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DESIGN OF BIOGAS STOVE FOR INJERA BAKING APPLICTION

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Abstract: **Matured biogas production technology has led to the development of a number of biogas appliances used for lighting, electricity and cooking. The most hopeful among them is the biogas stove, to achieve the energy necessity for cooking application at domestic as well as at the community level. In this research work an attempt has been made to design biogas stove used for Injera baking. This stove was designed by covering the flame under an insulated material. The new gas burner also designed based on: the demand of energy needed during baking and the amount of gas supplied from the biogas plant. This new designed gas burner is used to supply pressure equally on the holes of the burner port. Heat transfer analysis between burner and mitad by radiation, convection heat transfer between flame and wall and heat flow through insulation also determined.**

Keywords: **Injector Orifice, Aeration, Throttle tube, Burner port, Flame height.**

1. INTRODUCTION

Ethiopia is a country of approximately more than 80 million in habitants that is located in East Africa and encompasses 1.1 million square kilometers. The population is multi nation and nationalities that have different cultures, religious, languages and ethnic group. Ethiopia is the third largest user in the world of traditional fuels for household energy use, with 96% of the population dependent on traditional biomass (e.g., fuel wood and dung) to meet their energy needs; this is in comparison to 90% for Sub-Saharan Africa and approximately 60% for the African continent (Jargstorf, 2004). As reported by the Ethiopian Rural Energy Development and Promotion Center (1998), in 1996, the most recent year for which statistics on the energy sector are available, 77% of total final energy consumption consisted of firewood and charcoal while another 15.5% consisted of agricultural residues; only roughly 6% was met by modern energy sources such as petroleum and electricity, and only 1% of the population utilized electricity for cooking. From the total energy demand stated, approximately 89% was consumed by households, while an ordinary 4.6% was due to industry. While Ethiopian demand for modern energy sources is expected to grow faster than for any other energy source, biomass fuels will continue to govern total energy consumption [3]. In rural regions, nearly 85% of the population rely on fuel wood as their crucial fuel for cooking, with the next largest primary reliance ratio being 12.65 percent for crop residue; only 0.21 percent of the rural population depends on kerosene for their primary cooking fuel, while the percentage for electricity and LPG are 0.05 and 0.07%, respectively. Contrast this to the capital city of Addis Ababa, where 42% of residents depend on kerosene as their primary fuel, compared to just 6.5%, each, for LPG and electricity; approximately one quarter of the population in Addis Ababa depends on fuel wood for their primary fuel, with 8% depending on crop residue and 4.5% depending on charcoal (Ethiopian Central Statistical Authority, 2004) [3].

Clearly, as rural Ethiopian households migrate to urban centers, which they are doing at a rate of over 4% per year, the energy balance of the country will shift (growth of the capital is rumored to be greater than 6%); overall, fuel wood use will decline as households, in the non-presence of an appropriate alternative, switch to kerosene. This is beneficial in that it mitigates the pressure on fuel wood, but detrimental in that dependency on petroleum imports rises; as dependency on petroleum imports growths, so, too, will expenditures of valuable foreign exchange [3].

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The people in Ethiopia rely on Injera as their primary source of food. Injera is a flatbread made from Teff that is baked upon a griddle, which is most often heated by means of an open wood fire. Baking Injera accounts for over 50% of all primary energy consumption and over 75% of all household energy consumption. Due to the shortage of firewood in growing Ethiopian communities, baking Injera on open fire is becoming increasingly expensive. Women and young children have to walk many miles a day to collect firewood to feed their families.

Figure 1: Diagrams (Photo) Of Women and Young Children Have to Walk Many Miles to Collect Firewood [Fuel Efficiency Injera Baking In Ethiopia]

By buying fuel-efficient stoves, commercial clients and households can reduce their firewood consumption by approximately 60% and make a considerable saving on their financial budget. At the current market prices, the fuel-efficient stoves have a payback time between 2 and 3 months for an average household family [2].

2. PROBLEM FORMULATION

Many of under developed and developing countries get energy source for cooking and baking from wood. Biogas is one of the alternative energy sources produced from different waste materials, cow dung, excreta etc. Matured biogas production technology has led to the development of a number of biogas appliances for lighting, power generation, and cooking. The most promising among them is the biogas stove, to meet the energy requirement for cooking application at domestic as well as at the community level. Now days in Ethiopia, 75% energy is extracted from wood. Using firewood affects human and environmental problem. In order to reduce this problem different alternative energy sources are becoming on implement. From these biogas plants is one. In Ethiopia, the implemented biogas gives advantageous only for lighting and cooking purposes. However, previously implemented biogas stove used for Injera baking have a problem of consuming a large amount of gas from the plant, loses a large amount of heat out and pressures are not equally distributed on the holes. This affects unequal distribution of heat on mitad. The problem raised above can be overcome by proper designing of the system biogas stove used for baking Injera.

3. OBJECTIVE OF THE RESEARCH

- ✓ Collect necessary data's that is needed during designing the stoves.
- ✓ Designing of stove Parts needed: - design injector orifice, throttle size (mixing tubes), stability of flame, entrainments, burner ports, burner manifold and mitad supports.
- ✓ Designing of insulated materials to reduce extra energy loss.

4. DATA COLLECTION

Under this topic, the datas are collected through direct measurement, from literature reviews and some energy offices: - such as SNV and GIZ, are non-governmental organizations works on implement biogas plant for the community and household. The table shown below organizes these data's.

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i. Direct measurement data collection on temperature distribution.

Table 4.1: Temperature Distribution on Mitad, Injera and Sidewall of Mirte Stove in Bahirdar University Peda campus at 26°c of Ambient Temperature.

Test	Temp. mitad after heat (T°C)	Temp. of Injera after baked (T°C)	Temp.of Injera during baking (T°C)	Temp. side wall (T°C)
1	146	80	60	170
2	165	80	64	174
3	164	82	56	171
4	160	72	-	181
5	161	-	-	200
6	168	-	-	183
7	166	-	-	179
AVE.	161.4286	78.5	60	179.7143
STD	7.345228	4.434712	4	10.22602

ii. The data from GIZ expresses that the amount of water and Teff in dough is 60% and 40% respectively and dough measured at different test as shown in table 4.2 below.

Table 4.2: various weight of dough measured at different time.

Measurement	Weight of dough (gm)	60% Water (gm)	40% Teff (gm)
1	580	348	232
2	581	348.6	232.4
3	582	349.2	232.8
4	581	348.6	232.4
5	580	348	232
Average	581	348.6	232.4
STD	0.83666	0.501996	0.334664
Sigma @95% conf. level	0.8849	0.53	0.354

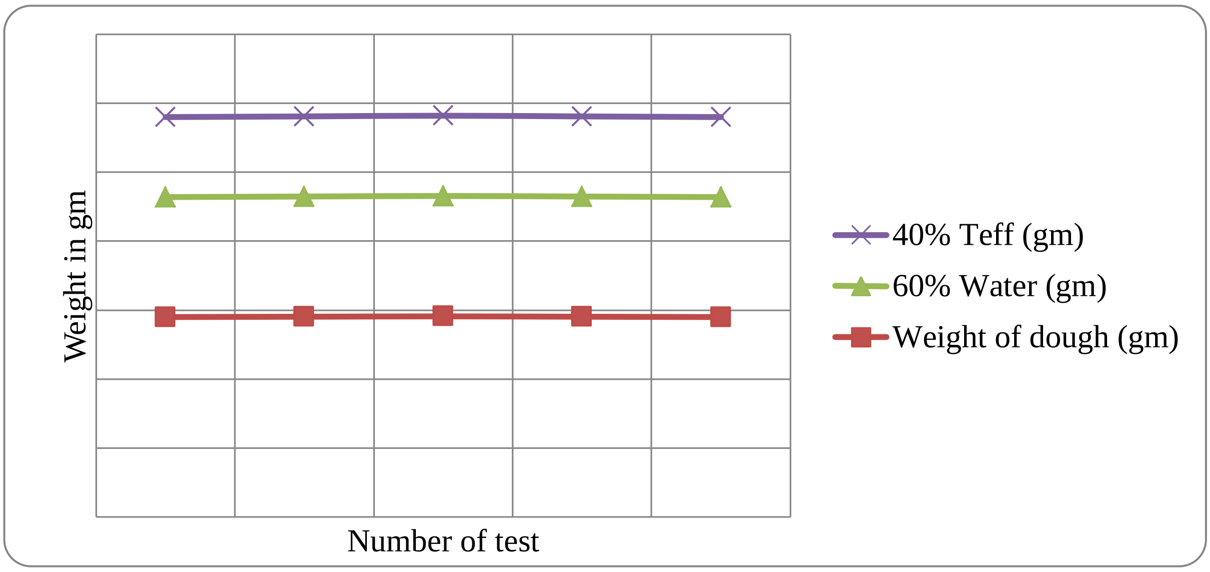

Figure 4.1: graphical expression of weight of dough, teff and water in each Injera at different test procedures of table 4.2

iii. These data's are collected from reference [9] that studies about properties of methane, CO_2 and biogas (40%CO_2 + 60%methane) they organized under table 4.3 and 4.4.

Table 4.3: Physical and Chemical Properties of Methane and Carbon Dioxide

Property	**Methane(CH_4)**	**Carbon dioxide(CO_2)**
Molecular weight	16.04	44.01
Specific gravity (Air=1)	0.554	1.52
Boiling point at 760mm	-161.49^{0}C	43.00^{0}C
Freezing point at 760mm	-182.48^{0}C	-56.61 ^{0}C
Critical temperature	-82.5^{0}C	31.11^{0}C
Critical pressure	47.363kg/m^3	75.369 kg/cm^2
Heat capacity ratio	1.307	1.303
Heat of combustion	38.13MJ/m^3	-
Limit of inflammability	5-15%by volume	-
Stoichiometric in Air	0.0947by volume or 0.0581 by mass	-
Ignition temperature	650^{0}C	-

Table 4.4: Properties of Biogas Relevant For Designing a Stove and Lamps

Property	**value**
Methane and carbon dioxide content 60%	60 % and 40 % (v/v) respectively
Calorific value	22 MJ/m3
Specific gravity	0.940
Flame speed factor	11.1
Air requirement for combustion	5.7 m^3/m^3
Combustion speed	40 cm/sec.
Inflammability in air	6-25 %

Table 4.5: Different Data Collections from Different Materials [4] [12] [15] [16].

No	**specifications**	**dimensions**
1	Mitad diameter	60 cm
2	Efficiency of three stone	7%
3	Efficiency of Lakech	25%
4	Efficiency of local closed mitad	40%
5	Temperature loses in mitad	150 to 180 ^oC
6	Biogas plants temperature work	35^{o}C
7	Efficiency of biogas burners for stoves	55%
8	Efficiencies of biogas burners for engines	24%
9	Efficiencies of biogas burners for lamps	3%
10	Efficiencies of biogas burners for heat – power	88%

5. DESIGN ANALYSIS OF BIOGAS STOVE AND ENERGY DEMAND DETERMINATION

5.1. Estimation of Energy Demand for Injera Baking

Energy demand needed to bake Injera from ambient temperature to heating temperature is given by;

$$Q = m_{dough} \times C_{p,dough}(T_2 - T_1)/t \qquad (5.1)$$

Where; M_{dough}= mass of dough=581gm

= ρ_{dough}* V_{dough}

C_{pw}= specific heat of water at 100 ^oC= 4.174kJ/kg ^oC

C_{pteff}= spec.heat of teff at 100^oC =1.046 kJ/kg ^oC

T_2=combustion temperature =126^oC

T_1= room/ambient temperature=26^oC

Estimated time to bake one Injera is =120 seconds

M_w=0.6*0.581=0.3486kg, since the dough contains 60% of water

M_{teff} =0.4*0.581=0.2324kg

Substituting the above value

$$Q_1 = \frac{m_w \times C_{pw} \times (T_2 - T_1)}{t}) \qquad (5.2)$$

$$Q_1 = \frac{0.3486 \times 4.174 \times (126 - 26)}{120\sec} = 1.212KW$$

$$Q_2 = \frac{m_{teff} \times C_{pteff} \times (T_2 - T_1)}{t}) \qquad (5.3)$$

$$Q_2 = \frac{0.2324 \times 1.046 \times (126 - 26)}{120} = 0.20KW$$

$Q=Q_1 + Q_2$ = 1.212 KW + 0.202 KW = 1.414 KW

5.2. Design Analysis of Biogas Stove Burner

The average power required for baking an Injera on mitad is 1.414 KW. A typical baking of mitad is 60 cm in diameter and an efficiency of mirte 25 % are indicated on above chapter 4
Power output = 1.414KW (Combination of eq. 5.2 and 5.3)

$$P_{required} = \frac{P_{out}}{efficiency} \qquad (5.4)$$

Then, the output power from flame combustion required to bake an Injera is 1.414 KW /.25 = 5.656 KW.

The energy needed at one hour is as follows. 5.656$\times$3600 = 20.36MJ/h

Biogas flow rate (Q_{gas}) is determined as follows $Q_{gas} = \dfrac{\text{Energy needed}}{\text{Calorific Value of biogas}}$ (5.5)

The biogas flow rate required is then: $\dfrac{20.36}{22} = 0.93m^3h^{-1}$

5.2.1. Design of Orifice Injector

An injector needs to be wisely designed and positioned in order to confirm the correct supply of gas and air to the burner. The size and shape of the injector orifice control the gas flow rate and hence heat input for a given gas composition and supply pressure. The siting of the injector with respect to the mixing tube affects air entrainment, so the injector must be positioned with a high degree of precision during manufacture of a burner or assembly of an appliance. Let, d_o is an

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injector orifice diameter, Q_{gas} is flow of biogas through the orifice, s is specific gravity of gas, C_d is coefficient of discharge and P is biogas pressure supply.

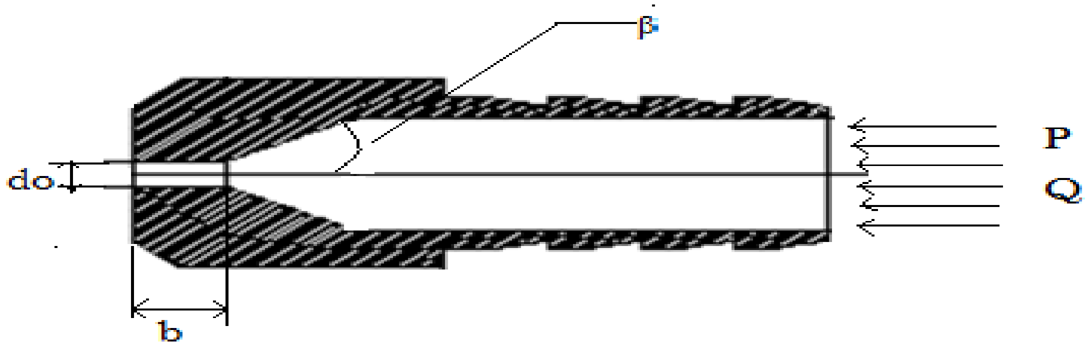

Figure 5.1: Injector Orifice

$$d_o = \sqrt{\frac{Q_{gas}}{0.036C_d}} \sqrt[4]{\frac{s}{P}} \qquad (5.6)$$

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

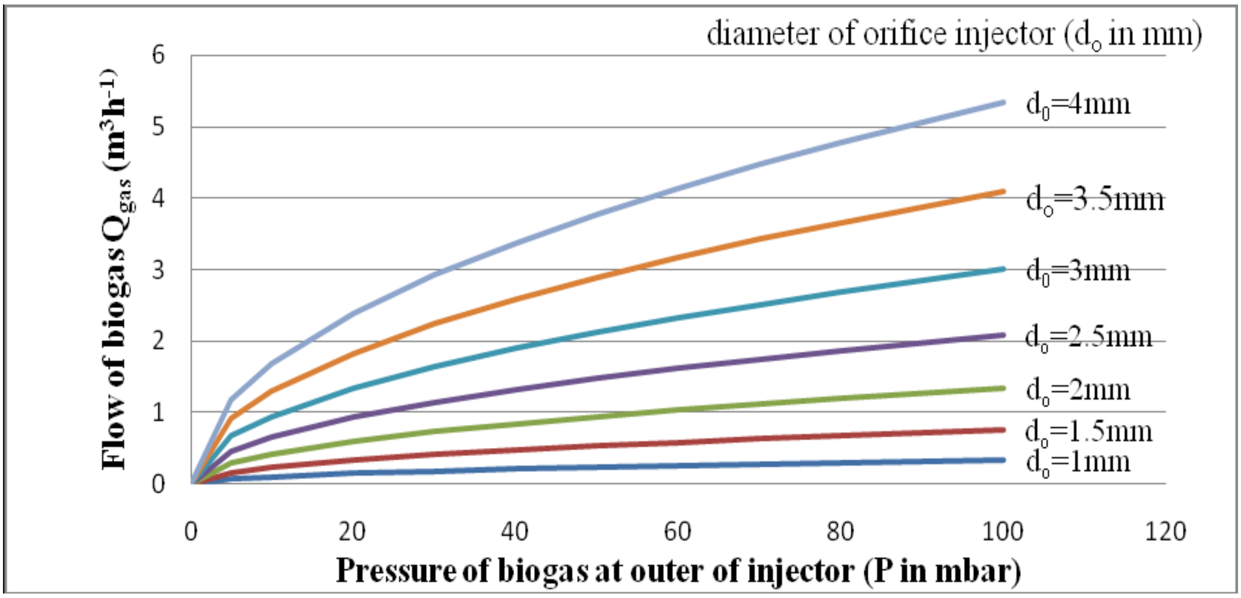

Figure 5.2: Graph of biogas flow versus pressure and orifice diameter at different variations.

Where, $Q_{gas} = 0.036C_d d^2 \sqrt{\frac{P}{s}}$ d is the orifice diameter (mm), c_d is taken as 0.9.

Using a suitable injector with a C_d of 0.9, and a gas supply pressure of 10 mbar (=102 mm water gauge), the injector size (d_0) is:

The injector size is determined as indicated on equation (5.6)

$$d_o = \sqrt{\frac{0.93}{0.036 \times 0.9}} \sqrt[4]{\frac{0.94}{10}} = 5.357 \times 0.554 = 3.0mm$$

The area of the injector orifice (A_0) is determined as follows.

$$A_0 = \pi d^2 / 4$$

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$$A_0 = \pi \times (3.0 \times 10^{-3})^2 / 4 = 7.068 \times 10^{-6} m^2$$

The velocity of gas in the orifice (V_0) is:

$$V_0 = \frac{Q_{gas}}{3.6 \times 10^3 A_0} \qquad (5.7)$$

$$V_0 = 36.55 ms^{-1}$$

N.B. Material selection of an injector is aluminum bar that is soft for drilling, cheap and available on market.

5.2.2. Throat Tube (mixing tube)

From the complete combustion process the Stoichiometric primary aeration required is 5.5, then the entrainment ratio r should be 5.5/2 = 2.75

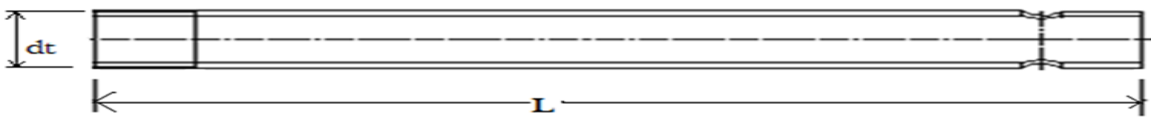

Figure 5.3: Throat Tube Diagram

By using prigs formula:

$$d_t = (\frac{r}{\sqrt{s}} + 1) d_0 \qquad (5.8)$$

$$d_t = (\frac{2.75}{\sqrt{0.94}} + 1) \times 3.0 = 11.50 mm. = 12 mm$$

However, it is better to increase this value to give aeration much greater than optimum and then use air controls to adjust the airflow. A suitable value for the throat diameter might be 21 mm, giving a maximum possible aeration of r = 5.5 which is Stoichiometric.

Then, the diameter of throat is

$d_t = (\frac{r}{\sqrt{s}} + 1) d_o = (\frac{5.5}{\sqrt{0.94}} + 1) \times 3.0 = 20.02 mm = 21 mm$ However, its exact size depends on the size of the pipe. The area of the throat determined is 346.36 mm^2

The air inlet ports must have an area similar to that of the throat.

The gas pressure in the throat can be calculated:

$$P_t = P_0 - \rho \frac{v_0^2}{2g} [1 - (\frac{d_0}{d_t})^4] \qquad (5.9)$$

$$P_t = 10^5 - 1.0994 \frac{36.55^2}{2 \times 9.81} [1 - (\frac{3.0}{21})^4]$$
$$= 10^5 - 74.825 pa$$

The mixture flow rate at optimum aeration is:

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$$Q_m = \frac{Q_{gas}(1+r)}{3600} \qquad (5.10)$$

$$Q_m = \frac{0.93(1+2.75)}{3600} = 0.97\times10^{-3}\,m^3s^{-1}$$

The pressure drop in the mixing tube, which should be at least 315 mm long (15 $\times d_t$), can be calculated:

$$R_e = \frac{4\rho Q_m}{\pi\mu d_t} \qquad (5.11)$$

$$R_e = \frac{4\times1.15\times9.7\times10^{-4}}{\pi\times1.71\times10^{-5}\times0.021} = 3955.162$$

$R_e > 2000$,

So, $f = \frac{0.316}{R_e^{1/4}} = \frac{0.316}{3655.162^{1/4}} = 0.03985$ and

$$\Delta P = \frac{f'}{2}\rho\frac{16Q_m^2}{\pi^2\times d_t^5}L_m \qquad (5.12)$$

$\Delta P = 0.03985\times1.15\times\frac{8\times(9.7\times10^{-4})^2}{\pi^2\times0.021^5}\times0.315 = 2.6955Pa$ This is much lower than the driving pressure in the throat (74.825Pa).

N.B. Material selection for throat tube is steel pipe that is used for water pipe having a diameter of 21mm with 315 mm length.

5.1.3. Burner Port

Burner port is at which the gas flows from it and burnt. It is more affected by high temperature and the material selected for this is stainless steel resists a temperature of flame.

The total burner port area can now be chosen:

$$A_p > \frac{Q_m}{0.25} \qquad (5.13)$$

$$A_p > \frac{9.7\times10^{-4}}{0.25} > 0.00388m^2$$

Using 5 mm diameter holes, the total number required will be

$$n_p = \frac{4A_p}{\pi d_p^2} \qquad (5.14)$$

$$n_p = \frac{4\times0.00408}{\pi\times0.005^2} = 198$$

Using the flame stabilization, it should be possible to reduce this number of burner ports, by up to 1/3, so 66 holes may be sufficient.

The biogas stove does use 66 holes at 5 mm diameters (total burner port area $=12.96\times10^{-4}m^2$), set at 90^0 to the vertical and the flames are fairly expanded on the left area. A larger burner port area would allow for greater flame stability.

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Using 66 holes, with 10 mm gaps between holes, arranged in a circular pattern, gives a total circumference of 66 × (10 + 5) = 990 mm. The holes centers are then placed around a circle of diameter 315 mm. using more burner ports of the same diameter would mean a larger circle and a larger area over which the heat is distributed.

5.3. Flame Height Determination

The flame height (L_f) be related to heat released (Q) rate with diameter of flame $D_{f.}$ The calculation of flame length is as follows.

$$L = 0.235\ Q^{2/5} - 1.02D \quad (5.15)$$

Where Q is given by KW and D is given by m

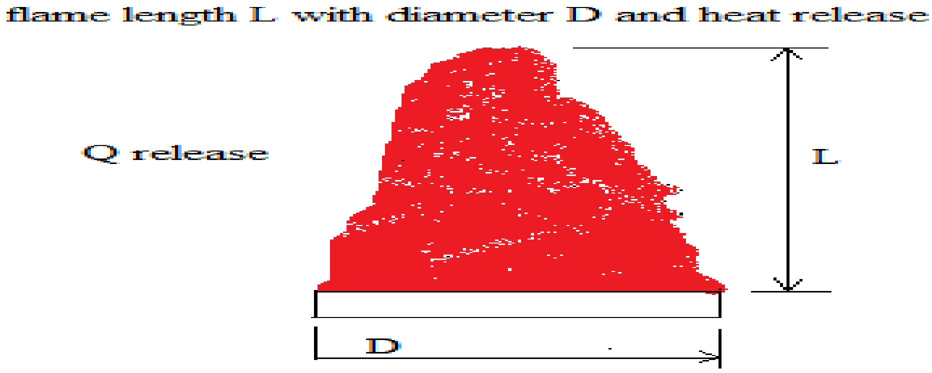

Figure 5.4: Flame Length

We determine the flame length as follows. $\dot{Q}$ is determined above by equation 5.4 and is 5.656 KW for all holes 66 and for one hole, which has a diameter of 5 mm, is 0.0857KW.

$$L = 0.235 Q^{2/5} - 1.02D$$

$$L = 0.235 \times (0.0857)^{2/5} - 1.02 \times 0.005 = 82.86mm$$

Of the 5.656 KW power, the length of the flame passes through each hole is determined 82.86 mm.

5.4. Determination of biogas digester in different biogas plants and amount of time needed to use the gas produced for the designed stove for Injera baking

In Ethiopia, different biogas digesters are implemented. Among these as it is indicated under chapter 4, a $6m^3$, $8m^3$ and $10m^3$ are implemented by non-governmental and governmental organizations. Especially the rural Ethiopian habitants use cow dung as a raw material for biogas production. The volume of biogas produced from these plants is calculated as follows [17].

The volume of biogas V_b, is given by

$$V_b = Cm_O \quad (5.16)$$

$V_b = Cm_O = 0.24m^3/kg \times 15kg/day = 3.6m^3/day$ Where C is the biogas per unit, dry mass of whole input (0.2 to 0.4-m^3 per kg) and m_O is the mass of the dry input (e.g. 2kg per day per cow).

The volume of fluid (V_f) in the digester is given by

$$V_f = \frac{m_0}{\rho_m} \quad (5.17)$$

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Where ρ_m is the density of dry material in the fluid (~50 Kg/m^3). The volume of the digester is given by

$$V_d = \dot{V}_f t_r \qquad (5.18)$$

Where, $\dot{V}_f$ is the flow rate of the digester fluid and t_r is the retention time in the digester (~8 to 20 days)

As it is indicated on above equations (5.16, 5.17 and 5.18), the volume biogas having 6m^3, 8m^3 and 10m^3 digester sizes are determined as follows.

$$\dot{V}_f = \frac{V_d}{t_r} = \frac{6m^3}{20days} = 0.3m^3/day$$

Substituting $\dot{V}_f$ value in to equation (5.17) and get the value of m_O equal to

$$m_O = V_f \times \rho_m = 0.3m^3/day \times 50Kg/m^3 = 15Kg/day$$

Then, substitute the value of m_O in to equation (5.16) and determine the value of V_b Therefore, for 6m^3 of digester size 3.6m^3/day biogas could be produced. Similarly, for 8m^3 and 10m^3 could be summarized in table below.

Table 5.1: Different digester sizes, flow of digester fluid, mass of dry input and volume of biogas flow

V_d (m^3)	V_f (m^3/day)	m_O (Kg/day)	V_b (m^3/day)
6	0.3	15	3.6
8	0.4	20	4.8
10	0.5	25	6

Table 5-2: Different digester size, volume flow rate through injector, volume of biogas produced in digester and time needed to bake Injera for biogas produced which is listed on table 5.1)

V_d(m^3)	V_O(m^3/hour)	V_b(m^3/day)	Time needed Injera for biogas produced per day (h)
6	0.93	3.6	3 hour and 52 min
8	0.93	4.8	5hour and 9min
10	0.93	6	6hour and 27min

5.5. Heat transfer through the wall of an insulated material

Heat loss through the insulative stove of cylindrical pipe is determined as follows. Insulative materials are: ashes and clay mixture and forms brick covered the wall side of biogas stove. This mixture has a thermal conductivity of 0.18W/m^{2}k. The diagram is shown as follows.

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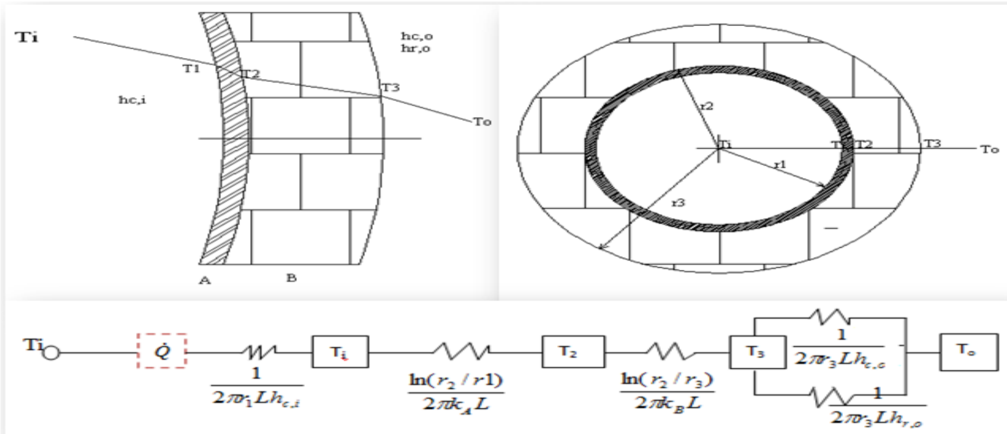

Figure 5.5: An insulated stove showing the temperature distribution and thermal circuit where, materials a and b are stainless steel and brick (clay + ashes)

$\dot{Q}$ is the heat flow across the face at r equal that the face r + Δr and U is over all heat transfer coefficient and A the circumferential area of the cylinder.

$$\dot{Q} = UA(T_i - T_o) = \frac{T_i - T_o}{\frac{1}{UA}} \qquad (5.19)$$

Then, summing the resistances in the thermal network,

$$\frac{1}{UA} = \frac{1}{2\pi r_1 L h_{c,i}} + \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi k_A L} + \frac{\ln\left(\frac{r_3}{r_2}\right)}{2\pi k_A L} + \frac{1}{2\pi r_3 L (h_{c,o} + h_{r,o})} \qquad (5.20)$$

Where, L is length of the insulated cylindrical pipe, r_1 is inner radius of the pipe, r_2 is the outer diameter of the inner plate, r_3 is outer radius of the insulated material, hc,i is interfacial conductance (W/m2K), K_A is thermal conductivity of the inner plate, K_B is the thermal conductivity of insulated material, hc,o is outer conventional heat transfer coefficient, hr,o is radiation heat transfer coefficient**.**

However, convectional heat transfer coefficient is determined as follows

$$h_c = \frac{N_u \times K}{L_n} \qquad (5.21)$$

For vertical plate and heat flows in horizontal direction

$$N_u = 0.062(G_r)^{0.327} \qquad (5.22)$$

$$Gr = \frac{g\beta \Delta T L_n^3}{\gamma^2} \qquad (5.23)$$

Where, L_n is the space between the flame and plate

K is thermal conductivity

ΔT is temperature difference between the two faces

β is volumetric coefficient of expansion of air

V is fluid velocity

γ is kinematic viscosity

g is gravitational force

C_p is specific heat capacity

ρ is fluid density

N_u is non-dimensional heat transfer coefficient

G_r is ratio of buoyant to viscous forces; replace Re in the cases of natural convection.

Take 19cm space between the flame and plate. hc, i = 5.674 W/m^{2}k at internal temperature of stove taking at 400^{0}c. Similarly at T = 20^{o}c, determine hc, o = 3.53W/m^{2}k.

By using equation 5.19 and 5.20 we can determine heat flow through an insulated material. Where r_1 = 30 cm, r_2 = 31cm, Ti of flame 800^{0}c, K_A = 16.26 W/m^{2}k, K_B = 0.16 W/m^{2}k.

$$\dot{Q} = \frac{T_i - T_o}{\frac{1}{2\pi L h_{c,i}} + \frac{\ln(r_2/r_1)}{2\pi k_A L} + \frac{\ln(r_3/r_2)}{2\pi k_B L} + \frac{1}{2\pi r_3 L(h_{c,o} + h_{r,o})}}$$

By substituting the values in each variable and get the following equation and graph, also see appendix 5. That determines the graph below.

$$\dot{Q} = \frac{800^o c - T_o}{0.056642 + 2\ln(r_3/0.3) + 0.09/r_3} \quad (5.24)$$

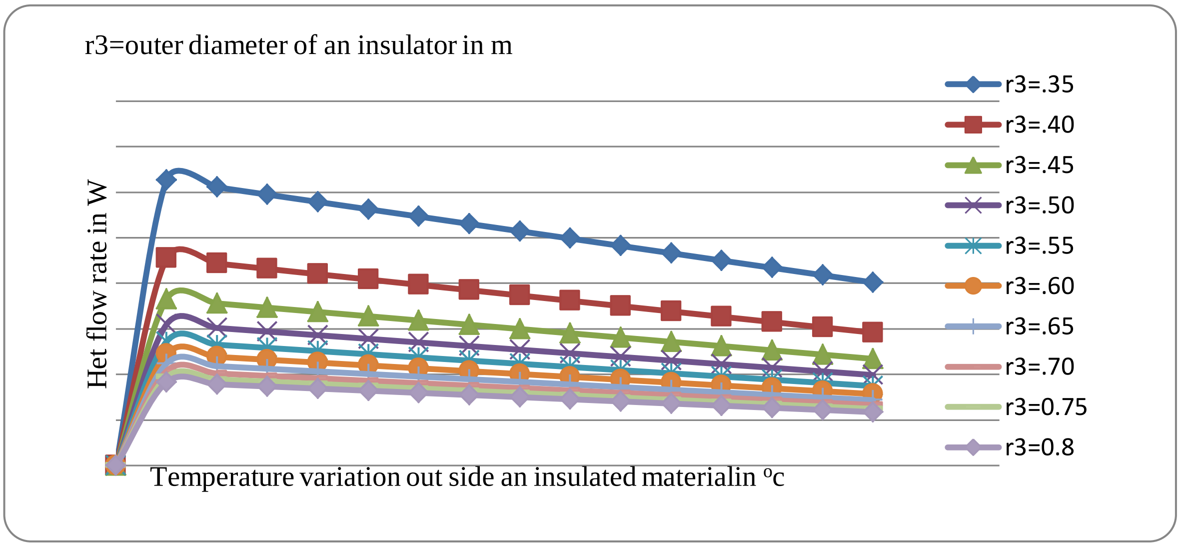

Figure 5.6: Graphical expressions of heat flow rate, temperature and radius variation of through an insulated material

5.6. Radiation intensity and shape factor determination between burner and mitad

The diagram below shows that mitad as A_2 and burner as A_1. The heat radiates from a burner of 21cm to mitad of 60cm diameter and the space between them is the height of flame determined 83 mm.

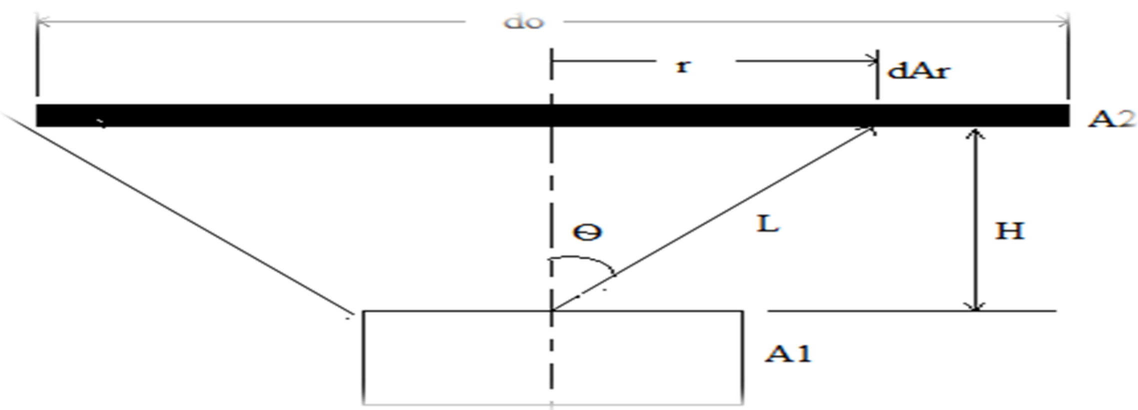

Figure 5.7: Diagram shows the rate of heat flow radiate between mitad and burner port

The heat flux q^+ leaving the burner is

$$q^+ = \frac{d\dot{Q}}{dA} = I\cos\theta d\omega \qquad (5.25)$$

$$d\dot{Q} = dAI^+\cos\theta d\omega \qquad (5.26)$$

$$d\omega = \frac{dA_s}{H^2} \qquad (5.27)$$

$$I = \frac{\sigma T^4}{\pi} \qquad (5.26)$$

Where, dω is differential solid angle, dA is elemental area on a body, I^+ is intensity of radiation leaving A_1, dA_s is elemental area on a body, H is space between burner and mitad, T is temperature of body A_1(temperature of flame = 1073^0k) and σ is Stefan boltz constant = $5.67*10^{-8}W/m^2k^4$

$$I = \frac{\sigma T^4}{\pi} = \frac{5.67\times10^{-8}\times1073^4}{\pi} = 2.3924\times10^4 W/m^2$$

The shape factor F_{12} between mitad and burner is determined as follows. Since the two bodies are parallel to each other,

$$F_{12} = \frac{1}{\pi}\int_{\omega_{12}}\cos\theta_1 d\omega \qquad (5.27)$$

Where dω = $dA_2\cos\theta_2/L^2$ and substituting in equation (6-13) get,

$$F_{12} = \frac{1}{\pi}\int_{A_2}\frac{\cos\theta_1\cos\theta_2}{L^2}dA_2 \qquad (5.28)$$

From above diagram we have cosθ = H/L, $L^2=r^2+H^2$, dA2 = 2πrdr and get equation (6.14)

$$F_{12} = \frac{1}{\pi}\int_{A_2}\frac{\cos\theta_1\cos\theta_2}{L^2}dA_2 = 2H^2\int_0^{d/2}\frac{rd_r}{(r^2+H^2)^2} = \frac{d_o^{\,2}}{d_o^{\,2}+4H^2}$$

Differential elements of heat flow on mitad $d\dot{Q}_{12}$ is a determined by multiplying F_{12} to $d\dot{Q}_1$.

$$d\dot{Q}_{12} = F_{12}\times d\dot{Q}_1 = \frac{d^2{}_o}{d_o^{\,2}+4H^2}\times dA_1 I_1 \qquad (5.29)$$

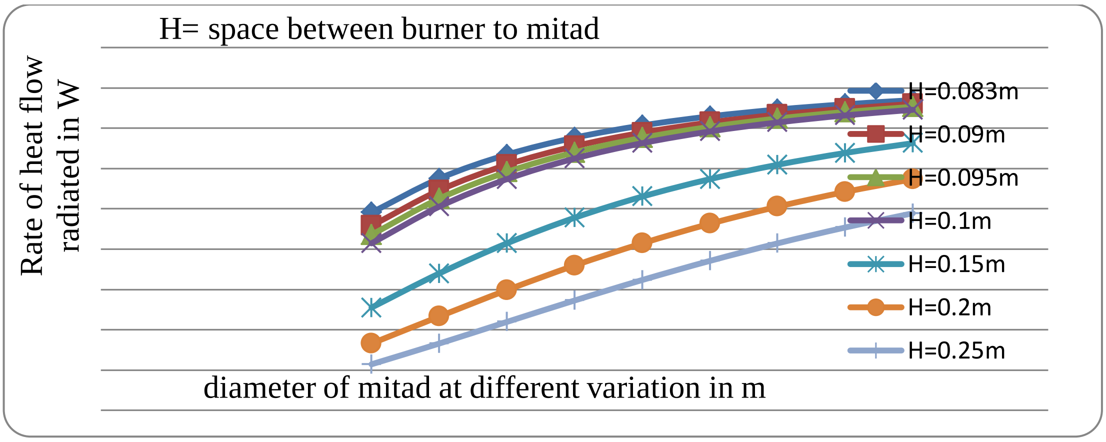

Figure 5.8: Graphical variation of rate of heat flow, variation diameter of mitad and space between burner and mitad by

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$$d\dot{Q}_{12} = \frac{d^2{}_o}{d_o{}^2 + 4H^2} \times 828.633W$$

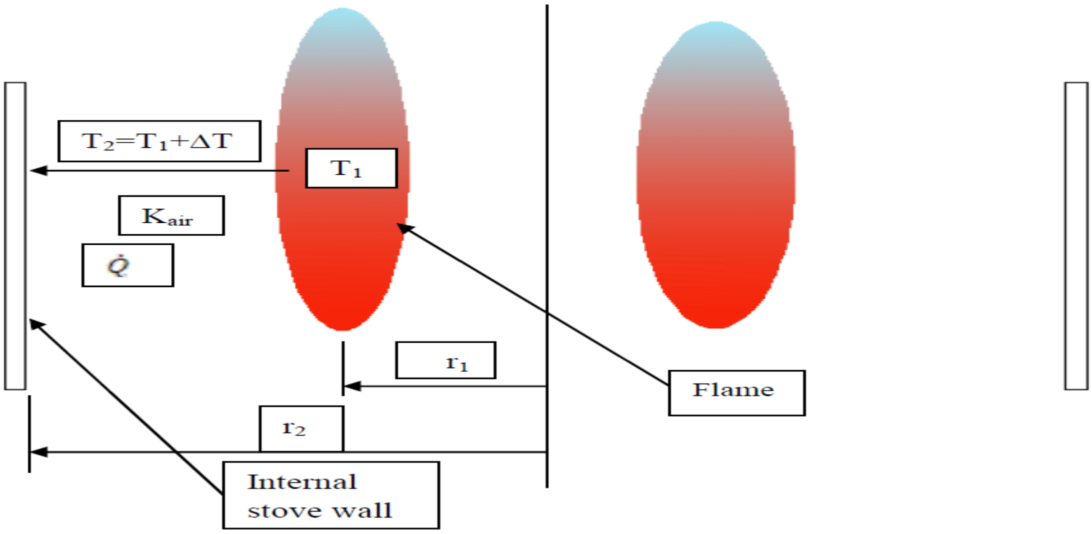

Figure 5.9: Diagrammatical expression of heat flow from flame to wall stove

Since the internal wall of stove has cylindrical shape, the equation of heat flow rate is given as follows.

$$\dot{Q} = \frac{2\pi k_{air} L(T_1 - T_2)}{\ln(r_2 / r_1)} \qquad (5.30)$$

L is given as 0.5m, r_1 is 0.11m, T_1 is 800^{o}C and K_{air} equals 0.072W/mK.

Substituting each values in equation 5.30 and get the following equation.

$$\dot{Q} = \frac{2\pi k_{air} L(T_1 - T_2)}{\ln(r_2 / r_1)} = \frac{0.2262(800 - T_2)}{\ln(r_2 / 0.11)} \qquad (5.31)$$

The graphical expression of heat flow rate as functions of temperature and space between them is shown below.

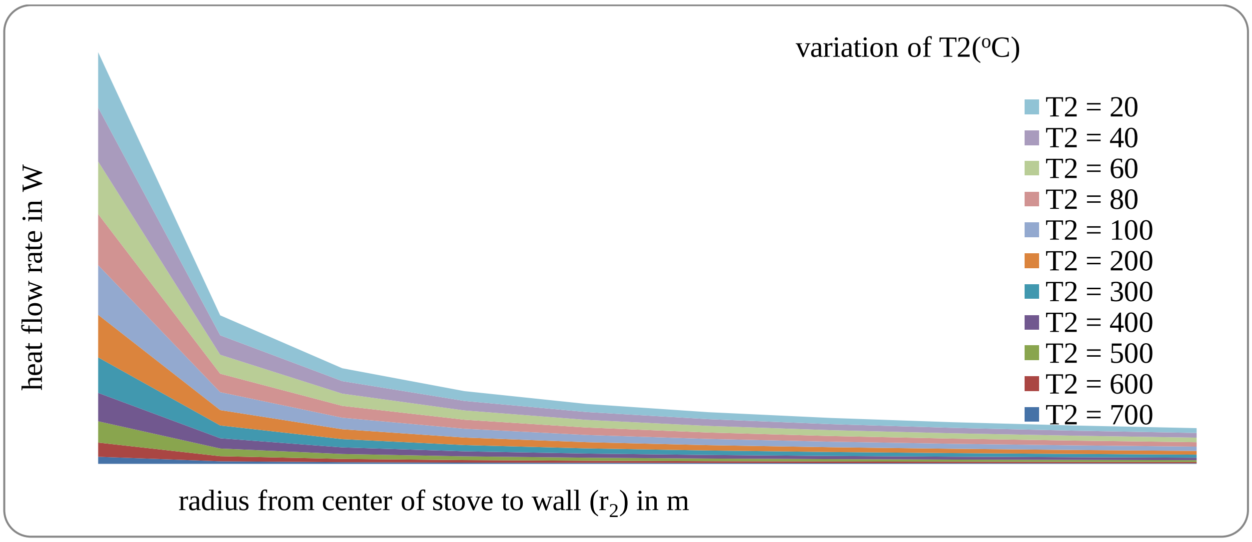

Figure 5.10: Graphical expression of conduction heat flow in air between wall side and flame at various temperature and space between them.

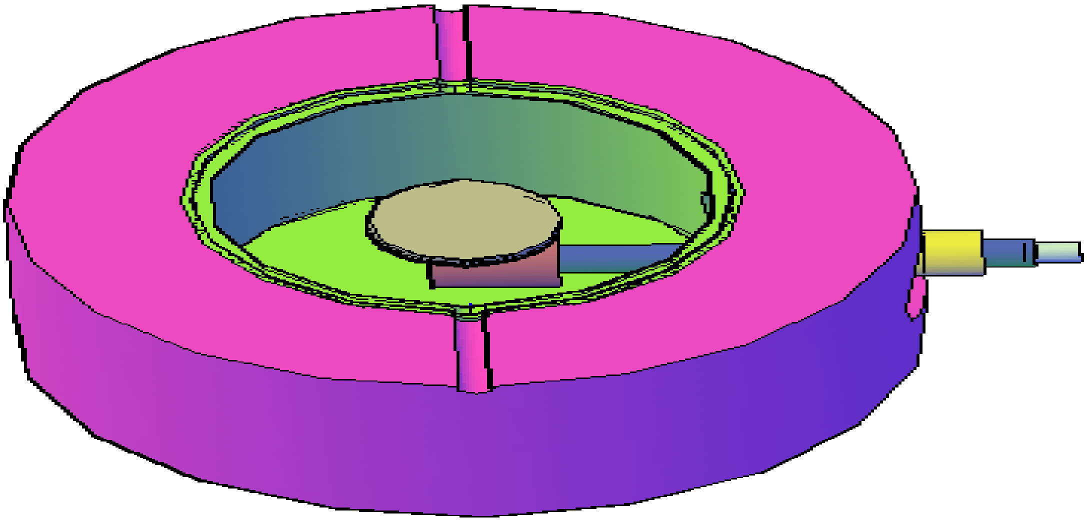

Figure 5.11: the final 3D model of biogas stove

6. CONCLUSION

The growing need for petroleum fuels and their limited availability in the world has necessitated the search for a replacement for these fuels. Biogas is a relatively clean gaseous fuel produced mainly from cattle dung and other biomass waste in an anaerobic digester. It is a non-toxic, non-poisonous gas. It can be used for direct combustion in cooking, in lighting applications, to power combustion engines for motive power or electricity generation, but due to the above mentioned properties, its use as a cooking fuel is more advantageous. The new burner was designed based on energy demand for baking. A biogas stove having gas consumption rating of $0.93m^3/h$ for community baking was designed for Injera baking. The time required to bake Injera for different biogas digesters was calculated. An insulation material used in this design was prepared from the mixture of different insulation materials.

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