



JIMMA UNIVERSITY
JIMMA INSTITUTE OF TECHNOLOGY
DEPARTMENT OF MECHANICAL ENGINEERING
CHAIR OF THERMAL SYSTEMS ENGINEERING

**DESIGN MODELLING AND SIMULATION OF INJERA BAKING
STEAM OVEN BY USING BIOGAS BOILER FOR SMALL SCALE
INDUSTRY**

*A thesis submitted in partial fulfillment of the requirements for the award of
the degree of*

**MASTER OF SCIENCE IN
THERMAL SYSTEM ENGINEERING**

to

School of graduate studies of Jimma University

By

SHEWATATEK HAILEGEBRAEL HAILU

Jimma, Ethiopia

April 2024

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of Masters of Science in Thermal System Engineering**

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DECLARATION

I hereby certify that the work which is being presented in this thesis, entitled, “**Design Modelling And Simulation Of Injera Baking Steam Oven By Using Biogas Boiler For Small Scale Industry**” is my original work and has not been presented by any other person for an award of a degree in this or any other university and all sources of materials used for the thesis have been duly acknowledged.

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ABSTRACT

Energy is one of the basic things needed to satisfy basic human needs. The main energy source of developing countries for cooking comes from biomass. Like most developing countries, Ethiopia is also dependent on using traditional fuels. The conventional injera baking method of using open three tone baking stove has a lot of limitations such as much energy consumption house and environmental pollution, deforestation, lung, skin and eye attacks due smoke etc. The second alternative energy that currently used is electricity. The cost of electric energies is increase year to year because of world cost inflation. This paper proposes the design and implementation of a steam energy-powered injera baking oven. The oven utilizes steam generated from a steam boiler or similar heat source to cook injera. The design incorporates elements such a steam generator, temperature control mechanisms, steam injection system, baking surface, ventilation, safety features, and monitoring systems. By harnessing steam energy, the oven offers a sustainable and efficient solution for baking injera, with potential applications in both domestic and commercial settings. The objective of this study was the analytical calculation, CAD modeling and simulation of the heat transfer during the process of injera baking. The design parameters are, capacities of boiler (proposed for five ovens which can bake 600 injeras within 6 hours), geometry of baking pan (diameter 60 cm and thickness 1 cm). The required energy power is 1980kW. The capacity may be depending on the specific requirements of the target users and market demand. Injera baking machine is powered by steam energy. The temperature of steam is equivalent to in the range 180-230°C. The component design and the assembly have been done by Solidworks, AutoCAD. The components are Lamont boiler and baking ovens. This oven is highly recommended to save energy, totally avoid individual energy bill per month and minimize operation costs of hotels and restaurants. Therefore, this injera baking oven saves labor, energy and it is hygienic.

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NOMENCLATURE

Abbreviations

ASME	American society of mechanical engineers
BBTC	Bureau basic technology center
CAD	Computer-aided design
CFD	Computational fluid dynamics
EELPA	Ethiopian electric light and power authority
ESCOA	English shepherd club of America
FEM	Finite element method
GIZ	German international cooperation
LMTD	Log mean temperature difference
MCR	Maximum continuous rating
NGO	Non-governmental organization
SNV	Netherlands development organization

CHAPTER ONE

INTRODUCTION

1.1 BACKGROUND

Energy is a fundamental prerequisite for development and a key requirement for the wellbeing of human society. Energy has been a primary driver of economic growth, wealth creation, and poverty eradication throughout much of the world, but Africa has been largely left behind (1). The global shortage of fossil fuels for energy generation and the harmful effects of greenhouse gas (GHG) emissions on the environment are well documented. This has forced policy-makers worldwide to devise strategies for reducing the environmental impact of the human race (2).

Energy for cooking refers to the energy sources used to generate heat for cooking food. These energy sources can vary depending on factors such as availability, cost, cultural preferences, and environmental considerations. Some common energy sources used for cooking include:

1. Electricity: Electric stoves and ovens use electricity to generate heat through resistive heating elements or induction technology. They are convenient and easy to use but may be more expensive to operate compared to other energy sources depending on electricity prices.
2. Natural gas: Natural gas is commonly used in gas stoves and ovens. It is relatively affordable and provides precise control over cooking temperatures. However, it requires a gas supply infrastructure and may not be available in all areas.
3. Propane: Propane is commonly used in areas where natural gas is not available. It is stored in tanks and can be used for gas stoves, ovens, and outdoor grills. Propane provides similar benefits to natural gas but requires periodic refilling of tanks.
4. Wood: Wood has been used for cooking for centuries and is still prevalent in many parts of the world, especially in rural areas. Wood stoves and open fires are used for cooking, but they can produce smoke and require a constant supply of wood (3).
5. Charcoal: Charcoal is often used for outdoor grilling and barbecuing. It produces high heat and imparts a smoky flavor to food. However, it requires time to heat up and may produce ash that needs to be disposed of properly.

6. Solar energy: Solar cookers harness sunlight to generate heat for cooking. They are environmentally friendly and can be particularly useful in sunny regions with limited access to other energy sources. However, they may not be suitable for cooking during cloudy days or at night.
7. Steam energy: steam energy injera baking ovens are a modern innovation designed to address both energy efficiency and traditional cooking methods. These ovens typically utilize steam generated from water to cook injera, traditional Ethiopian flat bread, which is a staple food in Ethiopia cuisine. The design aims to reduce energy consumption compared to traditional injera baking methods, such as using charcoal or wood-fired stoves. This innovation combines traditional cooking techniques with sustainable energy practices making it an environmentally friendly option for preparing injera.

The difficulties in energy provision necessitate a new strategy for the delivery of electricity. Luckily, the opportunities to transform the status quo in Ethiopia are numerous. The potential for hydropower and other low carbon energy options such as solar, wind and geothermal are substantial, and to-date largely under-exploited, and come with the added benefit of mitigating future greenhouse gas emissions. Low-carbon energy sources will ensure long-term sustainable access to safe, modern, reliable energy services in Ethiopian cities. Scaling up a renewable energy industry and driving energy efficiency would also aid the creation of green jobs and generate wealth in urban communities involved (4).

Injera productions occurs on a number of different scales from bakeries serving the local community to the large commercial bakeries serving entire regions, as well as supermarket, hotels, university, prison, Soldier etc. the focus of this thesis is small scale bakeries; i.e. bakeries producing injera at a rate of several injeras of finished product per hour on a continuous process (5).

Although the core ingredients of injera are: flour, water, yeast. Injera as a product varies vastly depending on the ratio of these ingredients (6).

Steam energy is uncommon in Ethiopia for baking injera. This research paper deals on steam energy used for injera baking. By using indirect baking method which means first biogas burns in combustion chamber of boiler then the water is converted to steam. After that the steam transported through well insulated pipe to baking oven.

1.2 STATEMENT OF THE PROBLEM

A problem statement for an injera baking oven could be framed as follows:

"Injera, a traditional Ethiopian flatbread, is a staple food in Ethiopian cuisine and is typically prepared on specialized clay plates over an open fire. However, traditional methods of injera baking can be labor-intensive, time-consuming, and environmentally unsustainable due to the reliance on wood or charcoal for fuel. The demand and price for Electric Power has been constantly growing in Ethiopia due to the rapid economic growth and inflation. The global shortage of biomass and fossil fuels for energy generation for cooking food and the harmful effects of greenhouse gas (GHG) emissions on the environment are well documented. Additionally, the inconsistency in heat distribution and control often leads to variations in the quality and texture of the injera produced. This has forced policy-makers worldwide to devise strategies for reducing the environmental impact of the human race.

To address these challenges, there is a need for an efficient, reliable, and environmentally friendly injera baking oven that can replicate the traditional baking process while offering improved heat control, energy efficiency, and convenience. Various academics have made an effort to create and refine injera baking oven that used different energy sources such as biogas, solar power, thermal and electricity. Because due to Ethiopia's comparatively widespread access to electricity (hydropower), electricity based mitads are a good substitute that are gaining popularity, particularly in cities.

Such an oven should be capable of evenly distributing heat across the cooking surface, maintaining consistent temperatures, and minimizing fuel consumption. Moreover, it should be user-friendly, safe to operate, and suitable for both commercial and household use.

By developing an innovative injera baking oven that meets these criteria, we can enhance the accessibility, quality, and sustainability of injera production, benefiting both customers as well as the surroundings."

This study focused on steam energy for cooking injera. The reasons why steam energy is chosen for baking injera are: Steam is a critical energy resource in today's food industry. Steam can be produced anywhere and anytime in the world by using the heat that comes from the fuels that available in the area. The following characteristics of steam make it so popular and useful to the industry: Highest specific heat and latent heat, highest heat transfer coefficient, easy to control and distribute and cheap and inert. Steam can hold five or six times as much potential energy as an equivalent mass of water. When water is heated in a

boiler, it begins to absorb energy. Depending on the pressure in the boiler, the water will evaporate at a certain temperature to form steam.

1.3 RESEARCH QUESTIONS

The following research questions are addressed in this study.

- 1) What is the possible impact of the implementation of efficient steam oven for Injera baking?
- 2) What are the key design parameters that influence the performance and efficiency of an injera baking oven?
- 3) How can heat distribution be optimized across the cooking surface of the injera baking oven to ensure uniform baking?
- 4) What materials are best suited for constructing the injera baking oven to achieve durability, heat retention, and environmental sustainability?
- 5) What are the realized savings on energy and materials compared to current cooking materials when a switch to efficient steam oven is made?
- 6) To what extent steam oven Injera baking an alternative to current cooking ovens in practice?
- 7) How can model improve the environmental and economic performance of household steam ovens?
- 8) What is the existing knowledge on steam Injera baking oven?
- 9) What are the major benefits of the utilization of steam energy technology for household purpose in respect to climatically, social-economic and environmental?
- 10) What are the economic feasibility and market potential of introducing improved injera baking ovens, considering factors such as production costs, affordability for users, and potential for adoption and market penetration?

1.4 OBJECTIVES

This study sets the following general and specific objectives

1.4.1 General Objectives

The general objective of this thesis is to design modelling and simulation of injera baking steam oven by using biogas boiler for small scale industry.

1.4.2 The Specific Objectives

The objectives of the study are to:

1. CAD Design each components of baking oven such as biogas burner, LeMont boiler, baking oven and other accessories
2. To perform CFD analysis of baking oven components
3. Calculating heat loss and the performance of baking stoves
4. Design of well insulated materials to reduce extra energy losses of boiler, steam pipe and oven

1.5 SCOPE OF THE STUDY

The scope of design modelling and simulation of injera baking steam oven by using biogas boiler for small scale industry conducted to CAD design and simulation of oven. This study starting from design of biogas not included biogas digester. The proposed production capacity is up to 600 injeras within 6 hrs by using 5 injera baking steam ovens. The required energy power is 1980kW. The capacity may be depending on the specific requirements of the target users and market demand. Injera baking machine is powered by steam energy. The temperature of steam is equivalent to in the range 180-230°C (7). The watertube boiler is designed for five injera baking ovens. The ovens are similar in size, energy consumption, and operating time. Under this study essential components with their proper materials are studied. In this design considered the following points, size, efficiency, automation, flexibility, cost-effectiveness and market considerations.

1.6 SIGNIFICANCE OF THE STUDY

The findings of this study would be a great relieve to Ethiopians, who could not afford the high cost of biomass and electricity used for baking products. This research is geared towards addressing some of the challenges out lined in the problem statement and proffer solutions to them. It will give a comprehensive starting point for more studies on application of steam oven for cooking service especially for Injera baking oven. Also services like review for other researchers who attempt use a steam oven to cut down on CO₂ emissions, indoor air pollution, and energy use. Public and governmental organization also will participate to manufacturing this steam oven Injera baking. The recommendations forwarded at the end of the study will also put light on what approaches to follow of steam oven. Generally, this steam oven is used alternatively for Injera baking. After finish all requirements design of steam oven local NGO's will have interest to work improving access and use of modern and clean energy to urban and rural households. Consequently, when this households better off in

terms of health, income and other social and economic welfare aspects the by contribute to minimize the incidence of Ethiopian poverty.

This baking steam oven applicable in both urban and rural area and also, used for individual and group activities. So, the use of this baking oven minimizes their time and the health problem. The study is also believed to significantly contribute to the efforts of the country to save the future energy needs of the society.

1.7 EXPECTED OUTCOMES

The study is expected to clear the path for the development of efficient and reliable injera baking process by providing information on alternative energy which is steam to replace the existing traditional biomass which could significantly reduce the energy loss of the traditional way of injera baking process. Using steam energy in an injera baking oven is likely to result in several benefits.

1. Even cooking: steam helps ensure that heat is distributed evenly that the injera cooks uniformly without any burnt spots or under cooked areas.
2. Improved texture: the steam helps create a moist environment, which can lead to softer and more tender injera
3. Faster cooking times: steam can accelerate the cooking process, potentially reducing the time it takes to bake a batch of injera.
4. Energy efficiency: steam ovens are often more energy-efficient than traditional baking methods, as they require less energy to reach and maintain the desired temperature.
5. Consistency: using steam can help maintain consistent baking conditions, resulting in consistent quality and flavor of the injera overall, incorporating steam energy into an injera baking oven can lead to better quality, efficiency and consistency in the baking process.

1.6 LIMITATIONS

Limitations of any particular study concern potential weaknesses that are usually out of the researcher's control, and are closely associated with the chosen research design, statistical model constraints, funding constraints, or other factors. In this respect, a limitation is an 'imposed' restriction which is therefore essentially out of the researcher's control.

The major limitation of this paper is that the research is limited to the design and simulation analysis so further prototyping and testing could not be done due to lack of budget and the

variation of the data on the injera possessing and baking is also the other limitation faced in this research.

Generally the limitations are listed below

- Issues with research samples and selection
- Insufficient sample size for statistical measurements
- Lack of previous research students on the topic
- Methods/instruments/techniques used to collect the data
- Limited access to data
- Time constraints
- Too little or unavailable evidence or previous research on the topic

1.7 ORGANIZATION OF THESIS

This research will study the injera processing system and design an efficient machine that will solve the problems of Injera baking. The design includes selecting the best design concept for the machine, selecting different material for the machine, conducting the structural analysis by ANSYS software. The result of the research will be included in the report with the design paper, part drawing and assembly drawing which will be drawn by using Solidworks and Autocad software. The structure of this thesis work is categorized in to eight chapters preceded by the preface of the research which shortly describes the summary of the whole work.

The first chapter introduces the research paper describing the background of the work, statement of the problem, objectives of the thesis, significance of the thesis, significance, scope and finally limitation of the thesis.

In the second Chapter, reviews on injera baking will be carried out by referring research papers, journals, related documents and discussed research gaps.

The third chapter of the research discusses about methods and material, data collection methodologies, description of the study area, sampling size, type of data use and data analysis techniques used to achieve the objective of the research.

The fourth chapter: analytical calculation of watertube boiler and burner

The fifth chapter: Energy and heat transfer analysis of injera baking pan and thermal efficiency of baking pan.

Chapter 6 summarizes the thesis with conclusions and suggestions for future research.

CHAPTER TWO

LITERATURE REVIEW

2.1 BACKGROUND OF INJERA BAKING

The background of injera baking is deeply rooted in the culinary traditions of Ethiopia and Eritrea, where injera holds significant cultural and dietary importance. Here's a brief overview:

Injera has been a common dish in the Horn of Africa for centuries, with its origins dating back to ancient times. Its unique fermentation process and spongy texture have made it a beloved and enduring component of Ethiopian and Eritrean cuisine (8).

Teff flour is used to make traditional injera, a gluten-free grain indigenous to the region. Teff flour is combined with water, then left to ferment for several days, resulting in a tangy flavor and distinctive bubbles on the injera's surface when cooked (3).

Injera is greater than only a food; it is a representation of hospitality, community, and identity. It is often shared communally, with friends and family gathering around a large platter of injera topped with various stews, vegetables, and condiments. The act of tearing off pieces of injera and using them to scoop up the accompanying dishes is a central part of Ethiopian and Eritrean dining etiquette.

Historically, injera was baked on large clay or metal plates called mitad or mogogo, which were placed over an open fire or traditional, stove, fueled by wood or charcoal. The baking process requires skill and precision to achieve the right balance of heat and timing to produce the characteristic texture and flavor of injera.

While traditional methods of injera baking have been passed down through generations, they can be labor-intensive, time-consuming, and environmentally unsustainable. In recent years, there has been a growing interest in developing modern injera baking ovens that offer improved efficiency, consistency, and environmental sustainability. These innovations aim to preserve the traditional art of injera baking while making it more accessible and feasible for contemporary households and commercial producers.

2.1.1 Injera Baking Method in Ethiopia

Injera baking method is can classified in to four based on the source of energy

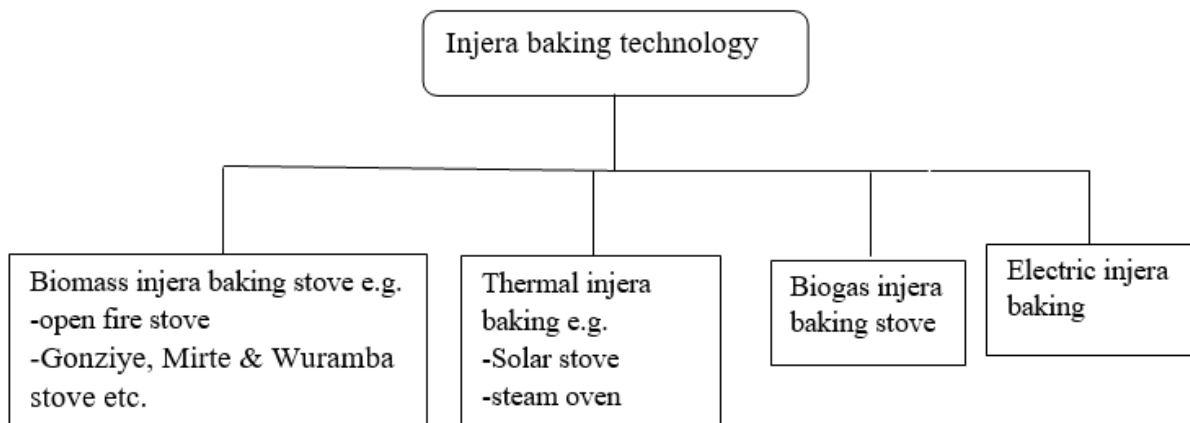


Figure 2.1: Types of injera baking stoves based on energy sources

2.1.4 Reviews of Injera Baking Stoves

The majority of Ethiopians use biomass and open-fire stoves to bake injera. The ineffective open-fire stove uses a lot of firewood, pollutes interior air significantly, and CO₂ emission. This includes early efforts by the Ministry of Agriculture, Water Resources, and Energy, i.e. by the GIZ-Energy Coordination Office and the Alternative Energy Development and Promotion Directorate by the GIZ-Energy Coordination office-Ethiopia improved see the reduction in specific fuel consumption of Mirt compared to open fire baking which was in the range of 30-40% (3). Researchers are still working to improve the efficiency of biomass burners, particularly for baking injera. .

This review discusses the following issues

- i. Injera baking stoves' initial development
- ii. Recent studies, as well as

The way forward in terms of research and development regarding injera baking stoves Since, the information collected is mostly from reports from published in peer reviewed journals.

It had been a while since the demand for effective injera baking ovens was addressed, but in the 1980s, government agencies set the groundwork with their research, and. Early initiatives included the Burayou Basic Technology Center (BBTC), a division of the Ministry of Education, producing mud injera baking stoves in the early 1980s. The stove was called a "Burayou mud-stove." In 1981, a consultant was employed by the Ethiopian Science and Technology Commission, which is currently the Ministry of Science and Technology of Ethiopia, to evaluate traditional closed stoves in specific regions of the nation (9). This study's primary goal was to survey the various stove types that are currently in use across the

nation. There were 113 clay stoves (mitad) with a diameter of 45–65 cm and a thickness of less than 1.5 to 2.5 cm that were surveyed (3). The performance evaluation was made based on water boiling test. It was performance at Appropriate Technology Center of the Adult Education Department of the Ministry of Education.

As a result, to construct the fire chamber enclosure, the mud construction of Mirt is replaced with a cement-mortar mixture. Mirt is an enclosed Injera stove designed by the former Ethiopian Energy studies and Research center of the Ministry of Mines and Energy. According to GTZ test results show that the latest Mirt stoves save fuels over 50% and carbon monoxide (CO) emission reduction of about 90%, when compared to the traditional biomass stove, and have efficiency 35%. Researchers and developers have discovered the precise fuel consumption of the Mirt stove. The typical amount of fuel used Mirt stove is 535g per kilogram of injera (9).

Clay is used to make the other upgraded injera baking stove, and its name is Gonziye. The Ethiopian Rural Energy Development and Promotion Center develop Gonzie stove in 1994, which serve as cooking and baking to make available affordable fuel saving stoves to the rural areas. The Gonzie multi-purpose stove attains an efficiency of 23%. It is a multipurpose improved cooking stove to be used both for injera baking and other types of cooking such as water heating, coffee making and wot preparation. The specific fuel consumption of Gonziye injera baking stove is 617g/kg of injera (10).

The Awramba injera baking stove, named for the Awramba settlement in Amhara Regional State, is the other modified model. The community has been using this injera baking furnace since 1971. Beyond baking injera, this stove incorporates other cooking functions as well. When it comes to fuel consumption, it is specifically 35% less than the open-fire injera baking stove (11).

Biogas cooking stove is widely used in developing countries; however it is not popular for injera baking. The Ministry of Water, Irrigation and Electricity's Alternative Energy Development Directorate, in collaboration with SNV, is working on many projects to create a biogas stove for the purpose of baking injera. Baking injera was used to gauge the concentric three-ring injera stove's performance. The stove produced 11kW of electricity, which translates to 41 L/min of gas usage. 193 L (3108 kJ) of gas were used for each injera. Within the range of improved biomass stoves, the three-ring biogas stove's average efficiency was 16% (13).

Combustion of methane using proper fuel-to-air ratio will create a non-polluting and comfortable kitchen for injera baking. The amount of biogas or the feed stock require for baking injera for a household, on average 25-30 injera at a time, is quite significant. Thus, improving the development of an injera baking biogas stove requires further efforts by researchers and developers in the area.

Cheru Zeleke studied about development of efficient biogas injera baking stove. The designed (developed) injera baking stove has nine vertical burners. According to the diameter of injera baking pan and amount of energy needed for baking injera, the nozzle diameter, number of burner and number of flame ports are designed for uniform heat distribution all over the radius of baking pan. The developed (designed) stove is manufactured and the test is done using LPG as a source of energy due to unavailability of biogas (12).

Taha Abdella Geda*, Demis Alemu Ambie, Kamil Dino Adem also study experimental investigation of biogas injera baking stove using circular ring pipe burner. The purpose of this study is to solve this problem by developing injera baking biogas stove with better performance. The experimental investigation was conducted using a biogas plant in Addis Ababa University 4 Kilo Campus. A detailed energy analysis of energy demand, energy output, and baking performance has been performed. Injera baking time is 3 minutes; idle time is 2 minutes between successive injera baking. Thus, the designed injera baking biogas stove achieved uniform heat distribution and improved performance (13).

The electric injera baking burner, also known as the electric injera mitad, was first brought to Ethiopia forty years ago by the Ethiopian Electric Light and Power Authority (EELPA). The electric injera stove was created and marketed to the market by a number of public and private groups in order to be distributed when the market was introduced to the electric injera baking stove. The performance of the electric injera baking stove is dependent on the company's experience and the caliber of the craftsmanship because it is not standardized. An electric injera baking stove used by one household typically consumes between two and four kW (16).

2.1.5 Solar Energy

The systems that are chosen must not only have technological features that conveniently solve specific cooking requirements, but also be socially and economically acceptable to their end users in order for solar energy to be used for cooking to be effective. Solar cooking can result in (i) better health for women and children, (ii) less local pollution, (iii) less local forest

degradation, and (iv) reduced contributions to global greenhouse gas emissions when it replaces cooking fuels used in developing nations. This book discusses the wide variety of thermal and photovoltaic solar cooking systems that are either in development or currently accessible in various parts of the world. The social, cultural, and economic aspects that have hindered the widespread use of solar cookers are given special consideration. It has been demonstrated that technological advancements aimed at overcoming these constraints are only beneficial when they enable customary practices of preparing specified foods in desired temperatures in designated climates (17).

A solar-powered injera baking burner was yet another alternative that several researchers looked into. The two primary results of the investigation inquiry were the potential use of steam, which can bake injera between 180 and 200 degrees Celsius, and the hopeful outcome of using a direct solar radiation reflector to bake injera using solar energy. In order to heat the thermic fluid that circulates during baking, an electric heater simulation was used in the experiment.

Table 2.1: Solar thermal injera baking stoves and injera baking pans research outputs

No	Authors	Year	Title of the research	Major outputs	Reference
1	Assefa Ayalew Tareke	2010	Heat transfer analysis of Injera baking pan by FEM	Using a clay plate, the heat transmission process during injera baking is modeled using FEM. Using an electric injera baking stove, this was experimentally confirmed. If the thickness of the cooking plate is reduced or the heat conductivity is raised, significant improvements are expected.	(14)
2	Asfafaw Haileselassie, Mulu Bayray Kahsay, and Ole Jorgen Nydal	2013	Design and development of solar thermal injera baking: steam based direct baking	Presents the indirect solar stove. To bake injera, use a stainless steel heat exchanger and well-insulated pipes in a parabolic dish with a 2.54 m ² aperture. The experiment proved that injera may be baked at a high temperature using indirect steam. According to the authors, injera may be satisfactorily baked at temperatures between 135 and 160 °C.	(15)

3	Abdulkadir Aman Hassen and Demiss Alemu Amibe	2013	Finite element modeling of solar powered injera baking oven for indoor cooking	A novel solar-powered injera baking system was the subject of a 2D transient finite element study. Under the suggested technique, heat transfer oil circulates through the area in the kitchen beneath the baking pan after being heated by a parabolic trough using solar energy. A novel kind of baking pan made of ceramic with an 8 mm thickness was produced and employed for the suggested system based on earlier finite element research on currently available electric injera baking pans. When compared to current traditional baking methods, the suggested solar-powered baking pan provides a reasonable heat-up and baking time. The temperature distribution during initial heat up and cyclic baking is well predicted by the finite element model.	(16)
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2.1.5 Modern Machine Production Commercial Injera

Modern injera producing machines have the ability to shape, bake, grout, and chill automatically. The machine's automated adsorption of the batter into the baking pan is the basis for its operation. The batter is then heated in the baking pan to solidify and mature. The machine's shovel automatically shoveled the Injera down and allowed it to cool on the conveyor belt once the baking pan had turned around. Additionally, this machine has multiple heating methods. Other heating techniques include gas heating, resistance wire heating, electromagnetic heating, and electric heating (21).

Advantages of automatic Injera making machine

1. We can customize round and square molds to meet customer needs
2. The thickness and size of the wrappers produced by this machine are consistent making them appropriate for commercial production of injera.
3. The injera making machine has a variety of models that can make 10-60 cm injera
4. A variety of heating methods can meet the needs of different customers;
5. The injera maker can adjust the thickness of the final product by adjusting the amount of grouting and the viscosity of the slurry. The machine can be equipped with folding counting and filling equipment according to customer requirements to form an automatic injera making machine.

2.3 GAS BURNERS

Gas burners are generally classified as premixed or turbulent jet diffusion burners, as defined below:

Premixed burners, where the fuel and most of the air are mixed together prior to passing through the nozzle. Note that these burners often introduce a small proportion of the air separate from the fuel as secondary air (17).

The following characteristics typify premix flames:

- Flames are small in volume relative to heat release rate.
- High combustion efficiency, i.e. small fraction of unreacted or partially reacted fuel.
- Non-luminous (little soot formation) and low radiant heat transfer efficiency.
- Noisy (relative to heat release rate).

The nozzle mixing system requires that both the air and gas are supplied under pressure. Subject to the pressures available, high velocities can be achieved. Indeed high velocities are required to prevent 'burn back' into the mixture tube. The mixing is complete prior to the refractory tunnel where the combustion reactions occur. Provided some excess air has been supplied, combustion is also complete or nearly complete at the tunnel exit.

Turbulent jet diffusion burners: where the fuel and most of the air are introduced separately and are mixed within the furnace. Combustion is then controlled by the rate at which the air mixes with the fuel. Note that these burners often mix a small proportion of the air with the fuel as primary air (17).

2.4 BOILERS

A boiler, as the name suggests, is a device that boils (water to produce steam). More scientifically, a boiler converts the chemical energy in fuel into the heat energy in steam, or the heat energy of hot gases into the heat energy of steam when no firing is involved. The earliest form of boiler is the kettle.

1. Classification of boilers is based on what flows through the boiler tubes.
 - Fire or smoke or flue tube or shell-type boilers in which flue gases are inside and water is outside
 - Water tube boilers in which water is inside and flue gases are outside

In fire tube boilers, hot gases pass through tubes which are surrounded with water. Examples: vertical, Cochran, Lancashire and Locomotive boilers. There may be single tube as in case of Lancashire boiler or there may be a bank of tubes as in a locomotive boiler.

In water boilers, water circulates through a large number of tubes and hot gases pass around them. Eg. Bobbcock Wilcox boiler

2. According to the axis of the shell
 - a. Vertical boilers
 - b. Horizontal boilers
3. according to location or position of the furnace
 - a. Internally fired boilers: the boilers the furnace forms an integral part of structure. The vertical tubular, locomotive and the scotch marine boilers are well known examples.
 - b. Externally fired boilers: externally fired boilers have a separate furnace built outside the boiler shell and usually below it. The horizontal return tube boiler is probably the widely known example of this type.
4. According to the application-
 - a. Stationary: a stationary boilers is one of which is installed permanently on a land installation
 - b. Mobile boilers: a marine boiler is a mobile boiler meant for ocean cargo passenger ships with an inherent fast steaming capacity.
5. According to steam pressure
 - a. Low pressure boilers: a boiler which produces steam at a pressure of 15-20 bar is called a low-pressure boiler. This steam is used for process heating.
 - b. Medium pressure boilers: Medium-pressure boiler: It has a working pressure of steam from 20 bars to 80 bars and is used for power generation or combined use of power generation and process heating.
 - c. High pressure boilers: High-pressure boiler: It produces steam at a pressure of more than 80 bars.
 - d. Sub-critical boiler: If a boiler produces steam at a pressure which is less than the critical pressure, it is called as a subcritical boiler.
 - e. Supercritical boiler: These boilers provide steam at a pressure greater than the critical pressure. These boilers do not have an evaporator and the water directly flashes into steam, and thus they are called once through boilers.

2.3 LITERATURE GAPS

Various Injera baking oven types that rely on the energy source were invented and investigated by earlier academics. The energy technology of the future will be interested in producing goods that are efficient, safe for the environment and high-performing while utilizing an endless supply of resources. Future research priorities in the areas of solar thermal, gasification, biogas, and steam could significantly enhance the health of women and children who are exposed to indoor air pollution. Nevertheless, the majority of research on Injera baking ovens was directed toward enhancing the use of particular fuels, with minimal focus on lowering carbon emissions and indoor air pollutants like CO. Future studies should therefore focus on problems like these, which are crucial elements in.

Reducing fuel usage for both financial and environmental reasons is the main goal of improving Injera baking ovens. Other goals include introducing fundamental terminology and the use of steam in the baking process. When processing food, steam is an easy way to transfer energy. It is made with readily available, reasonably priced water. Steam temperature can be precisely controlled and maintained with the use of pressure control valves. Since steam has large energy content in a relatively small mass, heat-transfer apparatus can be made tiny. Steam may be transported across considerable distances and to isolated areas of the process with ease and at a low cost.

Table 2.2: Summary of Gap identified in Review of injera baking technology

Baking stove		Efficiency	Gaps	Source
Biomass	Open fire	10-15%	<ul style="list-style-type: none"> • Inefficiency • Unhealthy, cause damage on lungs and eyes and • Allow excessive deforestation 	(18)
	Mirt & Gonziye	35~50%		
Solar			<ul style="list-style-type: none"> ✓ Difficulty to use in winter seasons and rainy days ✓ Baking will take longer time ✓ Cooker must be scheduled 	
Biogas		51-55%	<ul style="list-style-type: none"> • Lower quality of injera • Inefficient due to poor design • Face unequal distribution of pressure on the holes 	
Electric		50-55%	<ul style="list-style-type: none"> ○ Poor design ○ Energy inefficiency ○ High operating cost 	

CHAPTER THREE

MATERIALS AND METHODS

3.1 INTRODUCTION

The available technologies for the baking oven method are examined, and their limitations and drawbacks are noted, in the assessment of the literature that was covered in the preceding chapter.

3.1.1 Indicators and Variables

Concepts and variables found in the literature review are here translated to signal for data collection as part of the study process. The variables that influence household cooking energy choice can be broadly classified into three categories: socio-economic, socio-cultural, and factors connected to cooking energy supply. In order to address the study topics, specific variables were found under each of the major areas.

3.1.2 Research Type

Generally speaking, the exported answers to the research questions require more details than merely a summary of the several factors influencing home cooking energy choice. Thus, it is possible to argue that an applied and quantitative research strategy was employed in this study. Additionally, the review of empirical literature showed that variables such as gender roles, taste preferences, and culturally specific home design may have an impact on the amount of energy used for cooking in households. These variables are in addition to easily quantifiable ones like fuel prices and household income. It is essential to collect the data required to answer these study questions.

3.1.3 Research Strategy

A research strategy introduces the main components of a research project such as the research topic area and focus, the research perspective, the research design, and the research methods. This study combined reviews of the literature with CFD simulation and CAD modeling.

3.1.4 Identification of Study Areas

Since biogas is a readily available energy source everywhere in Ethiopia, the study regions can encompass the entire country. Thus, power and other auxiliary fuels have no bearing on the results of this investigation. A collection of facts, such values or measurements, is called data. It can consist of words, numbers, measurements, observations, or even just item

descriptions. Several data collection techniques were used to primarily use secondary data sources for this investigation. Additionally, semi-structured observation was utilized as one of the primary data collection methods to gather information about local cooking customs, household energy use behaviors, and the availability of cooking energy. Semi-structured interviews were conducted to elicit detailed information about the potential interactions and impacts of the various factors identified on household practices.

In this thesis, secondary data is incorporated. Secondary data is gathered from a variety of sources and official company documents for analysis. This study employs both qualitative and quantitative analysis.

Publicly available or unpublished secondary data are also possible. Typically, published data can be found in:

- a) A variety of publications from the federal, state, and local governments;
- b) a variety of publications from foreign governments, international organizations, and their affiliate groups
- c) Trade and technical journals d) Books, periodicals, and news sources
- d) Publications and reports from different business and industry-related associations, banks, stock exchanges, etc.
- e) Reports written by academics, researchers, economists, etc. in various subjects.
- f) Historical records, official statistics, and other public information sources

Workshop manuals, libraries, and electronic media on the internet are the sources of this data. Secondary data needs to demonstrate that they have the following traits. Data quality criteria: appropriateness, sufficiency, filtration, and reliability

3.2 METHODS

For this study, the analytical and numerical approach was selected since it is the only workable strategy. The two main types of software needed are simulation programs (like Ansys, Comsol Multiphysics, etc.) and CAD programs (like Solidworks, AutoCAD). Computational Fluid Dynamics (CFD) is a branch of numerical methods for flow simulations that was employed in this study.

The process of working with charts is described in the approach, as illustrated below

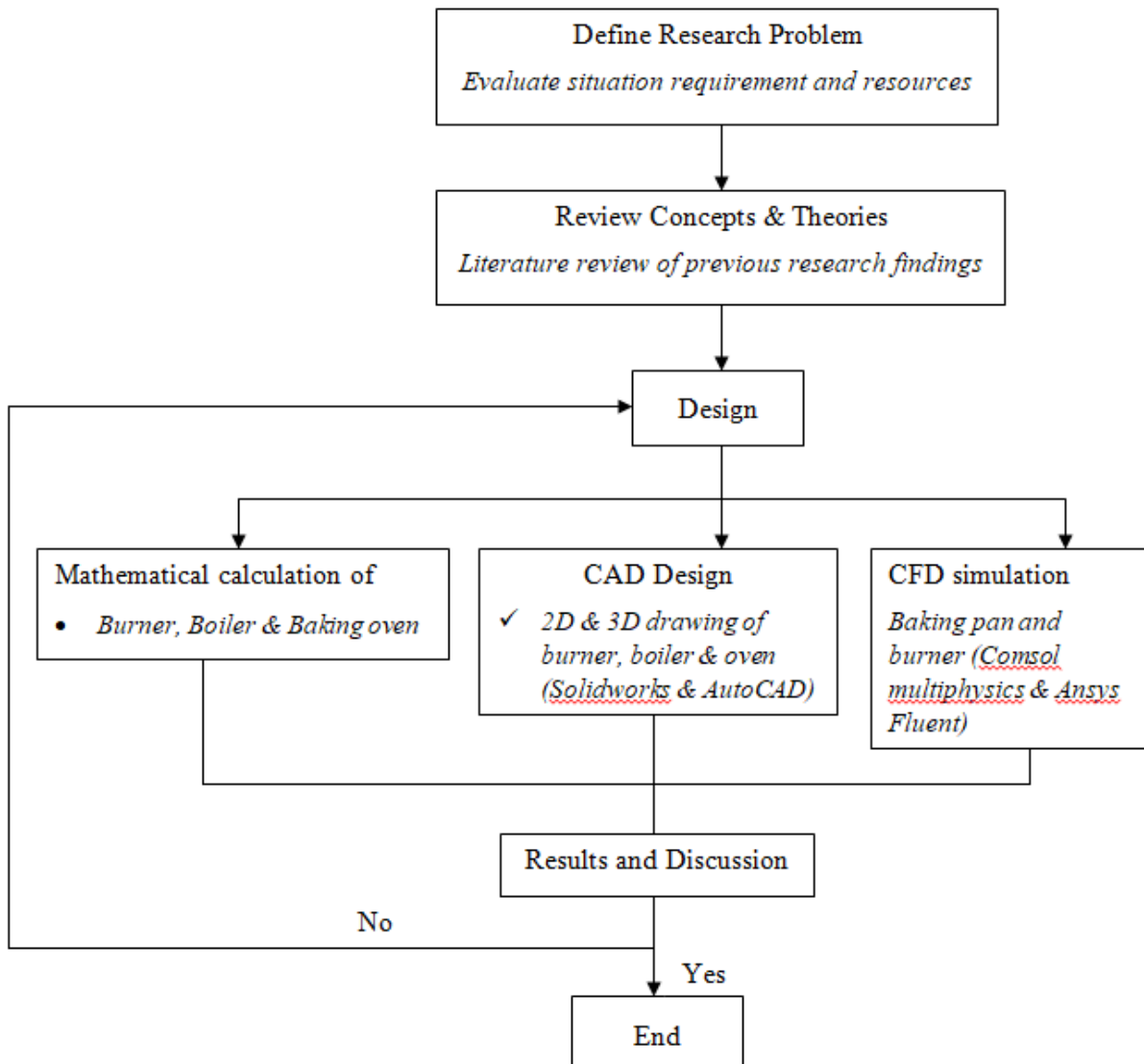


Figure 3.1 Flow diagrams for methodology

CHAPTER FOUR

ANALYTICAL DESIGN ANALYSIS OF WATERTUBE BOILER

4.1 INTRODUCTION

This design focuses on the fundamentals of a Lamont watertube boiler with the right measurements, components, and combustion heat. Watertube boilers are water inside and flue gasses outside that use water tubes (23).

4.1.1 La-Mont Boiler

The first forced convection boiler was the Lamont boiler. It's an internally fired boiler with forced circulation and high pressure water tubes. Because Douglas La-Mont introduced it in 1925, it was given the moniker Lamont Boiler. The Lamont boiler uses an external water tube pump to move water via its water tubes. As a result, the water pressure in the tubes is higher than the water pressure in the boiler with natural circulation. The main components of Lamont boilers are feed pump, economizer, steam drum, evaporator, superheater, blower and grate.

a. Construction:

The combustion chamber and air blower in the Lamont boiler are connected, providing air to the boiler. The economizer, which heats the water, is connected to the steam separator drum by the water feed pump. The steam separator drum is connected to the radiant evaporator, which is situated close to the combustion chamber. The superheater, which raises the air's temperature to the necessary level and transfers the steam to the oven to bake injera, is also connected to the steam separator drum.

b. Working:

The upper part of the steam separator drum collects steam, while the lower part of the drum settles the residual water. It is not possible to send the saturated steam from the steam separator drum straight into the oven since it will cause corrosion. Thus, the saturated steam is pushed through a superheater, which raises the steam's temperature to the necessary amount and transforms the saturated steam into superheated steam. The superheated steam was then transported to the oven to make injera. Additionally, outside air enters the combustion chamber where the fuel is burned. The evaporator is heated by the furnace's flue gases before they pass by the super heater. This gas enters the economizer (19).

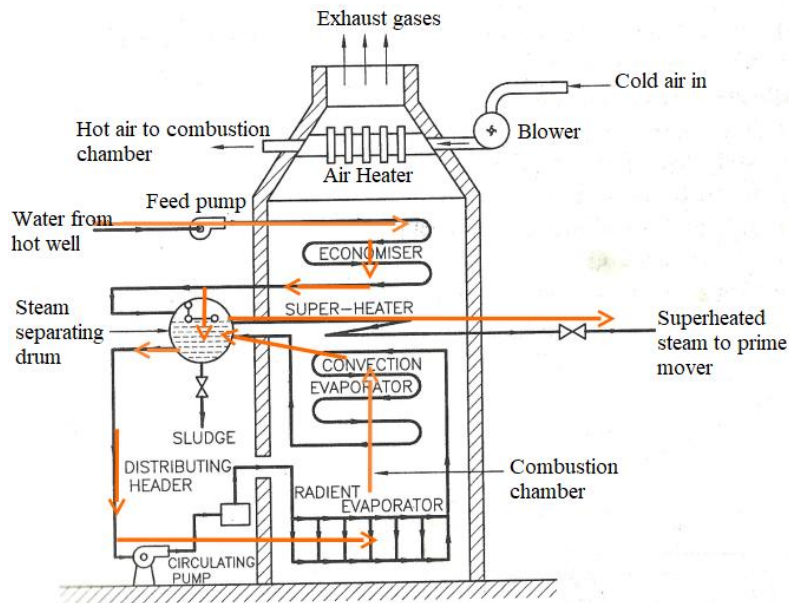


Figure 4.1 La Mont Boilers (24)

4.2 MODEL OF HEAT EXCHANGER FOR FURNACE-EQUIPMENT BOILERS

On the water/steam side, the heat transfer devices are always arranged in downstream order: economizer, evaporator, and superheater.

The water reaches the necessary temperature and pressure to enter the boiler after the feedwater pump. Through the economizers, pressurized water is injected into the boiler. Heat exchangers, often in the shape of tube packages, are the economizers. In the flue gas channel, economizers are positioned above the superheater zone. Typically, they are built as a collection of tubes that are affixed to the flue gas channel walls (20).

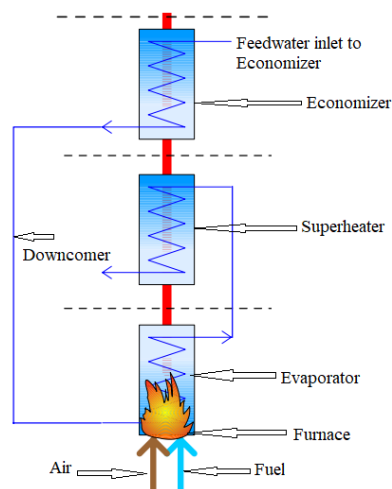


Figure 4.2: Furnace equipped boiler (20).

4.4 WATERTUBE BOILER'S INPUT PARAMETERS

One injera baking requires 2.2kW of energy. It took 11kW of energy to baking five injeras. 16.5 kW is the result of multiplying the wasted energy by the safety factor of 1.5. The Power needed, for 600 injera baking for 6 hours of baking time 1980 kW.

Chemical Power = Volume flow rate of gas × Calorific Value of gas

$$\text{Volume flow rate of gas} = \frac{\text{Power Input}}{\text{Calorific value of biogas}}$$

$$\text{Volume flow rate of gas} = \frac{0.0165\text{MJ}}{22.6\text{MJ}/\text{m}^3} = 7.3009 \times 10^{-4}\text{m}^3$$

The theoretical air required for complete combustion of biogas can be calculated by using the following formula.

The entire energy required for injera is utilized as boiler input, resulting in 16.5 kW of energy being absorbed by a steam boiler powered by biogas (CH₄ and CO₂). On a higher heating value (HHV) basis, boiler efficiency is 83%. Assuming a 15 percent surplus air usage, ascertain the overall air and flue-gas output. The biogas has an approximate HHV of 23,890 Btu/lb (55,568.14kJ/kg). The air temperature is 68°F (20°C), with a relative humidity of 60%, and the height is 4 meters (21).

Type of boiler: vertical, installed indoors

-High pressure steam requirement: 8 T/H

-Steam pressure requirement: 304 Psi

-Steam Temperature Requirement: 260°C

-Hot water temperature: 90 °C.

-One burner, please Table

Table 4.1 Calorific value of biogas

	% mol
LHV (kcal/kg)	21.5 MJ/kg
HHV (kcal/kg)	23.5 MJ/kg

It should be remembered that all calculations made in the first step are estimate, and they will need to be repeated once the detailed design is complete. Higher heating value is represented by HHV and lower by LHV.

4.5 BOILER HEAT DUTY

The quantity of heat that is transmitted in a unit of time from a hot side to a cool side is known as the heat duty. Initially, we must compute the boiler heat duty using the formula below (22):

$$Q_{to-steam} = h_{outlet\ steam} - h_{inlet\ water} \quad (4.1)$$

$$h_{outlet\ steam} = 388.925 \text{ kcal/kg}$$

$$h_{inlet\ water} = 55 \text{ kcal/kg}$$

$$Q_{to\ steam} = 333.93 \text{ kcal/kg}$$

$$Q_{duty} = Q_{to-steam} \times m'_{boiler\ at\ 100\% \ MCR} \quad (4.2)$$

$$= 333.925 \times 30 = 10,017.75 \text{ Mkcal/hr}$$

$$Q_{duty} = 10,017.75 \text{ Mkcal/h} = 39.75 \text{ MBtu/hr} = 11.65 \text{ MW}$$

4.6 REQUIRED FUEL

Thus, we should determine the fundamental boiler data, such as the necessary fuel and combustion air, after calculating our boiler heat duty.

$$m'_{fuel} = (Q_{duty} \times 1.05) / LHV \quad (4.3)$$

$$m'_{fuel} = (10,017.75 \times 1000 \times 1.05) / 5,138.63$$

$$m'_{fuel} = 2,046.97 \text{ kg/h}$$

4.7 BAROMETRIC PRESSURE

Moreover, the air is lighter at greater elevations than it would be below due to the absence of particles in the atmosphere. Similarly, the gravitational force diminishes with height. These explanations help to explain why the barometric pressure is lower at greater elevations.

$$P = \frac{101,325 \times (1 - 2.25577 \times 10^{-5} \times H)^{5.25588}}{6,894.76} \quad (4.4)$$

$$P = 14.69 \text{ psia}$$

4.8.1 At the Real Temperature, Saturation Pressure

The saturation pressure of water vapor in moist air is influenced by the temperature of mixing air vapor, which is expressed as follows:

$$P_{ws} = \frac{e^{\left(\frac{77.345+0.0057 \times T - 7,235}{T}\right)}}{(T^{8.2}) \times 6,894.76} \quad (4.5)$$

Where T is the dry bulb temperature of moist air (K), e is the constant 2.718, and Pws is the water vapour saturation pressure (psia)

$$P_{ws} = \frac{e^{\left(\frac{77.345+0.0057 \times T - \frac{7,235}{(20+273.15)}}{(20+273.15)}\right)}}{(20+273.15)^{8.2} \times 6,894.76} \quad (4.6)$$

$$P_{ws}=0.3381 \text{ (psia)}$$

4.9 MOISTURE IN AIR

We need to know the total amount of water vapor in flue gases—some of which come from combustion air—in order to calculate nonluminous heat transfer.

In addition, water will have a higher saturated vapor pressure in the presence of atmospheric pressure, and water can condense in the presence of cooling below the water dew point temperature.

Because the total amount of moisture in the air or gas affects the water dew point temperature, it is imperative to know the total amount of water vapor in the air or flue gas:

$$M = 0.622 \times \frac{RH \% \times SVP}{(P_{barometric} - RH \% \times SVP)} \quad (4.7)$$

When SVP (saturated vapor pressure) is represented by SVP, relative humidity by RH, and moisture in the air by M

$$M = 0.622 \times \frac{0.6 \times 0.4861}{(14.69 - 0.6 \times 0.3381)}$$

$$M = 0.009 \text{ lb/lb dry air}$$

$$W'_{\text{total combustion air}} = 72,655 \text{ (kg/h)}$$

4.10 TOTAL REQUIRED COMBUSTION AIR

The specification of actual cubic feet per minute (ACFM) vs standard cubic feet per minute (SCFM) and the precise conversion from one to the other are the main issues in blower performance calculations. While some engineers use ACFM, others utilize SCFM. Pressure, temperature, and humidity flow are commonly taken into consideration when designing SCFM.

To ensure that the full amount of oxygen needed to function effectively will be supplied, the following adjustments should be made:

$$W'_{\text{total combustion air}} \text{ (kg/h)} = \text{Total air required (lb/lb dry air)} \times m' \text{ fuel (kg/h)} \times 1 + \text{excess air}$$

$$W'_{\text{total combustion air}} = 8.1 \times 958.734 \times 1.17$$

$$W'_{\text{total combustion air}} = 9,085.922 \text{ (kg/h)}$$

4.10.1 Air Density

Any gas's density can be calculated using

$$\rho_g = 492 \times MW \times \frac{P}{359 \times (460 + T) \times 14.7} \quad (4.8)$$

Where, T is the gas temperature (°F), P is the gas pressure (psia), MW is the molecular weight of the gas, and ρ_g is the gas density (lb/cu.ft).

$$\rho_g = 492 \times MW \times \frac{14.69}{359 \times (460 + (1.8 \times 20 + 32)) \times 14.7}$$

$$\rho_{\text{air}} = 0.0752 \text{ lb/cu.ft}$$

4.10.2 Standard Combustion Air Flow

Variations in standard temperature will result in a large volumetric variation at the same mass flow rate:

$$V_s = \frac{W'_s}{\rho_{\text{air}} \times 60} \quad (4.9)$$

$$V_s = \frac{9,085.922 \left(\frac{\text{kg}}{\text{h}}\right) \times 2.20462}{0.0752 \times 60}$$

$$V_s = 4,439.496 \text{ (SCFM)}$$

4.10.3 Actual Combustion Air Flow

For a gas to flow, either a vacuum or positive pressure must be created. When a standard cubic foot of gas is connected to positive pressure, it becomes compressed. When a standard cubic foot of gas is linked to a vacuum, it expands. The volume of gas after rarefying or pressurizing will be the "actual" volume.

$$\frac{V_a}{V_s} = \frac{P_s}{P_a} \times \frac{T_a}{T_s} \quad (4.10)$$

Where P_s is standard air pressure at sea level (p sia), P_a is air pressure at actual level (p sia), V_a is actual combustion air volume SCFM, V_s is standard combustion air volume SCFM, T_s is standard ambient air temperature (R), and T_a is actual ambient air temperature (R).

$$V_a = \frac{14.7}{14.69} \times \frac{(18 \times 20 + 491.87)}{520} \times 4,439.496$$

$$V_a = 7,420.486 \text{ (SCFM)}$$

4.10.4 Forced Draft Fan Discharge Mass Flow

$$W'a = 60 \times \rho_{air} \times V_a \quad (4.11)$$

$$W'a = 60 \times 0.0752 \times 7,420.486$$

$$W'a = 33,481.233 \text{ (lb/h)}$$

4.10.5 Flue Gas Flow Rate

$$W'_{flue \text{ gas}} \text{ (lb/h)} = \text{Total fuel required (lb/lb flue gas)} \times m'_{fuel} \text{ (lb/h)} \times (1 + \text{excess air}) \quad (4.12)$$

$$W'_{flue \text{ gas}} = 11.51 \times 2,046.97 \times 2.20462 \times 0.78$$

$$W'_{flue \text{ gas}} = 40,514.94 \text{ (lb/h)}$$

4.10.6 Flue Gas Molecular Weight

$$MW_{flue \text{ gas}} = MW_{CO_2} \times lb_{CO_2} + MW_{H_2O} \times lb_{H_2O} + MW_{N_2} \times lb_{N_2}$$

$$MW_{flue \text{ gas}} = 44.01 \times 1.74 + 18 \times 1.39 + 28.016 \times 8.29$$

$$MW_{flue \text{ gas}} = 19.49$$

4.10.7 The actual temperature of the flame

Adiabatic combustion temperature is the highest temperature at which fuel and air may burn. In any case, this limit is never reached because of radiation losses and separation. In order to determine the ultimate temperature, multiple equations need to be resolved. In practical terms, the difference between the adiabatic and actual combustion temperatures is 3–5%.

It can be displayed as follows using an energy balance:

$$t_c = \frac{LHV + A \times \alpha \times HHV \times C_{pa} \times \left(\frac{t_a - 80}{10^6} \right)}{\left(1 - \% \text{ ash} / 100 + A \times \alpha \times HHV / 10^6 \right) \times C_{PG}} \quad (4.13)$$

A is the theoretical air needed per million Btu burnt, expressed in pounds; LHV and HHV are the lower and higher calorific values of fuel, expressed in Btu/lb; A = excess air factor = $1 + (E\%/100)$; t_a , t_c are air and combustion temperatures, in degrees Fahrenheit; C_{pa} , C_{pg} are the specific heats of air and combustion products, in Btu/lb.F.

$$t_c = \frac{5,138.63 + 650 \times 0.78 \times 5,616.64 \times 0.24 \times (68 - 80)/10^6}{(1 - \% / 100 + 650 \times 0.793 \times 5,616.64 / 10^6) \times 0.36}$$

$$t_c = 1,853.99 \text{ } ^\circ\text{F}$$

Actual flame temperature = adiabatic flame temperature \times 0.95

$$t_{ca} = 1,761.29 \text{ } ^\circ\text{F}$$

4.10.8 Flue Gas Temperature inside Furnace

There are several different methods to define the flue gas temperature (tg); some writers describe it as the temperature of the exit gas. According to some, it represents the theoretical flame temperature's mean. Experience seems to indicate that when measured and calculated values concur better, the winner is.

$$t_g = t_c + 300 \text{ to } 400 \text{ } ^\circ\text{F} \quad (4.14)$$

$$t_g = 1,761.29 + 300$$

$$t_g = 2,061.29 \text{ } ^\circ\text{F} = 1,127.383 \text{ } ^\circ\text{C} = 2,521.96 \text{ } ^\circ\text{R}$$

4.10.9 Furnace Pressure

Energy balance of mixtures of steam and water and fluid (28). We might estimate the length and width of the furnace and determine the flame length and width from the burner manufacturer's catalog.

$$P_{furnace} = P_{fan \text{ outlet}} - P_{loss \text{ air duct}} - P_{loss \text{ burner}} \quad (4.15)$$

$$P_{furnace} = 10.59 - 0.1295 \times 0.0241 - 166.67 \times 0.00094822$$

$$P_{furnace} = 10.43 \text{ psia}$$

4.10.10 Flue Gas Inlet Density

$$\rho_g = 492 \times MW \times \frac{P_{furnace}}{359 \times (460 + T) \times 14.7} \quad (4.16)$$

$$\rho_g = 492 \times 19.49 \times \frac{10.43}{359 \times (460 + 2,061.29) \times 14.7}$$

$$\rho_g = 0.00752 \text{ lb/cu.ft}$$

4.10.11 Flue Gas Volumetric Flow Rate

$$V'_{flue \text{ gas}} = \frac{m'_{flue \text{ gas}}}{\rho_{flue \text{ gas}}} \quad (4.17)$$

$$V'_{flue \text{ gas}} = 5,387,625 \text{ } ft^3/h$$

4.11 FURNACE DESIGN

4.11.1 Furnace Heat Surface Area

Furnace height =70 in

Length =55 in

Width =45in

4.11.2 Furnace Volume

$$V = L \times W \times H \quad (4.18)$$
$$V = \frac{55 \times 45 \times 70}{12^3}$$
$$V=100.26 \text{ ft}^3$$

4.11.3 Furnace Volumetric Heat Release

Furnace volumetric heat release should be lower than 300,000 kcal/m³h

$$\text{Volumetric heat release} = \frac{\text{Heat input to furnace}}{V_{\text{furnace}}} \quad (4.19)$$

$$\text{Volumetric heat release} = 22,736.625 \text{ Btu/ft}^3 \text{ h} = 202,201.255 \text{ kcal/m}^3 \text{ h}$$

4.11.4 Furnace Heat Surface Area

$$A_{\text{furnace}} = \frac{2 \times (H \times W + L \times H + W \times L)}{144} \quad (4.20)$$
$$A_{\text{furnace}} = 65.8 \text{ ft}^2$$

4.11.5 Furnace Exit Temperature

The bright portion of the flame and nonluminous gases in a boiler furnace cause transcendent radiation, which is insufficient for heat transmission. Using a vitality method, a common imprecise term for furnace absorption may be written.

$$Q_F = A_p \varepsilon_g \sigma (T_g^4 - T_0^4) = W_f LHV - W_g h_e \quad (4.21)$$

Where:

Q_F is furnace heat absorption(Btu/h),

A_p is furnace heat surface (ft²),

σ is the Stefan Boltzmann constant = 0.173×10^{-8} ,

T_{exit} is furnace exit gas temperature (R),

ε_g is emissivity of gas at T_g ,

T_0 is absolute temperature of tube surface (R) ,

W_f is fuel gas flow rate (lb/h),

W_g is flue gas flow rate (lb/h) and

$h_{average}$ is enthalpy at flue gas average temperature (Btu/lb).

The furnace exit temperature was assumed in the previous equation, and the enthalpy will be calculated using the average temperature. T_g will then be determined by solving both sides (left and right), and it should equal the previous assumption (28). T_{exit} was first taken to be equivalent to 648 °C:

$$T_{average} = \frac{T_g + T_{exit}}{2} \quad (4.22)$$

$$= 1,629.89 \text{ } ^\circ\text{F} = 887.72 \text{ } ^\circ\text{C}$$

Enthalpy is shown in Table:

$$h_{average} = 256.49 \text{ kcal/kg} = 461.37 \text{ Btu/lb}$$

The temperature drop across the steam film tube's internal temperature is added to the temperature drop across the gas film and metal conductivity to determine the tube's outside temperature (28).

It should be noted that a 67 °F difference in the tube's outer temperature has a maximum 7 °F impact on furnace exit temperature (28) and is used in the next calculation, which is done iteratively. The tube's outer temperature will be considered to be 430 °F.

$$A_p \varepsilon_g \sigma (T_{exit}^4 - T_o^4) = W_f LHV - W_g h_{average} \quad (4.23)$$

Table 4.2: Enthalpy of combustion product (kcal/kg)

Temp (°C)	Natural gas	Fuel oil
1,900	606.9	583.9
1,800	570.4	549.2
1,700	534.3	514.8
1,600	498.4	480.6
1,400	428.0	413.1
1,200	359.3	347.0
1,000	292.7	282.8
800	228.2	220.5

$$65.8 \times 0.16 \times 0.173 \times 10^{-8} \times (T_{exit}^4 - 430^4) =$$

$$= 2,046.97 \times 2.20462 \times 5,138.63 - 40,514.94 \times 461.37$$

$$T_{exit} = 1,731 \text{ } ^\circ\text{F} = 943.89 \text{ } ^\circ\text{C}$$

Thus, it is discovered that the temperature we imagined is rather close to the one that was computed.

4.12 EFFICIENCY OF COMBUSTION NONLUMINOUS HEAT TRANSFER

Nonluminous heat transmission is crucial in heat transfer equipment like boilers and superheaters, where gases transfer energy to fluid inside the tubes at high temperatures. The following can be used to describe the radiation compatibility of gases with their surroundings (such as a wall, tube bundle, or cavity):

$$\frac{Q}{A} = \sigma(\epsilon_g T_g^4 - \alpha_g T_o^4) \quad (4.24)$$

Where Q is heat transfer in furnace (Btu/h),

A is furnace heat surface (ft²),

σ is the Stefan Boltzmann constant = 0.173×10^{-8} ,

ϵ_g is emissivity of gas at T_g ,

T_g is the absolute temperature of gas, R

α_g is absorptivity at T_o ,

T_o is the absolute temperature of tube surface, R.

In spite of the fact that it is alluring to calculate heat flux, it is repetitive to estimate α_g at temperature T_o . We can use the following simplified equation by considering the reality that T_o^4 will be very smaller than T_{exit}^4 :

$$\frac{Q}{A} = \sigma(\epsilon_g T_g^4 - \alpha_g T_o^4) = h_N (T_g - T_o) \quad (4.25)$$

The nonluminous heat transfer coefficient h_N can be composed as follows:

$$h_N = \frac{\sigma \epsilon_g (T_g^4 - T_o^4)}{(T_g - T_o)} \quad (4.26)$$

$$h_N = \frac{0.173 \times 10^{-8} \times 0.16 \times (1,731^4 - 430^4)}{(1,731 - 430)}$$

$$h_N = 11.25 \text{ Btu/ft}^2 \text{ h F}$$

4.13 PRODUCT PROPERTIES OF FLUE GAS

Flue gas properties at average temperature can be

$$T_{\text{average}} = \frac{2,061.29 + 1,731}{2}$$

$$T_{\text{average}} = 1,896.15^\circ \text{F} = 1,035.64^\circ \text{C}$$

Table 4.3: Flue gas product properties

Temp (°F)	Cp	μ	K
2,200	0.33875	0.1236	0.0548
2,100	0.32335	0.11798	0.05231
2,000	0.30795	0.11236	0.04982
1,900	0.29255	0.10674	0.04733
1,800	0.2722	0.10113	0.044837

$$C_p = 0.2918 \text{ Btu/ft F}$$

$$\mu = 0.106523 \text{ lb/ft h}$$

$$K = 0.04723 \text{ Btu/ft h F}$$

Gas mass velocity, which can be computed as follows, is the mass flow rate over a unit range that is perpendicular to the direction of the velocity vector.

$$G = \frac{\dot{m}_{\text{flue gas}}}{A_{\text{furnace}}} \quad (4.27)$$

$$G = \frac{40,514.94}{65.8} = 615.73 \text{ lb/ft}^2 \text{ h}$$

4.13.1 Reynolds Number for the Flue Gas Product

The study of fluid flow is done with the use of the Reynolds number. The Reynolds number establishes the stability or unsteadiness of a fluid flow. (Laminar and turbulent) Fluids in a laminar flow follow streamlined paths. These streamlines disintegrate in turbulent flow, causing the fluid to travel erratically.

$$Re = \frac{Gd_{\text{tube}}}{12\mu} \quad (4.28)$$

$$Re = \frac{615.73}{12 \times 0.106523} = 481.69$$

4.13.2 Product Prandtl Number for Flue Gas

The ratio of momentum diffusivity to heat diffusivity is known as the Prandtl number (Pr), a dimensionless quantity that bears the name of the German physicist Ludwig Prandtl. It can be calculated as follows:

$$Pr = \frac{\mu C_p}{K} \quad (4.29)$$

$$= \frac{0.106523 \times 0.2918}{0.04723} = 0.65813$$

4.13.3 Product Nusselt Number for Flue Gas

The Nusselt number (Nu), a dimensionless quantity, can be defined as the ratio of convective to conductive heat transfer at a fluid barrier. Convection combined conduction with fluid motion. Measurements are made of conduction for a theoretically immobile fluid under the same circumstances as convection.

Heat transmission through pure conduction is represented by a Nusselt number. A laminar stream or slug stream has a characteristic value between 1 and 10. More dynamic convection is correlated with larger Nusselt numbers, which are often found in the 100–1,000 range, inside turbulent flow. Wilhelm Nusselt is the inspiration behind the Nusselt number, which is located as follows (28):

$$Nu = 0.33 \times Re^{0.6} \times Pr^{0.33} \quad (4.30)$$

$$Nu = 0.33 \times 481.69^{0.6} \times 0.65813^{0.33}$$
$$Nu=11.701$$

4.13.4 Coefficient of Combustion Convection Heat Transfer

The heat transfer coefficient, also known as the film coefficient or film effectiveness in thermodynamics and mechanics, is the complementary of thermal insulation and is expressed as follows (28): It is the steady-state proportionality between the heat flux and the thermodynamic driving force for the stream of heat (i.e., the temperature difference, ΔT):

$$h_c = \frac{12K \times NU}{d_{tube}} \quad (4.31)$$

$$h_c = \frac{12 \times 0.04723 \times 11.701}{1.3} = 5.10 \text{ Btu/ft}^2 \text{ h F}$$

4.13.5 Combustion outside Heat Transfer Coefficient

$$h_o = hc + h_N \quad (4.32)$$

$$h_o = 5.10 + 11.25 = 16.35 \text{ Btu/ft}^2 \text{ h F}$$

4.14 BOILER DESIGN PRESSURE

A normal condition's expected temperature and pressure are connected to design pressure. The highest pressure that is permitted in equipment or a vessel under typical operating conditions is known as the maximum authorized working pressure.

4.14.1 Steam Drum Saturated Pressure and Temperature

The term "boiler operating pressure," sometimes referred to as "working pressure," describes the operating pressure in the shell of a fire tube boiler and the steam drum of a water tube boiler. The following presumption can be used to compute steam drum pressure:

Table 4.4: Design specification of superheater

Superheater outlet pressure	28.55
Pressure drops in main stem stop valve + nonreturn valve	0.613
Pressure drops in main header	0.006
Spray attemperator loss	0.2267
Pressure drops in superheater bank	0.08
Pressure drops in boiler bank	0.0007
Steam drum operating pressure	29.48

Please take note that while the pressure drop indicated above is computed and can be used, we can estimate and calculate the drum pressure in the first iteration and determine the precise drum pressure by computing the data. The water temperature in the drum and bank tubes is 329.2 °F because the steam drum pressure is saturated.

4.14.2 Steam Drum First Safety Valve Setting Pressure

$$P_{PSV1} = P_{drum} \times 1.05 \times 1.03 \quad (4.33)$$

$$P_{PSV1} = 29.48 \times 1.05 \times 1.03 = 31.883 \text{ kg/cm}^2$$

4.14.3 Steam Drum Second Safety Valve Setting Pressure

$$P_{PSV2} = P_{PSV1} + 0.34 \quad (4.34)$$

$$P_{PSV2} = 31.883 + 0.34 = 32.22 \text{ kg/cm}^2$$

4.14.4 Boiler Design Pressure

$$P_{Design} = P_{PSV2} \quad (4.35)$$

$$P_{Design} = 32.22 \text{ kg/cm}^2$$

4.15 SUPERHEATER

Saturated steam is overheated (superheated) by the superheater, a heat exchanger. Saturated steam can be superheated to raise its temperature above that of the saturated steam and so boost the process's efficiency of producing energy. In addition, superheated steam finds application in non-electric facilities.

Using superheated steam has the following advantages: • Lower moisture content; • Reduced condensate in steam pipes; • Enhanced energy production efficiency

Typically, the superheater is made up of tubes that carry steam, which is heated by flue gases that travel outside the tubes. Headers are typically used to connect the tubes in parallel, with steam entering through one header and leaving through another. A boiler that serves as a superheater for heating external steam—steam that has already been used in a process outside the boiler—can contain multiple superheater units.

Superheaters raise the saturated steam's temperature to provide the necessary process conditions. Superheaters are one-phase heat exchangers where flue gas exits the tube and steam enters the tube.

Saturated steam is separated from water in the steam drum area at boiling point and passes via the superheater tubes. The surface temperatures of superheaters are higher than those of boiler tubes. The intended steam temperature determines the tube material. Up to 400°C, carbon steel tubes can be used, and up to 660°C, chrome-moly steel tubes. The superheating circulation and the kind of heat transfer—convection, radiation, or a combination of the two—are taken into consideration during design. It is possible to use counterflow, parallel flow, or combination parallel and counter flow.

4.15.1 Superheater Heat Duty Prediction

It is necessary for some heat to go from the hot side to the cold side over a period of time stated in terms of heat. Using the following calculations, we should first determine the heat duty of the superheater package:

Table 4.5: Superheater heat duty values

Superheater steam inlet temperature	319.3 °F
Superheater inlet pressure	419.34 Psig
Superheater inlet enthalpy	801.97 Btu/lb
Superheater steam outlet temperature (controlled)	525.33 °F
Superheater outlet pressure	466.1 Psig
Superheater outlet enthalpy	933.43 Btu/lb
Steam mass flow	88,184.8 lb/h

$$Q_{duty} = (933.43 - 801.34)\text{Btu/lb} \times 88,184.8\text{lb/h} = 11.65\text{MMBtu/h}$$

4.15.2 Superheater Tube Thickness

To determine the minimum required thickness of tubing, you will use the following formula:

$$t_{min} = \frac{PD}{2S + P} + 0.005D + e \quad (4.36)$$

Where, according per ASME 1, PG-27.4, t is the minimum required tube thickness in inches, D is the tube diameter in inches, S is the maximum permitted stress for A213-T11 in pounds, and e is the thickness factor for expanded tube ends.

$$t_{min} = \frac{32.22 \times 14.2233 \times 1}{2 \times 6,850 + 32.22 \times 14.2233} + 0.005 \times 1 + 0.04$$

$$t_{min} = 0.0774 \text{ in}$$

As per tube manufacturer catalog for 1 in diameter, they produce tube with thickness of 0.1 in. So: $t_{selected} = 0.1 \text{ in}$

4.15.3 Superheater Tube Area

$$A_{Superheater \ tube} = \frac{\pi}{144} \times \left(\frac{D-t}{2} \right)^2 \quad (4.37)$$

$$A_{Superheater \ tube} = \frac{\pi}{144} \times \left(\frac{1 - 0.1}{2} \right)^2$$

$$A_{Superheater \ tube} = 0.00349 \text{ ft}^2$$

4.15.4 Superheater Tube Rows and Deep Number

Steam velocity range, 1,300–3,200 fpm [41]

Selected steam velocity = 1,300 fpm

Saturated steam specific volume = 0.473 ft³/lb

Superheater steam volumetric flow rate = 62,526.87 ft³/h

Selected tube row deep no. = 4

$$Tube \ row \ no. = round_{up} \left(\frac{V'_{Superheater \ steam}}{V_{selected} \times 60 \times (tube \ row \ no.)} \right) \quad (4.38)$$

$$Tube \ row \ no. = round_{up} \left(\frac{62,526.87}{1,300 \times 60 \times (4)} \right)$$

$$Tube \ row \ no. = 13$$

4.16 SUPERHEATER CONVECTIVE HEAT TRANSFER COEFFICIENT

The term "heat transfer coefficient" refers to a proportionality constant between the heat flux and the thermodynamic driving force for heat flow (28).

4.16.1 Superheater Flue Gas Outlet Temperature Prediction

$$T_{g \text{ outlet}} = T_{g \text{ inlet}} - \frac{Q_{\text{superheater}}}{m'_{\text{flue gas}} \times C_{P \text{ average as temperature}} \times (1 - \text{heat loss} / 100)} \quad (4.39)$$

The primary flue gas outlet temperature and the specific heat at average temperature must be determined at this step. The flue gas outlet temperature can then be calculated by repeating this process (28):

$$T_{g, \text{outlet}} = 1,731 - \frac{11.65 \times 10^6}{40,514.94 \times 0.3449 \times (1 - 2/100)}$$

$$T_{g \text{ outlet}} = 880.27^\circ\text{F} = 471.26^\circ\text{C}$$

4.16.2 Log Mean Temperature Difference Prediction

The temperature driving force in heat transfer areas, such as heat exchangers, can be calculated using the log means temperature difference, or LMTD. Heat exchangers transmitted more heat by greater LMTD (28) for a certain size and heat transfer coefficient:

$$\Delta T_{\log} = \frac{(T_{g \text{ outlet}} - T_{\text{outlet steam}}) - (T_{g \text{ inlet}} - T_{\text{inlet steam}})}{LN \frac{T_{g \text{ outlet}} - T_{\text{outlet steam}}}{T_{g \text{ inlet}} - T_{\text{inlet steam}}}} \quad (4.40)$$

$$\Delta T_{\log} = \frac{(880.27 - 500) - (1,731 - 329.2)}{LN \frac{(880.27 - 500)}{(1,731 - 329.2)}}$$

$$\Delta T_{\log} = 783 \text{ F}$$

4.16.3 Superheater Average Flue Gas Temperature Prediction

$$T_{\text{average}} = \frac{T_{g \text{ inlet}} + T_{g \text{ -predicted outlet}}}{2} \quad (4.41)$$

$$T_{\text{average}} = \frac{880.27 + 1,731}{2}$$

$$T_{\text{average}} = 1,305.64^\circ\text{F}$$

4.16.4 Superheater Average Flue Gas Properties

Superheater average flue gas properties can be found from gas tables at average temperature:

$$\mu = 0.1298 \text{ lb/ft h}$$

$$C_p = 0.3449 \text{ Btu/lb F}$$

$$K = 0.0585 \text{ Btu/ft h F}$$

4.16.5 Superheater Tube Longitudinal and Transverse Pitch

Other than heat absorption and gas flow resistance, convection surface considerations include the ideal tube spacing and layout.

Transverse and longitudinal space will be chosen initially for the superheater package design. The ideal space will then be found by evaluating the draft pressure loss and performance outcome. Transverse and longitudinal pitches are now taken into consideration as 4 and 4.03125, respectively.

4.16.6 Superheater Package Long

$$L_{\text{superheater}} = \text{Round} \left[\text{ref. No. } S_L \times (N_{\text{tube row}} - c1) \right] \quad (4.42)$$

$$L_{\text{superheater}} = \text{Round} [\text{ref. No. } 4.03125 \times (13-1)]$$

$$L_{\text{superheater}} = 48.375 \text{ in}$$

4.16.7 Superheater Primary Heat Surface Area

By verifying the primary length selection, the tube length connected to superheater duty and outlet temperature must be at its ideal. Heat surface is determined by assuming that each tube is 30 feet long.

$$A_{\text{superheater}} = \pi \times D_{\text{tube}} / 12 \times N_{\text{tube row}} \times N_{\text{tube row deep}} \times L_{\text{superheater tube}} \quad (4.43)$$

$$A_{\text{superheater}} = \pi \times 1.5/12 \times 4 \times 13 \times 27$$

$$A_{\text{superheater}} = 551.35 \text{ ft}^2$$

4.16.8 Superheater Gas Mass Velocity

The velocity of gas mass can be calculated as (28): The mass flow rate across a unit area that is vertical to the direction of the velocity vector is known as the gas mass velocity.

$$G = \frac{W'}{N_{\text{row deep}} \times L_{\text{tube}} \times (S_T - d)} \quad (4.44)$$

$$G = 12 \times \frac{40,514.94}{4 \times 26 \times (4 - 1.5)}$$

$$G = 1,869.92 \text{ lb/ft}^2\text{h}$$

4.16.9 Superheater Convective Heat Transfer Coefficient

$$h_c = 0.9 \times \frac{G^{0.6}}{D_{tube}^{0.4}} \times \frac{K^{0.67} \times C_p^{0.33}}{\mu^{0.27}} \quad (4.45)$$

$$h_c = 0.9 \times \frac{1,869.92^{0.6}}{1.5^{0.4}} \times \frac{0.0585^{0.67} \times 0.3349^{0.33}}{0.1298^{0.27}}$$

$$h_c = 12.69 \text{ Btu/ft}^2 \text{ h F}$$

4.17 ECONOMIZER HEAT DUTY PREDICTIONS

Heat exchangers called economizers raise the temperature of fluids, usually water. Economizers increase the boiler's efficiency by recovering more enthalpy from waste gas. By preheating the cold water that will be used as the feed water, they conserve energy from exhaust gasses (28).

Table 4.6: Specification of economizer

Economizer water inlet temperature	172.5	°F
Economizer inlet enthalpy	149.85	Btu/lb
Economizer water outlet temperature	258.9	°F
Economizer outlet enthalpy	238.298	Btu/lb
Blow down rate	3%	%
Economizer mass flow	102,184.14	lb/h

$$Q_{duty} = (h_{outlet \text{ water}} - h_{inlet \text{ water}}) \times \text{Evaporation @ 100\% MCR} \quad (4.46)$$

$$Q_{duty} = (238.298 - 149.85) \text{ Btu/lb} \times 102,184.14 \times (1 + 3\%) \text{ lb/h}$$

$$Q_{duty} = 12.045 \text{ MM Btu/h}$$

4.17.1 Economizer Design Pressure

The following standards will be used to determine the design pressure, with the company's approval being the exception (23).

For maximum normal operating pressure less than 1.125 barg, use 2.625 bar gage.

For maximum normal operating pressures between 1.125 and 15 barg, use the maximum normal operating gage pressure + 1.5 bar.

For maximum normal operating pressures between 15 and 64 barg, use 110% of the maximum normal operating gage pressure.

Economizer design pressure: $541.8 \times 1.1 = 595.98 \text{ psig}$

4.17.2 Economizer Tube Thickness

$$t_{min} = \frac{PD}{2S + P} + 0.005D + e \quad (4.47)$$

Where P is the economizer design pressure in psig; D is the tube diameter in inches (assuming 1.5 in); and t is the minimum needed tube thickness in inches. As per ASME 1, PG-27.4, e is the thickness factor for expanded tube ends and S is the maximum allowed stress for A178-Gr.A, expressed in psi.

$$t_{min} = \frac{595.98 \times 1.5}{2 \times 12,400 + 595.98} + 0.005 \times 1.5 + 0$$

$$t_{min} = 0.0427 \text{ in}$$

As per tube manufacture catalog for 1.5-in diameter, they produce tube with thickness 0.085 in. So, $t_{selected} = 0.064$ in.

4.17.3 Economizer Tube Area

$$A_{economizer\ tube} = \frac{\pi}{144} \times \left(\frac{D-t}{2} \right)^2 \quad (4.48)$$

$$A_{economizer\ tube} = \frac{\pi}{144} \times \left(\frac{1.5 - 0.064}{2} \right)^2$$

$$A_{economizer\ tube} = 0.0112 \text{ ft}^2$$

4.17.4 Economizer Tube Arrangements

The most common configurations for economizers are bare tube, inline, and crossflow. We can reduce erosion and ash trapping by utilizing bare tube and inline configuration. The simplest geometry that soot blowers can maintain clean is this configuration. Please be aware that the weight, volume, and cost of this arrangement are greater (28).

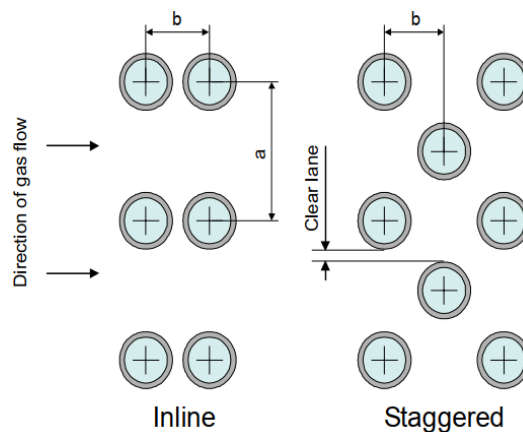


Figure 4.3: Economizer tube arrangement.

4.18 ECONOMIZER CONVECTION HEAT TRANSFER COEFFICIENT

4.18.1 Economizer Obstruction Surface Area

$$A_{obs} = \frac{d}{12} + \frac{n_{fin} b H_{fin}}{6} \quad (4.49)$$

$$A_{obs} = \frac{1.5}{12} + \frac{6 \times 0.03 \times 0.563}{6}$$

$$A_{obs} = 0.142 \text{ ft}^2/\text{ft}$$

4.18.2 Economizer Gas Mass Velocity

$$G = \frac{G_{flue\ gas}}{\left(\frac{S_T}{12} - A_{obs}\right) \times N_{row\ deep} \times L_{fin\ tube}} \quad (4.50)$$

$$G = \frac{128,114.19}{\left(\frac{3.375}{12} - 0.1967\right) 9 \times 7.5}$$

$$G = 22,448.114 \text{ lb/ft}^2 \text{ h}$$

4.18.3 Economizer Inlet Flue Gas Properties

Flue gas properties at inlet temperature can be found

$$T_{\text{gas inlet}} = 496.973 \text{ }^\circ\text{F}$$

Table 4.7: Flue gas product properties

Temp (°F)	C _p	μ	K
800	0.29070	0.07500	0.02870
700	0.28695	0.07063	0.02680
600	0.28320	0.06625	0.02490
500	0.27945	0.06188	0.02300

$$\mu = 0.0706 \text{ lb/ft h}$$

$$C_p = 0.2870 \text{ Btu/lb F}$$

$$K = 0.0268 \text{ Btu/ft h F}$$

4.18.4 Economizer Flue Gas Reynolds Number

$$Re = \frac{5,986.643 \times 2}{9 \times 0.0706}$$

$$Re = 18,843.698$$

4.18.5 Economizer Overall Heat Transfer Coefficient

Overall heat transfer coefficient, U, for extended surfaces can be obtained from [5]

$$\frac{1}{U} = \frac{A_t}{h_i A_i} + ff_i \frac{A_t}{A_i} + ff_o \frac{A_t}{A_w} \times \frac{d}{24K_m} \times \ln \frac{d}{d_i} + \frac{1}{\eta h_o} \quad (4.51)$$

Where A_t is surface area of finned tube, ft²/ft;

A_i is tube inner surface area = $\pi d/12$, ft²/ft;

A_w is average wall surface area = $\pi (d + d_i)/12$, ft²/ft;

K_m is thermal conductivity of the tube wall, Btu/ft.h.F;

d_i, d is tube inner and outer diameter, inch;

ff_i and ff_o are fouling factor inside and outside of the tube, ft² h F/Btu;

h_i, h_o is tube side and gas side coefficient, Btu/ft² h F; and

η is fin effectiveness.

Equation can be simplified t:

$$\frac{1}{U} = \frac{A_t}{h_i A_i} \quad (4.52)$$

$$U = \eta h_o$$

$$U = 0.72 \times 6.18$$

$$U = 4.4496 \text{ Btu/ft}^2 \text{ hF}$$

4.19 ECONOMIZER TUBES ROW NUMBER

4.19.1 Economizer Log Mean Temperature Difference Prediction

$$\Delta T_{log} = \frac{(T_{g \text{ inlet}} - T_{outlet \text{ water}}) - (T_{g \text{ outlet}} - T_{inlet \text{ water}})}{\ln \frac{(T_{g \text{ inlet}} - T_{outlet \text{ water}})}{(T_{g \text{ outlet}} - T_{inlet \text{ water}})}} \quad (4.53)$$

$$\Delta T_{log} = \frac{(496.97 - 258.9) - (240 - 172.5)}{\ln \frac{(496.97 - 258.9)}{(240 - 172.5)}}$$

$$\Delta T_{log} = 135.33 \text{ } ^\circ\text{F}$$

4.19.2 Economizer Heat Surface Prediction

$$A_{economizer} = \frac{\text{Heat duty}}{h_c \times \Delta T_{log}} \quad (4.54)$$

$$A_{economizer} = \frac{12.067 \times 10^6}{10.75 \times 135.33}$$

$$A_{economizer} = 8,294.63 \text{ ft}^2$$

4.19.3 Economizer Tubes Row Number

$$N_{\text{tube row}} = \text{Round up} \left(\frac{A_{\text{economizer}}}{A_t \times L_{\text{fin tube}} \times N_{\text{tube row deep}}} \right) \quad (4.55)$$

$$N_{\text{tube row}} = \text{Round up} \left(\frac{8,294.625}{7.09 \times 7.5 \times 9} \right)$$

$$N_{\text{tube row}} = 10$$

4.19.4 Economizer Final Heat Surface

$$A_{\text{economizer}} = N_{\text{tube row deep}} \times A_t \times L_{\text{fin tube}} \times N_{\text{tube row}} \quad (4.56)$$

Tube row number should be finalized by the economizer outlet water temperature.

$$A_{\text{economizer}} = 12 \times 7.09 \times 10 \times 10$$

$$A_{\text{economizer}} = 8,512.54 \text{ ft}^2$$

4.20 ECONOMIZER PACKAGE PERFORMANCE

4.20.1 Economizer Package Performance Prediction

Duty and exit temperatures can be predicted by number of transfer units (NTU) method. Fundamentally, the duty Q is given by (23)

$$Q = \varepsilon C_{\min} (T_{\text{gas inlet}} - T_{\text{steam inlet}}) \quad (4.57)$$

Where, ε depends on the type of flow, which can be counter flow or parallel flow or cross flow. In economizer, usually a counter flow arrangement is adopted. ε for this is given by

$$\varepsilon = \frac{1 - \exp[-NTU \times (1 - C)]}{1 - C \times \exp[-NTU \times (1 - C)]} \quad (4.58)$$

Where

$$NTU = \frac{UA}{C_{\min}}$$

$$C = \frac{C_{\min}}{C_{\max}}$$

$$C_{\min} = (W' C_p)_{\min}$$

$$C_{\max} = (W' C_p)_{\max}$$

At our design case,

Inlet water heat capacity @ constant volume = 0.886 Btu/lbF

Inlet flue gas heat capacity = 0.2870 Btu/lbF

$$W'_{\text{BFW}} = 102,184.14 \text{ lb/h}$$

$$W'_{\text{flue gas}} = 128,113.65 \text{ lb/h}$$

$$C_{\text{max}} = 102,184.14 \times 0.886 = 90,535.15$$

$$C_{\text{min}} = 0.2870 \times 128,113.65 \text{ lb/h} \times \left(1 - \frac{2}{100}\right) = 36,768.62$$

$$C = \frac{36,768.62}{90,535.15} = 0.406$$

Since gas temperature is low, nonluminous heat transfer is also low and can be neglected.

$$NTU = \frac{7.74 \times 6,384.405}{36,768.62} = 1.36$$

$$\varepsilon = \frac{1 - \exp[-1.36 \times (1 - 0.4)]}{1 - 0.4 \times \exp[-1.36 \times (1 - 0.4)]} = 0.67$$

$$Q = 0.67 \times 36,768.62 \times (496.97 - 172.5)$$

$$Q = 7.14 \text{ MM Btu/h}$$

4.20.2 Economizer Outlet Water Temperature

$$T_{\text{water outlet}} = T_{\text{water inlet}} + \frac{Q}{C_p \times W'_{\text{water}}} \quad (4.59)$$

$$T_{\text{water outlet}} = 172.5 + \frac{7.14 \times 10^6}{90,535.15 \times 0.886}$$

$$T_{\text{water outlet}} = 251.36 \text{ }^\circ\text{F}$$

Therefore, it is okay that the exit temperature is higher and closer to what we anticipated from our economizer design. If not, the tube's length or diameter should be adjusted.

$$T_{\text{gas outlet}} = T_{\text{gas inlet}} - \frac{Q}{C_p \times W'_{\text{gas}}} \quad (4.60)$$

$$T_{\text{gas outlet}} = 496.97 - \frac{7.14 \times 10^6}{0.2870 \times 128,113.65}$$

$$T_{\text{gas outlet}} = 302.783^\circ\text{F}$$

4.21 CIRCULATION RATIO (CR)

CR usually at low-pressure boilers (<750 psia) will be in the range from 15 to 37.5, and high-pressure boilers (750–2,025 psia) will be in the range from 6.75 to 3.75. Calculation will be as follows:

$$CR = \frac{1}{x} \quad (4.61)$$

Flow through the evaporator = CR the steam generated

$$CR_{\text{assumed}} = 30$$

$$Q_{\text{leaving steam from drum}} = \frac{1}{30} = 0.0333$$

4.21 FURNACE HEAT ABSORPTION

$$Q_{\text{furnace absorption}} = W_f LHV - W_g h_{\text{average}} \quad (4.62)$$

$$Q_{\text{furnace absorption}} = 2,156.12 \times 2.20462 \times 19,747.8 - 128,114.19 \times 653.08$$

$$Q_{\text{furnace absorption}} = 10,200,041.22 \text{ Btu/h}$$

4.22 DRUM LEAVING STEAM ENTHALPY

$$h_{\text{drum leaving steam}} = \frac{1}{CR} \times h_{\text{saturated vapour}} + \left(1 - \frac{1}{CR}\right) \times h_{\text{saturated liquid}} \quad (4.63)$$

$$h_{\text{drum leaving steam}} = \frac{1}{30} \times \frac{2,098.56}{2.326} + \left(1 - \frac{1}{30}\right) \times \frac{908.18}{2.326}$$

$$h_{\text{drum leaving steam}} = 374.95 \text{ Btu/lb}$$

4.23 BOILER EFFICIENCY

The boiler's efficiency will be determined by calculating a number of losses, such as radiation, unburned fuel, heat loss from molten ash, and leaving flue gas losses. The reader has the option to look up more ASME power test code details. There are two methods available for calculating boiler efficiency (28).

Boiler efficiency will be calculated using the following sequence:

Boiler efficiency based on input–output method

Boiler efficiency based on heat loss method but this one needs

4.23.1 Boiler Efficiency Based On Input–Output Method

$$\eta_{\text{input-output}} = 1 - \frac{Q_{\text{Free to atmosphere}}}{Q_{\text{Produce by burner}}} \quad (4.64)$$

Where: $Q_{\text{Free to atmosphere}} = Q_{\text{Produce by burner}} - 1.02 \times (Q_{\text{Absorption by furnace}} + Q_{\text{Absorption by super heater}} +$

$Q_{\text{Absorption by bank tube}} + Q_{\text{Absorption by economizer}} + Q_{\text{Loss from stack}})$

$$Q_{\text{Produce by burner}} = 183.66 \times 10^6 \text{ Btu/h}$$

$$Q_{\text{Absorption by furnace}} = Q_{\text{Input heat}} - Q_{\text{Free heat}} = m_{\text{fuel}} \times LHV - W_{\text{Flue gas}} \times h_{\text{average}}$$

$$Q_{\text{Absorption by furnace}} = 3,835.1 \times 2.20462 \times 19,747.8 - 170,818.92 \times 810.54$$

$$Q_{\text{Absorption by furnace}} = 28,511,278 \text{ Btu/h}$$

$$Q_{\text{Absorption by super heater}} = Q_{\text{heat duty}} = 35,773,872 \text{ Btu/h}$$

$$Q_{\text{Absorption by bank tube}} = Q_{\text{flue gas}} = 57,036,325 \text{ Btu/h}$$

$$Q_{\text{Absorption by economizer}} = Q_{\text{duty}} = 14,141,758 \text{ Btu/h}$$

$$Q_{\text{Loss from stack}} = Q_{\text{stack outer wall}} = 749,687 \text{ Btu/h}$$

$$Q_{\text{Free to atmosphere}} = 183.66 \times 10^6 - 1.02 \times (28,511,278 + 35,773,872 + 57,036,325 + 14,141,758 + 749,687)$$

$$Q_{\text{Free to atmosphere}} = 44,726,479 \text{ Btu/h}$$

$$\eta_{\text{input-output}} = 1 - \frac{44,726,479}{183.66 \times 10^6}$$

$$\eta_{\text{input-output}} = 75.64\%$$

4.24 BOILER MATERIAL

4.24.1 Introduction

The primary components of a boiler are steels and refractories, which are heated to high temperatures. Structural steels are utilized for support, while sheet materials are used for the transportation of gas and air (29).

Thus, a range of steels and refractories are utilized in the construction of boilers. Steel rounds in the shape of tubes, pipes, and headers are typically used to construct boilers. Due to their enormous diameters, even the drums and heads are rounds rolled from plates (29).

Bricks, refractories, insulation, and lagging (BRIL) make up the majority of the nonmetallic materials, and they are mostly used to cover boilers.

Table 4.8 Materials of boiler parts

Selected material	Used to	Code(ASME)
Alloy steel	Economiser	A/SA213T ₁
High carbon steel	Steam Drum	SA-178
Low-carbon steel	Superheater	SA213 T ₁₂
Steel	Furnace evaporator	SA 299
Firebrick and structural steel	Furnace wall	-
Stainless steel	Steam pipe	B36
Fiberglass	Insulated material for steam pipes	-

2.25 DESIGN OF GAS BURNER

The boiler is currently using an anaerobic digester and flaring off the biogas. The biogas, mainly consisting of methane, is a potential fuel source for the brewery, animal dug etc. The system designed makes use of this methane in a biogas powered steam boiler. Biogas is

generated when organic compounds containing carbon, hydrogen, and oxygen are consumed by bacteria in the absence of oxygen, which produces carbon dioxide and methane. Under this study not study about biogas digester because of scope of study.

2.25.1 Essential Subject for Burner Design

The burner in an industrial furnace is an essential component that produces and regulates the heat required for a variety of industrial processes. An industrial burner's main function is to create a regulated flame that releases heat. The burner is in charge of starting the fuel (biogas, for example) and making sure it burns completely inside the furnace chamber. The creation of industrial burners requires an understanding of basic combustion as well as topics like turbulent and multiphase flow, heat and mass transfer, and so forth.

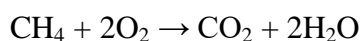
4.25.2 DESIGN OF BIOGAS BURNER

Table 4.9: Properties of biogas

Property	Value
Methane and carbon dioxide content	60% and 40% (v/v) respectively
Calorific value	22MJ/m ³
Specific gravity	0.940
Viscosity μ	$1.71 \times 10^{-5} \text{kg/m}^3$
Flame speed factor	11.1
Air requirement for combustion	$5.7 \text{ m}^3/\text{m}^3$
Combustion speed	40cm/sec
Inflammability in air	6-25%
Wobbe No.	732
Ignition temperature	650-750°C
Critical temperature	-82.5°C
Critical pressure	75-89 bar

4.25.3 Biogas Combustion

Biogas burns in oxygen to give carbon dioxide and water:



To produce one volume of carbon dioxide and two volumes of steam, one volume of methane needs two volumes of oxygen. Given that biogas contains 60% methane and air contains 21% oxygen:

$$\frac{1}{0.6} = 1.67 \text{ Volumes of biogas required}$$

$$\frac{2}{0.21} = 9.52 \text{ Volumes of air, or}$$

The stoichiometric air demand for 1 volume of biogas is $9.52 / 1.67 = 5.7$ volumes. Consequently, 5.7 is the stoichiometric air-fuel ratio.

Biogas will burn over a fairly narrow range of mixtures from 9% to 17% biogas in air (24).

4.25.4 Design of Biogas Burners

The burner designed to supply about 20 kW (Output power) for baking injera using five baking ovens for one cycle. For the biogas the estimated calorific efficiency is 75%. So the required heat input power is calculated as follows.

$$\text{Efficiency} = \frac{\text{Output power}}{\text{Input power}} * 100 \quad (6.65)$$

$$75 = \frac{20kW}{\text{Input power}} * 100$$

Therefore, the required input power is 26.67 kW or 96.012MJ/h. The calorific value (Cv) of biogas is 21.6MJ/m³.

Thus the flow rate required is given as:

$$\dot{V} = \frac{\text{Input power}}{\text{CV}} = \frac{96.012\text{MJ/h}}{21.6 \text{ MJ/m}^3} = 4.445 \text{ m}^3/\text{hr} \quad (6.66)$$

The biogas flow rate required is $\dot{V} = 4.445 \text{ m}^3/\text{hr}$ and a gas supply pressure of 60 mbar.

4.25.5 Injector Design for Burner

The injector size is calculated:

$$d_o = \sqrt{\frac{\dot{V}}{0.036 * C_d}} * \sqrt[4]{\frac{S}{P}} \quad (4.67)$$

$$d_o = \sqrt{\frac{4.445}{0.036 * 0.9}} * \sqrt[4]{\frac{0.94}{60}}$$

Where: $S_o = 4.1439 \text{ mm} = 5 \text{ mm}$; $A_o = \text{open area (mm}^2)$; $p = \text{gas pressure before opening (mbar)}$; $S = \text{specific gravity of gas (0.94)}$; and $C_d = \text{nozzle discharge factor (0.9)}$.

The area of the injector becomes:

$$A_o = \pi \frac{d_o^2}{4} = \frac{\pi \times 5^2}{4} \quad (4.68)$$

$$A_o = 19.635 \text{ mm}^2$$

4.25.6 Determine the Velocity of the Gas in the Orifice

The velocity of the gas in the orifice is calculated:

$$v_o = \frac{\dot{V}}{3.6 \times 10^{-3} \times A_o} \quad (4.69)$$

$$v_o = 80.5 \text{ m/s}$$

4.25.7 Design of Burner Throat

Determine the Size of Throat

The size of the throat is calculated using Prigg's formula:

$$d_t = \left(\frac{r}{\sqrt{S}} + 1 \right) * d_o \quad (4.70)$$

The stoichiometric air ratio requirement is 5.7 based on a complete combustion analysis, in which case the penetration rate (r) or primary aeration should be 50% of stoichiometric air.

So, r=50% of stoichiometric air

$$r = 0.5 \times 5.7 = 2.85$$

$$d_t = \left(\frac{2.85}{\sqrt{0.94}} + 1 \right) * 5$$

$$d_t = 98.49 \text{ mm}$$

4.25.8 Determine the Gas Pressure just after Nozzle

The gas pressure just after the nozzle or at the beginning of the throat then calculated as:-

$$P_A = P_o - \rho \frac{V_o^2}{2g} \left[1 - \left(\frac{d_o}{d_t} \right)^4 \right] \quad (4.71)$$

$$P_A = 10^5 - 1.0994 * \frac{80.5^2}{2 * 9.81} \left[1 - \left(\frac{5}{98.49} \right)^4 \right]$$

$$P_A = (100,000 \text{ Pa} - 396.797 \text{ Pa}) = 99,603 \text{ Pa} = 99.603 \text{ kPa}$$

Where: - p_t is the gas pressure (N m^{-2}), P_o is atmospheric pressure, as the throat is open to the air, ρ is the gas density (kg m^{-3}), and g is the acceleration due to gravity (9.81 ms^{-2})

Due to in this way weight distinction the essential discuss can connect into the throat from the encompassing. The discuss gulf ports must have an zone comparable to that of the throat zone.

4.25.9 Determine the Mixture Flow Rate at Optimum Aeration

The mixture flow rate at optimum aeration is given as:-

$$Q_m = \frac{Q * (1 + r)}{3600} \quad (4.72)$$

$$Q_m = \frac{4.445 * (1 + 2.85)}{3600} = 0.0183 \text{ m}^3/\text{s}$$

4.25.10 Determine the Pressure Drop in the Mixing Tube

The length of the mixing tube is given as:-

$$L_m = 10 \times d_t$$

$$L_m = 10 \times 98.49 \text{ mm} = 984.9 \text{ mm}$$

Calculate the Reynolds number:

$$Re = \frac{4\rho Q_m}{\pi\mu d_t} \quad (4.73)$$

Where: ρ and μ are the density and viscosity for the mixture (Use $\rho = 1.15 \text{ kg/m}^3$ and $\mu = 1.71 \times 10^{-5} \text{ Pa}$ at 30°C). \dot{Q}_m , is mixture flow rate in (m^3/s) and d_t is throat diameter in (m).

$$Re = \frac{4 * 1.15 * 0.0183}{\pi * 1.71 * 10^{-5} * 98.49 * 10^{-3}}$$

$$Re = 15,910$$

Re is >2000 , therefore, the pressure drop in the mixing tube is given as:-

$$\Delta P = \frac{f}{2} \rho \frac{16Q_m^2}{\pi^2 d_t^5} L_m \quad (4.74)$$

$$f = \frac{0.316}{15,910^{1/4}}$$

$$f = 0.0281$$

$$\Delta P = \frac{0.0281}{2} \frac{16 * 0.0183^2}{\pi^2 * 0.09849} * 0.984$$

As a result, $\Delta p=5.23$ pa is the pressure drop in the mixing tube. Compared to the driving pressure in the throat (363 pa), the pressure drop in the mixing tube is substantially smaller. Additionally, the tube's length is adjustable at $L_m=984.9$ mm.

4.25.11 Determine the Number of Flame Ports for a Burner

Determine the Flame Port Area

The mixture supply velocity (v_p) is given by:

$$v_p = \frac{Q_m}{A_p} \ll 0.25m/s \quad (4.75)$$

The total burner area can now be chosen:

$$\begin{aligned} A_p &> \frac{Q_m}{0.25} > \frac{0.97 \times 10^{-3}}{0.25} \\ &= 3.88 \times 10^{-3} \approx 0.004m^2 \end{aligned}$$

4.25.12 Determine the Number of Flame Port

The number of flame port is calculated as follow:-

$$n_p = \frac{4 \times A_p}{\pi d_p^2} \quad (4.76)$$

Where: n_p is estimated number of flame ports.

Using 4 mm diameter holes, the total number of holes required for one burner is given as:

$$\begin{aligned} n_p &= \frac{4 * 0.004}{\pi \times (0.004)^2} \\ n_p &= 318.4 \approx 318 \end{aligned}$$

It should be feasible to lower the number of flame ports for flame stabilization. Therefore, by completing Prigg's formula, the appropriate flame port for a single burner can be found. If the entire flame port area (A_p) is 1.5 to 2.2 times the throat's area, the Prigg's formula is valid.

$$x_r = \frac{A_p}{A_t} = n_p \left(\frac{dP}{dt} \right)^2 \quad (4.77)$$

Where: - x_r is the ratio of the total flame port area to the area of throat.

Choose $x_r = 2$ which is between 1.5 to 2.2 then calculate the sufficient number of flame ports for a single burner.

$$2 = n_p \left(\frac{4mm}{15.8mm} \right)^2$$

$$n_p=31.2\approx 32$$

4.26 BURNER'S CAD MODEL

The ANSYS flame port combustion model was fed the mixing model's output, which used 170 liters per hour. The emissions species, flame and zone temperatures, and other parameters generated by this model were utilized in an equally weighted ranking process to determine which three designs performed the best.

By using contemporary software tools, it was hoped to facilitate quick design iterations and optimization without requiring manufacturing investment.

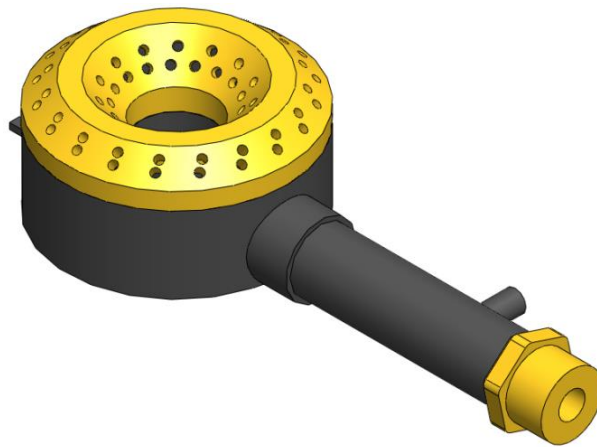


Figure 4.4: CAD modeling of burner

4.18.1 Graphical Representations

Through the 3-D rendering, the mixing model also monitored the movement of various species, such as carbon dioxide and methane in biogas and oxygen and nitrogen in the air. The entrainment effect was visualized by calculating the nitrogen mass fraction at each point in the 3-D rendering. The homogeneity of the fuel and air at the flame port was confirmed by this image.

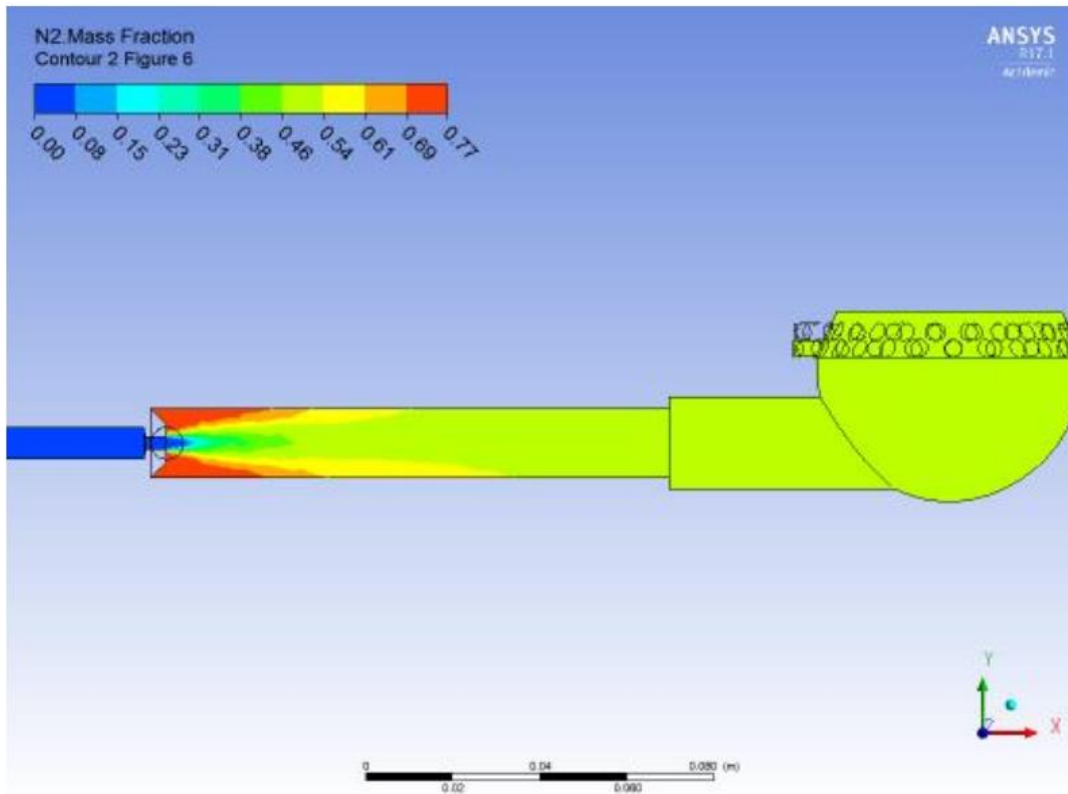


Figure 4.5: A visualization of the entrainment effect of air and extent of homogeneity in the mixture for design

An even better comprehension of the entrainment effect was attained by plotting the velocity vectors inside the three-dimensional rendering. According to the continuity equation, cross sectional area significantly decreases as gas enters the burner's injector orifice, increasing velocity. The cross-sectional area widens again after the gas leaves the injector orifice, lowering the gas velocity. Air is drawn into the burner throat by this change in velocity, which also produces a low-pressure zone and gradient between the interior of the burner and the surrounding environment.

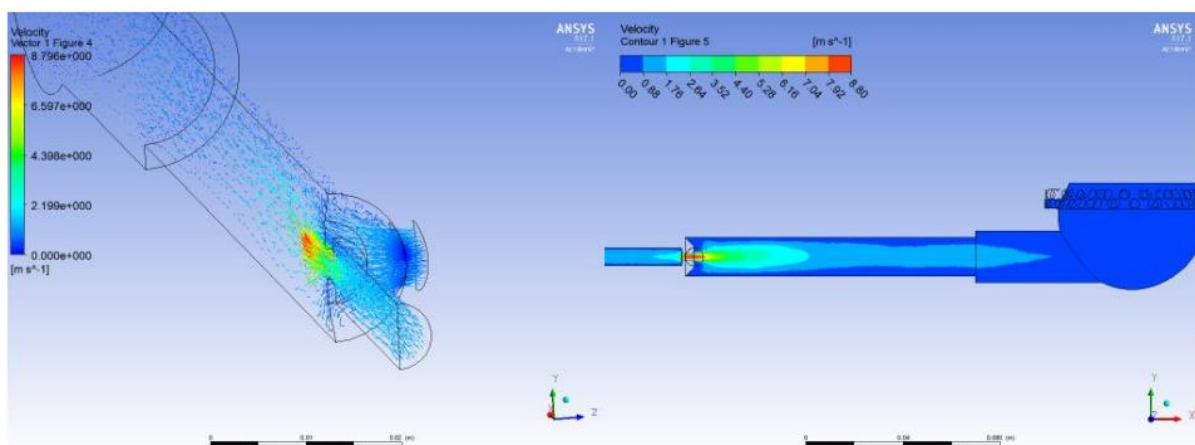


Figure 4.6: For a partially premixed flame, velocity contours and vectors showing the entrainment of ambient air into the burner throat

4.18.2 Visual Representations

The flame's high-temperature areas and the amount of the external volume's increased temperature were depicted visually by the simulation. Because the maximum temperature results stayed below Chemkin's calculated adiabatic flame temperature of 2100 K, they successfully passed one level of validation. This indicates that, as predicted, a variety of losses prevented the temperature at the stoichiometric mixture front in the simulation from rising to adiabatic levels. Only the design selection process used the simulation's temperature values.

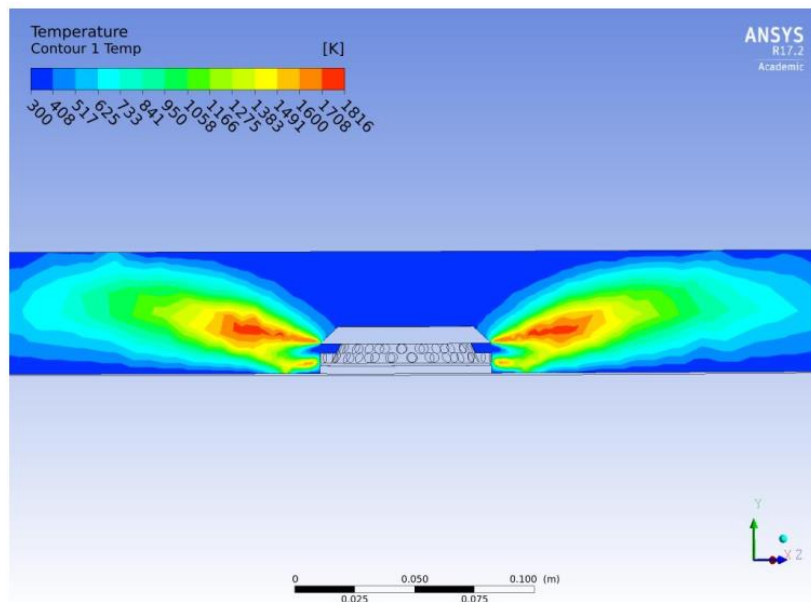


Figure 4.7: Temperature contour due to combustion within design

Moreover, the simulation generated informative graphics that described the velocity flow profiles in circular ports. Surface friction at the port walls, as predicted, decreased velocity and produced a parabolic profile, with the port's center seeing the highest flow velocity. Due to their larger aspect ratio, rectangular ports had a profile that was noticeably more homogeneous.

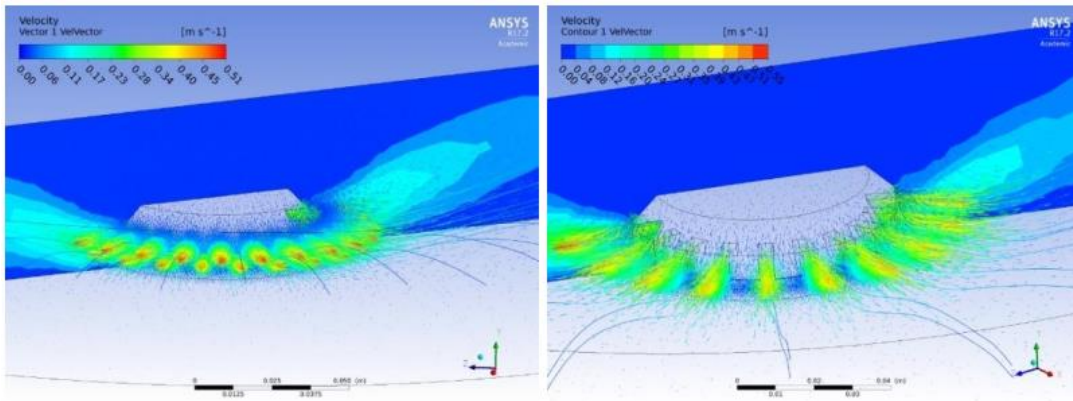


Figure 4.8: Design velocity contours displaying a parabolic state inside circular ports

CHAPTER FIVE

AN ANALYSIS OF THE HEAT TRANSFER AND STEAM ENERGY OF AN INJERA BAKING PAN

5.1 AN ANALYSIS OF THE ENERGY CONSUMPTION OF THE INJERA BAKING PAN (MITTAD)

The thermophysical characteristics of the Injera baking pan—such as heat capacity and thermal conductivity—are what primarily determine how much energy it uses. Reviews of the literature are where those properties are found. The amount of heat needed to bring the batter's temperature from room temperature to the boiling point of water and evaporate some of the water on the batter is known as the energy required for baking. The total energy required for baking and heating the pan is the sum of those two energy requirements (31).

5.1.1 Heat Required for Injera Baking

The heat used to bake Injera can be defined as the energy necessary to raise the batter to a particular temperature, and evaporate the amount of water that is observed 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 subtracting 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 batter from room temperature to the boiling point of water which is called sensible heat, plus the energy required to evaporate the water which is called latent heat. It is also assumed that the heat capacity of Injera batter is the same as water in order to calculate the energy required to raise the batter temperature to boiling point of water [17]. Therefore, the utilized heat energy is

$$Eu = m_{batter} C_{pwater} (T_{boil} - T_{room}) + (m_{batter} - m_{injera}) h_{vaporization} \quad (5.1)$$

Where; Eu=energy utilized; m_{batter} =mass of batter=400g; m_{injera} =mass of injera which is 320g
And C_{pwater} Specific heat capacity of water =4.187kJ/kg.k

This is due to heat capacity of batter is the same as that of water (batter contains 70% of water and 30% of cereals)

T_{boil} =boiling temperature of water at room condition which is 100°C

T_{room} =ambient temperature of Jimma which is 25°C

*h*_{vaporization}- Heat of vaporization of water $h_{fg}=2269.86 \text{ kJ/kg}$

$$E_u = 0.4 \text{ kg} * 4.187 \text{ kJ/kg} * (100 - 25) \text{ K} + (0.4 \text{ kg} - 0.32 \text{ kg}) 2269.86 \text{ kJ/kg}$$

$$E_u = 307.2 \text{ kJ}$$

By considering the energy loss in baking the required total energy can be calculated by assuming safety factor of 1.2. Thus the total energy required becomes 368.64 kJ. The time taken for cooking of Injera is about 3 minutes on average. The heat transfer rate (power) required for Injera baking can be calculated as:

$$\dot{Q} = \frac{E_{utilized}}{\Delta t}$$

$$\dot{Q} = \frac{368.64 \text{ kJ}}{2 \times 60 \text{ sec}} = 2.05 \text{ kW}$$

5.1.2 The Energy Needed to Preheat a Baking Pan

The energy needed to heat up the injera to the baking temperature on the pan's surface is known as heating up energy.

$$Q_h = m_{pan} C_{pan} (T_{baking} - T_{room}) \quad (5.2)$$

Where Q_h =heating up energy; m_{pan} =mass of pan (average mass of pan is 4.5 kg) [24]; C_{pan} =heat capacity of pan =0.88kJ/kg. K; T_{baking} =temperature of the baking surface of the pan 220°c

$$Q_h = 4.5 \text{ kg} * 0.88 \text{ kJ/kg} * (220 - 25) \text{ K}$$

$$Q_h = 772.2 \text{ kJ}$$

Total amount of energy=Energy utilized + Heating up energy

$$Q_T = E_u + Q_h \quad (5.3)$$

$$= 307.2 \text{ kJ} + 772.2 \text{ kJ}$$

$$= 1079.4 \text{ kJ}$$

But there is loss during baking as well as heating up process so taking 1.2 as factory of safety

$$\text{Total amount of energy} = 1079.4 \text{ kJ} * 1.2 = 1,294.8 \text{ kJ}$$

5.2 TRANSFER OF HEAT

Three directions are possible for heat loss from the baking pans /,mitadll/. These are the baking pans' bottom, lateral, and top directions, where energy disperses into the surroundings. Heat loss is thought to be zero because the lateral side and bottom are well insulated. All three of the basic heat transfer modes—conduction, radiation, and convention—as well as

different combinations of them, are used in ovens. Just a small portion of the heat is transmitted from the injera's bottom surface to its upper surface through the mechanisms of convection and radiation. The majority of the heat is generated by conduction.

I measure the well-functioning electric mitad to obtain the data. The temperature of the surrounding air is measured with a glass thermometer, and the surface temperatures of the clay pan and lid cover are measured with an infrared thermometer. The instrument is calibrated to obtain an accurate reading, and the aluminum lid cover and baking plate's emissivity are adjusted on the infrared thermometer to $\epsilon=0.11$ and 0.98 , respectively.

Seven injeras, including the time for initial heating and polishing, were baked during the thirty-minute experiment. In order to quantify the losses accounted for during the baking process, temperature measurements were made at each surface at intervals of three minutes. Ultimately, the average temperature of all measurements at each surface was taken.

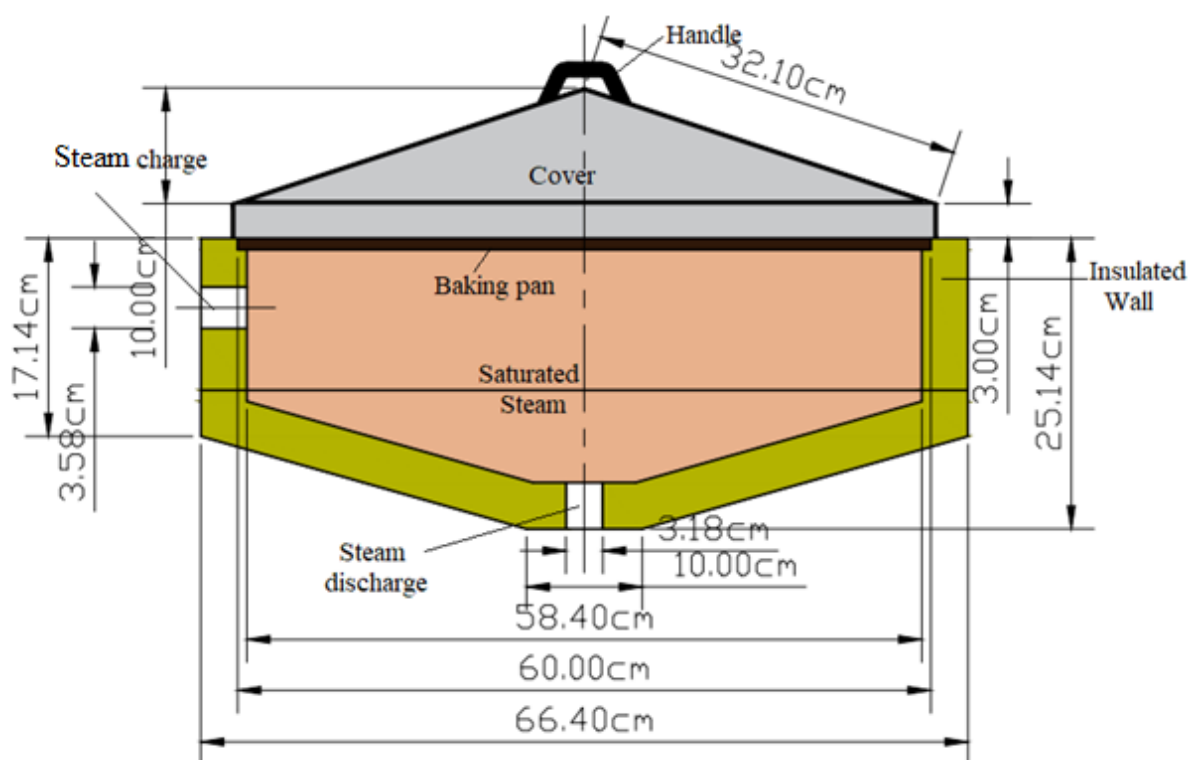


Figure 5.1: Schematic diagram of steam oven for injera baking

Assumptions

1. Steady-state conditions
2. The baking pan is large relative to its thickness
3. Thermal conditions on both sides of the plate are uniform
4. One-dimensional heat transfer by conduction through the baking pan.

5. Free convection
6. Radiation exchange between the baking pan surface and lid cover and also between lid cover to the surrounding
7. Liquid air is opaque to thermal radiation
8. Solar radiation is negligible
9. There is no heat generation in the plate
10. Ignore thermal contact resistance between two layers of stainless steel and clay.

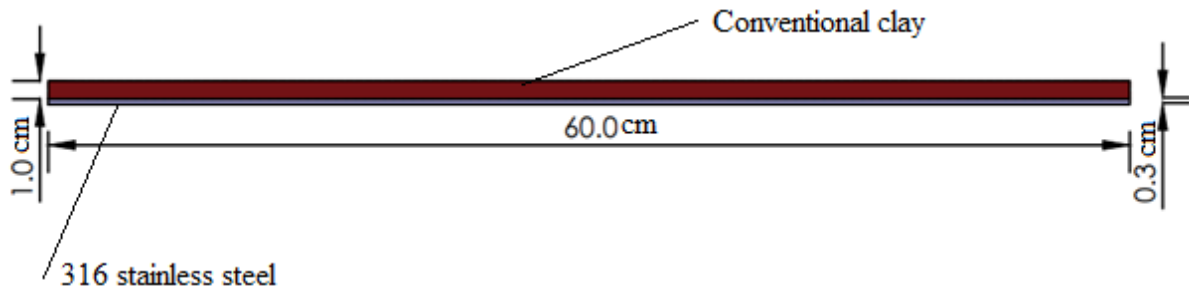


Figure 5.2: Thickness of baking pan

Thermal conductivity of conventional clay is composed of 316 stainless steel 1.3W/mK and 15.9W/mK respectively.

Thermal resistance of composite walls

$$R_{Total} = R_{ss} + R_{clay} = \frac{L_{ss}}{k_{ss}A_{ss}} + \frac{L_{clay}}{k_{clay}A_{clay}} = \frac{1}{A} \left(\frac{L_{ss}}{k_{ss}} + \frac{L_{clay}}{k_{clay}} \right) \quad (5.4)$$

Where: R_{total} total thermal resistance, R thermal resistance; L is the thickness of composite wall; A is surface area and k is thermal conductivity.

The subscript represents stainless steel and clay.

$$A = \frac{\pi}{4} D^2 = \frac{\pi}{4} (0.6m)^2 = 0.283m^2$$

$$R_{total} = \frac{1}{0.283m^2} \left(\frac{3 \times 10^{-3}m}{15.9 W/mK} + \frac{0.01m}{1.3 W/mK} \right) = 0.027848K$$

The overall heat transfer coefficient

$$U = \frac{1}{R_{Total}A} \quad (5.5)$$

$$U = \frac{1}{0.283m^2 \times 0.027848K/W} = 126.89 W/m^2K$$

The heat transfer rate is

$$\dot{Q} = UA\Delta T \quad (5.6)$$

$$\dot{Q} = 126.89 \frac{W}{m^2K} \times 0.283m^2 \times (220 - 25)K = 7002.42W = 7kW$$

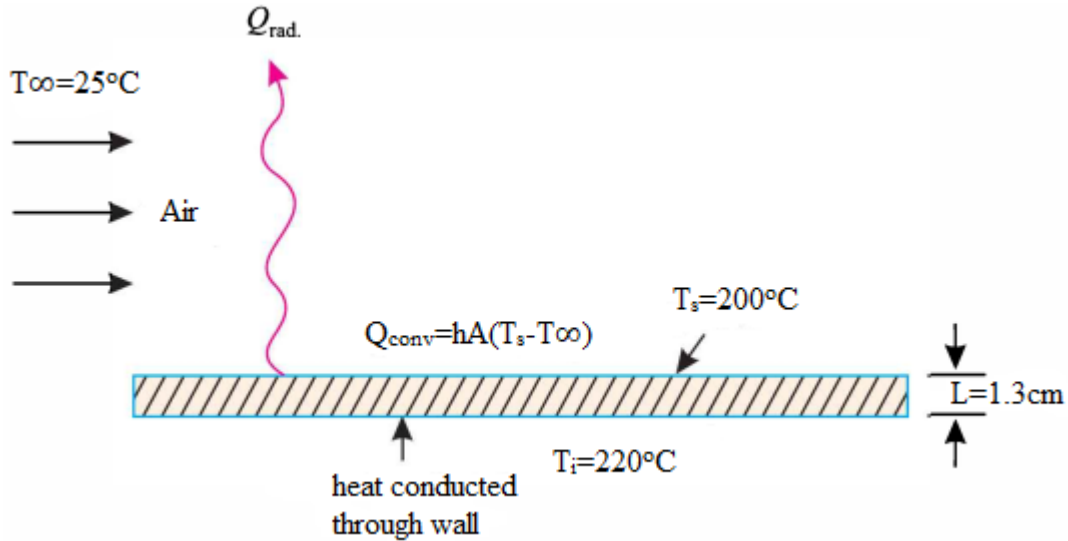


Figure 5.3 Combination of conduction, convection and radiation heat transfer

Surface energy balance

$$E_{in} - E_{out} = 0 \quad (5.7)$$

$$Q_{cond} - Q_{conv} - Q_{rad} = 0 \quad (5.8)$$

Convection: Newton's Law of Cooling

$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) \quad (5.9)$$

Radiation: Stefan-Boltzmann Law

The radiation effects, which we have overlooked thus far, can be substantial when the wall is surrounded by a gas and may need to be taken into account. The radiation heat transfer rate between an area and a surface with emissivity ϵ The surrounding surfaces at a certain average temperature T_{surr} can be expressed as at temperature T_s .

$$\dot{Q}_{rad} = \epsilon\sigma A_s(T_s^4 - T_{surr}^4) = h_{rad}A_s(T_s - T_{surr}) = \frac{T_s - T_{surr}}{R_{rad}} \quad (5.10)$$

$$\text{Where, } R_{rad} = \frac{1}{h_{rad}A_s}$$

Is the thermal resistance of a surface against radiation, or the radiation resistance, and

$$h_{rad} = \frac{\dot{Q}_{rad}}{A_s(T_s - T_{surr})} = \varepsilon\sigma(T_s^2 + T_{surr}^2) \quad (5.11)$$

Convection heat loss from baking pan

Data obtained from experiment

- Average temperature of bare plate $T_p = 473K$
- Surrounding temperature $T_\infty = 298K$
- Film temperature, $= \frac{T_p - T_\infty}{2} = \frac{473 + 298}{2} = 386K$

Properties of air at film temperature

$$\nu = 24.8728 \times 10^{-6} \text{m}^2/\text{s}; k = 32.736 \times 10^{-3} \text{W/mK}; Pr = 0.6928 \text{ and } \beta = 2.57 \times 10^{-3} \text{K}^{-1}$$

Plate Data

- Characteristic length of plate; $L_c = D = 60\text{cm} = 0.60\text{m}$
- Surface area; $A_s = \pi r^2 = 0.283\text{m}^2$
- Emissivity of plate; $\varepsilon = 0.98$
- Stefan- Boltzmann constant; $\sigma = 5.67 \times 10^{-8} \text{W/m}^2\text{K}^4$

Convection heat loss from flat plate can be defined as.

$$\dot{Q}_{conv} = hA_p(T_p - T_\infty)$$

Where, h = convection heat transfer coefficient, $\text{W/m}^2\text{K}$; A_p = heat transfer surface area of plate, m^2 ; T_p = temperature of the plate surface, K ; T_∞ = temperature of the air sufficiently far from the surface, K

Convective heat transfer coefficient over flat plate can be calculated by eqn.

$$h_c = \frac{Nu k}{L} \quad (5.12)$$

The recommended correlations for the average Nusselt number over upper surface of hot plate are

$$Nu = 0.54 Ra_L^{1/4} \dots\dots\dots (10^4 \leq Ra_L \leq 10^7), pr \geq 0.7 \quad (5.13)$$

$$Nu = 0.54 Ra_L^{1/3} \dots\dots\dots (10^7 \leq Ra_L \leq 10^{11}), all pr \quad (5.14)$$

Where; (Ra_L) is the Rayleigh number, which is the product of the Grashof (Gr_L) and Prandtl (Pr) numbers:

$$Ra_L = Gr_L Pr = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} Pr \quad (5.15)$$

Surface energy balance

$$E_{in} - E_{out} = 0$$

$$Q_{cond} - Q_{conv} - Q_{rad} = 0$$

Convection: Newton's Law of Cooling

$$\dot{Q}_{conv} = hA_s(T_s - T_\infty)$$

Radiation: Stefan-Boltzmann Law

The radiation effects, which we have not addressed thus far, can be substantial when the wall is encircled by a gas and may require consideration. The pace at which radiation heat transfers from an area to a surface with emissivity ϵ The surrounding surfaces at some average temperature T_{surr} and as at temperature T_s can be expressed as

$$\dot{Q}_{rad} = \epsilon\sigma A_s(T_s^4 - T_{surr}^4) = \frac{T_s - T_{surr}}{R_{rad}}$$

Where,

$$R_{rad} = \frac{1}{h_{rad}A_s}$$

Is the thermal resistance of a surface against radiation, or the radiation resistance, and

$$h_{rad} = \frac{\dot{Q}_{rad}}{A_s(T_s - T_{surr})} = \epsilon\sigma(T_s^2 + T_{surr}^2)$$

Convection heat loss from baking pan

Data obtained from experiment

- Average temperature of bare plate $T_p = 473K$
- Surrounding temperature $T_\infty = 298K$

$$\text{Film temperature, } T_f = \frac{T_p + T_\infty}{2} = \frac{473 + 298}{2} = 386K$$

Properties of air at film temperature

$$\nu = 24.8728 \times 10^{-6} \text{ m}^2/\text{s}; k = 32.736 \times 10^{-3} \text{ W/mK}; Pr = 0.6928 \text{ and } \beta = 2.57 \times 10^{-3} \text{ K}^{-1}$$

Plate Data

- Characteristic length of plate; $Lc = D = 60cm = 0.60m$
- Surface area; $A_s = \pi r^2 = 0.283\text{m}^2$
- Emissivity of plate; $\epsilon = 0.98$
- Stefan- Boltzmann constant; $\sigma = 5.67 \times 10^{-8} \text{ W/m}^2\text{K}^4$

Convection heat loss from flat plate can be defined as.

$$\dot{Q}_{conv} = hA_p(T_p - T_\infty)$$

Where, h = convection heat transfer coefficient, W/m^2K

A_p = heat transfer surface area of plate, m^2

T_p = temperature of the plate surface, K

T_∞ = temperature of the air sufficiently far from the surface, K

Convective heat transfer coefficient over flat plate can be calculated by eqn.

$$h_c = \frac{Nuk}{L}$$

The recommended correlations for the average Nusselt number over upper surface of hot plate are

$$Nu = 0.54Ra_L^{1/4} \quad (10^4 \leq Ra_L \leq 10^7), Pr \geq 0.7$$

$$Nu = 0.15Ra_L^{1/3} \quad (107 \leq Ra_L \leq 10^{11}), \text{ all } Pr$$

Where; (Ra_L) is the Rayleigh number, which is the product of the Grashof (Gr_L) and Prandtl (Pr) numbers:

$$Ra_L = Gr_L Pr = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} Pr$$

$$Ra_2 = \frac{9.8 \times (2.57 \times 10^{-3}) \times (200 - 25)(0.6)^3}{(25.12 \times 10^{-6})^2} = 1.5404 \times 10^9$$

Therefore; average Nusselt number is calculated as;

$$Nu = 0.15Ra_L^{1/3} = 0.15(1.5404 \times 10^9)^{1/3} = 173.24$$

$$h_c = \frac{Nuk}{L} = \frac{173.24 \times 32.736 \times 10^{-3}}{0.6} = 9.45$$

Therefore, at each baking (180sec) heat energy lost from the flat plate by convection heat transfer can be obtained as;

$$\dot{Q}_{conv} = hA_p(T_p - T_\infty)$$

$$= 9.45 \times 0.2827(473 - 298) = 467.52W \times 1.2 \text{ (Factor of safety)} = 561W$$

5.2.1 Heat Loss from Radiation on Bare Plate

At each baking of injera (180sec) heat energy lost from the bare plate by radiation heat loss is determined as;

$$\dot{Q}_{rad} = \varepsilon\sigma A_s(T_c^4 - T_{surr}^4)$$

$$\dot{Q}_{rad} = 0.98 \times 5.67 \times \frac{10^{-8}W}{m^2K^4} \times 0.2827m^2(473^4 - 298^4)K^4 = 662.40W \times 1.2$$

$$= 794.88W$$

According to study research, of the 180 seconds required for injera to bake thoroughly, lifting the cover was left open for 25% of the baking time (45 seconds) in order to remove the baked injera and pour the dough. Thus, if we assume that the lifting cover is open for 25% of the baking process, then eqn. define the total heat energy lost from the clay plate's top.

$$\begin{aligned}\dot{Q}_{tp} &= 0.25(\dot{Q}_{conv} + \dot{Q}_{rad}) \\ &= 0.25 \times (561 + 794.88\text{W}) = 338.98\text{W}\end{aligned}$$

5.2.2 Heat Distortion from the Cover

The lifting of the cover which remains closed for more than half of the baking cycle and becomes heated, resulted in heat loss from the top of the clay plate. Thus, radiation and convection transfer heat from the heated cover to the surrounding air.

5.2.3 Heat Loss by Convection through the Cover

In this case thermal energy is lost due to transfer of heat to the cold environment by the bulk movement of the air.

a. Data obtained from measuring

- Average temperature of cover $T_c = 356\text{K}$
- Surrounding temperature $T_\infty = 298\text{K}$
- Film temperature $T_f = \frac{T_p + T_\infty}{2} = \frac{356 + 298}{2} = 322\text{K}$

b. Properties of air at film temperature

$$v = 18.606 \times 10^{-6} \text{m}^2/\text{s}; k = 31.998 \times 10^{-3} \text{W/mK}; Pr = 0.6962 \text{ and } \beta = 3.05 \times 10^{-3} \text{K}^{-1}$$

c. Cover data

- Characteristic length of cover is $L_c = s = 33\text{cm} = 0.33\text{m}$
- Surface area $A_s = \pi r s + 2\pi r h = \pi r (s + 2h) = 0.546\text{m}^2$
- Emissivity of lifting cover $\varepsilon = 0.11$
- Stefan- Boltzmann constant $\sigma = 5.67 \times 10^{-8} \text{W/m}^2\text{K}^4$

The rate of convection heat loss from the cover is found to be proportionate to the temperature differential, and Newton's law of cooling provides a convenient expression for this relationship, as demonstrated in equation below

$$\dot{Q}_{conv} = hA_s(T_c - T_\infty)$$

Where, h = convection heat transfer coefficient, $\text{W/m}^2\text{K}$

A_c = cover heat transfer surface area, m^2

T_c = temperature of the cover surface, K

T_∞ = temperature of the air sufficiently far from the surface, K

Convective heat transfer coefficient over horizontal enclosure can be calculated by using eqn.

$$h_c = \frac{Nuk}{L}$$

Where, h_c = convection heat transfer coefficient, [W/m²K]; Nu = Nusselt number; k = thermal conductivity of the fluid (air) [W/mK] and L = characteristic length [m]

The empirical correlation for the average Nusselt number (Nu) in natural convection was shown in eqn.

$$Nu = C(Gr_L Pr)^n = CRa_L^n \quad (5.16)$$

The values of the constants C and n depend on the geometry of the surface and the flow regime, which is characterized by the range of the Rayleigh number.

Where; (Ra_L) is the Rayleigh number, which is the product of the Grashof (Gr_L) and Prandtl (Pr) numbers:

$$\begin{aligned} Ra_L = Gr_L Pr &= \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} Pr \\ &= \frac{9.81 \times (3.05 \times 10^{-3} K^{-1}) \times (83 - 25)K \times (0.33)^3}{(18.66 \times 10^{-6} m^2/s)^2} \times 0.6962 \\ Ra_L &= 3.779 \times 10^8 \end{aligned}$$

Hence, average Nusselt number for natural convection over the horizontal enclosure (cover) heated from below is determined from correlation proposed by Globe and Dropkin;

$$Nu = 0.069 Ra_L^{1/3} Pr^{0.074} \dots \dots \text{for } 3 \times 10^7 \leq Gr_L Pr \leq 7 \times 10^{11}$$

$$Nu = 48.57$$

$$h_c = \frac{Nuk}{L} = \frac{48.57 \times 31.998 \times 10^{-3}}{0.33} = 4.71 W/m^2 K$$

$$\dot{Q}_{conv} = hA_c(T_c - T_\infty)$$

5.2.4 Radiation Heat Loss from the Cover

The energy that is transferred by electromagnetic waves is known as thermal radiation. All matter with a temperature higher than absolute zero continuously emits it, and the amount increases as the body temperature rises.

From cover surface at temperature (T_s) to the surrounding (T_{sur}) will be given by Stefan-Boltzmann law.

$$\dot{Q}_{rad} = \varepsilon\sigma A_s(T_s^4 - T_{surr}^4) = 0.11 \times 5.67 \times \frac{10^{-8}W}{m^2K^4} \times 0.54m^2(356^4 - 298^4)K^4$$

$$= 27.536W$$

Where: ε = is emissivity of surface (Aluminum, $\varepsilon = 0.11$); σ = is Stefan - Boltzmann constant = $5.67 \times 10^{-8} W/m^2K^4$; A_s = is surface area of cover; T_c = is temperature of cover surface and T_{sur} = is surrounding temperature

According to study research, out of the 180 seconds required for injera to bake thoroughly, the cover was left open for 25% of the baking time (45 seconds) in order to remove the baked injera and pour the dough. The total rate of heat energy lost from the top of the cover is therefore defined as follows, taking into account that the cover can be closed for 75% of the baking process:

$$\dot{Q}_t = 0.75(\dot{Q}_{conv} + \dot{Q}_{rad})$$

$$\dot{Q}_{rad} = 0.75(89.145W + 27.536W) = 87.511W \times 1.2 = 105W$$

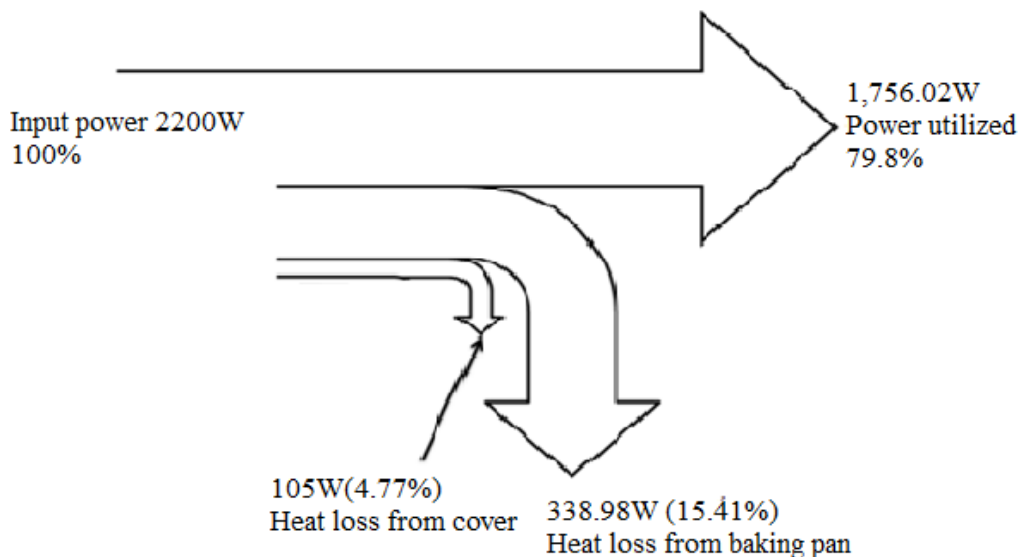


Figure 5.5: Result obtained from analytical analysis (a) Energy flow (Sankey) diagram of electric injera baking stove

5.3 STEAM USAGE ASSESSMENT

Important information to identify when estimating steam usage for a food process is given in table 5.1. Rough steam usage calculations are simple to make, when certain assumptions are

made and the English system of units is employed. Numbers obtained using rough estimates are valuable for sizing equipment, pipelines and estimating capacity. Key simplifying assumptions are:

1. Only the latent heat of vaporization of steam is transferred to the product (ignore sensible heat) during processing. The latent heat of vaporization of steam “ h_g ” is estimated as 1,000 BTU/ (pound of steam). This number can assumed to be constant (within 10 percent) for most steam sources (zero to 60 psig).
2. The specific heat of the product “ c_p ” is estimated to be 1.0 BTU/(pound·°F), or about the same as water.

Table 5.1: Information required computing steam usage in food processes

Variable or Unknown	Sym bol	Units	Explanation
Heat energy required	Q	kJ	Energy needed to accomplish the desired process
Mass of product to be heated	m_p	kg	Total amount of the product to be heated in a given amount of time
Specific heat of product to be heated	c_p	kJ/(kg.C)	Characteristic of product. Consult Rahman for information on specific heats of foods.
Temperature differential of product heated	T	°C	Initial product temperature subtracted from final product (heated) temperature
Latent heat of evaporation of steam	h_g	kJ/kg	Energy required to evaporate a given quantity of pure water at given conditions
Mass of steam required to heat product	m_s	kg	Amount of steam required to accomplish the desired heating process
Efficiency of heating process	eff	Decimal %	Accounts for heat losses to the environment

For more rigorous calculations of steam usage, see one of the many texts regarding this subject as applied to food processes.

Key equations for heat energy and steam usage calculations are:

$$Q = (W_p \cdot c_p \Delta T) / \text{eff}$$

And

$$Q=h_g \cdot W_s$$

5.4 TRANSFER OF HEAT AND INSULATION

Conduction and convection mode heat transfer are the two basic processes that heat transfer during baking when using bake ware. The interaction of molecules and atoms of non-moving liquid or solid materials results in heat transfer through conduction. Convectonal heat transfer also requires a solid and moving fluid with a temperature differential. Since the temperature reached during cooking is still fairly moderate, the radiation mode of heat transfer process is typically disregarded. Between the top surface of the bake pan, the food (such as Injera), and the outside surface of the bake pan, heat convection occurs. The largest resistance element in a heat loss system is usually insulation.

Table 5.2: Properties and selection of thermal insulation materials

Characteristics				
No.	Types of insulation	Density (kg/m ³)	Thermal conductivity (W/m.K)	Maximum working temperature (°C)
1	Glass fiber wool	11 to 45	0.032 to 0.044	260 °C
2	Rock wool	30 to 200	0.035 to 0.039	750 °C
3	Calcium Silicate	220	0.06	650 °C
4	Cera wool Blanket	64 to 128	0.11	1260 °C

Oven insulation typically consists of the materials listed in Table 5.2. Nonetheless, glass fiber wool is chosen for this application due to its accessibility and suitable operating temperature. The standard material of choice for thermal insulation in appliances like ovens, range cookers, and dual fuel cooking tops is glass fiber. In electric, gas, and dual fuel models, it is primarily used to stop heat loss through conduction and prevent heat transfer to the nearby work surfaces. When cooking in less demanding conditions, fiber glass performs better and offers respectable levels of safety and energy efficiency. This performance is more than sufficient for cooking at standard temperatures between 149 and 260 degrees Celsius.

5.4.1 Determining the critical thickness of insulation

The thermal conductivity of an insulation material's thickness divided by its convective coefficient of heat transfer yields the critical insulation thickness. The following formulas are used to calculate the effective insulation thickness.

$$T_1 = \frac{k}{h} \quad (5.17)$$

$$T_2 = \frac{1-k}{h} \quad (5.18)$$

Where, T_1 is critical side Insulation thickness, T_2 is critical bottom Insulation thickness, k is thermal conductivity coefficient of fiber glass insulation ($k = 0.043 \text{ W/m.K}$), and h is convective coefficient of heat transfer ($\text{W/m}^2\text{K}$), $h = 5.7 + 3.8v$, where v is wind speed around (2 m/s), $h = 5.7 + 3.8 * 2 = 13.3 \text{ W/m}^2\text{K}$.

The critical insulation thicknesses T_1 and T_2 are obtained as follows using Eqs. (2) and (3):

$$T_1 = \frac{0.043 \text{ W / mK}}{13.3 \text{ W / m}^2\text{K}} = 0.32 \text{ cm}$$

$$T_2 = \frac{1 - 0.043 \text{ W / mK}}{13.3 \text{ W / m}^2\text{K}} = 7.2 \text{ cm}$$

The critical insulation thickness values are calculated to be 0.0032 meters in the side direction and 0.072 meters in the bottom direction. Consequently, an analysis is conducted on the heat transfer on an electric Injera baking mitad with critical insulation thicknesses. The ideal insulation thickness dimension from the fundamental directions of heat loss is shown in the figure. The following relation can be used to calculate heat transfer from the bottom direction.

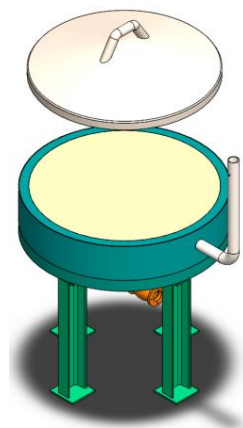


Figure 5.7: Isometric views of CAD model injera baking oven

5.5 ENERGY ESTIMATION OF OVENS

The boiler is designed for a 5 ovens. These ovens are similar in size and take 2 minutes to bake 1 minute for rotation or idle time. In 5 pans in 1 hour ($(60/3 = 20) * 5 = 100$ injeras) for 6 hours 600injeras can bake. The power required for 1 injera is 2.2 kWh, which is required for 100 injeras power required ($600*2.2\text{kW}=1320\text{kWh}$). We multiply it by 1.5 to get an idea of the energy wasted when baking. $1320*1.5=1980$ kW

If 600 injeras are baked in 1 day, it will be 18000 injeras in 30 days. The power consumption in 1 month is $1980\text{kW}*30=59,400$ kWh per month and 712800 kWh per year. When converted to Birr, 712800 kWh per year* 2.4=1,710,720 Birr.

Although inflation is difficult to predict, the initial price of the boiler is relatively expensive, but it is possible to save 1,710,720 Birr.

5.6 DESCRIPTION OF THE SYSTEM

The block diagram of the steam powered injera baking system is shown in Fig.6.1. The system consists of: boiler and the injera baking pan ('mitad'). They are integrated. The output steam temperature which exit from superheater is 230°C . The steam is transported through well insulated steam pipe to charge the each oven. Each oven has its control valve and safety valve.

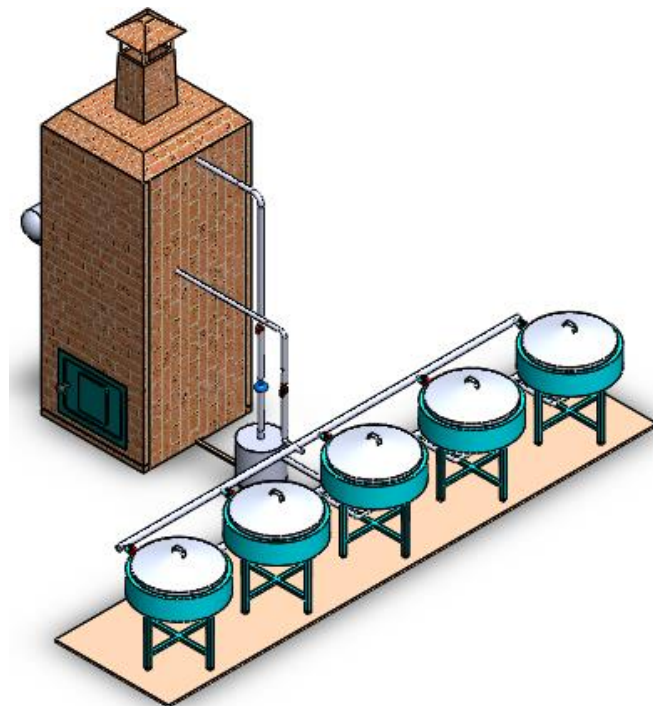


Figure 5.8: The assembled oven design with solidworks

5.7 INJERA PRODUCTION

Several basic biochemical, chemical, and physical processes are involved in the making of injera, including water evaporation, volume expansion, starch gelatinization, protein denaturation, crust formation, carbon dioxide production, porous structure formation, and browning reactions. Product quality, which encompasses food safety, produce consistency, taste, texture, appearance, and shelf life, is first and foremost non-negotiable. In addition, reducing manufacturing process interruptions to prevent waste and minimizing production time are critical components in maintaining low financial costs. Baking uses energy, which is becoming a bigger concern.

Injera is produced in five key stages:

1. Formation of dough: mixing and binding of raw ingredients and shaping the dough pieces.
2. Proving: supplying the dough with heat and humidity to encourage the yeast to ferment and the dough to rise.
3. Baking: heating the dough at high temperature to evaporate moisture and convert fragile dough to stable injera.
4. Cooling: lowering the temperature of the injera to ambient.
5. Packaging and distribution: the final preparations are made before the injera is delivered to the customer.

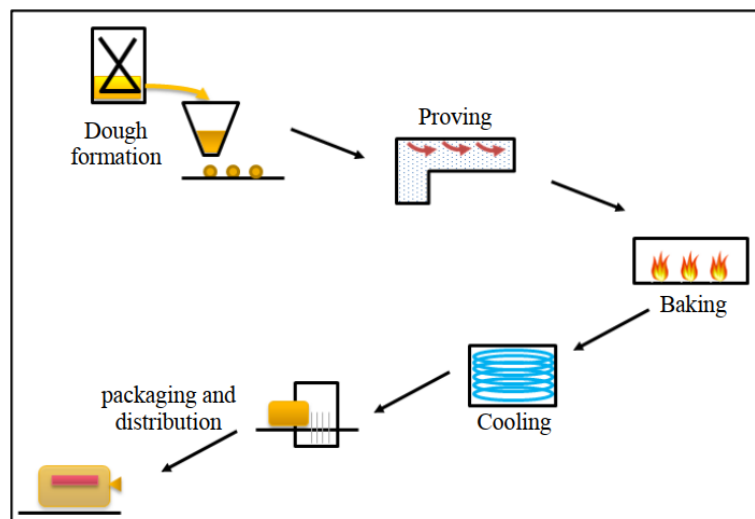


Figure 5.9: Schematic diagram of the injera baking process (25)

5.8 THERMAL CONDUCTIVITY OF INJERA AND BATTER DURING BAKING

A clearer understanding of the thermal diffusion through these substances over time can be obtained by graphically representing the thermal conductivity. Imagine a graph where the y-axis represents temperature and the x-axis represents time. The thermal conductivity of batter and injera could look like this:

5.8.1 Injera

At the beginning of baking, the temperature of the injera batter rapidly raises as heat is quickly conducted through its thin structure and high moisture content. As baking continues, the temperature curve might gradually increase at a slower rate as moisture evaporates from the surface cooling it down slightly. Towards the end of baking, the temperature might rise again as remaining moisture evaporates and the injera reaches its desired level of doneness.

5.8.2 Batter

Because of its thicker consistency and possibly lower moisture content, the batter initially warms up more slowly than injera. The temperature rises gradually during baking because heat transfers through the batter more slowly than it does with injera. The temperature curve may rise steadily until the batter is baked through, depending on the density and moisture content of the batter.

5.9 FEHT END PART TRANSFER OF HEAT VIA BAKING PAN

In many engineering applications, temperature distributions and contours are analyzed using FEHT (Finite Element Heat Transfer) software. Usually, the first step in using FEHT software to analyze temperature contours is to create a finite element mesh of the desired geometry. After the simulation is set up, you run the analysis and the program determines the distribution of temperatures across the domain by asking you to specify boundary conditions, material properties, and any heat sources or sinks that are present in the system. Once the analysis is finished, you can see the findings, along with temperature contours, to see how heat moves through the system.

Typically, you'll look for gradients that show heat flow as well as areas of high and low temperature when interpreting the temperature contours. For the purpose of improving designs, forecasting thermal performance, and guaranteeing system reliability, this information can be very important.

Item Count

29 nodes
0 lumps
55 line segments
27 triangular elements
This problem was solved with a bandwidth of 3

Specified Node Potentials

Node	X [m]	Y [m]	Spec. T [K]
1	7.990E-02	9.948E-02	473
2	5.995E-01	9.948E-02	473
6	1.201E-01	9.948E-02	473
10	2.000E-01	9.948E-02	473
12	2.408E-01	9.948E-02	473
14	2.805E-01	9.948E-02	473
16	3.207E-01	9.948E-02	473
18	3.604E-01	9.948E-02	473
20	4.001E-01	9.948E-02	473
22	4.403E-01	9.948E-02	473
24	4.805E-01	9.948E-02	473
26	5.207E-01	9.948E-02	473
28	5.599E-01	9.948E-02	473
29	1.598E-01	9.948E-02	473

Material Data

Outline	Material	Area [m ²]	Type	Generation [W/m ³]
1	Clay	1.072E-02	Distributed	0.0

Material properties**Clay**

Conductivity [W/m-K]: 1.300
Density [kg/m³]: 1460
Specific Heat [J/kg-K]: 880.0

Specified heat-flux boundaries

Edge	I	J	Flux [W/m ²]
1	1	4	0
2	2	3	0

Convection boundaries

Edge	I	J	Conv. Coef. [W/m ² -K]	Tinf [K]
1	8	27	17	293
2	3	8	17	293
3	25	27	17	293
4	23	25	17	293
5	21	23	17	293
6	19	21	17	293
7	17	19	17	293
8	15	17	17	293
9	13	15	17	293
10	11	13	17	293
11	9	11	17	293
12	7	9	17	293
13	5	7	17	293
14	4	5	17	293

The left and right sides of baking pan is insulated the heat is directed form bottom surface upper surface. It has 29 nodes.

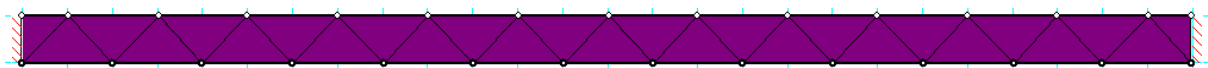


Figure 5.10: Discretized or mesh geometry

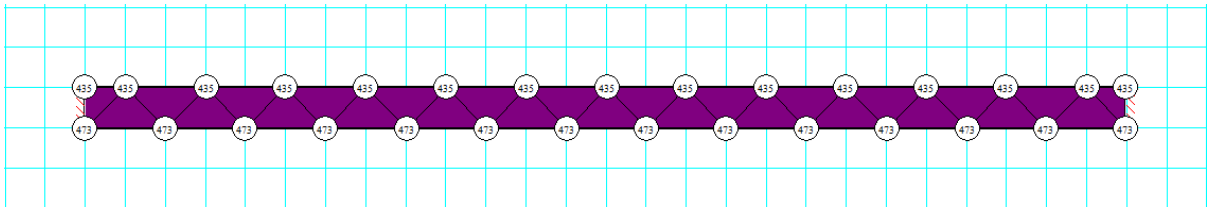


Figure 5.11: Temperature at different nodes

As shown the figure 5.10 below the red color show high temperature and the blue color shows low temperature because it exposed for environment.

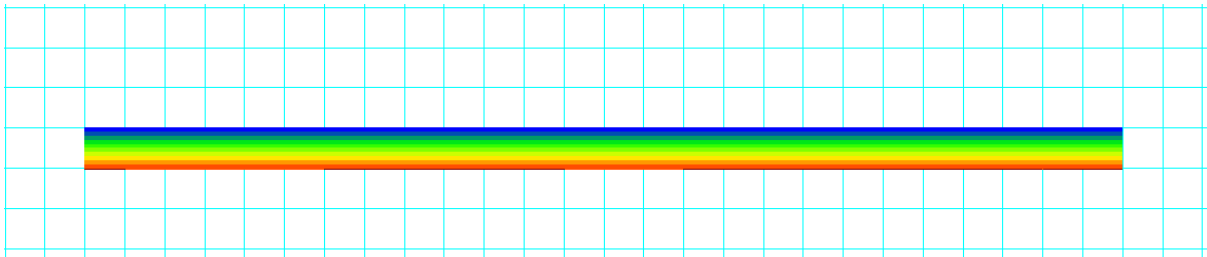


Figure 5.12: Temperature contour

CHAPTER SIX

CONCLUSIONS AND RECOMMENDATIONS

6.1 CONCLUSIONS

In conclusion, the steam energy injera baking oven showed significant promise as a modern and efficient alternative to traditional injera baking methods. Its performance, energy efficiency, and user-friendly features make it a valuable tool for both household and commercial applications. It is applicable in rural and urban areas. With ongoing refinement and optimization, steam energy injera baking ovens have the potential to contribute to the preservation of injera-making traditions while meeting the needs of modern consumers and producers.

The main design parameters of watertube boilers includes (types of boiler: gas fired; quantity of steam (dry) generated: 8TPH; steam pressure (gauge)/temp: 304.6 psi / 260°C; quantity of gas consumed: 1.8 TPH; feed water temperature: 90°C; GCV of biogas: 21.5 MJ/kg; Enthalpy of steam at 142.23 psi pressure: 665 kCal/kg (saturated); enthalpy of feed water: 85 kCal/kg and boiler efficiency 75%.

The boiler is designed for a 5 ovens. These ovens are similar in size and take 2 minutes to bake 1minute for rotation or idle time. In 5 pans in 1 hour ($(60/3 = 20) * 5 = 100$ injeras) for 6 hours 600injeras can bake. The power required for 1 injera is 2.2 kWh, which is required for 100 injeras power required ($600*2.2kW=1320kWh$). We multiply it by 1.5 to get an idea of the energy wasted when baking, $1320*1.5=1980$ kW

If 600 injeras are baked in 1 day, it will be 18000 injeras in 30 days. The power consumption in 1 month is $1980kW*30=59,400$ kWh per month and 712800 kWh per year. When converted to Birr, 712800 kWh per year* 2.4=1,710,720 Birr.

Although inflation is difficult to predict, the initial price of the boiler is relatively expensive, but it is possible to save 1,710,720 Birr.

Under this study analytical calculation and CAD modeling of burner, boiler components, baking ovens. And also arrangement or position of ovens well proposed.

This result and discussion section provides an overview of the findings from the hypothetical study on a steam energy injera baking oven, highlighting its performance, efficiency, user experience, limitations, and future research directions.

In conclusion, the steam energy injera baking oven presents a promising solution for more efficient and sustainable injera production. By harnessing steam energy, this oven reduces fuel consumption, minimizes environmental impact and enhances the overall baking process, with further development and implementation; it has the potential to revolutionize traditional baking methods, promoting both economic and ecological benefits for communities reliant on this staple food.

6.2 RECOMMENDATIONS

This injera baking machine is specially recommended massive production of injera in small scale industries or high community institutions such as military, schools, hotels and etc. this machine can minimize the energy biomass improves economy and introduce the cheap steam energy source alternatively.

This machine is highly recommended to replace other energy sources especially the wood fuels which are used to back injera so that we can reduce deforestation, dissertation and waste of electric energy. So, the researcher should be focus on cheap as well as free energy use for injera baking.

This research was focused on analytical study some part of CFD simulation and CAD modeling generally it will use grass root or basement for other works. In future research works, heat transfer simulation and well design specification and energy baking oven in order to increase thermal efficiency. In addition thermal conductivity improvement on the clay of mitad is required to minimize the startup heat load and saving of steam energy.

Many different adaptations, tests, and experiments have been left for the future due to lack of time and budget (i.e. the experiments with real data are usually very time consuming, requiring even days to finish a single run). Future work concerns deeper analysis of particular mechanisms, new proposals to try different methods, or simply well design of oven. In this study, validation of result predicted by experimentally is not carried out. So, anyone who is initiate and expert to this studying area will be validating the result predicted by tests and experimentally.

In this paper economic analysis of the system is not made detail due to shortage of time, however, it is better to make economic analysis of the system.

Boiler, pump, blower and burner etc. are available in the market with their specification so the researches focus on validation, optimization, and experimental testing.

Controlling system is not done yet so the researcher should focused on how to control the whole baking system such as steam generation mechanisms, baking capacity, energy efficiency, temperature and pressure etc.

Prototyping and Testing: While CAD modeling and simulation provide valuable insights, physical prototyping and testing are essential for validating the design and verifying its performance in real-world conditions. Use the CAD model to guide the prototyping process and compare experimental results with simulation predictions.

Documentation and Collaboration: Document the CAD model thoroughly, including design specifications, simulation setups, assumptions, and results. Foster collaboration between multidisciplinary teams involved in the design process, such as mechanical engineers, thermal engineers, and food technologists, to leverage diverse expertise and perspectives.

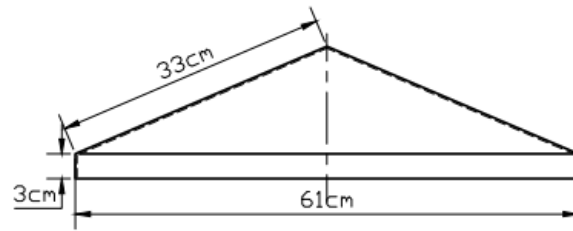
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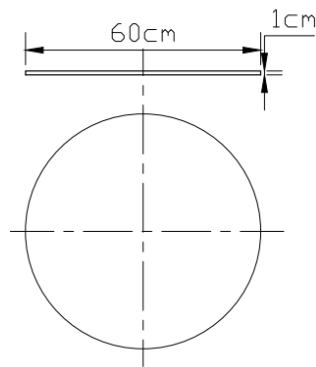
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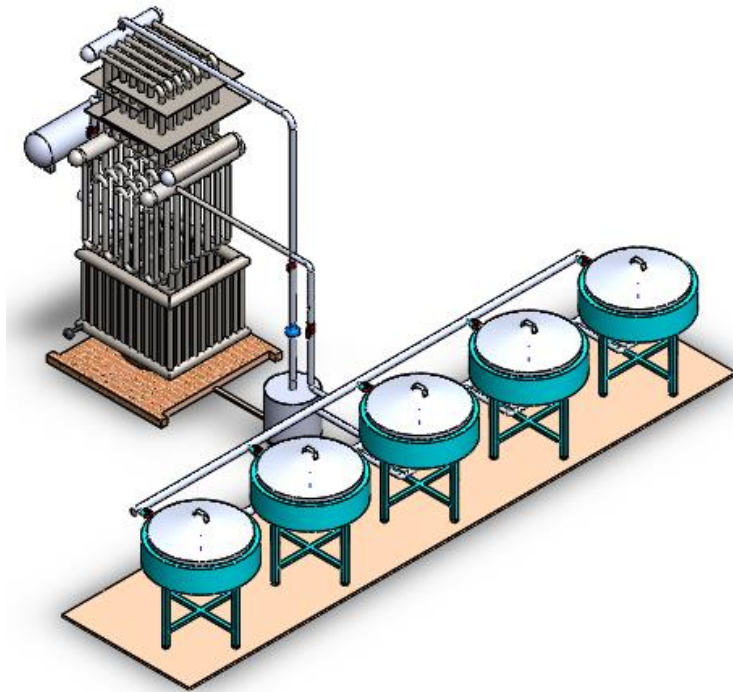
APPENDIX A



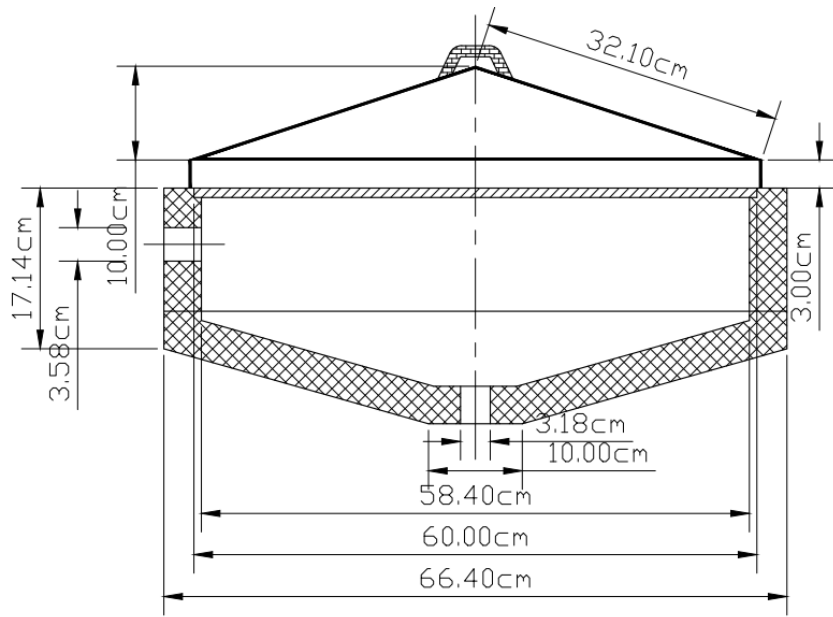
Apex1: Cover of baking pan



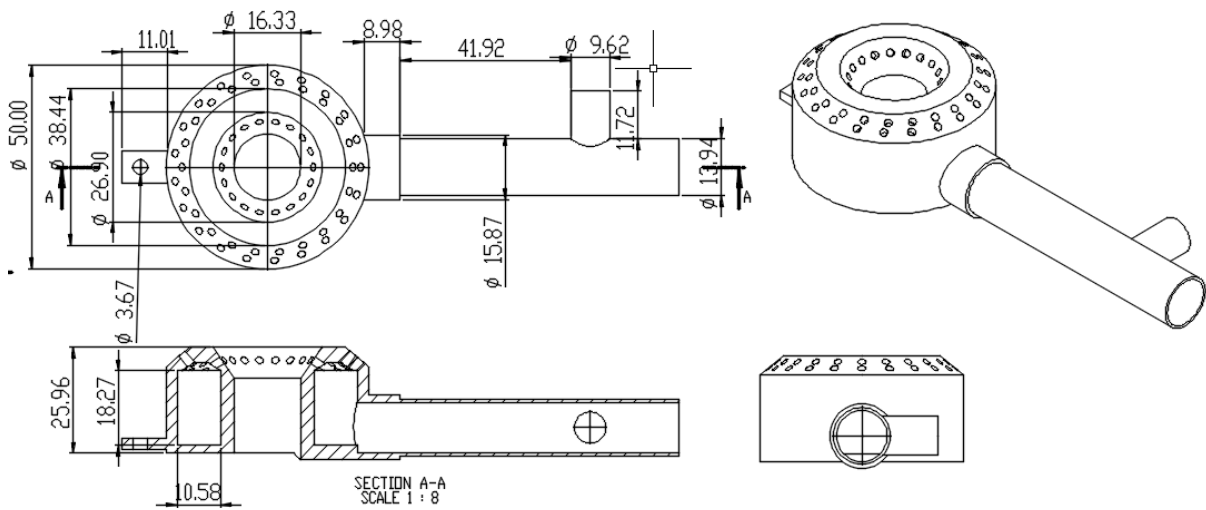
Appex2: Baking pan



Appex3. Expose view of baking pan assembly



Appex8. Drawing of oven



Appex9. Drawing of biogas burner