DESIGN AND PRODUCTION OF A BIOGAS STOVE BURNER FOR HOUSEHOLD USE

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UNIVERSITY OF BENIN

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A PROJECT SUBMITTED TO THE DEPARTMENT OF MECHANICAL ENGINEERING, FACULTY OF ENGINEERING, UNIVERSITY OF BENIN, BENIN CITY.

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CERTIFICATION

This is to certify that the project titled “Design and Production of a Biogas Stove Burner for Household Use” was carried out by EREKU ORITSETIMEYIN THOMAS, HABU NUVALGA ELISHA, IDUWE MARVELLOUS and OKOBIA EMMANUEL CHUKS with matriculation no ENG174292, ENG174297, ENG174299 and ENG1704325 respectively of the Department of Mechanical Engineering, Faculty of Engineering, University of Benin, Benin City.

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DEDICATION

We dedicate this work to God almighty for his unfailing love and to our parents for their never-ending support in this academic journey.

ACKNOWLEDGEMENT

Our gratitude goes to God Almighty for His protection and strength granted to us up till this day

We are indebted to our parents and families for their endless support throughout our academic years and our lives in general without which we would not be here today.

Sincere appreciation to the management and administration of the University of Benin, the Faculty of Engineering and our Department of Mechanical Engineering who deemed us worthy of undertaking this academic pursuit.

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We also recognize our course mates who we began this academic journey with, who have in more than one way motivated, shared knowledge and cheered us on.

Our appreciation also to many people who have contributed and mean a lot to us, whose names do not appear, we are grateful for all they did to make this successful.

Thank you all once more.

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ABSTRACT

Increasing demand for wood and fossil fuel which have limited availability has, over the years, contributed majorly in environmental pollution. The availability of energy for cooking remains a major concern in developing countries and cooking is a daily household activity. The negative environmental effect of wood and fossil fuel necessitates inquest for an alternative energy source that is sustainable. Biogas, over the years of research has shown favorable characteristics which make it an excellent option as an alternative fuel source. However, more research has to be made into designing and developing devices or appliances that utilize the biogas efficiently. This study details the design and fabrication of a biogas stove for domestic use with rural communities of developing countries such as Nigeria in mind focusing on characteristics such as efficiency, simplicity and cost-friendliness of the design. The biogas stove consists of the following major components: burner head, mixing tube, the injector burner support, etc. The Bernoulli's theorem was used to derive the flow rate of gas as well as key design dimensions to maintain this flow rate. The biogas stove was fabricated using stainless steel for the burner head, mild steel for the mixing chamber and a brass alloy for the injector component. The material selected were chosen based on considerations given to corrosion, local availability and then cost. The clearance between the cooking pot and the burner head is 45mm, while the clearance between the flame ports is 5mm. The injector is connected to the mixing chamber which tapers down to the throat diameter of 14.7mm which is maintained as the diameter of the mixing chamber. The mixing chamber is connected to the burner head which is a cylindrical component with a top having 32 burner ports each of 3mm diameter drilled into it, from which the gas can be ignited. The result of three water boiling tests places the heating efficiency of the stove at 58.51%

CHAPTER 1

1.1 BACKGROUND TO STUDY

The increasing demand for energy and the need to reduce greenhouse gas emissions globally have led to the exploration of renewable energy sources as the world becomes more conscious of the environmental impact of traditional energy sources such as fossil fuels. Furthermore, the energy problem in rural areas or communities in many developing nations like Nigeria has seen impressive change in the last three decades, leaving millions of people lacking enough energy inputs to sustain economic development (Stout and Best, 2001). Fossil energy, which is the main energy stay of Nigeria is estimated to be declining, a trend that will intensify after the year 2000 (Pimentel et al., 1998).

Biogas, a renewable energy source, is produced by *anaerobic digestion of* organic *matter* such as agricultural waste, food waste, and sewage sludge. Biogas has the potential to replace conventional fossil fuels for electricity and heat production as biogas is a sustainable alternative to traditional fossil fuels and can significantly reduce greenhouse gas emissions as well as the potential to offset energy costs incurred by households for cooking, which is currently dominated by biomass, Liquefied Petroleum Gas (LPG) and kerosene. Health and climate costs of biomass emissions can also be abated through the adoption and use of biogas. Adoption rates for decentralized biodigesters, however, are low in emerging markets (< 50%). Studies have indicated that this low technology adoption is mainly due to low ambient temperature (< 10^{o} C, resulting in reduced gas yield), feedstock issues, installation cost and complexity, low quality of gas production (methane content lower than 40 mole %), a lack of finance options and insufficient choices and low efficiencies of downstream use appliances.

Although biogas generation has been utilized since the 1950"s, and the principles of digestion are well documented, comparatively little has been done as regards designing appliances that are fueled by the burning of such gases. In most cases, burners are developed using a "trial-and-error" process, rather than consulting a text, or applying a formula. A gas burner generally is a device to generate a flame to heat up products using a gaseous fuel such as acetylene, natural gas or propane. Some burners have an air inlet to mix the fuel gas with air to make a complete combustion (Fulford, 1996). The main influencing factors in using biogas as a combustible gas are gas/air mixing rate, flame speed, ignition temperature and gas pressure.

Biogas contains methane, carbon dioxide, and impurities such as hydrogen sulfide and moisture. Biogas has a low calorific value compared to other fossil fuels, making it challenging to achieve high combustion efficiency. Hence, the utilization of biogas as a fuel source requires the development of *efficient burners* that can convert the biogas into useful energy with minimal environmental impact.

Biogas production technology has led to the growth of a number of biogas appliances for lighting, cooking, heating, incubating and electricity generation. The most commonly used appliance for cooking purposes in both households and institutions is the biogas stove. However, some households are using biogas lamps for lighting their homes.

The design of a biogas burner involves several considerations, including the characteristics of biogas, combustion efficiency, and safety.

The development of efficient biogas burners is critical for the widespread adoption of biogas as a fuel source. The aim of this research is to design a biogas burner that can efficiently convert biogas into useful energy while minimizing environmental impact and ensuring safety.

1.2 PROBLEM STATEMENT

The use of biogas as a fuel source requires the development of efficient burners that can convert biogas into useful energy with minimal environmental impact. The efficiency of biogas burners is crucial for the optimal utilization of biogas energy. However, the combustion of biogas poses several challenges due to the variable composition of the gas, including high levels of methane and impurities such as hydrogen sulfide and moisture. These impurities can affect the combustion process and reduce the efficiency of biogas burners. Therefore, it is essential to develop and optimize biogas burners to ensure safe and efficient combustion of biogas.

The project is therefore originated to design and manufacture an efficient biogas burner

1.3 AIM AND OBJECTIVES

1.3.1 AIM

The aim of this research is to design and construct a biogas burner that can efficiently convert biogas into useful energy while minimizing environmental impact and ensuring safety.

1.3.2 OBJECTIVES

The objective of the product includes:

1. To carry out mechanical studies on biogas
2. To prepare design specifications for the biogas burner.
3. To analyze the combustion characteristics of biogas and its impurities.
4. To develop conceptual design of the biogas burner.
5. To select one concept based on design specsism.
6. To carry out detailed design of the biogas burner.
7. To prepare manufacturing specifications of the burner.
8. To manufacture the biogas burner
9. To list the biogas burner constructed for performance.

1.4 SCOPE OF WORK

The scope of this research includes the design, development, and evaluation of a biogas burner. This research will focus on the development of a biogas burner designed to utilize the local biogas produced from anaerobic digestion of organic matter for small- scale applications, such as household heating and cooking.

The research will focus on the following areas:

1. **Analysis of Biogas Characteristics**: The study will analyze the properties of biogas, including calorific value, composition, combustion characteristics of biogas, flame stability, emissions and investigate the effect of impurities such as hydrogen sulfide and moisture on the combustion process which constitute challenges associated with the combustion of biogas.
2. **Biogas Burner Design**: The study will develop a design for a biogas burner that seeks to improve on the limitations of existing designs and ensures efficient combustion and safety. The design will be based on a thorough understanding of the characteristics of biogas and the requirements of different applications.

3. **Manufacturing and Testing**: Design, fabrication and testing of a biogas burner that can efficiently and safely combust biogas. The performance of the developed biogas burner will be evaluated in terms of efficiency and emissions.

CHAPTER 2

LITERATURE REVIEW

Biogas systems can be used in homes to supply energy for lighting and cooking and to make fertilizer. The ability to produce biogas depends on the availability of enough biomass energy, water, and digester space. Anaerobic digestion of organic material takes place in anaerobic environments to produce biogas. Methane and carbon dioxide make up the majority of biogas's composition.

Good combustion and heating efficiency of the biogas burner are the main focus of this study; hence a close observation of key factors is studied and inferences are drawn as presented.

2.1 BIOGAS COMBUSTION

Biogas contains 50 -75% methane and air contain 21% oxygen, for this study we will use 58% as the value of methane in biogas; thus, it is necessary and sufficient that the gas burns completely to convert methane to carbon dioxide and minimize the unburnt methane. Biogas burns in air (oxygen) to release energy with by-products of carbon dioxide and water, with the chemical reaction given as:

$$CH_4 + 2O_2 \longrightarrow CO_2 + 2H_2O$$

Volume:	1	2	1	2
% volume:	58	21		
Volume / %volume:	1/58	2/21		
	1.724	9.524		
Gas ratio:	1	5.52		

Hence, 1 volume of biogas requires 5.52 volumes of air.

The stochiometric requirement for biogas in air is given as

$$\frac{1}{1+5.52} = 15.33\%$$

A burner cannot be designed and then developed without considerations to: Air/fuel ratio, modulation range, reaction chemistry influence, temperature effect. (Bethold et al, 2011).

Biogas will burn over a range mixture range of 9% to 17% biogas in air (Fulford, 1960).

Thus, this study will give consideration to the burner being designed to operate with a small volume of excess air to ensure that this range is maintained and the flame is not too rich i.e having excess fuel, as this will lead to incomplete combustion and deposits of soot on the utensil being heated.

Air is supplied to the gas, mixed and then the mixture is ignited – this is known as pre-aeration. It is required that this primary air be greater or equal to 50% of the total air requirement for the combustion.

2.2 FLAME

Flame is the visible, hot result of the combustion process. Combustion occurs when a fuel reacts with oxygen in the air, producing heat, light, and various chemical by-products. A flame consists of several zones, each with distinct characteristics, namely:

- Ignition Zone: This is where the fuel and oxygen first combine and ignite and is usually the hottest part of the flame.
- Combustion Zone: Here, the fuel continues to react with oxygen, releasing energy in the form of heat and light. This is where most of the chemical reactions take place.
- Oxidation Zone: In this region, any remaining unburned fuel reacts with oxygen to ensure complete combustion. The flame may appear less bright here.

The color of a flame may vary depending on the type of fuel and temperature. A blue flame is often associated with complete combustion and high temperatures, while a yellow or orange flame may indicate the presence of impurities in the fuel.

It is crucial to control and manage combustion carefully to minimize pollution and ensure safety, hence flames can be classified into three on this criterion:

- Rich Flame: A rich flame occurs when there is an excess of fuel compared to the available oxygen. In this condition, the combustion process is incomplete because there is not enough oxygen for the fuel to react with. Rich flames typically produce a yellow or orange color due to the presence of unburned carbon particles (soot) emitting light, generate more carbon monoxide (CO) and are less efficient in terms of energy output.

- Optimal Flame (Stoichiometric Flame): An optimal flame is the ideal combustion condition where there is just enough oxygen to completely burn all the fuel. It results in the cleanest and most efficient combustion. The flame is typically blue in color, indicating a higher temperature and complete combustion. The ratio of fuel to oxygen is known as the stoichiometric ratio, which varies depending on the specific type of fuel.
- Poor Flame (Lean Flame): A poor flame occurs when there is an excess of oxygen compared to the available fuel. In this condition, the combustion process is also incomplete because there is not enough fuel to react with the excess oxygen. Poor flames tend to burn at higher temperatures and emit a blue or even colorless flame and are efficient in terms of fuel utilization but can produce more nitrogen oxides (NOx) due to the high combustion temperatures.

Maintaining the right air-to-fuel ratio is crucial in the biogas stoves, an optimal flame is desired for efficient cooking and minimal emissions. Controlling the combustion process ensures not only efficient energy use but also reduces environmental impacts by minimizing pollutants released into the atmosphere.
From works reviewed it is advisable to design for a flame that is between ideal and lean.

2.3 GAS FLOW

The pressure of the biogas is the driving force for the stochiometric mixing of the biogas and air in the burner. Assuming incompressible flow, Bernoulli's theorem which states that "*in a steady flow of an ideal fluid, the total energy per unit mass of the fluid remains constant along a streamline*".
Total energy can be expanded into three components – Kinetic, Potential and Pressure energy.

Mathematically:

$$P + \frac{1}{2}\rho v^2 + \rho g h = \text{constant}$$

For an ideal gas flow, the kinetic energy due to the velocity plus the potential energy due to the pressure of the flow is constant

Re-writing equation gives

$$\frac{P}{\rho} + \frac{v^2}{2g} + h = \text{constant}$$

where: p is the gas pressure (Nm^{-2}),

g is the acceleration due to gravity (9.81 ms^{-2}) and

ρ is the gas density (kgm^{-3}),

v is the gas velocity (ms^{-1}),

h is head (m). For a gas, head (h) can be ignored.

From compressible flow theory, flow through a nozzle of area, *A,* is given by:

$$\dot{m} = C_d\,\rho_0\,A\sqrt{2\left(\frac{\gamma}{\gamma-1}\right)\frac{p_0}{\rho_0}r^{2/\gamma}\left(1-r^{(\gamma-1)/\gamma}\right)}$$

where: p_o and r_0 are the pressure and density of the gas upstream of the nozzle

$r = p_1/p_0$

p_1 is the pressure downstream of the nozzle.

NOTE: A high value of *r* means that the gas can burn with the primary air with the flames being short in length as primary aeration is a decisive factor of flame height. Full aeration, that is *r* = 5.52 is avoided however to prevent "flash back" – flame passing back through the ports into the mixing tube to burn at the jet (Tumwesige et al 2014).

2.4 INJECTOR AND ORIFICE DESIGN

A biogas burner is composed majorly of four parts: the nozzle, air regulation components, injector and the burner manifold (Tumwesige et al 2014).

The Injector component controls the amount of gas used by a burner. The size of the hole drilled at the tip of the injector is a key design factor. The injector is secured onto the end of the gas line fitting, so that it is replaceable. The injector also serves a purpose of separating the burner from the gas supply and thus a properly designed injector inhibits flame from entering the gas supply pipe – Flashback (Fulford, 1996).

The flow rate (*Q*) of the gas is related to the gas velocity (*v*) by the area (*A*) of the pipe through which the gas flows and is given by:

Q = v A

A phenomenon known as *Vena contracta* is observed in the flow of the gas, wherein the diameter or area of the gas flow after it has passed through the orifice (a hole in a plate) is comparatively smaller as compared to the orifice diameter. A visual description of this is shown

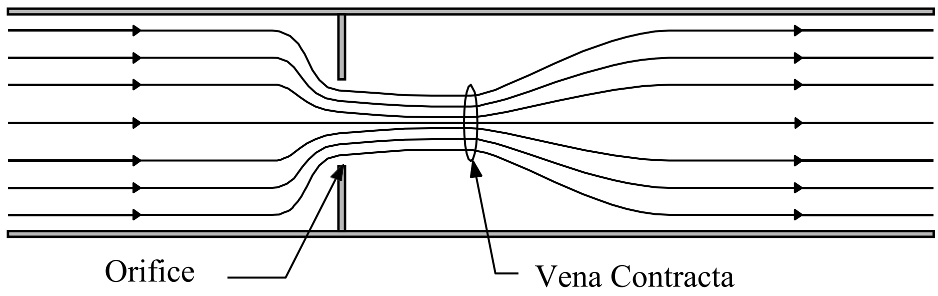


Figure 2.1 Flow Behavior Through Orifice

The flow of gas from the jet depends on the hole size and the gas pressure. From Bernoulli's theorem, the flow rate can be defined as

$$Q = 0.036\, C_d\, d^2 \sqrt{\frac{p}{s}}$$

where: Q = gas flow rate ($m^3\,h^{-1}$)

A_0 = area of orifice (mm^2)

p = gas pressure before orifice (mbar)

s = specific gravity of gas

C_d = coefficient of discharge for the orifice.

The vena contractor and friction losses through the orifice give rise to a factor known as the coefficient of discharge and usually has a value between 0.85 and 0.95.

The injector and orifice are designed as shown

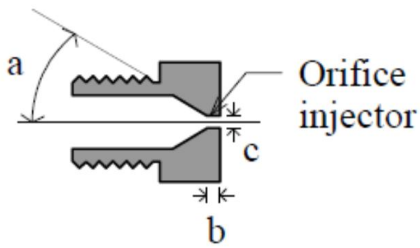


Figure 2.2 Injector Design

For a maximum value of the coefficient of discharge, C_d, the angle of approach, *a,* before the orifice should be 30° and the length of the orifice channel, *b,* should be a value between 1.5 - 2 times the orifice diameter, *c*.

2.5 ENTRAINMENT

After the gas passes through the injector orifice, it enters another channel called the mixing tube and approaches a region called the "throat" which has larger diameter than the injector, so the velocity of the gas stream is reduced. The design of the throat is cylindrical, and thus must be long enough to allow good mixing of the gas and air.

A length of $10d_t$ is usually recommended (Fulford, 1996).

The velocity, v_o , of the gas leaving the injector orifice is given by:

$$v_0 = \frac{Q}{3.6 \times 10^{-3}\ A_0} \ \mathrm{m\ s^{-1}}$$

And the reduced velocity observed at the throat:

$$v_t = v_0 \frac{A_0}{A_t} = v_0 \frac{d_0^2}{d_t^2}$$

Neglecting the vena contracta and friction effect, the gas pressure just after the nozzle becomes:

$$p_t = p_0 - \rho \frac{v_0^2}{2g}\left[1 - \left(\frac{d_0}{d_t}\right)^4\right]$$

The value of p_o is around atmospheric pressure, as the throat is open to the air, thus a pressure differential is set up and this sucks in primary air in through the air inlet ports to mix with the gas in the mixing tube.

Primary aeration depends on the "entrainment ratio" (r), which is determined by the area of the throat and the injector:

$$r = \sqrt{s}\left(\sqrt{\frac{A_t}{A_0}} - 1\right) = \sqrt{s}\left(\frac{d_t}{d_0} - 1\right)$$

The above relation is the Prigg's formula and holds true if the total flame port area (A_p) is between 1.5 and 2.2 times the area of the throat.

Where, A_t and d_t are the area and diameter of the throat

A_0 and d_0 are the area and diameter of the injector.

2.6 THROAT SIZE

The flow rate of the mixture in the throat (Q_m) is then given by:

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

SI unit for Q_m is m^3s^{-1} and Q is m^3hr^{-1}

The pressure drop due to the flow of the mixture down the mixing tube can be estimated by calculating the Reynolds number for the flow

$$\mathrm{Re} = \frac{\rho d_t v_t}{\mu} = \frac{\rho d_t}{\mu}\frac{4Q_m}{\pi d_t^2} = \frac{4\rho}{\pi\mu}\frac{Q_m}{d_t}$$

where ρ and μ are the density and viscosity for the mixture

The pressure drop (Δp) is then given by:

$$\Delta p = \frac{f}{2}\rho v_t^2 \frac{L_m}{d_t} = \frac{f}{2}\rho \frac{16Q_m^2}{\pi^2 d_t^5} L_m$$

where $f = \frac{64}{Re}$, when Re < 2000 and $f = \frac{0.316}{Re^{1/4}}$ when Re > 2000

2.7 BURNER PORTS DESIGN

In the design of biogas burners, considerations are made to address possible problems that may arise when burning the mixture, these are

2.7.1 Flashback

This is when the flame at a burner port, rather than staying at the burner tip, travels back down the mixing tube to the injector. This may lead to combustion within the mixing chamber and may arise as a result of improper air-fuel mixture or other factors that disrupt the stable combustion of the gas-air mixture.

To prevent flashback, a burner ports in thin metal are drilled to 2.5 mm diameter for natural gas. For thicker metal, larger port diameters can be used.

Biogas has a relatively low flame speed; thus, flashback does not often occur and burner port diameter of 5mm can be used in 5 mm thick metal.

2.7.2 Flame Lift

This is an opposite to flashback and occurs when the flame is lifted away from the burner tip. This is a major design consideration in biogas burner as it is undesirable since it results in unstable combustion as well as potential safety hazards. Factors that can be responsible for this are high gas velocity, the entrainment ratio being altered by adjustments in the primary air control, change in environment conditions such as strong wind, etc.

In the design, consideration is given to properly set the speed of the mixture passing through the burner ports. This is to ensure that the speed of the mixture leaving the burner port is not higher than the speed of the flame burning in the gas. Biogas has a stochiometric flame speed of about 0.25 ms^{-1}, the total flame port area is designed to ensure that the mixture velocity through the ports is much lower than 0.25 ms^{-1}.

An increase in the supply pressure will increase the mixture flow rate and velocity, also causing flame lift.

Given that,

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

based on the given conditions, the mixture supply velocity (v_p) can be calculated as

$$v_p = \frac{Q_m}{A_p} << 0.25\,\mathrm{m\ s^{-1}}$$

where the port Area, $A_{p,}$ can be expressed as

$$n_p \frac{\pi d_p^2}{4}$$

n_p is the number of ports

d_p is diameter of each port in meters.

The size and positioning of the individual burner ports are influenced by various factors, such as the heat pattern – circular or star; Cross-lighting and the need for adequate supply of secondary air. The heat output from the burner ports is limited to a design value to address flame lift. In practice, the heat output is set below 900 Wcm^{-2} (0.09 Wm^{-2}) of burner port area (Fulford, 1996).

2.7.3 Heat Pattern

Domestic stoves, used mainly for cooking, usually have burner ports arranged in a circular pattern, as most cooking pots have a circular base. The size of the circle depends on the average size of the cooking pots to be used.

2.7.4 Cross-Lighting

Cross-lighting is a method used to ignite multiple ports using flame from the initial ignition to light adjacent ports in a sequence to ensure all ports light simultaneously and safely. This is more efficient as well as convenient as compared to lighting
each port individually.

The burner is usually lit at one place, so the flames should "jump" from one burner port to the next, so the whole burner is alight.

Cross-lighting serves a dual purpose of reigniting the flames at individual burner ports when they go out. For biogas burner, the gaps between burner ports should be around 5 mm to ensure cross-lighting occurs.

2.7.5 Secondary Air Supply

Secondary air required to complete the stochiometric air requirement should, with ease, be able to reach each port. The pattern by which the ports are arranged ensures that this can happen without interference.

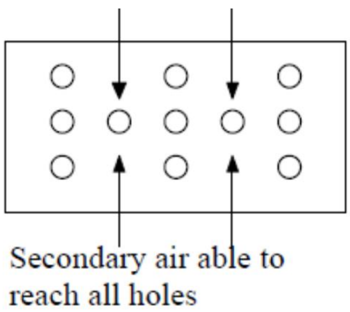


Figure 2.3 Pattern of Flame Ports

2.7.6 Flame Stabilization

The supply of secondary air to the flame can be increased by designing the burner manifold to have the burner ports set in a raised ledge or at an angle to the horizontal. This also mitigates the probability of flame lift to a good extent.

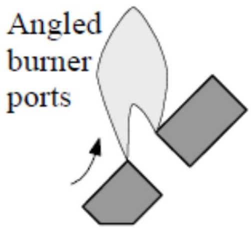


Figure 2.4 Flame Stabilization Technique

2.7.7 Pot Supports

For good heating and combustion efficiency to be obtained, the correct position of the object to be heated (e.g., a pot of food to be cooked) above the flame is estimated. If the object is too close to the flame, the flame is quenched and the result is an incomplete combustion and reduced efficiency of the stove. Likewise, the object being too far away from the flame results in heat being lost to the atmosphere and the stove is rendered less efficient.

The object being heated should be placed at the distance just above the tip of the visible part of the flame, intersecting the outer mantle of the flame as this ensures a clean and efficient combustion process.

In practice, the position of the object to be heated is designed after a prototype burner has been made and the flame length for typical conditions has been estimated. The design of the pot support height for domestic stoves lies within an average range of 25 to 30 mm for 5 mm burner ports, using biogas at 10 mbar pressure (Fulford, 1996).

2.8 REVIEW OF PREVIOUS WORKS

The burners and ovens used with biogas are comparable to those used with traditional appliances that use commercial gas fuels. The pressure generated by the biogas plant and the diameter of the entrance pipe both affect how much gas is consumed by a biogas stove's single or double burner, which is often equipped with one or two different gas consumption rates.

Tumwesige (Tumwesige et al 2014) investigated eight stove designs, closely observing the components that make up the biogas burners. The jet - a precisely sized hole that determines the burner's power output, regulates the amount of gas that comes into the burner. The results from this study showed that gas burners that were tested were shown to be of poor quality: they had low efficiencies and were shown to be made to designs that did not follow gas burner theory adequately. Recommendations made were that if local designs of a higher quality are able to meet well defined design standards and also be affordable biogas technology has potential positive impact on the livelihood rural communities.

Bajet, conducted a series of experiments to modify stove burner for biogas (Bajet, jr. et al, 2012), making biogas technology versatile and suitable for household use. The aim of this research was to provide different LPG customers with an alternative fuel source as Liquefied Petroleum Gas (LPG) is relatively expensive. Additionally, this addresses the issue presented by the farmer's biogas producer that just a portion of the cooking burner can burn and provide heat, with the majority of the emitted heat being absorbed by heating the excess methane. Biogas producers employed LPG burners; however, LPG's composition differs from that of biogas; as a result, inadequate flame was created when LPG burners were used with biogas. The technique also aims to demonstrate an atmospheric biogas stove burner that uses hog waste and has good combustion.

This research investigated data from two LPG cooking burner that had been modified to utilize biogas. The modification is made up of four parts: the head, which is similar to an LPG cooking burner in appearance, the injector, the air regulating sections, and the nozzle. Five trials were conducted to evaluate the burner's performance, focusing on the pressure, color, and length of the flame. Three levels of blue flame were produced by the modified

LPG stove burner. Since no flu gas is released by the burner, the risks to human health and the environment are reduced.

In the performance evaluation of a biogas stove for cooking carried out by Itodo (Itodo et al,) at the Teaching and Research Farm, University of Agriculture, Makurdi, Nigeria, a $3m^3$ continuous flow Indian style biogas plant was used to design, build, and assess the performance of a biogas stove. Cattle dung was used as the fuel source for the biogas plant, with a feedstock ratio of 1 part dung to 2 parts water, a retention period of 30 days, and a daily loading rate of 100 kg of slurry. The effectiveness of the stove was assessed by boiling water, cooking rice and beans; and measuring the time required to complete particular activities using a stopwatch. The working pressure of the plant, as measured by a manometer positioned between the stove and the plant, was used to calculate the amount of biogas required for boiling and cooking. According to the data:
5.13g of rice and 2.55g of beans were cooked in one minute while 0.14*l* of water boiled in one minute. Boiling water, cooking rice and cooking beans each required 0.69 m^3/min, 2.81 m^3/min, and 4.87 m^3/min of biogas, respectively. The stove was 20%, 56%, and 53% efficient at boiling water, cooking rice, and cooking beans, respectively.

Maged Kiriakos (Kiriakos, 2023) conducted a study on biogas combustion characteristics in a concentric flow slot burner to explore the stability and combustion properties of natural gas with various percentages of carbon dioxide from 0 to 40% emulating biogas fuel and use unique combustion technology. In this work, the stability properties and temperature profiles of turbulent planar flames at various levels of mixture inhomogeneity were examined. A newly invented concentric flow slot burner (CFSB) was used to produce the flames for various combinations of natural gas that contained 0, 10, 20, 30, and 40% CO2. For mixes of natural gas and CO2 and for varying levels of mixture inhomogeneity, measurements of the flame temperature and stability characteristics were developed. The mixture equivalence ratio, the Reynolds number, the degree of mixture inhomogeneity, the air-to-fuel velocity ratio, and the air-to-fuel velocity ratio are the key variables in the investigation. Results obtained showed that the flames exhibit the greatest stability for mixture inhomogeneity between completely premixed and non-premixed cases at a partial premixing ratio of 5 or higher. In conclusion, the study established the benefit of stabilizing biogas for use in actual combustion systems by utilizing turbulent planar flames with an inhomogeneous mixture. Therefore, biogas can effectively replace fossil fuels as a source of sustainable energy.

Yadav and Paul (Yadav & Paul, 2022) in their research, attempted to switch the LPG cook burner over to using biogas as the fuel for cooking. It was noted that the tendency of the flame lifting phenomena increases with an increase in burner port loading. The burner port loading was decreased up to an ideal value of 9.8 kJ/mm^{3}hr in order to balance the burning velocity and fuel velocity. The cook stove's maximum thermal efficiency was determined to be 67.53% at a power rating of 1.6 kW, which corresponds to a fuel injector size of 2.5 mm and a burner port size of 5 mm. The thermal efficiency was determined to be 5-9% higher than that of other biogas cook burners on the market.

Petro (Petro et al, 2020) made a study into the theoretical and experimental analysis of a novel domestic gas burner which was centered on developing and enhancing the performance of locally produced burners in order to obtain an even fuel flow in the mixing chamber, which will lead to an even distribution of heat throughout the cooking pot. Before fabrication, the burner was optimized using computational fluid dynamics (CFD) by adjusting the size of the jet, the number of flame portholes, and the number of air openings. A fuel distributor was included to boost the burner's efficiency. Results indicated that the manifold's and jet's optimal hole diameters were 100 mm and 2.5 mm, respectively. The currently developed biogas burner and the other two locally produced burners completed testing and comparison. The Centre for Agricultural Mechanization and Rural Technology (CARMATEC) and SIMGAS' thermal efficiencies were 54.61% and 58.82%, respectively, while the created burner's thermal efficiency was 67.01% according to the results of the water boiling test (WBT) on these three burners. Additionally, the newly produced burner consumed 736 g/*l* of fuel as opposed to 920 g/*l* for CARMARTEC and 833 g/*l* for SIMGAS.

Dàniel Füzesi and Viktor Jozsa, (Fuzesi and Jozsa 2019) conducted a numerical analysis of biogas combustion in a lean premixed swirl burner utilizing Computational Fluid Dynamics (CFD) of a 30 kW turbulent laboratory test burner. Four biogases were examined and were represented as mixtures of CH4, CO2, and H2 in varying proportions, with natural gas serving as the reference fuel. The combustion of natural gas among the fuels indicated flashback as a result of the bluff body located in the center of the mixing tube input.

Nevertheless, the problem's scope was limited, and adding some central purge air to the actual burner was sufficient to fix this problem. All of the flame patterns were V and W, which indicates the ideal loading of the combustion chamber. Natural gas combustion displayed the highest velocity and temperature in the flow field, despite the fact that the overall mass flow rates at the entrance rise as the heating value of the fuel decreases. The combustion chamber was modeled as a cylindrical body with a 150 mm diameter based on measurements and numerical simulations. As a result, the examined burner was able to run on fuels with low calorific value without requiring design changes or subjecting the combustion chamber to an elevated local thermal load. The results obtained from the different fuels were consistently of a high quality, the discrepancies were essentially insignificant, and the real-world application might include further standards.

As a conclusion, the analysis showed that the studied biofuels can be utilized successfully in the designed test equipment. In order to enable the current combustion appliances to function on fuels with low calorific values, hydrogen dilution in a modest quantity is an essential fuel component.

Decker (Thomas decker et al, 2018) used a study of mixed computational and experimental design strategy for improving the flame port geometry for biogas-fired domestic burner. The study used a multi-component simulation that combines three-dimensional computer-aided design (CAD) designs with computer-simulated chemical kinetics and computational fluid dynamics. Using a widely accessible biogas burner, the simulated flame port designs featured a variety of circular and rectangular shapes. To verify model results and compare against a reference port geometry, the three best-performing prototypes were constructed and put through an experimental test. Each of the three designs tested in the experiment appeared to have a higher thermal efficiency than the baseline. A layout of four-millimeter circular apertures led to an increase in thermal efficiency of 7.17%, from 53.0% to 56.8% on average. The findings suggested that port geometry design should include hydraulic diameter, velocity, and mixture density in order to increase a biogas burner's thermal efficiency. On the other hand, it was discovered that the model's estimates for emissions were incorrect and inconsistent with data from lab tests.

Bagaya (Bagaya et al, 2023) proposed a theoretical cylindrical design. The design's performance was examined in comparison to a conical design. On the basis of a steady-state thermal network, theoretical modeling of the cook stove was done. The model was solved using the MATLAB R2021B platform, which was used with permission (License No. 595687). The theoretical analysis's conclusion predicted a theoretical efficiency of 65%, a pot air temperature Tf of 220 ^{0}C, and a flame temperature Ta of 900 ^{0}C. Similarly, a validation with Kaushik's model for thermal efficiency and Sagouong's model for combustion chamber temperature. The maximal threshold (RMSE) between the two investigations was found to be 4%. The comparison revealed that the fire temperatures of the cylindrical shape are greater than those of the conical-shaped cook stove because the Tc temperature stagnates rapidly within 5 minutes at 600 ^{o}C. As a result, more testing and flue gas analysis can enhance the performance of the cylindrical cooking stove.

Huang (Xiaomei Huang et al, 2020) conducted research on a biogas-fired, partially premixed burner with a 25kW heat input that can be used for space heating and water heater tank heating was developed for gas-fired, wall- mounted boilers. According to the computation, the reference gas was supposed to be composed of 60% methane (CH4) and 40% carbon dioxide; and the burner port area was calculated to be 4744.5 mm2. The effectiveness of this burner was further examined through trials. The primary air ratio, and flame stability, jet diameters were varied to study their impacts on the burner's heat input. The outcomes demonstrate that the primary air ratio met the design requirement and the burner performed best when the jet diameter was 2.0 mm. Further research was done on a gas-fired wall-mounted boiler equipped with this burner to see how different biogas compositions affected exhaust emissions and thermal efficiency. When CH4 in the biogas fluctuated from 40% to 60%, the results showed that the concentration of CO in the flue gas met the safety standard. However, when the percentage of CH4 rises, the thermal efficiency of the wall-mounted biogas boiler significantly declines.

Oreko and Otancha (Oreko and Otancha, 2022) developed of Biogas stove from Aluminum alloy scraps. A single aluminum burner biogas stove was designed, manufactured and its efficiency was assessed using biogas produced by a mobile 200l biogas digester plant at the Federal University of Petroleum Resources in Effurun, Nigeria. The co-digestion of cattle dung and chicken droppings as feed stock in the ratio 1:2 and water substrate in the ratio 1:0.5 was used to operate the biogas digester plant. The system ran with a 30-day retention period. The sand-casting process was used to create the biogas burner made of aluminum alloy. The biogas burner stove underwent a thermal examination; from the analysis and evaluation of the burner's performance, the burner stove efficiency was estimated to be 50%, the burner port average flame transmission time was 3.10 seconds. Additionally, a biogas flow rate of 0.24 m^3 per minute on average boiled 1 liter of water in roughly 8.5 minutes.

Orhorhoro, Joel and Abubakar (Orhorhoro, Joel & Abubakar, 2018) carried out a study to design an improved biogas burner. The stove comprised of a combustion chamber natural convection is used to propel the fuel to be burned. The stove comprised of components such as the mixing tube, injector, the burner head and the burner supports; all constructed from medium carbon steel based on considerations given to financial factors and the availability of the material. The flame port diameter, clearance between flame ports, throat diameter, mixing tube length were 2.5mm, 5mm, 15mm and 80mm respectively with a clearance of 40mm between utensil and the burner head. The efficiencies of this design were 46.02%, 56.9% and 63.87% as the minimum, average and optimal recorded efficiencies respectively. Based on the result from the study, adoption of the stove was recommended.

CHAPTER 3
MATERIALS AND METHODS

3.1 DESIGN SPECIFICATIONS AND PARAMETERS

For this study, the determination following parameters and component dimensions were considered key to bringing out ensure proper fit of parts during assembly. These are:

- Composition of the gas produced
- Properties of the gas (density, velocity, pressure)
- Flame speed (velocity)
- Diameter of the jet (d_o)
- Length of the mixing pipe (L)
- Number and diameter of flame port holes (d_P)
- Height of the burner head from utensil.

Design specification may include:

- Cost
- Size
- Combustion efficiency / performers
- Etc.

During the design of the biogas burner, the following factors were put into consideration

- Efficiency and Flame Stability: Good thermal efficiency of burner to maximize energy conversion from biogas to heat with effective heat distribution to ensure even heating and stable and consistent flame performance across varying biogas compositions.
- Safety compliance and Environmental Impact: Design compliance with relevant safety and environmental regulations and standards to ensure low emissions of Carbon monoxide (CO) and Nitrogen oxides (NOx). Inclusion of biogas purification systems to remove impurities and reduce odor as well as efficient combustion to minimize greenhouse gas emissions.
- Materials and Durability: Use of corrosion-resistant materials suitable for biogas combustion. Durable design of burner to withstand continuous use and environmental conditions.

- Burner Size and Output: Appropriate sizing and output capacity of burner based on calculations to meet the specific heating requirements of the application.
- Compatibility: Burner design for integration with already existing biogas system and adaptable to fluctuations in biogas composition.
- Maintenance and Serviceability: Burner designed to enable easy access of components for cleaning and maintenance.

3.2 CONCEPTUAL DESIGN:

- Research on average gas pressure obtainable in biogas digesters, average specific gravity and percentage range of methane composition of biogas to aid informed estimates of data.
- Setting up design specifications
- Research and drawing of conceptual designs
- Selection of Design
- Calculation and detailing of part sizes.
- Production of design drawing
- Materials Selection
- Selection of Manufacturing methods, fabrication and assembly
- Performance evaluation and Comparative analysis.

3.3 DESCRIPTION OF THE BURNER ASSEMBLY

As earlier stated, four main components of the stove are the injector, the air/gas mixing chamber and the burner. The injector is connected to the mixing chamber which tapers down to the throat diameter of 14.7mm (given obtainable pipe diameter and manufacturing allowances) which is maintained as the diameter of the mixing chamber. The mixing chamber is connected to the burner head with two parts – the first being connected as a part of the mixing chamber and the second being a truncated cylindrical component having 32 burner ports, situated about 26^{o} to the horizontal axis, each of 3mm diameter drilled into it, from which the gas can be ignited. The frame of the stove supports the chamber and holds it in position by welded brackets to the frame. The combustion of biogas is regulated by the air shutter, which regulates the amount of air that enters the chamber. The frames and the stands are made from angle bars. The stove is to be connected to the gas holding unit of the biogas digester by a gas hose which conveys biogas from the gas holder of the plant to the stove.

3.4 CONCEPTAUL DESIGNS

During the course of this study, a number of designs were birthed and these will be discussed to explain concept, shortcomings and optimizations made.

3.4.1 CONCEPT 1

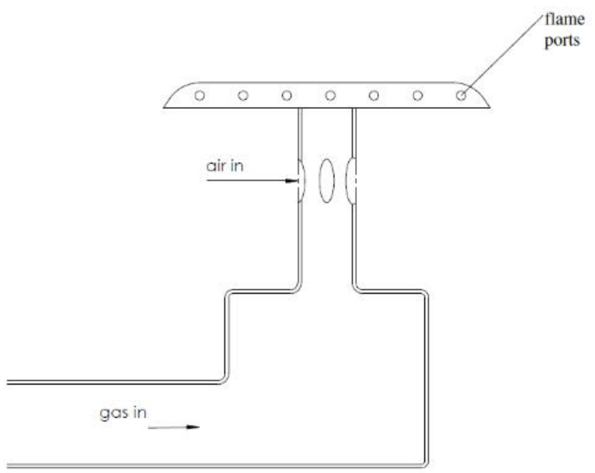


Figure 3.1 Conceptual Design 1

Brief Description of Concept

- This design was set up like a domestic LPG burner (commonly known as camping gas).
- The gas flows through an L-shaped channel and premixes with air from the air intake orifices before being ignited at the single row of flame ports at the burner head.

Considered unsuitable because:

- Length of mixing chamber considered too short as the mixing chamber should be designed with a recommended length of 10 times the throat diameter to allow proper mixing of air and gas.

- Design could not show satisfactorily that the air/gas mixture does not escape through the air intake orifices when approaching the burner head.
- Further research on gas burners showed that the "L" shape of the mixing chamber in this concept is not ideal as there should be no sudden changes in the geometry of the chamber because this affects the flow of the air/gas mixture. (US DoC, Circular No. 94)

3.4.2.i CONCEPT 2

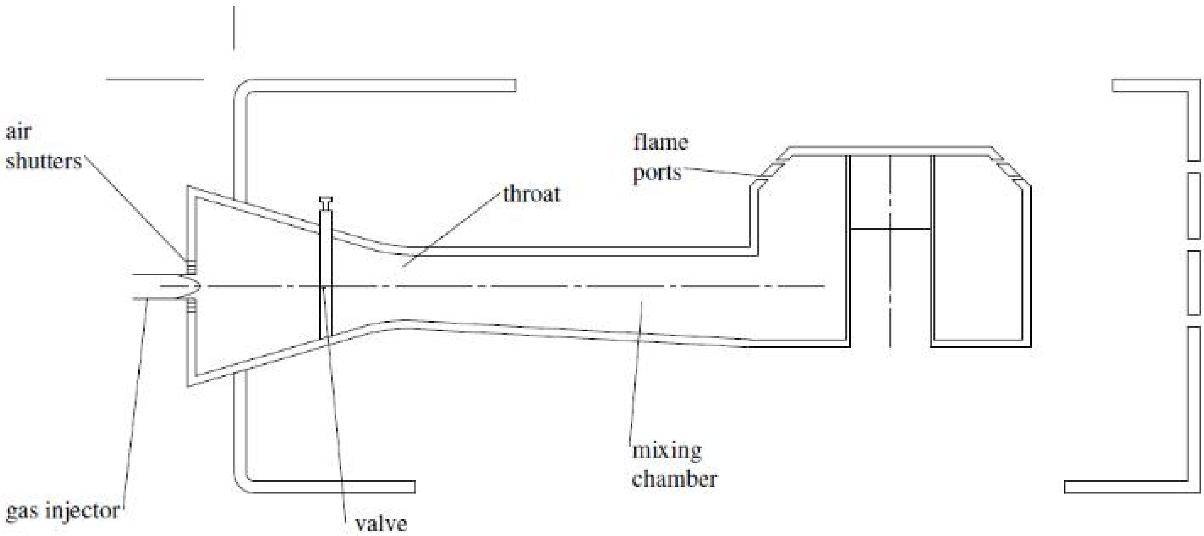


Figure 3.2 Conceptual Design 2

Brief Description of Concept

Gas flows into the mixing chamber through the gas injector, the high pressure creates a pressure differential which sucks in air through the air shutter to mix with the gas as it flows through the mixing chamber towards the flame ports. The mixing chamber geometry is a "venturi" or "diffuser", with a pipe that tapers into the throat and tapers smoothly away again (Fulford, 1996). The air flow in the venturi is also controlled by the throttle valve as it is moved in and out of the mixing chamber. The stove frame designed as an encasement with air gaps that allow surrounding air access to the burner head. Further descriptions are:

- Air shutters introduced as the primary air intake close to the injector nozzle
- Use of a venturi geometry with a throttle valve rather than a straight cylindrical mixing chamber

Considered unsuitable because:

- The position of the injector was considered to be brought further into the mixing chamber.
- Space beneath the burner head seen as unnecessary given fabrication considerations.
- Suggestions were made to redesign the stove frame given considerations to possible overheating of frame

3.4.3.ii DEVELOPMENT ON CONCEPT 2

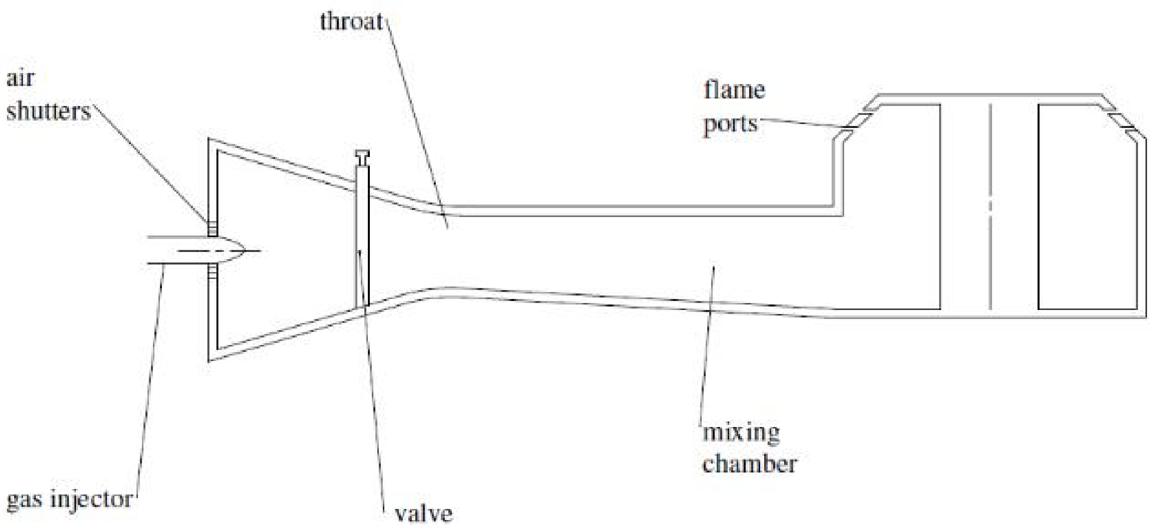


Figure 3.3 Development on Conceptual Design 2

Brief Description of Concept

- An optimization on Concept 3
- Injector brought further into the mixing chamber

Design was subjected to further optimization, based on research to

- Investigate the exact working of the air shutters.
- Consider a change of mixing chamber geometry.

3.4.3.i CONCEPT 3

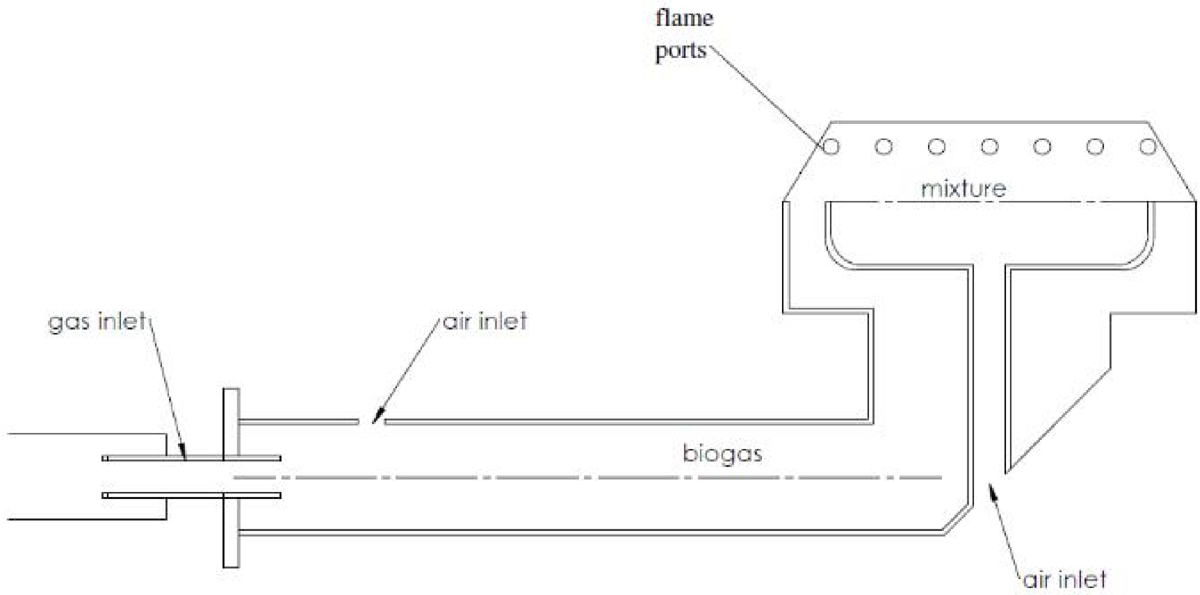


Figure 3.4 Conceptual design 3

Brief Description of Concept

Gas and air enter the mixing chamber through the gas inlet and air inlets respectively and flow towards the flame ports as they mix. Additional air is supplied from as air inlet directly below the burner head to join the premix before being ignited. Further descriptions are:

- Design set up like the standard LPG domestic burner
- Primary air intake channel positioned close to the gas inlet and directly below the burner manifold
- Single row of flame ports
- Gas flows in through the gas inlet and flows through the chamber to be premixed with primary air that rises into the mixing chamber that is directly below the flame ports

Considered because:

- No sufficient explanation could be made as to the mode of operation of the air intake located below the burner manifold.

- Complexity of manufacturing methods to fabricate this concept were considered.
- The design could not sufficiently show that the gas does not escape through the air intake orifice when the gas is throttled by the injector at high pressure.
- High possibility of improper mixing of air and gas.
- The geometry of the mixing chamber in this concept has abrupt changes and this could affect the gradual expansion of the gas at it approaches the ports. (US DoC Circular No. 94)

3.4.3.ii DEVELOPMENT ON CONCEPT 3

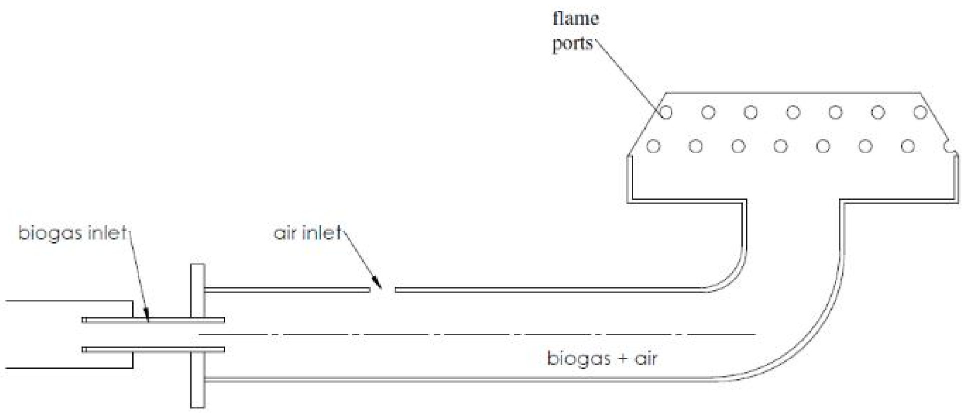


Figure 3.5 Development of Conceptual Design 3

Brief Description of Concept:

- An optimization on Concept 3
- Primary air intake positioned along the mixing chamber
- Double row of ports
- Consideration given to mixing chamber geometry (Circular No. 94 Bureau of Standards)

Considered unsuitable by the group because:

- The design could not sufficiently explain that the gas does not escape through the air intake orifice when the gas is throttled by the injector at high pressure.

3.4.3.iii FURTHER DEVELOPMENT ON CONCEPT 3

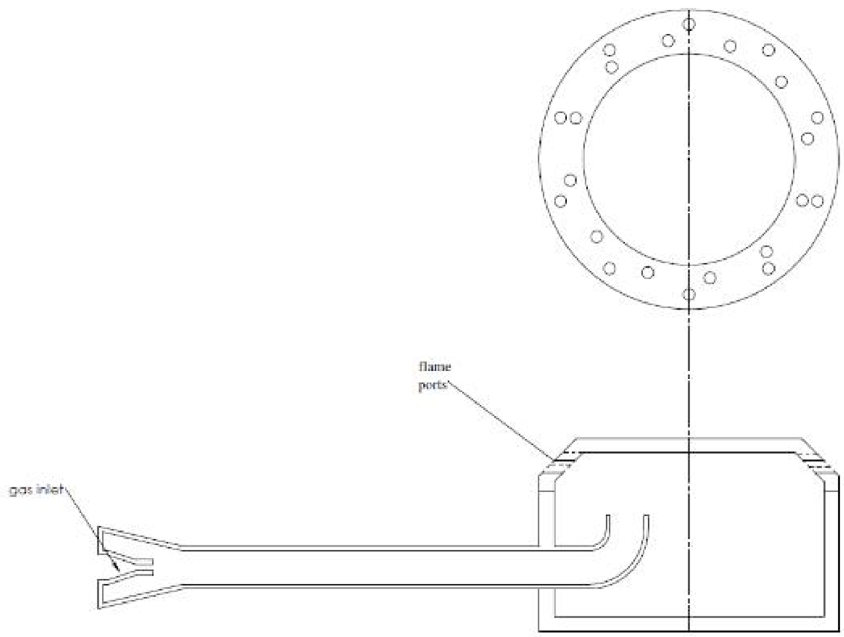


Figure 3.6 Further Development on Concept 3

Optimization made to design:

- Use of air shutters as the air inlet and control mechanism.

3.5 SELECTION OF CONCEPT

S/N		Concept 1	Concept 2	Concept 2(II)	Concept 3	Concept 3(II)	Concept 3(III)
1	Proper air/gas mixing	Low (0.5)	Moderate – High (0.76)	Moderate (0.7)	Moderate (0.72)	High (0.8)	High (0.8)

2	Position of flame ports	Side, Angled (0.7)	Side, Angled (0.7)	Side, Angled (0.7)	Side, Angled (0.7)	Side, Angled (0.7)	Top, Angled (0.65)
3	Air Control mechanism	Pressure induced (0.4)	Pressure and Suction (0.44)	Pressure induced (0.4)	Air Shutters (0.82)	Air Shutters (0.82)	Air Shutters (0.82)
4	Pattern of flame ports	Single Row (0.7)	Double Row, alternating (0.76)	Double Row, alternating (0.76)	Single Row (0.7)	Double Row, alternating (0.76)	Double Row, alternating (0.76)
5	Ease of Manufacture	High (0.85)	Medium (0.65)	Medium (0.65)	Low (0.4)	Medium (0.65)	Medium (0.65)
6	Ease of Maintenance	Low (0.4)	Low (0.4)	Low (0.4)	Low (0.4)	Low (0.4)	Medium (0.65)
7	Cost	Cheap (0.8)	Moderate (0.65)	Moderate (0.65)	Very expensive (0.3)	Moderate (0.65)	Moderate (0.65)
	TOTAL	4.65	4.36	4.28	4.04	4.78	4.98

Table 3.1 Selection of conceptual design

From the Table, Concept 3(iii) has the highest total, hence is selected and the calculations for the detailed design is carried out.

3.6 CALCULATIONS FOR DETAILED DESIGN

The study aims to design a burner that supplies 2.1kW and an average efficiency of about 65%.

For given system, flow rate Q is defined as the product of the velocity of flow (v) and the area (A) through which the flow passes, mathematically:

$$Q = vA$$

Biogas appliance	Power supply (kW)	Biogas consumption (10 mbar) ($m^3\ h^{-1}$)
Gas lamp	0.8	0.18
'fridge burner	0.8	0.18
Domestic burners	1.2 to 5.5	0.3 to 1.2
Commercial burners	5.5 to 17	1.2 to 4
Dual-fuel engines	per kW out	0.56
Spark engines	per kW out	0.7

Table 3.2 Gas consumption of various biogas appliances (Fulford, 1996)

Given the above,

Theoretical heat output of stove = $\frac{2.1}{0.65}$ = 3.23kW

using an injector with a discharge coefficient of 0.88, the biogas flowrate can be estimated to be

$$Q = 0.0467 C_d A_o \sqrt{\frac{p}{s}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots(1)$$

Where, A_o is area of orifice, p is pressure (10 mbar), *s* is specific gravity of biogas (0.94)

Interpolating from given data for domestic burners

$$Q = \left[\frac{2.1-1.2}{5.5-1.2} \times (1.2 - 0.3)\right] + 0.3 = 0.488\ m^3\ hr^{-1}$$

From (1), calculating for orifice diameter, d_o

$$Q = 0.036 C_d (d_o)^2 \sqrt{\frac{p}{s}}$$

$$d_o^{\ 2} = \frac{Q}{0.036 C_d \sqrt{\frac{p}{s}}} = \frac{0.488}{0.036 * 0.88 \sqrt{\frac{10}{0.94}}} = 4.72278$$

$d_o = 2.173 = 2.2$ mm

velocity of gas coming out of orifice

$Q = v_o A_o$

$$v_o = \frac{0.488}{3600*\left(\frac{\pi*\left(2.2*10^{-3}\right)^2}{4}\right)} = 35.66\ ms^{-1}$$

calculating the entrainment ratio, r,

given that stochiometric air requirement is 5.52

$$r = \frac{5.52}{2} = 2.76$$

calculating for throat diameter, d_t

using Prigg's formula $d_t = (\frac{r}{\sqrt{s}} + 1)d_o = (\frac{2.76}{\sqrt{0.94}} + 1) \times 2.2 = 8.5mm$

designing for maximum aeration i.e $r = 5.52$

$$d_t = (\frac{5.52}{\sqrt{0.94}} + 1) \times 2.2 = 14.7mm$$

length of mixing tube

$$L_m = 9.8d_t = 9.8 \times 14.7 = 144.06mm$$

calculating gas pressure at throat, p_t

given that atmospheric pressure, p_o, is 1.01 bar (10100 Pa)

$$p_t = p_o - \rho\frac{{v_o}^2}{2g}\ [1 - \left(\frac{d_o}{d_t}\right)^4\ ] = 101000 - 1.0994\frac{35.66^2}{2*9.81}\ [1 - \left(\frac{2.2}{14.7}\right)^4\ ]$$

$$p_t = 101000 - 71.22 = 1.0093\ bar$$

flow rate for mixture, Q_m

$$Q_m = \frac{Q(1+r)}{3600} = \frac{0.488(1+2.76)}{3600} = 5.097 \times 10^{-4}\ m^3s^{-1}$$

Checking flow condition in mixing tube (laminar or turbulent)

$$R_e = \frac{4\rho Q_m}{\pi\mu d_t} = \frac{4*1.15*5.097*10^{-4}}{\pi*1.71*10^{-5}*0.0147} = 2969$$

Since $R_e > 2000$ (turbulent), $f = \frac{0.316}{{R_e}^{\frac{1}{4}}} = \frac{0.316}{(2969)^{\frac{1}{4}}} = 0.0428$

pressure differential, Δp

$$\Delta p = \frac{16f\rho L_m {Q_m}^2}{2\pi^2 * {d_t}^5} = 2.174 = 2.174 \times 10^{-5}\ bar$$

Total port area, A_p

Given that biogas has a stochiometric speed of $0.25ms^{-1}$

$A_p > \frac{Q_m}{0.25}$

hence, $A_p > \frac{5.097 \ x \ 10^{-4}}{0.25} > 2.0388 \ x \ 10^{-3} \ m^2$

number of burner ports, n_p, required

using port diameter, d_p, of 4mm

$n_p = \frac{4A_p}{\pi {d_p}^2} = \frac{4*0.0020388}{\pi*0.004^2} = 162.24$

factoring flame stabilization, number of flame ports can be reduced to $\frac{n_p}{5}$

$\frac{n_p}{5} = \frac{162.24}{5} = 32.448$ (approximately 32 ports)

3.7 PRODUCTION OF DESIGN DRAWING

Concept 3(IV) was drawn and modelled using SolidWorks software to create the 2D and 3D detailed drawing of the stove. These are as shown below

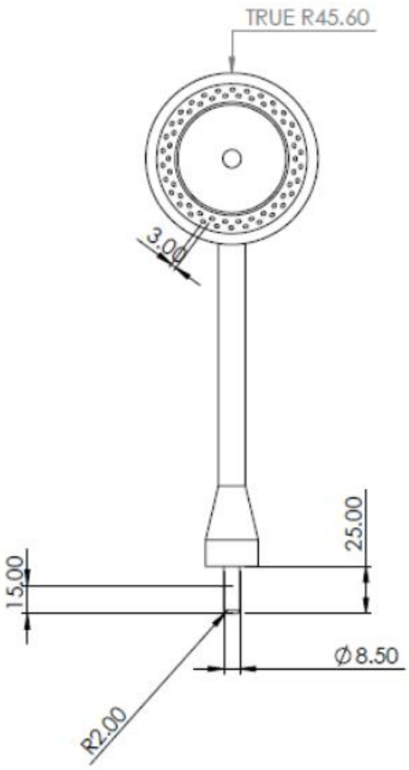
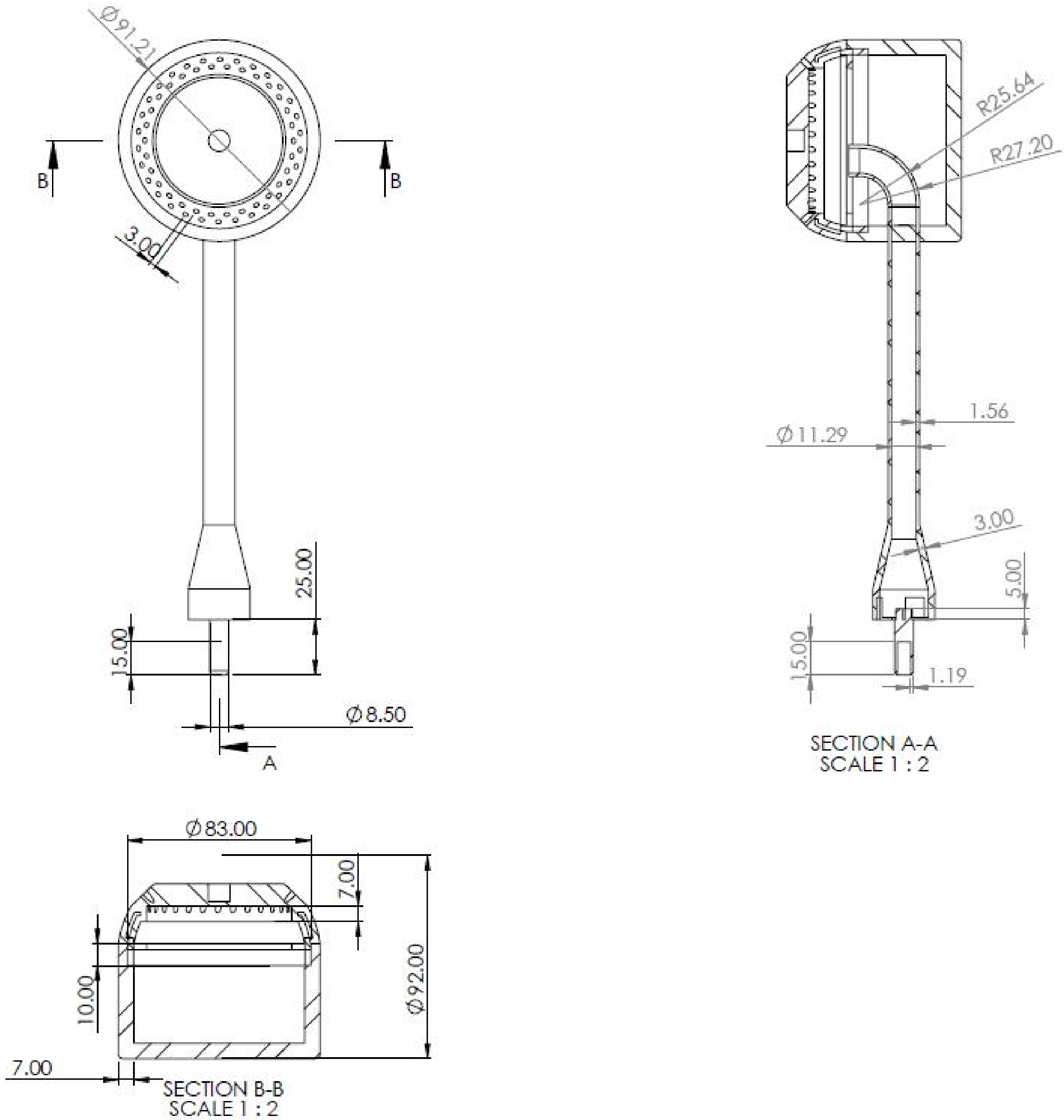


Figure 3.7 Top view of burner component



3.8 Sectioned view of burner component

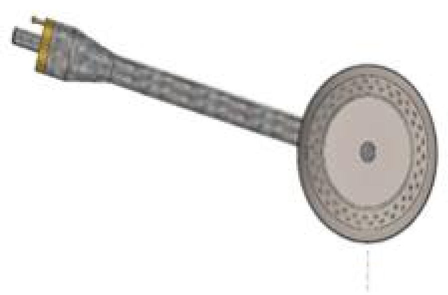


Figure 3.9 Pictorial Representation (2D render top view) of burner component

3.8 MATERIAL SELECTION

S/N	MATERIAL	ADVANTAGES	CONSIDERATIONS
1	Stainless Steel	- Corrosion Resistant - Durable: *can withstand high temperatures, mechanical stress, wear and tear* - High temperature resistance: *able to withstand high temperatures without deforming.* - Recyclable - Low maintenance requirement - Aesthetic appeal and Hygiene: *Non-porous, sleek looking finish of the metal enhances the overall look of the burner and makes ideal applications where hygiene is essential due to ease of cleaning.*	- Thermal Conductivity: *comparatively lower thermal conductivity to other materials which may affect heat distribution and efficiency* - Machinability and ease of fabrication: *might be more challenging to work with due to its hardness properties.* - Cost *implications on overall cost of the burner* - Comparative Weight
2	Cast Iron	- Heat Retention and Good Heat Distribution - Relatively Durable - Versatile: *can handle wide range of cooking heat requirement.* - Cost Effective	- Susceptible to Rust - Brittle - Comparative Weight - Comparatively higher maintenance requirement - Aesthetic consideration
3	Galvanized Mild Steel	- Corrosion Resistance: *provides good corrosion resistance owing to its zinc coating* - Durable - Cost effective and readily Available - Strength - Heat Resistant	- Zinc vapor hazard - Comparative weight *which can limit its potability and ease of installation* - Maintenance: *Zinc coating may degrade over lifetime of burner potentially reducing corrosion resistance. Increased*

			maintenance may be required to mitigate this. - Environmental concerns: *Galvanizing of steel has environmental considerations as well as disposal of the material after the burner's useful life*
4	Brass	- Corrosion Resistant - Durable - Good heat conductivity - Low maintenance requirement - Aesthetic appearance	- Relative Weight - Cost and limited availability - Tarnishing
5	Mild Steel	- **Strength and Durability with good heat resistance** - Readily available - Cost Effective - Ease of Machinability and Weldability	- Low corrosion resistance *that would require coating or treatments to improve the corrosion resistance.* - Comparatively low heat conductivity - Comparative weight
6	Aluminium	- **Excellent Corrosion Resistance** - **Lightweight** - **High thermal conductivity and energy efficiency:** ***This allows for efficient heat distribution and faster heating times and efficient heat distribution*** - **Recyclable** - **Good aesthetic property**	- Low material strength and Rigidity: *Compared to steel, aluminum is not as strong and thiscan affect the structural integrity of the burner* - Lower Melting Point *as compared to steel* - Thermal Expansion: *Relatively high coefficient of thermal expansion which can present the problem of deformation or dimensional changes with temperature variations* - Cost: *less expensive than stainless*

			steel but usually more costly than mild steel
7	Copper	- **Corrosion resistant with low maintenance requirement** - **Durable** - **Good thermal conductivity** - **Good aesthetic property**	- Cost considerations - Limited availability - Environmental concerns as regards disposal - Comparative Weight

Table 3.3 Advantages and Consideration of Various Materials for Material Selection of Biogas Stove

Given the above research findings, materials selected for the biogas stove were as follows

1. Stainless steel to be used for the burner head.
2. Galvanized mild steel for mixing chamber due to its corrosion resistant properties.
3. Brass or copper to be used (based on availability) for the injector component due to good corrosion resistant properties.
4. Mild steel to be used for the stove support frame.

NOTE: Given considerations to cost, galvanized mild steel can be used for the detachable burner head rather than stainless steel.

3.9 MANUFACTURING METHODS, FABRICATION AND ASSEMBLY

In the fabrication of the chosen design, three major manufacturing options available were casting, forging and welding. Given time financial considerations, welding was selected.

Manufacturing process carried out were:

1. Marking out and cutting: Mild steel sheets and pipes were measured, marked out and cut into the required sizes.
2. Forming: The cut components that required bending were bent to requirement,
3. Welding: the various components were joined through electric arc welding.
4. Drilling: drilling operation was carried out to create the flame ports.
5. Surface Finishing and Assembly

S/N	PART NAME	PART DESCRIPTION	QUANTITY	UNIT COST	TOTAL COST
1	Burner body	100mm stainless steel pipe (2mm thickness)	1	1200	1200
2	Burner cap	4mm by 100mm truncated cylindrical component with 3mm diameter drilled holes	1	1500	1500
3	Burner stem	20mm diameter steel pipe (7mm thickness); 140mm long	1	1000	1000
4	Control valve	Purchased off-shelf, made of brass alloy	1	2500	2500
5	Stand and pot support component	50mm x 5mm flat bar, 50mm x 5mm angle iron. Fabricated to form a tripod stand.	1	7500	7500
6	Bolt and nut	M13 bolt and nut (mild steel)	1	100	100
7	Paint	Chemical compound – blue	¼ cup	3000	750
8	Consumables	Electrodes, cutting disc, grinding disc, buffer, etc	-	5000	5000
10	Miscellaneous	Transportation, power, labour, etc	-	10800	10800
			Grand Total		**30,050**

Table 3.4 Bill of Engineering Measurement and Evaluation for Biogas Stove

CHAPTER 4
RESULTS AND DISCUSSSION

4.1 RESULTS

The outcomes of the performance of the biogas burner developed, that is, the experimental results are also presented here and demonstrate the strengths and deviations from the theoretical calculations when subjected to an experimental environment.

Table 4.1 shows the design requirement and specification used in this study.

	Design requirement	Design specification
1	Diameter of injector orifice(mm)	2.2
2	Area of injector orifice (mm^2)	3.8
3	Length of mixing tube(mm)	144
4	Inner diameter of tube (mm)	14.7
5	Number of flame ports (mm)	32
6	Diameter of each port (mm)	3
7	Area of flame ports (mm^2)	7.07
8	Clearance between flame ports(mm)	5
9	Distance between pot and biogas stove burner(mm)	45

Table 4.1 Design Requirement and Specification

Based on the design drawing, material and manufacturing selections made, a prototype was developed and tested as shown on the overleaf.

Figure 4.1 Fabricated Biogas stove

Table 4.2 gives a summary of the performance of the biogas stove for boiling water. The information recorded shows the boiling rate, biogas consumption rate and efficiency of the stove.

Parameter	Water Test			
	1	2	3	Average
Quantity of water (litre)	1.5	1.5	1.5	1.5
Time taken to boil water(min)	5.42	5.27	5.09	5.26
Time for all flame ports to ignite	2.51	1.37	1.17	1.68
Steady Flame?	No	Yes	Yes	Good
Boiling rate (litre/min)	0.277	0.285	0.295	0.286
Biogas consumption rate Q (m^3/min)	0.488	0.488	0.488	0.488
Efficiency of burner	0.567	0.584	.6045	.5851

Table 4.2 Performance of the Biogas Stove

4.2 DISCUSSION

Three tests were carried out to ascertain the heating efficiency of the stove using an aluminum pot. The gas inlet control knob was turned the exact number of times for each test to maintain to an extent, the same gas flow rate and consumption per test. It can be observed that the first test as compared to the other two tests had longer heating time recorded and lower heating efficiency as well as the time required for all the jets to ignite. This could be attributed to the cold start-up of the setup in the initial test.

The average boiling rate for the three tests was 0.286*l*/min while the biogas consumption rate was estimated at 0.488m^3/min. The time taken for the 32 flame ports of the burner to ignite was 2.51, 1.37 and 1.17 with the average time set at 1.68 seconds. The efficiency of the stove ranges from 56 – 61% with average recorded efficiency of the stove for the three tests was placed at 58.51%.

Boiling of water was ascertained by observing bubbling and steaming of the water, clear blue flames were observed for most part of the tests with the exception of ignition and when there was little wind. Flame lift was not observed during the period of testing.

Sources of error during the tests may arise from stopwatch timing, however various timers were used simultaneously to reduce the error and the average was recorded.

The fabricated stove had minor accuracy deviations from the design drawing due to manufacturing methods used but these fall within acceptable dimensional deviations.

4.3 COMPARISM AND IMPLEMENTATION

Table 4.3 shows the combustion efficiency and overall efficiency from a research study carried out by Smith (Smith et al, 1993)

Fuel/Stove	**Combustion efficiency %**
Biogas	99.4
LPG	97.7
Kerosene	96.5
Wood	90.1

Table 4.3 Comparison of combustion efficiency of different types of stoves

From this table, the combustion efficiency of biogas informs of its viability for being a relatively clean energy source.

Table 4.4 shows the design requirements, specifications and efficiency of eight biogas stoves studied by Tumwesige (Tumwesige et al 2014)

Parameter	**KEJS**	**Reo**	**Tusk**	**Bremmen**	**Ideal**	**Psem**	**Double**	**Psem Large**
Diameter jet d_0 (mm)	5	6	6	5	6	8	6	5
Diameter throat d_t (mm)	24	27.5	28.2	28	26.8	26.5	24	38
Area jet A_0 (mm^2)	19.6	28.3	28.3	19.6	28.3	50.3	28.3	19.6
Diameter jet A_t (mm^2)	452.4	594.0	624.6	615.8	564.1	551.5	452.4	1134.1
Diameter ports d_p (mm)	5	6	6	6	6	6	2	5
Number ports N	20	20	20	20	20	21	28	40
Area ports A_p (mm^2)	392.7	565.5	565.5	565.5	565.5	593.8	88.0	785.4
Mix pipe length (mm)	145	160	158	159	162	149	130	192
Efficiency % Cold Data	20.2%	21.4%	24.8%	28.0%	21.7%	20.2%	20.9%	20.2%
Efficiency % Hot Data	20.4%	25.6%	22.1%	23.4%	18.0%	22.5%	22.1%	20.7%

Table 4.4 Design requirements, specifications and efficiency of 8 stoves

From the information above it can be seen that the developed biogas stove performed higher than all eight stoves which are presented in Table 4.4. Tumwesige in his study, concluded that most times, biogas stoves are not designed properly.

Comparing the performance of the developed biogas stove to information from other works and literature, inference drawn was that the developed biogas stove performed satisfactorily.

CHAPTER 5
CONCLUSION

Biogas has in last few years become an important alternative fuel as the world turns its focus towards clean energy with multiple its benefits which include the diversification of cooking fuel supply, reduction of local pollutants as well as improved air quality index. The challenge facing biogas implementation is the needed is the research and documentation to improve performances of different biogas devices that properly follow gas burner theory.

From the study, a biogas burner was designed and tested and the results obtained placed the average recorded heating efficiency of the stove at 58.51%.

Well designed and affordable biogas technology will hasten the acceptability and adoption of biogas devices and this can have a much positive impact on the living standards of rural communities and the world at large.

There is room for more studies to consider a burner’s ability to have high thermal efficiency at a greater range of power as it is important to do better tomorrow what was done best today.

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