

OPTIMIZATION OF DOMESTIC BIOGAS STOVE BURNER FOR EFFICIENT ENERGY UTILIZATION

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ABSTRACT

The inefficient indoor burning of fuel wood on traditional cook stoves generates pollutants, primarily carbon monoxide and many other human health-damaging emissions. It is from this risk that it is necessary to have an immediate shift to alternative cleaner fuel sources. Biogas, which is among the biofuels from biomass, is one of the resources that play a considerable part in a more diverse and sustainable global energy mix. For domestic purposes in rural areas of Tanzania, biogas provides a better option that can supplement the use of fossil fuels such as wood, charcoal, and kerosene, which is nonrenewable. However, the low efficiency experienced in the locally made biogas burners hinders the large-scale use of biogas amongst the population in the country. With the locally made burners, the users of biogas for the domestic application face problems including heat loss and high gas consumption which affects the whole cooking process. It is against this backdrop that the current study objectives incline on designing and improving the efficiency of the locally manufactured burners to achieve the uniform flow of fuel in the mixing chamber, which will result to the consistent heat distribution around the cooking pot. The optimization of the burner was done by using computational fluid dynamics (CFD) through varying the number of flame portholes, air holes as well as the size of the jet before fabrication. The increased efficiency of the burner has also contributed by the addition of the fuel distributor. The results showed that the optimum hole diameter of the jet was 2.5 mm and that of the manifold was 100 mm. The currently developed biogas burner was tested and compared with the other two locally made burners. The water boiling test (WBT) on these three burners showed that the developed burner has a thermal efficiency of 67.01% against 54.61% and 58.82% of Centre for Agricultural Mechanization and Rural Technology (CARMATEC) and Simgas respectively. Additionally, the fuel consumption of the developed burner was 736 g/L as compared to 920 g/L for CARMATEC and 833 g/L for that of Simgas. The developed burner and its corresponding cook stove are both environmentally friendly and economical for household utilization in Tanzania and other developing countries.

Keywords: Biogas burner, Computational Fluid Dynamics, optimization, Injector, Efficiency, Water Boiling Test

DECLARATION

I, Lucia Petro do hereby declare to the Senate of the Nelson Mandela African Institution of Science and Technology that this dissertation is my own original work and that it has neither been submitted nor being concurrently submitted for a degree award in any other institution.

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CERTIFICATION

The undersigned certifies that they have read and hereby recommend for acceptance by the Nelson Mandela African Institution of Science and Technology a dissertation entitled “Optimization of Domestic Biogas Stove Burner for Efficient Energy Utilization” in fulfillment of the requirement for the degree of Master’s in Sustainable Energy Science and Engineering of the Nelson Mandela African Institution of Science and Technology.

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DEDICATION

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LIST OF ABBREVIATIONS AND SYMBOLS

ATC	Arusha Technical College
ANSYS	Analysis of System
CAMARTEC	Centre for Agricultural Mechanization and Rural Technology
CFD	Computational Fluid Dynamic
FP	Firepower
HPCSP	High Power Cold Start Phase
HPHSP	High power Hot Start Phase
ICEM	Integrated Computer Engineering and Manufacturing
LHV	Lower Heating Value
LPG	Liquefied Petroleum Gas
LPSP	Lower Power Simmering Phase
NM-AIST	Nelson Mandela African Institution of Science and Technology
PDF	Probability Density Function
SC	Specific fuel consumption
TDR	Turn Down Ratio
SIDO	Small Industries Development Organization
WBT	Water Boiling Test
p_t	Pressure of throat
ρ	Density of air
d_t	Diameter of throat
d_j	Diameter of injector
g	Acceleration due to gravity
$\dot{v}$	Volumetric gas flow rate from the orifice
A_j	Jet area
H	Column of gas
n_p	Number of burner ports
d_p	Diameter port
Q	Biogas flow rate
P	Output power
Re	Reynolds
f	Friction loss
ΔP	Pressure drop

η_t	Thermal efficiency
M_w	Mass of water
C_{pw}	Specific heat capacity of water
C_{vbg}	Calorific value of fuel (biogas
V_{bg}	The volume of biogas consumed
T_i	The initial temperature
T_f	Final temperature
H_v	Latent heat of evaporation of water
W_v	Water evaporated.
W_r	Water remaining
t_f	Time at the end
t_i	Time at start
T_b	The boiling point
T_h	Altitude of an area
Δt	Time to boil water
t_f	Time at the end of the test (min),
t_i	Time at the start of the test (min)
Δt^T	Temperature corrected time to boil
$T1_f$	Water temperature at the end of the test
$T1_i$	Water temperature at the start of the test
SE^T	Temperature corrected specific energy consumption
SC^T	Temperature corrected specific fuel consumption
TDR	Turndown ratio
r_b	Burning rate
Δt	Total time
η_{ov}	Overall efficiency

CHAPTER ONE

INTRODUCTION

1.1 Background of the problem

As a vital need for any growth of a country, about 80% of the population of Tanzania has no access to energy, specifically electricity. They, therefore, strongly rely on traditional energy sources including but not limited to charcoal and firewood for cooking, lighting and other thermal applications as an alternative (Felix & Gheewala, 2011). Approximately 90% of the entire energy consumption is biomass (Felix & Gheewala, 2011; Rosillo-Calle, 2012). Environmental destruction caused by the consumption of firewood and charcoal has grown dramatically and is causing deforestation and soil erosion (Yasar *et al.*, 2017). Heating and cooking activities in the households are mostly done by the utilization of firewood and charcoal which causes indoor air pollution resulting in diseases that affect women and children (Felix & Gheewala, 2011). Renewable energy is an alternative to biomass and will solve this problem if well exploited (Burke & Dundas, 2015). To realize a significant social and economic leap, a country should divert its attention to the wind, solar and biogas as reliable energy. Therefore, to meet the energy requirement, alternative sources of renewable energy are needed such as biogas to reduce the frequency of using biomass and consequently reduce over-dependence on the forest resources (Yasar *et al.*, 2017).

Biogas is obtained from inexpensive, readily available material wastes in rural areas, such as animal wastes and crop wastes (Chandra, Takeuchi & Hasegawa, 2012). In Tanzania, approximately 21 tons of cow dung are produced each year and can be gasified to generate heat (Mwakaje, 2008). Biogas is considered as one of the greatest substitutes fuel that can supplement the use of solid biomass and fossil fuel for domestic applications particularly in developing countries (Abbasi & Abbasi, 2010). It is currently being used in Tanzania for heating and cooking (Mwakaje, 2008). It is environmentally friendly, reduces workload and improves women's livelihood (Anderman *et al.*, 2015). The country's dependence on foreign energy will be reduced if biogas is utilized effectively, has no environmental effects and does not affect human health (Surendra, Takara, Hashimoto & Khanal, 2014). The challenge arises that Liquefied Petroleum Gas (LPG) burner does not work directly on biogas (Tumwesige, Fulford & Davidson, 2014). The Wobbe index for LPG is considered to be twice for that of biogas which means these burners are not interchangeable gases (Suwansri *et al.*, 2015).

Efforts have been put in place to produce and distribute renewable energy technologies by improving burner cook stoves to have efficient utilization of biogas for domestic cooking.

Appliances used for combustion of gaseous fuel like acetylene, propane and natural gas to produce a flame with an exception of a few that use both air and fuel for heating products are generally defined as gas burners (Beyler, 1986). Vital parameters to be observed for combustion of biogas are flame speed, ignition temperature, gas pressure as well as the rate of mixing gas and air. The steadiness of biogas flame in reference burners was studied by Suwansri *et al.* (2015). However, the biogas used 70% methane including varying the primary airflow. These stoves are poorly designed especially in areas such as an injector diameter and throat diameter. To use biogas as an alternative for liquefied petroleum gas (LPG) in domestic applications, rectifications are required for the burner stoves while considering health, economic, and environmental issues (Suwansri *et al.*, 2015). The available biogas burners have not only poor quality and low energy output, but their designs have also been reported that they do not follow the burner theory sufficiently (Tumwesige, Fulford & Davidson, 2014).

This study, therefore, will concentrate on adjustments and fabrication of the biogas burner parameters especially in increasing the diameter of the injector and manifold, improve throat size, throat diameter and to reduce the distance between burner head and bottom pot, to suit for the cooking application.

1.2 Statement of the problem

Biogas fuel is reported to be among the best alternative fuels to supplement the use of solid biomass and fossil fuel for domestic applications. However, their application is discouraged by the properties of fuel and poorly designed burner stoves which lose heat to the environment and the increased indoor air pollution due to poor combustion (Tumwesige, Fulford & Davidson, 2014). Furthermore, the domestic application of the current biogas plants face challenges of high gas consumption which affects the whole cooking process. Also, designs of biogas burners have some disadvantages to users such as low heating value which causes variation for biogas production in digesters and it is also reported that biogas burners consume more time in cooking compared to those of LPG. This, therefore, means that there is a loss of heat and flame lift which is normally observed during cooking. Other factors which cause low efficiency are pot to burner gap, gas pressure, and flame speed

(Bailis *et al.*, 2007). In this study, therefore, a biogas burner will be optimized and tested its performance by comparing with the existing ones.

1.3 Rationale of the study

The users of biogas for the domestic application face problems such as heat loss and high gas consumption which affects the whole cooking process. The available locally man made biogas burners have low efficiency, their application is discouraged by the properties of fuel and poor design which does not follow gas burner theory sufficiently (Tumwesige *et al.*, 2014). This study will focus on design and fabrication of a biogas burner by improving its parameters, in order to maximize its efficiency.

1.4 Objectives

1.4.1 Main objective

To optimize a domestic biogas stove burner for efficient energy utilization.

1.4.2 Specific objectives

- (i) To optimize the efficiency of biogas burner through simulation of key parameters.
- (ii) To fabricate and test the performance of the optimized biogas burner.

1.5 Research equations

- (i) What are the optimal performance parameters of the biogas burner?
- (ii) What is the efficiency of the designed biogas burner?

1.6 Significance of the study

Almost people of the developing country have limited access to grid power and depend on biomass fuel for cooking and lighting. However, the use of firewood contributes to air pollution, deforestation and social economic impacts. This research is aimed at improving the existing biogas burner to maximize its efficiency, reduce gas losses, as well as improve human health and environmental impact.

1.7 Delineation of the study

The delineation of the study in the area of domestic biogas burner stove for the household to maximize the efficiency of the burner for cooking application. Fabrication of the prototype was done in the workshops of Centre for Agricultural Mechanization and Rural Technology (CAMARTEC) and Arusha Technical College (ATC) in Arusha, Tanzania while experiments to determine the performance of the cook stoves were carried out in the energy lab at CAMARTEC in Arusha, Tanzania. The experiments were designed to simulate real-life cooking processes and were conducted using the water boiling test (WBT) software version 3.0 (Bailis *et al.*, 2007). The WBT consists of three components: A test at high power that is conducted with both cold and hot start conditions and a test at low-power to simulate slow cooking tasks or tasks that require low heat. The WBT assesses the thermal efficiency (η_{th}), the firepower (FP), and the specific fuel consumption (SC) of the stove.

CHAPTER TWO

LITERATURE REVIEW

2.1 Biogas

Biogas has varied composition depending on the characteristics of a feedstock, and amount of degradation (Karellas, Boukis & Kontopoulos, 2010). Methane is the most active part of biogas with a percentage of about 50-70% followed by 30-40% carbon dioxide and other trace gases (Lansing, Botero & Martin, 2008). The energy content of biogas depends on the composition of methane (Petersson & Wellinger, 2009). Table 1 and Table 2 are shown the composition and the properties of the biogas.

Table 1: Composition of biogas

Gas	Composition %
Carbon dioxide (CO ₂)	30-40
Methane (CH ₄)	50-70
Nitrogen (N ₂)	1-2
Hydrogen (H ₂)	5-10
Hydrogen Sulfide (H ₂ S)	Traces
Water vapour (H ₂ O)	0.3

Igoni *et al.* (2008)

Table 2: Properties of biogas

Properties	Range
The air required for combustion (m ³ /m ³)	5.7
Density (kg/m ³)	0.94
Net calorific value (MJ/m ³)	21.7
Ignition temperature (°C)	700

Tumwesige, Fulford and Davidson (2014)

2.2 Combustion and heat utilization

A burner is an appliance which is used to convert biogas into useful heating for domestic cooking (Tumwesige, Fulford & Davidson, 2014). Biogas produced is transported through a pipe from the small family digester to end-users and is burning directly in the burner. Unlike

other applications of biogas which undergo compression, particulate removal and filtration, cooking does not require compressed biogas and therefore, gas utilization is independent of gas contamination (Al Seadi *et al.*, 2013). The application of biogas is similar to other combustible gases such as LPG but with varying properties which means that care must be taken to ensure efficient combustion.

In comparison with LPG based on important features such as ignition temperature, flame, gas pressure and gas/air mixing rate, biogas requires less air per cubic meter. This, therefore, implies that, with a constant amount of air, more biogas is consumed. According to Itodo, Agyo and Yusuf (2007) about 1 liter of biogas requires 5.7 liters of air for complete combustion while propane and butane require 23.8 and 30.9 liters of air respectively (Sasse, Kellner & Kimaro, 1991). The flame speed of LPG is greater than that of biogas. This may be accomplished by using conical orifices but, the bottom of the cooking pot acts as a speed breaker for the flame. Given that the efficiency of utilizing biogas is 55% in stoves. This, therefore, means that using biogas is super alternative energy in farm energy demands (Sasse, Kellner & Kimaro, 1991).

2.3 Gas stoves

Biogas is burned directly in stoves whereby this is the simplest way of utilizing it in households. The gas stove is equipment which changes the state fuel of biogas and other gases such as liquefied fossil fuel gas (LPG). The small amount of hydrogen sulfide or carbon monoxide still exists when biogas is burned. A biogas stove comprises a mixer and multi-holed burning ports that operate at atmospheric pressure. A normal biogas stove consists of a gas supply tube, gas injector jet, gas tap/valve, primary air opening(s) or regulator, gas mixing tube/manifold, throat, burner head, pot supports, burner ports and body frame as shown in Fig. 1. Other biogas stoves may have one or double burners, the variable capability to consume from 0.22 to 0.44 m³ of gas per hour or more (Khandelwal & Gupta, 2009). Biogas is delivered to the stove using a supply pipe at a speed which depends on both the operating pressure and size of the pipe.

The speed of the gas is increased by the injector jet at the inlet of the stove and is dependent on the diameter of the injector. This results in a draft that consequently sucks in the primary air which mixes with the gas in the mixing chamber. The speed of the gas in the mixing chamber is reduced before approaching the burner head. The cone shape of the burner head is

modified to balance pressure before the mixture leaves the burner through the ports with a slightly higher specific flame speed of biogas. Finally, more oxygen (secondary air) is drawn for complete combustion (Sasse, Kellner & Kimaro, 1991).

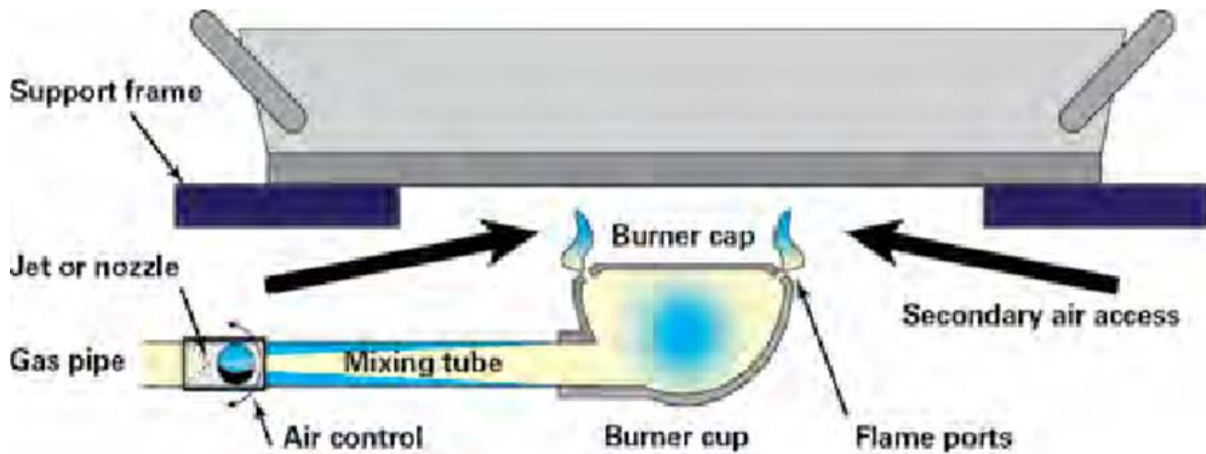


Figure 1: Typical biogas burner (Chandra, Tiwari, Srivastava & Yadav, 1991)

2.4 Types of gas stoves

Several types of biogas burners are used in the world including the Pecking stove used in China and Jackal stove which is common in Brazil. Others are KIE burner used in Kenya, and Patel Ge 32 stoves and Patel Ge 8 stoves generally in India (Itodo, Agyo & Yusuf, 2007). Two types of burners which are single flame burner and double flame as shown in Fig. 2 can be used to classify a biogas stove, although double burner is the most popular type used worldwide.

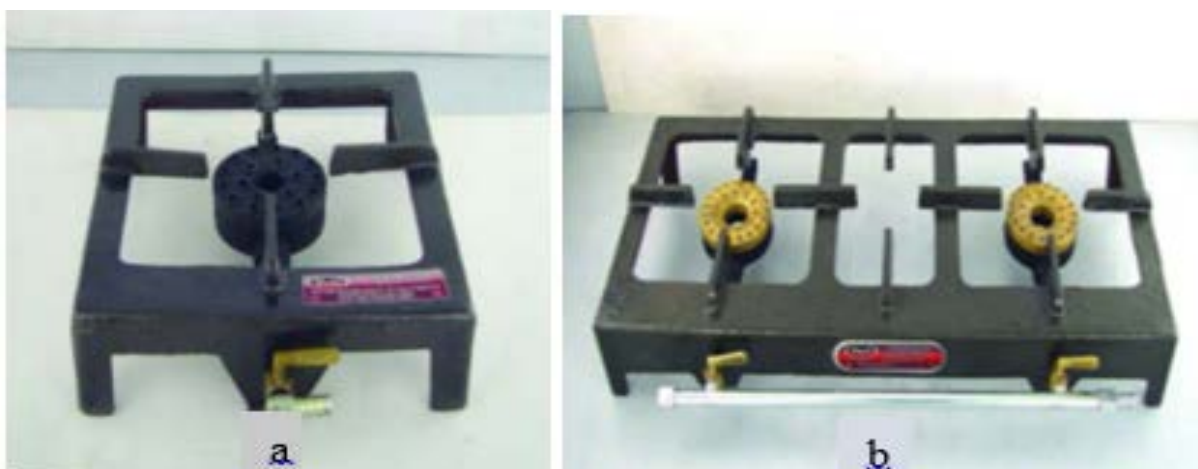


Figure 2: Gas stoves (a) Single flame (b) Double flame (Camera Photos)

2.5 Previous research on burners

Historical development of biogas burner began in the late 1970s (Decker, 2017a). Since then, researchers across the world have recognized the growing need for improvement of biogas burners to increase heating efficiency. In response to these, there have been several researchers and published journal articles and reports which describe the development and various modifications of biogas burners. In 1991, Chandra and colleagues at the Indian Institute of Technology in New Delhi developed a mathematical model which described the experimental performance of biogas burner by considering factors such as flame dynamic, burner head, the distance between the pot and burner head and reported the efficiency of 32 - 49% (Chandra, Tiwari, Srivastava & Yadav, 1991). In 1996 and 2014, Fulford and colleagues developed a biogas burner design guide and reviewed various biogas stoves designed in Uganda and reported low burner efficiency.

Furthermore, Itodo, Agyo and Yusuf (2007) modified biogas burners and tested their performance in cooking rice, boiling water and beans and the resulting efficiencies were 53%, 20% and 56% respectively. Olubiyi (2012) modified and upgraded biogas burners and obtained efficiencies of 60% and 21% in cooking rice and boiling water respectively. The Center for Research in Energy and Energy Conservation (CREEC) at Makerere University in Kampala, Uganda reported an efficiency of 20% tested for eight different biogas stove by boiling water although there were high levels of carbon monoxide emissions (Tumwesige, Fulford & Davidson, 2014). Although there have been recent research efforts, in the development of various domestic biogas burners, the problem of burner efficiency remains a bottleneck so this research will design the burner to increase its performance.

2.6 Gas burner

Biogas can be utilized as cooking fuel and in any gas-burning appliances that demand low pressure (such as a lamp, cooking stove, refrigerators etc.). Biogas is a clean gas that burns with no soot or other pollutants thus it can be used like any other flammable gas for household and industrial applications. The utilization of biogas requires special design parameters for biogas burners or modified consumer appliances to meet the properties of biogas.

2.7 Burner design

A common household burner within which biogas may be combusted consists of the subsequent main components: An injector, throat, primary air hole, burner head and mixing tube as shown in Fig. 3.

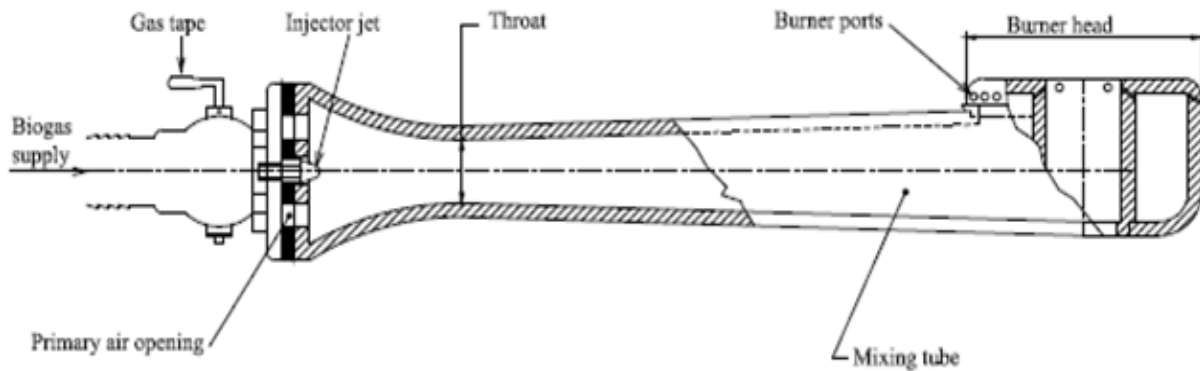


Figure 3: Detailed drawing of a biogas burner (Khandelwal & Gupta, 2009)

Generally, different fuels have different rates of burning and different oxidizing requirements and so designing of the injector and the burner head strongly depends on the fuel in question. The 56% mass flow of fuel to the burner head is affected by its lower heating value to achieve the specified firepower (Bhattacharya & Salam, 2002).

2.8 Mathematical modeling study by using computational fluid dynamic

Computational fluid dynamics (CFD) techniques help to analyse gas flow and other various parameters of the burner before the fabrication process. This process helps in studying the burner operation, and determining the parameters and phenomena that affect their performance from which the design can be optimized. Numerical analysis has brought further substantial improvements in the burner design parameters, in the detailed understanding of the flow and its influence on burner performance. The efficient application of advanced CFD is of great practical importance, as the design of the burner parameters. The main objective of this study is to optimize the efficiency of biogas burner through simulation of key parameters.

2.9 Biogas flame and impact of contaminants in biogas

2.9.1 Biogas flame

Figure 4 shows a biogas flame, as biogas emanates from the injector, the air is entrained and mixed with gas within the mixing tube before it is delivered to the burner ports. The unburned gas is warmed up in an inward cone and begins burning at the flame front. The cone shape of the flame may be a result of laminar flow in a round and hollow mixing tube, the blend at the middle of the tube moves at a higher velocity than at the exterior. Within the primary combustion zone, gas burns within the primary air and creates heat within the flame and combustion is completed at the external mantle of the flame with help of auxiliary air or the flame from the sides. As the combustion items rise vertically, i.e., carbon dioxide and vapours, heat is exchanged to the air near to the top of the flame. The hot air, which moves vertically away, draws in cooler auxiliary air to the base of the flame. The size of the inward cone depends on the primary air circulation. A high amount of primary air causes the flame to be smaller and concentrated, resulting in higher flame temperature. Complete combustion leads to a temperature greater than 850 °C and residential time 0.3 seconds (Sasse, 1988; Obada, 2016). The flame is dark blue and nearly invisible in the sunshine for complete combustion. Stoves are usually designed to operate with 75% primary air. If the air supply is low, there is incomplete combustion while with excess air, the flame cools off thus prolonging the operating time and increasing the gas demand (Tumwesige, Fulford & Davidson, 2014).

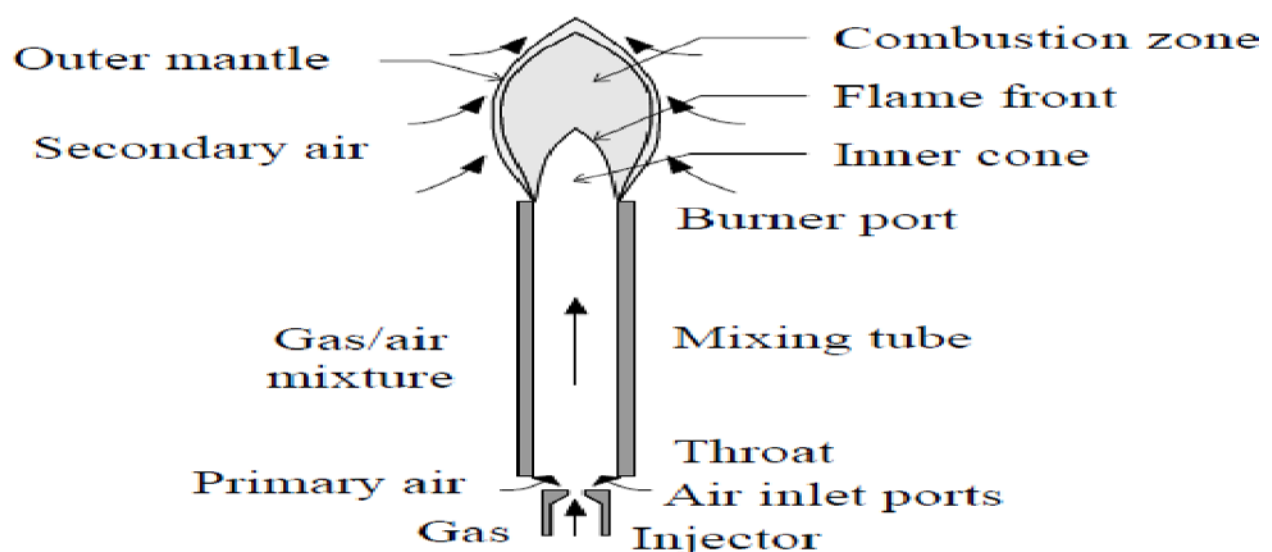


Figure 4: Biogas flame (Fulford, 1996)

2.9.2 Impact of contaminants in biogas

Significant volumes of carbon dioxide not only trap heat but also cool off the flame significantly to a point of unsteady state and blow out. The water vapour affects the flame temperature, lowers both the heating value and air/fuel proportion.

CHAPTER THREE

MATERIALS AND METHODS

3.1 Methodology

The methodology adopted in this work consists of analytical design procedures, 3D geometrical design of a burner, numerical design analysis using CFD, fabrication and development of a prototype as shown in Fig. 5. The procedure and analytical design of the burner included the findings of various important factors such as throat size, length of mixing chamber, injector jet diameter, the diameter of primary air inlets, number and diameter of flame port holes, and also the diameter and height of manifold (Demissie, Ramayya & Nega, 2016b; Fulford, 1996).

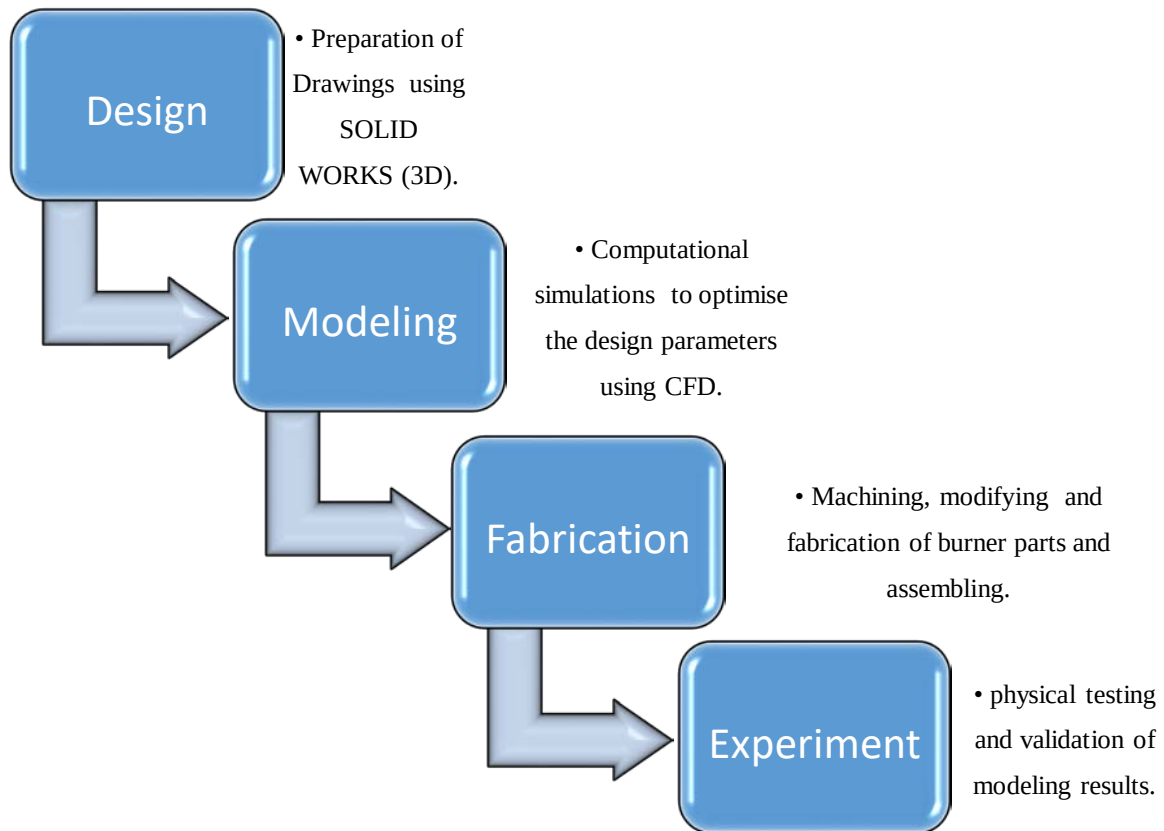


Figure 5: Methodological flow chart

3.2 Design of biogas stove

3.2.1 Analytical design

The following important parameters should be considered for the design of an efficient biogas burner was gas composition, gas pressure, flame speed and pot to burner gap (Obada, Obi, Dauda & Anafi, 2014). Other important considerations are the smooth gas channel to reduce air/gas resistance. Spacing and size of air gaps ought to coordinate with the necessity of gas combustion, a large volume of burner manifold to allow total mixing of the gas and air, size, shape and number of burner port holes ought to permit simple passage of the gas-air mixture, the arrangement of stabilized flame and complete combustion of gas without causing lifting up of flame, of the burner port or flame back stream from burner port to gas mixing tube and injector jet. The flame ought to be self-stabilizing i.e., flameless zones must re-ignite consequently within 2 to 3 seconds, in ideal situations, the port should be measured by the external cone of the flame with no contacts with the internal cone and burner shape and size.

(i) Injector

The size and the structure of the injector should be designed to control the rate of gas/air supplied and heat released for given gas composition and pressure (Demissie, Ramayya & Nega, 2016b).

(ii) Discharge from an orifice

An injector changes potential energy from the high-pressure gas supply into the kinetic energy of a developing gas jet (Demissie, Ramayya & Nega, 2016b). Assuming energy is conserved and no losses occur in the nozzle, this holds (per unit mass):

$$\frac{1}{2}v^2 = gh \quad (3.1)$$

Where:

$\dot{v}$ - Gas flow rate from the orifice (m^3s^{-1})

A_j - jet area (m^2)

g - Acceleration due to gravity

h - The column of gas required to exert gas pressure at the orifice

Gauge pressure (p) is given as:

$$p = h\rho_g g \quad (3.2)$$

Where, ρ_g = density of biogas (1.15 kg/m^3)

According to Jones (1989) specific gravity for CH₄ and CO₂ is 0.554 kg/m³ and 1.519 kg/m³ respectively. Considering the volumetric content of biogas to be 60% CH₄ and 40% CO₂ (Walsh, Ross, Smith & Harper, 1989) the specific gravity of biogas (s) is calculated as:

$$s = (0.554 \times \frac{60}{100}) + (1.519 \times \frac{40}{100}) = 0.94 \text{ kg/m}^3$$

The rate at which gas is delivered to the stove (Q, m³/h) depends on the size of the gas jet (Diameter: d_j, Area: A_j) and the pressure (p, mmHg):

From Equation 3.1 and 3.2:

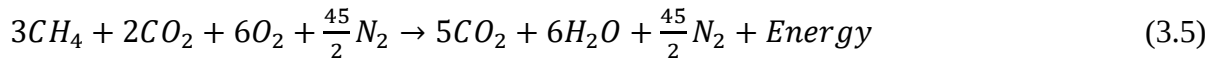
$$Q = 0.046101 \times A_j \times \sqrt{\frac{p}{s}} \quad (3.3)$$

Practically, the flow of gas before orifice is greater than after orifice due to friction losses and the vena-contracta effect. These two terms can be considered as a coefficient of discharge, C_d taken as 0.94 from (Jones, 1989) such that:

$$Q = 0.046101 \times C_d \times A_j \times \sqrt{\frac{p}{s}} \quad (3.4)$$

(iii) Biogas combustion

The biogas combustion with air mixture (nitrogen and oxygen) is shown in Equation 3.5:



Thus one volume of biogas requires 5.7 volumes of air or the stoichiometric necessity is

$$\frac{1}{(1+5.7)} = 0.149, \text{ i.e., } 14.9\% \text{ volume of biogas is required in air.}$$

Biogas will burn over a range of mixtures from roughly 9% to 17% in air. If the air-fuel mixture is 'too rich', i.e., has much fuel, it will undergo incomplete combustion, resulting in carbon monoxide (which is harmful) "lean". Therefore, it should be supplied with excess air to avoid this state (Fulford, 1996). In partially aerated burners, the air is mixed with the gas before it is burnt. The design of the burner affects the amount of primary air added to the gas before the flame.

(iv) Air entrainment

For a long time, theoretical and experimentally, the mechanism has been entrainment of air and domestic aerated burner designers are vital interest because primary air quantity taken up by considering the effect on flame stability, shape, ultimate, temperature, burner port design requirements and the design of the combustion chamber itself. The gas coming from the injector enters the beginning of the mixing tube in an area called the “throat”. The throat features a much bigger diameter than the injector, so the speed of the gas stream is much reduced. The velocity of biogas in an injector jet, V_{gas} is given by Equation 3.6:

$$V_{gas} = \frac{Q_{biogas}}{A_j} ms^{-1} \quad (3.6)$$

Where,

Q_{biogas} (m^3h^{-1}) is the flow rate of biogas, A_j (m^2) is the Area of injector jet.

Reduction velocity within the throat can be calculated by using Equation 3.7:

$$v_t = v_j \frac{A_j}{A_t} \quad (3.7)$$

Where,

v_t = Velocity of throat (m/s)

A_t = Area of the throat (m^2)

v_j = Velocity of the injector (m/s)

The vena contracta and friction loss the gas pressure after the nozzle can be calculated from Equation (3.8):

$$p_t = p_j - \rho_a \frac{v_j^2}{2g} \left[1 - \left(\frac{d_j}{d_t} \right)^4 \right] \quad (3.8)$$

Where,

p_t = Pressure at the throat (pa)

p_j = Pressure at the injector (pa)

ρ_a = Density of air (1.2 kg/m^3)

d_t = Diameter of throat (mm)

d_j = Diameter of the injector (mm)

g = Acceleration due to gravity (m/s^2)

The value of p_j is approximately atmospheric pressure, as the throat is open to the air, so this drop in pressure is adequate to draw primary air in through the air inlet ports to mix with the gas within the mixing chamber (Fulford, 1996).

The primary air circulation depends on the “entrainment ratio” (r), which is calculated by the area of the throat and the injector:

(v) Throat size

The flow rate of the gas/air mixture within the throat (Q_m) is expressed as:

$$Q_m = \left[\frac{Q_{biogas} \times (1+r)}{3600} \right] = 6.26 \times 10^{-4} \text{ m}^3 \text{ s}^{-1} \quad (3.9)$$

With Q_m in $\text{m}^3 \text{ s}^{-1}$ and Q_{biogas} in $\text{m}^3 \text{ h}^{-1}$

Where, r is the entrained air to gas volume ratio, given by $r = \frac{Q_{air}}{Q_{biogas}}$.

The pressure drop due to the stream of the blend down the blending tube depends on the Reynolds number as shown in Equation 3.10:

$$Re = \frac{\rho_m d_t v_t}{\mu} = \frac{4 \rho_m Q_m}{\pi \mu_m d_t} \quad (3.10)$$

Where, $\rho_m = (1.15 \text{ kg/m}^3)$ and $\mu_m = (1.71 \times 10^{-5} \text{ Pa})$ are the density and viscosity of the mixture at 30 °C.

In the above expression Pascal is also known N/m^2 as which is represented by Pascal:
hence $\text{N/m}^2 = \text{Pa}$.

The pressure drop (ΔP) is determined using Equation 3.11:

$$\Delta P = \frac{f}{2} \rho_m v_t^2 \frac{l_m}{d_t} = \frac{f}{2} \rho_a \frac{16 Q_m^2}{\pi^2 d_t^5} l_m \quad (3.11)$$

Friction loss (f) was calculated by using Equation 3.12:

$$f = \frac{Re}{64} \text{ When } Re < 2000 \text{ and } f = \frac{0.316}{Re^{\frac{1}{4}}} \text{ when } Re > 2000 \quad (3.12)$$

Hence, $Re > 2000$ (flow of biogas air mixture in the mixing tube is turbulent). The driving pressure should be greater than the pressure drop. Most burners are developed to have a throat that gives aeration larger than the optimum, with a device for confining the airstream, so the optimum air circulation can be set for a given circumstance.

(vi) Mixing tube

The mixing tube and diffuser as one unit impact on air entrainment of mixing tube length both downstream and upstream of the throat. The distance from the throat entrance to the injector ought to be 2 to 2.5 times the throat distance across, and that the mixing tube length ought to be 10 to 12 times the throat diameter (Harneit, 2004).

(vii) Burner ports

The greater benefit of a gas burner is that the heat can be coordinated to where it is required. When a biogas/air mixture has lighted, the fire front delivers products through the remaining unburnt gasses at a rate proportional to the mixture composition, temperature and pressure. The flame speed could be an essential property of the mixture and is linked to the general chemical reaction rate within the fire. Burning velocity is characterized as the speed normal to the fire front, relative to the unburnt gas, at which an infinite one-dimensional flame propagates through the unburnt gas mix. Biogas incorporates a stoichiometric burning velocity of 0.25 m/s (Fulford, 1996).

The mixing supply velocity (v) is given by Fulford (1996):

$$V_p = \frac{Q_m}{A_p} \ll 0.25 \frac{m}{s} \quad (3.13)$$

$$A_p = n_p \frac{\pi d_p^2}{4} \quad (3.14)$$

Where,

A_p (mm²) - total area of the burner port

n_p - Number of burner ports

d_p (mm)- Diameter of each port

N.B. Material selection of an injector is a brass shaft that resists corrosion and has good heat expansion.

3.2.2 Determining the power required

The power required for the process was calculated by the biogas flow rate to meet the power input required for the process, $C_{vbiogas} = 22 \text{ MJ/m}^3$ (Khandelwal & Gupta, 2009). From the experimental test results on the study conducted at CAMARTEC, the developed burner was observed to have a boiling time of 11 min to boil 2.5 litres of water, to a boiling point of 95.23 °C, the gas flow rate was 0.63 m³/h and thermal efficiency of 11.63% during in high power cold start phase (HPCSP). The stove output power (P , kW) was calculated by using Equation 3.15 (Foundation, 2012):

$$P = 5.2 \times \frac{V}{t} \quad (3.15)$$

Where,

V = Volume of water (2.5 litres)

$$P = 5.2 \times \frac{2.5}{11} = 1.18 \text{ kW}$$

The gas through the stove (Q , m³/h) can be determined by Equation 3.16 (Foundation, 2012).

$$Q = 0.45 \times P \quad (3.16)$$

Where,

Q = Biogas flow rate

P = Output power

So if the desired stove output power is 1.18 kW, the stove ought to have a gas flow of 0.68 m³/h.

The power required was calculated by the biogas flow rate to meet the power input required for the process, $C_{vbiogas} = 22 \text{ MJ/m}^3$ (Khandelwal & Gupta, 2009).

$$P_{required} = Q_{biogas} \times C_{vbiogas} = \frac{0.68 \times 22 \times 10^3}{3600} = 4.1555 \quad (3.17)$$

Where,

$C_{vbiogas}$ = Calorific value of biogas

$P_{required}$ = Power required

The power required of the process is 4.16 kW.

3.2.3 Design calculations of stove parameters

(i) Injector jet diameter or orifice jet diameter

The Injector jet diameter, d_j was calculated from Equation 3.18:

$$d_j = \sqrt{\frac{Q_{biogas}}{0.0361 \times C_d}} \times \sqrt[4]{\frac{s}{p}} = 0.0250 \text{ m} \quad (3.18)$$

Where Gas flow rate, $Q_{biogas} = 0.63 \text{ m}^3/\text{h}$ obtained from the experimental test results conducted at CAMARTEC for boiling 2.5 litres of water to a boiling point of 95.23 °C during HPCSP. The pressure has been considered from the pressure measurements taken for the bio digester at CARMATEC was 8.5 mbar and orifice diameter of 2.5 mm has been calculated from Equation 3.18 and as used as the input value for CFD simulation in Section 3.2.4. The area of the injector orifice (A_j) was determined as follows.

$$A_j = \frac{\pi \times (d_j)^2}{4} = 4.908 \times 10^{-6} \text{ m}^2 \quad (3.19)$$

The velocity of biogas in an injector jet, V_{gas} was then

$$V_{gas} = \frac{Q_{biogas}}{A_j} = 36.165 \text{ m s}^{-1} \quad (3.20)$$

$$\text{Gas flow rate, } Q = 0.046101 \times C_D \times d_j^2 \times \sqrt[4]{\frac{p}{s}},$$

(ii) Determination of throat size, d_t

For complete combustion, the stoichiometric air required is 5.7. At that point the entrainment ratio, r is $5.7/2 = 2.85$. The flow rate of the biogas and air mixture at ideal air circulation is determined by using Equation 3.20; throat diameter was computed using Prig's formula which shows in Equation 3.21:

$$d_t = \left[\left(\frac{r}{\sqrt{s}} + 1 \right) d_j \right] = 9.85 \text{ mm} \quad (3.21)$$

The stoichiometric air requirement value of primary air of 5.7 specifically instead of using $r = 2.85$ to perform good aeration and control utilizing primary air stream adjuster. So, the way to better design the diameter of the throat will be 18 mm, at that point the throat area gets to be 254.469 mm^2 . The area of air inlet ports must have the same to that of the throat (Fulford, 1996). The exact size depends on the standard pipe sizes available. The gas pressure within the throat can be evaluated by using Equation 3.8 as:

$$p_t = p_j - \rho_a \frac{V_j^2}{2g} \left[1 - \left(\frac{d_j}{d_t} \right)^4 \right] = 99920 \text{ Pa}$$

The value of p_j is 10^5 Pa as the throat is open to the air. This pressure drop is adequate to draw primary air by the air inlet ports to mix with the gas within the mixing chamber (Fulford, 1996). The Reynolds number was calculated by checking the pressure drop due to the stream of biogas and air mixing within the mixing chamber using Equation 3.10:

$$Re = \frac{4\rho_m Q_m}{\pi \mu_m d_t} = 2978$$

Hence, $Re > 2000$ (flow of biogas air- mixture in the mixing tube is turbulent). Friction loss was calculated using Equation 3.12:

$$f = \frac{0.316}{\left(Re^{\frac{3}{4}} \right)} = \frac{0.316}{(2978)^{\frac{1}{4}}} = 0.0428$$

Pressure drop (ΔP) may well be calculated using Equation 3.11 is $\Delta P = 1.85$ Pa. So, the driving pressure is higher than pressure drop Δp within the throat.

(iii) Design of burner port

The flame speed of biogas is only 0.25 m/s (Fulford, 1996). The velocity supply of mixture (v_p) using Equation 3.13 is $v_p = \ll 0.25 \text{ ms}^{-1}$. The total area of the burner port will be calculated as:

$$A_p > \frac{Q_m}{v_p} \gg 0.0025 \text{ m}^2 \quad (3.21)$$

Fulford (1996) and Itodo, Agyo and Yusuf (2007) in their study utilized 5 mm and 2.5 mm holes diameter respectively. The flame lift was recorded at a distance across less than 2.5 mm (Olubiyi, 2012). Utilizing 3 mm port distance across to reduce the problem of flame lift, the number of ports will be:

$$n_p = \frac{4A_p}{\pi d_p^2} = 354 \text{ Ports} \quad (3.22)$$

Using flame stabilization, the number of burner ports can be reduced up to 1/5, so 71 ports are sufficient (Fulford, 1996). Applying 40 holes, with 7.0 mm gap distance between centre holes, arranged in a circular pattern, gives a total circumference of $40 \times (2.5 + 7.0) = 380 \text{ mm}$. Using 31 holes, with 4.5 mm gap distance between centre holes are placed around a top of pitch circle diameter 54 mm.

3.2.4 Geometry for a burner

A 3D drawing is shown in Fig. 6, using Solid Works software. To proceed with modelling and simulation, the 3D drawing was imported in the analysis of system (ANSYS) workbench as shown in Fig. 7

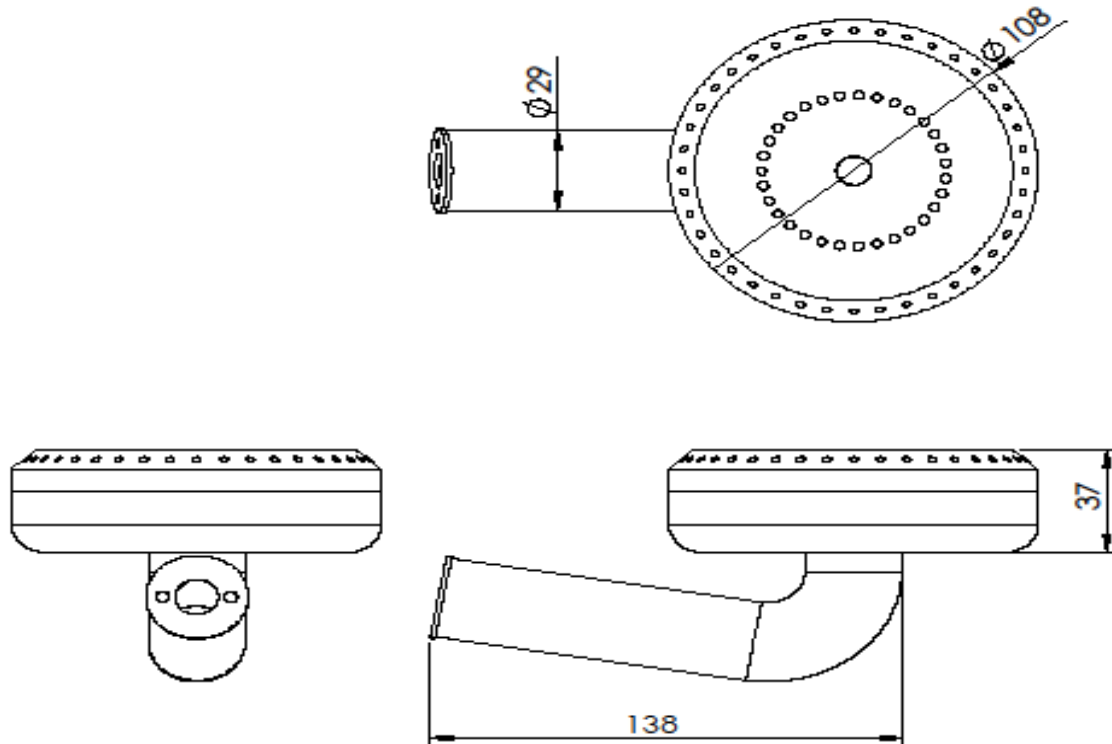


Figure 6: Designed drawing of a burner

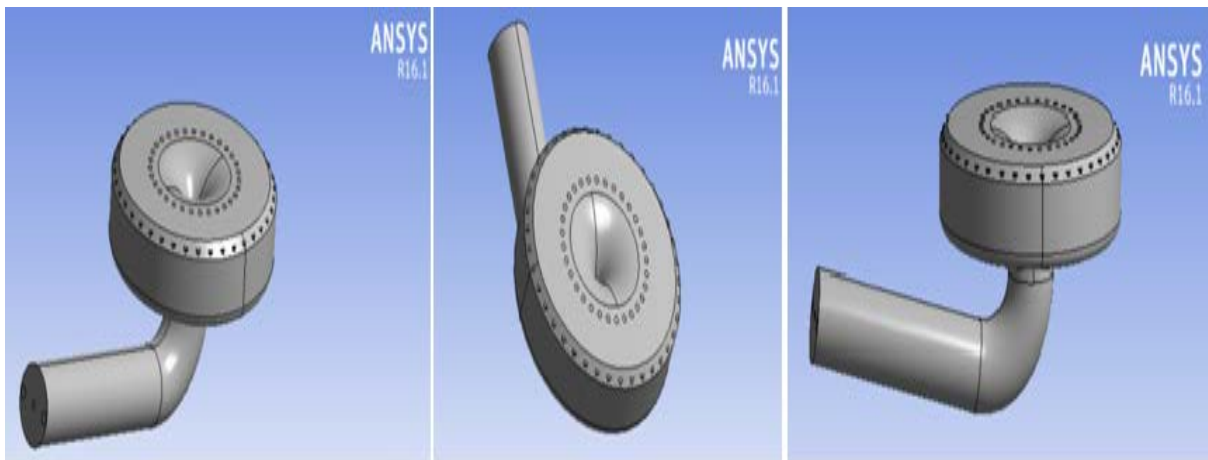


Figure 7: Elevation sides of a burner

3.3 Computational fluid dynamics (CFD) simulation and optimization of the burner

The CFD study is based on developing the simulation of biogas combustion to optimize the performance of the burner. Analysis of System 16.0 was used for the simulation and solid work was used to model the burner whereas the meshing was done by analysis of system integrated computer engineering and manufacturing-computational fluid dynamics (ICEM-CFD). Finally, the fluent was used to obtain the solution (Noor, Wandel & Yusaf, 2013;

Nordin, Yunus & Ani, 2014). Computational fluid dynamics can design and simulate a model without physically building the model.

Numerical modelling techniques including but not limited to CFD methods have gained fame in academic and industrial sectors because of the availability of efficient computer systems. It is possible to optimize the system design, operations and understand physicochemical properties inside the model using CFD simulations which can consequently reduce the number of experiments required for characterization of the system. Spatial and temporal profiles of different system variables that may be impossible to measure or are accessible only by expensive experiments can be obtained through simulations (Nordin, Yunus & Ani, 2014).

Figure 8 shows the 3D geometry obtained from solid works. Each dimension was marked on the Fig. 6 with corresponding input dimensions which can be varied. In this way, each dimension is easy and quickly modified and the entire construction can be modified, redrawn and further steps like mesh creation or design optimizations are possible (Jones & Launder, 1972; Launder & Spalding, 1983).

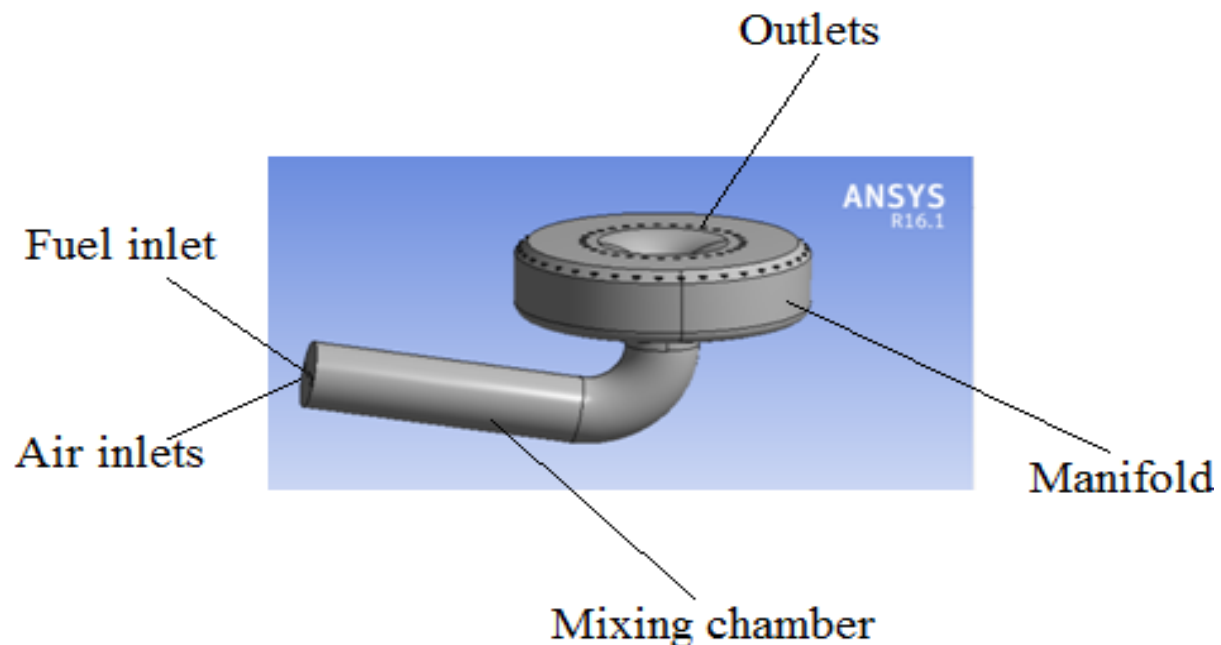


Figure 8: Boundary condition on a 3D model

There is a need to divide the geometry into grids or mesh since meshing is a vital part of the CFD process because the technique selected at the meshing stage determines the quality of the mesh. The meshing of the new and existing biogas burner models are shown in Fig. 9 and

they generate grids of elements used to solve fluid flow Equations. The ANSYS (ICEM-CFD) is used to generate the required grids (Nordin, Yunus & Ani, 2014). The grid size has both an impact on computational time which directly affects simulation cost as well as the speed of convergence and the accuracy of the solution (Noor, Wandel & Yusaf, 2013).

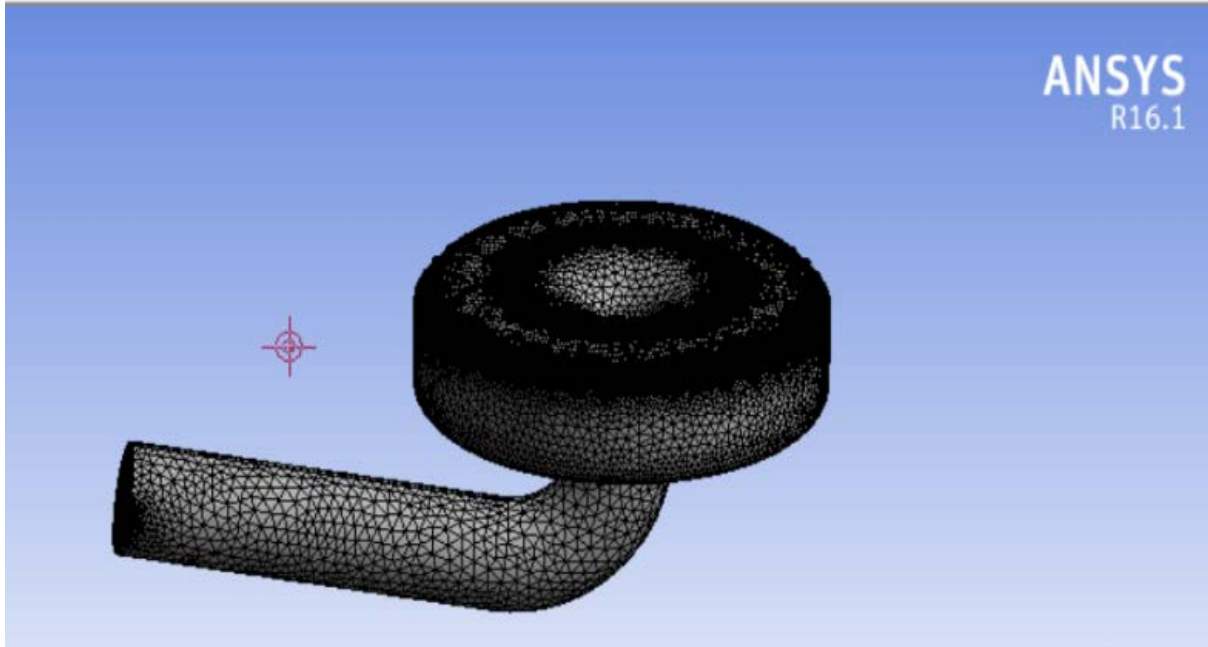


Figure 9: Mesh of geometry

To attain the fine resolution of flow and minimize the time taken by the computer to complete simulation, hexahedral cells are preferred. Additionally, refined grids not only result in a shorter time for calculation and number of iteration per time step but they also converge more easily.

The boundary conditions were determined based on the shape of the geometry as shown in Fig. 8. The Non-premix combustion with turbulent realizable K- ϵ and the model P1 were used in the simulation. The ANSYS-Fluent requires input parameters in terms of gas composition concerning CH_4 and CO_2 which are to be used in the simulation. The transient model was used as the initial setting in 3D geometry with the pressure-based solver. Table 3 shows the details of the model parameter settings and the input parameters for the simulation and Table 4 is the model input parameters used for the simulations.

Table 3: Settings of the fluent model

Model Parameters	Status
Multiphase	Off
Energy	On
Viscous	Standard k-e, Standard Wall Fn
Radiation	P1
Heat Exchanger	Off
Species	Non – Premixed Combustion
Inert	Off
NOx	Off
SOx	Off
Soot	Off
Decoupled Detailed Chemistry	Off
Reactor Network	Off
Reacting Channel Model	Off
Discrete Phase	Off
Solidification & Melting	Off
Acoustics	Off
Eulerian Wall Film	Off

Table 4: Model input data of the burner

Input parameters	Units	Value
Burner Dimensions	(L x H x W) mm	265 x 67.5 x 100
Species	CH ₄	0.6
	CO ₂	0.4
Air Inlets	Velocity (m/s)	5.2
	Temperature (K)	300
Biogas (Fuel)	Velocity (m/s)	35.01
	Temperature (K)	300
Operating Pressure	(Pa)	101325
Gravitational Acceleration	(m/s ²)	9.81

The Non-premix model was calculated by various types of species formed during combustion where the initial velocity of air and biogas was considered to be 5.2 m/s and 36.165 m/s, respectively. Table 5 shows the probability density function (PDF) creation boundary conditions for the fuel and oxidant. The biogas contains a mixture of CH₄ (60%) and CO₂ (40%) on a molar basis. As shown in the PDF Table 5, the oxidant consists of a mixture of nitrogen and oxygen 79% and 21% respectively, similar to normal combustion air configuration. The temperature for the fuel and oxidant was set at 300 K, and the CFD simulation was done by varying the geometry as well as manifold size to achieve optimum design to conform to the analytical results of Fig. 5.

Table 5: PDF for boundary conditions

Species	Fuel	Oxidant
CO ₂	0.4	0
H ₂	0	0
O ₂	0	0.21
CH ₄	0.6	0
N ₂	0	0.79

3.4 Governing Equations

The numerical analysis was carried out based on design and analysis techniques to obtain accurate solutions in solving the mass, momentum, and energy Equations which govern the gas flow (transport) and the combustion of biogas in the burner.

(i) Mass and the continuity Equation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho u) = 0 \quad (3.23)$$

Where: ρ – density of gas and u – velocity of gas

(ii) Momentum Equation

$$\rho \frac{\partial u}{\partial t} + \rho(u \cdot \nabla)u = -\nabla \rho + \nabla \cdot \{\mu[\nabla u + (\nabla u)^T] + \lambda(\nabla \cdot u)\} + \rho g \quad (3.24)$$

Where μ – viscosity of gas, T – Temperature, λ – wavelength of gas and
 g – Gravitational force of gas

(iii) Energy Equation

$$\rho C_v \frac{\partial T}{\partial t} + \rho C_v(u \cdot \nabla)T = -P(\nabla \cdot u) + \nabla \cdot (k \nabla T) + \lambda(\nabla \cdot u)^2 + \nabla u \cdot \{\mu[\nabla u + (\nabla u)^T]\} \quad (3.25)$$

Where C_v – Calorific value of gas, P – Pressure of gas and k – costant value

(iv) Species mass fraction

$$\frac{\partial \rho m_l}{\partial t} + \nabla \cdot \rho u m_l = \nabla \cdot D_e \nabla m_l - R_l \quad (3.26)$$

3.5 Fabrication of the optimized biogas stove burner

3.5.1 Materials used

Materials used for the fabrication of a biogas burner are cast brass, flat bar, fibre wood for legs, stainless steel and noncorrosive paint. Brass scraps and brass chips which were collected from metal scrapers at SIDO industrial area and other machine shops, in Arusha, Tanzania. Copper, mild steel and stainless steel materials were purchased from local hardware in Arusha.

3.5.2 Fabrication of a stove

Fabrication of the prototype was done in the workshops of CAMARTEC and ATC in Arusha, Tanzania. The stove parameters were obtained from the analytical design calculations. Tripod and stove frame were made by using mild steel and stainless steel respectively. Tripod was painted with corrosion-resistant materials to avoid corrosion during the lifetime and the gas burner was fabricated by using cast brass.

3.5.3 Prototype and physical layout

Figure 10 shows the biogas burner stove assembly named as developed burner which consists of seven main parts like frame, burner tripod, legs, gas pipe supply, knob, mixing tube and burner head. After designing all parts and components of the biogas burner stove, it is worthwhile to design the frame. The stove frame fabricated from standard stainless steel sheet good corrosion resistance and easy to bend, forming any shape, folding and dismantling while bolts and nuts are mainly used for joining. Welding, however, is done where necessary. The detailed drawing of each component is presented in Appendix 1-12.

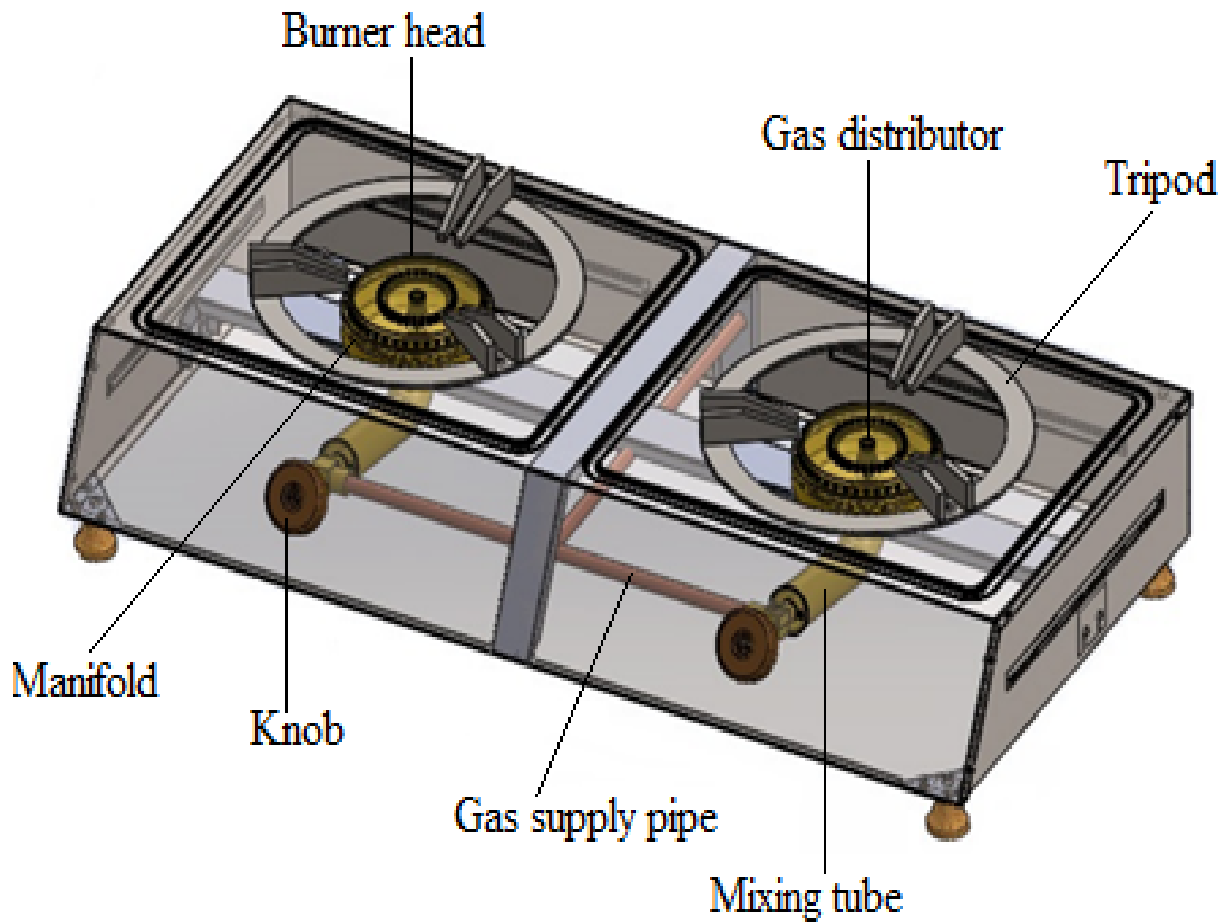


Figure 10: Biogas burner stove assembly

3.6 Evaluation of the performance of the cookstoves

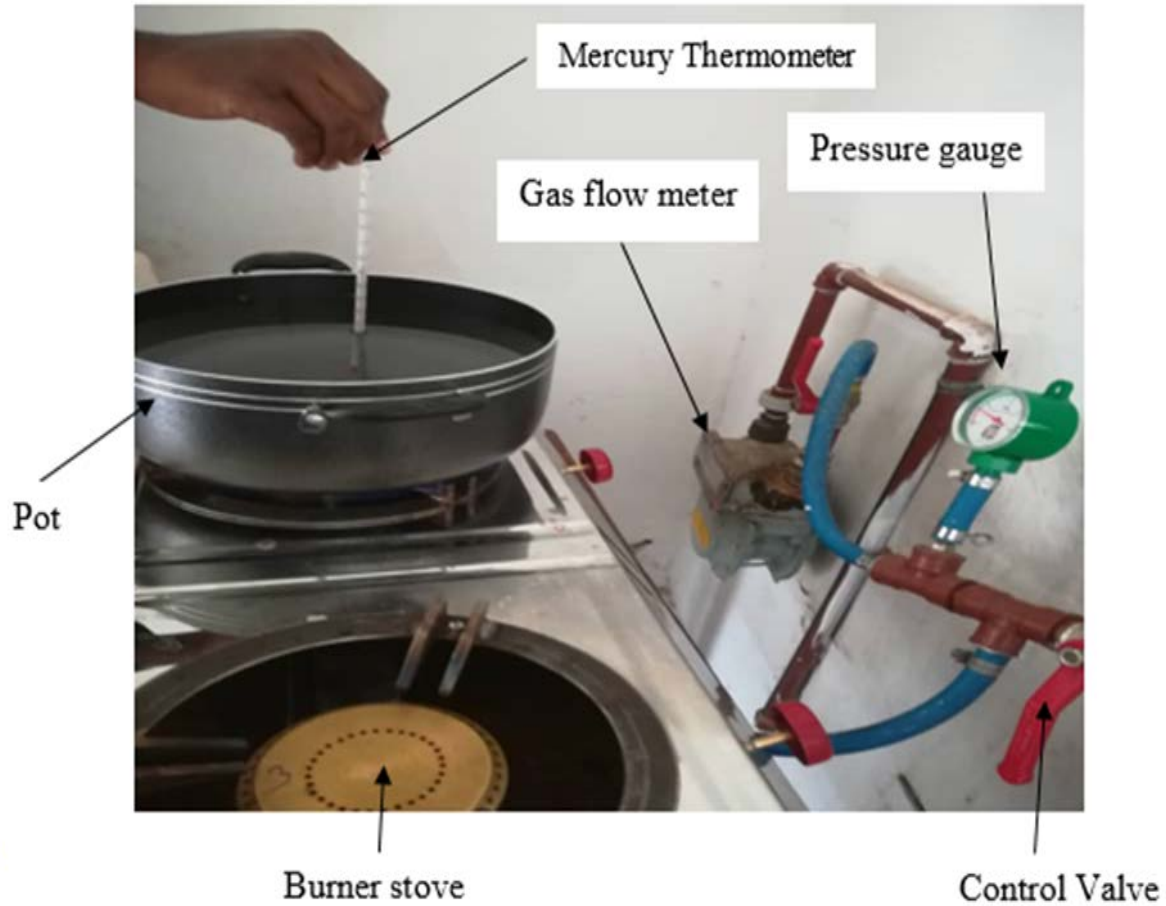
Experiments to determine the performance of the cook stoves were carried out in the energy lab at CAMARTEC in Arusha, Tanzania. The area is located at 1415 m above sea level. One window and a door were left open during experiments for good ventilation. The performance of the developed burner stove was compared with that of Simgas and CAMARTEC burner stoves. The experiments were designed to simulate real-life cooking processes and were conducted using the WBT software version 3.0 (Bailis *et al.*, 2007).

3.6.1 Experimental set-up and water boiling test procedure

The flow rate of biogas was measured before entering the mixing tube. After burning of biogas, the temperature was measured. A valve was used to control a stream of biogas. A pot of aluminium filled up with water was kept on the stove. Initial temperature (T_i) of water was measured by using a mercury thermometer. After time t , final temperature (T_f) of water was measured as shown in Fig. 11 and WBT procedure shown in Appendix 16. The apparatus required for the experiments included the following: Aluminium pot, mercury thermometer

(maximum capacity of 100 °C), electronic watch, infrared thermocouple and weighing balance.

Figure 11: Experimental setup (Camera)



3.6.2 The water boiling test (WBT)

Water boiling test has three phases namely; the cold start high power (CSHP), hot start high power (HSHP) and low power (simmering). The WBT was chosen in this study because of its simplicity and reliability. A standard 2.5 liters aluminium pot without a cover and weighing 0.549 kg was used in all the three experiments. The boiling test was assessed to determine the time to boil (Δt), specific fuel consumption (Sc), thermal efficiency (η_{th}), burning rate (r_b) and firepower (FP) of three stoves were compared. This involved measuring the burner capacity to move from high heating to low heating, and each experiment was repeated three times. All three burner stoves were tested at the same time on different days and average results were recorded (Berrueta, Edwards & Masera, 2008).

(i) Determination of local boiling point

The altitude above the sea level at CAMARTEC is 1415 m and this translates to the local boiling point of 95.23 °C according to Equation 3.28 (Bailis *et al.*, 2007).

$$T_b = \left(100 - \frac{h}{300}\right) ^\circ\text{C} \quad (3.27)$$

Where T_b the boiling point (Celsius) and h is the altitude of an area (m)

(ii) Phase 1; cold start high power (CSHP)

This is the primary stage of WBT where water at room temperature (23 °C) is heated up to a local boiling temperature (95.23 °C). The ambient temperature was measured with a mercury thermometer. The 2.5 litres of water was poured into an aluminium pot weighing 0.549 kg and the temperature of the water inside the pot was monitored by a mercury thermometer. The thermometer was suspended at the center of the pot by hand-holding, and the gap between the bottom surface of the pot and thermometer tip was 5 mm as recommended (Bailis *et al.*, 2007). Biogas fuel was utilized to heat the water from the initial temperature (T_i) to the local boiling point. After that, the water was left on the stove for around 5 min before the stove was switched off. The amount of water remaining after the test, final water temperature (T_f) and time to boil, Δt were recorded.

(iii) Phase 2: hot start high power (HSHP)

The hot start high power phase includes determination of fuel demand and time required by a cookstove, while still hot to heat a pot of water to the local boiling point of 95.23 °C. Similar steps as in phase 1 were followed, the only difference being that the stove was above ambient temperature when it was started after boiling, the hot water was not discarded but held for the third phase. Both temperature and time used in phase 2 were recorded.

(iv) Phase 3: low power (Simmering)

The simmering test was done after the HSHP test and included measuring the stove's capacity to move from high to low power. The temperature of the water was kept 3-6 °C below the boiling point for 45 min. This stage was utilized to simulate the long time required for cooking local food. At the end of the experiment, the remaining hot water was weighed.

(v) Time to boil water, (Δt)

The time to boil, $\Delta t = t_f - t_i$

Where

t_f : End time for test (min),

t_i : Start time for test (min)

This parameter was important for high power cold start phase (HPCSP) and high power hot start phase (HPHSP). For low power (simmering), the test was performed over a set time of 45 min.

(vi) Thermal efficiency, η_t

Thermal efficiency was calculated by the ratio of the energy used during heating and vaporizing water to the energy consumed by burning the biogas fuel (Demissie, Ramayya & Nega, 2016a).

$$\text{Thermal efficiency, } \eta_t = \frac{((C_{pw} \times M_w) \times (T_f - T_i)) + (H_v \times W_v)}{C_{v \text{ of biogas}} \times V_{bg}} \times 100 \quad (3.28)$$

Where,

V_{bg} : Volume of biogas consumed, m³/h

M_w : Mass of water boiled, kg

C_{vbg} : Calorific value of fuel (biogas) = 22 MJ/kg

C_{pw} : Specific heat capacity of water = 4186 J/kg °C

T_f : Boiling temperature of water = 95.23 °C

W_v : Water evaporated from the pot

T_i : The initial temperature of the water, °C

H_v : Latent heat of evaporation of water, 2260 J/g

(vii) Specific fuel consumption (SC)

Specific fuel consumption is the proportion of the sum of biogas fuel consumed to the sum of water remaining within the pot at the end of the test. Specific fuel consumption was calculated using Equation 3.30 (Bailis *et al.*, 2007).

$$SC = \frac{V_{bg}}{W_r} \quad (3.29)$$

where,

SC: Specific fuel consumption of the stove (g/L),

V_{bg}: Volume of biogas fuel expended during the phase of the test (g) and

Wr: The mass of water remaining at the end of the test (g)

(viii) Firepower (FP)

Firepower is the proportion of fuel energy utilization by stove per unit time. It was calculated by Equation 3.30 (Bailis *et al.*, 2007).

$$FP = \frac{V_{bg} \times LHV}{60 \times (t_f - t_i)} \quad (3.30)$$

Where,

FP: Power of stove in watts,

V_{bg}: Volume of biogas consumed during the stage of the test (m³/h)

LHV: Lower heat value of the biogas (kJ/kg)

t_f: Time at the end of the test (min)

t_i: Time at the start of the test (min)

(ix) Burning rate

Burning rate (r_b) is used to measure the rate of fuel consumption during the experiment.

$$r_b = \frac{v_{bg}}{\Delta t} \quad (3.31)$$

Where,

V_{bg} is the Volume of biogas consumed during the test (m³/h),

Δt is the total time taken to conduct the test (min),

(x) Temperature corrected time to boil (Δt^T)

This adjusts the results to a standard 75°C temperature change (from 25-100 °C). This adjustment standardizes the results and facilitates comparison between tests that may have used water with higher or lower initial temperatures. It's calculated by using the Equation below.

$$\Delta t^T = \Delta t \times \frac{75}{T_{1f} - T_{1i}} \quad (3.32)$$

Where Δt^T temperature corrected time to boil (min) is, Δt is time to boil (min), T_{1f} is the water temperature at the end of the test (°C), T_{1i} is the water temperature at the start of the test (°C).

(xi) Temperature corrected specific fuel consumption

Corrects specific fuel consumption to account for differences in initial water temperatures. This facilitates comparison of stoves tested on different days or in different environmental conditions. The correction is a simple factor that “normalizes” the temperature change observed in test conditions to a “standard” temperature change of 75°C (from 25 to 100). It was calculated using Equation 3.33:

$$SC^T = SC \times \frac{75}{T1_f - T1_i} \quad (3.33)$$

Where SC^T is the temp-corrected specific fuel consumption (grams of fuel/grams of water), SC is the specific fuel consumption (grams of fuel/liter of water), $T1_f$ is the water temperature at the end of the test (°C), and $T1_i$ is the water temperature at the start of the test (°C).

(xii) Temperature corrected specific energy consumption

This metric is a measure of the amount of fuel energy required to produce one litre (or kilogram) of boiling water starting with a cold stove. It is the temperature corrected specific fuel consumption multiplied by the energy content of the fuel. It's calculated by using Equation 3.34:

$$SE^T = SC^T \times \frac{LHV}{1000} \quad (3.34)$$

Where SE^T is the temperature corrected specific energy consumption (kJ/litre), SC^T is the temp- corrected specific fuel consumption (grams of fuel/grams of water), and LHV is the lower heating value of the fuel also known as net calorific value (kJ/kg) (Bailis *et al.*, 2007).

(xiv) Turn down ratio (TDR)

This is the ratio of average high firepower to average low firepower. It represents the degree to which the firepower of the stove can be controlled by the user. It's calculated using Equation 3.35:

$$TDR = \frac{FP_c}{FP_s} \quad (3.35)$$

Where,

FP_c and FP_s : Firepower (W) during cold and simmering phases respectively (Bailis *et al.*, 2007).

(xv) Overall efficiency and thermal efficiency

Overall efficiency was calculated by dividing output energy by input energy as shown in Equation 3.37 (Demissie, Ramayya & Nega, 2016a). The heat energy absorbed by the vessel is also considered.

$$Efficiency_{overall} = \left(\frac{(M_w \times C_{pw}(T_b - T_i)) + (M_v \times C_{pv}(T_b - T_i))}{V_{bg} \times C_{vf}} \right) \times 100 \quad (3.36)$$

Where,

C_{pw} : Specific heat capacity of water = 4186 J/kg °C

M_w : Mass of water boiled, kg

C_{vf} : Calorific value of fuel (biogas) = 22 MJ/kg or 22 MJ/m³

M_v : Mass of the vessel, kg

V_{bg} : Volume of biogas consumed, m³/h

T_i : The Initial temperature of the water, °C

C_{pv} : Specific heat capacity of aluminium vessel = 125 J/kg °C

T_b : Boiling temperature of water

CHAPTER FOUR

RESULTS AND DISCUSSION

4.1 Results of analytical design

The procedure and analytical design of the burner included the findings of various important parameters including throat diameter, length of mixing chamber, injector jet diameter, the diameter of primary air inlets, number and diameter of flame portholes and also the diameter and height of manifold. Design calculations using various formulas were applied in obtaining the different burner parameters presented in Table 6. The design of the parameters between analytical design calculation and specification values were compared well. In the comparison, there are some differences which have been observed such as biogas flow rate, throat and mixing tube length.

Table 6: The results of the analytical design calculation

S/N	Parameters	Analytical value	Specification values
1	Biogas flow rate (m^3h^{-1})	0.6398	0.648
2	Injector diameter (mm)	2.5	2.5
3	Throat diameter (mm)	18	25
4	Number of burner ports	71	71
5	Mixing tube length (mm)	180	138
6	Internal manifold diameter (mm)	100	100
7	Diameter of burner port (mm)	2.5	2.5
8	Primary air inlet area (mm^2)	25.13	25.13

4.2 Computational fluid dynamic modeling results

This Section presents results obtained from the analysis where CFD was employed to simulate the flow in the burner. During the simulation of the model, the convergence was reached easily when a set of Equations were solved and when the calculations were repeated several times until the flow and the temperature converged. The default criterion for convergence is that each residual was reduced to a value less than 10^{-3} except the energy residual for which the default criterion was 10^6 . Figure 12, shows the contours of temperature distribution in the burner. It is observed that the temperature distribution along the jet and mixing chamber is the same as the room temperature of the air and gas inlets in the boundary

conditions respectively. The temperature was identical throughout the mixing chamber and increased to a maximum of 997 K at the throat up to the upper Section of a manifold. The temperature in the middle Section of the manifold was 791 K and it was uniform throughout. Figure 13, indicates the distribution of temperature along the y-axis of the burner, the temperature increased in a circumference upwards from bottom to the upper part of a manifold.

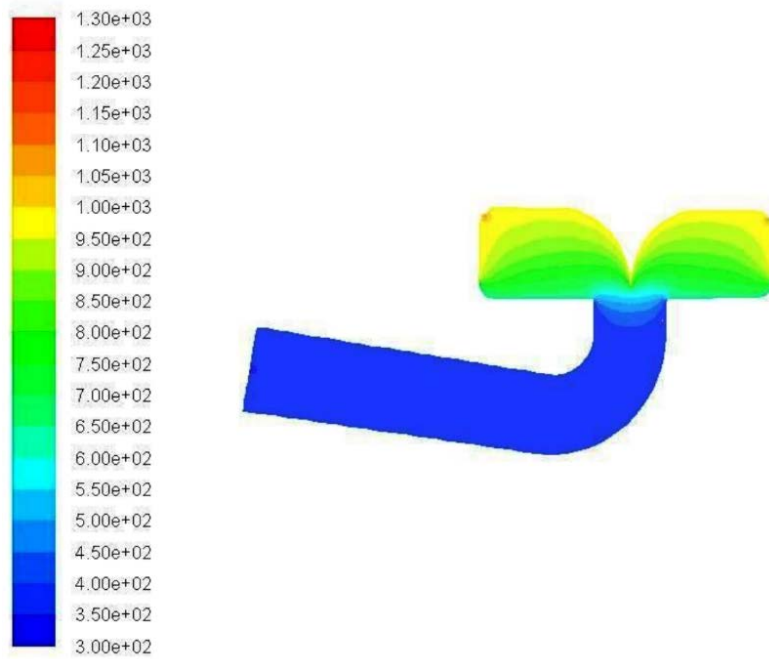


Figure 12: Contours of Static temperature along the burner

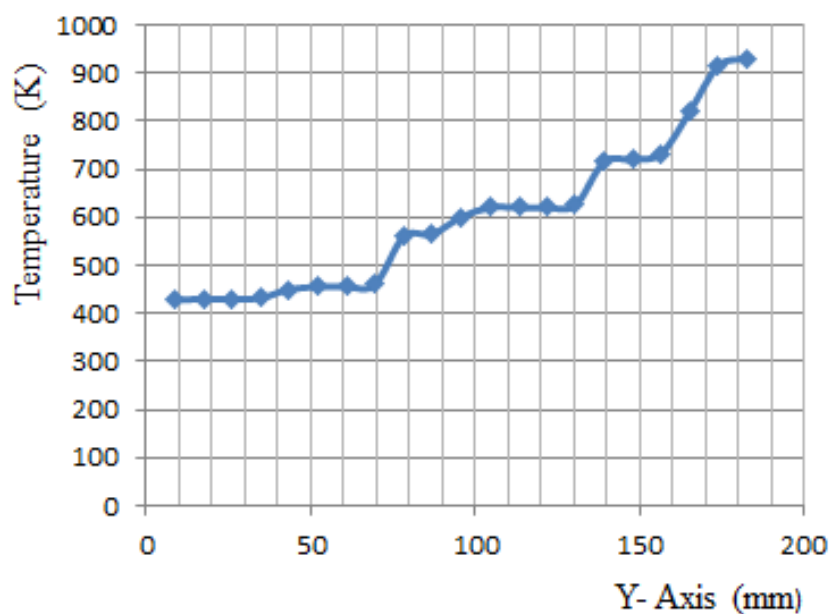


Figure 13: Distribution of temperature with position along the burner

Velocity distribution of fuel and air along the mixing chamber of the burner is shown in Fig. 14. The fuel comprises CH_4 and CO_2 which played a vital role in the combustion process.

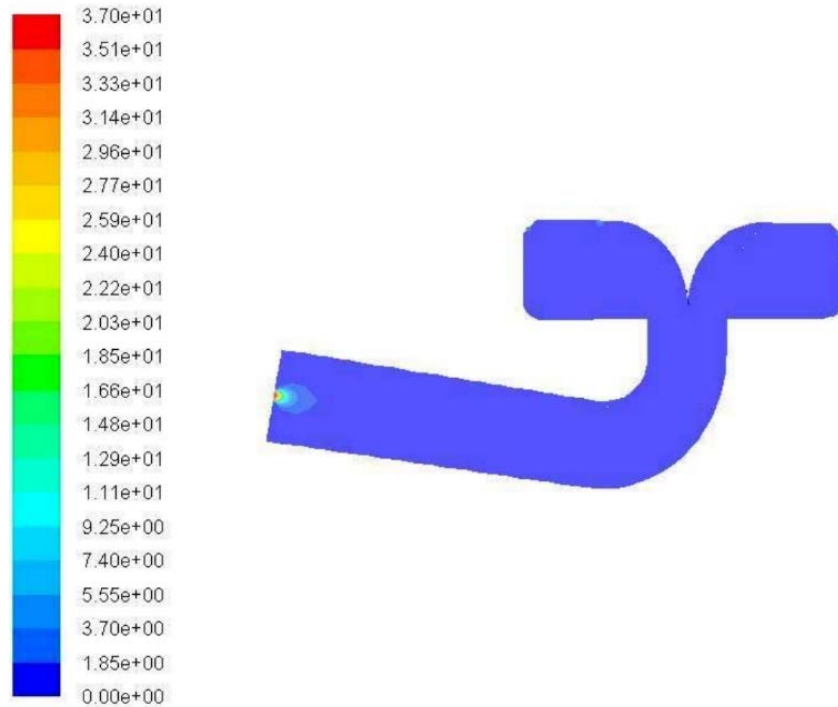


Figure 14: Contours of velocity distribution (m/s)

The velocity of the air and fuel was higher at the inlets and remained uniform at the mixing chamber and along the manifold as shown in Fig. 16. These findings are in agreement with other studies that involve experimental performance of biogas burners by application of mathematical model on key parameters including Decker (2017b) and Mulugeta, Demissie and Nega (2017). The maximum temperature distribution at a manifold in this research was 1270K (997 °C). The research done by Mulugeta, Demissie and Nega (2017) shows the maximum temperature distribution reached on the manifold was 1775K. Another research conducted by Noor, Wandel and Yusaf (2013) indicated the temperature reached was 537K, but this study was based on the preheating the fresh air through pipes and low oxygen dilution of a Moderate and intense low oxygen dilution (MILD) combustion burner. The mesh independent test was performed to verify and show that results are not dependent on mesh size. The influence of temperature change based on the mesh size is shown in Fig. 15. During the meshing process, the mesh size varied from 0.5 to 0.02 and in those various tests of mesh size 0.02 is considered to be appropriately consistent to ensure that temperature was independent of mesh size.

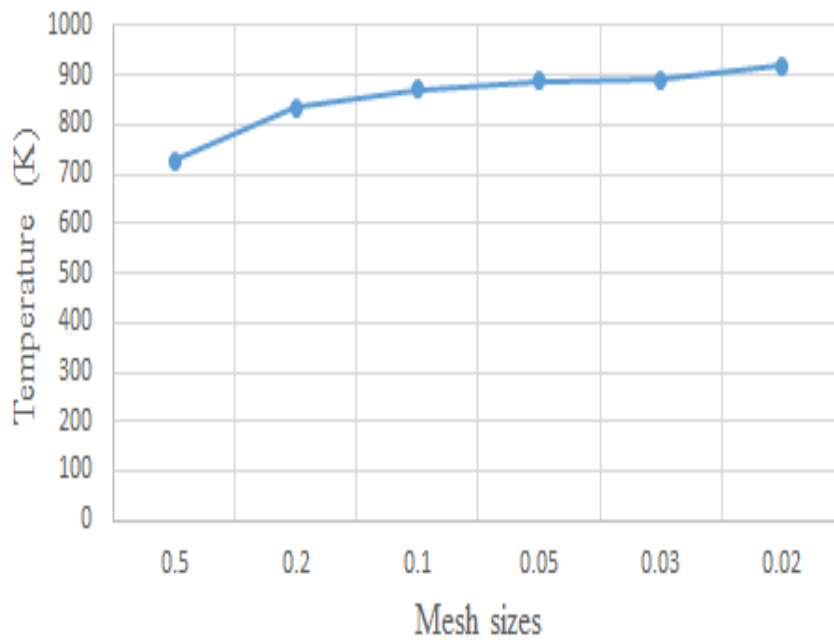


Figure 15: Influence of mesh size on Temperature

4.3 Results of experimental test

This Section presents test results and performance of biogas by deriving a comparative performance of developed cook stove with the Simgas and CAMARTEC cook stoves.

(i) High power cold start phase (HPCSP)

The developed burner took 7 min while Simgas and CAMARTEC burner took about 11 min respectively to boiling 2.5 litres of water to the boiling point of 95.23 °C (Fig. 18). The boiling temperature measured by the thermometer was approximately the same as the calculated local boiling point. The values of temperature corrected time to boil, temperature rectified specific fuel consumption for the three stoves are shown in Appendix 13, 14, and 15. The specific fuel consumption for Simgas and CAMARTEC stove was 0.51 g/L, and 0.92 g/L whereas thermal efficiencies performed was 9.94% and 5.58% respectively during HPCSP. While the developed stove was observed to have a boiling time of 7 min, the specific fuel consumption of 0.43 g/L and thermal efficiency of 11.63%. This is due to the good design of burner parameters which as a result kept the fire close to the cooking pot at most of the time during the test. Proper pot gap of about 30 mm was also maintained in this study.

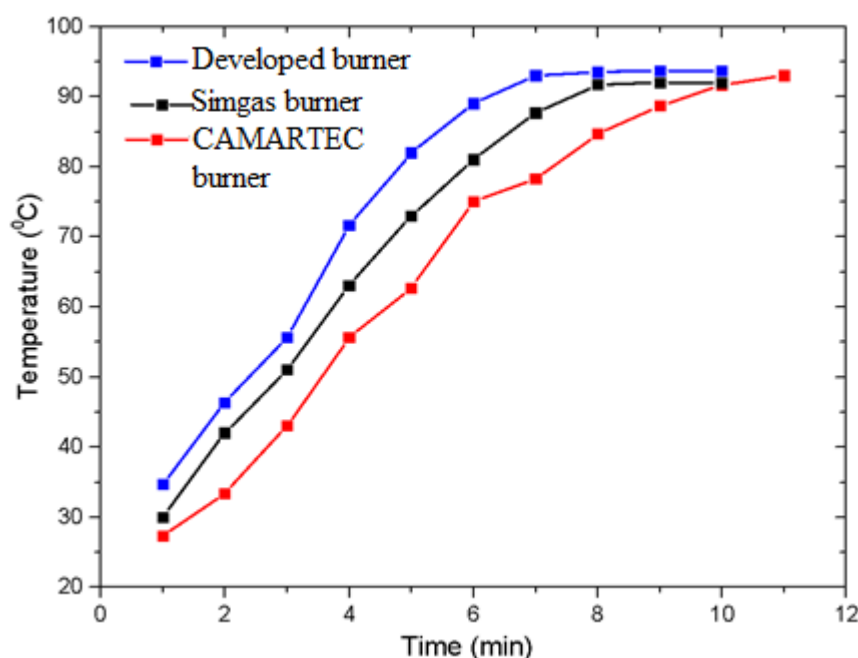


Figure 16: High Power cold start phase (HPCSP) water temperature profiles for Developed, Simgas and CAMARTEC stoves

(ii) High power hot start phase (HPHSP)

The hot start high power phase begins at the moment when the burner of the stove is hot, that is when the stove temperature is over room temperature. The method used for HPHSP was comparative to that of HPCSP only that the temperature of the stove was above the ambient temperature at the starting of the test. Figure 17, shows the plot of temperature versus time. The time taken by the CAMARTEC and Simgas burners to bring 2.5 litres of water to the local boiling point was 10 min. On the other hand, developed burner brings 2.5 litres of water to boiling point after 7 min. The maximum water temperature achieved for the developed stove was 93 °C compared to 95.3 °C of local boiling point. The time taken for developed and Simgas stove to boil water of 2.5 litres from ambient temperature of 25 °C to the local boiling point of 95.2 °C during HPHSP was 10 min compared to that of HPCSP. This is because in the HPHSP test, the stove temperature was over the room temperature and therefore, high heat energy had been retained by the stove body unlike with the HPCSP where some of the heat was taken up by the cooking pot.

The specific fuel consumption for developed and Simgas stove was 0.41 g/L, and 0.44 g/L whereas thermal efficiency performed was 11.38% and 12.27% respectively during HPHSP while the CAMARTEC stove had the highest specific fuel consumption i.e 0.63 g/L. This

implies that it consumes the most fuel, and its burning rate and firepower tends to be higher and thermal efficiency to be lower. Temperature redressed time to boil, temperature redressed specific fuel consumption, and temperature redressed specific energy consumption during HPHSP for the Developed, Simgas and CAMARTEC stoves are shown in Appendix 13, 14, and 15.

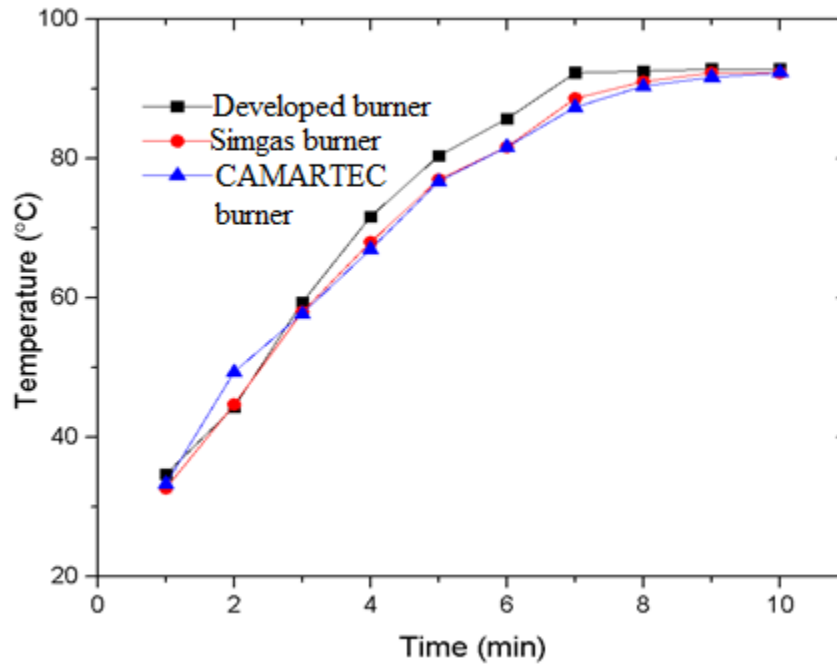


Figure 17: High power hot start phase (HPHSP) water temperature profiles for Developed, simgas and CAMARTEC burners

(iii) Low power simmering phase (LPSP)

This phase begins instantly after HPHSP, and it determines the capacity of a cookstove to move from the high to low heating. Low power simmering phase recreates the long cooking preparation and it is exceptionally significant within the assurance of the performance parameters such as firepower, specific fuel consumption and thermal efficiency. The test was conducted for 45 min utilizing the same hot water left during the HPHSP. The water temperature was kept up between 95.23 °C which is a local boiling temperature and 92 °C as recommended (Bailis *et al.*, 2007). When the experiment began, introductory water temperature and beginning time were measured and recorded together with the mass of biogas fuel and weight of remaining hot water. The temperature was maintained between 92 °C and 95.3 °C by adjusting the air opening utilizing burner regulator/shutter for Simgas whereas, for the developed stove, temperature was maintained within the recommended range

by adjusting both the air opening and fuel level utilizing burner air regulator/shutter and the fuel control valve respectively.

Since CAMARTEC burner could only reach to 78 °C for first testing instead of local boiling point which is 95.23 °C, simmering test was conducted by maintaining the water temperature between 92 °C –78 °C for 45 min. Figure 18, shows the water temperature profiles during simmering tests for all three stoves. The values of temperature redressed specific energy consumption during LPSP for the developed, Simgas, and CAMARTEC burners as shown in Appendix 13, 14 and 15.

Further, on the contrary, CAMARTEC burner, failed to bring the water to the local boiling point of 92 °C which conducted for 45 min but afforded to reach a maximum temperature of 78 °C while Simgas burner reaches a maximum temperature of 89 °C compared to 92 °C temperature afforded by the developed stove.

The outstanding performance afforded by the developed burner in low power start phase was qualified to maintain the temperature of 92°C for the long cooking process. The result indicates the good design procedures used for biogas burner designs were implemented by Fulford (1996). Developed burner ensured sufficient heat transfer to the cooking pot over the entire cooking process. This indicated that the developed burner is the best one to capable of cooking local food.

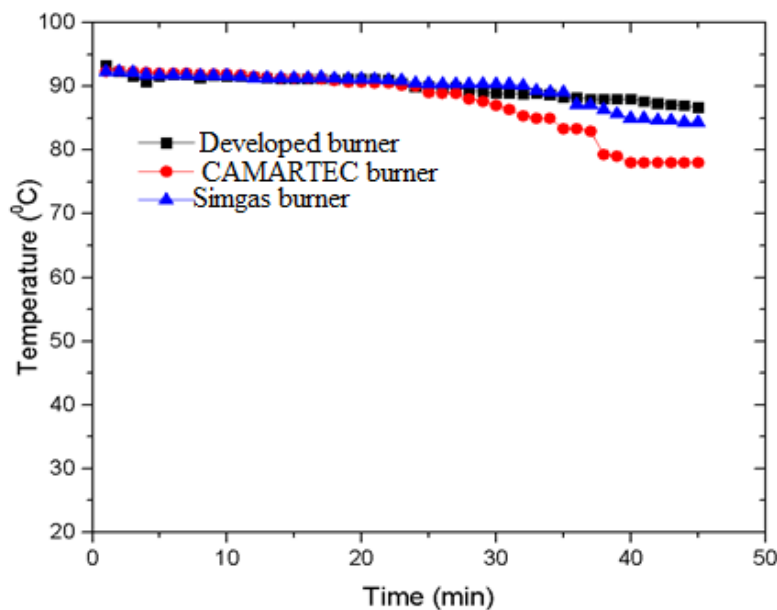


Figure 18: Low power simmering start phase (LPSP) water temperature profiles for the Developed, Simgas and CAMARTEC burners

4.3.1 Burning rate (rb)

Burning rate (r_b) are used to measure the rate of fuel consumption during heating of the water to a local boiling point. Figure 19, shows that the developed stove had the lowest burning rates compared to the other stoves tested. During the HPCSP, the developed stove burned fuel at a rate of 3.45 g/min and 3.26 g/min, in the cold and in hot phases while a burning rate of 8.18 g/min was achieved in the simmering phase. For Simgas stove, the average burning rates were 4.03 g/min, 3.55 g/min and 9.78 g/min in cold, hot, and simmering phases while the CAMARTEC burner appeared the highest average burning 7.28 g/min, 4.98 g/min and 10.22 g/min in cold, hot and simmering stage respectively. The overall burning rates for the whole water test were 4.96, 5.79 and 7.49 g/min for the developed, Simgas and CAMARTEC burners respectively.

The lower burning rates managed by developed stove was attributed to appropriate design of the stove, especially pot gap which minimized heat losses, and the adjustable primary air hole that ensured air mixing well to reduce flame lift or backfire to the mixing chamber. The high burning rate for CAMARTEC burner was likely due to the big flame ports diameter which consumes a large amount of biogas.

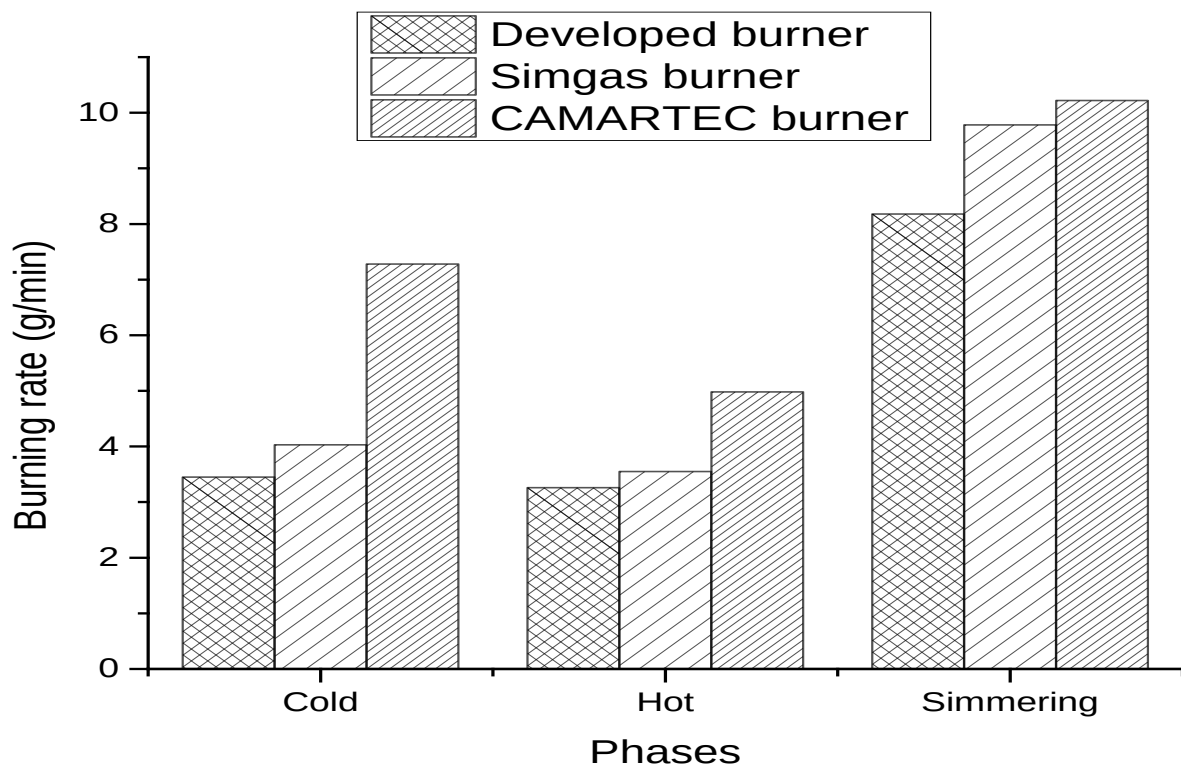


Figure 19: Burning rates of the Developed, Simgas and CAMARTEC burners

4.3.2 Thermal efficiency, η_{th}

Figure 20, shows the thermal efficiency of the three stoves tested. The developed burner performed the highest efficiency of 67.01% secondly by Simgas burner with 58.82% and lastly CAMARTEC burner with the thermal efficiency of 54.61% from simmer start. During simmering, the amount of fuel energy required to maintain water at 3°C below the boiling point was higher in the three stove types compared to boiling water for high power.

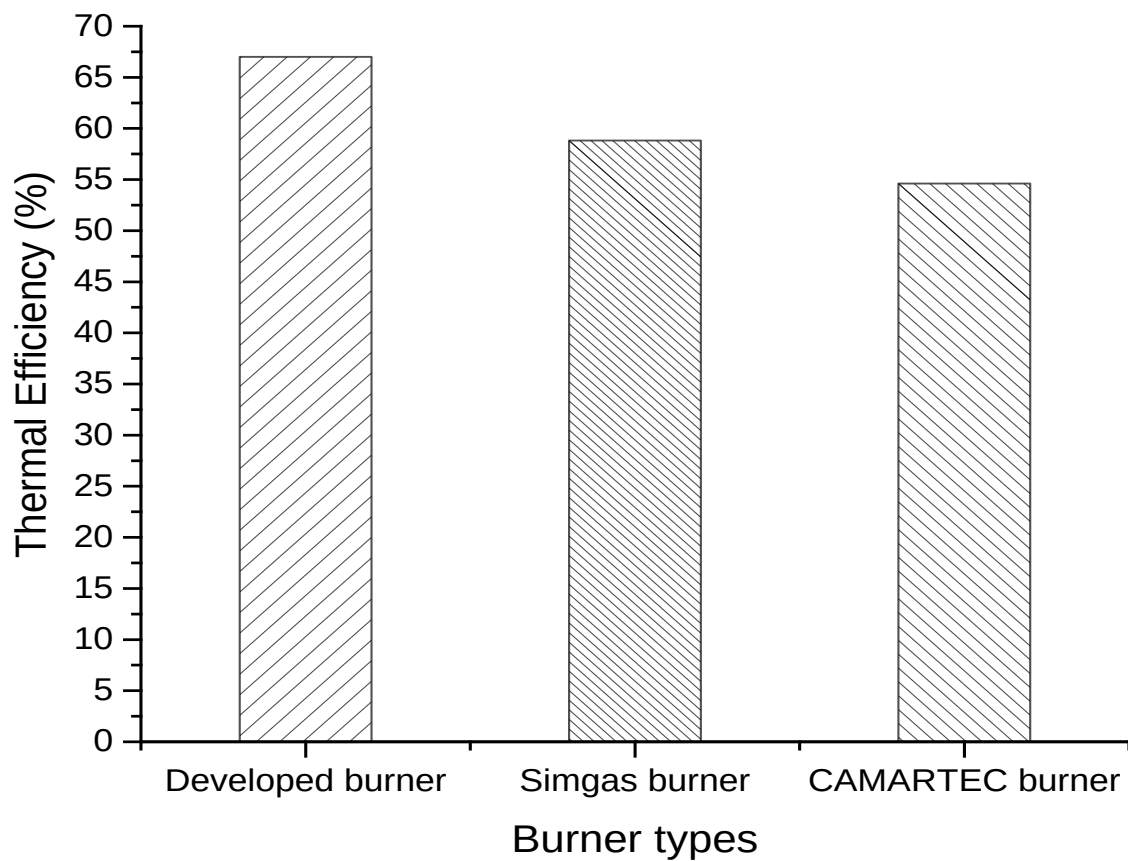


Figure 20: Thermal efficiency in the simmering phase of the Developed, Simgas and CAMARTEC burners

In a cook stove high thermal efficiency is accomplished when a stove uses a minimal amount of fuel to boil a given mass of water as was the case with the developed stove. This was attributed to the modification of the burner which improved parameters such as the diameter of the injector, throat size, throat diameter and height of burner head, to suit the cooking application.

Generally, it was noted that thermal efficiency of a stove depends on surrounding conditions, including wind speed, ambient temperature, specific heat capacity, bottom shape, weight, size of the vessel, and amount of specimen (Tumwesige, Fulford & Davidson, 2014). In this situation, different tests for efficiency could yield different results for the same stove.

4.3.3 Firepower (FP)

For the hot high power phase, the developed, Simgas and CAMARTEC stoves gave firepower of 17.31 watt, 18.6 watts and 24.56 watts respectively, whereas in low power the firepower for the individual stoves was 43.80 watt, 49.23 watts and 54.32 watts respectively as shown in Fig. 21. The contrast in firepower was related to the contrast in assignments the stoves were performing. On high power, the stoves boiled water faster with the fire regulator completely opened while on low power, the stoves simmered water for 45 min with the fire controlled to maintain a temperature range of 3°C to 6°C below boiling point.

On comparing the cold and hot start, the later gave more power output than the former. This is often due to greater fuel vaporization and superior airflow due to the heated stove body and surfaces. The overall firepower over the complete WBT test of the three stoves were 26.48, 29.76 and 39.19 watts, for the developed, Simgas and CAMARTEC burners respectively as shown in Fig. 22. The overall firepower of the developed burner was the lowest while that of the CAMARTEC was the highest. This was a result of the modification done on the burner parameters such as the diameter of the injector, throat size, throat diameter and distance between burner head top and bottom of pot. Miller-Lionberg (2011) reported that increasing the firepower of the stove decreases efficiency.

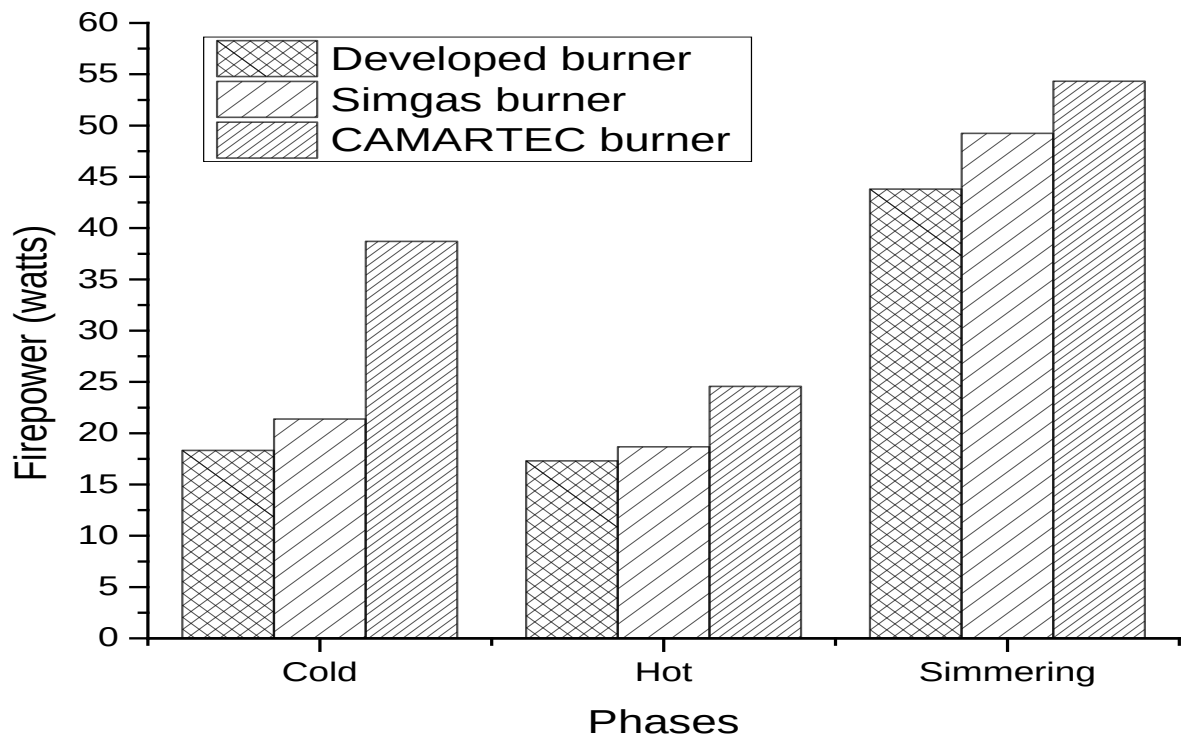


Figure 21: Firepower during cold, hot and simmering phases of the Developed, Simgas and CAMARTEC burners

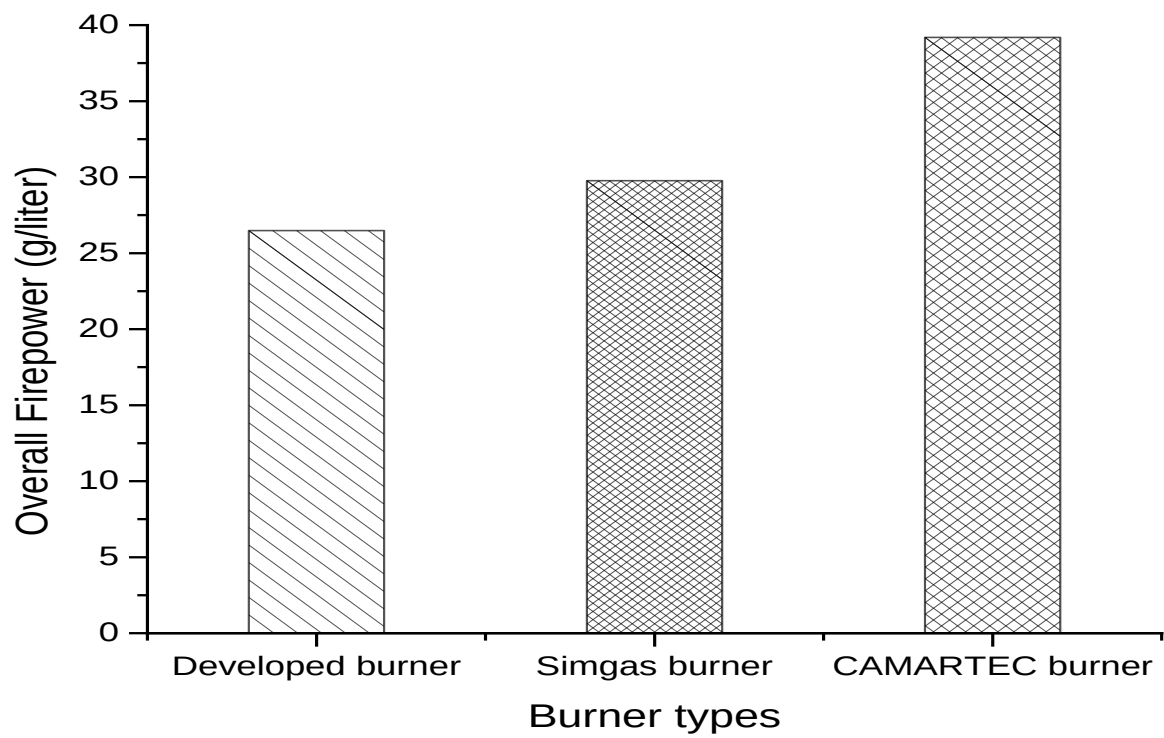


Figure 22: Overall firepower

4.3.4 Specific fuel consumption

Specific fuel consumption is the proportion of the amount of biogas fuel consumed to the amount of remaining water within the pot at the end of the test. The overall SC over the whole WBT for each stove was found to be 245.34, 277.68 and 306.68 g/L for the developed, Simgas and CAMARTEC burners respectively as shown in Fig. 23. The low SFC for developed burner resulted in higher thermal efficiency as shown in Fig. 20.

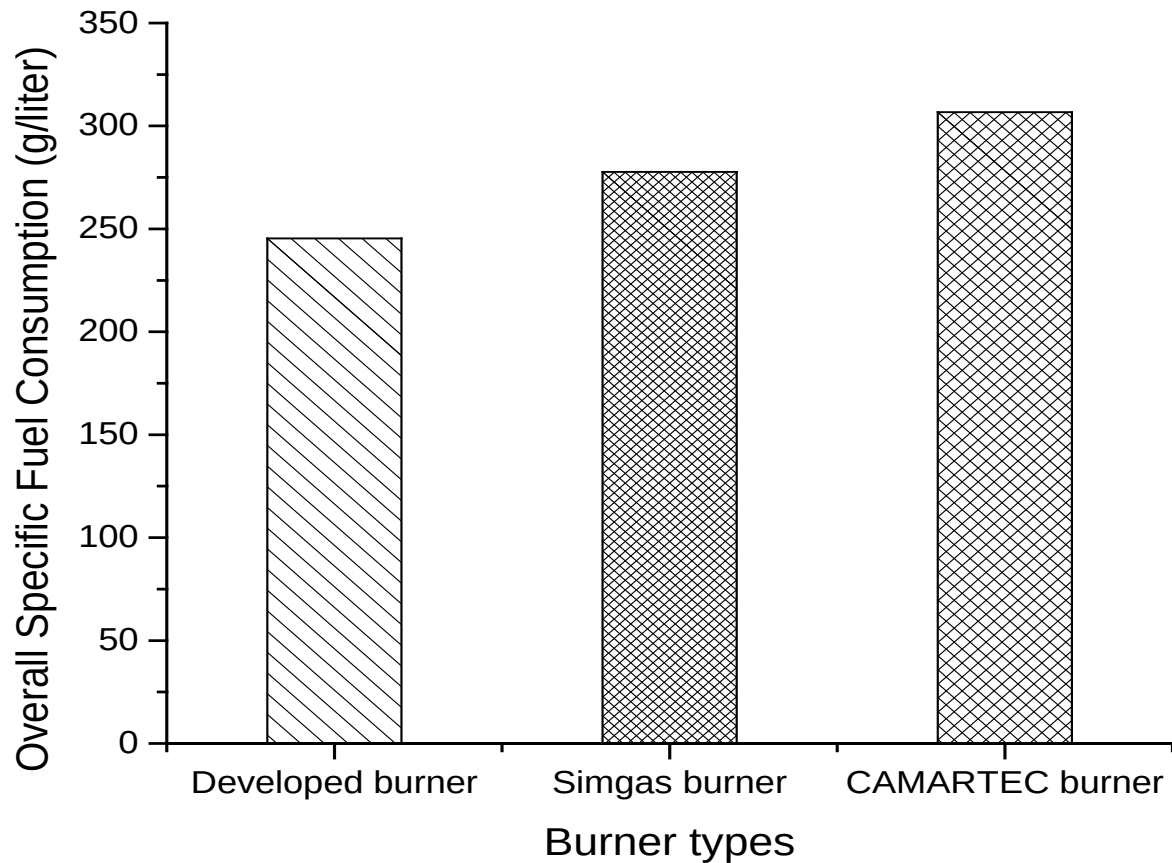


Figure 23: Overall specific fuel consumption over the entire WBT for the three burners

The developed burner stove shows a lower amount of fuel used compared to the Simgas and CAMARTEC burners. The rated fuel saved was higher in the cold start than in hot start with the variability of the fuel saved being inverse. The changes in the surrounding conditions affected the experiments and consequently the fuel utilization. In the hot start, the burner of the Simgas and CAMARTEC burners were at high-temperature values than in the cold start phase as shown in Fig. 24.

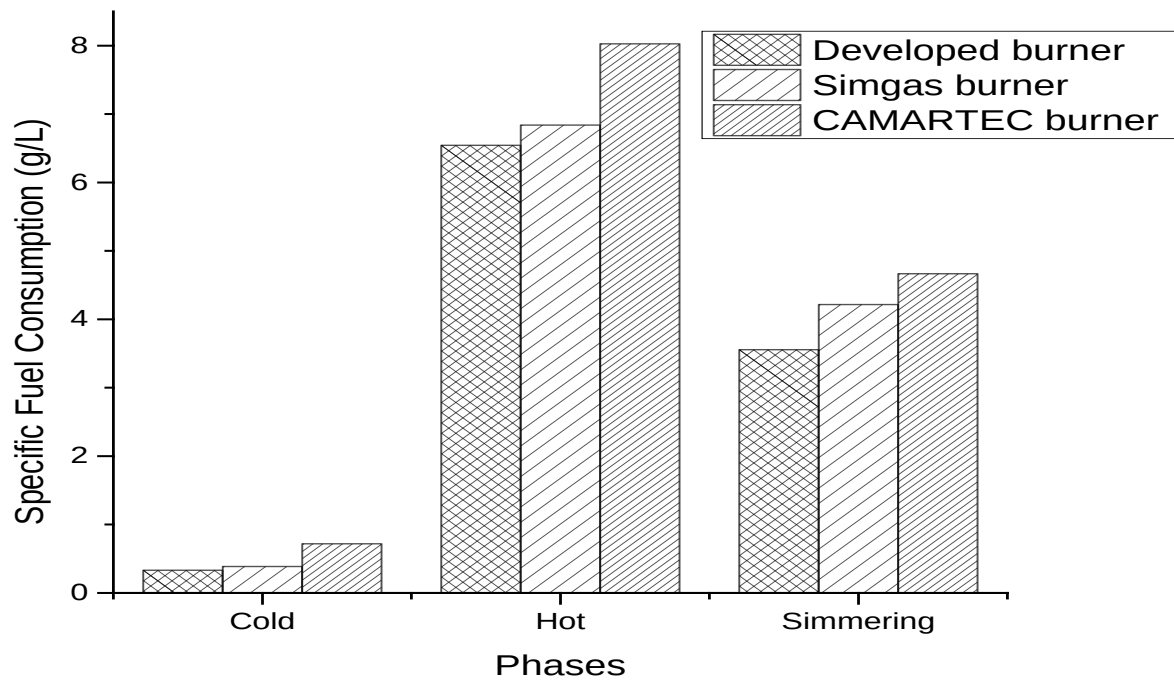


Figure 24: Specific fuel consumption during cold, hot and simmering phases of the Developed, Simgas and CAMARTEC burners

CHAPTER FIVE

CONCLUSION AND RECOMMENDATIONS

5.1 Conclusion

The outcomes of this study present perception in the modelling process of the newly designed burner and its performance. The design of the burner consisted of theoretical analysis, numerical design calculations and Computational fluid dynamics (CFD). As per the analytical design, several formulas were used to design the key parts of a burner. The developed burner head consists of 40 portholes located on its outer circumference having 2.5 mm diameter each, and 31 portholes with 2.5 mm diameter each at its top surface. The manifold of a developed burner has 100 mm internal and 138 mm external diameters respectively. The tripod and stove frame was made by using mild steel and stainless steel respectively. The tripod was painted with corrosion-resistant paint to avoid corrosion during the lifetime and the burner was fabricated by using casted brass. The simulation results show that the maximum flame temperature observed was 997K against 925K of the experimental measurements.

The water boiling test (WBT) was conducted for the developed burner, and its thermal efficiency performance was 67.01% against 54.61% and 58.82% of CARMATEC and Simgas burners respectively. The comparison for the average fuel consumption was also conducted between a developed burner and the above - mentioned burners. The test showed that the developed burner fuel consumption was 736 g/L compared to 920 g/L and 833 g/L for CARMARTEC and Simgas, respectively. The low fuel consumption observed from the developed burner as compared to others was due to the proper distribution of fuel on the cooking pot at most of the time during the test. In this study, the biogas stoves can be suggested as fetched successful options to the traditional stove.

5.2 Recommendations

Recommendations for future works are the following; assessment of improved Biogas Stove with preheating of biogas, CO₂, H₂S and moisture can be minimized by modifications of the biogas stove. The changes in the surrounding conditions affected the experiments and consequently, the fuel utilization. The improvement of efficiency of biogas stoves can be done in different areas of Tanzania.

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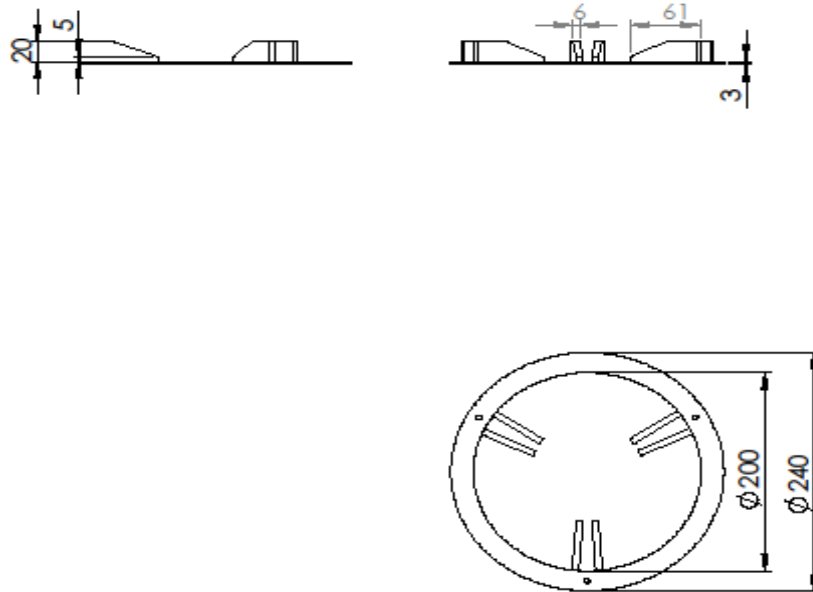
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APPENDICES

Detail Drawings of Biogas Burner Stove

The Biogas burner stove contains several components which are designed and drawn. The detail drawing of each component is presented in below Appendices to show dimensions, part list, quantity and name of the component. All these parts are completed assembly of the biogas burner stove.

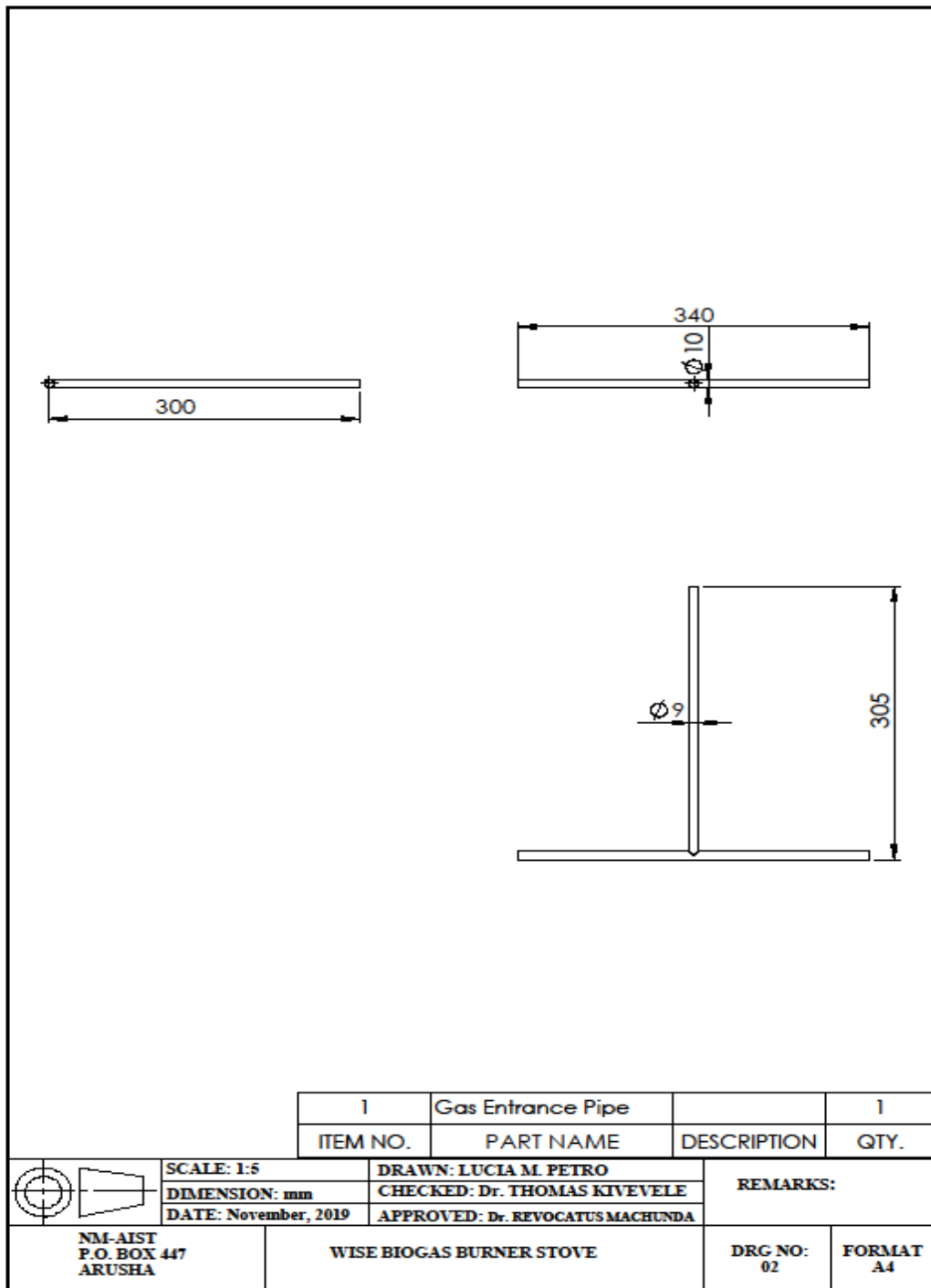
Appendix 1: Burner tripod



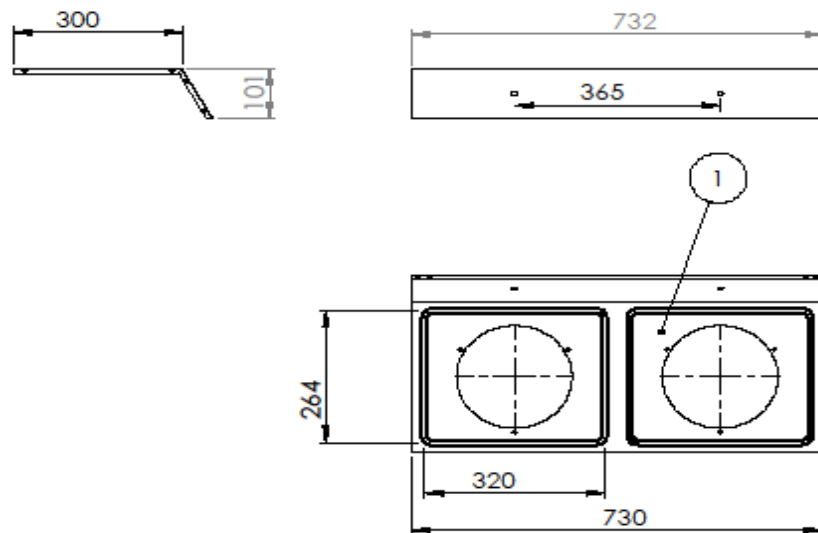
2	Tripod		6
1	Tripod Base		1
ITEM NO.	PART NAME	DESCRIPTION	QTY.

	SCALE: 1:5		DRAWN: LUCIA M. PETRO		REMARKS:		
	DIMENSION: mm		CHECKED: Dr. THOMAS KIVEVELE		Left & Right subassembly tripod stove		
	DATE: November, 2019		APPROVED: Dr. REVOCATUS MACHUNDA				
NM-AIST P.O. BOX 447 ARUSHA		WISE BIOGAS BURNER STOVE				DRG NO: 01	FORMAT A4

Appendix 2: Gas entrance pipe

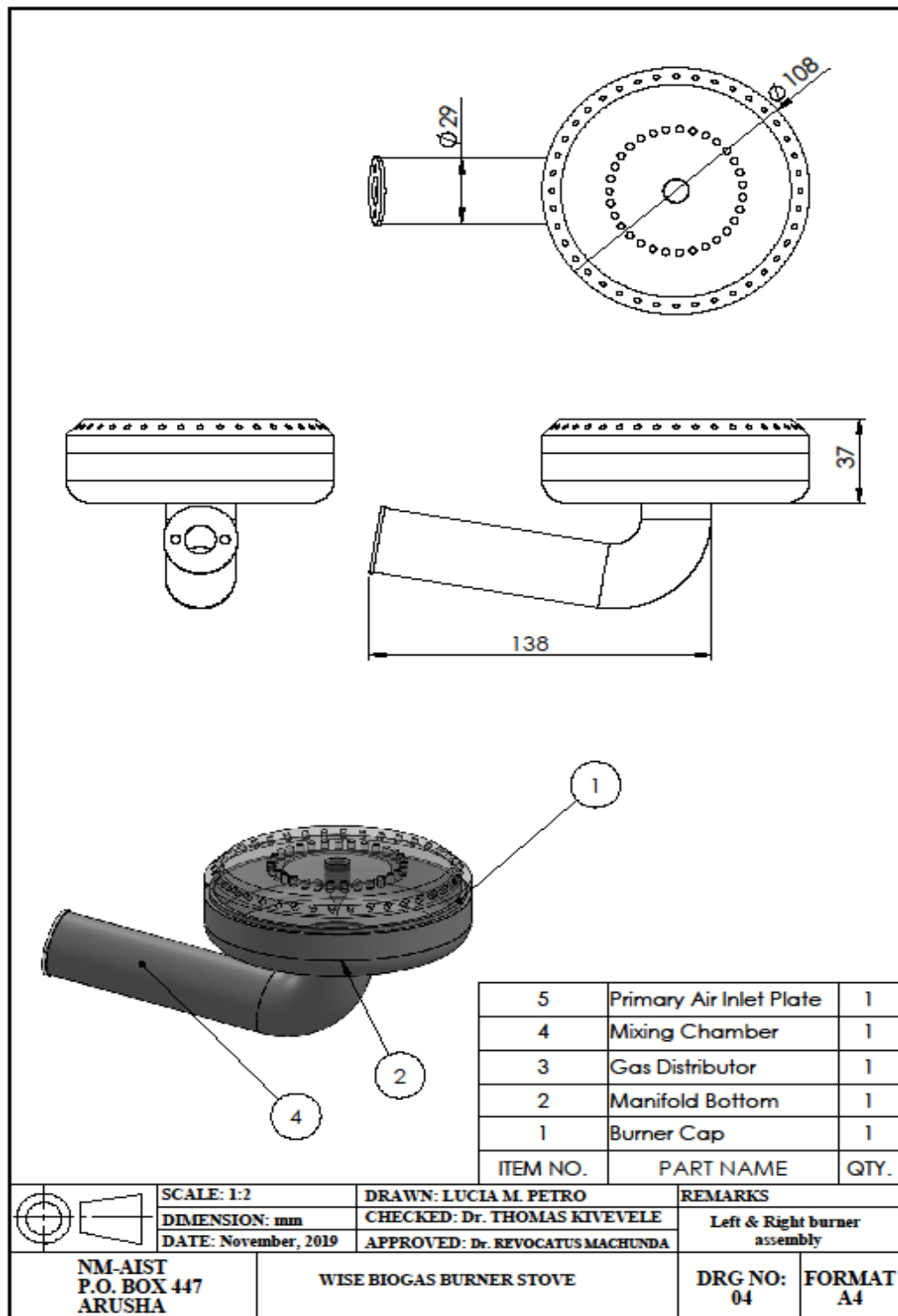


Appendix 3: Upper sheet

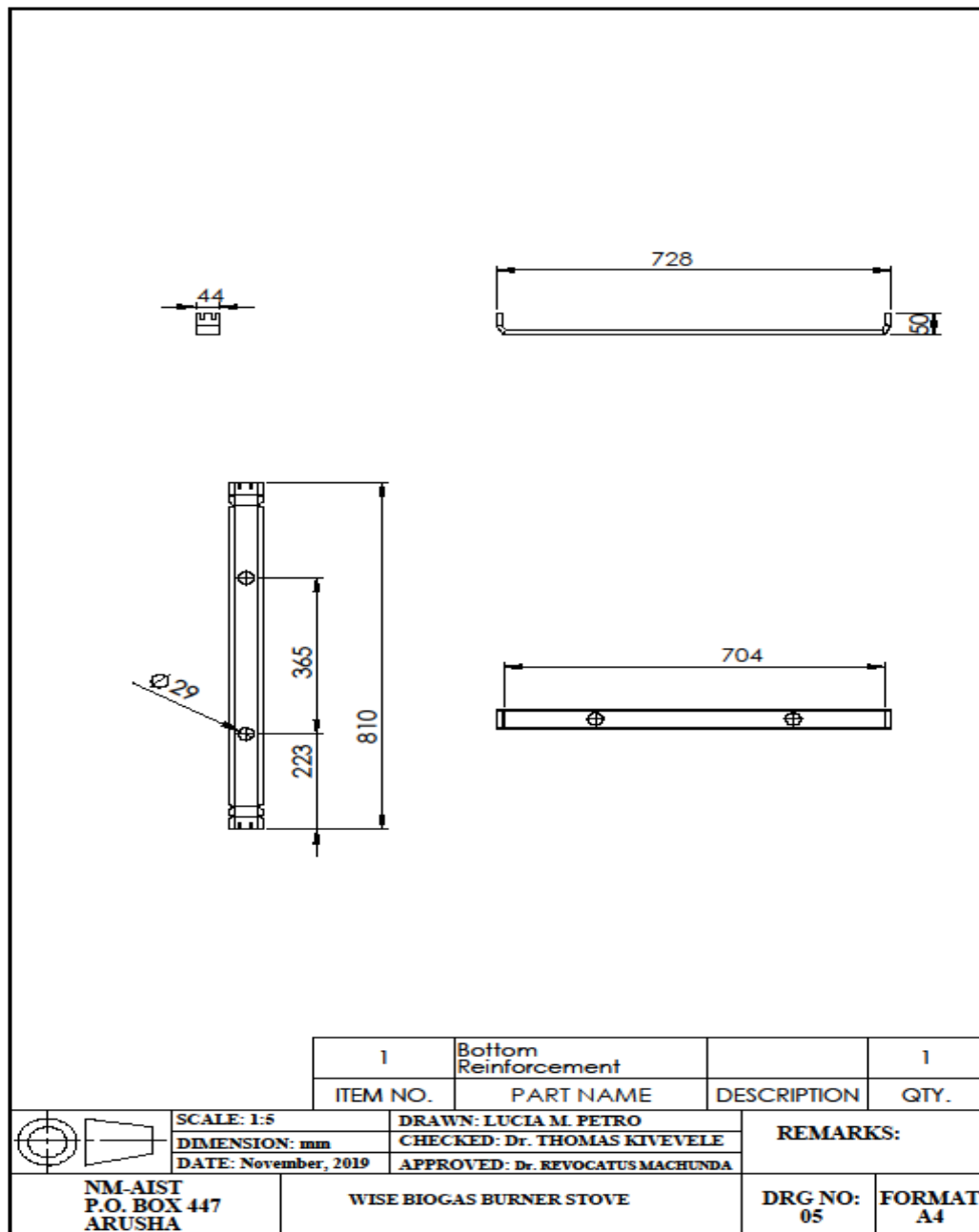


		1	Upper Sheet		1
		ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
	SCALE: 1:10		DRAWN: LUCIA M. PETRO		REMARKS:
	DIMENSION: mm		CHECKED: Dr. THOMAS KIVEVELE		
	DATE: November, 2019		APPROVED: Dr. REVOCATUS MACHUNDA		
NM-AIST P.O. BOX 447 ARUSHA		WISE BIOGAS BURNER STOVE			DRG NO: 03
					FORMAT A4

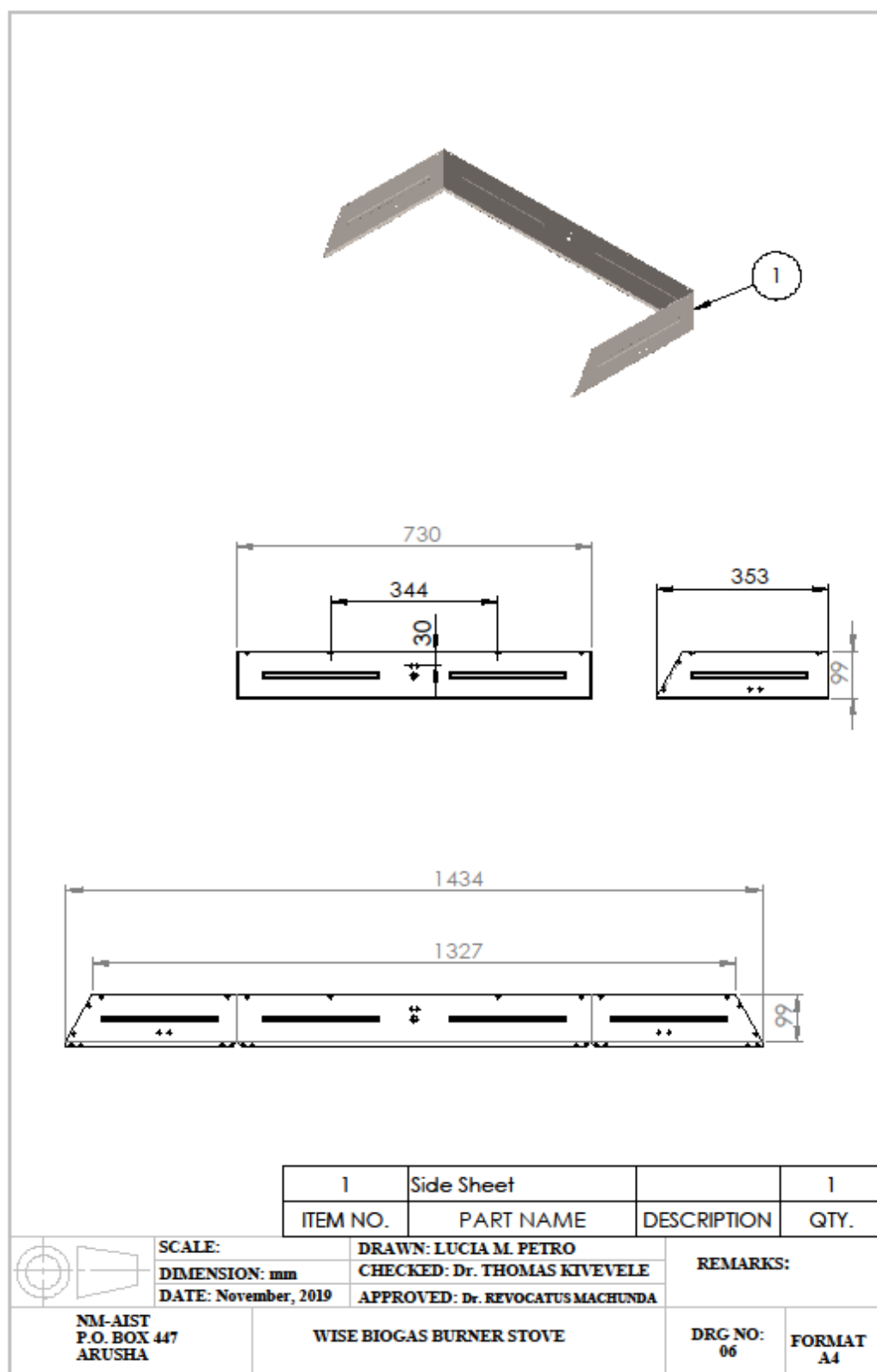
Appendix 4: Burner Assembly



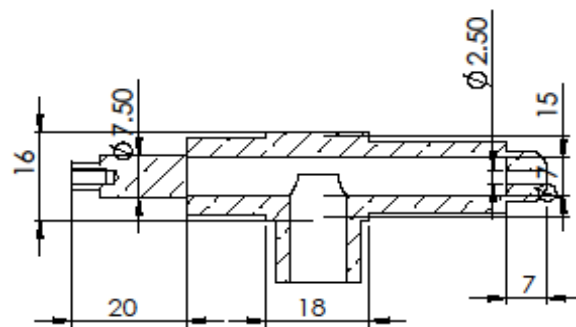
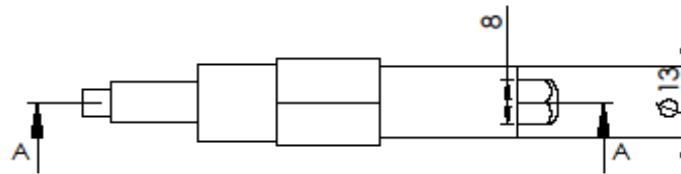
Appendix 5: Bottom Reinforcement



Appendix 6: Side Sheet



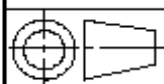
Appendix 7: Gas Flow Control Valve



SECTION A-A



1	Gas Flow Control Valve		1
ITEM NO.	PART NUMBER	DESCRIPTION	QTY.



SCALE: 1:1

DIMENSION: mm

DATE: November, 2019

DRAWN: LUCIA M. PETRO

CHECKED: Dr. THOMAS KIVEVELE

APPROVED: Dr. REVOCATUS MACHUNDA

REMARKS:

Left & Right gas flow control valve

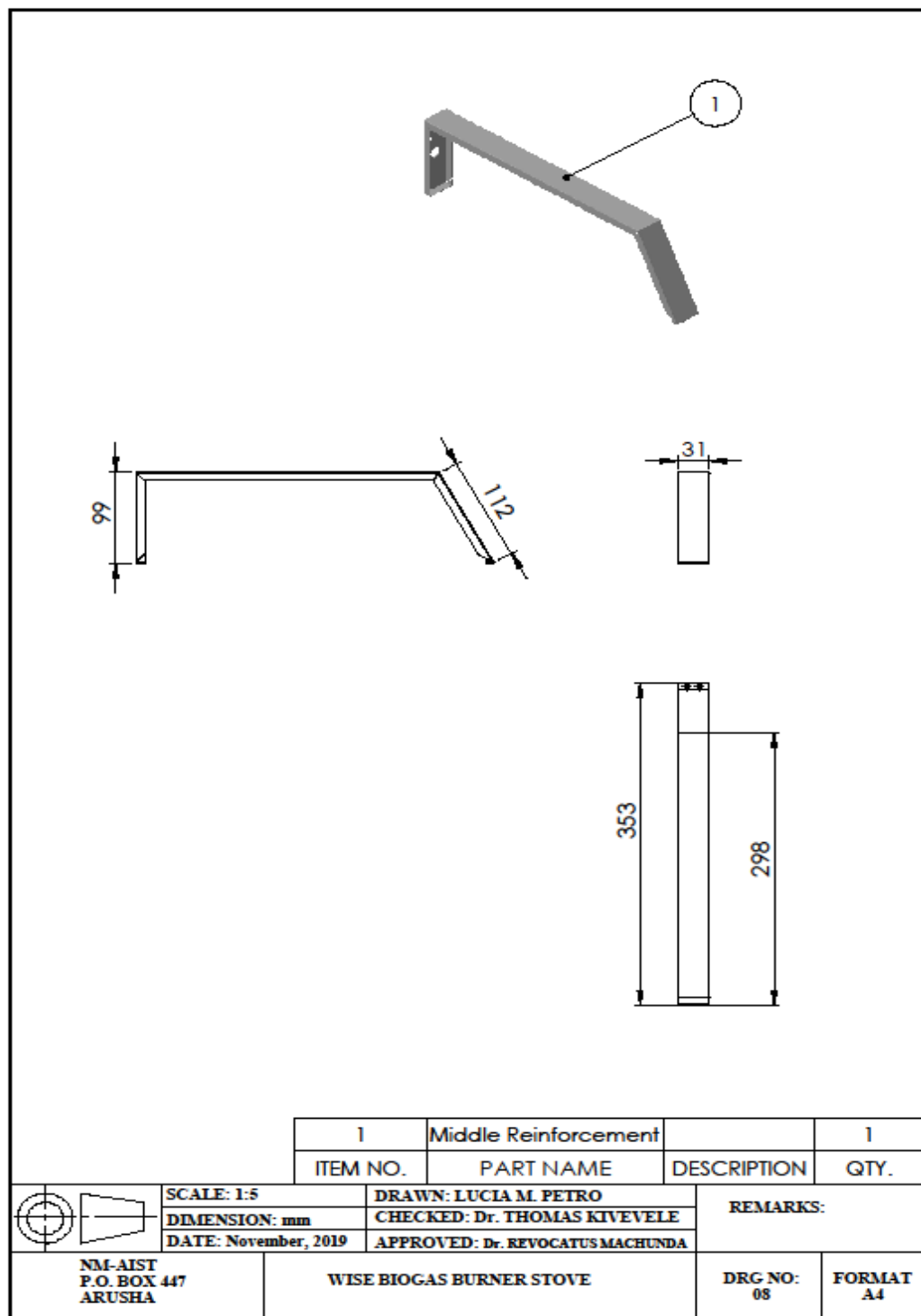
NM-AIST
P.O. BOX 447
ARUSHA

WISE BIOGAS BURNER STOVE

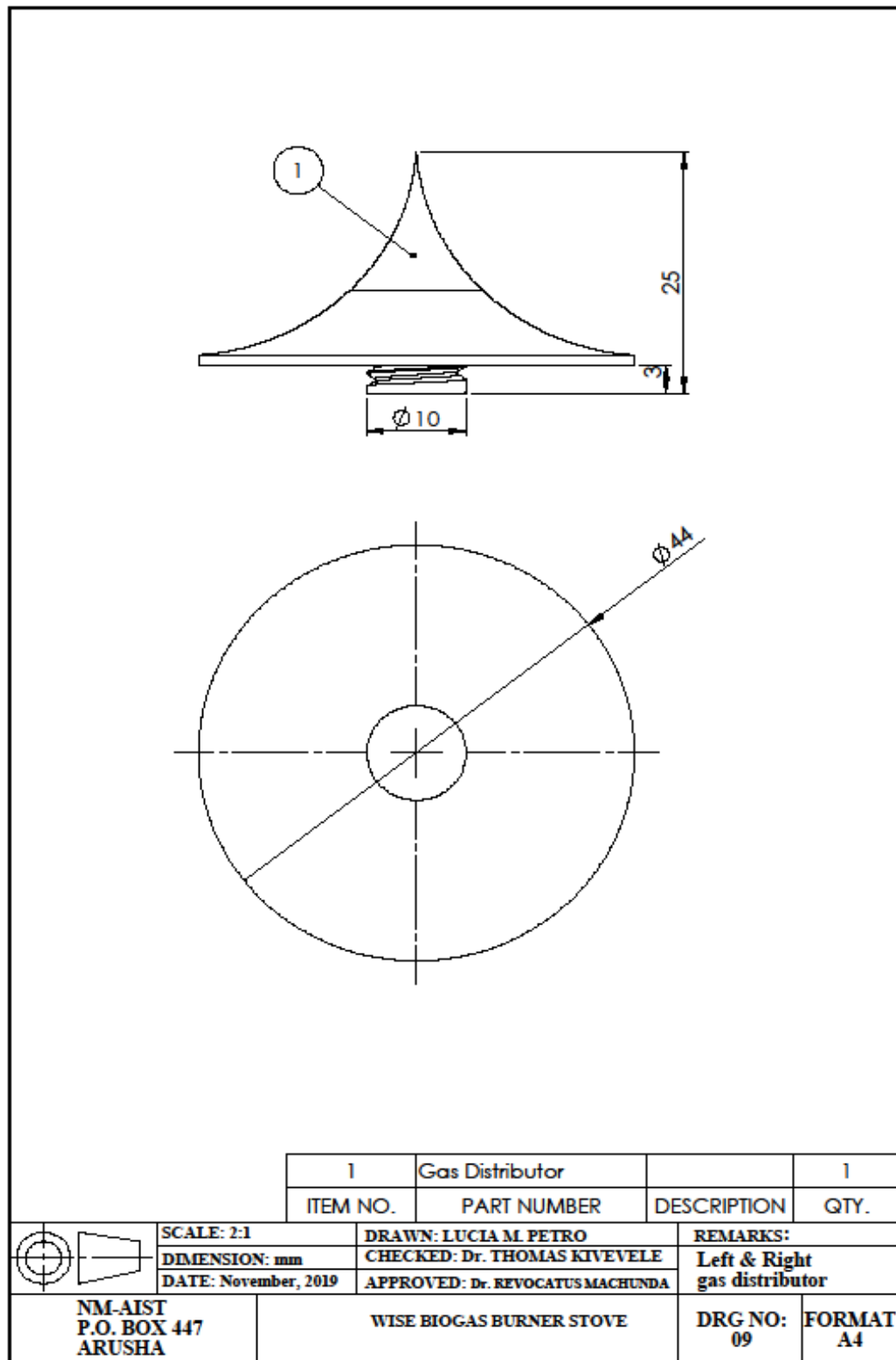
DRG NO:
07

FORMAT
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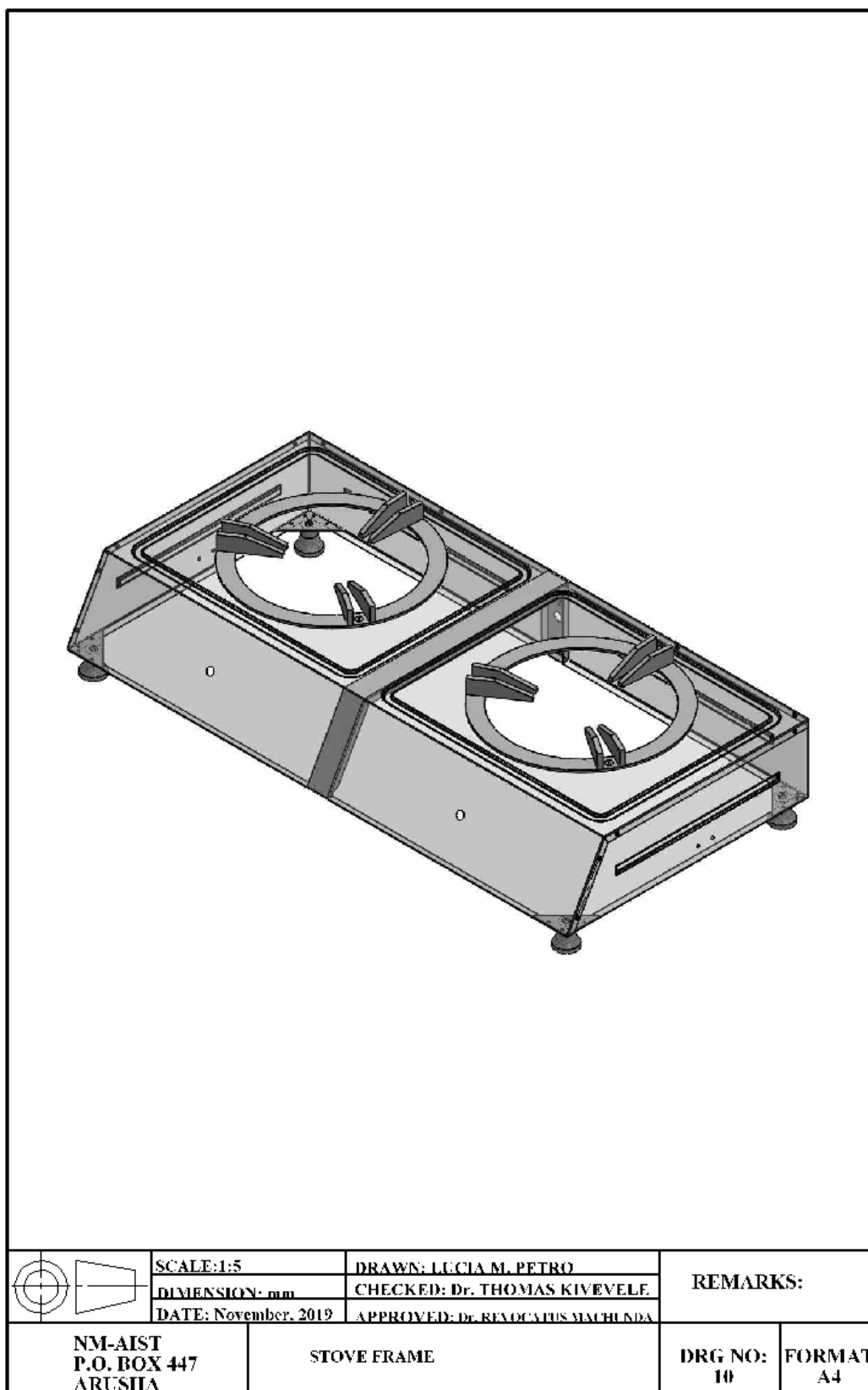
Appendix 8: Middle Reinforcement



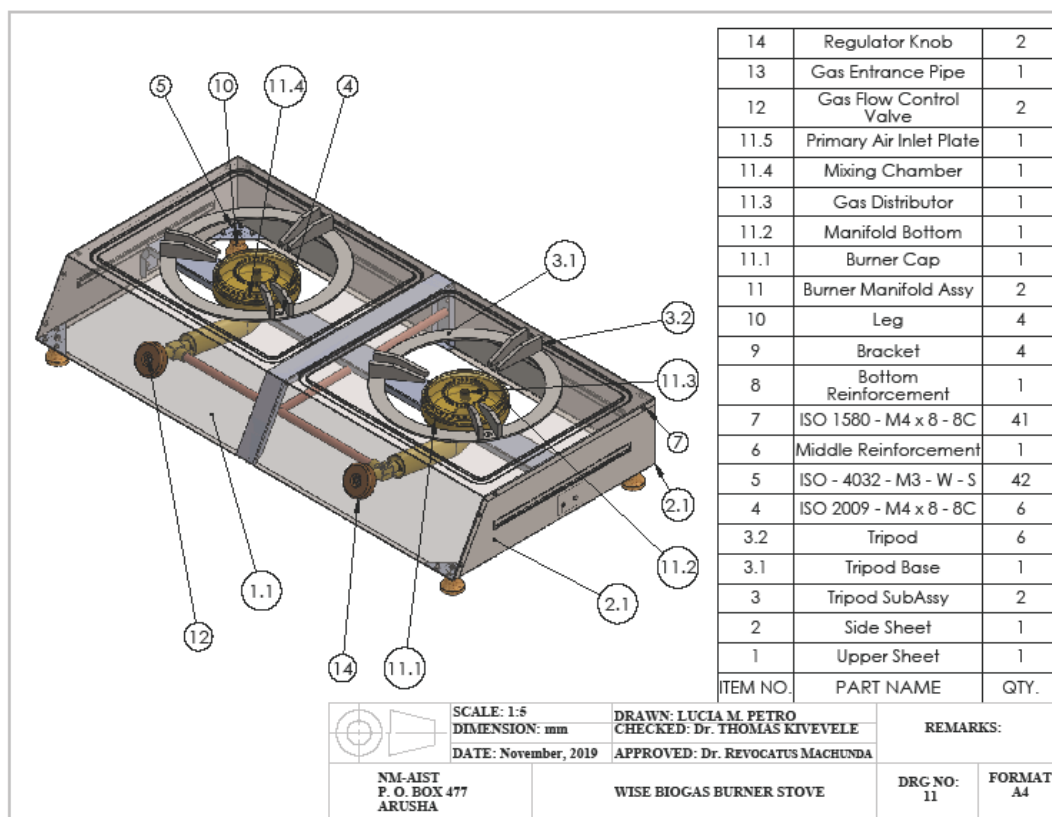
Appendix 9: Gas Distributor



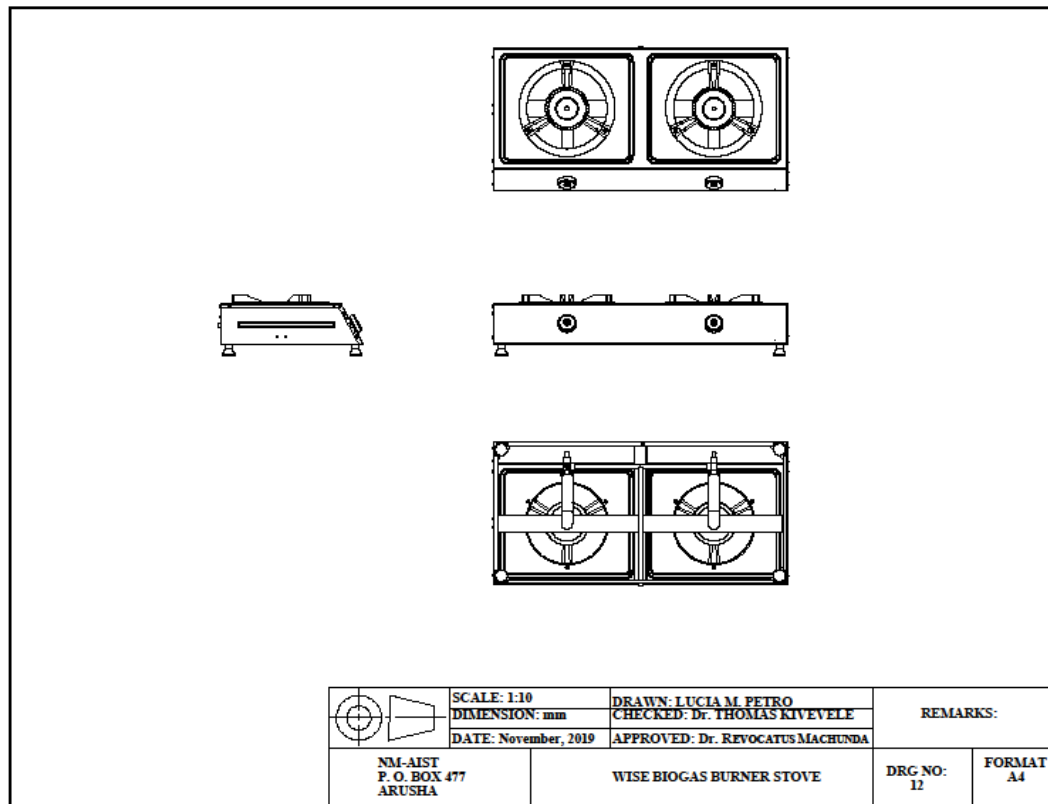
Appendix 10: Biogas stove frame



Appendix 11: Biogas burner stove assembly



Appendix 12: Burner stove 3D drawing



Appendix 13: WBT results for the WISE biogas stove

1.High Power Test (cold Start)	Units	Test 1	Test 2	Test 3	Average	St Dev
Time to boil water pot #1	min	9	8	9	8.667	0.577
Temp-corrected time to boil pot # 1	min	9.51	8.33	9.25	9.03	0.616
Burning rate	g/litre	3.26	3.45	3.64	3.45	0.192
Thermal efficiency	%	12.75	10.72	11.41	11.63	1.033
Specific fuel consumption	g/litre	0.41	0.43	0.46	0.43	0.024
Temp- corrected specific fuel consumption	g/liter	0.43	0.45	0.47	0.45	0.019
Temp-corrected specific Energy consumption	kJ/liter	24.19	22.77	27.04	24.67	2.174
Firepower	kWatts	17.31	18.33	19.35	18.33	1.002

2.High Power Test (Hot Start)	Units	Test 1	Test 2	Test 3	Average	St Dev
Time to boil water pot #1	min	8	8	8	8	0
Temp-corrected time to boil pot # 1	min	8.96	8.33	8.57	8.62	0.314
Burning rate	g/min	3.07	3.26	3.45	3.26	0.19
Thermal efficiency	%	12.06	11.35	10.72	11.38	0.670
Specific fuel consumption	g/liter	0.386	0.410	0.434	0.41	0.024
Temp- corrected specific fuel consumption	g/liter	0.432	0.427	0.465	0.44	0.021
Temp-corrected specific Energy consumption	kJ/liter	20.24	21.51	22.77	21.51	1.27
Firepower	kWatts	16.30	17.31	18.33	17.31	1.018

3. Low Power test (Simmer)	Units	Test 1	Test 2	Test 3	Average	St Dev
Burning rate	g/min	6.517	8.433	9.583	8.178	1.355
Thermal efficiency	%	81.69	63.41	55.93	67.01	13.25
Specific fuel consumption	g/liter	586.5	759	862.9	736	139.430
Temp- corrected specific Energy consumption	kJ/litre	967.73	1252.35	1423.13	1214.4	230.060
Fire power	kWatts	34.63	44.82	50.93	43.80	8.23
Turn down ratio	--	2.30	2.30	1.90	2.15	1.26

Appendix 14: WBT results for the Simgas biogas stove

High Power Test (cold Start)	Units	Test 1	Test 2	Test 3	Average	St Dev
Time to boil water pot #1	min	9	8	9	8.67	0.577
Temp-corrected time to boil pot # 1	min	9.93	8.82	9.64	9.46	0.573
Burning rate	g/litre	4.03	3.83	4.22	4.03	0.192
Thermal efficiency	%	10.32	9.65	9.85	9.94	0.345
Specific fuel consumption	g/litre	0.51	0.48	0.53	0.51	0.024
Temp- corrected specific fuel consumption	g/liter	0.56	0.53	0.57	0.55	0.019
Temp-corrected specific Energy consumption	kJ/liter	29.89	25.3	31.31	28.83	3.140
Firepower	kWatts	21.39	20.37	22.41	21.39	1.02

2.High Power Test (Hot Start)	Units	Test 1	Test 2	Test 3	Average	St Dev
Time to boil water pot #1	min	9	10	9	9.33	0.577
Temp-corrected time to boil pot # 1	min	10.07	11.19	10.23	10.50	0.607
Burning rate	g/min	3.45	3.64	3.45	3.55	0.11
Thermal efficiency	%	12.04	12.66	12.04	12.25	0.36
Specific fuel consumption	g/liter	0.43	0.46	0.43	0.44	0.014
Temp- corrected specific fuel consumption	g/liter	0.49	0.51	0.49	0.50	0.014
Temp-corrected specific Energy consumption	kJ/liter	25.62	30.04	25.62	27.09	2.56
Firepower	kWatts	18.33	19.35	18.33	18.67	0.588

3. Low Power test (Simmer)	Units	Test 1	Test 2	Test 3	Average	St Dev
Burning rate	g/min	8.242	11.117	8.433	9.775	1.607
Thermal efficiency	%	64.30	48.32	63.84	58.82	9.09
Specific fuel consumption	g/liter	741.75	1000.5	759	833	144.667
Temp- corrected specific Energy consumption	kJ/liter	1223.89	1650.83	1252.35	1375.689	238.701
Fire power	kWatts	43.80	59.07	44.82	49.23	8.54
Turn down ratio	--	2.44	1.94	2.50	2.26	0.46

Appendix 15: WBT results for the CAMARTEC biogas stove

High Power Test (cold Start)	Units	Test 1	Test 2	Test 3	Average	St Dev
Time to boil water pot #1	min	8	8	10	8.67	1.155
Temp-corrected time to boil pot # 1	min	8.45	8.33	10.42	9.07	1.170
Burning rate	g/min	8.43	6.52	6.9	7.28	1.0
Thermal efficiency	%	4.39	5.68	6.68	5.58	1.151
Specific fuel consumption	g/liter	1.06175	0.8205	0.8687	0.92	0.128
Temp- corrected specific fuel consumption	g/liter	1.12	0.86	0.91	0.96	0.142
Temp-corrected specific Energy consumption	kJ/liter	55.66	43.01	56.93	51.87	7.69
Firepower	kWatt	44.81	34.63	36.67	38.70	5.39

2.High Power Test (Hot Start)	Units	Test 1	Test 2	Test 3	Average	St Dev
Time to boil water pot #1	min	8	9	7	8	1
Temp-corrected time to boil pot # 1	min	8.70	9.78	7.95	8.81	0.919
Burning rate	g/min	3.83	6.13	4.98	4.98	1.15
Thermal efficiency	%	9.65	6.77	6.51	7.64	1.74
Specific fuel consumption	g/liter	0.48	0.77	0.63	0.63	0.145
Temp- corrected specific fuel consumption	g/liter	0.52	0.84	0.71	0.69	0.158
Temp-corrected specific Energy consumption	kJ/liter	25.3	45.54	28.78	33.21	10.82
Firepower	kWatts	20.37	26.07	27.24	24.56	3.68

3. Low Power test (Simmer)	Units	Test 1	Test 2	Test 3	Average	St Dev
Burning rate	g/min	8.241	11.691	10.733	10.222	2.439
Thermal efficiency	%	66.21	46.78	50.84	54.61	10.25
Specific fuel consumption	g/liter	741.75	1052.25	966	920	160.279
Temp- corrected specific Energy consumption	kJ/liter	1223.89	1736.21	1593.9	1518	264.46
Firepower	kWatts	43.80	62.13	57.04	54.32	9.46
Turn down ratio	--	1.02	0.56	0.64	0.74	4.68

Appendix 16: The water boiling procedure

The 2.5 L of water was poured into an aluminium pot weighing 0.549 kg. Both the ambient and water temperatures were monitored by using a mercury thermometer during WBT. The temperature of the water inside the pot was monitored by a mercury thermometer and flame temperature measured by using Infrared thermometer. The thermometer was suspended at the centre of the pot by hand-holding, the gap between the bottom surface of the pot and thermometer tip was 5 mm.



RESEARCH OUTPUT

- 1. Research Article**
- 2. Patent**
- 3. Poster**

Theoretical and experimental performance analysis of a novel domestic biogas burner

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Abstract The inefficient indoor burning of fuelwood on traditional cookstoves generates pollutants, primarily carbon monoxide and many other human health-damaging emissions. It is from this risk that it is necessary to have an immediate shift to alternative cleaner fuel sources. Biogas, which is among the biofuels from biomass, is one of the resources that play a considerable part in a more diverse and sustainable global energy mix. For domestic purposes in rural areas of Tanzania, biogas provides a better option that can supplement the use of fossil fuels such as wood, charcoal, and kerosene, which is nonrenewable. However, the low efficiency experienced in the locally made biogas burners hinders the large-scale use of biogas amongst the population in the country. With the locally made burners, the users of biogas for the domestic application face problems including heat loss and high gas consumption which affects the whole cooking process. It is against this backdrop that the current study objectives incline on designing and improving the efficiency of the locally manufactured burners to achieve the uniform flow of fuel in the mixing chamber, which will result to the consistent heat distribution around the cooking pot. The optimization of the burner was done by using computational fluid dynamics (CFD) through varying the number of flame portholes, air holes as well as the size of the jet before fabrication. The increased efficiency of the burner has also contributed by the addition of the fuel distributor. The results showed that the optimum hole diameter of the jet was 2.5 mm and that of the manifold was 100 mm. The currently developed biogas burner was tested and compared with the other two locally made burners. The water boiling test (WBT) on these three burners showed that the developed burner has a thermal efficiency of 67.01% against 54.61% and 58.82% of Centre for Agricultural Mechanization and Rural Technology (CARMATEC) and Simgas respectively. Additionally, the fuel consumption of the developed burner was 736 g/L as compared to 920 g/L for CARMARTEC and 833 g/L for that of Simgas. The developed burner and its corresponding cook stove are both environmentally friendly and economical for household utilization in Tanzania and other developing countries.

Keywords: Biogas burner, Computational Fluid Dynamics, optimization, Injector, Efficiency, Water Boiling Test

1. Introduction

Energy is a crucial requirement for the development of a country. Around 80% of the population in Tanzania have no access to power, particularly electricity. They, subsequently, strongly depend on conventional energy sources such as charcoal and firewood for cooking, lighting and other applications as an alternative [1]. Approximately 90% of the energy needs of the rural people in Tanzania comes from biomass [1, 2]. Environmental degradation due to deforestation for firewood and charcoal has increased drastically, leading to soil erosion [3]. Heating and cooking exercises within the family units are generally done by the utilization of firewood and charcoal, which causes indoor air pollution leading to infections that, in most cases, affect women and children [1]. The inefficient burning of biomass on conventional cookstoves produces emissions, mainly carbon monoxide and other human health-damaging pollutants such as nitrogen oxides, benzene, butadiene, formaldehyde, polyaromatic hydrocarbons. Biogas is a renewable energy resource and an alternative biomass option that can counteract these effects [4] and is obtained from inexpensive, readily available material wastes in rural areas, such as animal wastes and crop wastes [5]. In Tanzania, around 21 tons of cow dung are produced each year and can be gasified to create heat energy [6]. Biogas is considered one of the greatest substitute fuel that can supplement the utilization of solid biomass and fossil fuel for household applications, especially in developing countries, Tanzania inclusive [7]. It is environmentally, and it eliminates workload on women livelihood, hence improve lives [8]. The successful use of biogas reduces the country's dependence on fossil fuels [9].

Few studies have specifically investigated the thermal performance of locally made biogas burners theoretically and experimentally. Mulugeta et al. [10], did research on biogas injera baking burner with a manifold external diameter of 170mm. The experiment and analysis results demonstrated that the burner was small, and therefore, insufficient for baking the injera pieces of breads. This was caused by the nonuniform distribution of heat along with the baking pan.

Decker [11], conducted research on burner flame portholes. The burner consisted of rectangular flame port holes which caused the friction during air - fuel mixture flow, hence reducing the flame stability. Orhorohoro et al. [12] designed an improved biogas fuel stove because the existed burner had low efficiency. The study focused on the gap between flame portholes, mixing tube length, flame port diameter and distance between cooking pot and stove burner. After the analysis and improvement, the maximum efficiency obtained was 63.87%. Sukhwani et al. [13] designed and conducted a performance evaluation of an improved biogas stove. In the analysis, the variation of burner flame port diameter was done as well as preheating of the biogas to increase its efficiency. The results showed an increased performance efficiency of 42.99%.

The existing burners consist of flame ports located on their outer circumference which, allow only cooking and cannot be used for preheating. This arrangement increases the costs for users because even preheating consumes more fuel. Therefore, domestic users face challenges including heat loss, low heating value, and high gas consumption which affect the

whole cooking process. Other factors which cause low efficiency are flame speed (velocity), manifold size and pot to burner distance [14].

The information provided in this paper is useful for generating knowledge and findings on improving the performance of locally made burners. Therefore, the main aim of this study is to optimize the performance of a novel biogas burner made using local and low-cost materials.

2. Materials and Methods

2.1 Analytical design of biogas burner

The technique and analytical design of the burner involved the determination of various important factors such as jet diameter, throat diameter, area of the burner port, number of ports and area of the jet [15, 16].

2.1.1 Determining output power and biogas flow rate.

The strategy adopted in this study comprised the 3D geometrical design of a burner, numerical design analysis using CFD, analytical design procedures, and manufacture of a prototype. The analytical design adopted for biogas burner [17].

The output power of the stove (kW) was calculated using Equation 1 [18]:

$$P = 5.2(V/t) \quad (1)$$

Where V is the volume of water in litres and t is the time consumed to boil such amount of water.

The gas flow rate, Q can be determined by Equation 2 [18]:

$$Q = 0.45(P) \quad (2)$$

Where Q biogas is the flow rate in m³/h and P is the output power in kW.

2.1.2 Jet diameter, d_j

The jet diameter, d_j was calculated by using Equation 3.

$$d_j = \sqrt{\frac{Q_{biogas}}{0.0361 \times C_d}} \times \sqrt[4]{\frac{s}{p}} \quad (3)$$

Where C_d is the coefficient of discharge for injector jet, p is the gas pressure before injector jet, and s is the specific gravity of the gas.

The area of injector jet (A_j) was calculated using Equation 4.

$$A_j = \frac{\pi \times (d_j)^2}{4} \quad (4)$$

The biogas velocity in the injector jet, V_j was determined according to Equation 5.

$$V_{gas} = \frac{Q_{biogas}}{A_j} \quad (5)$$

2.1.3 Determination of throat diameter, d_t

The throat diameter was determined using Prig's formula, shown in Equation 6.

$$d_t = \left[\left(\frac{r}{\sqrt{s}} + 1 \right) d_j \right]$$

(6)

Where r is entrainment ratio, and s is the specific gravity of the gas.

According to [Fulford \[16\]](#), it is recommended to double the throat diameter for best design.

The exact size depends on the standard pipe sizes available.

2.1.4 Determination of burner port design

As suggested by [Fulford \[16\]](#), the stoichiometric flame speed of biogas is 0.25 m/s.

The blending supply velocity, v_p is given by [Fulford \[16\]](#):

$$V_p = \frac{Q_m}{A_p} \ll 0.25 \text{ m/s} \quad (7)$$

The burner port area was solved as follows:

$$A_p > \frac{Q_m}{V_p} \gg 0.0025 \text{ m}^2$$

(8)

From the design calculations, the centre flame port distance from one hole to another is 3mm to avoid the flame lift. Then, the number of flame portholes was calculated according to Equation 9

$$n_p = \frac{4A_p}{\pi d_p^2} \quad (9)$$

Where n_p is the number of holes and d_p is the diameter of each hole in meters.

2.2 Burner drawings and Geometry modelling

The parameters obtained from the analytical design were used to generate 3D drawings of the burner using SOLID WORKS software, as shown in Figure 1.

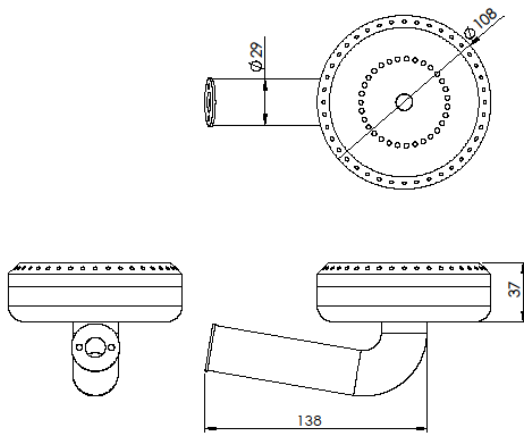


Figure 1: 3D Geometry of a burner.

The 3D geometry of the model has then imported in ANSYS Workbench for further analysis, as shown in Figure 2.

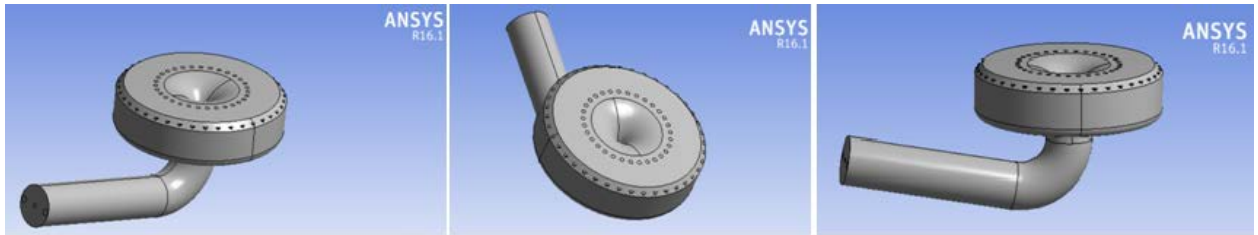


Figure 2: 3D geometry of a burner

2.2.1 CFD simulation of the burner

The CFD study is based on creating the simulation of biogas combustion of the optimized burner. The simulation was performed by applying ANSYS 16.0 software and the modeller was used to model the burner while the meshing was done by ANSYS (ICEM-CFD), and the solution was computed by the use of the fluent (Post Processor). It is important to mention that CFD can design and simulate a model without physically constructing the model. Numerical modelling technique including but not limited to CFD methods has gained fame in academic and industrial sectors because of the availability of efficient computer systems. It is possible to optimize the system design operation and understand physiochemical properties inside the model using CFD simulations, which can consequently reduce the number of experiments required for characterization of the system. Simulations produce spatial and temporal profiles of different system variables that are either impossible to measure or are accessible only by expensive experiments [19].

Figure 3 shows the 3D geometry based on the actual dimensions. Each measurement was checked on the plan with corresponding input measurements which can be changed. In this way, each measurement is simple and rapidly modified and the whole construction can be modified, redrawn and advanced steps like mesh creation or design optimization are conceivable in real-time [20, 21].

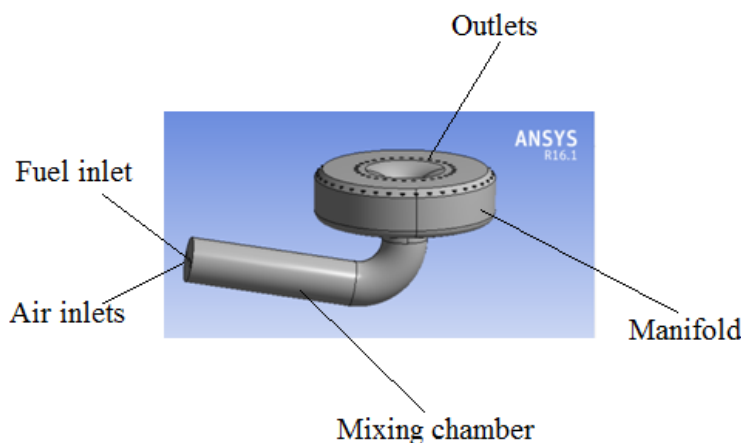


Figure 3: Boundary condition of a burner on a 3D model.

It is required to provide grids or mesh into the geometry, where meshing is an essential part of the CFD process because the method chosen on the meshing stage decides the quality of the mesh. Figure 4 shows the meshing of the developed biogas burner with grids of elements employed to solve liquid flow Equations [19]. The grid size has an effect on computational time which directly influences simulations cost as well as the speed of convergence and the accuracy of the solution [22].

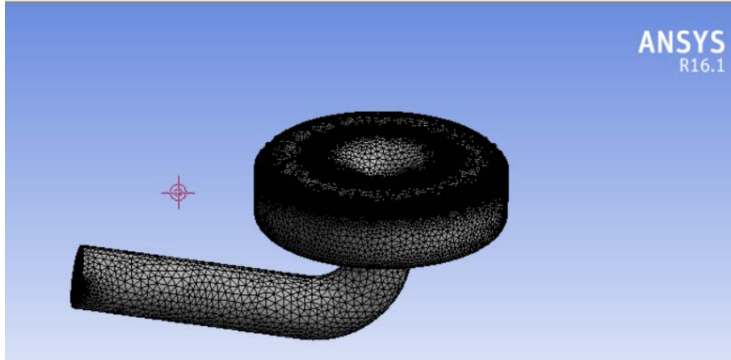


Figure 4: Burner mesh Geometry generated using ANSYS software.

To realize sufficient determination of the stream and reduce the time taken by the computer to run the simulation, hexahedral cells are chosen. Also, refined grids reduce the period for calculation, and the convergence is reached more easily [22]

The boundary conditions were indicated according to the profile of the geometry, as shown in Figure 3. The non- Premix combustion with turbulent realizable K- ϵ and the model P1 were used in the simulation of combustion in the newly designed burner. ANSYS-FLUENT requires input parameters in terms of gas composition concerning CO₂ and CH₄, which are to be used in the simulation. The transient model was used as the initial setting in 3D geometry with the pressure-based solver. The details of the model parameter settings and model input parameters for the simulation are shown in Table 1 and Table 2, respectively.

Table 1: Settings of the fluent model

Model Parameters	Status
Multiphase	Off
Energy	On
Viscous	Standard k-e, Standard Wall Fn
Radiation	P1
Heat Exchanger	Off
Species	Non – Premixed Combustion
Inert	Off
NOx	Off
SOx	Off
Soot	Off
Decoupled Detailed Chemistry	Off
Reactor Network	Off
Reacting Channel Model	Off

Discrete phase	Off
Solidification & Melting	Off
Acoustics	Off
Eulerian Wall Film	Off

The non-premix combustion model was employed, where the initial velocity of air and fuel was considered 5.2 m/s and 36.1m/s, respectively. Table 3 shows the Probability Density Function (PDF) of boundary conditions for the fuel and oxidant. The biogas contains a mixture of 60% of CH₄ and 40% of CO₂ on a molar basis [23]. As presented in Table 3, the oxidant consists of 21% and 79% mixture of oxygen and nitrogen, respectively, similar to the normal combustion air configuration. The temperature of the fuel and oxidant was set at 300 K, and the CFD simulation was performed by varying the geometry dimensions as well as manifold size to achieve the optimum design.

Table 2: Model input data of the Burner

Parameter		Value
Burner Dimensions	(L x H x W) mm	265 x 67.5 x 100
Species	CH ₄	0.6
	CO ₂	0.4
Air inlets	Velocity (m/s)	5.2
	Temperature (K)	300
Biogas (Fuel)	Velocity (m/s)	36.01
	Temperature (K)	300
Operating pressure	(Pa)	101325
Gravitational Acceleration	(m/s ²)	-9.81

Table 3: PDF for boundary conditions

Species	Fuel	Oxidant
CH ₄	0.6	0
O ₂	0	0.21
N ₂	0	0.79
H ₂	0	0
CO ₂	0.4	0

2.3. Fabrication of a burner

The fabrication of the prototype was done in the workshops of Centre for Agricultural Mechanization and Rural Technology (CAMARTEC) and at Arusha Technical College (ATC), Arusha, Tanzania. A burner was fabricated basing on the results of CFD analysis. Figure 5 shows the cookstove assembly fabricated to accommodate the burner. The cookstove consists of a burner jet for the fuel flow, a mixing tube comprising of air holes for proper mixing of fuel and air, and a manifold at which the combustion mixture is distributed to each flame porthole in the burner head. A distributor is provided at the manifold to balance the pressure and ensure equal distribution of air-fuel mixture. Additionally, the cookstove has a gas supply pipe which supplies fuel from the source to the control valve. Tripods are fixed on the stove frame to hold the pots as well as to provide the suitable position between the burner top and the pot bottom.

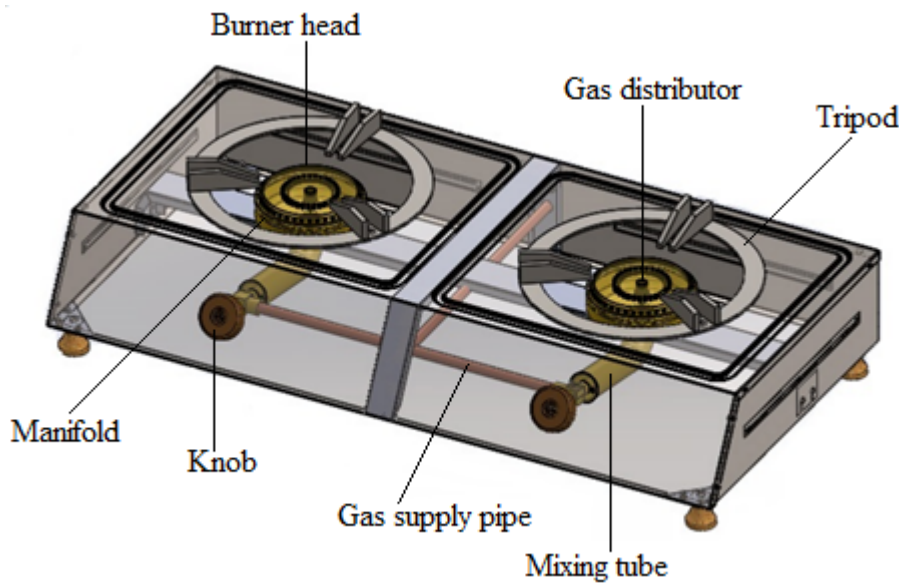


Figure 5: Stove Assembly.

2.4. Evaluation of the performance of the burners

The experiments to determine the performance of the burners were carried out in the energy lab at CAMARTEC in Arusha, Tanzania. The performance of the developed burner was compared with that of Simgas and CAMARTEC burners. The experiments were designed to simulate real-life cooking processes and were conducted using the water boiling test (WBT).

2.4.1 Theoretical Analysis

The water boiling test (WBT) method was used to assess the thermal efficiency (η_t), the firepower (FP), the specific fuel consumption (SC) and the burn rate (r_b) of burner [24]. Several formulae relating to burner performance have been developed. The methods adopted based on the approach done by Berrueta et al. and Ahuja et al. [24, 25].

1. Thermal efficiency (η_{th}) was calculated by the ratio of the energy used during heating and vaporizing water to the energy consumed by burning the biogas fuel [26-28]. Mathematically,

$$\text{Thermal efficiency, } \eta_{th} = \frac{((C_{pw} \times M_w) \times (T_f - T_i)) + (H_v \times W_v)}{C_v \text{ of biogas} \times V_{bg}} \times 100 \quad (10)$$

Where, V_{bg} is Volume of biogas consumed, m^3/h , M_w is the mass of water boiled, kg, C_{vbg} is the calorific value of fuel (biogas) = 22 MJ/kg, C_{pw} is the specific heat capacity of water = 4186 J/kg °C, T_f is the boiling temperature of water = 95.23 °C, W_v is the water evaporated from the pot., T_i is the initial temperature of the water, °C and H_v is the latent heat of evaporation of water, 2260 J/g.

2. Firepower (FP) is the proportion of fuel energy utilization by burner per unit time. It was calculated by Equation 11[24, 27]. Mathematically,

$$FP = \frac{V_{bg} \times LHV}{60 \times (t_f - t_i)} \quad (11)$$

Where FP is the power of burner in watts, V_{bg} is the volume of biogas consumed during the stage of the test (m^3/h), LHV is the lower heat value of the biogas (kJ/kg), it is the time at the end of the test (min), and t_i is the time at the start of the test (min).

3. Specific fuel consumption (SC) is the proportion of the amount of biogas fuel consumed to the amount of water remaining within the pot at the end of the test. SC was calculated using Equation 12 [26]. Mathematically,

$$SC = \frac{V_{bg}}{W_r} \quad (12)$$

Where SC is the specific fuel consumption of the burner (g/L), V_{bg} is the volume of biogas fuel expended during the phase of the test (g), and W_r is the mass of water remaining at the end of the test (g).

4. Burning rate (r_b) is used to measure the of fuel consumption to bring water to boil during the experiment. r_b was calculated using Equation 13 [27, 29]. Mathematically

$$r_b = v_{bg} / \Delta t \quad (13)$$

Where V_{bg} is the Volume of biogas consumed during the test (m^3/h), and Δt is the total time taken to conduct the test (min).

2.4.2 The water boiling test (WBT)

The water boiling test consisted of three stages. The first stage conducted a test with both cold and hot start conditions in high power start. The second stage was that of lower power (simmering) which simulate slow-boiling tasks [27]. A standard aluminium pot of 3 litres capacity was used in all three experiments. At the high power cold start phase, 2.5 litres of water was boiled at room temperature up to 95°C local boiling point. In the high power hot start, the fuel was reset immediately after the cold start phase. The test was repeated to note the difference in performance when the burner is cold and when it is hot. The third stage involved the simmering test, which was done after the high power hot start phase test. This involved measuring the burner capacity to move from high heating to low heating, and each experiment was repeated three times. All three burner stoves were tested at the same time on different days, and average results were recorded [24].

3. Results and Discussions

3.1 Computational Fluid Dynamics (CFD) Analysis

Table 4 shows the results of the CFD analysis employed in the study to simulate the fluid flow in the burner. During the simulation of the model, the convergence is reached easily when a set of Equations are solved and when the calculations are repeated several times until the flow and the temperature are converged. The default criterion for convergence is that each residual is reduced to a value less than 10^{-3} except the energy residual for which the default criterion is 10^{-6} .

Table 4: The results of the analytical design and CFD input values

s/no	Parts	Analytical values	CFD values
1	Injector diameter (mm)	2.5	2.5
2	Internal throat diameter (mm)	18	25
3	Number of burner ports	71	71
4	Mixing tube length (mm)	170	138
5	Internal manifold diameter (mm)	100	100
6	Flame port diameter (mm)	2.5	2.5
7	Air inlet diameter (mm)	5.0	5.0

Figure 6 shows the contours of temperature distribution in the burner. It was observed that the temperature distribution along the jet and mixing tube was the same as the room temperature of the air and gas inlets as in the boundary conditions. The temperature was identical throughout the mixing tube and increased to a maximum of 997 K at the manifold. The temperature in the middle Section of the manifold was about 757 K, and it was uniform

throughout. Figure 7 is a graph which indicates the distribution of temperature along with the burner. The temperature increased circumferential upwards from the bottom to the upper part of a manifold.

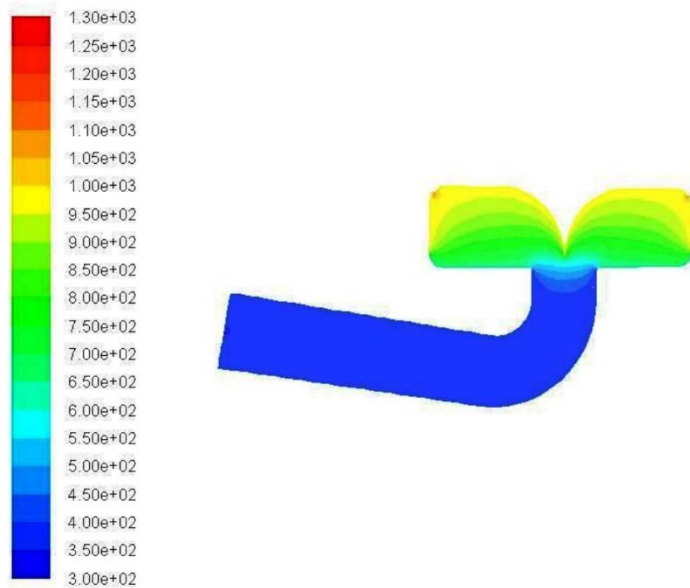


Figure 6: Contours of Static temperature along the burner.

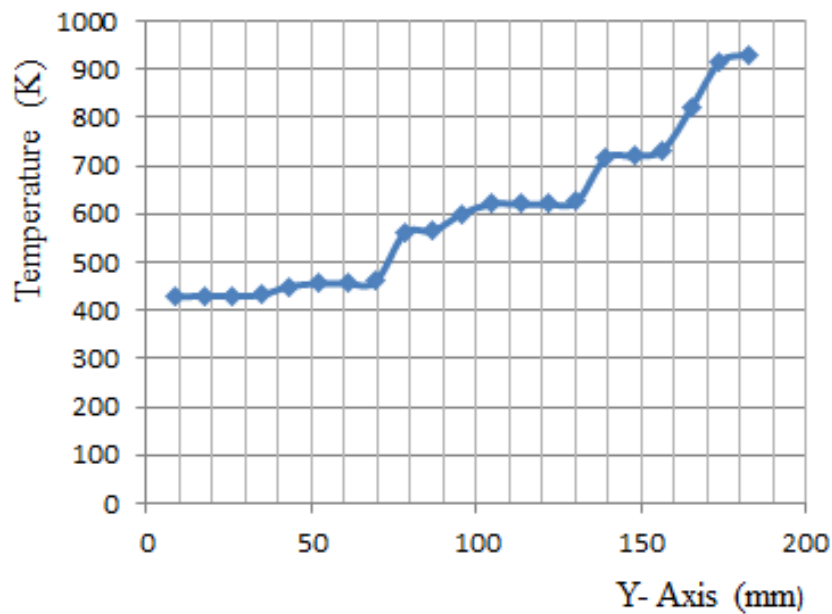


Figure 7: Temperature distribution with position along with the burner.

Velocity distribution of air-fuel mixture along the mixing chamber of the burner is shown in Figure 8. The fuel contains CH_4 and CO_2 , which play a critical part in the burning process.

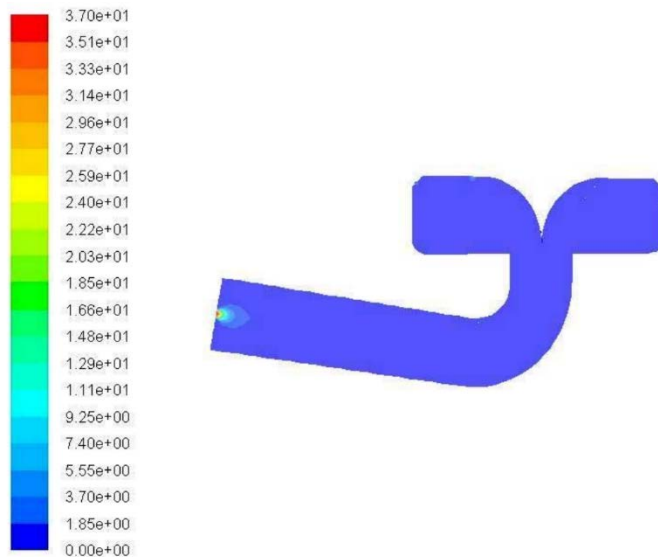


Figure 8: Contours of velocity distribution (m/s).

The velocity of the air and fuel was higher at the inlets about 5.2 m/s and 36.01 m/s, respectively, and remained uniform at the mixing chamber and along the manifold as shown in Figure 8. These findings are in agreement with other studies done by [Decker \[30\]](#), and Mulugeta et al. [31] and thus indicates that the distribution of velocity is uniform in the manifold. The independent mesh test was performed to verify and show that results are not dependent on mesh size. The influence of temperature change basing on the mesh size is shown in Figure 9. During the meshing process, the mesh size was varied from 0.5 to 0.02, and in those various tests of mesh size, 0.02 is considered appropriate consistently to ensure that temperature was independent of the mesh size.

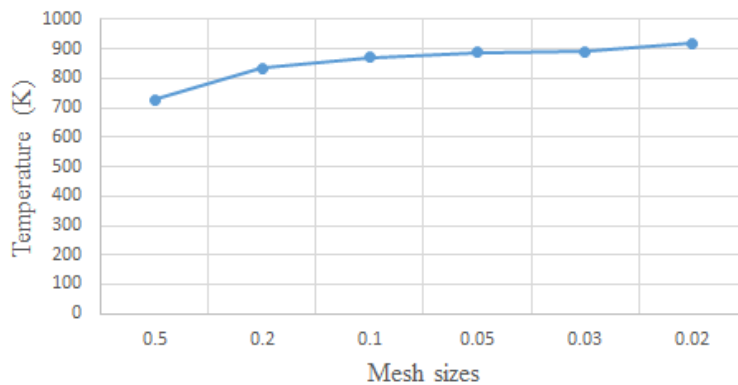


Figure 9: Influence of mesh size on Temperature

3.2 Results of water boiling Test

The results show that in the HPCS phase, the time taken to boil 2.5 litres of water to the local boiling point was 7 minutes for the developed burner, whereas it took 10 and 11 minutes for Simgas and CAMARTEC burners respectively. For the HPHSP, the time taken by the CAMARTEC and Simgas burner to bring 2.5 litres of water to the local boiling point was 10 min, while 7 min taken by the developed burner.

The lower power (simmering) was conducted for 45 minutes, and in a CAMARTEC burner, the water temperature decreased to 78°C compared to 89°C and 92°C of Simgas and developed burner respectively compared to the local boiling point of 95.3°C. The flame temperature was recorded by Infrared thermo-couple, and it was about 652°C during the peak hour of combustion.

Figure 10 shows the thermal efficiency in the simmering phase of the three burners tested. The developed burner performed the highest efficiency of 67.01%, followed by Simgas burner with 58.82%, and lastly, CAMARTEC burner exhibited thermal efficiency of 54.61%. Other researchers such as Orhorohoro et al. [12] did research, and find that, the optimal cooking efficiency of biogas stove is 63.87 %. Another study was conducted by Sukhwani et al. [13] on a designed burner to improve efficiency. The results showed that the performance efficiency was 42.99%. Also, Demissie [28] reported the overall efficiency of a burner stove evaluated through Water Boiling Test (WBT) which was found to be 54.8% and 43.6% at the higher flame intensity and lower flame intensity respectively.

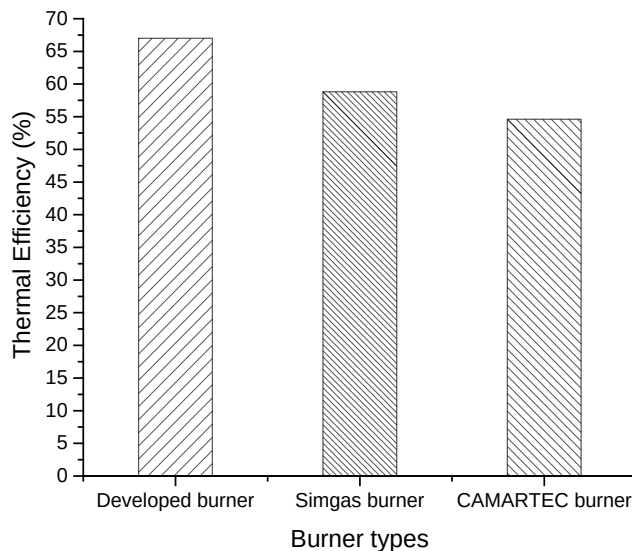


Figure 10: Thermal efficiency in the simmering phase of the developed burner, Simgas and CAMARTEC burners.

The overall Specific fuel consumption over the whole WBT for each burner was found to be 245.34, 277.68, and 306.68 g/L for the developed burner, Simgas and CAMARTEC burners respectively as shown in Fig. 11. The low SFC for developed burner resulted in higher thermal efficiency which is contributed by the addition of a gas distributor in the manifold to balance the pressure and ensure equal distribution of air-fuel mixture in the flame portholes. Furthermore, the small gap between the burner head and the pot bottom was made 30mm, which keeps the fire closer to the cooking pot most of the time during the tests.

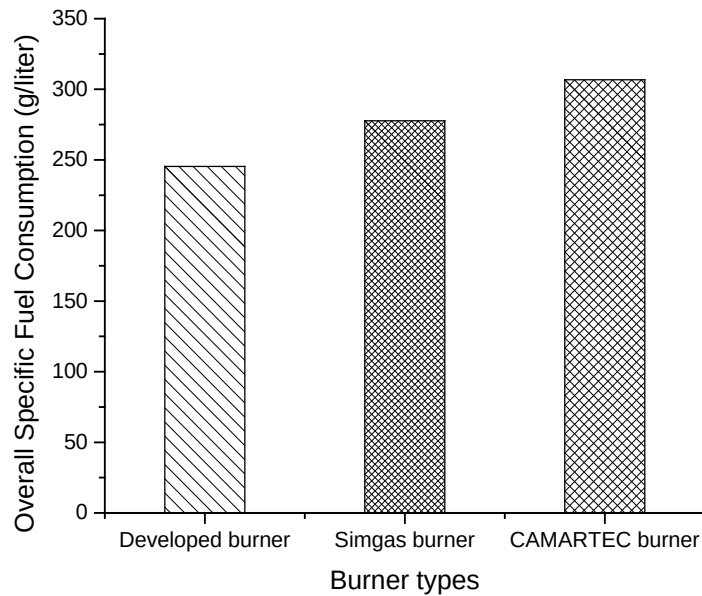


Figure11: Overall specific fuel consumption over the entire WBT for the three burners.

During the high power phase, the developed burner burned fuel at a rate of 3.45 g/min and 3.26 g/min in the cold and hot phases, while a burning rate of 8.18 g/min was achieved in the simmering phase. For Simgas burner, the average burning rates were 4.03 g/min, 3.55 g/min, and 9.78 g/min respectively, for cold, hot, and simmering phases. While the CAMARTEC burner appeared the highest average burning of 7.28 g/min, 4.98 g/min, and 10.22 g/min in cold, hot, and simmering stage, respectively, as shown in figure 12. The overall average low burning rate of 4.96 g/min managed by the developed burner was attributed to the appropriate design of the burner especially the pot gap that minimized heat losses, and the adjustable primary air hole that ensured well air mixing to reduce flame lift or backfire to the mixing chamber. The overall average high burning rate for Simgas burner 5.79 g/min and 7.49 g/min for CAMARTEC burner is due to the big gap between burner head top and bottom pot which consumes a large amount of biogas.

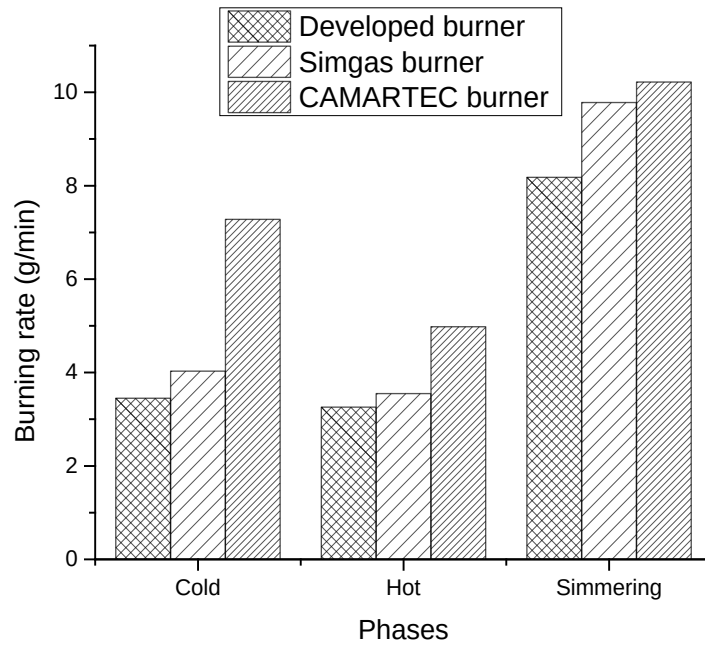


Figure12: Burning rates of the developed burner, Simgas and CAMARTEC burners.

For the cold high power phase, the developed burner, Simgas, and CAMARTEC burner gave firepower of 18.33, 21.39, and 38.70 watts, respectively. In hot high power phase, the developed burner, Simgas, and CAMARTEC burner gave firepower of 17.31, 18.6, and 24.56 watts, respectively, whereas in low power the firepower for the individual burners were 43.80, 49.23 and 54.32 watts respectively, as shown in Figure 13. Developed burner displayed better results in terms of firepower because of the modification done on the burner parameters such as the diameter of the injector, throat size, throat diameter, and the height of burner head. According to Miller-Lionberg [32] reported that increasing the firepower of the burner decreases efficiency.

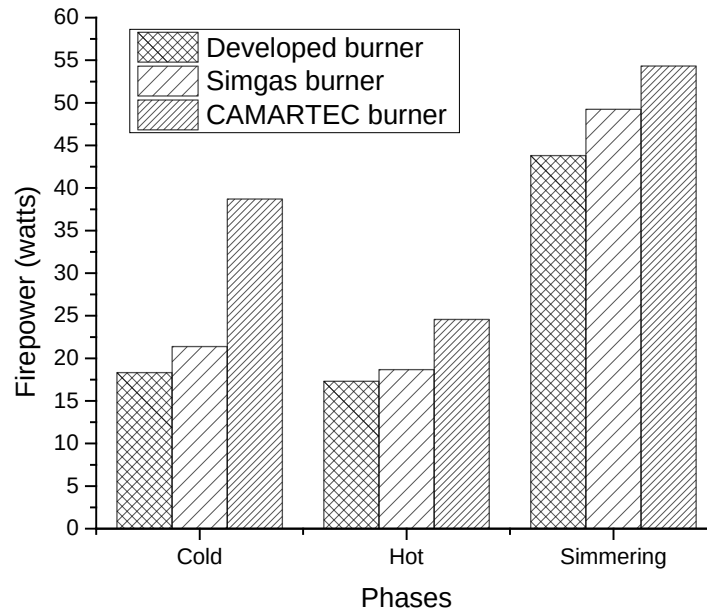


Figure 13: Firepower during cold, hot, and simmering phases of the developed burner, Simgas, and CAMARTEC burners.

4. Conclusion

The outcomes of this study present perception in the modelling process of the newly designed burner and its performance. The design of the burner consisted of theoretical analysis, numerical design calculations, and Computational Fluid Dynamics (CFD). As per the analytical design, several formulas were used to design the key parts of a burner. The developed burner head consists of 40 portholes located on its outer circumference having 2.5 mm diameter each, and 31 portholes with 2.5 mm diameter each at its top surface. The manifold of a developed burner has 100mm internal and 138 mm external diameters, respectively. The simulation results show that the maximum flame temperature observed was 997K against 925K of the experimental measurements.

The water boiling test (WBT) was conducted for the developed burner, and its thermal efficiency performance was 67.01% against 54.61% and 58.82% of CARMATEC and Simgas burners respectively. The comparison for the average fuel consumption was also conducted between a developed burner and the above - mentioned burners. The test showed that the developed burner fuel consumption was 736 g/L compared to 920 g/L and 833 g/L for CARMARTEC and Simgas, respectively. The low fuel consumption observed from the developed burner as compared to others was due to the proper distribution of fuel on the cooking pot most of the time during the test.

Conflicts of Interest

The authors declare no conflict of interest.

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In the matter of

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