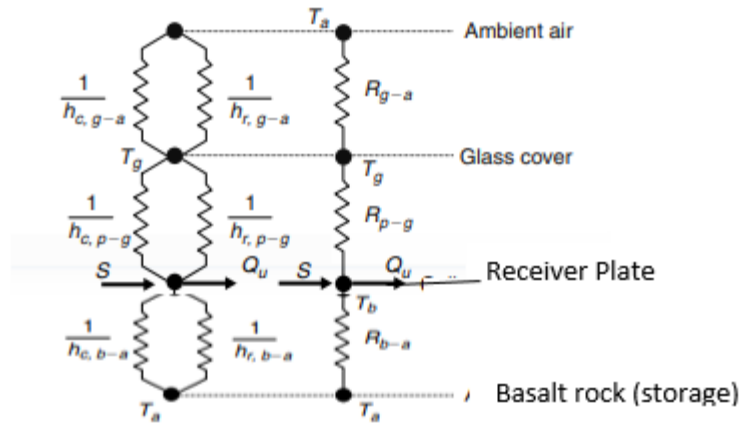


Fig. 56. Heat transfer network and Resistance network[7].

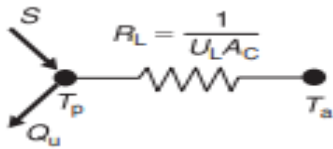


Fig. 57. Simple collector network

To model the collector shown in Fig. 42, a number of assumptions, which simplify the problem, need to be made.

1. All sides are insulated with the exception of the receiver which is directly facing the Scheffler-Reflector
2. No solar energy is absorbed by the cover.
3. Covers are opaque to infrared radiation.
4. Dust effects on the cover are negligible.
5. There is no shading of the receiver.
6. As the oven works by natural convection by buoyancy force convection inside the oven is not considered as loss, the convection losses from receiver to cover are negligible.

So that the energy losses from the collector can be written as:

$$Q_{abs} = Q_u + Q_L \text{-----}(28)$$

$$Q_u = A_c [G_t(\alpha \cdot \tau) - U_L(T_r - T_a)] = \dot{m} \cdot c_p [T_o - T_i] \text{.....}(29)$$

$$Q_{loss} = \frac{T_r - T_a}{R_L} = U_L \cdot A_c (T_r - T_a) \text{.....}(30)$$

Where,

U_L overall heat loss coefficient is based on cover Area A_c and Receiver A_r ($W/m^2 K$).

T_r Receiver temperature ($^{\circ}C$).

The overall heat loss coefficient is a complicated function of the collector construction and its operating conditions, given by the following expression:

$$U_L = h_w + h_r + h_c \text{.....}(31)$$

Where, h_w = heat loss coefficient by convection ($W/m^2 K$).

h_r = heat loss coefficient by Radiation ($W/m^2 K$).

h_c = heat loss coefficient by conduction ($W/m^2 K$).

In this case, U_L , is given by

$$U_L = \left[\frac{A_r}{(h_w + h_{r,c-a}) \cdot A_r} + \frac{1}{h_{r,r-c} + h_{c,r-c}} \right]^{-1} \text{.....}(32)$$

Where, $h_{r,c-a}$ radiation heat loss coefficient from cover to ambient (W/m^2-K).

A_c external area of glass cover (m^2).

$$h_{r,c-a} = 4 \cdot \sigma \cdot \epsilon_c \cdot T_r^3 \text{.....}(33)$$

For h_w (wind loss coefficient), the Nusselt number can be used.

For $0.1 \leq Re \leq 1000$,

$$Nu = 0.4 + 0.54(Re)^{0.52} \text{.....}(34)$$

For $1000 \leq Re \leq 50,000$

$$Nu = 0.3(Re)^{0.6} \text{.....}(35)$$

$$Re = \rho \cdot V \cdot L_c / \mu \text{.....}(36)$$

Where, ρ density of air at T_m ($T_m = (T_c + T_a)/2$)

V , velocity of wind

μ Viscosity of air at T_m

L_c length of cover

$$h_w = Nu * K / L_c \dots\dots\dots(37)$$

K conductivity of air at T_m

h_{r,r-c} radiation heat loss coefficient from receiver to cover,

h_{cr-c} convective heat transfer from receiver to the cover

$$h_{r,r-c} = \frac{\sigma * (T_r^2 + T_c^2) + T_r + T_c}{\frac{1}{\epsilon_r} + \frac{A_c}{A_r} (\frac{1}{\epsilon_c} - 1)} \dots\dots\dots(38)$$

To estimate the glass cover properties, the temperature of the glass cover, T_c, is required. This temperature is closer to the ambient temperature than the receiver temperature. The thermocouple was inserted at a distance of 15 cm between the holes.



Fig. 58. Oven prototype

3.5 Environmental Benefits of Solar Bread Baking System

Solar bread baking systems offer numerous environmental advantages over traditional ovens. By harnessing the power of the sun, these systems eliminate the need for fossil fuels or electricity, significantly reducing carbon emissions and environmental pollution.

Emission saving

The relation between the amount of firewood burnt and the amount of carbon dioxide produced was found to be 1 kg of firewood to 1.6 kg of CO₂[50]. On the other hand, to produce 1 kg of bread, 0.9 kg of firewood is required [51]. Using these two relations, the amount of carbon dioxide saved when using the Scheffler reflector was calculated.

Cost saving.

The Ethiopian electric utility provides information that the cost of 1 kWh of electric power is 1.625 birr [52]. Additionally, the production of 1 kg of bread requires 1.9 kWh of electric power [53]. Using these two pieces of information, a calculation was performed to determine the amount of cost savings that can be achieved by using the Scheffler reflector.

CHAPTER FOUR

4. Result and Discussion

4.1 Result of Ray trace Simulation for the optical performance of Scheffler reflectors.

The Ray tracing analysis of the proposed Scheffler reflectors for the baking system was done using Comsol Multipysics 5.6. 3D simulations were performed for each model to evaluate the optical performance of Scheffler reflectors, in order to study various types of deviation to the surface during manufacturing inaccuracies or imperfections. For simulation purposes, both the reflector and sun conditions can be adjusted to any day of the year and hour of the day. It is possible to estimate the size of the focus and the power delivered to a certain area. For analyses 5m² aperture area 1.4 m focal length and 1m*3m focus area are considered.

4.1.1 Reflector at equinox at noon at 0° polar axis

This simulation was performed at the equinox at noon when all conditions are ideal and light shines down vertically from above. The reflected light rays converge perfectly at the focus since the conditions are the same as when we designed the reflector, and there are no sources of deviation. The two figures below represent the outcomes of this simulation from two different points of view. The direction of the resultant reflected rays converging at the focus can be seen. This reflector has no defects and is fully adjusted for equinox conditions, so the focus is on a point with no scattering.

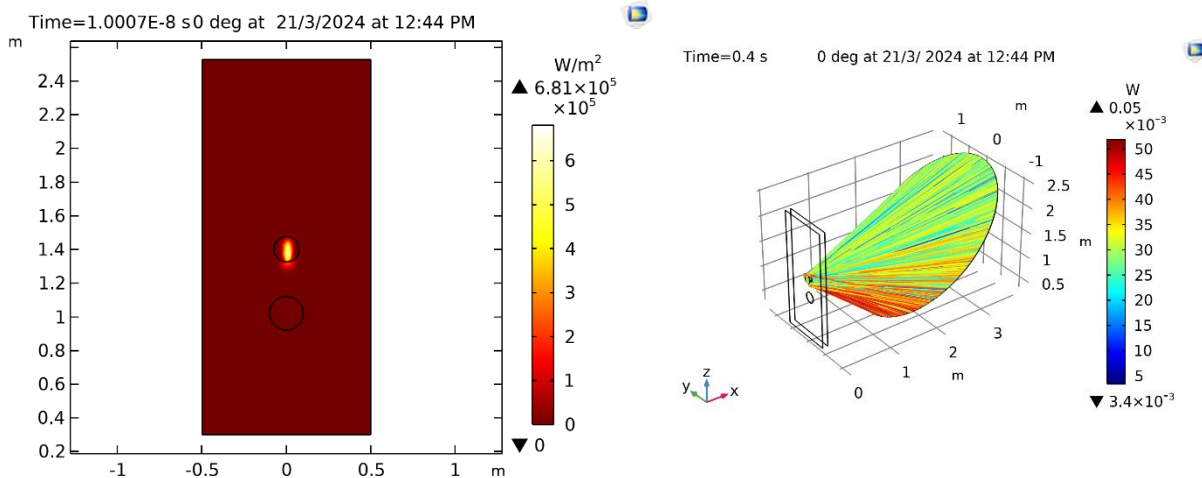


Fig. 59. The reflector at equinox at noon at 0° polar axis

4.1.2 Reflector at equinox at noon at 7.6° polar axis

This simulation was done at the equinox at noon at the place of installation. This is the area we mostly focused on because the Scheffler reflector was installed in this polar axis. The two figures below show the results of this simulation from two different views.

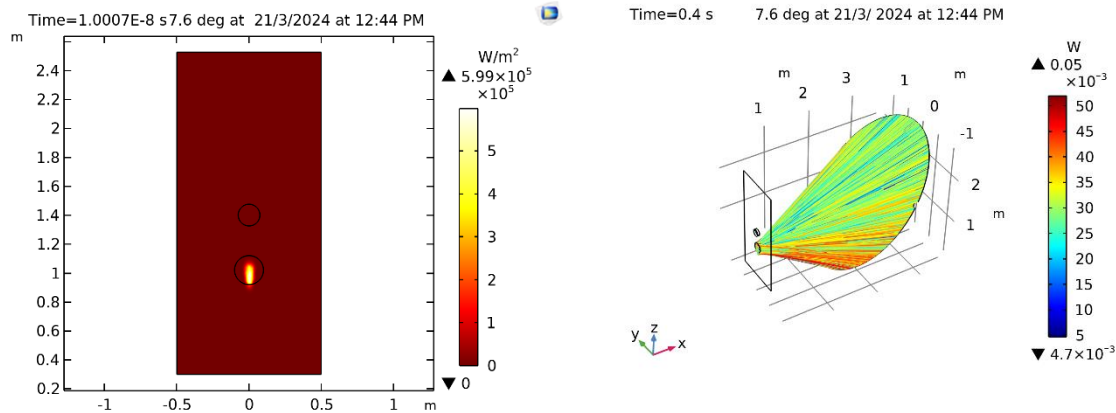


Fig. 60. The reflector at equinox at noon at 7.6° polar axis

The figure above shows the newly created focal point at the equinox at noon at 7.6° appearing at a distance of 0.4m below the focal point created at equinox at noon at 0° . This deviation is caused by the polar axis. The flux intensity in the receiver also decreased by $0.82 \times 10^5 \text{ W/m}^2$ from the equinox at noon at 0° . However, this is what we want, because our Scheffler reflector is installed in this polar axis.

4.1.3 The sun's position at summer solstice, reflector at equinox without adjusting and reflector at summer solstice with adjusting declination angle at noon at 7.6°

These simulations were done when the sun's position was at the summer solstice (21 June), with adjusting and without adjusting the declination angle of the Scheffler reflector at noon.

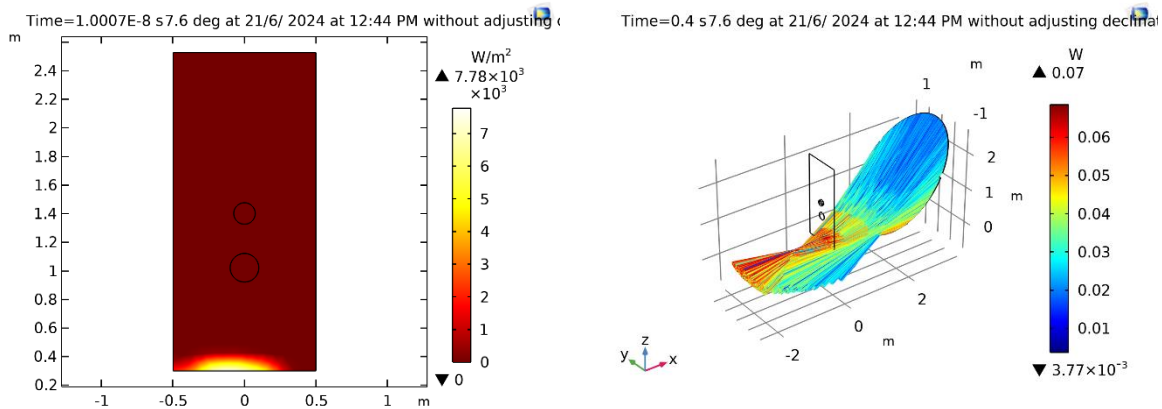


Fig. 61. The sun's position at summer solstice (21, June), reflector at equinox without adjusting to declination angle.

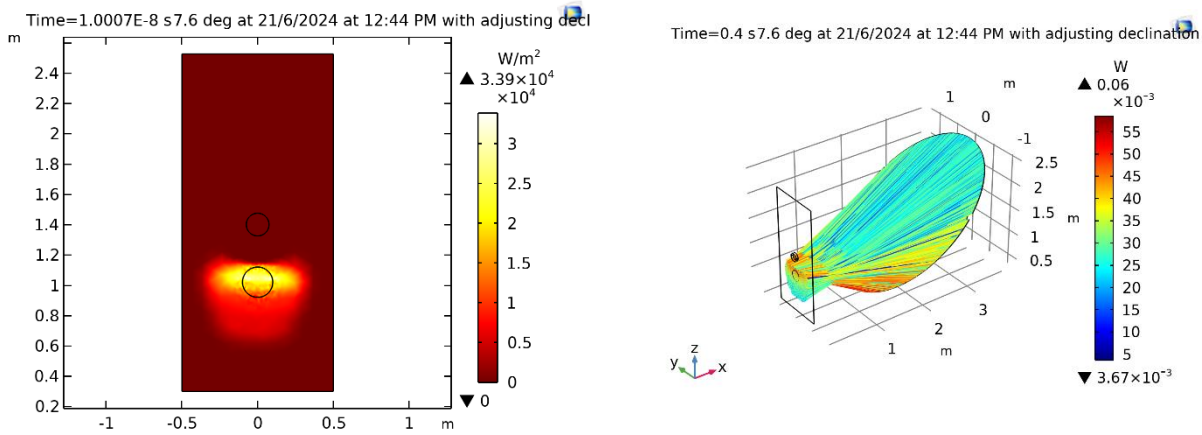


Fig. 62. The sun's position at the summer solstice (21 June), the reflector adjusted to declination angle (38.35°).

The figure above shows the newly created focal point is appearing at a distance of 0.8m below the focal point created at equinox at noon at 7.6°. This deviation is caused by the declination angle. The flux intensity in the receiver increased by $2.612 \times 10^4 \text{ W/m}^2$ when the adjusted to the declination angle. Even when adjusted to the declination angle, the flux intensity in the receiver is not the same as at equinox. This indicates that the resultant reflected rays are not converging at the focus perfectly. During the summer season, the sun's intensity is generally higher than at other times of the year. However, during the summer season, the concentration ratio is low. This means that the sun's rays are spread out over a larger area, which helps to decrease the temperature occurring in the receiver. By reducing the temperature in the receiver, the solar Scheffler reflector can operate at optimum temperature.

4.1.4 The sun's position at the Winter solstice and reflector at the equinox without adjusting and reflector at winter solstice with adjusting declination angle at noon at 7.6°.

These simulations were done when the sun's position was at the winter solstice (21, December) with adjusting and without adjusting the declination angle of the Scheffler reflector at noon.

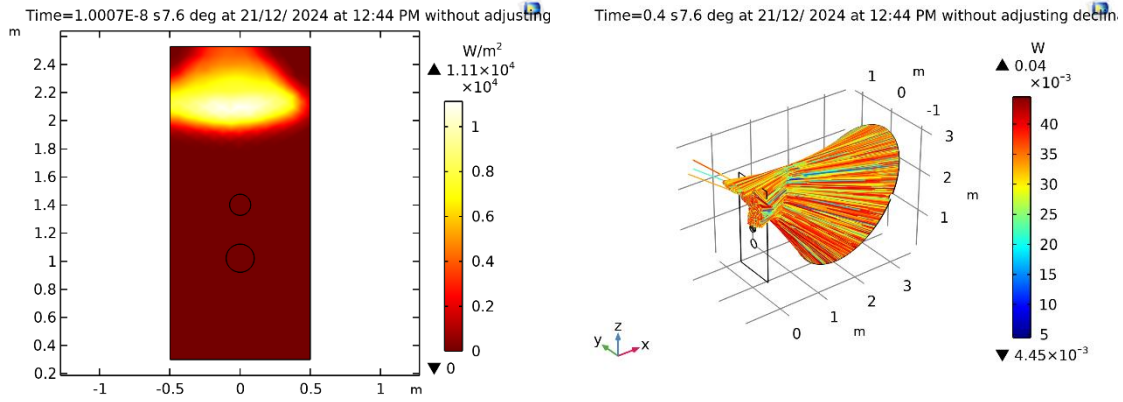


Fig. 63. The sun's position at winter solstice (21, December), reflector at equinox without adjusting to declination angle.

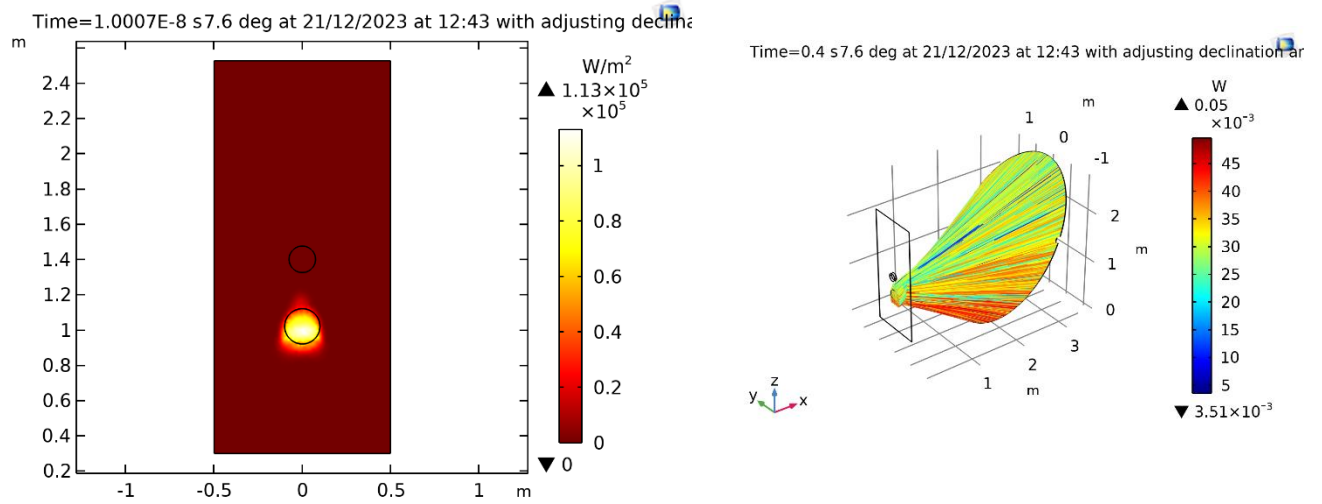


Fig. 64. The sun's position at the winter solstice (21 December), the reflector adjusted to declination angle (61.8°).



Fig. 65. Effects caused by not adjusting the declination angle

The figure above shows the newly created focal point appearing at a distance of 2m above the focal point created at the equinox at noon at 7.6° . This deviation is also caused by the declination angle. The flux intensity in the receiver increased by $1.019 \times 10^5 \text{ W/m}^2$ when adjusted to the declination angle. Even when adjusted to the declination angle the flux intensity in the receiver is not the same as at equinox. This indicates the resultant reflected rays are not converging at the focus perfectly. During the winter season, the sun's intensity is generally lower than at other times of the year. However, during the winter season, the concentration ratio is high. This means that the sun's rays are focusing sunlight on a small area, which helps to increase the temperature occurring in the receiver. By increasing the temperature in the receiver, the solar Scheffler reflector can operate at optimum temperature with lower intensity.

Seasonal Adjustments

The figure below shows how the change in declination closely resembles a sine curve, the greatest gradient occurs at the spring and fall equinoxes (green and red, respectively) and zero gradient at the summer and winter solstices (Yellow and blue, respectively).

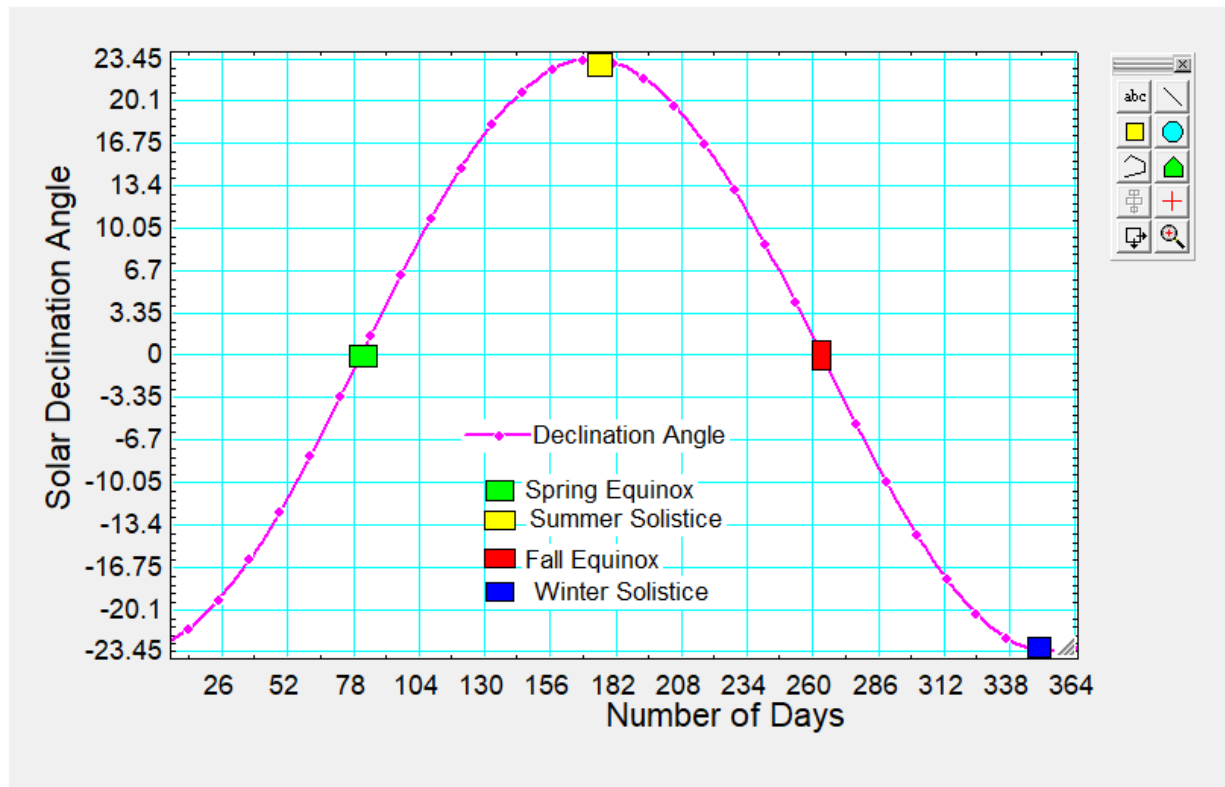


Fig. 66. Solar Declination vs Number of days

At the equinoxes, the gradient of the solar declination changes more rapidly, necessitating more frequent adjustments to the reflector. On the other hand, when at the solstices, the gradient changes more slowly, requiring less frequent adjustments to the reflector. It is therefore safe to assume that adjustments should be done every three days when close to the equinoxes and around once a week when close to the solstices. The adjustment is made with telescopic bars.

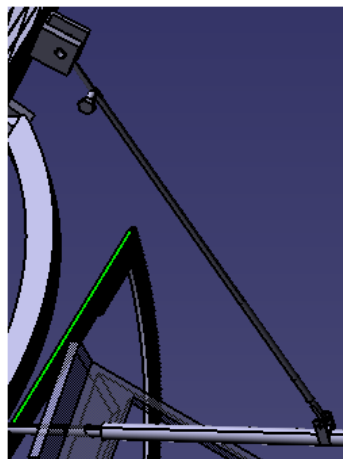


Fig. 67. Telescopic bar

4.1.5 Reflector rotates on polar axis with adjusting hour angle at -60° (8:47 AM) and without adjusting hour angle at 7.6° polar axis.

These simulations were done the sun's position at the equinox (21, march) at -60° hour angle without adjusting the hour angle and with adjusting of the Scheffler reflector.

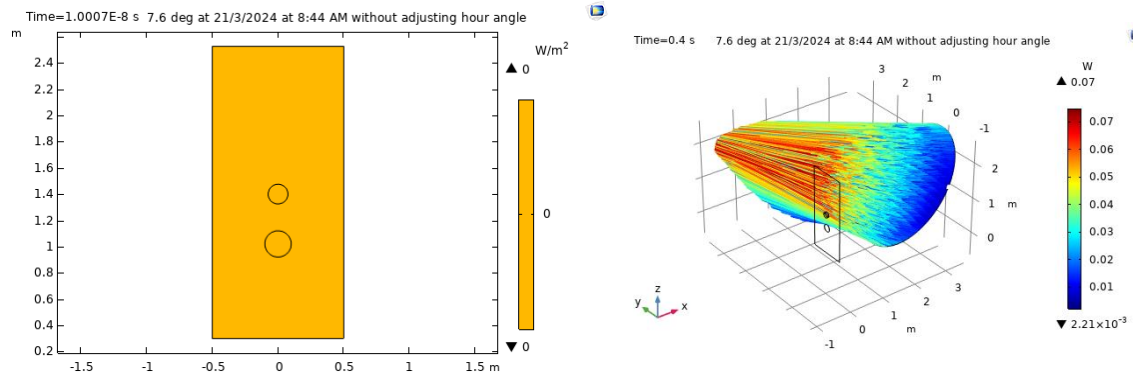


Fig. 68. The sun's position at equinox (21, march) at -60 hour angle without adjusting hour angle of Scheffler reflector to hour angle (reflector at noon).

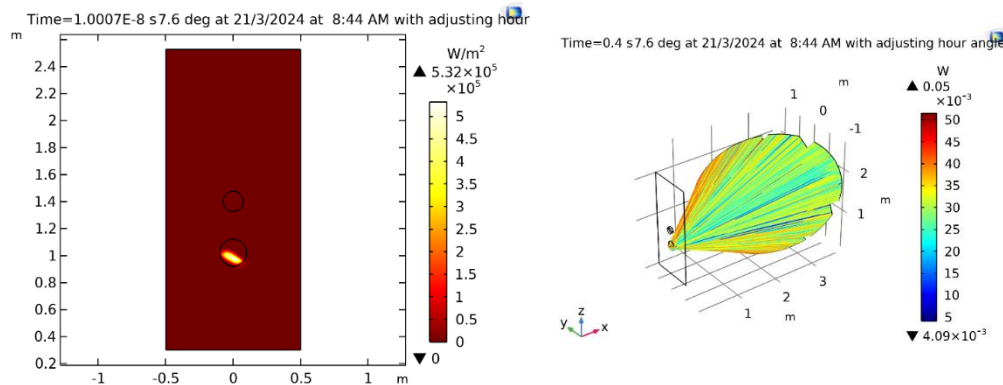


Fig. 69. The sun's position at equinox (21, march) at -60 hour angle with adjusting hour angle of Scheffler reflector to hour angle -60° (8:47 AM) .

The figure above shows that there is no focal point created in the receiver area without adjusting the hour angle. This deviation is caused by the hour angle. The flux intensity in the receiver also shows 0 W/m^2 . This shows there is no rays falling on the receiver at that time. But when adjusted to the hour angle, the flux intensity in the receiver is $5.32 \times 10^5 \text{ w/m}^2$, only $0.27 \times 10^5 \text{ w/m}^2$ differs from noon. This indicates the resultant reflected rays are converging at the focus when compared with others.

4.1.6 Reflector rotates on the polar axis without adjusting the hour angle at -15° (11:47 AM) at 7.6° polar axis.

This simulation was done when the sun's position at the equinox (21, march) at -15° hour angle without adjusting the hour angle of the reflector.

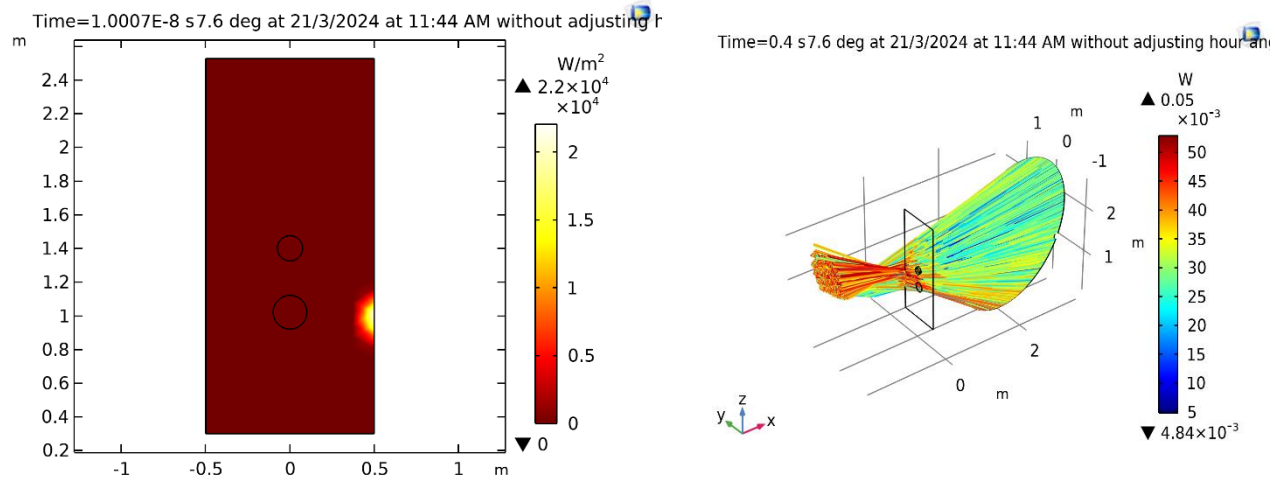


Fig. 70. The sun's position at the equinox (21, march) at -15° hour angle without adjusting the hour angle (reflector at noon).

The figure above shows the newly created focal point without adjusting the hour angle is appearing at a distance of 0.5m to the left of the focal point created at the equinox at noon at 7.6° . This deviation is also caused by hour angle. The flux intensity in the receiver also shows $2.2 \times 10^5 W/m^2$. This shows there are missed rays not falling on the receiver at that time. The adjustment is made with daily tracking mechanism.



Fig. 71. Daily tracking

4.1.7 Reflector rotates on 0° polar axis but installed in 7.6° location at equinox with adjusting hour angle at -60° (8:47 AM) at 7.6° polar axis.

This simulation was done when sun's position at equinox (21, March) at -60 hour angle, reflector rotates on 0° polar axis but installed in 7.6° location with adjusting hour angle.

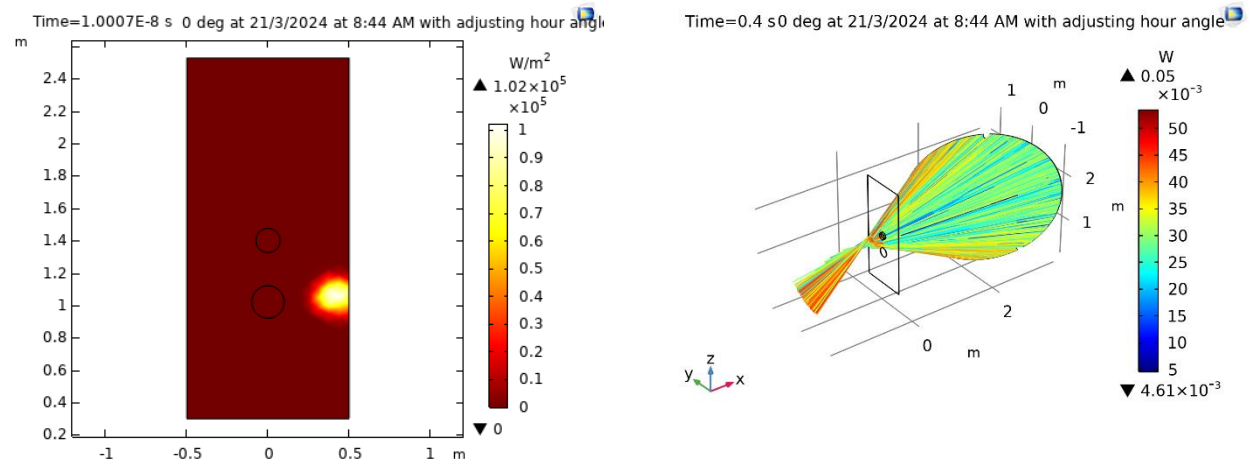


Fig. 72. The sun's position at equinox (21, march) at -60 hour angle (8:47 AM), reflector rotates on 0° polar axis but in installed in 7.6° location with adjusting hour angle.

The figure above shows the newly created focal point with adjusting the hour angle is appearing at a distance of 0.45m to the right of the focal point created at the equinox at noon at 7.6° . This deviation is also caused by axis of rotation. The flux intensity in the receiver also shows 1.02×10^5 W/m². This shows there are missed rays not falling on the receiver at that time. This is the problem that needs attention during installation. The adjustment is made with level measurement mechanism.



Fig. 73. Level measurement tool

4.1.8 Reflector deviates 10° from true north to east at equinox at noon at 7.6° polar axis

This simulation was done when sun's position at equinox (21, march) at noon reflector deviates 10° from true north to east.

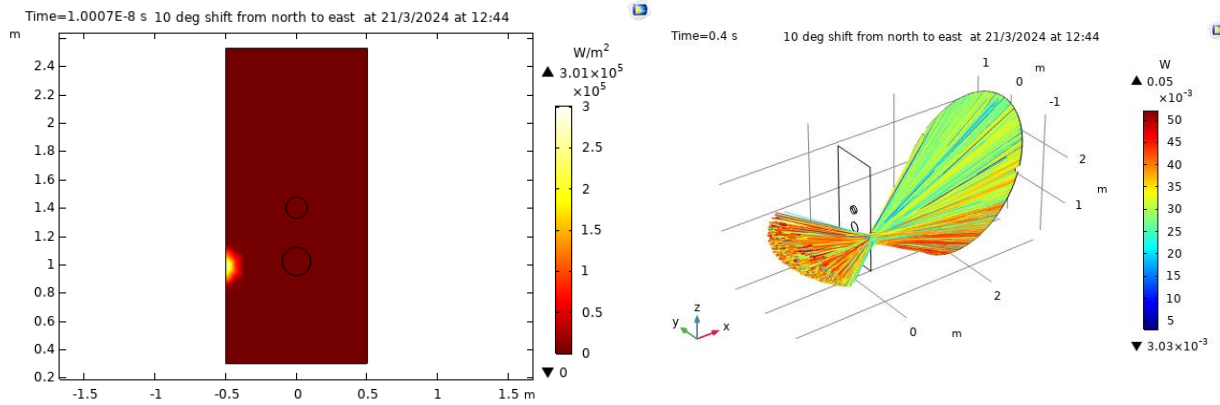


Fig. 74. The sun's position at equinox (21, march) at noon, reflector deviates 10° from true north to east.

The figure above shows the newly created focal point at noon appearing at a distance of 0.5m to the left of the focal point created at the equinox at noon at 7.6° . This deviation is caused by true north. The flux intensity in the receiver also shows $3.01 \times 10^5 \text{ W/m}^2$. This shows there are missed rays not falling on the receiver at that time. This is also the problem that needs attention during installation.

4.1.9 Reflector deviate 10° from true north at 11:45 am at 7.6° polar axis

This simulation was done when the sun's position at equinox (21, March) at 11:45 pm (15° before noon) reflector deviates 10° from true north to east.

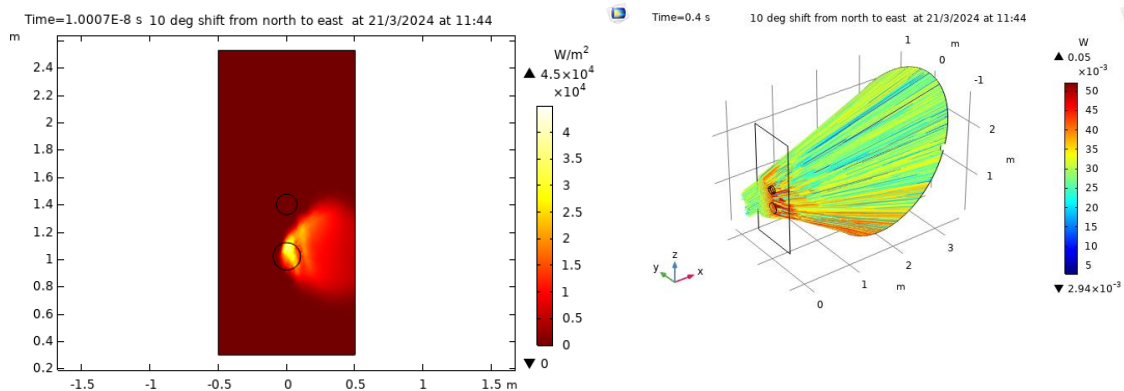


Fig. 75. The sun's position at equinox (21, march) at 11:45 pm (15° before noon) reflector deviates 10° from true north to east.

The figure above shows the newly created focal point at 15° before noon hour angle appearing at a distance of 0.2m to the left of the focal point created at the equinox at noon at 7.6° . The flux intensity in the receiver also shows $4.5 \times 10^4 \text{ W/m}^2$. This shows there are missed rays not falling on the receiver at that time. This is also the problem that needs attention during installation.

4.1.10 Reflector deviate 10° from true north at 1:45 pm at 7.6° polar axis

This simulation was done when the sun's position at equinox (21, march) at 1:45 pm (15° after noon) reflector deviates 10° from true north to east.

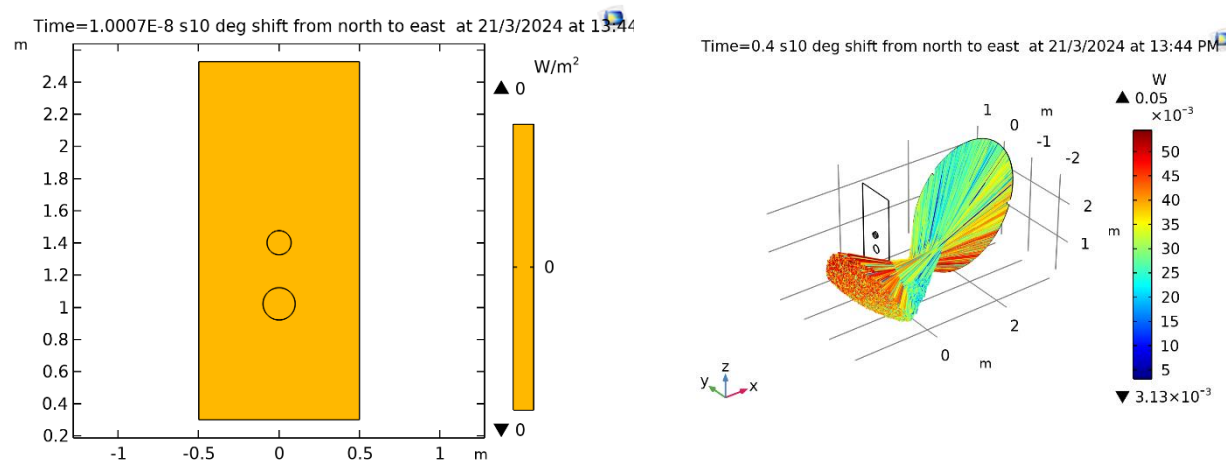


Fig. 76. The sun's position at equinox (21, march) at 1:45 pm (15° after noon) reflector deviates 10° from true north to east.

The figure above shows that there is no focal point created in the receiver area at 15° before noon hour angle. This deviation is also caused by true north. The flux intensity in the receiver 0 W/m^2 . This shows there is no rays fall on the receiver at that time. This is also the problem that needs attention during installation. The adjustment is made with digital compass.

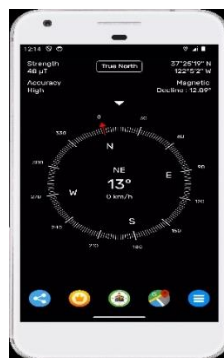


Fig. 77. Digital Compass

4.2 Result of ANSYS Fluent Simulation

4.2.1 Prediction of temperature and heat flux values on the receiver surface

The computational fluid dynamics analysis of the receiver was done using ANSYS Fluent 19.2. A series of 3D simulations were performed for models to evaluate the temperature and heat flux distribution on the surface of the receiver. The estimated result of concentrated solar radiation heat flux was used as input parameters.

The direction in which the heat flux is applied is the direction in which the receiver receives the concentration of solar energy. The heat flux was applied in the x direction.

These simulations were done at different heat fluxes. Because the heat flux varies with the aperture area. The aperture area varies with the declination angle. The aperture area varies from 5.851 m^2 to 3.941 m^2 at the winter solstice (Dec 21) to the summer solstice (June 21) respectively. Assume for simulation, 2500 W/m^2 and 3500 W/m^2 flux were taken.

Temperature Variation at 3500 W/m^2

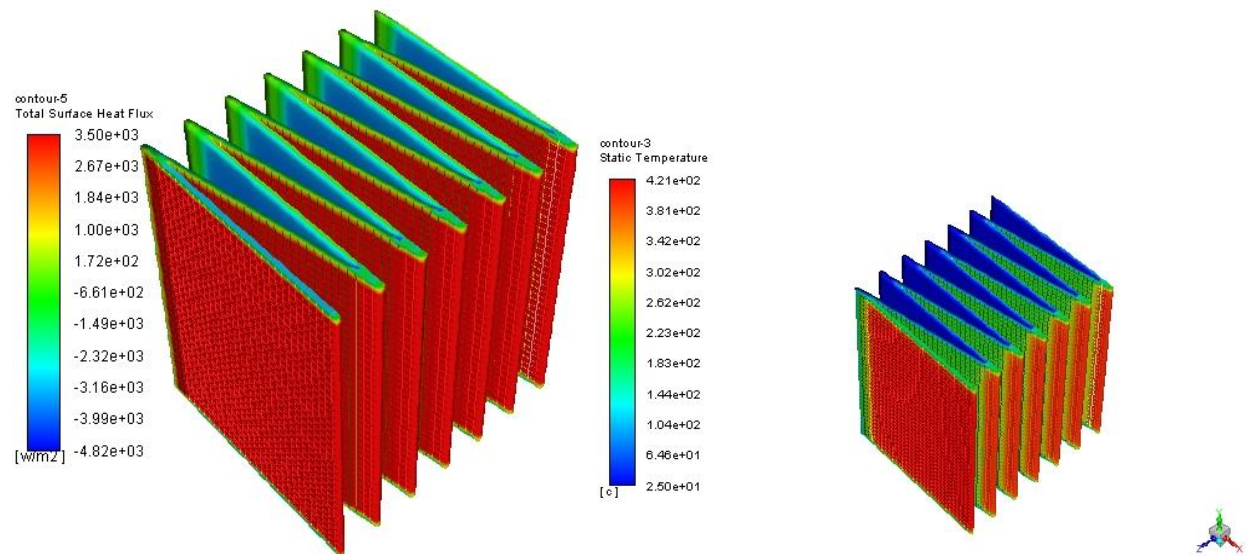


Fig. 78. Temperature Variation at 3500 W/m^2 heat flux.

The figure above shows that the heat flux and temperature are directly related. As the heat flux varied from -4.82 Kw/m^2 to $+3.5 \text{ Kw/m}^2$, the temperature also varied from $25 \text{ } ^\circ\text{C}$ to $421 \text{ } ^\circ\text{C}$.

Temperature Variation at 2500 w/m²

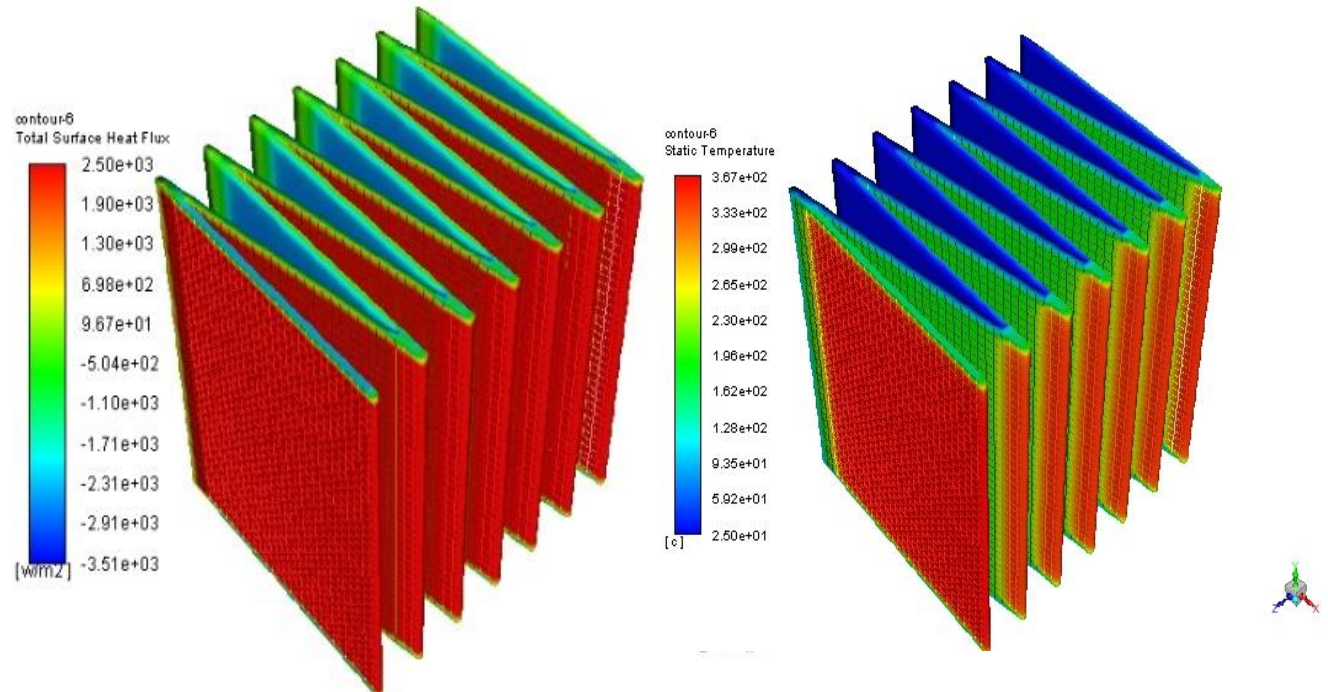


Fig. 79. Temperature Variation at 2500 w/m² heat flux.

As the heat flux varied from -3.51 Kw/m² to +2.5 Kw/m² the Temperature also varied from 25 °C to 361 °C.

From above the two figures, we conclude that as heat flux increase temperature also increases.

4.2.2 Prediction of temperature, Velocity, and Pressure distribution inside the oven

The computational fluid dynamics analysis of the oven for the baking system was done using ANSYS Fluent 19.2. 2D simulations were performed to study the behavior of the air domain in terms of temperature, velocity, and pressure distribution inside the oven. The simulation result of temperature from the surface receiver was used as input parameters. From the simulation result, 350 °C and 300 °C temperatures of surface receiver was taken for these simulations.

Temperature, velocity, and Pressure distribution inside the oven at 350 °C

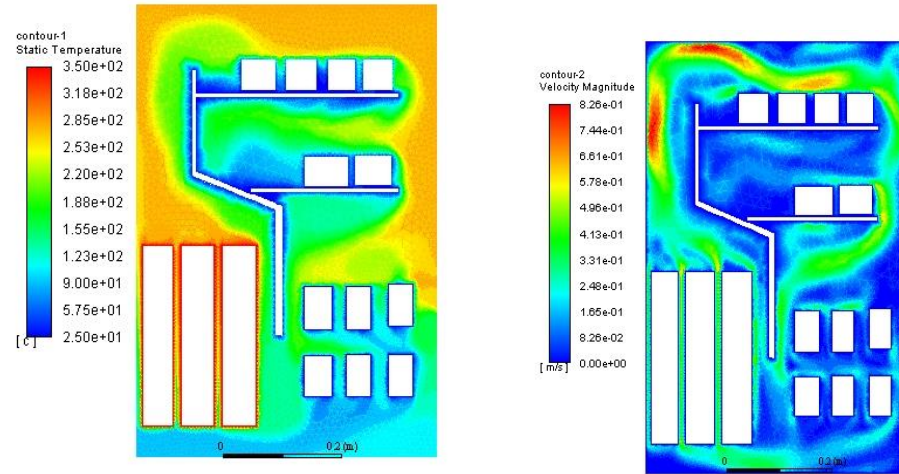


Fig. 80. Temperature and Velocity Distribution inside the oven when both trays are closed to the left

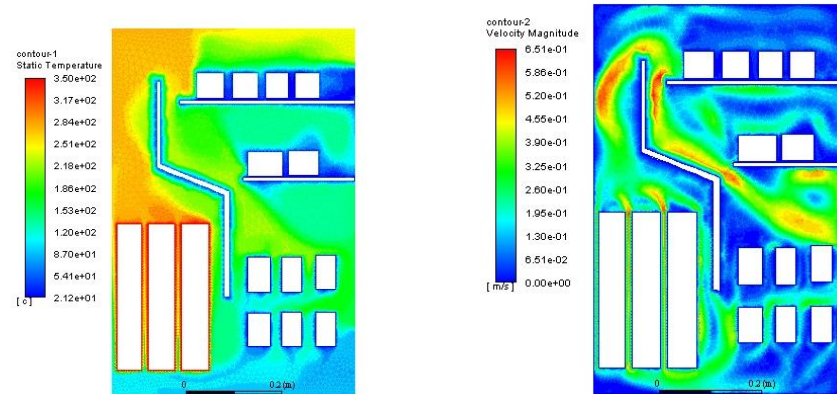


Fig. 81. Temperature and Velocity Distribution inside the oven when both trays are closed to the right

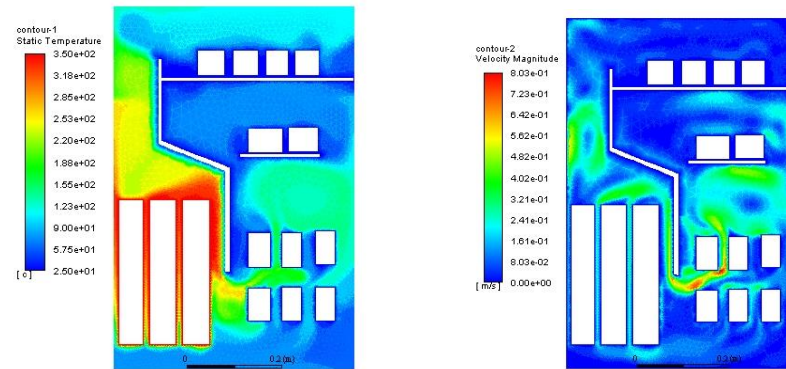


Fig. 82. Temperature and velocity Distribution inside the oven when top tray closed air flow

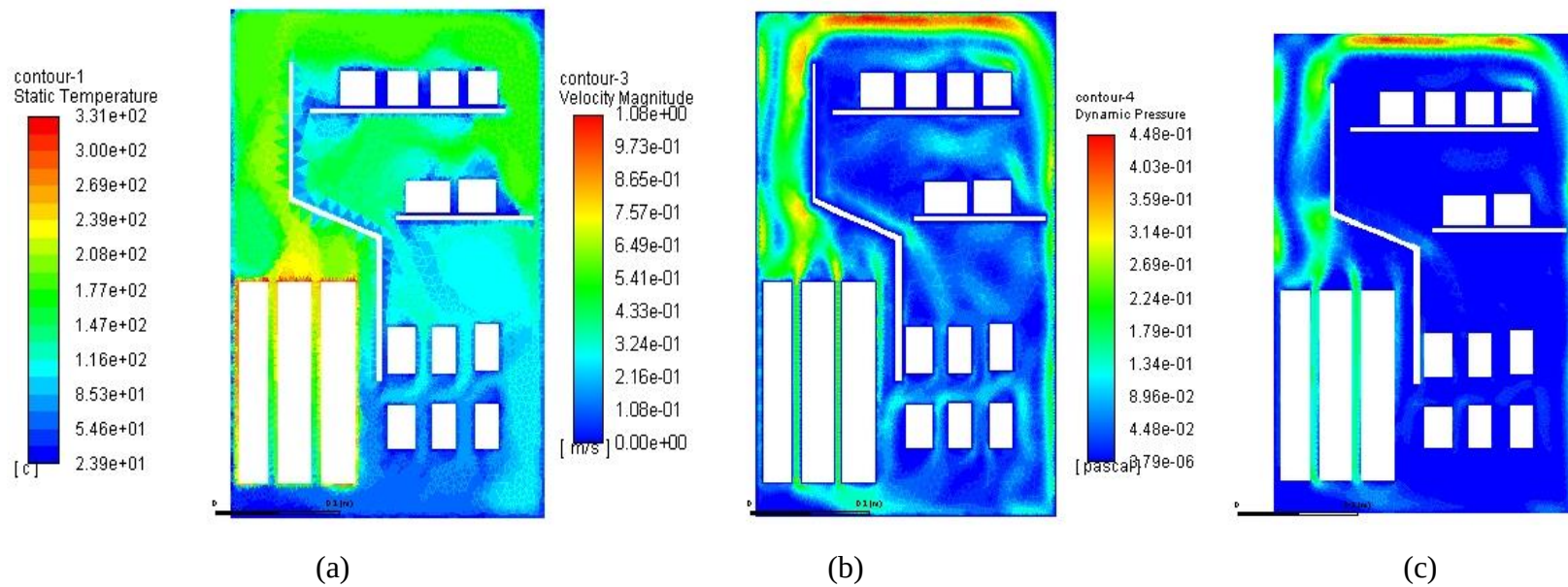


Fig. 83. Temperature, velocity, and Pressure distribution inside the oven when top tray is closed to left and bottom tray is closed to the right

In a solar oven, the temperature distribution can vary depending on factors such as the design of the oven, the position of the food inside. Fig 67, show that when both trays are closed to the left, the temperature on the right side increases, leading to uneven baking of the bread. Fig 68, show that when both trays are closed to the right, the temperature on the left side increases, leading to uneven baking of the bread. Fig 69, shows that when the top tray closed airflow, low temperature occurred at the top tray and also at the bottom tray because there was no air circulation inside the oven, leading to uneven baking of the bread.

Fig 70, shows that when the top tray is closed to the left and the bottom tray is closed to the right, the temperature distribution is fair. This ensures that the bread is baked evenly, taking into consideration the amount of dough placed on the tray.

The overall steady-state temperature profile of the oven when top tray is closed to the left and bottom tray is closed the right at receiver temperature 350 °C is shown in figure 70 a.

At the top of the receiver, closest to the heat source (Receiver) the temperature might reach the highest level, up to 300 °C. This area receives direct heat from convection and is not recommended for baking, because it burns bread. It must be decreased to recommended temperature for baking 175°C-225°C [54].

These is done by using chimney. The decreasing temperature has an effect on utilization efficiency. After passing through the chimney wall vertically, the hot air temperature decreases. At the top, the oven wall changes the direction of the hot air to horizontal.

At the top the first tray and the second tray the temperature distribution is the oven is within the range of between 177 °C to 239 °C. This is the temperature needed for baking purpose. After the bottom tray temperature decreased below 177 °C because heat is absorbed by first tray and second tray. This area receives the least amount of heat and is best for storage and reheating air.

The overall Velocity profile of the oven is shown in figure 70 b. The air is flowing in a clockwise direction inside the oven. At the top of the oven wall, the velocity might reach the highest level, up to 1.08 m/s. At the right side of the oven wall, the velocity might reach, up to 0.541 m/s. inside of the oven the velocity reach up to 0-0.216 m/s. This velocity distribution demonstrates that the air movement inside the solar oven is faster near the top of the oven and right side of the oven when hot air movement obstacles (The region where the air changes direction) by the top wall and right side respectively and minimum inside where there is no obstacle. From the distribution of velocity and temperature inside the oven, we can understand how much bread to put in the tray and where to move the tray. This means that we need to put more bread in the top tray than in the bottom tray to allow for an even baking. Additionally, it's important to note that the velocity on the right side is greater than the velocity on the left side. To ensure that the bread is baked evenly, we need to move the top tray towards the left and the bottom tray towards the right. However, it's important to ensure that the trays are not closed completely, as this could lead to uneven baking and result in burnt or undercooked bread.

The overall Dynamic Pressure profile of the oven is shown in Figure 70 c.

From the fig dynamic pressure increases where velocity increases. At the top of the oven wall, the Dynamic pressure might reach the highest level, up to 0.448 Pascal. At the right side of the oven wall, the Dynamic pressure might reach, up to 0.224 Pascal. Inside the oven, the Dynamic Pressure reaches up to 0-0.0896pascal.

Temperature, velocity, and Pressure distribution 300 °C.

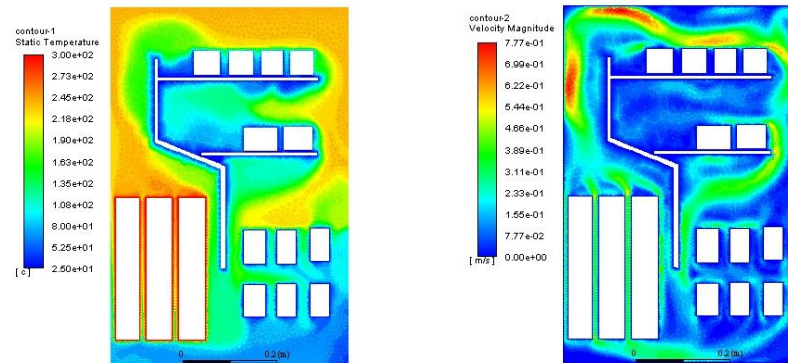


Fig. 84. Temperature and Velocity Distribution inside the oven when both trays are closed to the left

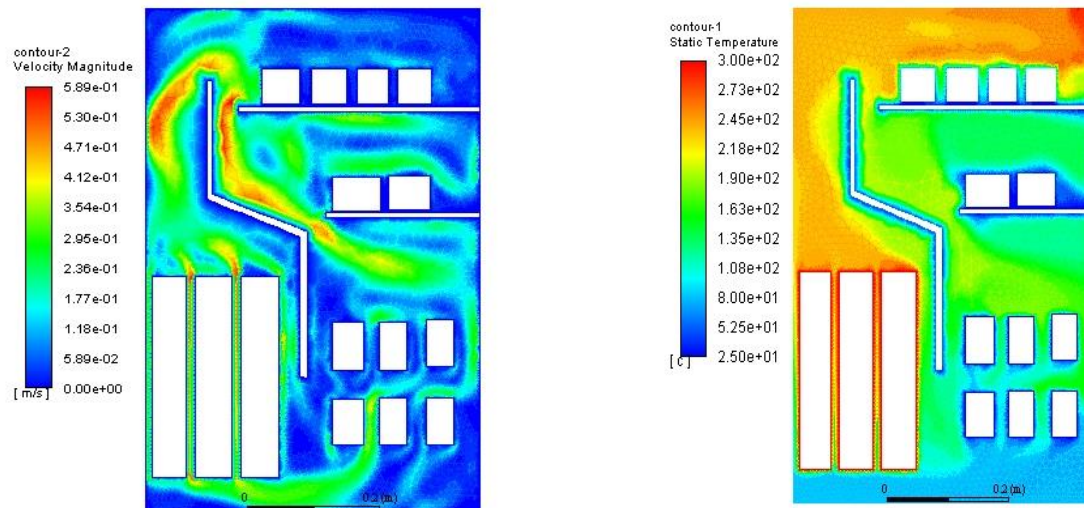


Fig. 85. Temperature and Velocity Distribution inside the oven when both Trays are closed to the right

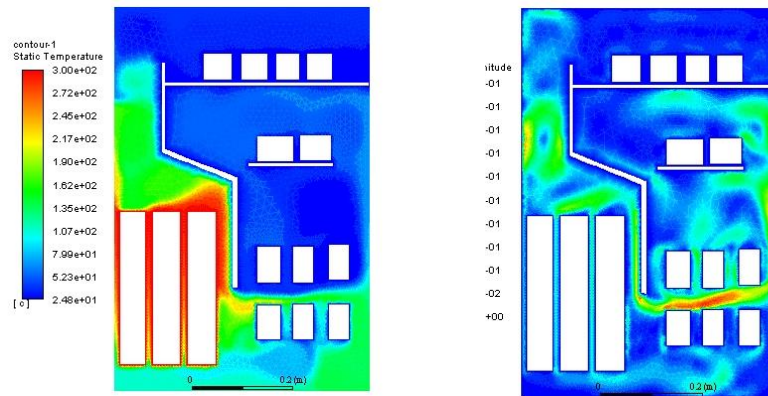


Fig. 86. Temperature and velocity Distribution inside the oven when top tray closed airflow

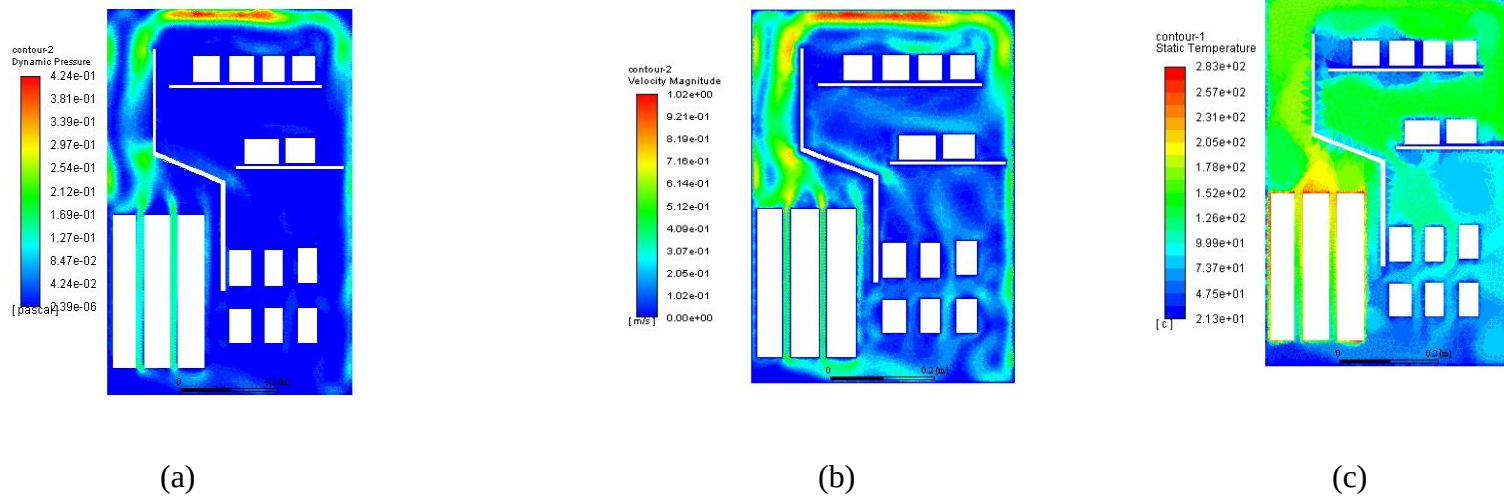


Fig. 87. Temperature, velocity, and Pressure distribution inside the oven when top tray is closed to the left and bottom tray is closed the right

In a solar oven, the temperature distribution can vary depending on factors such as the design of the oven, the position of the food inside. Fig 71, show that when both trays are closed to the left, the temperature on the right side increases, leading to uneven baking of the bread. Fig 72, show that when both trays are closed to the right, the temperature on the left side increases, leading to uneven baking of the bread. Fig 73, shows that when the top tray closed airflow, low temperature occurred at the top tray and also at the bottom tray because there was no air circulation inside the oven, leading to uneven baking of the bread.

Fig 74, shows that when the top tray is closed to the left and the bottom tray is closed to the right, the temperature distribution is fair. The overall steady-state temperature profile of the oven when top tray is closed to the left and bottom tray is closed the right at a receiver temperature of 300 °C is shown in figure 74 a.

At the top of the receiver, closest to the heat source (Receiver) the temperature might reach the highest level, up to 230 degrees Celsius. When compare within the temperature of the receiver 350 °C there is 70 °C difference. At the top the first tray and the second tray the temperature distribution is the oven is within the range of between 152 to 205 degrees Celsius. When compared within the temperature of the receiver 350 there is 25 °C to 34 °C difference. This indicates that the oven design should been changed, the heat flux should be increased or the area of reflector should be increased.

The overall Velocity profile of the oven is shown in figure 74 b. The air is flowing in a clockwise direction inside the oven

At the top of the oven wall, the velocity might reach the highest level, up to 1.02 m/s. At the right side of the oven wall, the velocity might reach, up to 0.512 m/s. Inside of the oven the velocity reach up to 0-0.205 m/s. This velocity distribution demonstrates that the air movement inside the solar oven is faster near the top of the oven and right side of the oven when hot air movement obstacles (The region where the air changes direction) by the top wall and right side respectively and minimum inside where there is no obstacle. When compared with the temperature of receiver at 350 °C velocity decreased by 0.04 m/s/. This indicates that as temperature of the receiver increase velocity is increased

The overall Dynamic Pressure profile of the oven is shown in figure 74 c.

From the fig dynamic pressure increases where velocity increases. At the top of the oven wall, the Dynamic pressure might reach the highest level, up to 0.424 Pascal. At the right side of the oven wall, the Dynamic pressure might reach, up to 0.212 Pascal. Inside the oven the Dynamic Pressure reach up to 0-0.047 Pascal.

Overall, the temperature, Velocity and Dynamic Pressure distribution inside a solar oven can be uneven, this indicates that the oven works by natural convection and is effective in storing energy and reheating air.

4.3 Experimental Results

4.3.1 Temperature distribution inside the oven without load and without storage

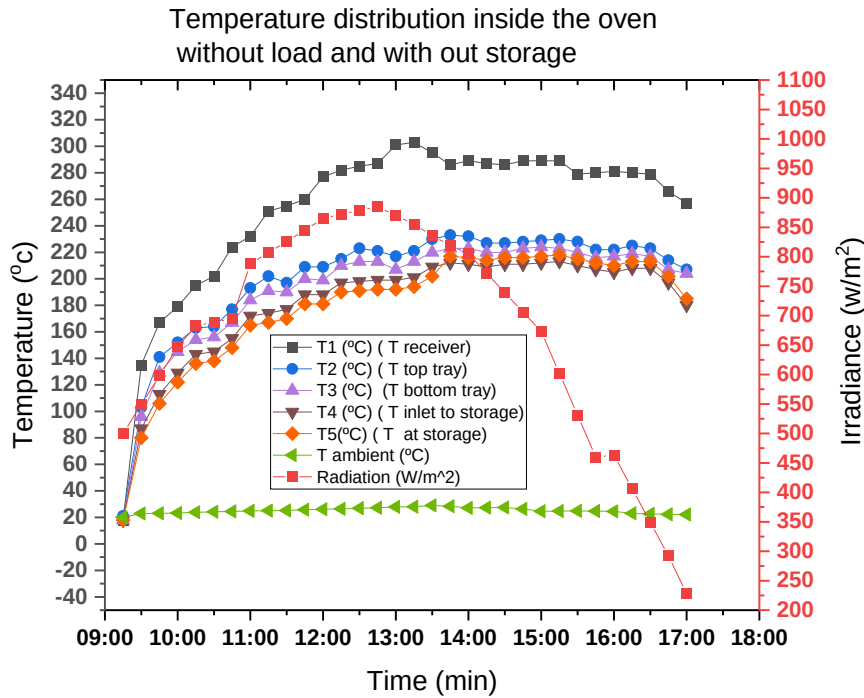


Fig. 88. Variation of temperature with time for no-load and no-storage test for 6 February 2024

The Graphs above show the temperature increases over the receiver surface with the increase in the solar irradiance. However, the maximum temperature obtained on the surface of the receiver is 303 °C at 1:15 pm at 855 W/m². After 1:15 PM temperature in the receiver decreased from 303 °C to 257 °C and solar irradiance decreased from 855 W/m² to 228 W/m². From this relation, as solar irradiance decreased rapidly, temperature decreased minimally. This indicates that the receiver absorbs the heat and the air inside the oven gets hot. The graph shows that the temperature at the top tray, bottom tray, at the inlet to the storage, and at the storage increases as the receiver temperature increases. However, after 1:15 pm, the temperature at the storage is greater than the temperature at the inlet to the storage. From this, we understand that there is radiation heat transfer between the back of the receiver and the storage area.

4.3.2 Temperature distribution inside the oven without load and with storage

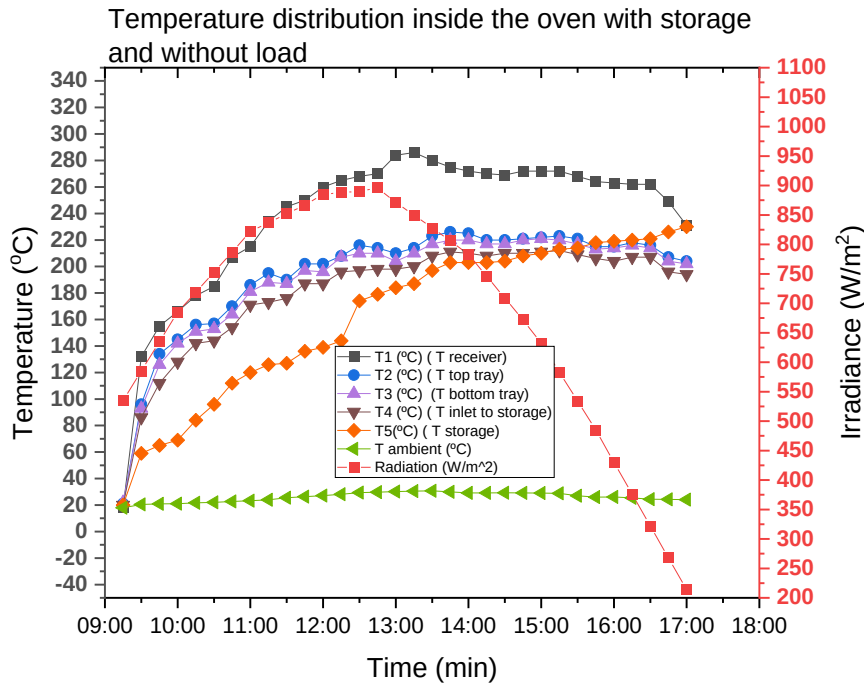


Fig. 89. Variation of temperature with time for no-load and storage test for 7 February 2024

The Graphs above shows the temperature increases over the receiver surface with the increase in the solar irradiance. However, the maximum temperature obtained on the surface of the receiver is 286 °C at 1:15 pm at 850 W/m². After 1:15 PM temperature in the receiver decreased from 286 °C to 231 °C and solar irradiance decreased from 850 W/m² to 214 W/m². From this relation, as solar irradiance decreased rapidly, temperature decreased minimally. This indicates that the receiver absorbs the heat and the air inside the oven gets hot. The graph shows that the temperature at the top tray, bottom tray, at the inlet to the storage, and at the storage increases as the receiver temperature increases. The graph shows that the temperature of storage is minimal up to 2:45 pm. After 2:45 pm, the storage temperature is greater than the inlet to the storage. At 9:45 pm, the storage temperature is greater than the top tray temperature. This indicates that there is radiation heat transfer between the back of the receiver and the storage area. At this stage, our storage is fully charged. From this, we understand that to fully charge our storage, at least 6:30 is needed.

4.3.3 Temperature distribution inside the oven with load and with storage

Temperature distribution inside the oven with storage and with load

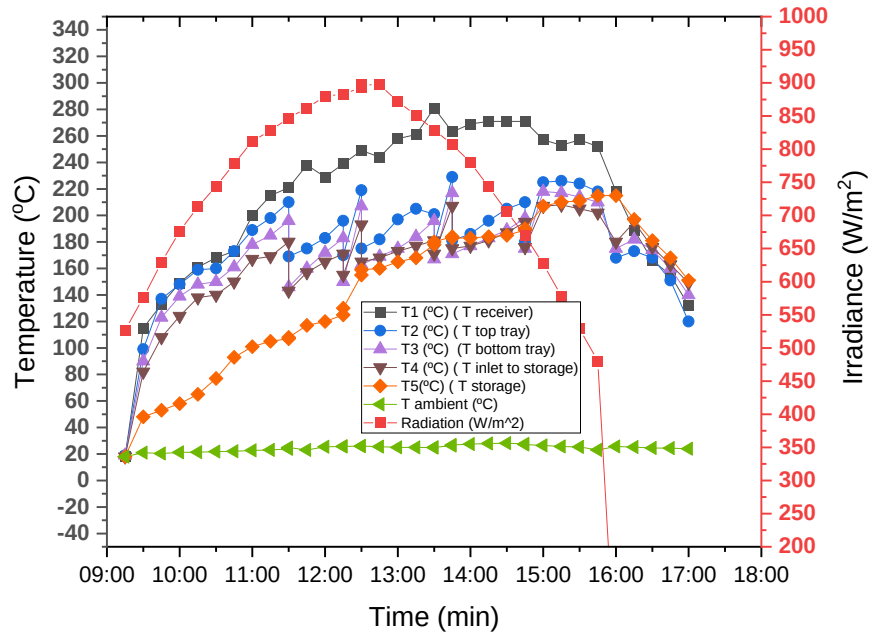


Fig. 90. Variation of temperature with time for load and storage test for 8 February 2024

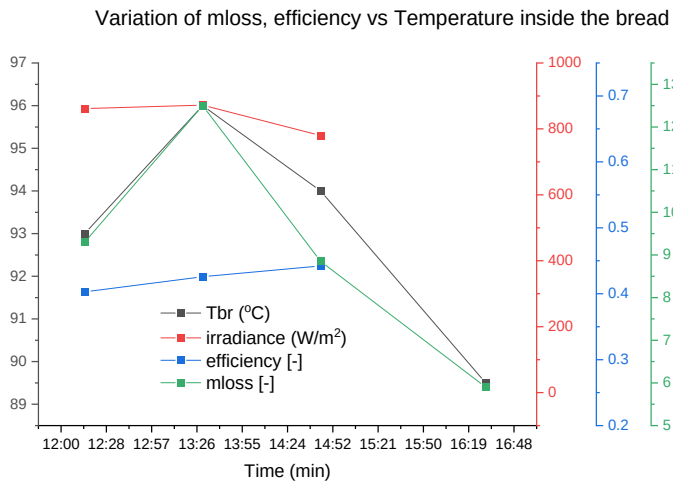


Fig. 91. Efficiency and moisture losses

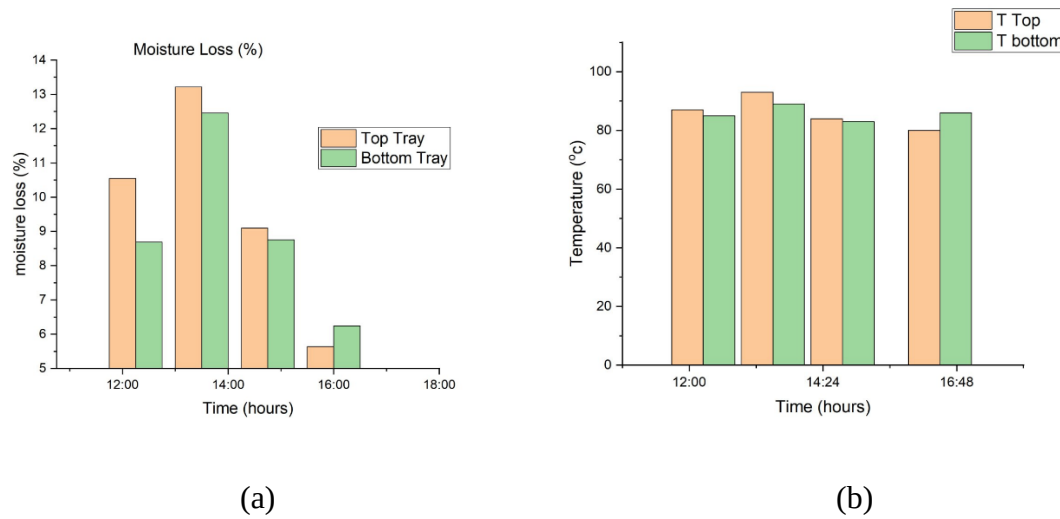


Fig. 92. Moisture losses (a), Temperature inside the Bread (b)

To perform a baking test, 2.6kg of dough was prepared. The dough was then placed on a flat table, divided into required weight, and kneaded into a ball shape. The kneaded dough was divided and weighed on digital mass balance into 100g. 1.6 kg (16 loaves) was placed on the top and 1kg (10 loaves) was placed on the bottom tray because, as we observed from the unloaded test, the temperature at the top is greater than at the bottom. A thermocouple was inserted in the dough center to record a temperature variation, and the dough was placed in the preheated oven. The dough is inserted into the oven when the temperature of the top and bottom oven reaches 210°C and 196°C respectively. When the oven reaches this stage, the oven is considered hot. The temperature reached this stage at 11:30 am. The figure above shows that the temperature of the top tray, bottom tray, and inlet to the storage dropped from 210 °C to 169 °C, 196 °C to 146 °C, and 180 °C to 143 °C respectively when opening the door and inserting dough. The impact was caused by the outward flow of hot air. These effects decrease the efficiency of the oven. At that time, the temperature inside the dough is 25 °C. After 15 minutes, the temperature of the top tray, bottom tray, inlet storage, temperature inside the dough at top tray and, temperature inside the dough at bottom tray increased from 169 °C to 175 °C ,146 °C to160 °C, 143 °C to 157 °C, 25 °C to 34 °C and 25 to 34 °C respectively. At the end of 45 minutes, the temperature of the top tray, bottom tray, inlet the storage, temperature inside the dough at top tray and, temperature inside the dough bottom reached 196 °C, 183 °C ,171 °C, 94 °C and 92 °C respectively. The moisture loss of the dough at top tray and, the dough of bottom tray

reached 10% and 8% respectively. The 2nd cycle was conducted at the 12:45pm. To perform a baking test for 2nd cycle, 2.6kg dough was prepared. The same criteria that were applied to the first one are also applied to this one. At the end of 45 minutes the temperature of top tray, bottom tray, inlet the storage, temperature inside the dough at top tray and, temperature inside the dough bottom reached 201 °C , 196 °C ,181 °C, 97 °C and 95 °C respectively. The moisture loss of the dough at top tray and, the dough of bottom tray reached 13% and 12% respectively. The 3rd cycle was conducted at the 2 pm. To perform a baking test for 3rd cycle, 2.6kg dough was prepared. The same criteria that were applied to the first one are also applied to this one. At the end of 45 minutes the temperature of top tray, bottom tray, inlet the storage, temperature inside the dough at top tray and, temperature inside the dough bottom reached 210 °C, 198 °C ,195 °C, 96 and 92 °C respectively. And the moisture loss of the dough at top tray and, the dough of bottom tray reached 9% and 8% respectively. The 4th cycle was conducted at the 4 pm. This test was performed to evaluate the storage performance by defocusing reflector from the receiver. To perform a baking test for 4th cycle, 2.6kg dough was prepared. 1.6 kg or 16 loaf was put on the bottom tray and 1kg 100g was put on the top tray because the heat flow direction was changed. The same criteria that were applied to the first one are also applied to this one. At that time (4pm) the receiver and storage temperature were reached 252 °C and 218 °C respectively. At the end of 45 minutes the temperature of top tray, bottom tray, inlet the storage, temperature inside the dough at top tray and, temperature inside the dough bottom reached 151 °C, 160 °C ,163 °C, 89 °C and 90 °C respectively. The moisture loss of the dough at top tray and, the dough of bottom tray reached 5.6% and 6.2% respectively. The average energy efficiency of the oven with three cycles are shown in fig 78. Results of the first cycle baking test started at 11:30 AM to 12:15 PM for a solar average irradiance of 862 W/m² , the oven raised the temperature of bread/dough from 25°C to 93 °C in 45 min and 9.3% moisture loss was occurred. From equation number 9, the efficiency of the oven for bread baking is 40 %. The second cycle baking test started at 12:45 PM to 1:30 PM for a solar average irradiance of 872 W/m², the oven raised the temperature of bread/dough from 25°C to 96 °C in 45 min and 12.5% moisture loss was occurred. From equation number 9, the efficiency of the oven for bread baking is 42.5 %. The 3rd cycle baking test started at 2:00 PM to 2:45 PM for a solar average irradiance of 780 W/m², the oven raised

the temperature of bread/dough from 25°C to 94 °C in 45 min and 8.8% moisture loss was occurred. From equation number 9, the efficiency of the oven for bread baking is 44 %. The test showed that the oven has an overall efficiency of 42% at an average solar irradiance of 838 W/m². The fourth cycle baking test started at 12:45 PM to 1:30 PM for testing storage performance by defocusing the reflector from receiver. The oven raised the temperature of bread/dough from 25°C to 89 °C in 45 min, 5.9% moisture loss was occurred, and the temperature of receiver and storage reached 155 and 168 respectively. This indicates that the storage can only last for one cycle, because temperatures below this are not recommended for baking.

Heat loss

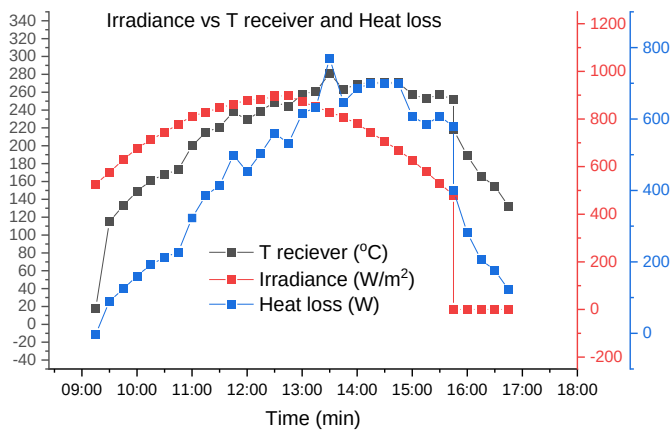
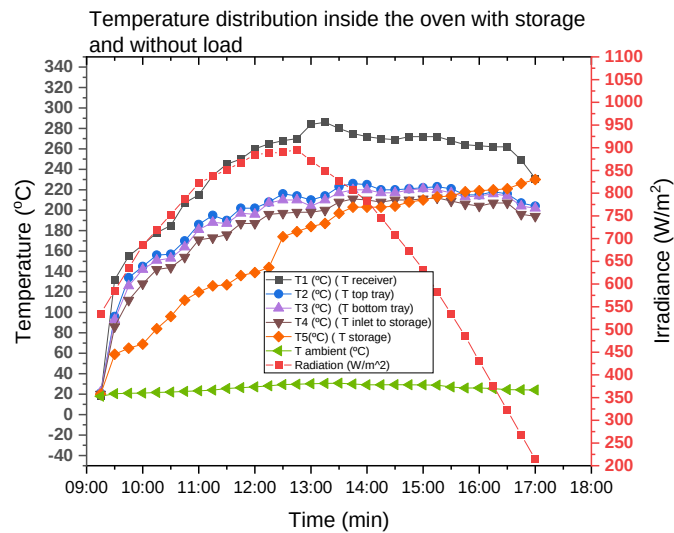
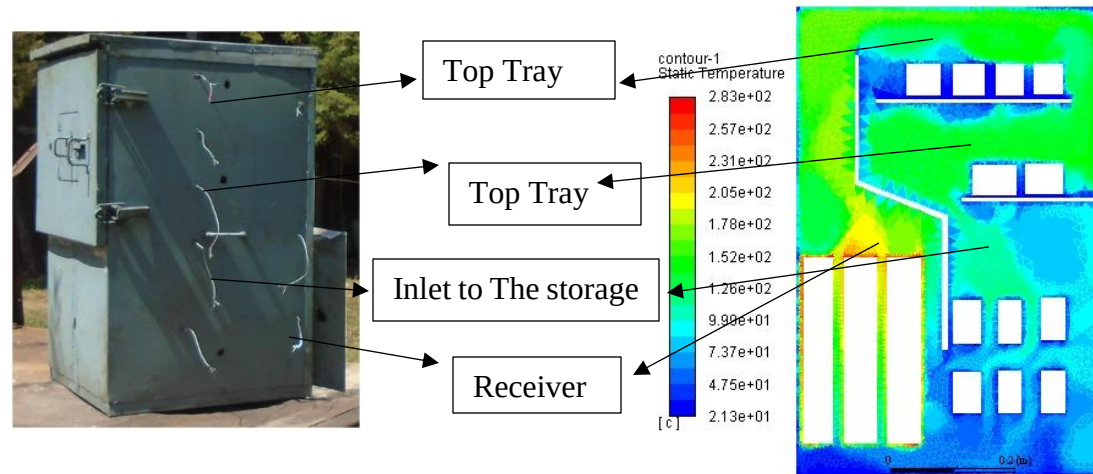


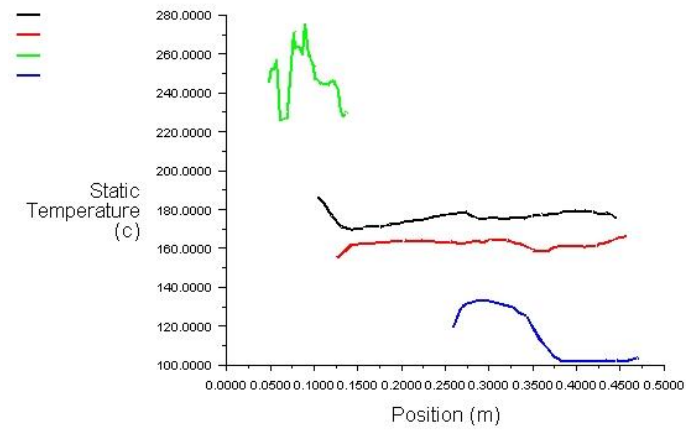
Fig. 93. Variation of Receiver temperature and Heat losses

The performance of concentrated solar power is influenced by direct normal irradiance. Due to variations in solar radiation, the amount of solar energy converted to the thermal heat of working fluid also varies throughout the day. Figure 80, indicates the variation of solar irradiance and temperature of the receiver with a time of the day. As seen, solar irradiance increases from sunrise to a maximum at noontime and then decreases till sunset. Figure 80, shows the variation in receiver temperature and heat loss during the baking test. The intensity of solar irradiance increases linearly from 9:15 Am to 12:45 Pm are observed. The maximum receiver temperature and heat loss are recorded during the test is 281 °C and 769 W at 1:30 pm respectively at solar Irradiance is 879 W/m².

Comparison of experimental data with simulation result



(a)



(b)

Fig. 94. Comparison experimental (a) and simulation result (b) of Temperature Distribution inside the oven

This comparison was conducted to compare the temperature distribution at the top tray, bottom tray, and inlet of the storage. The comparison was performed when the temperature of the receiver reached 286°C at 1:15 PM, when the irradiance reached 850 W/m². At 1:15 PM, the temperatures were recorded as follows: the top tray was at 214°C, the bottom tray at 210°C, and the inlet temperature was 200°C.

Based on the simulation results, the temperature in the top tray ranged from 170 °C to 190°C, the bottom tray ranged from 150°C to 170°C, and the storage ranged from 130°C to 100°C. Both results indicated that the temperature at the top tray was not the same as the bottom tray.

Emission saving

According to the design specifications, the production of bread is estimated to be 10kg per day. This means that in a year, assuming a baking time of 245days (clear days), the total production of bread would be 2450kg. To produce this amount of bread, $2450 \times 0.9\text{kg}$ or 2205kg of firewood is required. However, burning this amount of firewood results in the production of $2205 \times 1.6\text{kg}$ or 3528kg of CO₂ emissions. This is a significant amount of greenhouse gas emissions, which contribute to climate change and environmental degradation. By utilizing the Scheffler reflector, it is possible to save up to 3.5 tons of CO₂ emissions annually. This is a significant reduction in greenhouse gas emissions, and it helps to protect the environment and mitigate the effects of climate change.

Cost saving

Based on the design, the production of bread is estimated to be 10kg per day. Assuming that the baking process takes 245 days per year (clear days), the total annual bread production is estimated to be 2450 kg. This estimation is crucial in determining the amount of electricity needed to produce the bread. To produce 2450 kg of bread, the bakery requires $2450 \times 1.9\text{kwh}$ of electricity, which amounts to 4655kwh of electric energy consumed. This energy consumption has a significant impact on the bakery's electricity costs, which can be reduced through the use of Scheffler reflector technology. 1kwh of the electric energy cost in Ethiopia is 1.625 Ethiopian birr. The cost savings from using a Scheffler reflector are estimated to be 7564 birr. This technology utilizes a Scheffler reflector to concentrate solar energy onto the bread oven, reducing the electricity required for baking. The cost savings from using this technology can be a significant boost to the bakery's profitability.

Conclusion

The demand for energy for baking is increasing with a growing population worldwide. However, traditional fuels have negative environmental impacts. Therefore, there is a need for innovative techniques to meet energy needs and improve living standards without further damaging the environment. This study focused on a solar thermal baking system incorporating a Scheffler reflector and basalt rocks as an energy storage medium. The study involved designing, manufacturing, and evaluating the performance of the system through experimentation and simulation. The results showed that the Scheffler reflector and oven design were successful in efficient and effective bread baking. Factors such as true north, polar axis, declination angle, and hour angle were found to significantly impact the performance of the reflector. The findings also highlighted the importance of proper bread placement in the oven to ensure even baking. The system has an overall efficiency of 42% and has the potential to save significant amounts of CO₂ emissions and electricity costs. The Scheffler reflector proved to be an environmentally friendly and cost-effective solution for reducing carbon emissions and energy consumption in baking.

Recommendations

During Scheffler reflector installation, it is essential to consider factors like true north, polar axis, declination angle, and hour angle as they significantly impact system performance. Proper alignment ensures optimal efficiency.

Further research should focus on scalability, adapting to various baking capacities, and optimizing the solar thermal baking system. Additionally, studies on improving solar thermal baking performance with different materials and mass manufacturing designs are crucial.

Developing training programs for bakers and communities on system operation and maintenance would enhance efficiency and promote sustainability.

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Appendix



Photo shows during manufacturing Central bar





Photo shows during assembling



Photo shows during checking polar axis



Photo shows zig-zag receiver



Photo shows basalt rock used for storage



Photo shows temperature obtained in flat iron during testing without oven

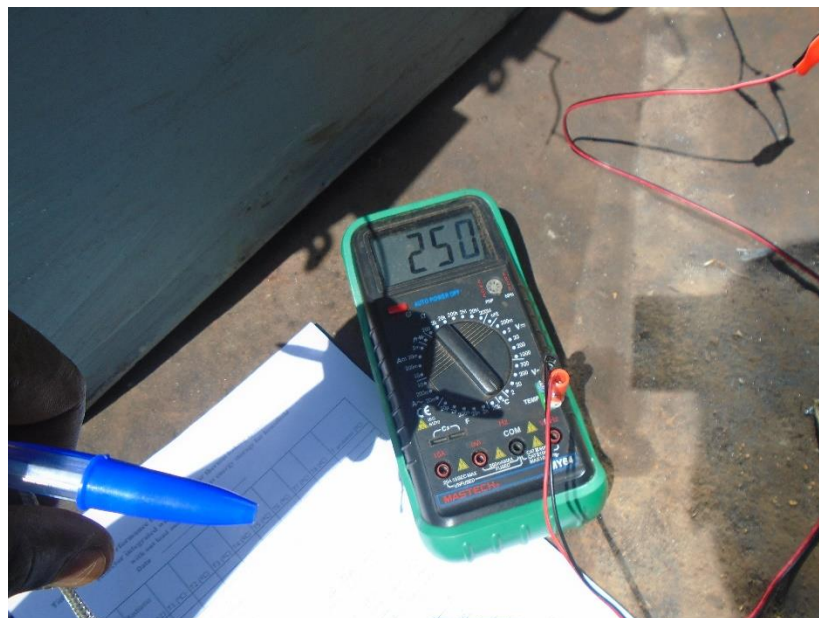




Photo shows during baking test