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Addis Ababa Institute of Technology

Department of Mechanical Engineering

Development of Efficient Biogas Injera Baking Stove

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Dedication

I dedicate this thesis to my beloved family and friends.

ACKNOWLEDGEMENT

I would like to express my sincere gratitude to Dr. -Ing. Demiss Alemu for his constant support and guidance throughout this thesis work. Without his precious help this thesis could have never achieved its current form.

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ABSTRACT

Injera baking is the most energy intensive process. The wide spread use of wood cutting for fuel is the primary cause of deforestation in Ethiopia. About 95% of the total energy consumption in Ethiopia mostly comes from biomass fuels. Historically Ethiopia was mostly covered with forest up to recently. In the past 50 years the land covered by forest has dropped from approximately 50% to less than 3%. Some experts attribute this is mostly due to forest clearing for cultivation and cutting wood for fuel. Biogas is just a clever way of exploiting nature without - destroying it. It is well suited to household cooking. Biogas has been successfully used to several food cooking stoves with single or double biogas burners.

In this thesis, biogas injera baking stove which uses biogas as a source of energy is presented. The main objective is to develop a biogas stove for baking injera from domestic biogas stoves. Most of the domestic biogas stoves have single or double horizontal biogas burners. The designed (developed) injera baking stove has nine vertical burners.

According to the diameter of injera baking pan and amount of energy needed for baking injera, the nozzle diameter, number of burner and number of flame ports are designed for uniform heat distribution all over the radius of baking pan.

The developed (designed) stove is manufactured and the test is done using LPG as a source of energy due to unavailability of biogas.

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NOMENCLATURE

A_o	Orifice area
A_p	Flame port area
C	Circle circumference
C_d	Coefficient of Discharge
d_c	Cylinder diameter
d_o	Orifice diameter
d_r	Radius distance
d_s	Arc distance
d_t	Throat diameter
d_{lo}	Lower diameter of cone-frustum
d_{up}	Upper diameter of cone-frustum
h_b	Cone frustum height
h_c	Cylinder height
h_m	Cone frustum height
L_m	Length of mixing tube
n_p	Number of flame port
P_0	Atmospheric pressure
P_t	Throat pressure
$\dot{V}$	Gas flow rate
$\dot{V}_m$	Mixture flow rate
R	Entrainment ratio
R_e	Reynolds number
S	Specific gravity of gas
V_0	Orifice velocity
V_t	Throat velocity

Abbreviations

LEL	Lower Explosive Limit
UEL	Upper Explosive Limit
HHV	Higher Heating Value
LHV	Lower Heating Value
CNG	Compressed Natural Gas
VS	Volatile Solids
SFC	Specific Fuel Consumption

Greek letters

μ	Viscosity

ρ	Density
η	Efficiency

CHAPTER ONE

1.1 Introduction

Injera is a specific type of bread that is part and parcel to the Ethiopian culture. Injera is a necessity food item for almost all Ethiopians and is utilized in almost all-traditional cuisine. Food in Ethiopia consists of a number of traditional dishes. Injera is typically served with either meat or vegetable sauces.

The process of making injera requires the teff flour to be mixed with water and allowed to ferment for few days. This fermentation is a delicate process and is temperature and humidity sensitive [25]. Traditionally the prepared teff is baked either on a specialized electric stove or on a clay pan on biomass stove. The average power of the baking pan is 1.867 – 3.0 kW for electric heating coil for conventional pans. A conventional baking pan is 58 - 60cm in diameter and has 25 mm thickness approximately [8]. Use of wood as energy source for baking injera led’s to deforestation and the amount of electric consumption is high for electric stove. So people are trying to use biogas as energy source and it has many advantages.

Biogas is a type of gas that is formed by the biological breakdown of organic matter in an oxygen deficient environment. It is counted as an eco friendly bio fuel. Biogas contains 60% methane and carbon dioxide. It can be employed for generating electricity and also as automotive fuel. Biogas can be used as a substitute for compressed natural gas (CNG)[1].

The advantages of biogas are manifold. Biogas by itself can positively affect the economy of rural areas. The principal benefits of biogas include:

- *Conversion of natural organic waste into fertilizer*: The conversion is carried out in a digester. Cow dung slurry is put into the digester. The product is organic fertilizer of high quality. The fertilizer obtained is rich in nitrogen.
- *Eco friendly energy production*: Biogas is fully capable of replacing other rural energy sources like wood, hard coal, kerosene, plant residues, and propane. The calorific value of biogas is equal to that of half liter of diesel oil (6kWh/m3). Methane is a key component of the gas [17].
- *Considerable workload reduction in rural areas*: This is particularly true for rural women engaged in day to day household work.

- *Environmental benefits on a global scale*: Biogas plants significantly lower the greenhouse effects on the earth's atmosphere. The plants lower methane emissions by entrapping the harmful gas and using it as fuel.

Due to all benefits of biogas mentioned above there is an interest to use biogas as energy source for injera baking. Therefore, in this thesis, an attempt is made to develop injera baking stove which use biogas as energy source.

1.2 Statement of the Problem

Injera baking is the most energy intensive process. Cooking and baking accounts for over 50% of all primary energy consumption in the country and 75% of the total energy consumed in households. The wide spread use of wood cutting for fuel is the primary cause of deforestation in Ethiopia. About 95% of the total energy consumption in Ethiopia mostly comes from biomass fuels. Historically Ethiopia was one of the “forest” rich nations in the world. In just the past 50 years the land covered by forest has dropped from approximately 50% to less than 3%. Some experts attribute this mostly to forest clearing for cultivation and cutting woods for fuel [25].

To reduce this deforestation problem and to increase quality of injera electric stove has been developed which produce injera continuously or one at a time in home. But the amount of electrical energy consumption is high. So it is important to use other renewable energy sources like biogas for injera baking.

1.3 Objective

This work has the objective of designing an injera baking stove which uses biogas as fuel in order to substitute wood and electrical energy.

1.3.1 General Objective

The general objective of the study is to develop biogas injera baking bio gas stove.

1.3.2 Specific Objectives

The specific objectives of the study are: -

- To develop uniform flame distribution mechanism.
- To reduce heat loses to the surrounding.

- To determine the diameter of the nozzle.
- To determine the number of burners needed.
- To determine the number of flame ports according to port arrangement for uniform heat supply to the baking pan.
- To manufacture and test the designed stove.

1.4 Methodology

The methods to be employed to achieve the objectives of the research are: -

- Literature review;
- Design of biogas stove;
- Manufacturing the biogas stove;
- Testing of biogas stove;
- Interpretation of the test results.

CHAPTER TWO

LITERATURE REVIEW

In this chapter review of research work along the areas of this study found in literature is presented. It is basically a review on biogas utilization which is on the biogas demand and benefits; on the domestic biogas cooking stoves on different type biogas burners.

2.1 Biogas Utilization

2.1.1 Gas Demand

In developing countries, the household energy demand is greatly influenced by eating and cooking habits. Gas demand for cooking is low in regions where the diet consists of vegetables, meat, milk products and small grain. The gas demand is higher in cultures with complicated cuisine and where whole grain maize or beans are part of the daily nourishment.

As a rule of thumb, the cooking energy demand is higher for well-to-do families than for poor families. Energy demand is also a function of the energy price. Expensive or scarce energy is used more carefully than energy that is effluent and free of charge.

In industrialized countries, biogas replaces existing energy sources like electricity, diesel or other gases. The objective of biogas production may be less to satisfy a certain demand, but to produce biogas as much and as possible with low cost. Whatever surplus is available can be fed as electricity into the grid. The gas demand is thus market-driven in developed countries, while in developing countries, the gas demand is needs-driven [18].

2.1.2 Gas Production

Biogas originates from bacteria in the process of bio-degradation of organic material under anaerobic (without air) conditions. The natural generation of biogas is an important part of the biogeochemical carbon cycle. Methanogens (methane producing bacteria) are the last link in a chain of micro-organisms which degrade organic material and return the decomposition products to the environment. In this process biogas is generated, a source of renewable energy.

a. Biogas Production Process

Biogas microbes consist of a large group of complex and differently acting microbe species, notable the methane-producing bacteria. The whole biogas-process can be divided into three steps: hydrolysis, acidification, and methane formation (Figure 2.1). Three types of bacteria are involved (Figure 2.2) [17].

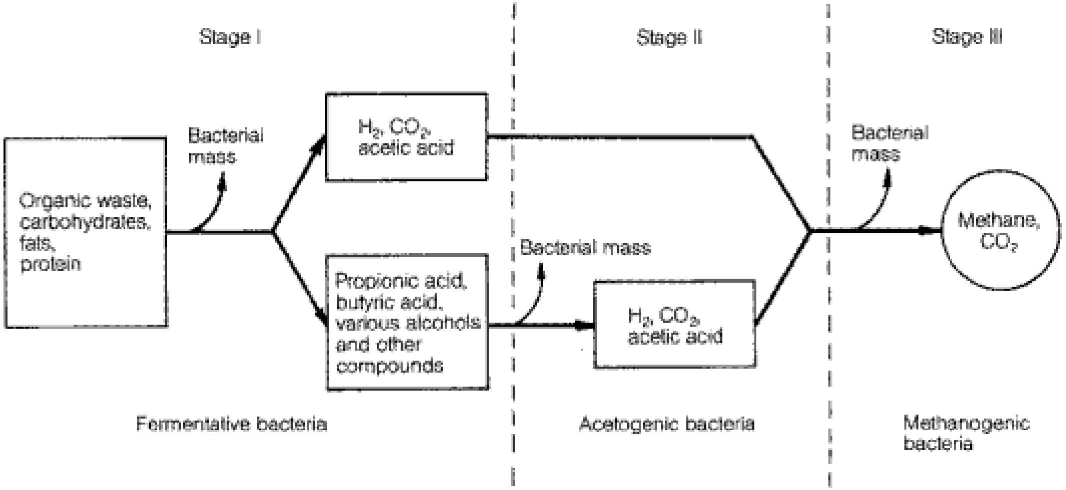

Figure 2.1: The three-stage anaerobic fermentation of biomass

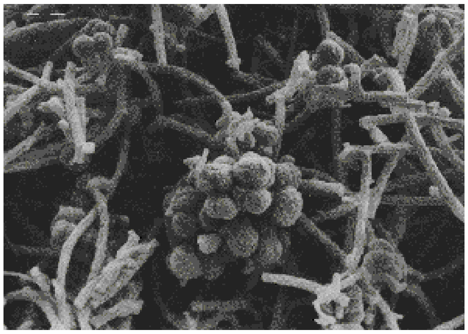

Figure 2.2: Various types of methanogenic bacteria.

i. Hydrolysis

In the first step (hydrolysis), the organic matter is enzymolyzed externally by extracellular enzymes (cellulose, amylase, protease and lipase) of microorganisms. Bacteria decompose the long chains of the complex carbohydrates, proteins and lipids into shorter parts. For example, polysaccharides are converted into monosaccharide. Proteins are split into peptides and amino acids.

ii. Acidification

Acid-producing bacteria, involved in the second step, convert the intermediates of fermenting bacteria into acetic acid (CH3COOH), hydrogen (H2) and carbon dioxide (CO2). These bacteria are facultative anaerobic and can grow under acid conditions. To produce acetic acid, they need oxygen and carbon. For this, they use the oxygen solved in the solution or bounded-oxygen. Hereby, the acid-producing bacteria create an anaerobic condition which is essential for the methane producing microorganisms. Moreover, they reduce the compounds with a low molecular weight into alcohols, organic acids, amino acids, carbon dioxide, hydrogen sulphide and traces of methane. From a chemical standpoint, this process is partially endergonic (i.e. only possible with energy input), since bacteria alone are not capable of sustaining that type of reaction.

iii. Methane Formation

Methane-producing bacteria, involved in the third step, decompose compounds with a low molecular weight. For example, they utilize hydrogen, carbon dioxide and acetic acid to form methane and carbon dioxide.

If the daily amount of available dung (fresh weight) is known, gas production per day in warm tropical countries will approximately correspond to the following values [18]: -

- 1 kg cattle dung 40 liters biogas
- 1 kg buffalo dung 30 liter biogas
- 1 kg pig dung 60 liter biogas
- 1 kg chicken droppings 70 liter biogas

b. Conditioning of Biogas

Sometimes the biogas must be treated /conditioned before utilization. The predominant forms of treatment aim at removing water, hydrogen sulfide, or carbon dioxide from the raw gas:

i. Reduction of the Moisture Content

The biogas is usually fully saturated with water vapor. This involves cooling the gas, e.g. by routing it through an underground pipe, so that the excess water vapor condenses at the lower temperature. When the gas warms up again, its relative vapor content decreases. The "drying" of biogas is especially useful in connection with the use of dry gas meters, which otherwise would eventually fill up with condensed water.

ii. Reduction of the Hydrogen-Sulfide Content

The hydrogen sulfide in the biogas combines with condensing water and forms corrosive acids. Water-heating appliances, engines and refrigerators are particularly at risk. The reduction of the hydrogen sulfide content may be necessary if the biogas contains an excessive amount, i.e. more than 2% H_2S. Since most biogas contains less than 1% H_2S, desulfurization is normally not necessary.

iii. Reduction of the Carbon-Dioxide Content

The reduction of the carbon-dioxide content is complicated and expensive.

c. Piping Systems

The piping system connects the biogas plant with the gas appliances. It has to be safe, economic and should allow the required gas-flow for the specific gas appliance. Galvanized steel (G.I.) pipes or PVC-pipes are most commonly used for this purpose. The operating pressure of most biogas handling systems will generally be less than 75 mms water column. However, if the system contains a compressor, some piping in the system could have an operating pressure as high as 30 bar [9].For a floating drum biogas plant that is required to operate at a gas pressure of 0.10 m W.C. [2]. Most prominently, the piping system has to be reliably gas-tight during the life-span of the biogas unit. In the past, faulty piping systems were the most frequent reason for gas losses in biogas units.

i. PVC Piping

PVC pipes and fittings have a relatively low price and can be easily installed. They are available in different qualities with adhesive joints or screw couplings (pressure water pipes). PVC pipes are susceptible to UV radiation and can easily be damaged by playing children. Wherever possible, PVC pipes should be placed underground.

ii. Galvanized Steel Piping

Galvanized steel pipes are reliable and durable alternatives to PVC pipes. They can be disconnected and reused if necessary. They resist shocks and other mechanical impacts. However, galvanized steel pipes are costly and the installation is labor intensive, therefore they are only suitable for places where PVC is unavailable or should not be used.

iii. Pipe Diameters

The necessary pipe diameter depends on the required flow-rate of biogas through the pipe and the distance between biogas digester and gas appliances. Long distances and high flow rates lead to a decrease of the gas pressure. The longer the distance and the higher the flow rate, the higher the pressure drops due to friction. Bends and fittings increase the pressure losses. G.1. pipes show higher pressure losses than PVC pipes. The table below gives some values for appropriate pipe diameters. Using these pipe diameters for the specified length and flow rate, the pressure losses will not exceed 5 mbar [18].

Table 2.1: Pipe diameter for different pipe lengths and flow-rate (maximum pressure loss <5 mbar)

	Galvanized steel pipe			PVC pipe		
Length [m]	20	60	100	20	60	100
Flow – rate [m^3/h]						
0.1	1/2"	1/2"	1/2"	1/2"	1/2"	1/2"
0.2	1/2"	1/2"	1/2"	1/2"	1/2"	1/2"
0.3	1/2"	1/2"	1/2"	1/2"	1/2"	1/2"
0.4	1/2"	1/2"	1/2"	1/2"	1/2"	1/2"
0.5	1/2"	1/2"	3/4"	1/2"	1/2"	1/2"
1.0	3/4"	3/4"	3/4"	1/2"	3/4"	3/4"
1.5	3/4"	3/4"	l"	1/2"	3/4"	3/4"
2.0	3/4"	l"	l"	3/4"	3/4"	l"

2.1.3 Biogas Benefits

a. Economy and Financing

In judging the economic viability of biogas programs and -units the objectives of each decision-maker are of importance. Biogas programs (macro-level) and biogas units (micro level) can serve the following purposes:

- the production of energy at low cost (mainly micro-level);
- a crop increase in agriculture by the production of bio-fertilizer (micro-level);
- the improvement of sanitation and hygiene (micro and macro level);
- the conservation of tree and forest reserves and a reduction in soil erosion (mainly macro-level);
- an improvement in the conditions of members of poorer levels of the population (mainly macro-level);
- a saving in foreign exchange (macro-level);
- provision of skills enhancement and employment for rural areas (macro-level).

b. Social Aspect

Biogas technology not only supports national economies and the environmental protection, but as its main outcome for the local population it provides for a wide range of improvements in overall living conditions. Sanitary and health conditions improve and the quality of nutrition is enhanced by improved energy availability. Through the provision of lighting and the reduction of time-consuming fuel gathering cultural and educational activities are supported. The women have benefited from the clear and smokeless energy for cooking, which has had a positive impact on the health and environment of the house. They are relieved of the drudgery and time spent to collect fuel wood. They have more time to take up other vocation and earn a better livelihood. They are able to spend quality time with their families at home [10, 19].

c. Benefits for the Environment

For many years the rational behind using biogas technology (or anaerobic technology) was the search for renewable sources of energy. In the meantime, other environmental protection aspects gain additional importance: A technology which previously just filled a "niche" is now becoming a key environmental technology for integrated, solid and liquid waste treatment concepts and climate protection both in industrialized and developing countries. Biogas technology is linked to the atmospheric budgets of many greenhouse gases. Another major environmental target is the mitigation of deforestation and soil erosion through the substitution of firewood as an energy source. The macro-economic benefits from biogas use in this field should be approached within the scope of the specific condition in the household energy sector and possible alternative protection measures.

d. Benefits for the Biogas Users

Individual households judge the profitability of biogas plants primarily from the monetary surplus gained from utilizing biogas and bio-fertilizer in relation to the cost of the plants. The following effects, to be documented and provided with a monetary value, should be listed as benefits:

- Expenditure saved by the substitution of other energy sources with biogas. If applicable, income from the sale of biogas;
- Expenditure saved by the substitution of mineral fertilizers with bio-fertilizer. Increased yield by using bio-fertilizer. If applicable, income from the sale of bio-fertilizer;
- Savings in the cost of disposal and treatment of substrates (mainly for waste-water treatment);
- Time saved for collecting and preparing previously used fuel materials (if applicable), time saved for work in the stable and for spreading manure (if this time can be used to generate income).

Table 2.2: The percentage benefit of biogas for households [10]

Benefit of Biogas	**Percent of Households**
Provides clean fuel for cooking	96
Cleanliness of environment	73
Improvement in the health of ladies	50
Saving in manure cost	74
Employment generation	8
Saving in cooking time	79
Saving in traditional fuel	60
Other benefits	3

2.1.4 Biogas Appliances

Biogas is a lean gas that can, in principle, be used like other fuel gas for household and industrial purposes, especially for:

- Gas cookers/stoves
- Biogas lamps
- Radiant heaters
- Incubators
- Refrigerators
- Engines

For the utilization of biogas, the following consumption rates in liters per hour (l/h) can be assumed:

- Household burners: 200-450 l/h
- Industrial burners: 1000-3000 l/h
- Refrigerator (100 l) depending on outside temperature: 30-75 l/h
- Gas lamp, equiv. to 60 W bulb: 120-150 l/h
- Generation of 1 kWh of electricity with biogas/diesel mixture: 700 l/h
- Plastics molding press (15 g, 100 units) with biogas/diesel mixture: 140 l/h

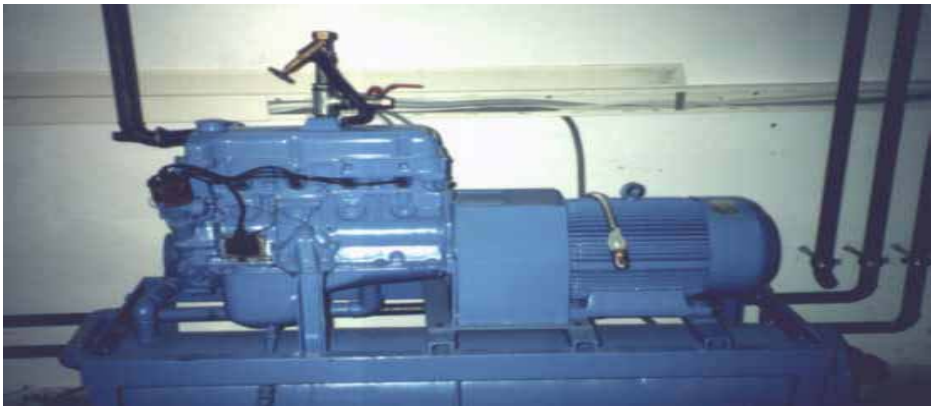

Figure 2.3: Co-generation unit (electricity and heat utilization)

2.2 Gas Cookers/Stoves

Biogas cookers and stoves must meet various basic requirements:

- Simple and easy operation
- Versatility, e.g. for pots of various size, for cooking and broiling
- Easy to clean
- Acceptable cost and easy repair

- Good burning properties, i.e. stable flame
- Attractive appearance

2.2.1 Biogas Burners

In developing countries, the main prerequisite of biogas utilization is the availability of specially designed biogas burners or modified consumer appliances. The relatively large differences in gas quality from different plants, and even from one and the same plant (gas pressure, temperature, caloric value, etc.) must be given due consideration.

The heart of most gas appliances is a biogas burner. In most cases, atmospheric-type burners operating on premixed air/gas fuel are preferable. Due to complex conditions of flow and reaction kinetics, gas burners defy precise calculation, so that the final design and adjustments must be arrived at experimentally. Compared to other gases, biogas needs less air for combustion. Therefore, conventional gas appliances need larger gas jets when they are used for biogas combustion [18].

The modification and adaptation of commercial-type burners is an experimental matter. With regard to butane and propane burners, i.e. the most readily available types, the following pointers are offered:

- Butane/propane gas has up to three times the caloric value of biogas and almost twice its flame-propagation rate.
- Conversion to biogas always results in lower performance values.

Practical modification measures include:

- Expanding the injector cross section by factor 2-4 in order to increase the flow of gas;
- Modifying the combustion-air supply, particularly if a combustion-air controller is provided;
- Increasing the size of the jet openings (avoid if possible).

The aim of all such measures is to obtain a stable, compact, slightly bluish flame.

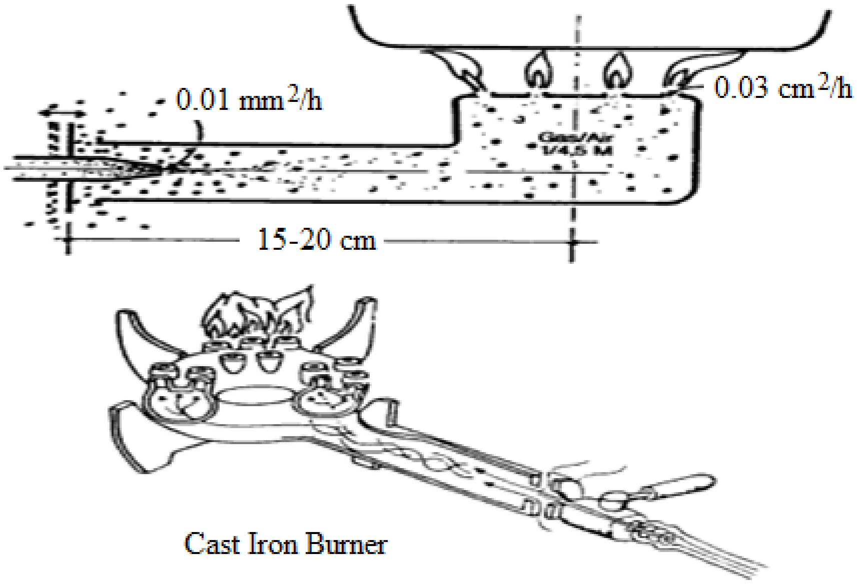

Figure 2.4: Schematic diagram of a gas burner

2.2.2 Two - Flame Burners

A cooker is more than just a burner. It must satisfy certain aesthetic and utility requirements, which can vary widely from region to region. Thus, there is no such thing as an all-round biogas burner. Most households prefer two-flame burners. The burners should be set initially and then fixed. Efficiency will then remain at a high practical level. Single-flame burners and lightweight cook-stoves tend to be regarded as stop-gap solutions until more suitable alternatives can be afforded.

Figure 2.5: Lightweight and stable 2-flame biogas burners

Biogas cookers require purposeful installation with adequate protection from the wind. Before any cooker is used, the burner must be carefully adjusted, i.e.:

- For a compact, bluish flame
- The pot should be cupped by the outer cone of the flame without being touched by the inner cone

- The flame should be self-stabilizing, i.e. flameless zones must re-ignite automatically within 2 to 3 seconds.

Test measurements should be performed to optimize the burner setting and minimize consumption.

Figure 2.6: Biogas burner in china

2.2.3 Efficiency

The calorific efficiency of using biogas is 55% in stoves, 24% in engines, but only 3% in lamps. A biogas lamp is only half as efficient as a kerosene lamp. The most efficient way of using biogas is in a heat-power combination where 88% efficiency can be reached. But this is only valid for larger installations and under the condition that the exhaust heat is used profitably. The use of biogas in stoves is the best way of exploiting biogas energy for farm households in developing countries [17].

Figure 2.7: Different types of Biogas burners at an agricultural exhibition in Beijing/China

Table 2.3: Efficiency of different biogas appliance

Appliances	Gas lamps	Engines	Gas stoves	Power-heat
Efficiency [%]	3	24	55	88

Table 2.4: Gas consumption of various biogas appliances [4]

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

2.3 Biogas Combustion

2.3.1 Approximate Fuel Value

Pure methane at standard temperature and pressure has a lower heating value of approximately 31.61 MJ/m^3. Typical biogas of 65% methane has a heating value of approximately 20.8 MJ/m^3 since only the methane portion will burn [9]. Approximate equivalents of biogas to other fuels are presented in Table 2.5.

Table 2.5: Fuel Equivalents of Biogas (per 28.3174 m^3)*

Fuel	Fuel Equivalents of Biogas(per 28.3174 m^3)*
Natural gas	16.9904 m^3
Propane	0.0250 m^3
Butane	0.0223 m^3
Gasoline	0.01780 m^3
#2 Fuel oil	0.01628 m^3
Bituminous coal	19.9583 kg
Medium-dry wood	45.3597 kg

* Biogas with 65% methane

2.3.2 Properties of Gases

The physical and chemical properties of biogas affect the choice of technology used for clean-up and combustion. Therefore, knowledge of these properties is useful for optimizing biogas utilization. Since biogas contains primarily methane and carbon dioxide their respective physical characteristics are listed in Table 2.6. Other components (nitrogen, hydrogen sulfide, and trace organics) are present in relatively small quantities, the magnitude of which vary greatly and depend on the composition of the material digested. Although the small concentration of these trace gases have little effect on the physical properties of the gas, they influence the choice of technologies. Therefore, individual components should be evaluated on a site-specific basis.

Table 2.6: Physical Constant of Methane and Carbon Dioxide [a] [9]

Property	Methane (CH_4)	Carbon Dioxide (CO_2)
Molecular Weight	16.04	44.01
Specific Gravity, Air = l [c]	0.554	1.52
Boiling Point @ 1 atm	399.3 K	316.15 K[b]
Freezing Point @ 1 atm	90.6 K	216.5 K
Specific Volume	1.51 m^3/ kg	0.55 m^3/ kg
Critical Temperature	319.8 K	304.3 K
Critical Pressure	45.8 atm	72.95 atm
Heat Capacity C_p 1 atm	2.2604 kJ/kg.K	0.8581 kJ/kg.K
Ratio C_p/C_v	1.307	1.303
Heat of Combustion	35.08 MJ/m^3	
	55529.71 kJ/kg	
Limit of Inflammability	5-15% by volume	
Stoichiometry in Air[c]	0.0947 by volume	
	0.0581 by mass	

a - Properties of pure gases given at 298.15K and atmospheric pressure

b - Sublimes

c - Air at 1 atm, 288.71K

a. *Volumetric Compensation*

Volumetric measurement of biogas, like any gas, must be compensated for pressure and temperature differences. The equation below (Salisbury 1950) illustrates a simple method of gas volume compensation for a saturated gas:

$$V_s \text{ (sat.)} = V \times 17.626 \times ((H - A)/ (459.6 + T))$$

Where:

V = observed volume

V_s = volume at standard conditions, 288.71 K and 0.762 m Hg

H = absolute gas pressure, m Hg

A = water vapor pressure, m Hg, for gas at temperature T

T = temperature of gas, K

b. Flame Velocities

A major consideration in analyzing gaseous fuels, particularly those such as biogas with low energy contents due to dilution with various non-combustible gases, is the flame velocity of that fuel during combustion. Flame velocity is defined as the speed at which a flame progresses into a mixture relative to the speed of the fuel mixture. It is important in the design of systems for feeding fuel and air to burners and in the setting of the spark advance for internal combustion engines.

The impact of carbon dioxide concentrations on flame velocities over the limits of inflammability of a methane/carbon dioxide mixture are illustrated in Figure 2.8. The information can be used to compare the performance of a combustion system designed for natural gas that will be modified for operation on biogas.

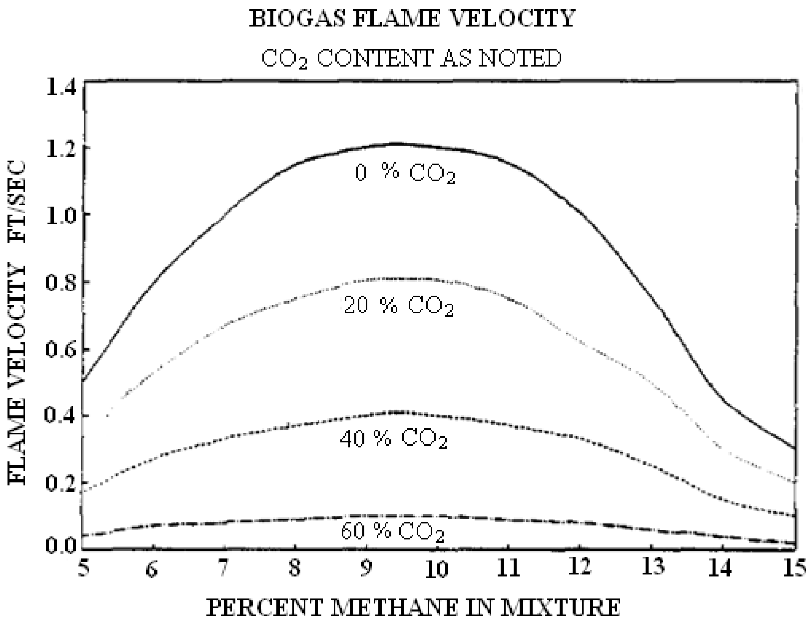

Figure 2.8: Flame velocity as a Function of Carbon Dioxide Concentration [9]

c. Flammability Limits

Flammability limits (or limits of inflammability) indicate the maximum and minimum percentages of a fuel in a fuel and air mixture at which the mixture will burn. This is a critical parameter in biogas combustion due to the dilution of methane in a biogas fuel with carbon dioxide and other inert gases.

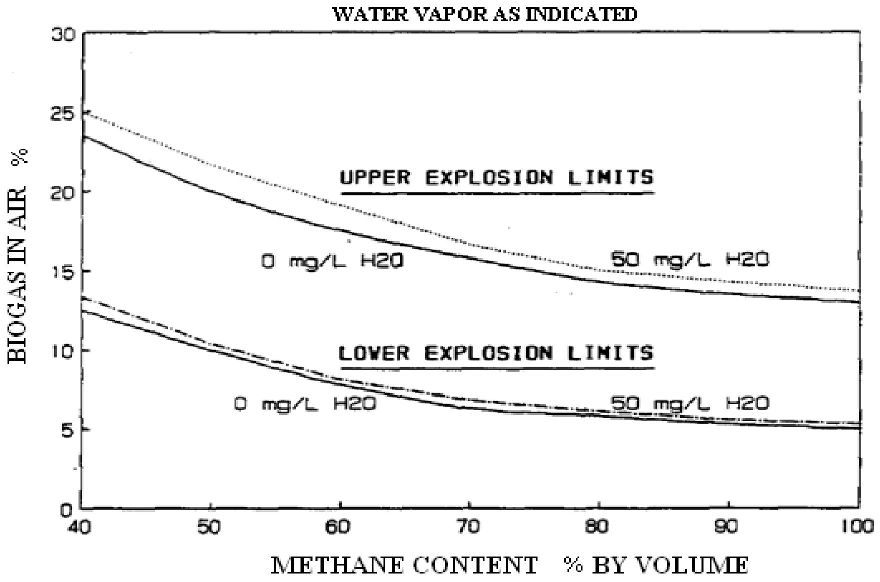

Figure 2.9: Flammability limits as a function of carbon dioxide and water vapor concentration [9]

The flammability limits of methane are listed in Figure 2.9, and range from 5% to 15% in air. These two values are also known as the lower explosive limit (LEL) and upper explosive limit (UEL), respectively. The impact of CH_4 dilution (by CO_2 and water vapor) on flammability limits are illustrated in Figure 2.9.

d. Flame Temperatures

The temperature of the flame front created by a combustible mixture is important with respect to the performance of all types of combustion systems. In the operation of boilers, flame temperature (sometimes referred to as hot mix temperature) is an indication of thermal efficiency. The temperature of the exhaust gases from a combustion system will affect the potential for heat recovery and the formation of nitrogen oxides in the exhaust. The theoretical flame temperature of methane in a stoichiometric mixture with air and including dissociation is 2190.93 K [9]. However, the theoretical flame temperature decreases as the concentration of non-combustibles increases; accordingly, the theoretical flame temperature as a function of water vapor and methane content is shown in Figure 2.10.

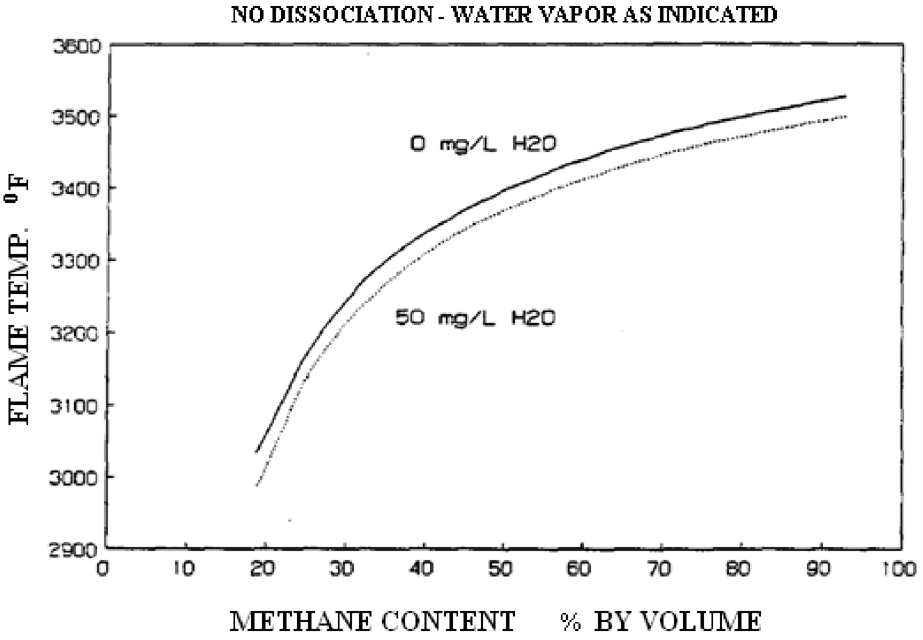

Figure 2.10: Theoretical flame temperatures as a function of methane and water vapor concentration [9]

e. Heating Value

The gross and net heating values of simple fuels are important in defining the energy available from different gases and are compared in Table 2.7. Since different gases have different heating values, calculation of the net heating value of a mixture such as biogas must take into account not only the amount of methane but also all other combustible and non-combustible gases. The higher heating value (HHV) is the energy released from a given mass of a fuel where the energy required to vaporize the water in the fuel is recovered. The HHV of methane, the primary combustible in biogas, is listed as 1067.66 kJ/ SCF. The lower heating value (LHV) is defined as the higher heating value less the energy required for the vaporization of water in the fuel and combustion products. For methane, the net or lower heating value is 962.16 kJ/SCF [9]. The effect of biogas moisture content and CH_4 content on the net heating value of biogas is illustrated in Figure 2.11**.** A comparison of energy values for several commercial fuels is provided in Table 2.8.

Table 2.7: Comparative fuel values for several simple fuels

Fuel	**Heating Values**		**Air-Fuel Ratio**	
	MJ/m^3	**kJ/kg**		
	Higher Heating Values (Lower Heating Values)		**Vol. Air / Vol. Fuel**	**Wt. Air / Wt. Fuel**
Butane, n-C_4H_{10}	113.38 (104.61)	49589.49 (45768.11)	31.0	15.50
Hydrogen, H_2	11.27 (9.53)	142097.9 (120067.4)	2.38	34.50
Hydrogen Sulfide, H_2S	22.39 (20.62)	16506.57 (15204.09)	7.15	6.08
Methane, CH_4	35.08 (31.58)	55529.71 (49994.19)	9.53	17.20
Octane, C_8H_{18}	216.98 (201.25)	48368.41 (44868.01)	-------	15.10
Propane, C_3H_8	87.49 (80.49)	50398.88 (46370.51)	23.8	15.70

Table 2.8: Comparison of fuel values for commercial fuels

Fuel	**Heating Values**		**Air-Fuel Ratio**	
	kJ/kg (MJ/m^3)			
	Higher	**Lower**	**Wt. Air / Wt. Fuel**	**SCF/ m^3**
Natural Gas	50773.35	45807.65	15.73	-------
Gasoline	46958.95	43702.76	14.80	312513.11
	(6056.86)	(29766.25)		
Diesel (#2)	44174.9	41528.08	14.35	357.69
	(35541.02)	(33412.14)		
Fuel Oil (#4)	43921.39	41376.9	13.99	366.67
	(37078.5)	(35005.1)		
Propane	50175.6	46251.89	15.35	224808.67
	(23723.4)	(21868.305)		

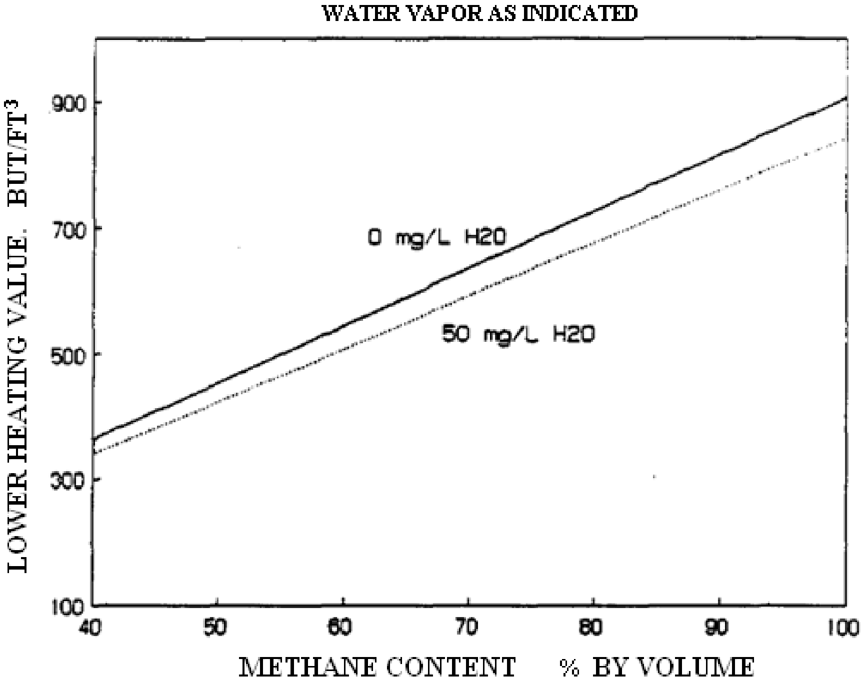

Figure 2.11: Lower heating values as a function of methane and water vapor content [9]

f. Fuel Mixtures

As described under flammability limits, methane and air mixtures will combust between 5% and 15% methane in air. The optimum concentration of CH_4 in air often referred to as the stoichiometric mixture.

- Air-Fuel Ratio (AF) is defined as:

$$\text{Air-Fuel Ratio (AF)} = \text{(mass flow rate of air)/ (mass flow rate of fuel)} \tag{2.1}$$

- Equivalence ratios (Φ): are used to describe the degree of variation from the stoichiometric ratio, from excess air to excess fuel. The equivalence ratio is defined as:

$$\text{Equivalence Ratio } \Phi = \text{(AF stoichiometric)/ (AF Actual)} \tag{2.2}$$

Where:

$\Phi = 1$ is a stoichiometric ratio

$\Phi < 1$ is a lean mixture, excess air

$\Phi > 1$ is a rich mixture, excess fuel

The stoichiometric ratio of biogas will obviously vary with the amount of noncombustible gases mixed with the methane. The variation Air-Fuel Ratio for biogas as a function of the methane and water vapor content are illustrated in Figure 2.12.

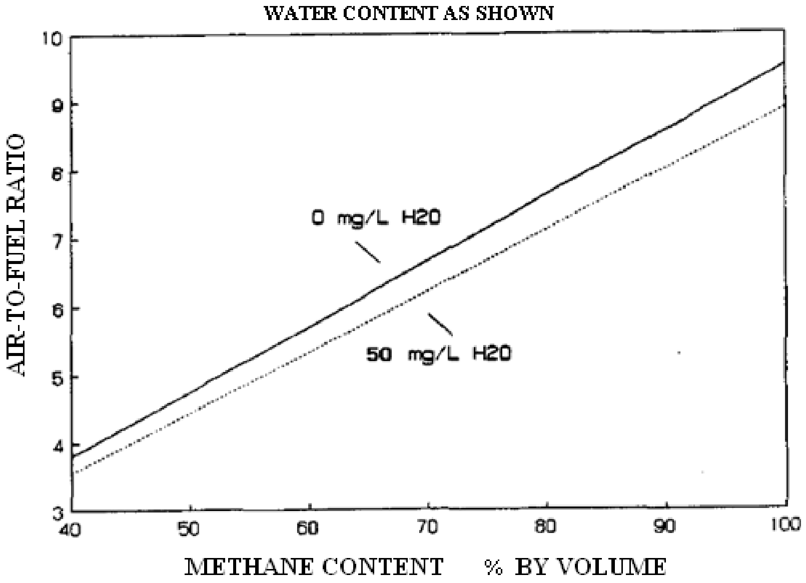

Figure 2.12: Air- Fuel ratio variation [9]

g. Water Vapor

While not as prevalent a diluents as carbon dioxide, water vapor can have a significant effect on biogas combustion characteristics. As illustrated in Figure 2.8 through 2.12, water vapor has a small but noticeable impact on flame temperature, flammability limits, lower heating value, and Air-Fuel Ratios of biogas.

These effects plus the potential for corrosion highlight the need for water vapor reduction in biogas. Depending on temperature, biogas samples immediately after the outlet from a digester may contain as much as 50 mg/L water vapor, which is near the saturation level [9].

CHAPTER THREE

BIOGAS STOVE DESIGN

3.1 Theory for a Simple Gas Burner Design

The force which drives the gas and air into the burner is the pressure of gas in the pipeline. The key equation that relates gas pressure to flow is Bernoulli's theorem (assuming incompressible flow):

$$\frac{p}{\rho g} + \frac{v^2}{2g} + z = \text{constant} \qquad 3.1$$

Where:

P is the gas pressure (N m^{-2}),

ρ is the gas density (kg m^{-3}),

V is the gas velocity (m s^{-1}),

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

Z is head (m). For a gas, head (*z*) can be ignored.

Bernoulli's theorem essentially states that for an ideal gas flow, the flow energy due to the pressure, plus the kinetic energy due to the velocity of the flow is constant. In practice, with gas flowing through a pipe, Bernoulli's theorem must be modified. An extra term must be added to allow for energy loss due to friction in the pipe:

$$\frac{p}{\rho g} + \frac{v^2}{2g} - \text{f(losses)} = \text{constant} \qquad 3.2$$

Using compressible flow theory, flow though a nozzle of area *A* is:

$$\dot{m} = C_d \rho_0 A \sqrt{2\left(\frac{\gamma}{\gamma-1}\right)\frac{p_0}{\rho_0} r^{2/\gamma}\left(1 - r^{(\gamma-1)/\gamma}\right)} \qquad 3.3$$

Where:

p_0 and ρ_0 are the pressure and density of the gas upstream of the nozzle and

P_1 is the pressure downstream of the nozzle.

$$r = {}^{p_1}/{p_0}$$

3.1.1 Burner Power

The equation used to find the burner power (kW) is given by:

$$\dot{Q} = 39000 * Cd * Ao * W * \sqrt{p} \qquad 3.4$$

Where:

$\dot{Q}$ is the burner power (kW)

Cd is the discharge coefficient

Ao is the jet area (m^2)

W is the wobbe number for biogas (= 22.2 kJ/l) and

p is the supply pressure (mm WG)

The efficiency of the burner is given by:

$$\text{Efficiency} = \text{Power Out} / \text{Power In} * 100 \qquad 3.5$$

3.1.2 Injector Orifice or Jet

The amount of gas used by a burner is controlled by the size of the gas “jet” or “injector orifice” (an orifice is a hole in a plate). This is usually a brass thimble with a hole drilled in the end, screwed onto the end of the gas line fitting, so that it can be easily replaced. As well as controlling the gas flow rate, the injector has the second important role of separating the burner from the gas supply. It should be impossible for a flame to enter the gas supply pipe. Injectors on larger burners may have more than one hole, mainly to reduce noise [4].

The gas flow rate $\dot{V}$ is related to the gas velocity (v) by the area (A) of the pipe through it is flowing:

$$\dot{V} = V * A \qquad 3.6$$

For gas flow through an orifice, a sudden change in flow area causes a “vena contracta”, a narrowing of the flow to an area smaller than that of the hole itself:

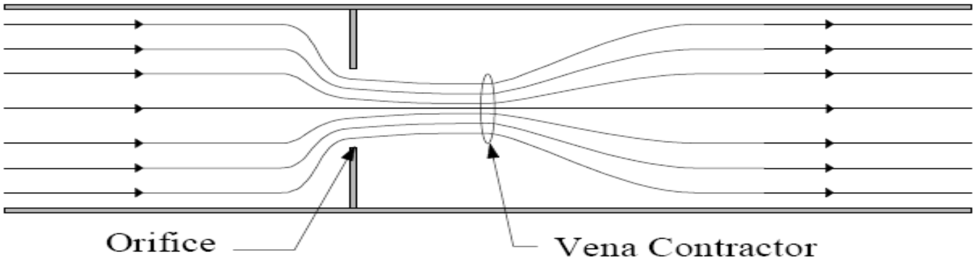

Figure 3.1: Gas flow stream through an orifice [4]

An orifice plate can be used to measure gas flow over a very wide range of flow conditions, including very high flow rates.

a. Gas Flow through an Injector Orifice (Jet)

An empirical version of Bernoulli's theorem is used to define the flow rate:

$$\dot{V} = 0.0467 C_d A_0 \sqrt{\frac{p}{s}} \tag{3.7}$$

Where:

$\dot{V}$ = gas flow rate (m^3h^{-1}),

A_0 = area of orifice (mm^2)

p = gas pressure before orifice (mbar)

s = specific gravity of gas

C_d = coefficient of discharge for the orifice.

The coefficient of discharge for the orifice takes into account the vena contractor and friction losses through the orifice. It usually has a value between 0.85 and 0.95[4].

b. Orifice Design

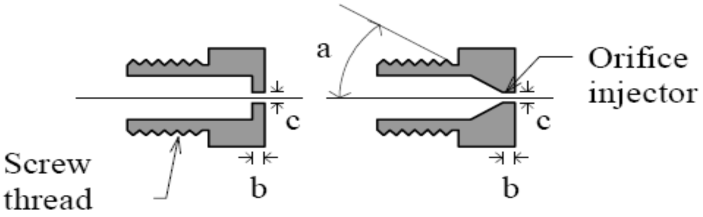

Figure 3.2: Types of orifice design [4]

To maximize C_d, the angle (*a*) of approach before the orifice should be 30° and the length of the orifice channel (*b*) should be between 1.5 and 2 times the orifice diameter (*c*).

To ensure accuracy, each jet is usually calibrated individually using a fixed pressure air supply and a flow meter and its value of C_d marked on it.

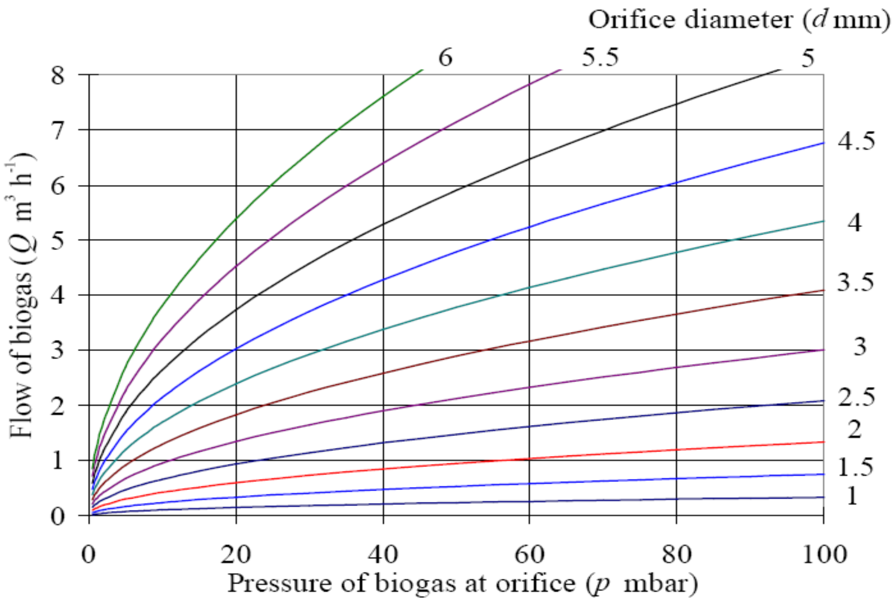

Figure 3.3: Graph of $\dot{V} = \mathbf{0.036C_d d^2}\sqrt{\frac{\mathbf{p}}{\mathbf{s}}}$ where d is orifice diameter (mm),Cd is 0.9

3.1.3 Flames

As gas comes out of the injector, air is “entrained” into the stream and is mixed in the mixing tube with the gas before it comes out of the burner port. The unburned gas is heated up in an “inner cone” and starts burning at the “flame front”. The cone shape is a result of laminar flow in a cylindrical mixing tube, the mixture at the centre of the tube is moving at a higher velocity than that at the outside.

The main “combustion zone” is where the gas burns in the primary air and generates the heat in the flame. The “Outer mantle” of the flame is where combustion is completed with the aid of the secondary air that is drawn into the flame from the sides.

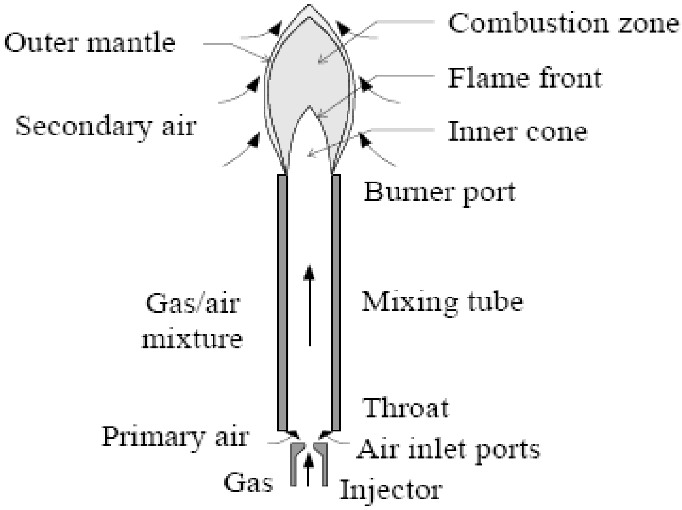

Figure 3.4: Combustion process in the gas burner [4]

The combustion products (carbon dioxide and steam) are at a high temperature, so rise vertically away from the flame, transferring heat to the air close to the top of the flame. It is this air moving vertically away that draws in the cooler secondary air to the base of the flame.

The size of the inner cone depends on the primary aeration. A high proportion of primary air makes the flame much smaller and concentrated, giving higher flame temperatures.

3.1.4 Entrainment

The gas emerging from the injector enters the end of the mixing tube in a region called the "throat". The throat has a much larger diameter than the injector, so the velocity of the gas stream is much reduced.

The velocity (v_0) of the gas in the injector orifice is given by:

$$v_0 = \frac{\dot{V}}{3.6 * 10^{-3} A_0} {}^{m}/_{s} \qquad 3.8$$

Where: -

$\dot{V}$ is in $({}^{m^3}/_{h})$ and A_0 is in (mm^2).

Ignoring the vena contracta and friction, the velocity in the throat is reduced to:

$$v_t = v_0 \frac{A_0}{A_t} = v_0 \frac{d_0^2}{d_t^2} \qquad 3.9$$

The gas pressure just after the nozzle then becomes:

$$p_t = p_o - \rho \frac{v_o^2}{2g}\left[1 - \left(\frac{d_o}{d_t}\right)^4\right] \qquad 3.10$$

The value of P_0 is around atmospheric pressure, as the throat is open to the air, so this drop in pressure is sufficient to draw primary air in through the air inlet ports to mix with the gas in the mixing tube. The value of ρ is the density of biogas (use $\rho = 1.0994\ \text{kg m}^{-3}$) [4].

The primary aeration depends on the "entrainment ratio" (r), which is determined by the area of the throat and the injector (Prigg's formula):

$$r = \sqrt{s}\left(\sqrt{\frac{A_t}{A_0}} - 1\right) = \sqrt{s}\left(\sqrt{\frac{d_t}{d_0}} - 1\right) \qquad 3.11$$

Where: A_t and d_t are the area and diameter of the throat and A_0 and d_0 are the area and diameter of the injector.

Prigg's formula holds if the total flame port area (Ap) is between 1.5 and 2.2 times the area of the throat. This ratio is approximately independent of the gas pressure and the flow rate. The primary air supply is rarely enough to give a stochiometric mixture.

a. Throat Size

The size of the throat (d_t) is can be found using Prigg's formula. The flow rate of the mixture in the throat $\dot{V}_m$ is then given by:

$$\dot{V}_m = \frac{\dot{V} * (1+r)}{3600} \qquad 3.12$$

Where:

$\dot{V}_m$ is in ($m^3\ s^{-1}$) and $\dot{V}$ is in ($m^3\ h^{-1}$).

The pressure drop due to the flow of the mixture down the mixing tube should be checked, by first calculating the Reynolds number:

$$R_e = \frac{\rho d_t v_t}{\mu} = \frac{\rho d_t}{\mu}\frac{4\dot{V}_m}{\pi d_t^2} = \frac{4\rho \dot{V}_m}{\pi \mu d_t} \qquad 3.13$$

Where: ρ and μ are the density and viscosity for the mixture (use $\rho = 1.15\ \text{kg m}^{-3}$ and $\mu = 1.71 * 10^{-5}$ Pa at 30°C) [4].

The pressure drop (Δp) is then given by:

$$\Delta p = \frac{f}{2}\rho v_t^2 \frac{L_m}{d_t} = \frac{f}{2}\rho \frac{16\dot{V}_m^2}{\pi^2 d_t^5} L_m \tag{3.14}$$

$$\text{When Re} < 2000,\ f = \frac{64}{R_e} \tag{3.14.a}$$

$$\text{When Re} > 2000,\ f = \frac{0.316}{R_e^{1/4}} \tag{3.14.b}$$

The pressure drop should be much less than the driving pressure.

Most burners are designed to have a throat that gives aeration greater than optimum, with a device for restricting the air flow, so the optimum aeration can be set for a given situation:

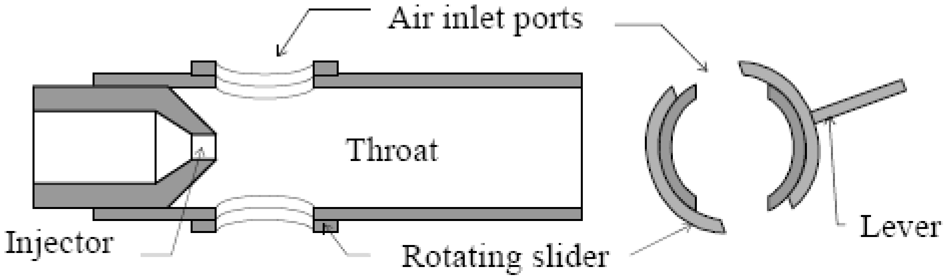

Figure 3.5: Lever and rotating slider system of air flow control mechanism

A simple method of air control on a cylindrical mixing tube is to make the air inlet ports as holes in the cylinder wall, at right angles to the length of the cylinder. These holes should be horizontal, rather than vertical, to prevent gas seeping out at low flow rates. The holes can be partially covered by a concentric section of cylinder, with identical holes in it that can be rotated by a lever. The maximum area of the holes should be larger than the cross-sectional area of the throat.

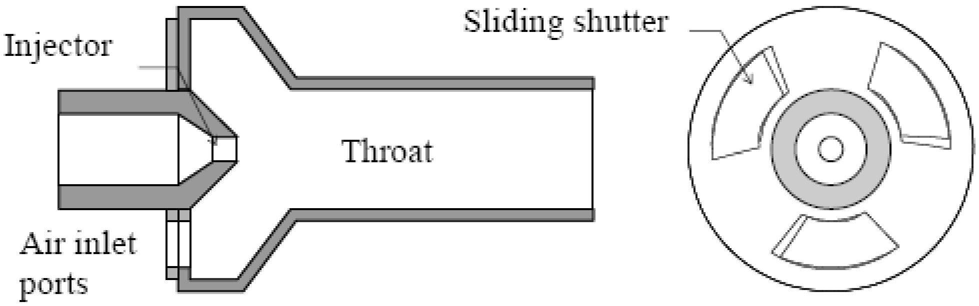

Figure 3.6: Sliding shutter and slots in a flat disk system air flow control mechanism

A more complex method to do the same job is to make slots in a flat disk that fits behind the gas injector or to mount a disk on a thread on the injector pipe, so the air port can be opened and closed by rotating the disk up and down the screw.

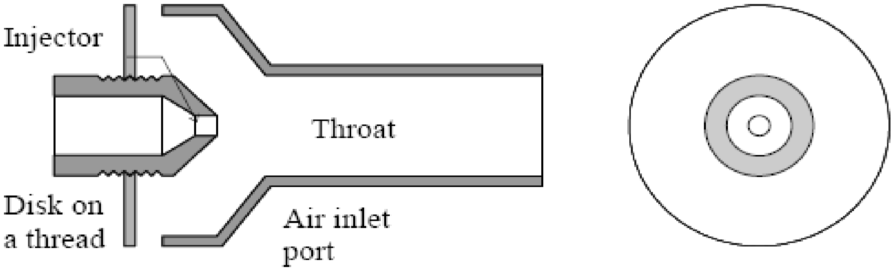

Figure 3.7: Screw system of air flow control mechanism

b. Mixing Tube

For a cylindrical throat, the mixing tube must be long enough to allow good mixing of the gas and air. The recommended length of mixing tube is given by:-

$$L = 10 * d_t \qquad 3.15$$

c. Venturi

Another way of making the mixing tube is as a "venturi" or "diffuser", with a pipe that tapers into the throat and tapers smoothly away again:

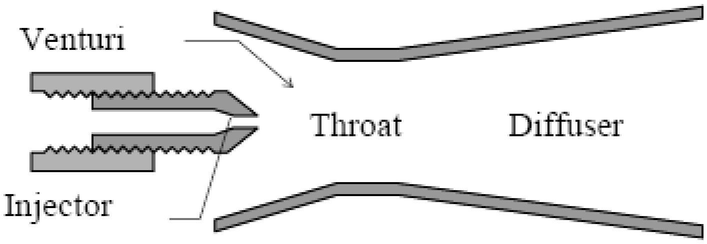

Figure 3.8: Screwing system in the venturi or diffuser of air flow mechanism

The air flow can be adjusted by screwing the injector into or out of the throat, or by moving the throat relative to the injector.

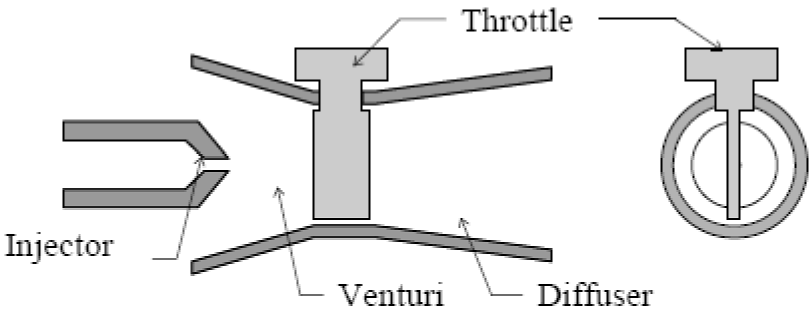

Figure 3.9: Throttle system in the venturi or diffuser of air low control mechanism

The air flow in the venturi can also be controlled by fitting a “throttle”, either a vane that can be turned or a screw that can be screwed in to block the throat.

A venturi can be shorter than a cylindrical mixing tube (6*d_t), so is often used where space is limited, such as in lamps.

3.1.5 Burner Ports

The big advantage of a gas burner is that the heat can be directed to where it is needed, by designing the burner properly. However, the design must allow for particular problems that can occur when burning gas, especially biogas.

I. *Lighting Back*

It is possible for the flame at a burner port to travel back down the mixing tube to the injector. This is called “lighting back”.

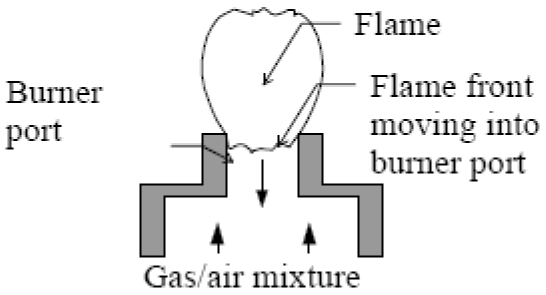

Figure 3.10: Lighting back at the burner port

The way to overcome lighting back is to choose a burner port size smaller than a certain size. For ports in thin metal; this will be 2.5 mm diameter for natural gas. If the burner port is drilled in thicker metal, then it can be larger.

Because biogas has such a low flame speed, lighting back is not usually a problem. 5 mm diameter holes in 5 mm thick metal do not seem to light back [4].

II. Flame Lift

The opposite effect is a real problem with biogas, that of the flame lifting off the burner port:

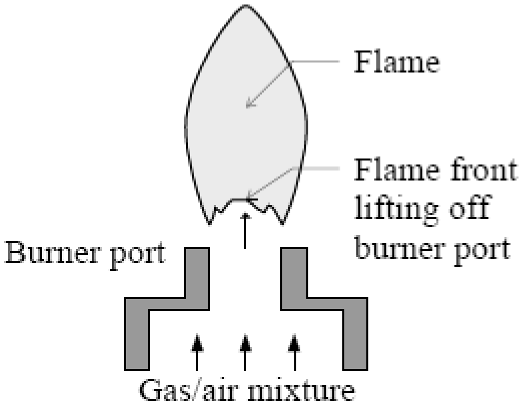

Figure 3.11: Flame lift off from the burner port

The flame lifts off from the port and can "blow-off" and go out. "Flame lift" occurs when the speed of the gas/air mixture through the burner port is higher than the speed of the flame burning in the gas. Biogas has a stochiometric flame speed of only 0.25 m s^{-1}, so the total flame port area must be chosen so that the mixture velocity through the ports is much lower than this Figure. The flame velocity at the flame front is likely to be 50% of the stochiometric value, as the flame is not fully aerated at this point [4].

Even if the burner port size is designed correctly for a particular situation, a variation in conditions can result in flame lift. Alterations in the entrainment ratio, caused by adjustments in the primary air controls, or by partial blockage of the air inlets by dirt, can cause the flame velocity at the flame front to change. Increased supply pressure will increase the mixture flow rate and velocity, also causing flame lift.

The mixture supply velocity (v_p) is given by:

$$v_p = \frac{\dot{V}_m}{A_p} \ll 0.25 \ {}^{m}/_{s} \qquad 3.16$$

The total burner port area (A_p) is given by:

$$A_p = n_p \frac{\pi d_p^2}{4} \ m^2 \qquad 3.17$$

Where: -

n_p is the number of ports, each of diameter d_p in m.

a. Burner Port Design

The total area of the burner ports is limited by the need to prevent flame lift, as above. The size and positioning of the individual burner ports are defined by various factors, such as the heat pattern required, the need for burner ports to be close enough together for cross-lighting and the need for an adequate supply of secondary air.

Domestic stoves, used mainly for cooking, usually have burner ports arranged in a circular pattern, as most cooking pots have a circular base. The size of the circle depends on the average size of the cooking pots to be used. Water heaters usually use one or more bar burners arranged under a rectangular boiler.

i. Cross-Lighting

A burner is usually lit at one place, so the flames should jump from one burner port to the next, so the whole burner is alight. Also the flames at individual burner ports may go out, so cross-lighting is essential.

For biogas, the gaps between burner ports should be around 5 mm to ensure cross-lighting occurs.

ii. Secondary air supply

The pattern of burner ports should allow secondary air to reach each port without interference.

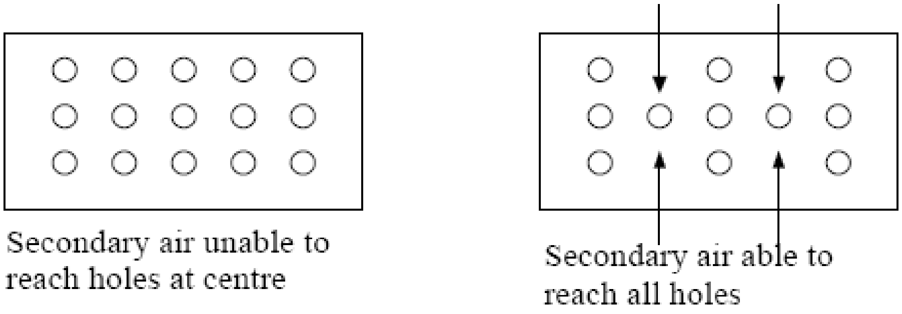

Figure 3.12: Flame port arrangement for the secondary air supply

The first pattern would produce a poor burning pattern, with the flames from the central burner ports being much higher than those at the edge because secondary air is prevented from reaching them. The second pattern allows air to reach each of the burner holes.

iii. Flame Stabilization

Several methods can be used to reduce the problem of flame lift. The supply of secondary air to the flame can be increased by putting the burner ports in a raised ledge, or by putting them at an angle to the horizontal:

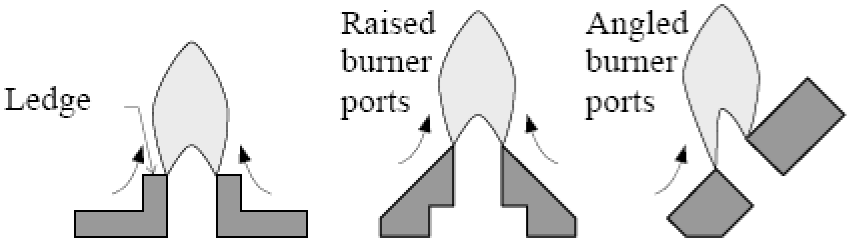

Figure 3.13: Flame stabilization methods by putting the burner ports in a raised ledge or at an angle to the horizontal

The second method uses retention flames, small flames arranged around the main flame to hold it onto the burner port. The velocity of the mixture entering these smaller burner ports is often reduced by increasing the friction losses into these ports, using "metering orifices":

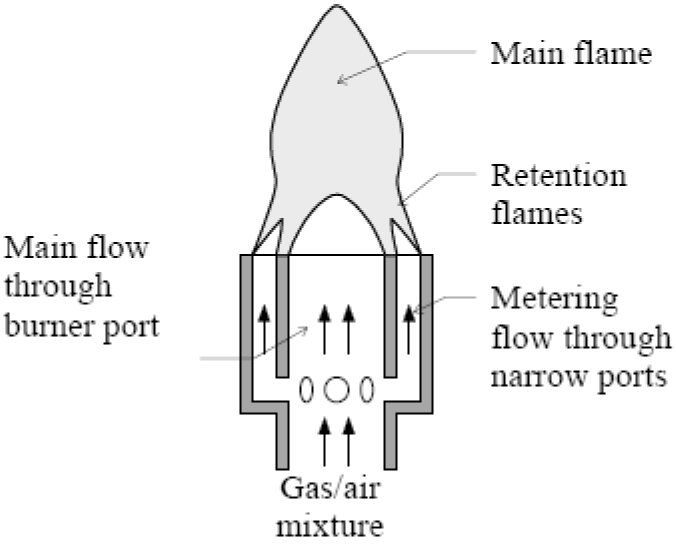

Figure 3.14: Flame stabilization method by using retention flames

The third method uses a sudden change in flow area at the burner port to give eddies, that help in flame stabilization:

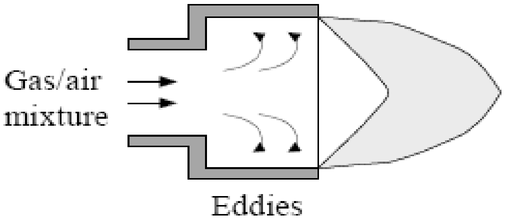

Figure 3.15: Flame stabilization by using sudden changes in the flow area

b. Burner Manifold

The flow of the gas/air mixture through each of the burner ports must be uniform, so each burner port should be of the same size. Also the pressure drop in the supply pipes leading to the burner ports must be of the same value. The usual way to ensure this is to use a manifold that is symmetrical and with a cross-sectional area that is much larger than the total flame port area:

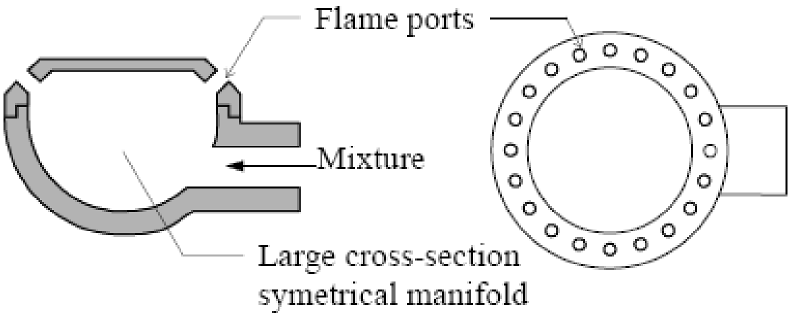

Figure 3.16: Circular burner manifold

For a bar burner, with the flame ports arranged in line on a cylindrical or rectangular tube, it is common to place the mixing tube so the mixture comes out at the centre of the manifold:

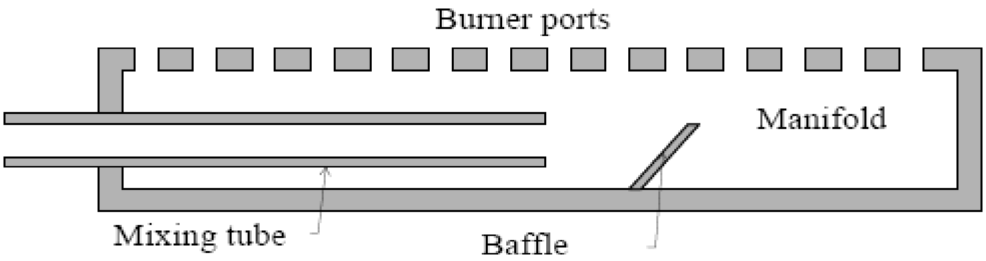

Figure 3.17: A bar burner manifold

c. Baffle

Baffles may be required to balance the flow patterns within the manifold, so the flame size is uniform. Burner ports are often round in shape, but can be made any shape.

3.1.6 Pot Supports

The flame must be at a high temperature for the combustion reaction to be sustainable. If the flame is cooled, the reactions are "quenched" and the reactions are incomplete. Biogas burning in air will produce carbon monoxide and carbon particles (soot) if the reaction is quenched.

Quenching is useful, as it prevents lighting back in burner ports that are of the correct size. The flame cannot pass through the port as the metal cools it.

The correct positioning of the object to be heated (e.g. a pot of food to be cooked) above the flame is therefore important. If the object is too close to the flame, the flame is quenched and the combustion is incomplete and the efficiency of the stove is reduced. If the object is too far away from the flame, heat is lost to the atmosphere and the stove is again less efficient. The best position for the base of the object being heated is just above the tip of the visible flame, just outside the outer mantle, above the hottest part of the flame [4].

The flame height, though, depends on different parameters. The major parameter is the velocity of the gas/air mixture through the burner ports, which depends on the size of the burner ports and the gas pressure. The degree of primary aeration of the burner affects both the mixture velocity and the height of the inner core of the flame, which in turn affects the full flame height. Greater primary aeration will reduce the flame height.

In practice, the position of the object to be heated needs to be designed once a prototype burner has been made and the flame length for typical conditions has been measured. The design of the pot support height for domestic stoves, for example, may need to be left until the rest of the stove can be made and tested.

A typical value for the height between the flame ports and the pot base was 25 to 30 mm for 5 mm burner ports, using biogas at 10 mbar pressure. Smaller flame ports should lead to shorter flames.

3.2 Design of Biogas Stove

3.2.1 Biogas Combustion

Biogas burns in oxygen to give carbon dioxide and water:

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

One volume of methane requires two volumes of oxygen, to give one volume of carbon dioxide and two volumes of steam. Since there is 60% methane in biogas and 21% oxygen in air:

$$\frac{1}{0.60} = 1.67 \text{ Volumes of biogas required.}$$

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

1 volume of biogas requires $\frac{9.52}{1.67} = 5.7$ volumes of air (stoichiometric air requirement).

Therefore, the stoichiometric air-fuel ratio is 5.7.

Biogas will burn over a fairly narrow range of mixtures from 9% to 17% biogas in air. If the flame is “too rich”, has too much fuel, then it will burn badly and incompletely, giving carbon monoxide (which is poisonous) and soot (carbon particles). Burners are usually run “slightly lean”, with a small excess of air, to avoid the danger of the flame becoming rich. In most burners, air is mixed with the gas before it is burnt in a flame (pre-aeration). Post-aerated flames, where the gas is ignited at the end of the gas line, give very poor combustion. The amount of “primary air” added to the gas before the flame, varies depending on the design of burner, but is usually around 50% of the total air requirement [4].

3.2.2 Biogas Burner Design

The stove designed to supply about 3 kW (Output power) for baking injera using biogas. For the biogas stove the estimated calorific efficiency is 55% as indicated in Table 2.3. So the required heat input power is calculated using equation 3.5.

$$\text{Efficiency} = \frac{\text{output power}}{\text{input power}} * 100$$

$$55 = \frac{3 \text{ kW}}{\text{input power}} * 100$$

Therefore, the required input power is 5.45 kW or 19.6 MJ/h. The calorific value (Cv) of biogas is equal to that of half liter of diesel oil 6kWh/m^3 (21.6MJ/m^3).

Thus the flow rate required is given as:

$$\dot{V} = \frac{\text{input power}}{\text{Cv}} = \frac{19.6\ \text{MJ/h}}{21.6\text{MJ/m}^3} = 0.91\frac{\text{m}^3}{\text{h}}$$

The biogas flow rate required is $\dot{V}= 0.91\frac{m^3}{h}$ and a gas supply pressure of 50 mbar.

3.2.3 Design of Burner Injector

a. Determine the Size of Injector

The injector size is calculated using equation 3.7:

$$\text{d}_\text{o} = \sqrt{\frac{\dot{V}}{0.036 * \text{Cd}}} * \sqrt[4]{\frac{\text{s}}{\text{p}}}$$

$$\text{d}_\text{o} = \sqrt{\frac{0.91}{0.036 * 0.9}} * \sqrt[4]{\frac{0.94}{50}}$$

$$\text{d}_\text{o} = 1.96 \approx 2\text{mm}$$

Where:

$\dot{V}$ = gas flow rate (m^3h^{-1}),

A_0 = area of orifice (mm^2)

p = gas pressure before orifice (mbar)

s = specific gravity of gas (0.94)

C_d = coefficient of discharge for the orifice (0.9).

The area of the injector becomes:

$$\text{A}_\text{o} = \pi\frac{\text{d}_\text{o}^2}{4}$$

$$\text{A}_\text{o} = 3.14\ \text{mm}^2$$

b. Determine the Velocity of the Gas in the Orifice

The velocity of the gas in the orifice is calculated using equation 3.8:

$$V_o = \frac{\dot{V}}{3.6 * 10^{-3} * A_o}$$

$$V_o = 80.5\ \frac{m}{s}$$

3.2.4 Design of Burner Throat

a. Determine the Size of Throat

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

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

The stoichiometric air ratio requirement is 5.7 from complete combustion analysis then the entrainment ratio (r) or the primary aeration should be 50% stoichiometric air. So

$$r = 50\% \text{ of stoichiometric air}$$

$$r = 0.5 * 5.7$$

$$r = 2.85$$

$$d_t = \left(\frac{2.85}{\sqrt{0.94}} + 1\right) * 2$$

$$d_t = 7.88\text{mm}$$

For better combustion, the primary aeration should be greater than the optimum. So the stoichiometric air ratio can be used as primary aeration. The air flow is controlled with airflow controller.

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

$$d_t = \left(\frac{5.7}{\sqrt{0.94}} + 1\right) * 2$$

$$d_t = 11.76\text{mm}$$

However, the exact size of throat depends on the standard pipe sizes available. A standard steel pipe sizes available is 1/2" which has inside diameter of 15.8mm. Therefore, the size of the throat d_t is taken equal to be 15.8mm.

b. Determine the Gas Pressure just after Nozzle

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

$$p_t = p_o - \rho \frac{V_o^2}{2g}\left[1 - \left(\frac{d_o}{d_t}\right)^4\right]$$

$$p_t = 10^5 - 1.0994 * \frac{80.5^2}{2 * 9.81}\left[1 - \left(\frac{2}{15.8}\right)^4\right]$$

$$p_t = 10^5 - 363 \text{ pa}$$

Where: - p_t is the gas pressure (N m^{-2}),

P_0 is atmospheric pressure, as the throat is open to the air,

ρ is the gas density (kg m^{-3}),

g is the acceleration due to gravity (9.81 m s^{-2})

Due to thus pressure difference the primary air can inter into the throat from the surrounding. The air inlet ports must have an area similar to that of the throat area.

c. Determine the Mixture Flow Rate at Optimum Aeration

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

$$\dot{V}_m = \frac{\dot{V} * (1 + r)}{3600}$$

$$\dot{V}_m = \frac{0.91 * (1 + 2.85)}{3600} = 0.97 * 10^{-3} \frac{m^3}{s}$$

d. Determine the Pressure Drop in the Mixing Tube

The length of the mixing tube is given as:-

$$L_m = 10 * d_t$$

$$L_m = 10 * 15.8\, mm$$

$$L_m = 158\, mm$$

Calculate the Reynolds number:

$$R_e = \frac{4\rho \dot{V}_m}{\pi\mu\, d_t}$$

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

$$R_e = \frac{4 * 1.15 * 0.97 * 10^{-3}}{\pi * 1.71 * 10^{-5} * 15.8 * 10^{-3}}$$

$$R_e = 5259.5$$

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

$$\Delta p = \frac{f}{2}\rho \frac{16\dot{V}_m^2}{\pi^2 d_t^5} L_m$$

$$f = \frac{0.316}{R_e{}^{1/4}}$$

$$f = 0.0371$$

Therefore, the pressure drop in the mixing tube becomes$\Delta p = 5.23$ pa.

The pressure drop in the mixing tube is much lower than the driving pressure in the throat (363 pa). And the length of the tube can be adjusted as: $L_m = 80$ mm.

3.2.5 Determine the Number of Flame Ports for a Burner

a. Determine the Flame Port Area

The mixture supply velocity (Vp) is given by:

$$V_p = \frac{\dot{V}_m}{A_p} \ll 0.25\frac{m}{s}$$

The total burner port area can now be chosen:

$$A_p > \frac{\dot{V}_m}{0.25} > \frac{0.97 * 10^{-3}}{0.25} = 3.88 * 10^{-3} \approx 0.004\ m^2$$

b. Determine the Number of Flame Port

The number of flame port is calculated as follow:-

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

Where: -

n_p is estimated number of flame ports.

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

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

$$n_p = \frac{4 * 0.004}{\pi * (0.004)^2}$$

$$n_p = 318.4 \approx 318$$

For flame stabilization, it should be possible to reduce the number of flame ports. So for a single burner the sufficient amount of flame port can be determined by satisfying the Prigg's formula. The Prigg's formula holds if the total flame port area (Ap) is between 1.5 and 2.2 times the area of the throat.

$$x_r = \frac{A_p}{A_t} = n_p \left(\frac{d_p}{d_t}\right)^2$$

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

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

$$x_r = n_p \left(\frac{d_p}{d_t}\right)^2$$

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

$$n_p = 31.2 \approx 32$$

3.2.6 Determine the Number of Burners for the System

The total number of flame ports of injera baking stove depends on the area of injera baking pan size. To distribute equal amount of heat to the pan surface most of the flame ports are arranged in three concentric circles. For cross-lighting from one circle to the other the rest flame ports are arranged in three straight lines in triangular pattern on the burner port plate as shown in Figure 3.18.

Figure 3.18: Flame ports arrangement

The three concentric circle diameters d_1, d_2 and d_3 are 100, 260 and 420 mm from the center respectively. And the diameter of each flame ports is 4 mm. The arc gap distance between each burner port is 6 mm which means the arc distance from the center of one flame port to the other is 10 mm.

The number of flame port in one circle is given as:

$$c = n_p * d_s \qquad 3.18$$

Where:

c is circumference of the circle ($c = \pi d$).

n_p is number of flame port in a circle.

d_s is the arc distance from the center of one flame port to the other.

For the inner circle (d_1=100 mm) the number of flame port is: -

$$c = n_p * d_s$$

$$\pi d_1 = n_p * d_s$$

$$n_p = \frac{\pi d_1}{d_s} = \frac{\pi * 100}{10} = 31.4 \approx 31$$

For the two circle d_2 = 260 mm and d_3 = 420 the flame ports are 82 and 132 respectively. So the total flame port on the three circles is 245.

The number of flame port in one straight line is given as:

$$n_p = \frac{d_r - d_s}{d_s} \tag{3.19}$$

Where:

n_p is number of flame port in a straight line.
d_r is the radius distance from one circle to the other.
d_s is the arc distance from the center of one flame port to the other.

To facilitate cross-ignition, holes on three radial lines connect the outer circle to medium circle and from medium circle to the inner circle, the total number of flame port with radius distance of 80 mm each is: -

$$n_p = 2 * \frac{d_r - d_s}{d_s} = 2 * \frac{80 - 10}{10} = 14$$

So the total flame ports on the three straight line is 3 * n_p = 42.

Therefore, the total flame port of the stove is 245 + 42 = 287.

From the above calculation, one burner is sufficient for 32 flame ports which is 4 mm in diameter.

$$1 \text{ burner} = 32 \text{ flame ports}$$

So for 287 flame ports, 9 burners required. Therefore, the total number of burners for the stove is nine.

CHAPTER FOUR

MANUFACTURING PROCESS AND TESTING OF INJERA BAKING STOVE

4.1 Manufacturing Process Stove

The injera baking stove has the following different parts.

- Fuel distributor tank
- Injector
- Mixing tube
- Union
- Manifold
- Baffle
- Flame ports plate
- Clay pan
- Pan cover
- Support structure
- Hose

Figure 4.1: Biogas injera baking stove

4.1.1 Fuel Distributor Tank

It distributes the biogas to all burner injectors which come from the supply hose. It has a cylindrical shape and has one inlet hole at the center of the bottom cover plate and nine outlet holes on the upper cover plate, the holes arrangement is shown in the Figure.

The dimensions of cylindrical shape of fuel distributor tank are given as:

- The cylinder diameter (d_c) is $d_c = 300$ mm.
- The cylinder height (h_c) is $h_c = 40$ mm.

The material used to manufacture the distributor tank is 2 mm thick steel sheet metal. To make the cylinder, the steel sheet with dimension of width equal to the circumference of the cylinder and length equal to the height of the cylinder was required

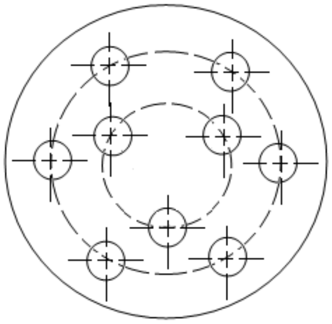

Figure 4.2: Holes arrangement on fuel distributor tank for injector assembly

The procedure to manufacture the cylinder consists of the following steps:

- Prepare the required plate using a sheet cutting machine.
- Using sheet rolling machine make a cylindrical shape.
- Then connect the end edge of the plate by welding.

To manufacture the cylinder upper and lower cover, a steel sheet with a dimension of width equal to two times the diameter of cylinder and length equal to the diameter of the cylinder was required.

The procedure to manufacture the circular cylinder cover consisted of the following steps:

- Prepare the required sheet using a sheet cutter machine.
- Draw two circles on the sheet with the required diameter using divider.

- Make circle sheets by cutting the sheet tangent to the drawn circles.
- Drill a hole at the center of lower circle cover sheet with a diameter (10 mm).
- Drill nine holes on the upper circle cover sheet with a diameter (7 mm).

Finally the upper and lower circle cover sheets are welded to the cylinder.

4.1.2 Injector

The injector controls the amount of gas used by a burner which is distributed from the fuel distributor tank. It has two parts for easy maintenance and manufacturing purpose. Those parts are nozzle (injector orifice) and injector tube (threaded hallow tube).

Nozzle: - A nozzle that is available in the market was used with minor modification of diameter. It has a thread on the outer part of the nozzle and the thread pitch length is 1 inch. The material of the nozzle was brass.

Injector tube: - The material of the injector tube was mild steel rod with a diameter of 25mm and a length of 65mm. It has internal thread for the purpose assembly to the nozzle.

The procedure of manufacturing the injector tube consisted of the following steps:

- Mount the steel rod on the lath machine then,
- Make facing operation for a good surface finishing and to reduce the length of rod up to the required dimension.
- Make turning operation for a good surface finishing and to reduce the diameter of the rod up to the required dimension.
- Drill the center of the rod with a drill bit of 7mm in diameter.
- Dismount the steel rod from the lath machine then, make internal thread hallow tube rod with a thread tap of 8mm in diameter and has a thread pitch length of 1 inch.

Finally the two parts are assembled with their internal and external threads.

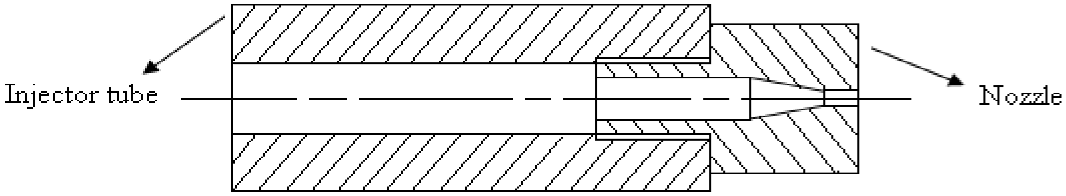

Figure 4.3: Nozzle and injector tube assembly

4.1.3 Mixing Tube and Union

Mixing tube: - it is a long cylindrical tube which helps for a good mixing of air and biogas. It has a tube length of 80 mm and two holes on the side of tube. The standard galvanized steel pipe available in the market was used, which is 1/2 inch galvanized steel pipe. The two holes are drilled with 8mm diameter of drill bit by using drill machine. Those holes are used for the primary air inlet.

Union: - union is used to connect the injector to the mixing tube. It helps for easy assemble and disassemble process. And the standard union available in the market was used, which is 1/2 inch galvanized steel pipe union.

4.1.4 Manifold and Baffle

It is used to distribute the biogas which supplied from the mixing tube, to the flame burner port. It has a shape of cone-frustum like shown in the Figure 4.4.

The dimensions of cone-frustum are given as:

- The cone frustum height is h_m = 100 mm.
- The upper diameter of the cone-frustum is $d_{m.up}$= 460 mm.
- The lower diameter of the cone-frustum is $d_{m.lo}$= 300 mm.

To make the manifold 2 mm steel metal with a length of 740 mm and a width of 510 mm was needed.

The procedure of manufacturing the manifold consisted of the following steps:

- Prepare the required sheet metal by using sheet cutter machine.
- Draw the development of the cone-frustum on the sheet metal using divider.
- Cut out the required development of the cone-frustum using sheet cutter machine.
- Make the cone frustum by rolling the sheet metal using rolling machine.
- Then connect the end edge of the sheet by welding.

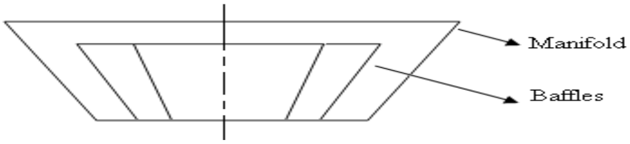

Figure 4.4: Manifold with two baffles inside

The bottom circle surface is covered with 2mm thickness steel sheet and has nine holes. It has the same dimension to the upper cover circle sheet of fuel distributer tank. So the manufacturing process was also the same.

Baffle: - it used to distribute the biogas in the manifold, which come through burner tubes to the all flame port proportionally. It has a shape of cone-frustum as shown in the above Figure 4.4. There are two baffles inside the manifold. The first baffle is small in size and used to distribute the biogas from three burner tubes to the flame ports. The second baffle is of larger size and used to distribute the biogas, which comes from the other six burner tubes, to the flame ports.

The dimensions of cone-frustum shape of baffles are given as:-

- For the smaller baffle: cone frustum height is h_b = 60 mm
  upper diameter of the cone-frustum is $d_{sb,up}$ = 180 mm
  lower diameter of the cone-frustum is $d_{sb,lo}$ = 50 mm
- For the larger baffle: cone frustum height is h_b = 60 mm
  upper diameter of the cone-frustum is $dl_{b,up}$ = 340 mm
  lower diameter of the cone-frustum is $d_{lb,lo}$ = 180 mm

To make the smaller baffle 1mm sheet metal with a length of 123 mm and a width of 83 mm was used and to manufacture the larger baffle 1mm sheet metal with a length of 213 mm and a width of 172 mm was used. The procedures of manufacturing the baffles are the same as that of the manifold.

Finally, the two baffles were welded concentrically to the bottom circle sheet of manifold. Then the edge of the bottom circular sheet of manifold was welded to the manifold cone frustum.

4.1.5 Flame Ports Sheet

It is a thick circular steel sheet at which the combustion of biogas took place. The diameter of the flame port sheet is 460 mm. it has 287 holes on the surface of the sheet and the holes diameter is 4 mm.

The holes are arranged in three concentric circles and straight lines as shown the Figure 3.18. The procedure of making a circular flame port sheet is the same as the procedure of making a circular cylinder cover. But the holes are drilled by milling machine with the help of rolling disk sheet. Finally the flame port sheet is welded to the upper part of cone-frustum.

4.1.6 Clay Pan and Pan Cover

Clay pan: - it is a working media for the injera baking s and it transfer the thermal energy from the the flame to the injera batter. The standard clay pan available in the market was used; the pan has a diameter of 58cm and a thickness of 25 mm.

Pan cover: - it is used to cover the baking injera during heat up time. It helps to reduce heat loss to the surrounding. It has a shape of cone.

The dimensions of a cone shape structure of pan cover are given as:

- The cone height is h_c = 50 mm.
- The cone diameter is d_c = 560 mm.

To manufacture the pan cover a 0.8mm aluminum sheet was used. The length and width of the sheet is 570 mm.

The procedure of manufacturing the pan cover consisted of the following steps:

- Prepare the required aluminum sheet by using sheet cutter machine.
- Draw the development of the cone on the aluminum sheet using divider.
- Cut out the required development of the cone using sheet cutter machine.
- Make the cone by rolling the aluminum sheet using rolling machine.
- Then connect the end edge of the sheet by welding.

4.1.7 Hose and Support Structure

Support structure: - It helps to support all part of the system. The system has two major parts which are burner support structure and pan support structure. The two support structure can be assemble or disassemble easily.

The pan support (pot support) structure holds only the clay pan and the pan cover. But the burner support structure holds the all parts of the system including the pan support structure.

Hose: - The hose use for supply of biogas to the fuel distributor tank. It is a flexible tube, so that the stove can be easily displaced.

4.2 Costs of Production

The cost of production as a function of material and labor cost is described in Table 4.1.

Table 4.1: Production costs for the injera baking stove

No	Description	Qty	Unit Price [Birr]	Material required	Total amount [Birr]
1	Material cost				
	a. Metal sheet (mm)				
	• Fuel distributor tank	1	550.00	650x950x2	137.50
	• Manifold	1	“	740x810x2	137.50
	• Baffle	2	220.00	173x96x1	55.00
	• Flame ports sheet	1	1000.00	460x460x5	225.00
	b. Aluminum sheet				
	• Pan cover	1	500.00	570x570x1	62.50
	c. RHS (20x20)				
	• Support structure	1	110.00	7200x20x2	55.00
	d. Burner				
	• Injector tube	9	3.00	Ø10	27
	• Nozzle	9	5.00		45.00
	e. Welding				
	• Electrode	1	144.00	½	72.00
	• Oxy-acetylene				50
	f. Paint				
	• Red spray	1	55.00	½	27.50
2	Labor costs				
	(three employees)	3	52birr/ 8hr	(6.5birr/hr*8hr)*3	156
	Total Production Costs				***1055.00***

4.3 Operating Procedure of Biogas Stove

a. Make sure all the primary air inlet valves are closed.

b. Open the biogas flow control valve.

c. Ignite the stove at the flame port.

d. Open the primary air inlet valves to insure the complete combustion.

e. Wait for some minutes until the clay pan surface temperature reach around 190-200^{o}c.

f. Start baking injera then close the clay pan cover.

g. After 2-3 min open the clay pan cover and take out the baked injera from the pan.

h. To bake the remaining injera wait until the pan surface temperature reaches 190-200 ^{o}C.

4.4 Performance Test of Injera Baking Stove

The test was done with injera baking clay pan of 15 mm in thickness and 56 cm in diameter. Due to unavailability of biogas the test is done using LPG as source of energy. To see whether the heat distribution is uniform or not, the temperature measurements were taken at different radius on the upper surface of clay pan. When injera baking starts, the baking pan surface temperature ranges from 190 – 200 ^{o}C.

Table 4.2: Clay pan top surface temperature distribution at different radius

Number of Test	Time (min)	Temperature in (^{o}C) at a Distance r in (mm)						
		r = 0	r = 50	r = 90	r = 130	r = 170	r = 210	r = 250
1	0	18	18	18	18	18	18	18
2	1.5	50	52	58	55	65	70	67
3	3	79	80	82	90	85	90	90
4	4.5	105	101	114	126	120	110	100
5	6	130	135	146	152	148	138	120
6	7.5	145	147	150	154	152	151	140
7	9	160	165	165	170	178	176	160
8	10.5	180	185	190	200	195	200	185
		The surface temperature of pan just before the first injera baking starts.						
9	12	200	195	210	222	219	220	219
		The surface temperature of pan just after the first injera is taken out.						
10	14.5	159	161	180	174	190	180	168
		The surface temperature of pan just before the second injera is baking.						
11	16	193	191	211	218	232	231	200
		The surface temperature of pan just after the second injera is taken out.						
12	18.5	150	157	170	180	185	185	170
		The surface temperature of pan just before the third injera is baking.						
13	20	198	191	220	238	230	222	212
		The surface temperature of pan just after the third injera is taken out.						
14	22.5	155	163	172	186	179	176	170

4.4.1 Test Result and Analysis

For three injera baking cycle, the heat up time, baking time and ideal time are shown in Figure 4.6. The ambient temperature during the beginning of test was 18 ^{o}C, so at time t = 0 the pan surface temperature at every radius of the pan was equal to 18 ^{o}C.

For baking the first injera, the baking pan takes a heat up time of around 12 min, at which the pan surface temperature reaches around 190 – 200 ^{o}C. The average baking and ideal time per injera is 2.5 min and 1.5 min respectively.

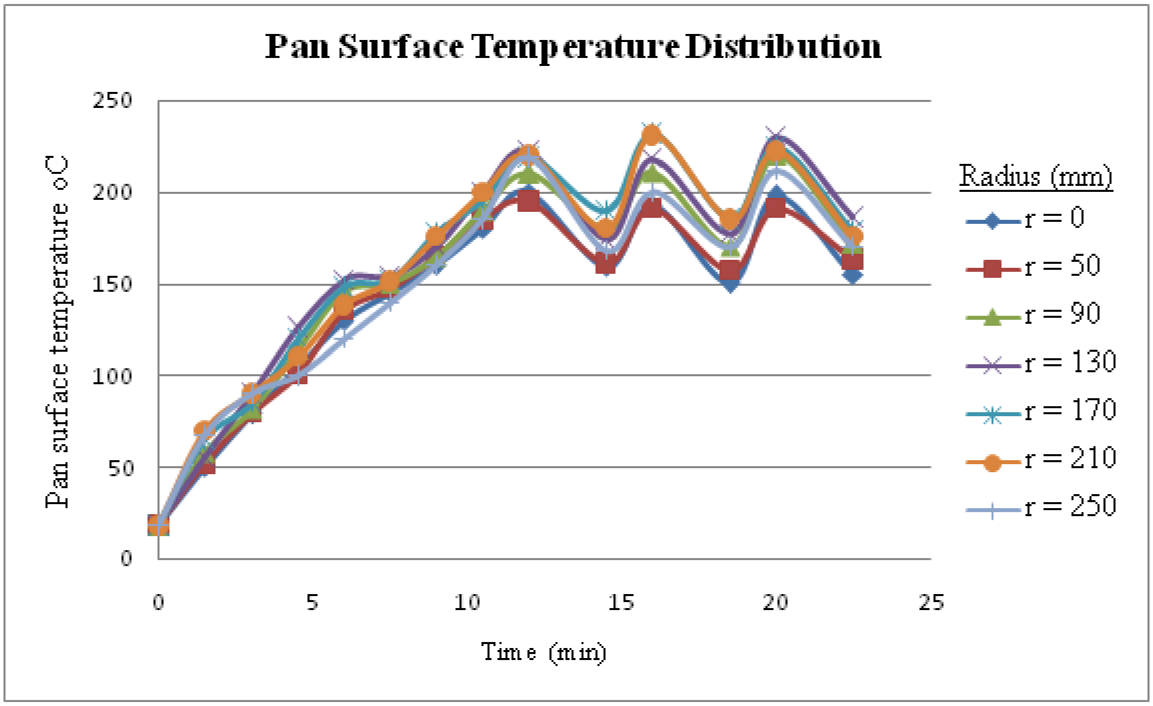

Figure 4.6: Clay pan top surface temperature distribution

During the heat up time of baking pan the expected graph was a smooth curve. But the actual test data shows rough curve. This was caused due to opening of primary air inlet to insure complete combustion, wind effect during baking and adjusting the biogas flow rate to supply sufficient biogas. After the primary air inlet valve is opened, complete combustion takes place and the flame becomes stable.

CHAPTER FIVE

CONCLUSION AND RECOMMENDATION

Biogas is just a clever way of exploiting nature without destroying it. It is well suited to household cooking. One of the most important advantages of biogas is its feasibility in rural areas where it offers the potential for sustainable development. The scale of digesters can vary to suit the energy needs of a household or a small community, and the only input required organic waste is readily available in rural areas.

Biogas is one of renewable energy and it used for many applications. So this thesis demonstrated that biogas can be used for injera baking by developing a biogas injera baking stove. However, the gas pressure must be sufficient to overcome the pressure drop in the system.

The biogas injera baking stove is developed from a simple biogas domestic cooking stove. The functionality of the design stove is checked by baking three injera using LPG as energy source.

The following conclusions are made from the development (design) of stove.

- The developed (designed) stove has nine burners to uniformly distribute sufficient heat to the baking pan.
- The positions of nine burners should be in a vertical position to prevent gas leakage through the primary air inlet valve when there is minimum biogas flow rate. If the burners are in a horizontal position the gas can leak to the environment with the primary air inlet valve.
- Conical baffles should be used in the manifold to distribute the biogas to the flame ports uniformly.
- If the diameters of the flame ports are large, back firing can occur. For 5 mm thick metal plate, 5 mm diameters of flame ports are recommended. But to make sure that back firing does not occur, 4 mm diameter hole on 5 mm thick metal plate are used for the injera baking stove.

The following conclusions are made from the performance test of the stove.

- The air-fuel mixture flow rate should be controlled to prevent flame lifting on the flame port due to high flow rate.

- The distance between the flame port sheet and the baking pan should be optimized to reduce heat loss to the surrounding and to insure complete combustion by supplying secondary air to the flame port.
- During baking time the baking pan surface temperature should be approximately equal at every radius of the baking pan. This insures the heat distribution is uniform and the flame is stable.

The recommendation for further work to enhance efficiency concentrates on the following:

- Reduce the existing sheet thickness and conduct test experiment for better baking cycle time and initial heat up time.
- Investigate better way of primary air inlet mechanism for better combustion.
- Optimize the number of burner and the diameter, number and arrangement of flame ports.
- Investigate the stove efficiency using biogas as a source of energy.

APPENDIX

A. Conversion Factors

Acceleration	$1\ m/s^2 = 4.2520 \times 10^7 ft/h^2$
Area	$1\ m^2 = 10.764\ ft^2$
Density	$1\ kg/m^3 = 0.06243\ Ibm/ft^3$
Energy	1J = 0.239 cal $= 9.4787 \times 10^{-4}$Btu
Force	1 N = 0.22481 Ibf
Heat capacity flow rate	$1\ kW/K = 1\ kW/^0C$ $= 1895.6\ Btu/h.^0F$
Heat flux	$1W/m^2 = 0.3171\ Btu/h.ft^2$
Heat generation rate	$1W/m^3 = 0.09665\ Btu/h.ft^3$
Heat transfer coefficient	$1W/m^2. K = 0.17612\ Btu/h.f\ t^2.\ ^0F$
Heat transfer rate	1W = 3.4123 Btu/h
Kinematic viscosity and thermal diffusivity	$1m^2/s = 3.875 \times 10^4 ft^2/h$
Latent heat and specific enthalpy	1kJ/kg = 0.42995 Btu/Ibm
Length	1m = 3.2808 ft
Mass	1kg = 2.2046 Ibm
Mass flow rate	1kg/s = 7936.6 Ibm/h
Mass flux	$1kg/s.m^2 = 737.35\ Ibm/h.ft^2$
Power	1kW = 3412 Btu/h = 1.341 hp
Pressure (stress)	$1Pa\ (1N/m^2) = 0.020886\ Ibf/ft^2$ $= 1.4504 \times 10^{-4}$ psi $= 4.015 \times 10^{-3}\ in.H_20$
Pressure	1.01325×10^5 Pa = 1atm = 14.696 psi = 760 torr $= 406.8\ in.H_20$

Specific heat	1kJ/kg. K = 0.2389 Btu/Ibm. ^{0}F
Surface tension	1N/m = 1000 dyne/cm = 0.068523 Ibf/ft
Temperature	K = ^{0}C + 273.15 = (5/9)(^{0}F + 459.67) = (5/9) (^{0}R)
Temperature difference	1 K = 1 ^{0}C = 1.8 ^{0}F = 1.8^{0}R
Thermal conductivity	1W/m. K = 0.57782 Btu/h. ft. ^{0}F
Thermal resistance	1 K/W = 0.52750^{0}F.h /Btu
Viscosity	1 kg/m. s = 1000 cp = 2419 Ibm/ft.h
Volume	1 m^3 = 35.314 ft^3 = 264.17 gal
Volumetric flow rate	1 m^3/s = 2118.9 ft^3/min(cfm) = 1.5850 x 104 gal/min(gpm)

Ibf: pound force and Ibm: pound mass.

B. Ghhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhhjhghghh

(a). Ignition process.

(b). Adjusting primary air inlet valve.

(c). Measuring mid pan surface temperature.

(d). Measuring temperature at different radius.

(e). Just after baking injera.

(f). Just before the baked injera is taken out.

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C. Clay pan top surface temperature distribution at different radius

Number of Test	Time (min)	Temperature in (^oC) at a Distance r in (mm)						
		r = 0	r = 50	r = 90	r = 130	r = 170	r = 210	r = 250
1	0	18	18	18	18	18	18	18
2	1.5	50	52	58	55	65	70	67
3	3	79	80	82	90	85	90	90
4	4.5	105	101	114	126	120	110	100
5	6	130	135	146	152	148	138	120
6	7.5	145	147	150	154	152	151	140
7	9	160	165	165	170	178	176	160
8	10.5	180	185	190	200	195	200	185
		The surface temperature of pan just before the first injera is baking.						
9	12	200	195	210	222	219	220	219
		The surface temperature of pan just after the first injera is out.						
10	14.5	159	161	180	174	190	180	168
		The surface temperature of pan just before the second injera is baking.						
11	16	193	191	211	218	232	231	200
		The surface temperature of pan just after the second injera is out.						
12	18.5	150	157	170	180	185	185	170
		The surface temperature of pan just before the third injera is baking.						
13	20	198	191	220	238	230	222	212
		The surface temperature of pan just after the third injera is out.						
14	22.5	155	163	172	186	179	176	170

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