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DEPARTMENT OF ENERGY CENTER

DESIGN AND MANUFACTURE OF LABORATORY MODEL FOR SOLAR POWERED INJERA BAKING OVEN

By

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ABSTRACT

Currently millions of people around the world rely on wood as a source of fuel for cooking. Although this situation not only pertains to impoverished rural communities, it is within these communities and in industrializing countries that mostly occurs. The burning of wood in open fires is causing a number of health problems but is also deteriorating for the rural household economy as well as for the local and global environment.

Injera is the stable bread in Ethiopia; it is perhaps consumed by almost all Ethiopian people on a daily basis. Biomass fuel can consist of either wood, crop residues, or dried animal dung. Using these energy sources can result in negative effects such as deforestation, environmental pollution, and health problem. Women and children are the main groups exposed to the indoor smoke produced while cooking.

The overall aim of this thesis project is to use solar powered injera baking oven, so as to avoid the problems that are caused due to burning of fossil fuels and to assure the environmental sustainability. In this project a laboratory model for solar powered injera baking oven system is designed and manufactured; the laboratory model consists of the oil storage and heating tank, the piping and pumping system, the baking pan assembly, and supporting frame and legs as its main components.

The system uses electrical heater to heat the heat transfer oil (shell thermia B) to the required temperature, and then the heated oil is pumped to the baking pan assembly to heat the pan surface and re-circulates in the system using an electrical driven pump. To protect heat loss; ash insulation system for the heat transfer oil gallery, the oil storage and heating tank and fiber glass insulation for the piping lines are used.

During the experimental test a temperature of about 215^{o}C on the baking pan surface is achieved and injera is baked on this surface. To increase the heating up time of the pan surface, the pan supporting plate should be a high thermal conductivity material.

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NOMENCLATURE

$A-$ Area [m^2]

c - Heat capacity of liquid [kJ/kg.k]

D - Diameter [m]

E- Energy [kJ]

g - Acceleration due to gravity [m/s2]

h - Convection heat transfer coefficient

h_{fg} - Heat of vaporization [kJ/kg]

HCE-Heat Collection Element

k - Thermal conductivity [W/m.k]

L - Length [m]

m- Mass [kg]

Nu - Nusselt number

PTSC-parabolic trough solar collector

P – Power [kW]

$\Pr -$ Prandlt number

Q- Heat flow [watt]

$Ra -$ Rayleigh number

Re - Reynolds number

RHS- Rockwall hardness steel

T - Temperature [^oC or ^oK]

T ∞ - Temperature of the environment

ΔT – Temperature gradient (change in temperature) [^oC or ^oK]

$\beta -$ Volumetric thermal expansion coefficient

$\sigma -$ Stefan-Boltzman constant [$W/m^2.K^4$]

$\varepsilon -$ Emissivity of a material

$\eta -$ Efficiency

$\alpha -$ Absorbance

*All measurements indicated on the drawings are in mm.

CHAPTER ONE

INTRODUCTION

1. 1 Introduction

Food is a universal need, which uses energy for its preparation. Gathering, preparing, and consuming food is subject to many cultural considerations. It is known that, the energy requirements in developing countries are largely met from forests and agricultural waste, and the options for cooking food is limited, Most of the time they rely on wood as a primary fuel and cooking on simple open fires.

Technical advances in energy efficiencies are critical for developing countries like Ethiopia, whose populations depend primarily on biomass fuels such as wood, charcoal and agricultural residues. Overuse of these fuels deplete resources, degrade local environments, spent time needed to collect fuel, and creates indoor air pollution that cause health problems.

Injera is the staple flatbread and is a necessity food item in Ethiopia. When one thinks of cooking in Ethiopia, one thinks immediately of Injera bread, in which its preparation usually takes two to three days and in most of the cases the crop teff is used for injera.

Because injera is such a large part of our diet, any alternative cooking method must be taken into account. One alternative is to use kerosene, or liquefied petroleum gas; but these are expensive and transporting them to remote areas is also difficult. So to reduce and thus to avoid the addressed problems, it is important to address the escalating need for a new fuel source by utilizing energy from the sun.

The sun is an inexhaustible source of energy; the amount of solar energy an area receives depends on the time of day, the season of the year, the cloudiness of the sky, and how closes the area to the earth’s equator. Ethiopia is close enough to the equator to get plenty of solar power; in a fourth rank in the world in terms of solar cooking potential, with thirteen months of sunshine [19, 16]. So it is important to use the technology of using solar energy for injera baking system in order to solve the addressed problems and to have sustainable environment.

1.2 Background of the study

In Ethiopia injera baking requires a bulk of domestic energy demand. In most households this injera baking system is carried out using an open fire /three stone/ baking system which is inefficient and wasteful technique.

To bake injera, the heat supplied to the baking pan either comes from burning fuel wood, dung or agricultural residue in biomass cookers using the traditional mittad stoves for centuries. The cultural attachments of the people with the mittad and its product of baking injera is so strong that people are not expected to get rid of it in the short term.

Ethiopian energy consumption is mainly satisfied by biomass fuels. One serious disadvantage is that the mittad consumes considerable quantities of firewood, estimated to be at least 50% of the biomass energy consumption per household per year [13]. Due to the growing scarcity of wood fuel, people are forced to travel farther from their homes to Search for it; this burden of subsistence is carried almost entirely by women spending more times in fuel wood collection.

Recent field tests have shown that improvements in air control and firebox design is currently the most effective and socially acceptable stove design modification. However, their usefulness with regard to saving the resource base from depletion is limited in the face of growing population and urbanization because they use fire wood as the source of energy.

The use of solar energy for the purpose of cooking food presents a viable alternative to the use of fuel wood, kerosene, and other fuels traditionally used in developing countries. While certainly solar cookers cannot entirely halt the use of combustible fuels, it can be shown that properly applied solar cooking can be used as an effective mitigation tool with regards to global climate change, deforestation, and economic debasement of the world’s poorest people.

Solar collectors convert solar radiation in to heat and they rely on an energy source that is free, abundant and renewable, but the reason that they are not widely spread as needed is because of the draw back that cooking must occur when, and where the sun is shining.

Thus to reduce the problems associated with using biomass, kerosene, or liquefied petroleum gas, in this thesis solar powered injera baking oven, which uses oil as heat transfer fluid is expected to include the following benefits:

- Reduced concentrations of smoke and indoor air pollution
- Saves money and time in acquiring fuel
- Reduced biomass use, ability to use animal dung as fertilizer instead of for fuel
- Less pressure on forest and other non-renewable energy resources
- Reduced greenhouse gases

1.3 Problem Statement

In Ethiopia gathering wood for fuel and burning in inefficient stoves leads to local scarcity and ecological damage in areas of high population density where there is strong demand of wood for domestic purpose. The annual per capita consumption of biomass fuels for domestic purpose in rural areas is estimated 0.7 to 1.0 ton; the share of woody biomass from the total is about 80%, the rest comes from crop residue and animal dung [13].

The extensive use of biomass in traditional and inefficient ways and in some extent the modern fuels causes drought, environmental pollution and restrains economic and social development. To avoid or minimize these problems and increased concerns over the local, regional and global environmental impacts of conventional energy systems; promotion to renewable energy resources is necessary. Thus designing and manufacturing of solar powered injera baking oven will have important role to solve the problem. In this thesis focus was given on injera baking oven. This is because baking injera consumes the largest share of biomass fuel for domestic purpose.

1.4 Objectives

1.4.1 General Objectives

The general objective of this thesis is to design, manufacture and study the performance of a heat transport system for solar powered injera baking oven that uses oil as a heat transfer fluid from the solar receiver to baking pan.

1.4.2 Specific Objectives

The specific objectives of this thesis are:

- To design and manufacture an air free heat transport system loop from oil storage tank to the injera baking compartment.

- To determine the optimum oil temperature in the oil storage tank and at the baking compartment for injera baking pan.
- To determine the optimum pipe diameter and the pump capacity.
- To determine the heat loss from different parts of the systems.
- To perform a series of experiments during initial heat up and baking using the laboratory model of the system.

1.5 Significance of the Project

- The majority of the people in Ethiopia live in rural areas, in which electricity power source is not available, so the use of solar power for baking injera plays an important role in their life.
- Women and children spent much of their time in collection of fuel wood and they are exposed to health problems due to smoke from the burning wood; so the use of solar power conserves their time and avoids the health problems.
- The solar baking system will solve the problem of energy sustainability by replacing the current dependency on fuel wood, dung and crop residues and this contributes reduction in air pollution and green house gas emissions in the long range.

1.6 Limitations of the Project

Even though the final result was almost successful by using some modifications and alternatives during manufacturing and experimental testing of the laboratory model there were limitations as:

- The required materials like high temperature resistant gasket, oil pump for high temperature application, high capacity electrical heater, fiber-glass for insulation system and high thermal conductivity sheet metal plate were not found in the local market easily.
- The heat transfer oil decreases its viscosity as temperature increases and become very thin which was difficult to control the leakage through the fittings.
- Another limitation of the project was; because of time limitation and financial problems it was not possible to construct the solar baking system except the laboratory model using oil heating tank.

CHAPTER TWO

LITERATURE REVIEW

This chapter reviews research work along the areas of the study found in literature. It is basically a review about the modes of heat transfer, the types of solar cookers, the Injera baking system in Ethiopia, the parabolic trough solar collector and the heat transfer fluids.

2.1 Modes of Heat Transfer

Heat transfer, also known as heat flow, heat exchange, or transfer of thermal energy is the movement of heat from one place to another. When an object is at a different temperature from its surroundings, heat transfer occurs so that the body and the surroundings reach the same temperature at thermal equilibrium. Such spontaneous heat transfer always occurs from a region of high temperature to another region of lower temperature. There are three basic modes of heat transfer mechanisms defined as follows:

2.1.1 Conduction Heat Transfer Mechanism

Transfer of energy between objects in physical contact is the transfer of heat from a hot side to a cooler side through a dividing medium. The hot side heats the molecules in the dividing medium and causes them to move rapidly, heating the adjacent molecules until the cool side is heated. The transfer of heat stops when the temperature of the hot side equals that of the cool side. In heat transfer by conduction heat flow (Q) through the area A in a plane normal to the direction of heat transfer in time (δt) given by Fourier's law of conduction as: [5, 8, 4, 25].

$$\dot{Q} = -k\,A\frac{\delta T}{\delta X} \qquad (2.1)$$

Conduction through a plane wall: The details of conduction are quite complicated but for engineering purposes may be handled by a simple equation, usually called Fourier's equation. For the steady flow of across a plane wall with the surfaces at temperatures of T_1 and T_2 where T_1 is greater than $T_{2;}$ the heat flow Q per unit area A; (the heat flux) is:

$$\frac{Q}{A} = k\left(\frac{T_1 - T_2}{X_1 - X_2}\right) = k\frac{\Delta T}{\Delta X} \qquad (2.2$$

Equation (2.1) can be written in a more general form if the temperature gradient term is written as a differential:

$$\dot{Q} = -k\,A\frac{dT}{dX} \qquad (2.3$$

Where: $\dot{Q}$ - Is the heat flow (W)

dT/dX - Temperature gradient in plane of heat transfer in the direction of heat flow (^{o}C/m)

k- Thermal conductivity (W/m^{o}C)

A – Area perpendicular to heat flow (m^2)

The negative sign in the equation is introduced to account for the fact that heat is conducted from a high temperature to a low temperature, so that (dT/dX) inherently negative; therefore the double negative indicates a positive flow of heat in the direction of decreasing temperature.

2.1.2 Convection Heat Transfer Mechanism

Convection is transfer of energy between an object and its environment, due to fluid motion. Convection can be forced convection in which the flow is caused by a pump or a fan or it may be natural convection in which the flow is caused by density differences due to differences in temperature. It is found that the heat flux is approximately proportional to the temperature difference between the wall and the bulk of the fluid.

$$\frac{Q}{A}\alpha\,(T_f - T_s) \qquad (2.4)$$

This causes to define a constant proportionality called “convection heat transfer coefficient” denoted by, h

$$\frac{Q}{A} = h(T_f - T_s) \qquad (2.5)$$

Thus the rate of convection heat transfer is given by [2, 8, 4, 25].

$$\dot{Q} = h\,A\Delta T \qquad (2.6)$$

Where: h -convection heat transfer coefficient (W/m^2 ^{o}C)

A - Surface area where convection takes place (m^2)

ΔT - Temperature difference between the fluid and the wall surface (^{o}C)

2.1.3 Radiation Heat Transfer Mechanism

Transfer of energy from or to a body by the emission or absorption of electromagnetic radiation. All objects with a temperature above absolute zero radiate energy at a rate equal to their emissivity multiplied by the rate at which energy would radiate from them if they were a black body. According to Stefan–Boltzmann law, ideal radiators emit energy at a rate proportional to the fourth power of the absolute temperature. And the net rate of exchange of energy between two ideal radiators A and B related as; [5, 8].

$$\dot{Q} = \sigma A (T_A^4 - T_B^4) \tag{2.7}$$

Where:

σ - Stefan- Boltzmann constant = 5.67x10-12w/cm^2.k^4

T_A - Temperature of body A (^{o}C)

T_B - Temperature of body B (^{o}C)

In many physical situations, we are interested in radiation heat transfer from the surface of an object to the surrounding uniform temperature. Thus, the net radiation from a non-black surface to the surrounding is given by [2].

$$\dot{Q} = \varepsilon \sigma A (T_s^4 - T_\infty^4) \tag{2.8}$$

Where: T_s - Body surface temperature (^{o}C)

T_∞ - Surrounding or ambient temperature (^{o}C)

ε - Emissivity of the surface and has a value between 0 and 1, for a perfect reflector $\varepsilon = 0$ and for a perfect emitter a so called "black body", $\varepsilon = 1$.

2.2 Solar Energy Basics

The basic resource for all solar energy systems is the sun and all life on the earth depends on solar energy. Knowledge of the quantity and quality of solar energy available at a specific location is of prime importance for the design of any solar energy system. Although the solar radiation (*i*nsolation) is relatively constant outside the earth's atmosphere, local climate influences can cause wide variations in available insolation on the earth's surface from site to site. In addition, the relative motion of the sun with respect to the earth will allow surfaces with different orientations to intercept different amounts of solar energy.

The sun is a sphere of intensely hot gaseous matter with a diameter of $1.39x10^9$m and is, on the average, $1.5x10^{11}$m from the earth. The earth revolves around the sun every 365.25 days in an elliptical orbit, with a mean earth-sun distance of 1.496×10^{11} m (92.9×10^6 miles) defined as one astronomical unit (1 AU). This plane of this orbit is called the ecliptic plane. The earth's orbit reaches a maximum distance from the sun, or aphelion, of 1.52×10^{11} m (94.4×10^6 miles) on about the third day of July. The minimum earth-sun distance, the perihelion, occurs on about January 2nd, when the earth is 1.47×10^{11} m (91.3×10^6 miles) from the sun. [10]

Solar energy will provide an ever-increasing fraction of our future energy requirement. Although sun light or solar radiation is abundant and renewable, it is diffuse. The temperature from this diffuse source is sufficient to provide domestic hot water or home heating, but much higher temperature are necessary to displace fossil fuels for producing electricity or other industrial applications. Thus solar radiation must be concentrated to produce this elevated temperature; it must also be collected and moved to the point of use. The use of solar energy to cook food presents a viable alternative to the use of fuel wood, kerosene, and other fuels traditionally used in developing countries for the purpose of preparing food.

2.3 Solar Cookers

Solar food cookers use an arrangement of reflectors to concentrate solar energy on a cooking vessel. A number of innovative designs have been developed and many of the solar cooker designs are inexpensive and easy to build.

2.3.1 Types of Solar Cookers

Solar cookers can be classified in to the following main categories:

Direct Solar Cookers: Direct solar cookers are cookers; in which cooking system uses direct solar energy to cook different types of foods. It is a direct-focusing cooker which uses a reflector to focus sunlight directly onto a cooking pot either suspended or set on a stand at the focal point. Numerous arrangements of this cooker have been devised for allowing the reflector to be tilted to always point toward with the pot remaining at the focal point. The most common types of direct solar cookers are Box Cookers, Curved Concentrator Cookers and Panel Cookers.

Indirect Solar Cookers: In this case the cooking vessel is physically displaced from the collector and a heat transferring medium is required to convey the heat to the cooking utensils. The heat transfer medium is circulating in a closed loop in an insulated pipe and container. These cookers are advantageous than the direct cookers because they provide high thermal power and temperatures and allow cooking in indoors.

2.4 Injera Baking System in Ethiopia

Preparation of Injera has a long process; it usually takes two to four days from mixing to cooking. It can be produced from almost any staple grain, with sorghum, millet and teff being the most common in Ethiopia. The teff flour is mixed with water and left to ferment for two to four days, but can take less than this time in warmer locations. Starter (left-over batter from the previous baking time) may be added to trigger fermentation. Approximately four to six hours before baking, a layer of bitter fermentation product is removed and hot water is added to reactivate fermentation, then the batter is poured on top of the hot baking pan surface.

To bake injera, the heat supplied to the baking pan either comes from burning fuel wood, dung or agricultural residue in biomass cookers, by heating electrical resistance in the electric baking pan and by means of heating heat transfer fluids for solar powered baking pan as the case in this particular thesis. This heat is then conducted through the baking pan to the surface where the batter is cooked. The heat supplied to the injera baking pan is used for raising the temperature of the batter on the pan surface from room temperature (20 to 25^{o}c) to around boiling point of water. (In Addis Ababa, boiling point of water is about 92^{o}c). Conventional baking pans are 58 - 60cm in diameter.

2.4.1 Injera Baking Using Open Fire System

In most of the households of the country, Injera baking is carried out using an open fire (three stone) baking system and the fuel is biomass. The heat supplied to the mittad in this system is lost through a variety of paths such as: through the sides, through the exhaust gases from the fuel, through convective and radiative heat losses from the pan surface. The fraction of energy that flows into the Injera batter is very small and therefore this technique is inefficient and wasteful;

and also is unhealthy because it can damage the lungs and eyes of those in close proximity to the oven or fire.

Figure 2.1: open fire (three stone) injera baking system

2.4.2 Mirte Injera Stove

It is prefabricated stove from cement and local aggregate such as sand panels. The stove is suitable for mass production by casting the light concrete. Each Mirte saves approximately 5 kg of wood per injera baking session for the average household. Most household bakes injera twice a week. Thus, the Mirte saves at average per household nearly 260 kg of wood a year [20]. This is a significant savings for the average Ethiopian urban household. However, the Mirte saves commercial injera bakers over 3.5 tons of fuel wood per year [20].

Even though the mirt stove is better and efficient comparing to the open fire baking system; it is disadvantageous in:

a. Since it uses biomass (wood or animal dung) it has contribution to deforestation and limits advantage of dung as plant fertilizer; however in a lesser amount than the open fire system.
b. It can produce some smoke and may result in producing some pollution if baking is in door.

Figure 2.2: Mirt injera baking stove

2.4.3 Electrical Injera Baking Pan /Mittad/

Electricity is an important energy source, so the other type of technology for injera baking is electric ‘’mitad’’; which is mainly used by people in the urban and near urban towns where electricity is available. Thus, the majority of population (more than 80%) in Ethiopia uses wood or biomass fuel for injera baking.

Disadvantages of electrical baking system:-
a. If the source of energy is diesel fuel, it needs high cost and has contribution to the resource depletion and air pollution.
b. The electric baking system is used only for the urban areas where electricity is available; so that a lot of rural people do not have access of the electricity net work.
c. There is high energy loss through the sides and bottom of the baking assembly; and also it has maintenance and labor cost.

Figure 2.3: Electrical injera baking mitad assembly

Comparing the advantages and disadvantages of each baking system, the solar baking system should be selected since it includes the following benefits:

a. No smoke production, free of air pollution and therefore safe for health and environment.

b. Does not cause resource depletion

c. Can be used anywhere, there is solar radiation.

d. Uses abundant, free and renewable energy source.

2.5 Parabolic Trough Solar Collectors (PTSC)

A parabolic trough solar collector, actually, is a special kind of heat exchanger that transfers solar radiation energy into heat. PTSCs are linear focus concentrating solar devices suitable for working in the 150^{o}c – 400^{o}c temperature range [12].

2.5.1 The Operational Principles and Components of the PTSC

A parabolic trough solar collector is basically made up of:

- A parabolic trough-shaped mirror that reflects direct solar radiation on to the PTSC’s receiver tube

- Receiver tube located in the focal line of the parabola,
- Steel support structure and an axis drive mechanism.

The concentrated radiation heats the fluid that circulates through the receiver tube, thus transforming the solar radiation into thermal energy in the form of the sensible heat of the fluid. PTSCs are dynamic devices because they have to rotate around an axis, the so-called tracking axis, to follow the apparent daily movement of the sun.

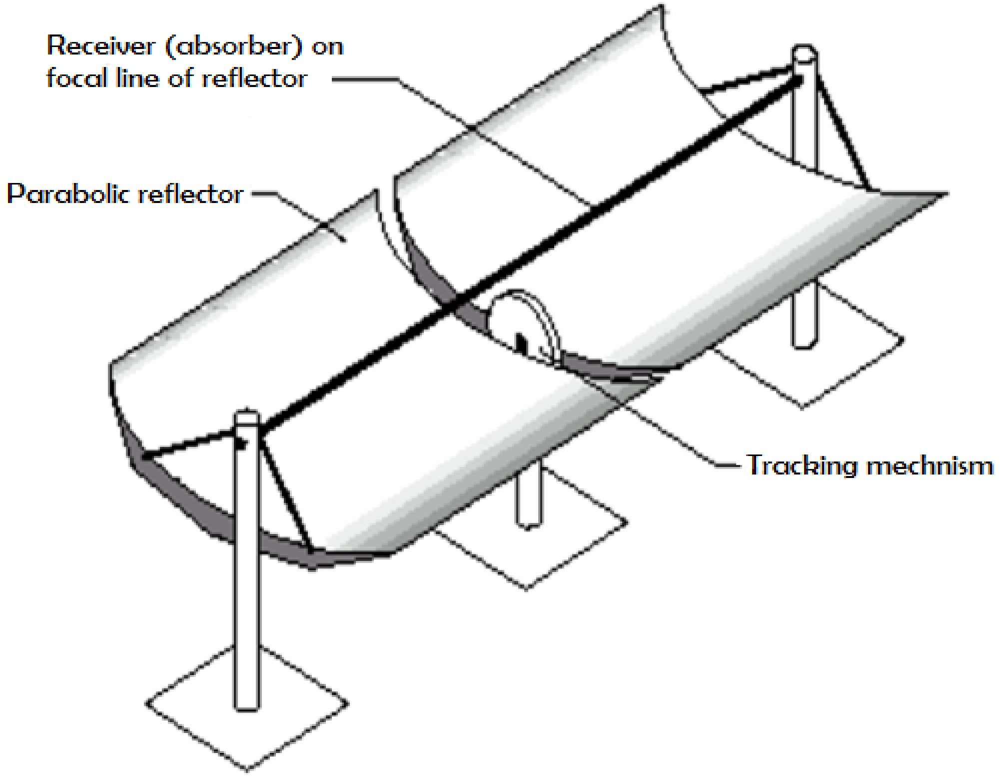

Figure 2.4: Main components of parabolic trough solar collector

Thermal oils are commonly used as the working fluid in these collectors for temperatures above 200^{o}C [12], because at these high operating temperatures normal water would produce high pressures inside the receiver tubes and piping. When choosing a thermal oil to act as working fluid, the main limiting factor to be taken into consideration for stability is the maximum oil bulk temperature. Above this temperature, oil cracking and rapid degradation might occur.

PTSCs are usually installed with the rotation axis oriented either north–south or east–west; however, any other orientation would be feasible too. Solar collector orientation influences the sun incidence angle on the aperture plane which, in turn, affects collector performance.

The main design parameters required for a PTSC are the geometric concentration ratio, the acceptance angle, and the rim angle. The concentration ratio is the ratio between the collector aperture area and the total area of the absorber tube, whereas the acceptance angle is the maximum angle that can be formed by two rays in a plane transversal to the collector aperture so that they intercept the absorber pipe after being reflected by the parabolic mirrors.

2.5.2 Energy Balance Equations

The heat collection element (HCE) performance uses an energy balance between the heat transfer fluid (HTF) and the atmosphere, and includes all equations and correlations necessary to predict the terms in the energy balance, which depend on the collector type, HCE condition, optical properties, and ambient conditions.

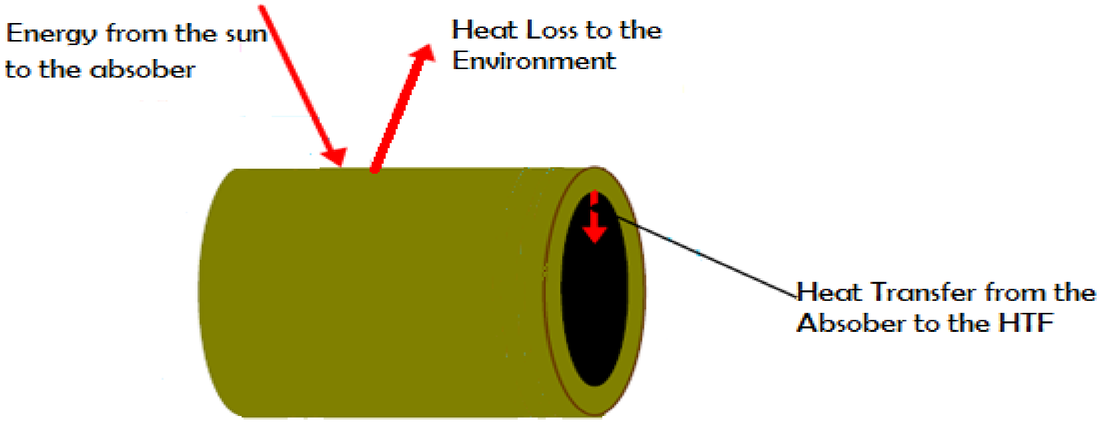

Figure 2.5: Heat transfer at the Heat Collection Element

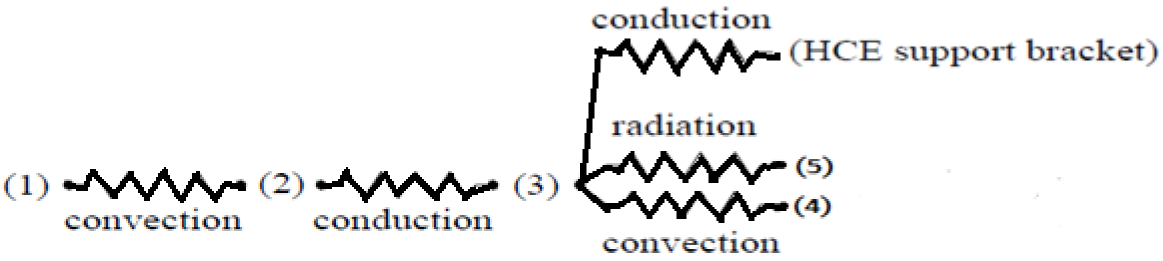

Figure 2.6: Thermal resistance circuit of the HCE

Where:

1 – Heat transfer fluid	4 – Surrounding air
2 – Absorber inner surface	5 - Sky
3 - Absorber outer surface	

Convection Heat Transfer between the HTF and the Absorber: From Newton's law of cooling, the convection heat transfer from the inside surface of the absorber pipe to the HTF is [15].

$$\dot{Q}_{12conv} = h_1 D_2 \pi (T_2 - T_1) \tag{2.9}$$

$$h_1 = Nu_{D2} \frac{k_1}{D_2} \tag{2.10}$$

Where:

$h_1 =$ HTF convection heat transfer coefficient at T1 (W/m^2-k)

$D_2 =$ Inside diameter of the absorber pipe (m)

$T_1 =$ Mean bulk temperature of the HTF (^{o}C)

$T_2 =$ Inside surface temperature of the absorber pipe (^{o}C)

$Nu_{D2} =$ Nusselt number based on D_2

$k_1 =$ Thermal conductivity of the HTF at T_1 (W/m-k)

Conduction Heat Transfer through the Absorber Wall: Fourier's law of conduction through a hollow cylinder describes the conduction heat transfer through the absorber wall.

$$\dot{Q}_{23cond} = 2\pi k_{23} (T_2 - T_3) \big/ \ln(D_3 / D_2) \tag{2.11}$$

Where:

$k_{23} -$ Thermal conductance at the average absorber temperature (T_2+T_3)/2 (W/m-k)

$T_3 -$ Absorber onside surface temperature (K)

$D_3 -$ Absorber outside diameter (m)

Heat Transfer from the Absorber to the Atmosphere: The heat will transfer from the Absorber to the atmosphere by convection and radiation. The convection will either be forced or natural, depending on whether there is wind. Radiation heat loss occurs due to the temperature difference between the Absorber and sky.

Convection Heat Transfer: The convection heat transfer from the Absorber to the atmosphere (q_{34conv}) is the largest source of heat loss, especially if there is a wind from Newton's law of cooling [14].

$$\dot{Q}_{34conv} = h_{34}\,\pi\,D_3\,(T_3 - T_4) \tag{2.12}$$

$$h_{34} = \frac{k_{34}}{D_3} Nu_{D3} \tag{2.13}$$

Where: $k_{34} -$ Thermal conductance for air at (T_3-T_4)/2 (W/m-k)

$T_4 -$ Ambient air temperature (K)

$h_{34} -$ Convective heat transfer coefficient for air at (T_3-T_4)/2 (W/m-k)

$Nu_{D3} =$ Nusselt number based on D_3

Radiation Heat Transfer: The radiation transfer between the absorber and sky, discussed here, is caused by the temperature difference between the absorber and sky. To approximate this, the Absorber is assumed to be a small convex gray object in a large blackbody cavity (sky). The net radiation transfer between the absorber and sky becomes:

$$\dot{Q}_{35} = \delta\,D_3\,\pi\,\varepsilon_3\,(T_3^{\,4} - T_5^{\,4}) \tag{2.14}$$

Where: $\delta -$ Stefan-Boltzman constant ($5.67*10^{-8}$) (w/m^2.k^4)

$T_5 -$ Effective sky temperature (K)

$\varepsilon_3 -$ Emissivity of the Absorber outer surface

Solar Irradiation Absorption

Optical Properties: Table 2.1 lists terms used to estimate the effective optical efficiencies. The data in the table are valid only for solar incidence irradiation normal to the collector aperture. An incident angle modifier term is added to account for incident angle losses, which includes trough end shading, changes in reflection and refraction, and selective coating incident angle effects.

Table 2.1: Estimates of Effective Optical Efficiency terms [15]

ε_1 = shadowing (bellows, shielding, supports)	0.974
E_2 = Tracking Error	0.994
ε_3 = Geometry Error (mirror alignment)	0.98
ρ_{cl} = Clean Mirror Reflectance	0.935
ε_4 = Dirt on Mirrors	Reflectivity/ ρ_{cl}
ε_5 = Dirt on HCE	$(1+ \varepsilon_4) / 2$
ε_6 = Unaccounted	0.96
Reflectivity is a user input (typically between 0.88 and 0.93)	

Incident angle modifier (K): is needed for cases when the solar irradiation is not normal to the collector aperture. It is a function of the solar incidence angle to the normal of the collector aperture.

$$K = \cos(\theta) + 0.000884\theta + 0.00005369\theta^2 \tag{2.15}$$

Solar Irradiation Absorption in the Absorber: The solar energy absorbed by the absorber occurs very close to the surface; therefore, it is treated as a heat flux. The equation for the solar absorption in the absorber becomes:

$$\dot{Q}_{3SolAbs} = q_{si}\ \eta_{abs}\ \alpha_{abs} \tag{2.16}$$

$$\eta_{abs} = \varepsilon_1\,\varepsilon_2\,\varepsilon_3\,\varepsilon_4\,\varepsilon_5\,\varepsilon_6\,\rho_{cl}\,K \tag{2.17}$$

Where: q_{si} − Solar irradiation per receiver length (w/m)

η_{abs} − Effective optical efficiency at Absorber

α_{abs} − Absorbance of absorber

Heat Loss through HCE Support Bracket: The HCEs are supported at the collector focal line by support brackets that run from the collector structure to the absorber pipe. The bracket losses are approximated by treating the support bracket as an infinite fin with base temperature less than the outer absorber surface temperature T_3 at the point where the bracket is attached. The bracket heat loss is estimated with the following equation:

$$\dot{Q}_{cond,bracket} = \sqrt{h_b\, p_b\, k_b\, A_{cs,b}}\,(T_{base} - T_4) \Big/ L_{HCE} \tag{2.18}$$

Where: $h_b -$ Average convection coefficient of bracket (W/m^2.k)

$p_b -$ Perimeter of bracket (m)

$k_b -$ Conduction coefficient (W/m^2.k)

$A_{cs,b} -$ Minimum cross-sectional area of bracket (m^2)

$T_4 -$ Ambient temperature (^{o}C)

$L_{HEC} -$ Heat collection element length (m)

2.5.3 Sizing and Layout of Solar Fields with PTSCs

A typical parabolic trough solar collector field is composed of a number of parallel rows of several collectors connected in series so that the working fluid circulating through the absorber pipe is heated as it passes from the inlet to the outlet of each row. The first step in the design of a parabolic trough solar collector field is the definition of the so-called design point, which is a set of parameters that determine solar field performance. Parameters to be defined for the design point are:

- The collector orientation
- The date (month and day) and time of design point
- The direct solar irradiance and ambient air temperature for the selected date and time
- The geographical location of the plant site (latitude and longitude)
- The total thermal output power to be delivered by the solar field
- The solar field inlet/outlet temperatures
- The working fluid for the solar collectors and its nominal flow rate.

2.6 Heat Transfer Fluids

Heat transfer fluids are used to convey energy from the source to the end use devices and the heat transfer medium can be water, air or heat transfer oil.

2.6.1 Heat Transfer Oils

Heat transfer oil is a non-corrosive heat transfer fluid that is formulated to provide fast and efficient heat transfer when used in a closed system application. Heat transfer oil is blended from the finest select high viscosity index severely solvent refined, severely hydro treated pure Paraffin base oils available. These pure paraffin base oils allow heat transfer oil to exhibit the following performance characteristics:

- ***High Viscosity Index* –** these results in a minimum change in viscosity over a broad temperature range.
- ***High Thermal and Oxidative Stability* –** This results in the product having resistance to cracking, carbon, sludge, varnish and lacquer formation during high temperature operation.
- ***Low Volatility Characteristics* –** The low volatility of paraffin base oils not only results in not only lower makeup requirements, but also helps eliminate vapor lock in circulating pump and reduces the possibility of cavitations.

Besides these performance benefits Heat Transfer Oil also offers the following benefits:

- High thermal efficiency for rapid and efficient transfer of heat.
- Low vapor pressure at elevated temperatures and high boiling point to prevent pressure build-up.
- Non-corrosive to system parts and Non-fouling on degradation.
- Excellent hydrolytic stability and resistance to emulsification with water.
- Excellent compatibility with other petroleum base heat transfer oils.
- Excellent compatibility with all types of seals, materials of construction and finishes commonly used in heat transfer systems.
- Virtually odorless and essentially non-toxic.
- Long service life for proven trouble-free operation.

2.6.2 Selection of Heat-Transfer Oils

Heat-transfer fluids carry heat through collectors and heat exchangers for the application process. When selecting a heat-transfer fluid, it is important to consider the following criteria:

- ***Coefficient of expansion*** **–** the fractional change in length or in volume of a material for a unit change in temperature
- ***Viscosity*** – resistance of a liquid to sheer forces (and hence to flow)
- ***Thermal capacity*** **–** the ability of matter to store heat
- ***Freezing point*** **–** the temperature below which a liquid turns into a solid
- ***Boiling point*** **–** the temperature at which a liquid boils
- ***Flash point*** –is the lowest temperature at which a liquid can give off sufficient vapors to form an ignitable mixture in air near the surface of the liquid. It is the lowest temperature to which a lubricant must be heated before its vapor.
- ***Pour Point:*** is the lowest temperature at which the oil will pour or flow when cooled under prescribed conditions.
- ***Fire point:*** is the lowest temperature at which a liquid can give off sufficient vapors to form a mixture in air that continuously supports combustion after ignition near the surface of the liquid.
- ***Volatility:*** For any high temperature processing application, volatility is an important consideration as oil fumes represent a serious health hazard. Oil volatility generally increases with decreasing viscosity.

2.6.3 Shell Thermia Oil B Heat Transfer Oil

Shell thermia oil B is based on carefully selected highly refined mineral oils chosen for its ability to provide superior performance in indirect closed fluid heat transfer systems. It is applicable for enclosed circulated heat transfer system for industrial application such as process industry, chemical plants, textile producers etc. Shell thermia oil B can be used in high temperature continuous heat transfer equipment with maximum Film temperature 340^{o}c and maximum bulk temperature 320^{o}c application limits [23].

An expansion tank is necessary to allow for the change in fluid volume upon heating. The volume of mineral oil at 300°c is about 20 per cent greater than at room temperature [23]. The tank should be large enough to accept the total heat expansion within its own dimensions.

The life of Shell thermia oil B depends on the design and usage of the system. If the system is well designed, and not subjected to abnormal workloads, the life can be for many years.

Table 2.2: Typical design data for shell thermia oil B heat transfer oil [23]

Properties	Shell Thermia oil B heat transfer oil								
Temp [^{o}c]	0	20	40	100	150	200	250	300	340
Density [kg/l]	0.876	0.863	0.850	0.811	0.778	0.746	0.713	0.681	0.655
c [kJ/kg.k]	1.809	1.882	1.954	2.173	2.355	2.538	2.72	2.902	3.048
k [W/m.k]	0.136	0134	0.133	0.128	0.125	0.121	0.118	0.114	0.111
Pradtle number	3375	919	375	69	32	20	14	11	9
β [^{o}C^{-1}]	0.0008								

Performance Features and Benefits of Shell Thermia oil B heat transfer oil

- ***High oxidation***; **Shell t**hermia oil B is based on carefully selected highly refined mineral oils. The rates of oil cracking and oxidation are very small, giving long oil life.
- ***Thermal stability***, Longer oil life at higher operating temperature
- ***Low viscosity and high heat transfer coefficient***, Low viscosity enables excellent fluidity and heat transfer also at lower temperatures.
- ***Very high operating temperature***
- ***Very high initial boiling point***
- ***Non-corrosive;*** does not attack the metal parts of the systems being a blend of pure base oils. Thus ensures long life of component parts.
- ***Low vapor pressure***
- ***Non-toxic*** , safe for handling, use and storage

CHAPTER THREE

SYSTEM COMPONENTS DESIGN AND MANUFACTURING

3.1 Solar powered Injera baking System

The solar powered injera baking system uses solar radiation as the source of energy. It uses parabolic trough solar collector to convert the solar radiation in to heat energy. The heat energy is conveyed from the source (collector) to the end use compartment (baking pan surface) using heat transfer oil.

The solar powered injera baking oven has the following main components:-

- The parabolic through solar collector with its tracking mechanism (to collect and reflect the solar radiation and heat up the heat transfer oil in the receiver tube).
- Highly insulated oil storage tank
- The baking pan assembly (for baking injera using the heat gained from solar system).
- The piping lines from the receiver tube to the oil gallery under the baking pan using oil pump to circulate the heat transfer oil through the system.

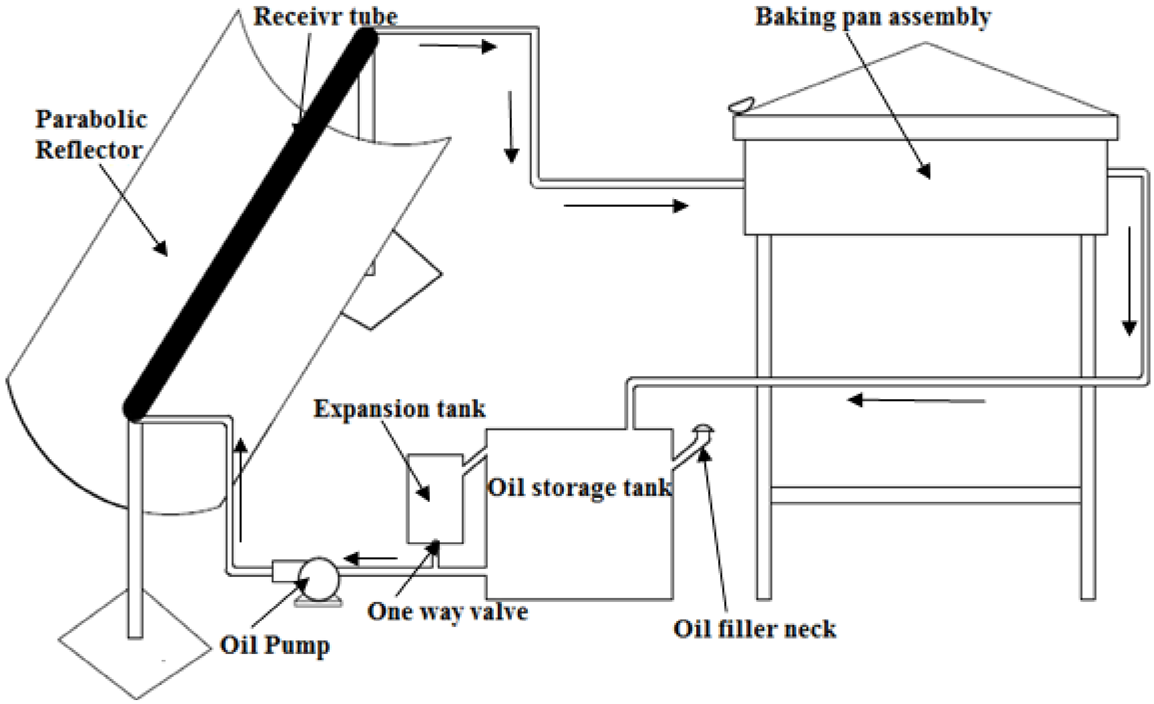

Figure 3.1: Main components of solar powered injera baking system.

3.2 Laboratory Model System Components

The main components of the injera baking system considered in this laboratory model are:

- Oil heating and storage system (heat transfer oil in the storage tank and electrical heater).
- Baking pan assembly (which consists of a stand to support, ash insulation system, oil gallery and baking pan).
- Insulation system (ash insulation system for the oil gallery and storage tank and fiber glass for the piping system were used).
- Heat transfer fluid piping and pumping system for the oil circulation (pipes from; storage tank to pump, pump to oil gallery, oil gallery to oil storage tank and excess return line from pump to oil storage tank were used and the pump was driven by an electric motor).
- Sealing system (gasket maker and high temperature resistance gasket sealing system were used to protect from oil leakage and air entering to the system).
- The pan cover (used to open and close baking pan surface when needed).

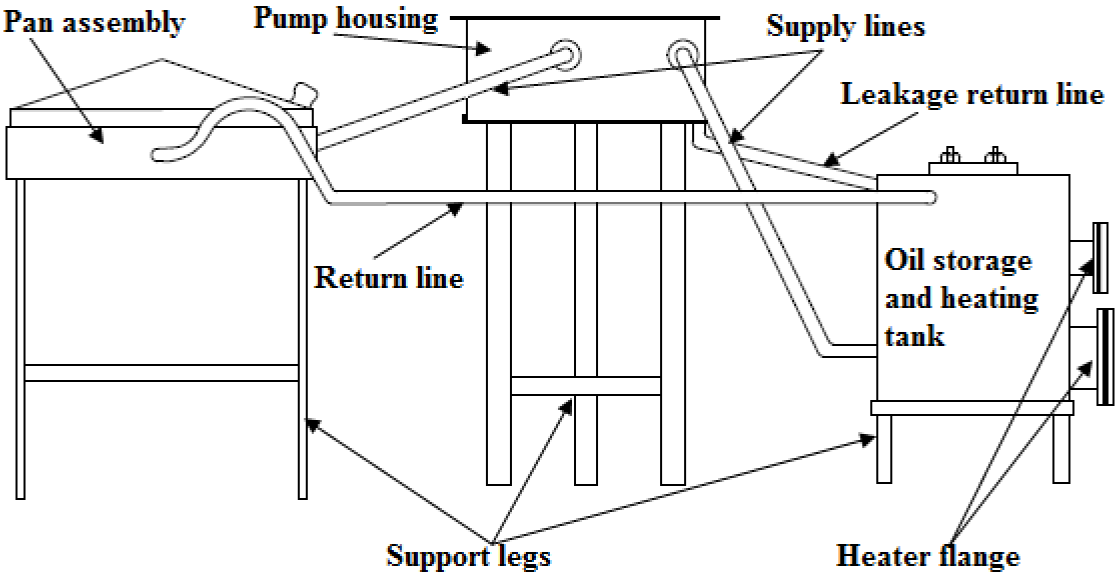

Figure 3.2: Main components of the laboratory model

3.3 Working Principle of the Laboratory Model

The working principle of the laboratory model as can be observed from figure (3.2); the baking pan (mitad) and the heat storage system are placed separately. Thus, heat was transferred from the heat storage unit to the baking pan assembly indirectly using heat transfer fluid. The Thermal oil (thermia oil B) was used as a heating medium as the injera baking process requires very high surface temperature usually 180^{o}c to 220^{o}c.

The heat transfer fluid stored in the storage tank, after it was heated by electrical heater it was is pumped to the baking pan assembly (oil gallery). Thus heat was transferred from the heated oil to the baking pan by convection and conduction heat transfer mechanisms. After giving the energy to the baking pan the oil from the oil gallery returned to the oil storage tank through the return pipe line in order to get heated and this circulation was continue for the required period of time.

3.4. Energy Required for Injera Baking

Baking injera requires intensive energy and this energy can be defined as the energy necessary to raise the batter to a particular temperature, and evaporate the amount of water that is to be lost during the baking process. To measure the energy utilized in cooking injera, the initial mass of batter, and the total amount of injera produced from this batter were measured.

Thus the mass of water vapor can be obtained by reducing the mass of the injera produced from the initial mass of batter. It is assumed that the energy utilized in cooking the injera is the energy required in raising the temperature of the batter from room temperature to the boiling point of water which is called sensible heat, plus the energy required to evaporate water which is called latent heat. It is also assumed that the heat capacity of injera batter is the same as that of water in order to calculate the energy required to raise the batter temperature to boiling point [6, 22].

Therefore, the utilized energy is:

$$E_{utilized} = m_{batter}\, c_{water}\,(T_{boil} - T_{room}) + (m_{batter} - m_{injera})h_{vaporzation} \tag{3.1}$$

Where:

m_{batter} -is the mass of the batter expected for one injera= 400g

T_{boil} - is the boiling temperature of water in Addis Ababa = 92^{o}C

T_{room} - is the room temperature in the baking pan test room = 20^{o}C

c_p - is the heat capacity of water = 4.187kJ/kg.k

m_{injera} - is the mass of the injera produced = 320g

$h_{vaporization}$ - is the heat of vaporization of water h_{fg} = 2260kJ/kg

$$E_{utilized} = 0.4kg \times 4.187kJ/kg.k \times (92 - 20)\,k + (0.4 - 0.32)\,kg \times 2260\,kJ/kg$$

$$= 120.6KJ + 180.8kJ$$

$$E_{utilized} = 301.4kJ$$

During injera baking there are losses in pipes, in storage tank, in pump housing and in the oil gallery, so considering the losses and assuming a safety factor of 1.3 the total energy required will be 301.4kJ x 1.3 = 391.82kJ.

The time taken for cooking of one injera is assumed to be about 2 to 3 minutes taking 3 minutes; the power required for injera baking can be calculated as:

$$P = \frac{E_{utilized}}{\Delta t} \qquad (3.2)$$

Where: P- the power required for baking one injera

$$\Delta t = 3x60 = 180\sec onds$$

Then; $P = \frac{391.82}{180} = 2.2kW$

Here the efficiency of the mittad and the energy required for heat up were ignored.

3.5 Design and Manufacturing of Laboratory Model System Components

3.5.1 Design of the Baking Pan Assembly

Design of *Baking Pan Cover*

To protect heat loss from injera during baking, pan cover is required. It was manufactured from aluminum with the following dimensions.

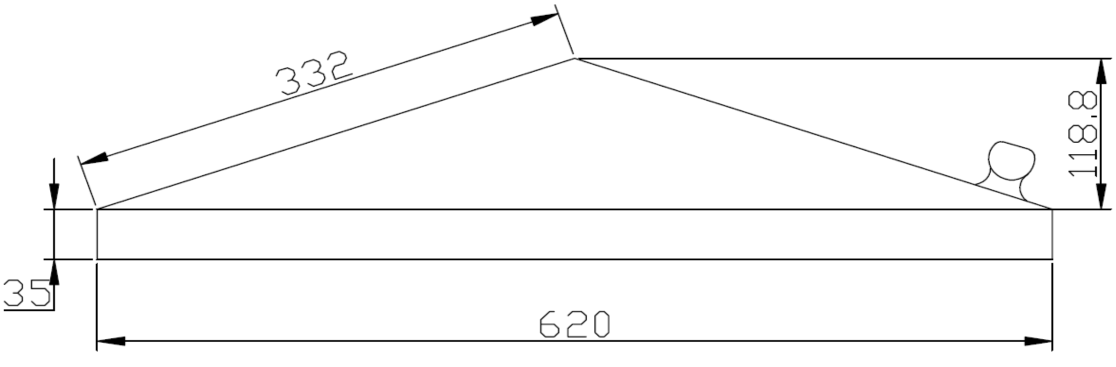

Figure 3.3: Baking pan cover

Design of Baking Pan

A baking pan is a flat and circular pan commonly about 50 to 60cm in diameter and traditionally used over large clay hearths to bake injera [20]. The baking pan 'mitad' considered in this case was 8mm thick and 580mm in diameter and because of its reduced thickness, it has high thermal conductivity than the one which is available in the local market. Direct contact of the pan with the hot heat transfer oil can cause cracking of the baking pan; therefore the baking pan was separated from the heat transfer fluid with conductive steel sheet metal.

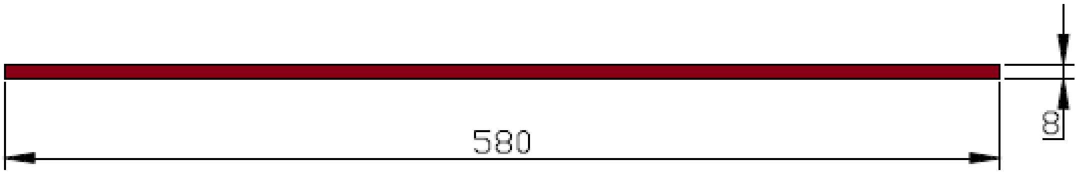

Figure 3.4: Baking pan Dimension

Figure 3.5: Ceramic baking pan (Mittad)

Design of Heat Transfer Fluid Gallery

To transfer heat energy uniformly from the heat transfer oil; the oil storage below the pan was used to overcome sudden drop of surface temperature during baking below the optimum baking temperature. The oil inlet and outlet ports were drilled with1/2 inch diameter size on opposite sides of the housing of the gallery; pipes with 1/2 inch diameter were welded to the drilled inlet and outlet ports. To increase the contact between the hot heat transfer oil with the pan supporting plate, fin like structures were welded inside the oil gallery to hinder direct oil flow and also to increase strength of the gallery. In order to decrease heat loss, the oil gallery was insulated by ash insulation system which has about 3.5cm thickness from below and the side wall. The volume of the oil gallery can determined as follows:

$$Vg = \frac{\pi}{4} D_g{}^2 \, h_g \tag{3.3}$$

$$Vg = \frac{\pi}{4} 0.58^2 x0.06 = 0.0159m^3 = 15.9liters$$

But there is fin structure inside the oil gallery and its volume is:

$$V_{fin} = l \; w \; h = 2(0.53x0.003x0.06) - (0.06x0.06x0.003)$$

$$V_{fin} = 0.00018m^3 = \underline{\underline{0.18}}liters$$

Hence the effective volume (V_{eff}) of the oil gallery is:

$$V_{eff} = (0.0159m^3 - 0.00018m^3) = 0.015.7m^3 = \underline{\underline{15.7}}liters$$

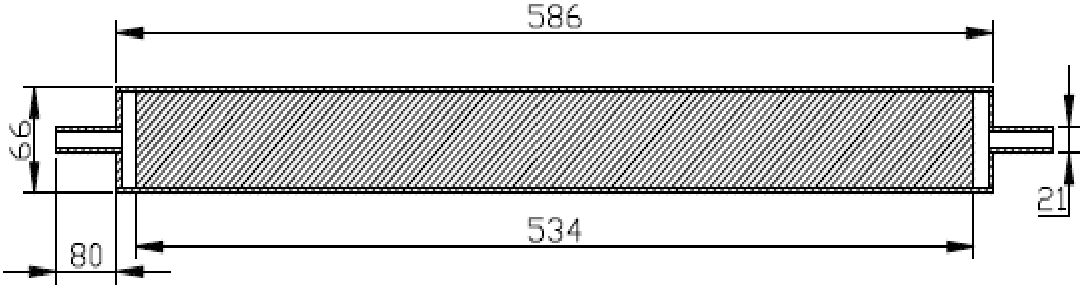

Figure 3.6: Heat transfer fluid gallery.

Calculation of Optimum Temperature of the Heat Transfer Fluid in the gallery

When a fluid comes in contact with a solid body, heat exchange will occur between the solid and the fluid whenever there is a temperature difference between the two. During heating and cooling of gases and liquids, the fluid streams exchange heat with the solid surfaces by convection. Energy transfer from heat transfer fluid to the baking pan was in the following two ways:

- Free convection from hot oil to pan supporting plate:
- Conduction from pan supporting plate to the baking pan**:**

Assumptions for the analysis:

- Steady state conditions
- Negligible contact resistance between the supporting plate and pan
- One dimensional heat flow through the y- direction
- The pan surface temperature is assumed to be 200^{o}c (the average)
- The effect of conductive heat transfer of fin like structure is assumed in significant.
- The oil gallery is well insulated from below and the sides using ash insulation system, and convection and radiation losses are negligible
- Assume that the required power for injera baking is 2.2kw

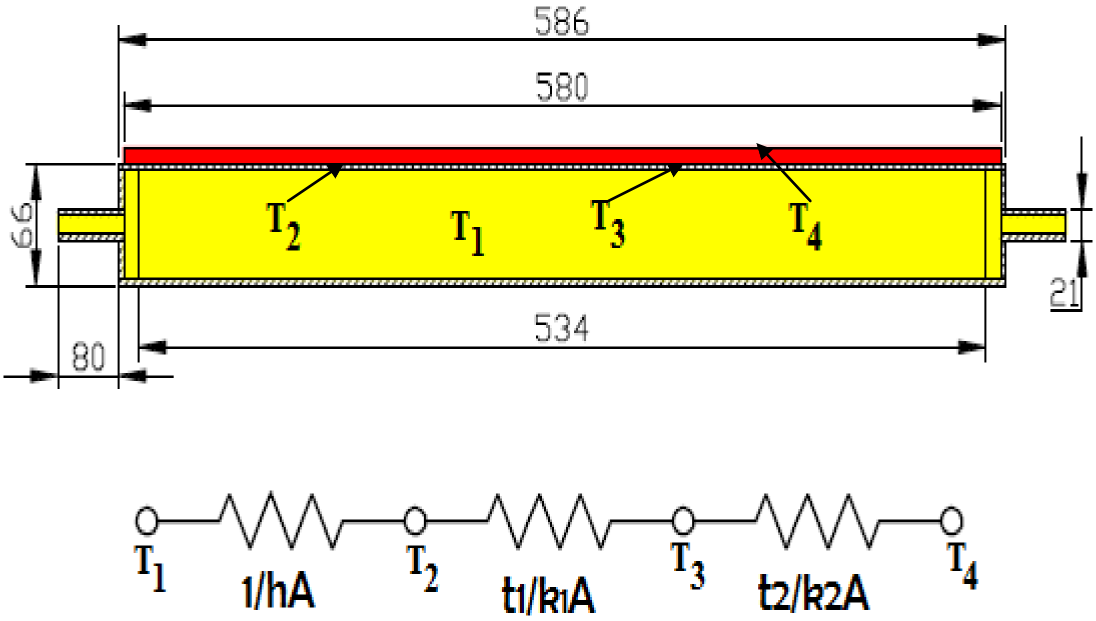

Figure 3.7: Heat flow from the hot oil to the baking pan surface and its thermal resistance model.

Where:

T_1- Hot oil temperature (^oC)

T_2- Supporting plate lower surface temperature (^oC)

T_3- Baking pan lower surface temperature (^oC)

T_4- Baking pan upper surface temperature =200^oC

Thus the rate of heat transfer from hot oil to baking pan surface can also be expressed as follows by combining Equations (2.3) and (2.6) we have:

$$\dot{Q} = \frac{T_1 - T_4}{\left[\frac{1}{hA} + \frac{t_1}{k_1 A} + \frac{t_2}{k_2 A}\right]} \tag{3.4}$$

Where:

h- Convection heat transfer coefficient

t_1- Supporting sheet metal thickness = 0.003m

t_2- Thickness of cooking pan = 0.008m

k_1- Thermal conductivity of supporting plate carbon steel 1%C, k_1 = 43w/m.K

k_2- Thermal conductivity of cooking pan = 0.8w/m.K

A- Area of the pan and is calculated as follows;

$$A = \frac{\pi D^2}{4} \tag{3.5}$$

$$A = \frac{\pi\, 0.58^2}{4} = \underline{\underline{0.2642}} m^2$$

Convection Heat transfer Coefficient: Convection coefficient, *h*, is the measure of how effectively a fluid transfers heat by convection. To determine the convection heat transfer coefficient first one should determine the Nusselt number and Raleigh number, using Churchill and Chu equations [3, 2].

$$h = \frac{Nu\, k}{L} \tag{3.6}$$

$$Nu = \left\{ 0.825 + \frac{0.387\, Ra^{1/6}}{\left[1 + (0.492/\mathrm{Pr})^{9/16} \right]^{8/27}} \right\}^2$$

Where:

k- Thermal conductivity

Pr-Pradtel number

Nu - Nusselt number

Ra - Raleigh number

Assuming that the velocity is small and using the formula for free convection; The Raleigh number which is a dimensionless parameter can be calculated from:

$$Ra = \frac{g\, \beta\, (T_o - T_\infty)\, L^3}{v\, \alpha} \tag{3.7}$$

Where: $\alpha = k / \rho\, c$ – Thermal diffusivity

$v = \mu / \rho$ - Kinematics viscosity

β - Coefficient of thermal expansion

To – Operating heat transfer fluid temperature assume to be about 300^{o}c

T ∞ - Temperature of the environment = 20^{o}c

L- Effective length of the heat transfer surface for a horizontal circular plate and can be calculated from the relation:

$$L = \frac{A}{P} \tag{3.8}$$

Where: A-Surface area

P- Perimeter of the heat transfer pan surface

$$L = \frac{A}{P} = \frac{\pi\, D^2}{4\, \pi\, D} = \frac{0.58}{4} = 0.145m$$

From heat transfer fluid properties data for initial guess of heat transfer coefficient at the film temperature ($T_{f)}$; take the temperature (T_o) = 300^{o}C and the atmospheric temperature (T_∞) is assumed to be 20^{o}C.

$$T_f = \frac{T_o + T_\infty}{2} \tag{3.9}$$

$$T_f = \frac{300+20}{2} = 160^o C$$

Using this film temperature the properties of the heat transfer oil from the Property Design Data table (2.2) for Shell Thermia Oil B are:

Thermal conductivity of oil; $k = 0.124 W/m.K$

Density of oil; $\rho = 774 kg/m^3$

Kinematics viscosity; $v = 2.85x10^{-6} m^2/s$

Specific heat capacity; $c \ = 2.40 kJ/kg.k$

Coefficient of thermal expansion, $\beta = 0.0008/^o C$

Pradtle number; $p_r \ = 29$

Thermal diffusivity (α) $= \dfrac{k}{\rho\, c}$ (3.10)

$$\alpha = \frac{0.124(w/m)k}{774 kg/m^3 x 2.4(kJ/kg)k} = 6.67x10^{-8} m^2/s$$

Now substituting the values of the oil properties; the Raleigh number and the Nusselt number becomes:

$$Ra = \frac{g\ \beta\,(To - T\infty)}{v\,\alpha} \tag{3.11}$$

$$Ra = \frac{9.81x0.0008(300-20)x0.145^3}{2.85x10^{-6}x6.67x10^{-8}} = \underline{\underline{3.525x10^{10}}}$$

$$Nu = \left\{0.825 + \frac{0.387\ Ra^{1/6}}{\left[1+(0.492/\Pr)^{9/16}\right]^{8/27}}\right\}^2$$

$$Nu = \left\{ 0.825 + \frac{0.387x(3.525x10^{10})^{1/6}}{\left[1+0.492/29)^{9/16}\right]^{8/27}} \right\}^2$$

$$Nu = \underline{\underline{586}}$$

Thus the average convection heat transfer coefficient can be calculated as:

$$h = \frac{Nu\ k}{L} = \frac{586x0.124}{0.145} = 501W/m^2.K$$

Now the required oil temperature can be calculated as

$$\frac{2.2kW/m^2}{0.2642} = \frac{T_1 - 473}{\frac{1}{501} + \frac{0.003}{43} + \frac{0.008}{0.8}}$$

$$T_1 = 573.5 = \underline{\underline{300.5}}^o C$$

Now for the next iteration we have the heat transfer oil temperature (To) 300.5°c and the environment temperature (T_∞) is 20°c; then the film teperature (T_f) can be found from:

$$T_f = \frac{T_o + T_\infty}{2}$$

$$T_f = \frac{300.5 + 20}{2} = 160.25^o C$$

The properties of the heat transfer oil are evaluated at this film temperature using linear interpolation on Property Design Data table (2.2), for Shell Thermia Oil B; and the values of the properties are as follows.

Thermal conductivity of oil; $k = 0.124W/m.K$

Density of oil; $\rho = 772kg/m^3$

Kinematics viscosity; $v = 2.862x10^{-6}m^2/s$

Specific heat capacity; $c \ = 2.392kJ/kg.k$

Coefficient of thermal expansion, $\beta = 0.0008/^o C$

Pradtle number; $p_r \ = 29.6$

Thermal diffusivity (α) $= \dfrac{k}{\rho\, c}$

$$\alpha = \frac{0.124\,(W/m)K}{772kg/m^3 x 2.392(kJ/kg)k} = 6.715x10^{-8}m^2/s$$

Now substituting the oil property values; the Nusselt number and the Raleigh number becomes:

$$Ra = \frac{g\,\beta\,(T_o - T_\infty)\,L^3}{v\,\alpha}$$

$$Ra = \frac{9.81x0.0008(299.2-20)x0.145^3}{2.862x10^{-6}x6.721x10^{-8}} = \underline{\underline{3.495x10^{10}}}$$

$$Nu = \left\{0.825 + \frac{0.387\,Ra^{1/6}}{\left[1+(0.492/\Pr)^{9/16}\right]^{8/27}}\right\}^2$$

$$Nu = \left\{0.825 + \frac{0.387x(3.495x10^{10})^{1/6}}{\left[1+(0.492/29.6)^{9/16}\right]^{8/27}}\right\}^2$$

$$Nu = \underline{\underline{499.009}}$$

Thus the average convection heat transfer coefficient can be calculated as:

$$h = \frac{Nu\,k}{L}$$

$$h = \frac{499.009x0.124}{0.145} = \underline{\underline{426.738}}W/m^2.K$$

$$\frac{2.2kw/m^2}{0.2642} = \frac{T_1 - 473}{\dfrac{1}{426.738} + \dfrac{0.003}{43} + \dfrac{0.008}{0.8}}$$

$$T_1 = 576.4 = \underline{\underline{303.4^oC}}$$

Now again for the next iteration we have the heat transfer oil temperature (To) is 302.2^{o}C and the environment temperature is (T_∞) is 20^{o}C; then the film teperature (T_f) can be found from:

$$T_f = \frac{T_o + T_\infty}{2}$$

$$T_f = \frac{303.4 + 20}{2} = 161.5^o C$$

The properties of the heat transfer oil are evaluated at this film temperature using linear interpolation on Property Design Data table (2.2) for Shell Thermia Oil B; and the values of the properties as follows.

Thermal conductivity of oil; $k = 0.124 W/m.K$

Density of oil; $\rho = 771 kg/m^3$

Kinematics viscosity; $v = 2.805 x 10^{-6} m^2/s$

Specific heat capacity; $c \ = 2.397 kJ/kg.k$

Coefficient of thermal expansion, $\beta = 0.0008 /^o C$

Pradtle number; $p_r = 29.24$

Thermal diffusivity (α) $= \dfrac{k}{\rho\, c}$

$$\alpha = \frac{0.124 (W/m)K}{771 kg/m^3 x 2.397 (kJ/kg)K} = 6.7 * 10^{-8} m^2/s$$

Now substituting the oil property values; the Nusselt number and the Raleigh number becomes:

$$Ra = \frac{g\ \beta\ (T_o - T_\infty) L^3}{v\ \alpha}$$

$$Ra = \frac{9.81 x 0.0008 (303.4 - 20) x 0.145^3}{2.805 x 10^{-6}\ x\ 6.7 x 10^{-8}} = \underline{\underline{3.608 x 10^{10}}}$$

$$Nu = \left\{ 0.825 + \frac{0.387\ Ra^{1/6}}{\left[1 + (0.492/\Pr)^{9/16} \right]^{8/27}} \right\}^2$$

$$Nu = \left\{ 0.825 + \frac{0.387\,(3.608x10^{10})^{1/6}}{\left[1+(0.492/29.24)^{9/16}\right]^{8/27}} \right\}^2$$

$$Nu = \underline{\underline{503.95}}$$

Thus the average convection heat transfer coefficient can be calculated as:

$$h = \frac{Nu\,k}{L}$$

$$h = \frac{503.95x0.124}{0.145}$$

$$h = 430.966W/m^2.K$$

Now the required oil temperature can be calculated as:

$$\frac{2.2kW/m^2}{0.2642} = \frac{T_1 - 473}{\frac{1}{430.966} + \frac{0.003}{43} + \frac{0.008}{0.8}}$$

$$T_1 = 576.2 = \underline{\underline{303.2^o C}}$$

The error between the assumed value and the final value is:

$$error = \frac{T_i - T_f}{T_f}$$

$$error = \frac{303.4 - 303.2}{303.2} = 0.00066$$

Error = 0.066%

Thus the error is neglegible and it is possible to stop the iteration with the optimum oil temperature (T_1) in the oil gallery $303.2^o C$

Design of Ash housing wall

The ash housing wall is used to enclose the ash insulating the oil gallery. It was constructed from 1mm thick carbon steel sheet metal.

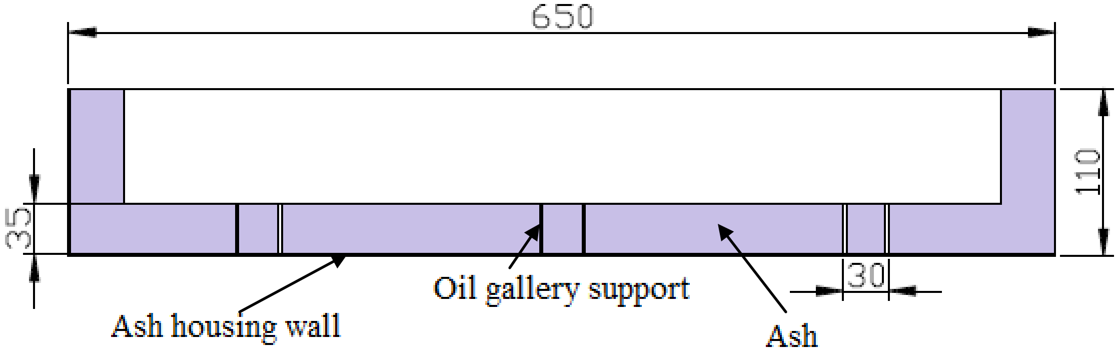

Figure 3.8: Ash hosing wall showing Ash and oil gallery stand.

3.5.2 Design of Baking pan Assembly Supporting Legs

To **c**arry the whole weight of pan assembly with oil in the oil gallery; four supporting legs were welded to the supporting ring of the baking pan assembly as shown in the figure bellow.

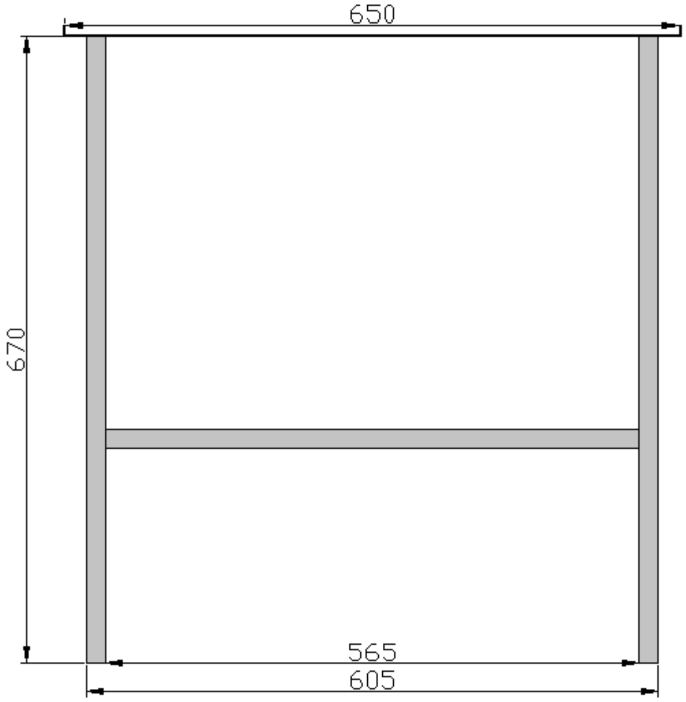

Figure 3.9: Baking pan assembly support legs

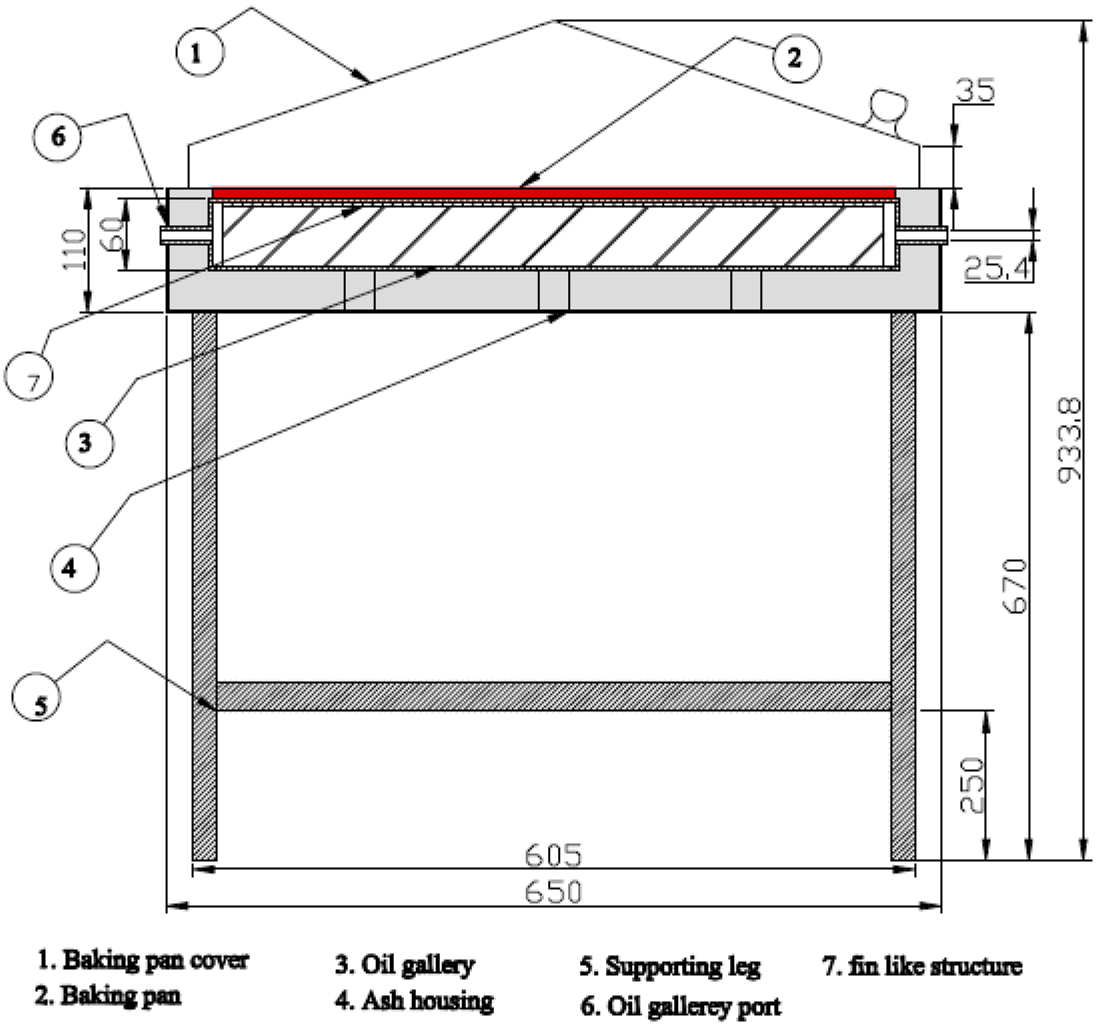

Figure 3.10: Baking pan assembly of the laboratory model

3.5.3 Design of Oil Heating and Storage System

The basic components of Heating and Storage System are storage tank and the electrical heater. In addition to this ash insulation and ash cover wall were included. The storage tank was manufactured from 3mm thick steel metal and the ash cover wall was manufactured from 1mm thick steel metal by welding. As the oil was being heated its volume increased but its density decreased, thus the volume of the storage tank was designed by considering thermal expansion of the heat transfer oil.

Design of The oil storage and heating tank

An expansion tank is necessary to allow for the change in fluid volume upon heating. The volume of mineral oil (Thermia oil B) at 300°C is about 20 per cent greater than at room temperature. In this particular case instead of expansion tank the volume of the tank was designed by considering the amount expected to expand at the optimum temperature, so that the tank was large enough to accept the total heat expansion within its own dimensions.

The density of the oil changes with change in temperature and the final density (ρ_f) of the fluid can be expressed as:

$$\rho_f = \frac{\rho_o}{[1+\beta\,(T_f - T_o]} \qquad (3.12)$$

Where:

ρ_f - Final density (kg/m^3)

ρ_0 - initial density (kg/m^3) of the oil is equal to 863kg/m^3 at T = 20^{o}C

β - Volumetric temperature expansion coefficient = 0.0008 (m^3/m^{3o}C)

T_f - Final oil temperature (^{o}C) =320^{o}C

T_o – Initial oil temperature (^{o}C) =20^{o}C

Assume that the initial fluid temperature is the ambient temperature in the surrounding and be 20^{o}C, thus the final density (ρ_f) will be:

$$\rho_f = \frac{863}{[1+0.0008(320-20]} = 695.968 kg/m^3$$

Since the mass of the fluid does not change with temperature, that means mass is constant the volume of the fluid at a temperature of 320^{o}C can be calculated from

$$m_f = m_o \Rightarrow V_o\rho_o = V_f\rho_f \qquad (3.13)$$

$$V_f = \frac{m}{\rho_f} = \frac{Vo\,x\,\rho_o}{\rho_f}$$

Where: v_f - final volume of the heat transfer fluid

v_o – Initial volume the heat transfer fluid

ρ_f- Final fluid density

ρ_o - Initial fluid density

m - Mass of heat transfer fluid

Considering the volume expansion of the oil, assume that volume of the heat transfer fluid in the storage tank is 35 litters.

$$V_f = \frac{m}{\rho_f} = \frac{V_o\,\rho_o}{\rho_f}$$

$$V_f = \frac{0.035x863}{695.968} = 0.0434m^3 = \underline{\underline{43.4}}liters$$

Considering thermal expansion of the heat transfer fluid the volume of the storage tank should have the dimensions: Length (L) = 400mm, Height (h) = 500mm and Width (w) = 230mm.

Even though some volume in the storage tank was occupied by the coil of the electrical heaters; the flange for the electrical heaters were additional volumes for the oil storage and can substitute the volume occupied by the coil of electrical heaters; therefore the volume of the storage tank (V_{st}) should be:

$$V_{st} = h\,L\,w \tag{3.14}$$

$$V_{st} = 500mm\,x\,400mm\,x\,230mm$$

$$V_{st} = 0.046m^3 = \underline{\underline{46}}liters$$

That is the storage and heating tank was manufactured to occupy a volume of about 46 liters of the heat transfer oil.

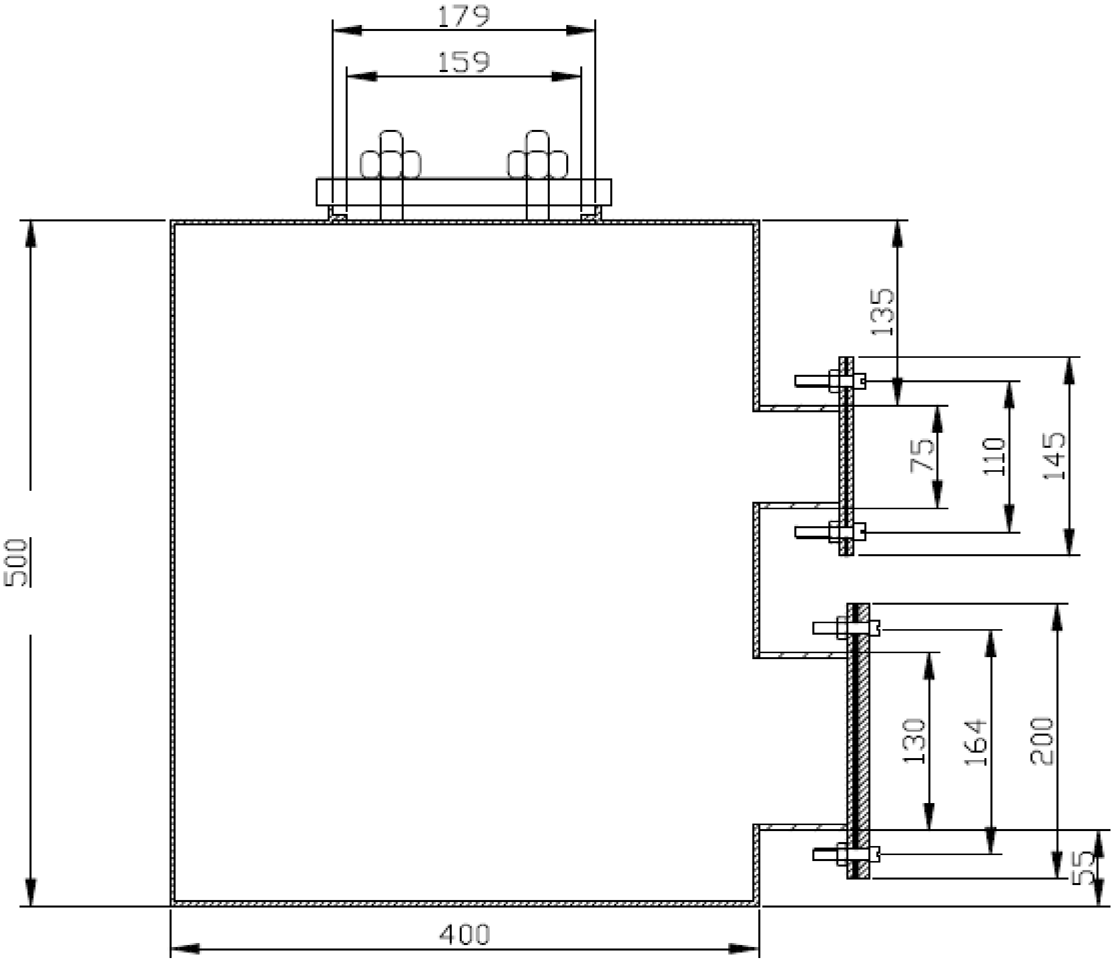

Figure 3.11: Oil storage and heating tank

3.5.4 The Electrical Heater

An electric heater is an electrical appliance that converts electrical energy into heat. The heating element inside every electric heater is simply an electrical resistor, and works on the principle of Joule heating; an electric current through a resistor converts electrical energy into heat energy. From a heat generation standpoint, electric heaters heat a fluid like a traditional heat exchanger by using electrical heat.

The heat transfer fluid heater was directly immersed into the heat transfer oil with the most commonly used element sheath being copper and is mounted in the form of a flanged immersion heater type. It is assumed that the electric heater is submerged in the heat transfer fluid at atmospheric pressure and the current is increased until the temperature of the oil is 320^{o}C.

Figure 3.12: The Electrical heater

Electric Power Required for the Electrical Heater

The heat transfer from the heater to the oil is through convection heat transfer.

Assumption for the analysis:

- The system is stationary and therefore there is no change of kinetic and potential energy, that is $\Delta KE = \Delta PE = 0$
- The pressure remains constant during the process
- The electric power is supplied continuously for about 60 minutes

By applying energy balance to a control surface of the storage tank; [17, 14]

$$\dot{Q}_{in} - \dot{Q}_{loss} = m\,c\,\Delta T \tag{3.15}$$

The heat loss (Q_{loss}) can be calculated as:

$$\dot{Q}_{loss} = \frac{T_o - T_\infty}{R_t} = \frac{T_o - T_\infty}{\dfrac{1}{h_1 A} + \dfrac{t_1}{k_1 A} + \dfrac{t_2}{k_2 A} + \dfrac{1}{h_2 A}}$$

Where:

$\dot{Q}_{in}$ – Electrical energy supplied by the heater

$\dot{Q}_{loss}$ – Heat loss to the atmosphere

T_o – The film oil temperature during heating up process

T_∞ – ambient temperature

R_t– Thermal resistance

h_1- Convection heat transfer coefficient of heat transfer fluid = 490W/m^2.k

h_2- Convection heat transfer coefficient of ambient air = 10W/m^2.k

k_1 – Thermal conductivity of housing material (carbon steel) = 43W/m.k

t_1- Thickness of housing sheet metal = 0.003m

k_2 – Thermal conductivity of Ash insulating material = 0.14 W/m.k [1].

t_2- Thickness of insulating material = 0.035m

A-Heat transfer surface area (A) exposed to heat loss

Considering the area of the bore (A_b) for the purpose of electrical heaters and is calculated as:

$$A_b = \frac{\pi}{4} D_1^2 + \frac{\pi}{4} D_2^2$$

Where; D_1-is the diameter of bore one

D_2 – is the diameter of bore two

$$A_b = \frac{\pi}{4} 0.075^2 + \frac{\pi}{4} 0.13^2 = \underline{\underline{0.018}} m^2$$

Hence; area (A) exposed to heat loss is:

$$A = 2(0.5 \times 0.4) + 2(0.5 \times 0.23) + 2(0.4 \times 0.23) - A_b$$

$$A = 0.814 - 0.018 = \underline{\underline{0.796}} m^2$$

$$R_t = \frac{1}{490x0.796} + \frac{0.003}{43x0.796} + \frac{0.035}{0.14x0.796} + \frac{1}{10x0.796} = \underline{\underline{0.442}} k/W$$

Thus; the heat supplied by electrical heater during heating up is:

$$\dot{Q}_{in} = \dot{Q}_{loss} + \frac{m \ c \ \Delta T}{\Delta t}$$

$$\dot{Q}_{in} = \dot{Q}_{loss} + \frac{V_o \ \rho \ c \ \Delta T}{\Delta t}$$

But the values of density (ρ) and specific heat (c) of the heat transfer oil at the film temperature (T_f) from the design data table are:

ρ = 778kg/m^3

c = 2.4kJ/kg.k

Hence, $\dot{Q}_{in} = 0.34 + \frac{0.035\,x\,778\,x\,2.4\,x\,(320-20)}{60\,x\,60} = \underline{\underline{5.786}}kW$

This power (5.786kW) was the power required to heat up the oil in the storage tank without pumping; during pumping there was loss in the pipes and through other parts of the system, so to recover the loss and to be safe it is better to increase the power of the heater to about 6 to 7 kW.

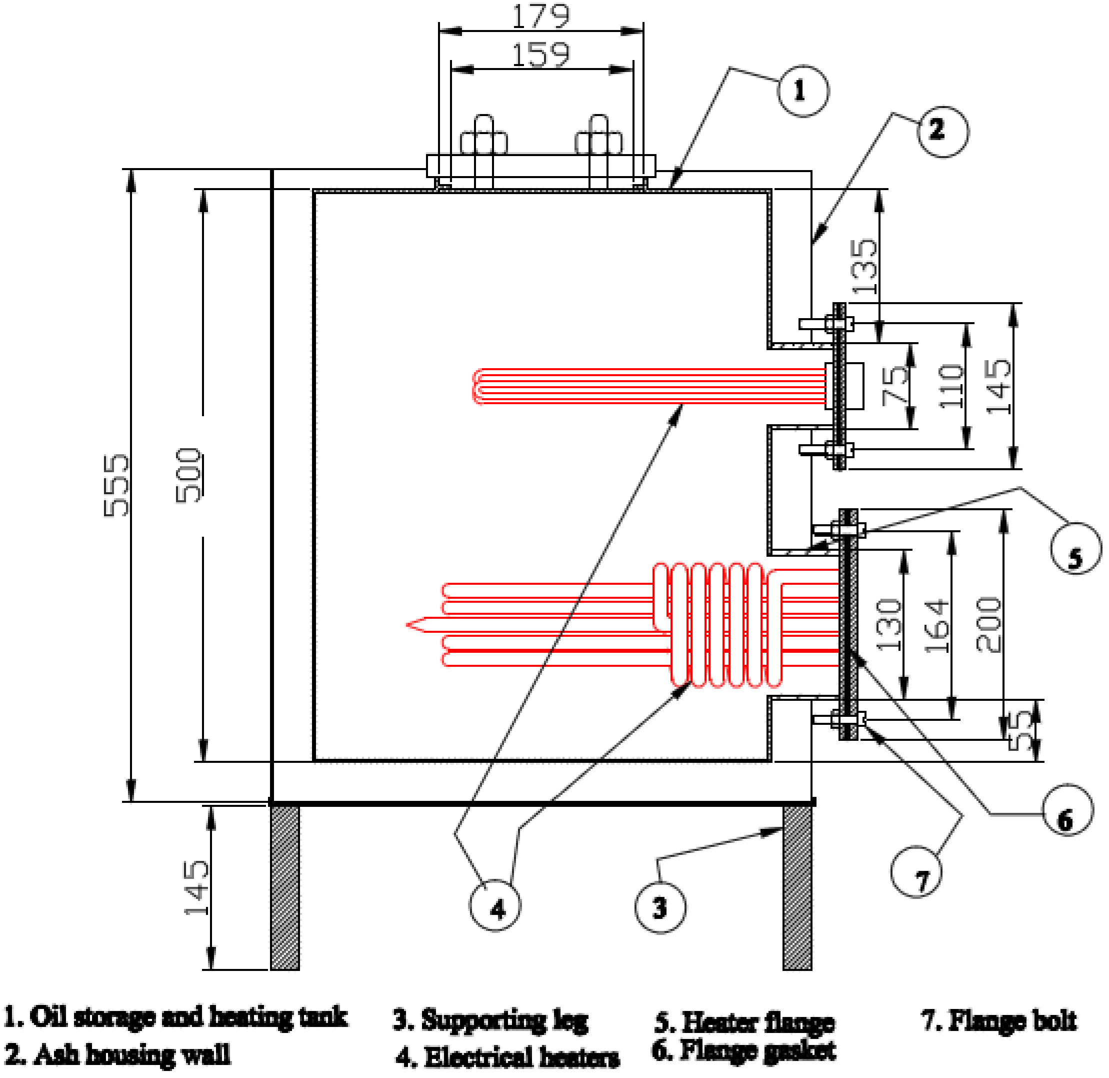

Figure 3.13: Oil storage and heating tank assembly of the laboratory model

3.5.5 Design of Piping System

This system comprises supply line from storage tank to pump and then from pump to inlet port of the oil gallery, return pipe form oil gallery to storage tank, leaked oil return line from pump to storage tank and the electrical driven pump. The heat transfer fluid was pumped from the storage tank to the inlet side of the cooking baking pan assembly using the pump through ½ inch galvanized steel pipe.

To protect heat loss from the pipes the piping system was insulated using fiber glass insulation system. The total length of piping system was about 4.50m; with 2.10m supply pipe, 1.40m main return pipe and about 1.0m excess (leakage) return pipe.

3.6 Machines and Equipments used in the Manufacturing of the System Components

The manufacturing process was performed using the design and the production drawings.
The components of the system that were manufactured are as listed below;

- The pan cover
- The heat transfer oil gallery
- Baking pan assembly Supporting legs
- Oil storage tank
- The oil pump housing
- The ash insulation system housing

To manufacture and finally to assemble these manufactured components; the following machines and tools were used in the workshop.

- Sheet metal cutting or shearing machine
- Bending machine
- Rolling machine
- Milling machine
- Lathe machine
- Drilling machine
- Arc welding machine
- Seam welding machines
- Hack saws
- Grinding machine
- Oxyacetylene welding set
- Metering and marking tools
- Different kinds of wrenches
- Different kinds of pliers
- Different kinds of files
- Different kinds of hammers

3.6.1 Manufacturing of Backing Pan Cover

The pan cover was manufactured with the specified dimension from aluminum coated flat iron sheet metal; cutting, folding and steam welding process were involved during the manufacturing of the pan cover.

Procedure:

- The aluminum coated flat iron sheet metal was cut with the required shape and size
- The sheet metal was folded with the required shape
- The folded sheet was welded with seam welding machine to give the required shape and size.
- The hinge type system was provided and bolted to the gallery housing so as possible to swing during the opening and closing of the pan cover.

3.6.2 Manufacturing of Heat Transfer Fluid Gallery

The oil gallery was manufactured with the specified dimension from 3mm thick carbon steel sheet metal; measuring, cutting, rolling, drilling, grinding and welding process were involved during the manufacturing of the oil gallery.

Procedures held in the manufacturing of the heat transfer fluid gallery:

- Selection of the sheet metal which fit the required thickness and material properties
- The selected sheet metal was cut into different pieces according to the determined size using cutting machine.
- Fin like structures were welded to the lower and upper sides of the gallery to hinder direct oil flow and to give strength to the oil gallery.
- Inlet and outlet ports were drilled to the required size and at the required position.
- The side wall sheet metal was rolled using rolling machine and welded to the upper and lower sides of the gallery frame.
- Then the inlet, outlet fittings were welded to the inlet and outlet ports.

3.6.3 Manufacturing of Supporting Legs of the Baking Pan Assembly

The supporting legs were manufactured from hollow section RHS with 20x20mm dimensions with a bed like structure giving strength to the legs and also used for holding baking accessories.

Cutting the hollow section RHS in to the required size, grinding and welding to the supporting ring process were involved during the manufacturing of the supporting legs.

Procedure:

- The four legs were cut from hollow section RHS with the designed size and dimension.
- cutting and rolling the supporting ring from 2mm thick sheet metal with 200mm height.
- The four legs were welded on equal distance around the supporting ring.

3.6.4 Manufacturing of Oil Storage Tank

The storage tank has a capacity of 46 liters. It was manufactured from 3mm thick Carbon steel sheet metal; Cutting, folding, drilling, grinding and welding process were involved during the manufacturing of the oil storage tank.

*Procedures held in the manufacturing of the oil storage tank***:**

- Selection of the sheet metal which fit the required thickness and material properties
- The selected sheet metal was cut into different pieces according to the determined size using cutting machine.
- Inlet, outlet, and electric heater mounting holes were drilled to the required size and at the required position.
- The sheet metal was bended or folded using bending machine and welded to give the required frame.
- Then, inlet and outlet fittings were welded and the heater flange was bolted to the tanker flange. The mounting frame of the storage tank was also manufactured with the same procedure as that of the baking pan supporting leg except the dimensions.

3.6.5 Manufacturing of the Pump Housing

The purpose of the pump housing is:

- To enclose the pump so that to protect the pump from dirty and dust.
- To protect oil wastage because the pump has leakage purposely for lubrication system so that the leaking oil from the pump was collected and was returned to oil storage tank.

It was manufactured from 3mm thick Carbon steel sheet metal; Cutting, folding, drilling, grinding and welding process were involved during the manufacturing of the pump housing.

3.6.6 Manufacturing of the Heat Transfer Fluid Pipe Lines

The pipe lines were manufactured from ½ inch steel pipe; and Cutting, bending, drilling, grinding threading and welding process were involved during the manufacturing of the fluid pipe lines.

*Procedures***:**

- Selection of the fluid pipe
- The selected pipe line was cut into different pieces according to the determined length.
- Threads or flanges were made to the pipe lines to fit to the required part.
- The pipe lines were bended to the required shape using bending machine and then treaded, welded or bolted to the required ports.

CHAPTER FOUR

ANALYSIS OF THE OIL CIRCULATION SYSTEM

4.1 Insulation System

Insulation is defined as a material or combination of materials, which retard the flow of heat and is used in heating, air conditioning and refrigeration systems to insulate piping, ducts, vessels and equipments in order to conserve energy, prevent surface condensation and control heat input to the contained fluids. Generally the thermal insulation can be used to control heat flow in wide temperature ranges when the proper material thickness is selected [11].

4.1.2 Types of Insulation Materials

There are different types of insulation materials among those; fiber glass, cellular glass, foamed plastic and calcium silicate are the most commonly used materials. In the laboratory model of this particular case the following two insulation systems were used:

***Fiberglass insulation system*:** Fiberglass insulation is fibrous glass, made either plain or with a heat-resistant binder in order for the fiberglass to hold its shape. Fiberglass is the most popular insulation, and it comes in many forms. In the form most commonly used for pipe, it is molded and shaped into semicircular sections. The binder is the critical factor for the ultimate temperature for which it can be used.

Fiberglass by itself is not strong enough to stay permanently on a pipe without falling off in layers. Since fiberglass is porous, there is no way to seal it to prevent water vapor from flowing freely from the air to the pipe and then condensing on the pipe and saturating the fiberglass.

In addition, there is no way to finish fiberglass that would be considered pleasant to look at. For these reasons, a covering, or jacket, must be added to protect it from physical damage, allow it to be firmly and permanently attached to the pipe, and prevent the penetration of water vapor. Fiberglass is recommended for temperatures up to 422^oC. A high temperature, flexible blanket can be used with temperatures up to 530^oC [7].

Ash insulation system: Ash is a by-product (waste) from the combustion of fuel wood especially in the preparation foods with the largest share in baking Injera. In some cases ash from fuel wood is used as nutrient for plants because it improves the fertility of soil.

In this particular case the wood ash was used for insulation system because:-

- Ash is completely burned material, so should have very low thermal conductivity
- Ash is a waste material, so there is no cost for it and available everywhere locally
- There is no fire hazard and is not toxic, so there is no problem of safety.

In more general case in the country sides; which is difficult to get matches easily for firing to prepare food early in the morning; what mothers used is that; they cover the fired charcoal with wood ash estimated about 3 to 5 cm thickness in the evening to get the charcoal with its fire early in the morning so that they can prepare food, this shows us that the ash conserves the energy in the charcoal, by protecting entering cold air from outside in to the fired charcoal and by protecting energy loss from the charcoal to the outside environment. So the idea of using ash as insulation material in this particular case is from this concept, and actually it was a good insulator.

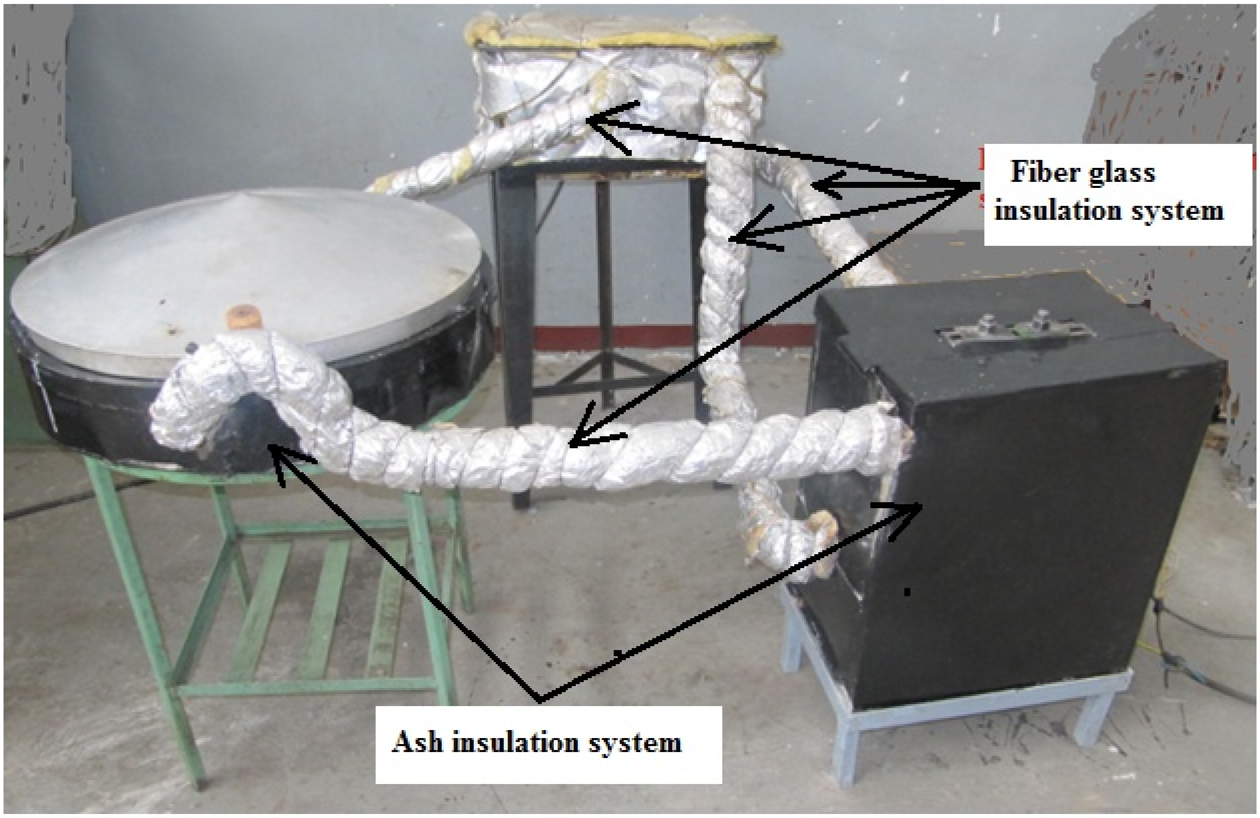

Figure 4.1: Fiber glass and Ash insulation systems of the laboratory model

Thus, in order to minimize heat loss due to cold air circulation over the surface and so, to save energy; in this particular case; the storage tank and the heat transfer fluid gallery were well insulated using 3.5cm thick ash insulation system and the supply, return and leakage return line pipes were well insulated using about 3.5cm thick fiber glass insulation system.

4.1.1 Insulation Material and Thickness Selection

The general criteria needed to make a choice among various insulation materials are as follows:

- The reason insulation is needed
- Service temperature expected
- The location where insulation will be installed
- Accessibility for the insulated pipe
- Installed cost of the complete insulating system

In selection of the thickness of the insulation material it is advisable to select the correct thickness. As thickness of insulation is increased, the cost of material and installation goes up. On the other hand, the energy cost savings also goes up, but at a slower rate of increase than the cost of materials and installation. At some point, then, the total cost, which is the sum of the lost energy cost and the material cost, reaches a minimum point, that amount of insulation is called the economic thickness.

4.1.3 Benefits of Insulation System

Energy Savings*:* Substantial quantities of heat energy are wasted because of un-insulated heated surfaces. Properly designed and installed insulation systems will immediately reduce the need for energy and benefits to industry include enormous cost savings, improved productivity, and enhanced environmental quality.

Process Control: By reducing heat loss or gain, insulation can help maintain process temperature to a pre-determined value or within a predetermined range.

Personnel Protection: Thermal insulation is one of the most effective means of protecting workers from second and third degree burns resulting from skin contact with surfaces of hot piping and equipment operating at higher temperatures. Insulation reduces the surface temperature of piping or equipment to a safer level, resulting in increased worker safety and the avoidance of worker downtime due to injury.

Fire Protection: Used in combination with other materials, insulation helps provide fire protection in fire stop systems designed to provide an effective barrier against the spread of flame, smoke, and gases at penetrations of fire resistance rated assemblies by ducts, pipes, and cables.

4.2 Head Losses

The fluid in a typical piping system passes through various fittings, valves, bends, elbows, tees, inlets, exits, enlargements, and contractions in addition to the pipes. These components interrupt the smooth flow of the fluid and cause additional losses because of the flow separation and mixing they induce.

In a typical system with long pipes, these losses are minor compared to the total head loss in the pipes (the major losses) and are called minor losses**.** Although this is generally true, in some cases the minor losses may be greater than the major losses.

Static head is simply the difference in height of the supply and destination reservoirs and it is independent of flow; and the Static head (Hs) in our case has a value of about 0.5m.

Friction head (h_f) is the sum of major and minor head friction losses on the liquid being moved in pipes, valves and equipment in the system. Friction tables are universally available for various pipe fittings and valves.

Table 4-1: Loss factor for piping system

Types of fitting	Factor (K)	Quantity
90^o elbow	1.75	5
Gate valve	0.27	0
Check valve	0.27	0
Slightly rounded bend	0.12	3
180^o return bend	0.2	1

4.2.1 Major Head Loss

The Major Head loss (h_j) can be calculated from:

$$h_j = \frac{f\ v_o{}^2\ L}{2\ g\ D_p} \tag{4.1}$$

Where: Vo- Fluid flow velocity and is assumed to be 0.5m/s

L- Total effective flow length = 4.5m

D_p- Pipe diameter = 12.7mm

v – Viscosity of the fluid = 2.02 x 10^{-6} m^2/s

f- Friction factor and can be read from moody chart using Reynolds number Re and relative roughness factor $\frac{\varepsilon}{D_p}$

The Reynolds number (Re)

$$\mathrm{Re} = \frac{V_o\ D_p}{v} \tag{4.2}$$

$$\mathrm{Re} = \frac{0.5x0.0127}{2.02x10^{-6}} = 2886.4$$

This value is between 2300 and 4000 which means it is in the transition from laminar to turbulent type of flow; since it is nearer to 2300 than 4000 it is better to use laminar flow equation to find the friction factor (f) and can be calculated as:

$$f = \frac{64}{\mathrm{Re}} \tag{4.3}$$

$$f = \frac{64}{2886.4} = 0.02$$

$$h_j = \frac{0.02x0.5^2\ x4.5}{2x9.81x0.0127} = 0.09m$$

4.2.2 Minor Head Loss

The Minor Head loss (Hn) can be calculated as:

$$h_n = \sum \frac{k\, v_o^{\ 2}}{2\, g} \tag{4.4}$$

Where: K = Minor loss coefficient read from the above table.

$$h_n = \frac{[(5x1.75)+(3x0.12)+(1x0.2)]0.5^2}{2x9.81} = 0.123m$$

The friction head (h_f) is:

$$h_f = h_j + h_n$$

$$h_f = 0.143 + 0.019 = 0.162m$$

Hence; total head loss (Ht) is:

$$h_t = h_j + h_n + h_s$$

$$h_t = 0.09 + 0.123 + 0.5 = 0.713m$$

4.2.3 The Pressure Drop

A quantity of interest in the analysis of pipe flow is the *pressure drop (* ΔP *)* since it is directly related to the power requirements of the fan or pump to maintain flow.

$$\frac{dp}{dx} = \frac{p_2 - p_1}{L}$$

The pressure drop for a fully laminar flow can be expressed as:

$$\Delta P = P_1 - P_2 = \frac{8\, v\, L\, V_o}{R^2} = \frac{32\, v\, L\, V_o}{D^2} \tag{4.5}$$

Where: ΔP - is pressure drop

v - is viscosity of the oil

L - Length of the pipe

V_o - Average velocity of the oil

$D = 2R$ -internal diameter of the pipe

The pressure drop is proportional to the viscosity (μ) of the fluid, and ΔP would be zero if there were no friction. Therefore, the drop of pressure from P_1 to P_2 in this case is due to entirely viscous effects, and Eq. 4.4 represents the pressure loss, when a fluid of viscosity (μ) flows through a pipe of constant diameter (*D)* and length *L* at average velocity V_o .

In practice, it is found convenient to express the pressure loss for all types of fully developed internal flows (laminar or turbulent flows, circular or noncircular pipes, smooth or rough surfaces, horizontal or inclined pipes) as:

$$\Delta P = f\,\frac{L\,\rho V_o^{\,2}}{2\,D}$$

Where: f - is friction factor

$$\Delta P = 0.02\,\frac{4.5x774x0.5^2}{2x0.0127} = 0.686bar$$

4.3 Oil Pumping and Circulation System

In a pumping system, the objective, in most cases, is either to transfer a liquid from a source to a required destination or to circulate liquid around a system. A pressure is needed to make the liquid flow at the required rate and this must overcome head losses in the system. The main components of a pumping system are:

- Pumps
- Prime movers: electric motors or diesel engines
- Piping, used to carry the fluid from some source to some destinations
- Valves, used to control the flow in the system
- Other fittings, controls and instrumentation
- End-use equipment, which have different requirements

To transfer the heat transfer fluid from the heating and storage tank to the oil gallery and then to circulate in the system an electrically driven pump was used in this laboratory model.

Once the pressure loss (or head loss) is known, the required pumping power *to overcome the pressure loss* is determined.

The pump power can be calculated from the relation:

$$P_p = \frac{\rho\, H_t\, g\, Q_v}{\eta_p} \tag{4.6}$$

Where: Pp- Pump Power

Qv- Volume flow rate

Ht- Total head

η_p - pump efficiency

The volume flow rate (Qv) through the pipe can be found from the relation:

$$Q_v = A_p\, V_o \tag{4.7}$$

$$Qv = \frac{\pi}{4} D_p{}^2\, V_o$$

$$Q_v = \frac{\pi}{4} 0.0127^2\, x 0.5 = 6.334 x 10^{-5} m^3 / s$$

Assuming the efficiency of the pump to be 60%:

$$P_p = \frac{774 x 0.804 x 9.81 x 6.334 x 10^{-5}}{0.6}$$

$$P_p = 0.645W$$

Therefore power input in the amount of 0.645W is needed to overcome the frictional losses in the flow.

It is difficult to get a pump used for this high temperature application according to the design in the local market, so it has tried to use a gear drive type oil pump from some truck's engine by modifying the gear on the pump input shaft to adjust the velocity of the fluid.

To know the RPM of the pump used for this application we can use the relation:

$$\frac{N_m}{N_p} = \frac{G_p}{G_m} \tag{4.8}$$

Where: N_m - RPM of electrical motor shaft =1425

N_p - RPM of oil pump shaft =?

G_m - Number of teeth on electrical motor shaft gear = 21

G_p - Number of teeth on oil pump shaft gear = 42

$$\Rightarrow \frac{1425}{N_p} = \frac{42}{21}$$

$$\Rightarrow N_p = \frac{1425 x 21}{42} = 712.5 RPM$$

4.4 Heat Loss Calculations

During oil Circulation in the System the rate of heat transfer through insulation is dependent upon the internal resistance offered by the insulation, temperature of the hot surface and the surrounding air. The total length of the piping line in this laboratory model is about 4.50m; out of this about 2.10m is supply line, about 1.40m is return line and about 1m is excess return line.

4.4.1 Heat Loss Due to Convection and Radiation from the Supply Line

The total area exposed to heat loss on the supply line (A_{sup}) is;

$$A_{sup} = \pi \, D_p \, L_{sup} \qquad (4.9)$$

where: D_p- diameter of pipe line

L_{sup}- the length of supply line

$$A_{sup} = \pi \, 0.0127 x 2.1 = 0.084 m^2$$

Assumptions:

Convection heat transfer coefficient of the ambient air is 10W/m^2.k,

The surface temperature on the fiberglass (Ts) is 45°C

The surface temperature on oil gallery is 38°C

Emissivity (ε) of aluminum foil of fiber glass is 0.04 [3, 2].

The return line oil temperature is about 5°C lower than the supply line oil temperature.

The Stefan Boltzmann constant (σ) is 5.67×10^{-8}

The environment air temperature (T∞) is 20°C

The oil temperature (To) inside the pipe line is 301.5°C

By using equation (2.2) and equation (2.5) the heat loss from the supply line is:

$$Q = U\ A_{\sup}(T_o - T_\infty) + \varepsilon\ \delta\ A_{\sup}(T_s^{\ 4} - T_\infty^{\ 4}) \tag{4.10}$$

But $$U = \frac{1}{\left[\frac{t_p}{k_p} + \frac{t_i}{k_i} + \frac{1}{h}\right]} \tag{4.11}$$

Where:

h- Convection heat transfer coefficient of ambient air

t_p- Pipe thickness

t_i- Insulation thickness

k_p- Thermal conductivity of pipe

k_i- Thermal conductivity of insulation thickness

$$U = \frac{1}{\left[\frac{0.002}{43} + \frac{0.025}{0.046} + \frac{1}{10}\right]}$$

$$U = 1.554 W / m.k$$

Hence the heat loss from the supply line (Q_s) is:

$$Q_s = 1.554 x 0.084 x (574.5 - 293) + 0.04 x 5.67 x 10^{-8} x 0.084 (318^4 - 293^4)$$

$$Q_s = 0.037 kW$$

4.4.2 Heat Loss Due to Convection and Radiation from the Main Return Line

Using the same procedure with the supply line and assuming the temperature of the oil in the return line is 5^{o}c lower than the supply line, the heat loss can be calculated as follows;

$$A_{ret} = \pi\ D_p\ L_{ret}$$

Where: A_{ret} -area of return line

D_p - Diameter of return line

L_{ret} - Length of the return line

$$A_{ret} = \pi\ 0.0127 x 1.40 = 0.056 m^2$$

Hence the heat loss from the main return pipe line (Q_r) is:

$$Q_r = U\ A_{ret}(T_o - T_\infty) + \varepsilon\ \sigma\ A_{ret}(T_s^{\ 4} - T_\infty^{\ 4}) \tag{4.12}$$

$$Q_r = 1.554x0.056\,(568.5 - 293) + 0.04x5.67x10^{-8}x0.056\,(313^4 - 293^4)$$

$$Q_r = 0.024kW$$

4.4.3 Heat Loss Due to Convection and Radiation from the Leakage Return Line

Using the same procedure with the supply line, the heat loss from the excess return line can be calculated as follows:

$$A_{leak} = \pi\ D_p\ L_{leak} \tag{4.13}$$

Where: A_{leak} -area of excess return line

D_p - Diameter of excess return line

L_{leak} - Length of excess return line

$$A_{leak} = \pi x0.0127x1 = 0.04m^2$$

Hence the heat loss from the leakage return line is:

$$Q_{leak} = U\ A_{leak}(T_o - T_\infty) + \varepsilon\ \sigma\ A_{leak}(T_s^{\ 4} - T_\infty^{\ 4}) \tag{4.14}$$

$$Q_{leak} = 1.554x0.04(568.5 - 293) + 0.04x5.67x10^{-8}x0.04(313^4 - 293^4)$$

$$Q_{leak} = 0.017kW$$

The total heat loss from the pipe line (Q_{tp}) is the sum of each individual pipe lines.

$$Q_{tp} = Q_{\sup} + Q_{ret} + Q_{leak}$$

$$Q_{tp} = 0.037 + 0.024 + 0.017$$

$$Q_{tp} = 0.078kW$$

4.4.4 Heat Loss Due to Convection and Radiation from Heat Transfer Fluid Gallery

Using similar steps as in the above cases the area of oil gallery exposed to heat loss is:

$$A_g = \pi D_g h_g + \frac{\pi D_g{}^2}{4} \tag{4.15}$$

Where: A_g - Area of the oil gallery

D_g - Diameter of the oil gallery

h_g -height of the oil gallery

$$A_g = \pi\, 0.58 x 0.06 + \frac{\pi\, 0.58^2}{4} = 0.374 m^2$$

Thus the heat loss from the heat transfer oil gallery (Q_g) becomes;

$$Q_g = U\, A_g (T_o - T_\infty) + \varepsilon\, \sigma A_g (T_s{}^4 - T_\infty{}^4) \tag{4.16}$$

But $U = \dfrac{1}{\left[\dfrac{0.003}{43} + \dfrac{0.035}{0.08} + \dfrac{1}{10}\right]}$

$$U = 1.86 W / m.k$$

$$Q_g = 1.86 x 0.38 (574.5 - 293) + 0.81 x 5.67 x 10^{-8} x 0.37 (310^4 - 293^4)$$

$$Q_g = 0.195 KW$$

The heat loss from the oil gallery surface area is about 0.195kW which is somewhat larger than the heat loss through the pipes; this is because of its larger area exposed to heat loss.

CHAPTER FIVE

EXPERMENTAL RESULTS AND DISCUSSIONS

This section discusses about the equipments used and the results obtained from the experiment during the session.

Figure 5.1: Main components of the laboratory model and thermocouples inserts during experimental test

5.1 Experimental Test Equipments and Procedures

5.1.1 Infrared Thermometer

An infrared thermometer is a non-contact temperature measurement device. It detects the infrared energy emitted by all materials at temperatures above absolute zero, (^{0}k) and converts the energy factor into temperature reading [21]. This configuration facilitates temperature measurement of the object from distance without contact with the object to be measured.

The infrared thermometer was used to measure the baking pan surface temperature, the outer surface temperature of pipe lines, and the surface of the oil gallery and storage tank during the heating and pumping time.

5.1.2 Data Logger

Temperature logger was used to record the temperatures of the surface of baking pan, the temperature of the oil in the pipe lines and the temperature of the oil in the oil storage tank. PC with Microsoft Excel and Lab View software program was used. The software program was stored and the temperature readings were written in Excel program data sheet Applications for printing data sheets.

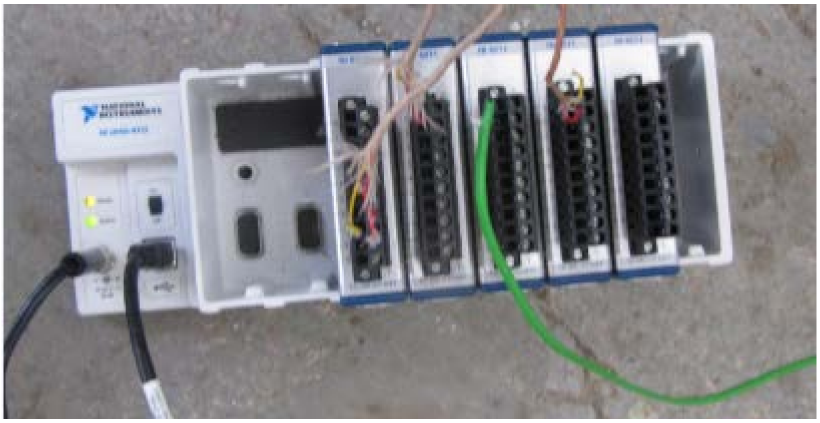

Figure 5.2: Lab view data logger

5.1. 3 Thermocouples

Temperature sensor k-type thermocouples were used during the experimental test to sense the temperature of the oil or the temperature of the surface. One end of the thermocouple was plugged to the cable from the date logger and the other end was to the oil or surface to be measured.

5.1.4 Clamp Meter

This meter measures AC/DC Current, AC/DC Voltage, Resistance and Continuity. During the experimental test the clamp meter was used to measure the current and the voltage in order to know and check the power capacity of the electrical heater, and also to measure the continuity of the coils in the electrical heater.

5.2 Pump Driving Methods in the Laboratory Model System

There are many ways of driving mechanisms; among the options to drive the pump; the belt with pulley and gear to gear drive mechanisms were tried to drive the pump.

5.2.1 Belt Drive Mechanism

In case of belt drive mechanism the system is easy, no noise and is cheap; but the problem with the belt drive system is, the belt cannot withstand the high temperature of the oil and therefore results with the following problems.

a. With increasing the temperature of the oil the belt expands, losses its tension and cannot drive the pump which results with decreasing flow rate.

b. With further increasing to higher temperature of the oil the belt might be tearing, cut and melt in which case the pumping system might stop totally; therefore no more energy is transferred to the pan surface.

c. The melted belt can mix and contaminates the heat transfer oil and may result in affecting the thermal properties of the oil and locking some flow lines.

5.2.2 Gear Drive Mechanism

In case of the gear driving mechanism the pumping system has sufficient flow rate and has no problem to withstand the higher oil temperature, but it has a drawback that there is noise which disturbs the environment around from the gears during operation, but this might be solved since the injera baking system is indoor but the driving system will be in the outdoor for the future solar powered baking system.

So comparing the advantages and disadvantages of both the driving systems, the gear driving system is the better one and is selected for this particular case.

5.3 Temperature Variation and Measurements

The oil in the storage tank was heated using 5.695kW electrical heater; the temperature had increased gradually but the increasing rate was higher during initial heat up time. The experimental tests were conducted and the results are as in the following graphs.

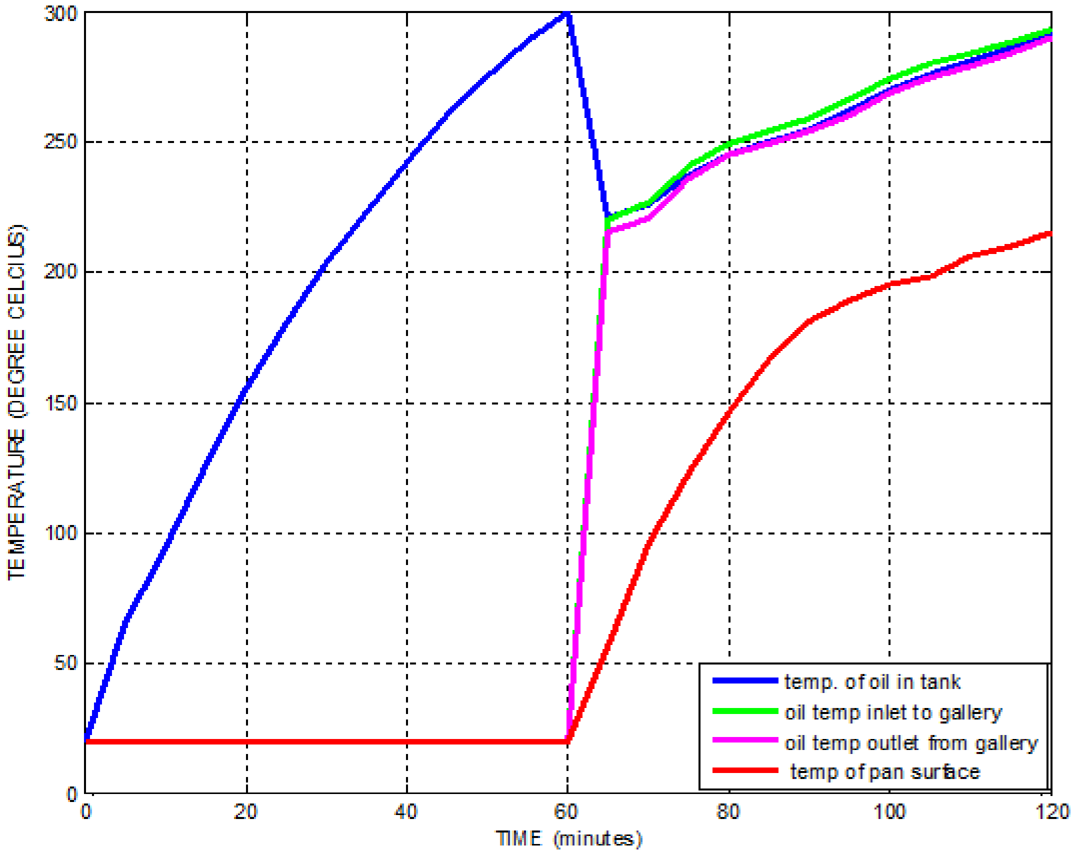

Figure 5.3: Heating up time of oil and baking pan surface before baking injera.

5.3.1 Heat up of Baking Pan

Figure 5.3 shows temperature variation during intial heating of the oil in the storage tank and heating up of the baking pan.

- Even though the design for oil temperature in the oil tank was 320^{o}C, in the figure above the maximum oil temperature is about 300^{o}C and was started to pump the oil at this temperature. This was because at this temperature the oil in the storage tank started to smoke and also expanded to over flow above the design level. This shows that even though the thermocouple's reading was 300^{o}C, the actual oil temperature was more than this temperature may be about 320^{o}C.
- The oil temperature at the outlet from the baking pan assembly was some what lower than the oil temperature at the inlet to the baking pan assembly; this was mainly due to some energy (temperature) loss to heat up the pan surface.

- When pumping was started, the temperature of the oil in the oil tank dropped from 300^{o}C to about 220^{o}C. This was because, in addition to the loss in piping lines; the temperature of the oil in the oil gallery was at room temperature. So during pumping the cold oil in the gallery was mixed with the hot oil in the tank and make the temperature to drop to 220^{o}C.

5.3.2 Baking of Injera on the Laboratory Model

After the fermentation process has been completed the activities during injera baking are as in the figures below.

Figure 5.4.1: Smoothing the baking pan

Figure 5.4.2 Pouring batter on to baking pan

Figure 5.4.3 Injera baked on the baking pan

Figure 5.4.4 Removing the baked injera

Figure 5.4: Pictures taken during baking of injera on the laboratory model of a baking cycle.

Figure 5.4 (above) shows the steps followed during injera baking cycle on the laboratory model:

- figure 5.4.1 shows the polishing of the baking pan in order to make it not to stick the baked injera using powder locally called “ Gomen zer”.
- figure 5.4.2 shows pouring of the batter locally called “lit” on to the baking pan surfce then after covering the pan cover and waits for two minutes of heating
- figure 5.4.3 shows the injera baked on the baking pan surface after two minutes of heating
- figure 5.4.1 shows removing of the baked injera from the baking pan surface and preparing the baking pan for another bakin cycle.

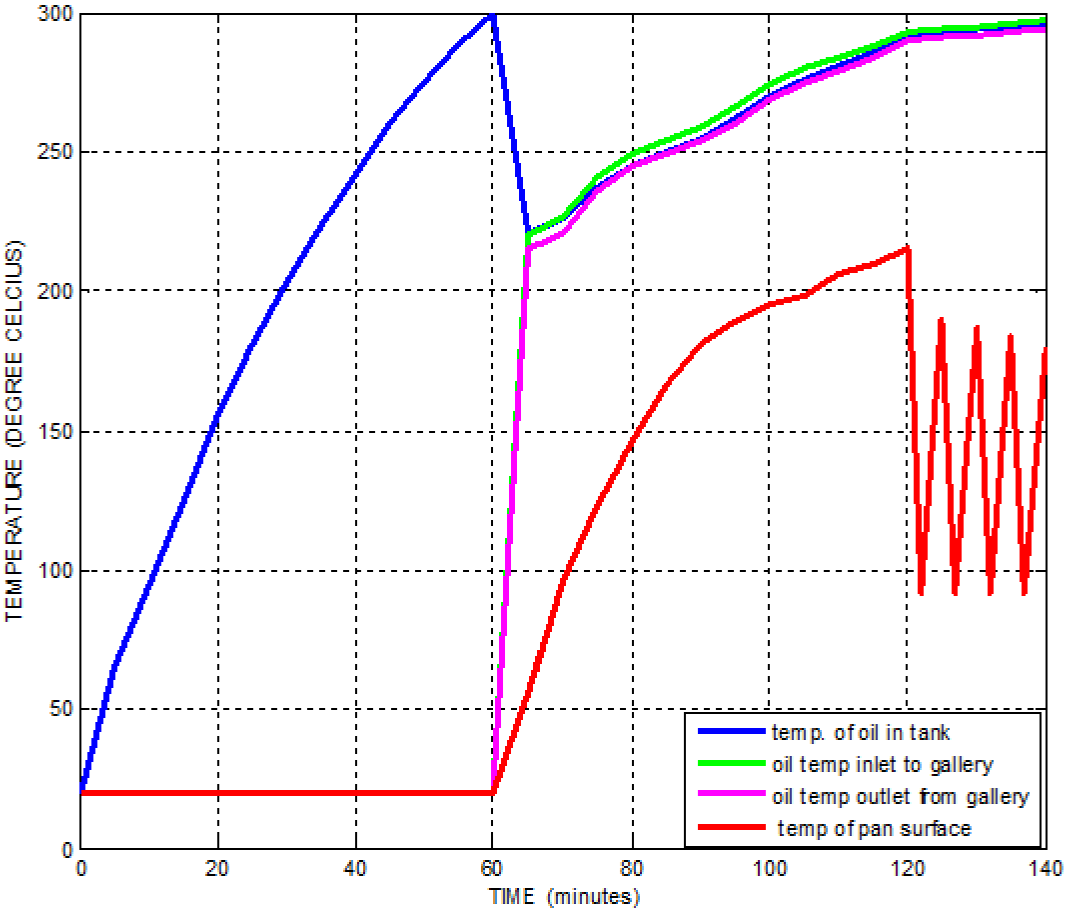

Figure 5.5: Heating up time of oil, baking pan surface and baking cycles.

Figure 5.4 shows temperature variation during heat up and baking. The baking of injera starts after the baking pan surface temperature was reached 215^{o}C, when the batter (locally called ‘lit’) was powered on to the pan surface, the temperature of the pan surface dropped to about 92^oC. Then after it took about two minutes to bake one injera and about three minutes to recover the pan surface to the baking temperature; that is each injera baking cycle took about five minutes. Generally as can be seen from the graph the recovering of the pan surface temperature decreases from one baking cycle to the next. This was because, even though there was enough oil temperature in the heating and storage tank and in the pipe lines; the heat transfer rate from the oil to the pan surface was slow. So a high thermal conductivity material was required to recover the pan surface temperature in a short period of time so that it was possible to bake the required amount of injera consecutively.

Figure 5.5 shows injera baked on the pan surface of the laboratory model after two minutes of heating.

Figure 5.6: Injera baked on the laboratory model

CHAPTER SIX

CONCULUSION AND RECOMMENDATIONS

6.1 Conclusion

- The utilization of solar energy for baking injera would reduce the dependence on fossil fuels and their adverse environmental effects. Moreover, given both the immediate and long-term harmful effects of injera baking system through the burning of fossil fuels and their insufficiencies. In Ethiopia using renewable energy source like solar energy for injera baking is the best solution since sunlight is usually abundant and free.

- Reducing heat loss by using insulation is often a cost effective Energy saving mechanism. However, the effect of adding insulation and understanding insulation economics is important since the addition of insulation produce cost-effective benefits, while there is a point of diminishing returns of savings for increasing the levels of insulation. In this particular case ash insulation system is used for oil storage tank and the oil gallery because of its availability, less costly (free), and good insulation material (low thermal conductivity).

- The Thermia B shell heat transfer oil was used as heat transfer fluid; because of its high operating temperature, high initial boiling point, low vapor pressure, non-corrosive, non-toxic, high heat transfer coefficient, and thermally stable. In addition to this, it has the ability to capture (absorb) the temperature it has for a long time (estimated 3 to 4 hours) after the power source is off; which is very helpful in case solar radiation is not available (or sudden cloudy condition is happened).

- The baking pan surface temperature in the experiment was measured and a temperature of about 215^{o}C on the pan surface was registered; so that it was possible to bake nice injera. The baking cycle took about five minutes. However, this can be reduced using higher temperature in the oil and higher thermal conductivity material. This implies that it was possible to bake injera using heat transfer oil heated by solar or other energy source by oil circulation system.

6.2 Recommendations

- A proper type of pump which is used for the purpose pumping high temperature heat transfer fluid according to the design should be used to circulate the oil without any leakage in order to save energy loss and to transfer to the end use device efficiently.

- The injera baking system needs time for the pan surface to recover its temperature after each injera baking cycle; so that it is better to recommend that the pan support plate should be a high thermal conductivity material like copper.

- To reduce the baking time; higher oil temperature circulation or baking pan with high thermal conductivity has to be used.

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APPENDIX:

Appendix –A Heat Loss from Pipe Surfaces

Heat loss from 1/2" to 12" steel pipes at various temperature differences between pipe and air can be found in the table below. [10]

Heat loss from fluid inside pipe(w/m)													
Nominal bore		**Temperature difference (^{o}c)**											
(mm)	(inch)	50	60	75	100	110	125	140	150	165	195	225	280
15	1/2	30	40	60	90	130	155	180	205	235	280	375	575
20	3/4	35	50	70	110	160	190	220	255	290	370	465	660
25	1	40	60	90	130	200	235	275	305	355	455	565	815
32	11/4	50	70	110	160	240	290	330	375	435	555	700	1000
40	11/2	55	80	120	180	270	320	375	420	485	625	790	1120
50	2	65	95	150	220	330	395	465	520	600	770	975	1390
65	21/2	80	120	170	260	390	465	540	615	715	910	1150	1650
80	3	100	140	210	300	470	560	650	740	860	1090	1380	1980
100	4	120	170	260	380	585	700	820	925	1065	1370	1740	2520
150	6	170	250	370	540	815	970	1130	1290	1470	1910	2430	3500
200	8	220	320	470	690	1040	1240	1440	1650	1900	2440	3100	4430
250	10	270	390	570	835	1250	1510	1750	1995	2300	2980	3780	5600
300	12	315	460	670	980	1470	1760	2060	2340	2690	3370	4430	6450

Appendix-B Physical Characteristics of Thermia B Shell Heat Transfer Oil [23]

Thermia		B
Density at 15°C kg/l	ISO 12185	0.868
Flash point PMCC °C	ISO 2719	220
Flash point COC °C	ISO 2592	230
Fire point COC °C	ISO 2592	255
Pour point COC °C	ISO 3016	-12
Kinematic viscosity	ISO 3014	
At 0°C mm^2/s		230
At 40°C mm^2/s		25
At 100°C mm^2/s		4.7
At 200°C mm^2/s		1.2
Initial boiling point	ISO 13771	>355
Auto ignition temperature	DIN 51794	360
Neutralization value mgKOH/g	ASTM D974	<0.05
Water content %m/m	ISO 3733	<0.1
Ash (oxide) %m/m	ISO 6245	<0.01
Carbon residue %m/m	ISO 10370	0.02
Copper corrosion (3h/100°C)	ISO 2160	Class I
Coefficient of thermal expansion l/°C		0.0008

Appendix –C Thermal Properties of Some Selected Materials [8]

Materials	Density ρ (kg/m3)	Specific heat Cp(J/kg.k)	Thermal conductivity(K)
Selected solids			
Aluminum	2700	903	237
Copper	8930	385	388
Iron	7870	447	80.2
Lead	129	11,300	35.3
Silver	235	10,500	429
Soil	2050	1840	0.52
Tin	227	7310	66.6
Steel, Carbon 1%	7830	460	43
Steel, AISI302	8060	480	15.1
Selected building Materials at 300k			
Brick, common	1600	840	0.72
Concrete (stone mix)	2240	880	1.13
Glass, plate	2500	750	1.4
Hard board, siding	840	1170	0.094
Lime stone			
Plywood			
Softwood			
Insulating materials at 300k			
Asbestos	580	1050	0.16
Blanket (fiberglass)	16	800	0.046
Cork	160	1680	0.043
Duct liner (glass fiber, coated)	32	835	0.038
Polystyrene (extruded)	55	1210	0.027

Vermiculite fill(flaks)	80	835	0.068
Gypsum			0.15
Selected liquids at 300k			
Water	996.5	4179	0.613
Used engine oil	884.1	1909	0.145
Ammonia	599.8	4818	0.645
Mercury	13,529	139	8.54

Appendix-D Table of Emissivity Values of Common Materials

Material	**Emissivity**
Aluminum*	0.30
Asphalt	0.95
Basalt	0.70
Brass*	0.50
Brick	0.90
Carbon	0.85
Ceramic	0.95
Concrete	0.95
Copper*	0.95
Dirt	0.94
Frozen food	0.90
Hot food	0.93
Glass (plate)	0.85
Ice	0.98
Iron*	0.70
Lead*	0.50
Limestone	0.98
Oil	0.94
Paint	0.93
Paper	0.95
Plastic**	0.95
Rubber	0.95
Sand	0.90
Skin	0.98
Snow	0.90
Steel*	0.80
Textiles	0.94
Water	0.93
Wood***	0.94
* oxidized **opaque, over 20 mils ***natural	

CANDIDATE'S DECLARATION

I, the undersigned, declare that this thesis entitled "Design and Manufacturing of a Laboratory Model for Solar Powered Injera baking oven" is my original work and has not been presented for a degree in any University, and that all the source of materials used for the thesis has been duly acknowledged.

Mekonnen Mesele
(Candidate)

Date

This is to certify that the above declaration made by the candidate is correct to the best of my knowledge.

Dr. –Ing Demiss Alemu
(Thesis Advisor)

Date