American Journal of Engineering and Applied Sciences

Original Research Paper

Biogas Clay Brick Kiln Burner

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Article history
Received: 25-04-2018
Revised: 12-05-2018
Accepted: 22-05-2018

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Abstract: Biogas burners were designed in different geometrical for different purpose like for cooking, beking injera and the likes. This biogas burner for clay brick firing is designed for firing clay brick in the clay brick kiln. Biogas clay brick burner was specifically designed to replace firewood, caw dung and other non-renewable energy source used in local and mechanized clay brick manufactures. Because of the arrangement of clay brick in the clay brick kiln, bar types of biogas burner was selected and all components of biogas burner in the biogas burner design is considered.

Keywords: Clay Brick, Kiln, Burner, Biogas

Introduction

Brick firing is primary process used to make the clay brick stronger, fire resistance and less absorbent. Biomass (fuel wood, coal, sawdust and dung cake) is the main energy source in traditional brick firing, while modern brick manufacture uses oil fuel, natural gas and propane gas as energy sources Alam and Degree (2006).

According to the study by Alam and Degree (2006) on deforestation and greenhouse gas emissions associated with the fuel wood consumption of the brick making industry in Sudan, CO_2 emitted during the process of brick firing using biomass (fuel wood, crop residue, dung cake) to be carbon neutral the amount of trees cut for the fuel in brick firing were planted/replaced. Even though we say all biomass is collected in sustainable ways, there are carbon monoxide, CH_4 and non-methane hydrocarbon emission as a result of incomplete combustion.

Most brick firing kilns are low efficient, so large amounts of biomass or fossil fuel is necessary to fire brick in a standardized manner. This leads the people to use more trees or large amounts of fossil fuel, which results in deforestation and non-renewable energy sources being exhausted. Deforestation is a big problem in natural ecosystem balance and even leads to world climate change. World climatic change comes from natural concentration deviation of (GHG) which brings about Global Warming (GW).

Environment is the host of all living and non-living things; it is the common home of all nations and nationalities and it is the common property of the peoples of the world. Unless we keep our environment clean and neat, it is impossible to talk about sustaining what we have in hand to day or planning for future in a sustainable manner; it is impossible to talk about sustainable development.

Emission resulted from different factories and even those released from local activities were very serious problematic for our environment. Non-renewable energy sources were naturally limited; they were depleted if we were used it continuously. Treating those non-renewable source of energy before, during combustion may solve problem of emission but the problem of sustainability were another issue. Unless substitution of those non-renewable energy with renewable energy sources to satisfy the energy need of those activities, our environment will continue to suffer because of emission released.

To solve problems of emission during the brick firing process, people started using propane gas or natural gas as the main energy source. This energy source is clean and it contributes a lot in emission reduction. But brick firing using propane gas has the problem of sustainability and cost affordability. From this point of view, this paper design biogas burner for brick firing considering biogas as the main energy sources.

Burner Design

To gain optimum energy and utilize the biogas produced in efficient way for brick firing process bar type of Bolivial was selected Kurchania *et al*. (2010). For complete combustion there are things that are considered to be design in resealable ways. Injector orifice, primary aeration, flame port size, mixture tube, flame velocity, manifold size and throat size are components of burner that were considered Jones (2005) in the burner design.

Combustion of Biogas

Since simulation of transient temperature distribution of bricks require knowledge of temperature of hot gas conditions Prasertsan *et al*. (1997), Vitázek *et al*. (2016), seeing the combustion reaction of biogas with air is very important. The combustion equation is:

$$0.6CH_4 + 0.4CO_2 + 1.2O_2 + 4.5N_2 \rightarrow CO_2 + 1.2H_2O + 4.5N_2$$

The general hydrocarbon combustion equation of 60% gas and 40% of CO_2 is Jones (2005):

$$\begin{aligned} &0.6C_xH_y + 0.4CO_2 + (x/5 + y/20)O_2 \\ &+3.76(x/5 + y/20)N_2 \rightarrow \\ &(0.6x + 0.4)CO_2 \ + (x/5 + y/20)H_2O \\ &+3.76(x/5 + y/20)N_2 \end{aligned}$$

The theoretical amount of air required is defined as the number of air required for stoichiometric combustion per unit volume of the fuel. The theoretical values of air required can be obtained from the general hydrocarbon combustion which is given as Jones (2005):

$$TAR = 14.28(x/5 + y/20) \tag{1}$$

When we come to the combustion of methane gas, $x = 1m^3$; and from the general hydrocarbon combustion equation $y = 4m^3$.
Therefore the theoretical air required would be:

$$TAR = 14.28(x/5 + y/20) = 14.28 * 0.4 = 5.7$$

The quantity of oxidizer that required to completely burn a quantity of fuel is stoichometric ratio. It is found by:

$$\left(\frac{A}{F}\right)_{stioc} = \frac{m_{air}}{m_{fuel}} = \left(\frac{4.76}{0.6}\right)(a)\left(\frac{M_{air}}{M_{fuel}}\right) \tag{2}$$

Where:
M_{air} = Molecular mass of air
M_{fuel} = Molecular mass of fuel

And 'a' stand for the coefficient of oxygen and nitrogen in the general chemical balance of methane combustion in the above chemical formula. It is obtained by:

$$a = \frac{x}{5} + \frac{y}{20}$$

Since from general chemical formula of methane gas a would be:

$$a = \frac{1}{5} + \frac{4}{20} = 0.4$$

The molecular mass of fuel and air becomes:

$$\begin{aligned} &Mfuel = 0.6(12 + 4 * 1.008) = 9.6192 \\ &Mair = 28.85 \end{aligned}$$

The stoichometric ratio is:

$$\left(\frac{A}{F}\right)_{stioc} = \frac{4.76}{0.6}(0.4)\frac{28.85}{9.6192} = 9.5$$

Equivalence ratio (ϕ) is expressed as:

$$\phi = \frac{\left(\frac{A}{F}\right)_{stioc}}{\left(\frac{A}{F}\right)} = \frac{\left(\frac{F}{A}\right)}{\left(\frac{A}{F}\right)_{stioc}} \tag{3}$$

Air to fuel ratio of combustion of biogas in the air is:

$$\left(\frac{A}{F}\right) = \left(\frac{N_{air}}{N_{fuel}}\right)\left(\frac{M_{air}}{M_{fuel}}\right) = \left(\frac{4.76}{0.6}\right)(0.4)\left(\frac{28.85}{9.6192}\right) = 9.5$$

Equivalence ratio of combustion of biogas in the air:

$$\phi = \frac{\left(\frac{A}{F}\right)_{stioc}}{\left(\frac{A}{F}\right)} = \frac{9.5}{9.5} = 1$$

As it is seen from ϕ analysis, the mixture of biogas and air is stoichometric.

To find the combustion temperature, we have assumed the flame is adiabatic in controlled pressure. From first laws of thermodynamics and replacement of internal energy with enthalpy, we can write as Turns (2000):

$$H_{reac}(T_i, p) - H_{proc}(T_{ad}, p) = 0 \tag{4}$$

Where:
T_i = Temperature at 298K
T_{ad} = Adiabatic flame temperature

Table 1: Enthalpy and specific heat of some species (source: Turns (2000))

Species	Enthalpy of Formation at 298K $h^{o}_{f,i}$ (KJ/kmol)	Specific Heat at 1200K $C_{p,i}$(KJ/kmol-K)
CH_4	-74,831	
CO_2	-393,546	56.21
H_2O	-241,845	43.87
N_2	0	33.71
$O2$	0	

Adiabatic temperature of biogas with 60% CH_4 and 40% of CO_2 with combustion in the air was calculated Díaz-González *et al*. (2009) and it is equal to 2145K.

In this study it is calculated as follow.

By taking the initial condition as T_{ini} = 298K and pressure P_{ini} = 1atm = 101, 325 pa and from Table 1.

Substituting the values of enthalphy and specific heat of reactant and product in the simplified equation from Table 1:

$$H_{reac} = (0.6)(-74{,}831) + 1.2(0) + (4.5)(0) + (0.4)(-393{,}546) = -202{,}317KJ$$

$$\begin{aligned} H_{proc} &= (1)\left[-393{,}546 + (56.21)(T_{ad} - 298)\right] \\ &+ (1.2)\left[-241{,}845 + (43.87)(T_{ad} - 298)\right] \\ &+ (4.5)\left[0 + (33.71)(T_{ad} - 298)\right] \\ &= -761{,}403.602KJ + 260.549T_{ad} \end{aligned}$$

By equating these heat of reaction and heat of product we can find the adiabatic flame temperature as:

$$T_{ad} = 2145.8K$$

Adiabatic flame temperature of combustion of biogas was varies with distance over the burner. The flame temperature at 35 mm distance above a plate (burner) can be estimated to 1800K Kishore *et al*. (2007).

Injector Orifice

In order to supply fuel and air to the burner, appropriate design of injector is major activity that has to be done Jones (2005). The main purpose of injector is converting the potential energy of fuel to kinetic energy. Therefore, the speed and flow rate of gas would be:

$$\frac{1}{2}u^2 = gh \tag{5}$$

$$\dot{V} = A_j\sqrt{2gh} \tag{6}$$

Where:
u = Gas velocity
$\dot{V}$ = volume flow rate
A_j = Injector area
h = Column height

The gage pressure is obviously given by:

$$P = \rho_g gh$$

where, ρ_g-gas density.

So that we can more modify the volume flow rate of the gas of Equation 6:

$$\dot{V} = A_j\sqrt{\frac{2P}{\rho_g}} \tag{7}$$

Since density of gas is the product of density of air (ρ_a) and relative density of gas δ, we can simplify the volume flow rate of Equation 7 as:

$$\dot{V} = A_j\sqrt{\frac{2P}{\rho_g\delta}} \tag{8}$$

Density of air ρ_a = 1.225 kg/m^3 and with A_j, in mm and P, in mbar we can write the volume flow rate m^3/s as:

$$\dot{V} = 1.278*10^{-5}A_j\sqrt{\frac{2P}{\delta}}$$

Since the specific gravity of gas is:

$$\delta = \frac{\rho_g}{\rho_a} = \frac{1.2}{1.225} = 0.98$$

$$\dot{V} = 1.29*10^{-5}A_j\sqrt{P} \tag{9}$$

Because of friction effect, the flow of gas from orifice is less than expected result Jones (2005). The ratio of gas flow charge after orifice entrance to that of gas flow charge before orifice entrance is constant and it is called coefficient of discharge. We can express it as:

$$\frac{\dot{V}_{af}}{\dot{V}} = C_d$$

$$\dot{V}_{af} = 12.78C_dA_j\sqrt{\frac{P}{\delta}}$$

If $\dot{V}_{af} = Q$, then:

$$Q = 0.04601C_dA_j\sqrt{\frac{P}{s}} \tag{10}$$

Table 2: Variation of coefficient of discharge with and ratio of length to that of the diameter of orifice (source: Jones (2005))

Angle (degree)	Angle in radian	Orifice length/diameter	C_d
45	0.7850000	0.58	0.81
55	0.9594440	3.50	0.84
33	0.5756670	0.80	0.93
40	0.6977780	1.00	0.94

The optimum value of coefficient of discharge is one or number which approach to one. The experimental data obtained in the Jones (2005) is presented in the Table 2 as follow.

In this study we have four biogas combustion box, the energy transfer rate to the 3094 bricks is 646.6 *MJ/h* Munson and Young (2002). Since the burner type selected is bar, unless we provide the fuel in both side of the bar we could not find similar gas distribution in the burner port. The flow rate of the gas from each biogas plant (Volume one) is 29.38 m^3/h. The flow rate of gas that could be given to each burner is 29.34/4 = 7.345 m^3/h as in the Fig. 1 and 7.345 m^3/h of gas flow rate again divided in to two combustion box by pipe, as a result the flow rate of gas in a pipe would be 7.345/2 = 3.67m^3/h. But again, in each combustion box we have three equally spaced bartype burner as Fig. 2, that is the flow rate of the three gas pipe that provide gas to the burner is 3.67/3 = 1.22 m^3/h.

By taking the efficient of bar burner as 43% and C_d = 0.95 the area of orifice would be:

$$A_j = \frac{1.22}{0.0467 * 0.95\sqrt{\frac{10}{0.98}}} = 8.6mm^2$$

And from this value we can estimate the diameter of orifice as:

$$d_j = \sqrt{\frac{4A_j}{\pi}} = \sqrt{\frac{4 * 8.3}{\pi}} = 3.3mm$$

Throat Size

The diameter of mixing pipe was greater than the diameter of injector. Diameter of throat should be six times that of diameter of the injector Itodo *et al*. (2007):

$$d_t = 6d_j = 6 * 3.3mm = 20mm$$

Entrainment

The place where the emerged gas from injector enters to the mixing tube is throat Jones (2005). Since the area of throat is larger than the area of injector, the velocity of gas in the throat is reduced. To relate them, we have used:

$$Q_t = Q_j$$

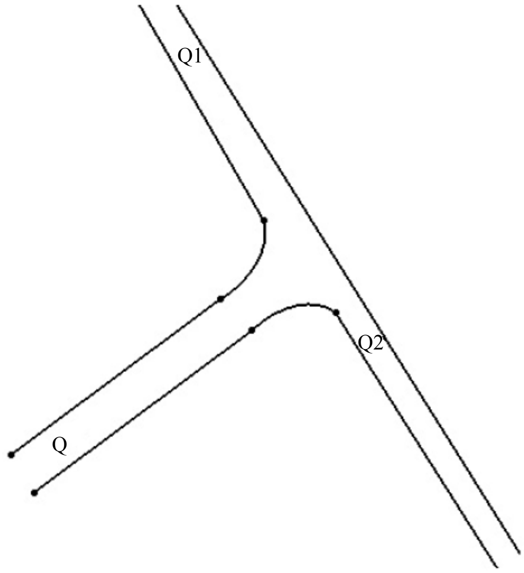

Fig. 1: Gas transportation to the brick kiln gas pipe

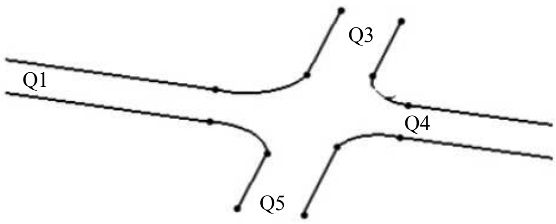

Fig. 2: Gas transportation pipe to the burner

Then the flow rate of each area is expressed as:

$$Q_t = V_t A_t$$
$$Q_j = V_j A_j$$

From these two equations we can write the velocity of the gas in the throat as:

$$V_t = V_j \frac{A_j}{A_t} \tag{11}$$

As it is seen from Equation 11, velocity of gas in the throat is reduced by the factor of $\frac{A_j}{A_t}$. The velocity of gas at the injector is:

$$V_j = \frac{1.22}{8.6*3600*10^{-6}} = 39.4 m/s$$

Using Equation 11, while the velocity of the gas in the throat is reduced to:

$$V_t = V_j \frac{A_j}{A_t} = V_j \frac{d_j^2}{d_j^2}$$

$$V_t = 39.4 \frac{3.3^2}{20^2} = 1.07 m/s$$

The gas pressure just after the throat becomes:

$$P_t = P_o - \rho \frac{V_j^2}{2g}\left[1 - \left(\frac{d_j}{d_t}\right)^4\right]$$

$$P_t = 10^5 - 1.0645 \frac{39.4^2}{2x9.81}\left[1 - \left(\frac{3.3}{20}\right)^4\right] = 10^5 Pa - 84Pa$$

To draw primary air through air inlet and to mix with gas in the mixing tube a pressure drop of 84 Pa is enough Jones (2005).

To carry out complete combustion of the fuel, the primary aeration required should be in between 40-60%. In our case, the primary air aeration is taken to be 50%. Therefore, entrainment ratio (r):

$$r = \frac{5.7}{2} = 2.85$$

Mixing Tube

Unless air and gas mixed appropriately, the consumption of gas is very large and burner efficiency was reduced. In this biogas brick firing burner systems, we have selected bar type burner without diffuser. For such burner, the mixing tube must be long enough and approximated as Kurchania *et al*. (2010):

$$L = 10 d_t \tag{12}$$

The mixing tube size is:

$$L = 10 * 20mm = 20cm$$

Burner Port Number and Size

The physical assembly and the relation between the port area of injector and that of the area of the port area has taken attention.

Small flame port is recommended in order to minimize the probability to light back. The diameter of port is quoted in between 2.5-5 mm and the distance between the port should be considered so as to prevent delay of ignition Kurchania *et al*. (2010).

The biogas designed was planned to fire brick of 3,094 per two day with tolerance of ±100 bricks in the arrangement. The length, width and height of 3,094 brick arrangement estimated from single brick size and the space between bricks in the arrangement. It is arranged with 2.0175*2.115*1.92 m.

The type of burner selected is bar and its length should be the same to the width of brick arrangement, that is 2.115 m. The inner diameter of bar tube as it is stated in the Kurchania *et al*. (2010) could be 19 mm, by selecting 5mm port diameter and 5 mm gap between each port the number of port in one row is:

$$5*10^{-3} m(2n) = 2.115m$$

$$n = 211$$

As it is seen in the Fig. 3 the arrangement of ports are, three 211 of ports are arranged on the line with an angle of 1200 difference to each other. In general we have 633 ports on the 2.115 m of bar that has 19 mm diameter.

The number of bar burner in biogas burner design was three. Those three bars pipe are out from manifold with angle difference of 1200 with one another. Therefore the total number of port is 1899. In general the parameter of biogas fired brick kiln bar burner is seen in the Table 3.

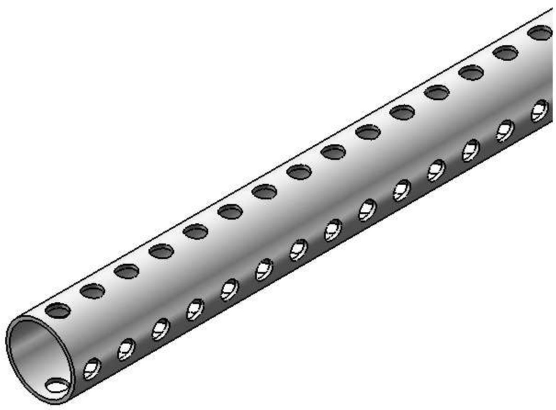

Fig. 3: Arrangement of ports on the bar burner

Table 3: Specification of different parameter of designed biogas burner

No.	Parameter	Specification
1	Inner diameter of tube	19 mm
2	Number of perforated tube	3
3	Number of port	1899
4	Length of mixing tube	200 mm
5	Diameter of injector	3.25 mm
6	Diameter of throat	20 mm
7	Diameter of port	5 mm
8	Diameter of primary air inlet	18 mm
9	Gap (distance) between brick and burner	35 mm
10	Length of tube	2.115 m

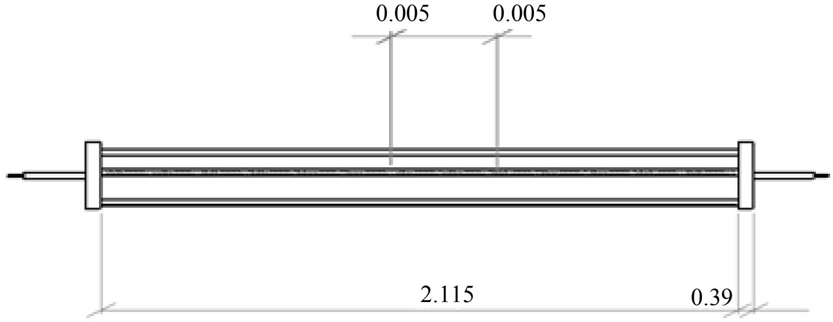

Fig. 4: Biogas brick kiln burner design

Sizing Manifold

The flow of air/mixture to the port should be uniform; in the same way the pressure in each and every port Fulford (1988) should be the same. This was done by designing reasonable manifold size, the temporary gas storage, for the biogas burner shape and port size.

Bar burner selected was tube which has to be taken as a cylinder of 19 mm in diameter. The height of cylindrical bar tube which served as a burner is seen in the section II-6 and it has a length of 2.115 m. we have taken cylindrical shape manifold of diameter 22 cm and estimated height of 35 cm. Its detail design drawing for full burner is seen in the Fig. 4.

Flame Lift

As stated by Yustia (2015), Obada *et al*. (2016) flame lift is one of the problems seen in the biogas burner designed. It was happened when the speed of gas to air mixture is higher than that of the speed of flame. Even though flame speed was depend on different quantities, the optimum flame speed is 0.25 m/s.

To ensure absence of flame lift for biogas burner designed, we have to check the speed of mixture gas with air at the port could be less or equal to flame speed. It could be expressed mathematically from flow rate of mixture gas and area of port as:

$$V_p = \frac{Q_m}{A_p} \leq 0.25 m/s$$

Entrainment ratio given as:

$$r = \sqrt{0.98}\sqrt{\left(\frac{A_j}{A_t}\right)} - 1 = \sqrt{0.98}\left(\sqrt{\left(\frac{20^2}{3.25^2}\right)} - 1\right) = 5.1$$

Therefore we can find the flaw rate of gas/air mixture as:

$$Q_m = \frac{Q(1+r)}{3600} = \frac{1.22(1+5.1)}{3600} = 0.002 m^3/s$$

The maximum mixture speed can be found as:

$$V_m = \frac{Q_m}{A_p}$$

$$= \frac{0.002}{\frac{1899\pi * 25x10^{-6}}{4}} = 0.05 m/s$$

Thus, we did not afraid for flame lift in the biogas burner designed.

Conclusion

The research shows how much biogas can be used for big or mechanized sector if we have enough biomass. It is seen that we can use biogas to different activities equivalent to the energy satisfied using different biomass. In the energy intensive activities like clay brick firing kiln, it is possible to use biogas as energy source to safe the environment from different emission and peoples from air born diseases. The design of biogas burner for the purpose of clay brick kiln firing is different from the local biogas burner for cooking, since in biogas clay brick firing the burner selected is bar as per the length of the brick arrangement and the biogas hole arrangement is designed by considering the flow of methane gas to all direction.

Acknowledgement

I would like to express my grateful, gratitude and sincere appreciation to my advisor, Prof. Venkata Ramayya and my co-advisor Dr. Ing. Getachew Shunki for their guidance, valuable advice and consistent support at all the stage of this article work.

Funding Information

This manuscript was funded by Jimma University Institute of Technology of Masters Program Coordinator. They funded all of the money needed to carry out all activities of this paper.

Author's Contributions

Alemayehu Beyene: Contribute a lot such as doing the research, simulating, data analysis and create this structure of the paper and making ready the paper for publication. The second and the third authors were also contributed in different ways.

Vinkata Ramaya: Did a lot in correcting the ideas, editing the concept and scientific knowledge. He contribute in organizing the research and contents that included in the paper.

Getachew Shunki: Contribute in design part and the way the burner holes arranged.

Ethics

This material is original and contains new unpublished design. We assure you that not ethical issues and no conflict of interest that may comes after the publication of this paper.

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