Design of 22KW LPG Burner for an Oil Refinery Boiler

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Authors' contributions

This work was carried out in collaboration between both authors. Author BSK design the study, wrote the introduction, design the methodology and select the design parameters. Author GOA designed the figures, generated the data points in Table 1 and managed the analysis of results including conclusion. Both authors read and approved the final manuscript.

Original Research Article

ABSTRACT

The oil refining process is energy-intensive since every aspect of the process consumes energy. The need to minimize energy consumption when raising steam in boilers using Liquefied Petroleum Gas (LPG) burner was the focus of this study, to proffer techniques for improving optimum thermal efficiency via proper burner design and positioning. Burner design models were utilized to evaluate parameters for optimum combustion, to deliver the expected thermal output, including thermal efficiency. The results of this study suggest that, to design a 22KW LPG burner for an oil refinery boiler, the optimum values estimated for the burner parameters for efficient combustion at a gas flow rate of $1.89 \times 10^{-4} m^3$/sec, including Wobbe Index ($83285.7 KJ/m^3$), size of burner nozzle (1.9 mm), gas supply pressure (0.80 psi), length of burner slot for air entrainment (137.61 mm), size of burner pipe (46.48 mm), total orifice diameter (400.53 mm), and number of 3 mm. Studies elsewhere also suggest that if a proper angle between the burner axis and the boiler surface is achieved, significant changes in the amount of gas used can results positively in the direction of fuel utilization efficiency, thereby saving the cost of steam production in an LPG fired refinery boiler.

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1. INTRODUCTION

When fluid emerges from a nozzle, it interacts with fluid from the surroundings to form a jet. Jet flows are classified as fully separated flow because, after separation from the solid surface, the solid surface no longer play a significant role in their development. Fig. 1, represents regions of turbulent free jets and immediately downstream from the nozzle, there is a region (potential core), with which the velocity and concentration of nozzle fluid remain unchanged. Outside this region, a free boundary layer develops in which momentum and mass transfer are perpendicular to the direction of flow. The fully developed region of the jet is preceded by a transition region. The lengths of the potential core and transition region are about 4" to 5" and 10 nozzle diameters respectively. These values also depend on initial conditions, such as velocity distribution and turbulence level at the nozzle exit [1].

The fully developed regions of turbulent jets are similar and therefore axial and radial distribution of velocities and concentration can be described by relatively simple and general relationships, [2,3,4,5,5,6].

The last few years have witnessed an unprecedented increase in the price of energy available to the industry globally, following the first united action of the Organization of Petroleum Exporting Countries (OPEC) members on crude oil pricing in 1973. Any energy-saving measure adopted, offer a substantial reduction in operating costs and national energy conservation. As an energy-intensive process, the oil refining industry has always been aware of the need to minimize energy consumption, long before the energy crisis.

There are several approaches to the problem of reducing the specific energy consumption in the oil refining process. In the analysis of the oil refining process, two factors indicate the likelihood of process changes that might be instituted by the industry because of energy considerations. First, the steam raising process using boilers is by far the most significant energy-consuming unit. Second, the availability and cost of LPG have been enough of a major problem. It is, therefore, the focus of the study to proffer techniques and measures aimed at optimum thermal efficiency improvement through proper burner design, proper burner position, and other techniques in the spirit of energy conservation and also, as cost-saving measure.

The velocity at any point on the jet axis is independent of the nozzle diameter if the distance from the nozzle to the point is measured in terms of nozzle diameter. Fig. 2. represents the reciprocal of the axial velocity as a function of distance from the nozzle. The equation corresponding to the straight line in Fig. 2, can be given as:

$$\frac{\bar{U}_0}{\bar{U}_m} = 0.16\frac{x}{d_0} - 1.5 \qquad (1)$$

Where $\bar{U}_0$ is the mean velocity at nozzle exit axial direction, $\bar{U}_m$ is the maximum velocity in the axial direction, x is axial distance from nozzle exit and d_0 is nozzle diameter.
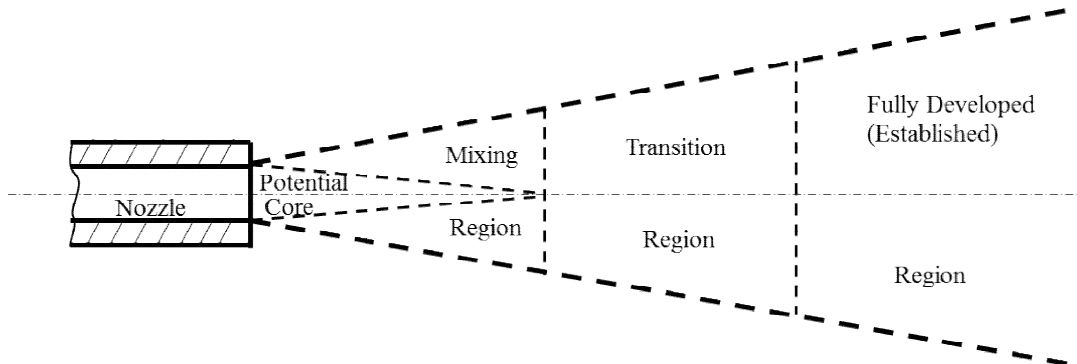
Fig. 1. Jet stream showing region of flow
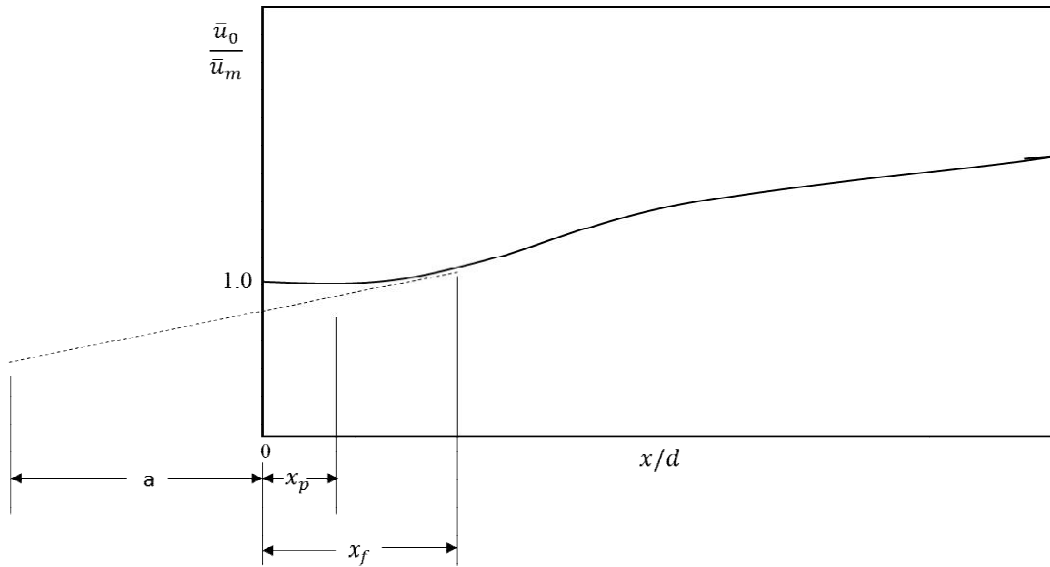
Fig. 2. Determination of apparent origin of a round jet through extrapolation of velocity

Because the mechanism of transfer of momentum and mass are the same, a similar relationship can be given for the axial distribution of concentration:

$$\frac{\bar{C}_0}{\bar{C}_m} = 0.22\frac{x}{d_0} - 1.5 \qquad (2)$$

Where $\bar{C}_0$ is the axial distribution of concentration and $\bar{C}_m$ is the maximum concentration

The horizontal part of the curve in Fig. 2, represents the distribution within the potential core and the difference between the slopes of the lines of velocity and concentration (Equations 1 and 2) is due to the difference in the value of the coefficients for the turbulent transfer of momentum and mass respectively.

For an enclosed jet such as a jet issuing into a duct, two extreme cases can be considered:

a) There is ample of supply of secondary flow surrounding the jet so that entrainment by jet is uninterrupted until it expands to reach the wall of the duct and
b) The surrounding secondary flow is less than that which the jet can entrain.

As a consequence of momentum exchange between the jet and surrounding, fluid is entrained from the surrounding across the boundary of the jet. This entrainment is due to friction, which depends upon the exchange coefficient and the velocity gradient. The turbulent exchange coefficient is several orders of magnitude higher than the molecular exchange coefficient. The magnitude of entrainment can be illustrated by the fact that a turbulent jet entrains fluid, equivalent to the mass flow rate of the nozzle fluid, for about every three nozzles diameter distance along the jet axis.

If no external force act on the free jet, the total momentum is conserved in every direction. When an external force is acting in any one direction, momentum is not conserved in that direction, but a force balance by the rate of change of momentum.

The general relationship, which has been developed for single free round isodensity jet, can be extended to more complex systems by using the concept of equivalent nozzle diameter. In its complex form, this base on the conservation of momentum flux. In a non-constant density system, the jet entrains fluid of ambient density and because of the high rate of entrainment, the density of fluid within the boundary of the jet, approaches the density of the surrounding within a short distance from the nozzle exit.

Beer and Chigier [1], reported that for non-constant density systems, the ratio of mass entrained to the mass of gas is given by the general relationship:

$$\frac{m_e}{m_0} = 0.32\left(\frac{\rho_a}{\rho_0}\right)^{\frac{1}{2}}\frac{x}{d_0} - 1 \qquad (3)$$

Where $\frac{m_e}{m_0}$ = ratio of mass of air-entrained to the mass of gas, ρ_a is the density of air, ρ_0 is the density of gas, d_0 is the nozzle diameter and x is the distance from nozzle exit.

Fig. 2 shows that

$$\frac{x}{d_0} = \frac{x_a}{d_0} + \frac{x'}{d_0} \tag{4}$$

Where x_a is the distance from nozzle exit to effective origin and x' is the distance from nozzle exit to the beginning of a fully developed region.

Equation 3, then becomes:

$$\frac{m_e}{m_0} = 0.32 \left(\frac{\rho_a}{\rho_0}\right)^{\frac{1}{2}} \left(\frac{x_a}{d_0} + \frac{x'}{d_0}\right) - 1 \tag{5}$$

But entrainment starts only at the nozzle exit. It is, therefore, necessary to find the apparent origin where the mass of air-entrained is zero. This means that:

$$0 = 0.32 \left(\frac{\rho_a}{\rho_0}\right)^{\frac{1}{2}} \frac{x_a}{d_0} - 1$$

$$3.125 \left(\frac{\rho_a}{\rho_0}\right)^{\frac{1}{2}} = \frac{x_a}{d_0}$$

$$3.125 (SG)^{\frac{1}{2}} = \frac{x_a}{d_0}$$

Substituting this value of $\frac{x_a}{d_0}$ in Equation 4, we have

$$\frac{m_e}{m_0} = 0.32 \left(\frac{\rho_a}{\rho_0}\right)^{\frac{1}{2}} \left(3.125(SG)^{\frac{1}{2}} + \frac{x'}{d_0}\right) - 1 \tag{6}$$

$$\frac{m_e}{m_0} = 0.32 \left(3.125 \frac{(SG)^{\frac{1}{2}}}{(SG)^{\frac{1}{2}}} + \frac{x'}{d_0(SG)^{\frac{1}{2}}}\right) - 1 \tag{7}$$

Equation 6 simplifies to:

$$\frac{m_e}{m_0} = \left(1 + \frac{0.32x'}{d_0(SG)^{\frac{1}{2}}}\right)$$

Equation 7 is based on the nozzle exit and not the apparent origin

In terms of volumetric flow rate, we have:

$$\frac{m_e}{m_0} = \frac{V_e}{V_0 SG} = \left(\frac{0.32x'}{d_0(SG)^{\frac{1}{2}}}\right)$$

Simplifying we have"

$$\frac{V_e}{V_0} = \left(\frac{0.32x'(SG)^{\frac{1}{2}}}{d_0}\right) \tag{8}$$

$$\frac{x'}{d_0} = \left(\frac{3.125V_e}{V_0(SG)^{\frac{1}{2}}}\right) \tag{9}$$

Equation 9 is the relationship for determining the slot length to be given from nozzle exit on the burner pipe for air entrainment.

For a free jet with no external force acting on it, the momentum is conserved (Fig. 3). For a non-constant density system.

$$m_0 U_0 = (m_e + m_o)\overline{U}_x = Constant \tag{10}$$

Where m_e is the mass of entrained air, m_0 is the mass of gas and $\overline{U}_x$ is the average velocity of the mixture.
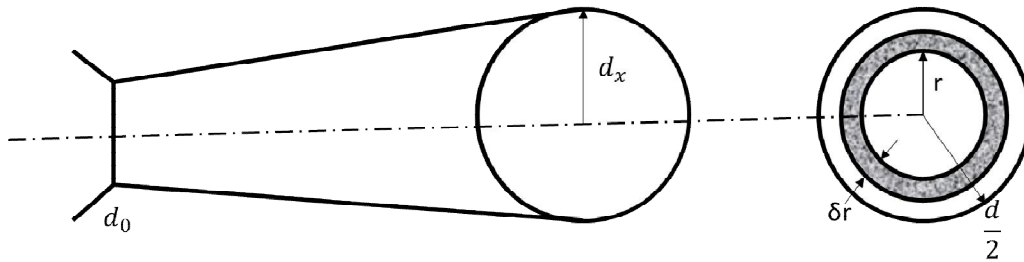
Let momentum

$$dI = dm\overline{U}$$

Fig. 3. Section of a jet

Where dm is the mass flowing around the small area,

$$dI = dm\bar{U} = \rho\bar{U}dv$$
$$dv = UdA = \bar{U}\pi(r + dr)^2 - \pi r^2$$
$$= 2\bar{U}\pi rdr \qquad (11)$$

Hence

$$dI = \rho\bar{U}^2(2\pi rdr)$$
$$I_x = \int_0^r dI = 2\pi\bar{U}^2\rho \int rdr$$
$$= 2\pi\bar{U}^2\rho \frac{r^2}{2} = \pi\bar{U}^2\rho r^2 = \pi\bar{U}_x^2\rho_x \frac{d_x^2}{4} \qquad (12)$$

At the nozzle exit

$$I_0 = \pi\bar{U}_0^2\rho_0 \frac{d_0^2}{4} \qquad (13)$$

Since momentum is conserved, $I_x = I_0$ and we have after simplifying

$$\frac{d_x}{d_0} = \frac{\bar{U}_0}{\bar{U}_x}\left(\frac{\rho_0}{\rho_x}\right)^{\frac{1}{2}} \qquad (14)$$

Where d_x is the diameter of the jet just at the point it touches the wall of the confining pipe.

Velocity at nozzle exit:

$$U_0 = \frac{jet\ volume\ at\ nozzle\ exit\ V_o}{Area\ of\ nozzle}$$

Therefore,

$$U_0 = \frac{4\ V_o}{\pi d_0^2} \qquad (15)$$

Volume flow at distance x from nozzle exit is the sum of jet flow V_0 and entrained flow V_e. Then

$$\bar{U}_x = \frac{(V_0 + V_e)}{\pi d_x^2} \qquad (16)$$

Substituting equations 15 and 16 into 14, we have:

$$\frac{d_x}{d_0} = \frac{4V_0\pi d_x^2}{4\pi d_0^2(V_0 + V_e)}\left(\frac{\rho_0}{\rho_x}\right)^{\frac{1}{2}}$$

After simplification, we have:

$$\frac{d_x}{d_0} = \frac{(V_0 + V_e)}{V_0}\left(\frac{\rho_x}{\rho_0}\right)^{\frac{1}{2}} \qquad (17)$$

Simplifying further, Equation 17 becomes

$$\frac{d_x}{d_0} = 1 + \frac{V_e}{V_0}\left(\frac{\rho_x}{\rho_0}\right)^{\frac{1}{2}} \qquad (18)$$

Where ρ_x is the density or specific gravity of mixture of gas and air and ρ_0 is the density or specific gravity of jet at the exit.

Equation 20, is useful when it is required to determine the size of the burner pipe. It gives the diameter of the jet just at the point it touches the wall of the confining pipe.

2. METHODOLOGY

The methodology adopted here includes evaluation of the burner parameters for optimum combustion, to deliver the expected thermal output. Other thermal efficiency improvement measures for this process have also been suggested.

2.1 Proper Burner Design

The fuel volume flowrate, the nozzle size and its flow characteristics, the length of burner slot for air entrainment and the size of burner pipe are evaluated to deliver the expected thermal output of the LPG fired industrial burner.

2.2 Evaluation of the Fuel Volume Flowrate

If the thermal capacity of the burner and the calorific value of the gas used are known, then the volume flow rate can be obtained by applying the relationship:

$$\dot{V} = \frac{q}{CV} \qquad (19)$$

Where q is the thermal output of the burner, $\dot{V}$ is the volume flowrate of the gas and CV is the calorific value of the gas.

The maximum thermal output of the burner q, is given as 22KW. The burner is been design to use LPG and the calorific value of this fuel has been obtained to be 11600kJ/m3 (see Appendix 1). Substituting these values into Equation 19, the volume flow rate of the fuel become:

$$\dot{V} = \frac{22kJ/sec}{11600kJ/m^3} = 1.89x10^{-4}m^3/sec$$

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2.3 Determination of Burner Parameters

The burner parameters: size (area and diameter) of the nozzle needed to give the required gas flow rate and thermal output, size of the burner slot for air entrainment, size of the pipe and the number of the 3 mm orifices needed on the burner rim for flame stability are evaluated. These parameters are used in the design of the burner and accurate estimation of these parameters will result in proper design of the burner that will give the desired output. These burner parameters are determined through the use of established models for burner design.

2.3.1 Evaluation of the size (area and diameter) of the nozzle burner

In evaluating the size of the burner nozzle needed to give the required gas flow rate and therefore the required thermal output of the burner, the coefficient of discharge of the nozzle, C_D can be calculated according to the correlation:

$$C_D = \frac{\dot{V}}{A \left(\frac{2\Delta P}{\rho}\right)^{0.5}} \qquad (20)$$

Rearranging, to calculate for A, we have

$$A = \frac{\dot{V}}{C_D \left(\frac{2\Delta P}{\rho}\right)^{0.5}} \qquad (21)$$

Where A is the area of the nozzle (m^2), $\dot{V}$ is the volume flow rate of the gas (m^3/sec), ΔP is the static pressure drop, ρ is the density of the gas (kg/m^3) and C_D is the discharge coefficient.

Having obtained the value of the volume flowrate to be 1.89x10^{-4} m^3/sec, the density of the gas was found to be 1.97 kg/m^3 (see Appendix 2), and the static pressure drop taken as the regulator pressure output, the discharge coefficient can be determined experimentally by studying the flow characteristics of a typical nozzle [7]. The value of C_D obtained corresponding to the gas flow rate is 0.88.

Therefore, applying Equation 19 and substituting the known values, the nozzle area and diameter are 3.828x10^{-7} m^2 and 6.98x10^{-4} m (0.698 mm).

2.3.1.1 Variation of nozzle size with gas supply pressure

Equation 21, gives the relationship between the gas supply pressure and the nozzle size at a given fluid flow rate. The pressure output of the regulator is usually taken as the gas supply pressure. If a particular flow rate is desired and using a regulator of known pressure output, then an appropriate nozzle size can be obtained according to Equation 21. To maintain the gas flow rate at 1.89x10^{-4} m^3/sec, Table 1 was developed using excel spreadsheet, to enable the user to choose an appropriate nozzle size at a supply pressure that corresponds to the pressure output of the pressure regulator intended for use.

The pressure regulator chosen for this design has a pressure output of 0.8 psi (5516 N/m^2). From Table 1, the corresponding nozzle diameter is approximately 1.9 mm.

Table 1. Variation of nozzle size with gas supply pressure at the flow of 1.89x10^{-4} m^3/sec

S/N	Pressure (N/m^2)	Nozzle area (m^2)	Nozzle diameter (m)	Nozzle diameter (mm)
1	310050	3.82808E-07	6.98E-04	0.70
2	275600	4.0603E-07	7.19E-04	0.72
3	241150	4.34064E-07	7.43E-04	0.74
4	206700	4.68843E-07	7.73E-04	0.77
5	172250	5.13591E-07	8.09E-04	0.81
6	137800	5.74212E-07	8.55E-04	0.85
7	103350	6.63043E-07	9.19E-04	0.92
8	68900	8.12059E-07	1.02E-03	1.02
9	34450	1.14842E-06	1.21E-03	1.21
10	5516	2.87106E-06	1.91E-03	1.91
11	3445	3.63164E-06	2.15E-03	2.15

2.3.2 Determination of the length of burner slot for air entrainment

The slot length from the nozzle exit allowed on the burner pipe for air entrainment, can be determined according to Equation 9

$$\frac{x'}{d_0} = \left(\frac{3.125V_e}{V_0(SG)^{\frac{1}{2}}}\right)$$

Where x' is the axial distance from the nozzle exit, d_0 is the nozzle diameter, $\frac{V_e}{V_0}$ is the volumetric air requirement and SG is the specific gravity of the gas.

The volumetric air requirement for LPG at 10% excess air has been calculated in Appendix 3 to be 32.45. The nozzle diameter is 1.9 mm and the specific gravity of the LPG was also determined to be 1.97. Substituting these values into Equation 9 and solving for the axial distance we have:

$$x' = 137.61 \ mm$$

2.3.3 Determination of size (diameter) of the burner pipe

The size of the burner pipe can be determined according to Equation 18:

$$\frac{d_x}{d_0} = 1 + \frac{V_e}{V_0}\left(\frac{\rho_x}{\rho_0}\right)^{\frac{1}{2}}$$

The parameters in Equation 18 are:

$$\rho_x = 1.03, \ \rho_0 = 1.97 \ (form \ Appendix \ 3), \ d_0$$
$$= 1.9 \ mm \ and \ \frac{V_e}{V_0} = 32.45$$

Substituting these values into Equation 18, and solving for d_x, we have

$$d_x = 46.48 \ mm$$

2.3.4 Determination of number of orifice on burner rim

According to [8], to achieve a stable flame on the burner rim, it would require that, the jet Velocity should be equal to or slightly lower than the burning velocity. The total area of the burner

orifice can be calculated according to the correlation:

$$Total \ Orifice \ Area = \frac{Volume \ flow \ rate \ of \ gas}{velocity \ of \ Gas}$$

The volume flow rate of gas is $1.89 \times 10^{-4} \ m^3$/sec.

The design is primarily concerned with a relatively slow uniform velocity of flame propagation and therefore, a flame speed of 6.0015 m/sec has been assumed which is quite adequate for the burner. Substituting values, we have

$$Total \ Orifice \ Area = 0.126 \ m^2 = 126000 \ mm^2$$

Then the

$$Total \ Orifice \ Diameter \ will \ be = \left(\frac{4A}{\pi}\right)^{0.5} =$$
$$\left(\frac{4x1 \ 26 \ 000}{\pi}\right)^{0.5} = 400.53 \ mm$$

If this is divided into 3mm diameter individual orifice, then

$$Number \ of \ such \ orifices = \frac{400.53}{3}$$
$$= 133.51 \ (\cong 134)$$

Therefore, about 134 uniformly spaced 3 mm burner orifice are required on the rim to achieve a stable flame at a slow uniform velocity of 0.0015 m/sec.

2.4 Evaluation of Wobbe Index of LPG

The calorific value is the most important property of fuel gas and is expressed on the volumetric basis. Given the dependence of volumetric flow rate through an orifice on specific gravity, it is desirable to combine both parameters into one term. This is known as the Wobbe Index of gas and is defined as:

$$Wobbe \ Index = \frac{calorific \ Vaue}{\sqrt{specific \ gravity}} \tag{25}$$

$$relative \ to \ air = 1$$

This represents the potential heat flow through an orifice at constant pressure:
For LPG

$$C_V = 116600kJ/m^3$$

$$SG = 1.97$$

Therefore,

$$Wobbe\ Index = \frac{116600}{\sqrt{1.97}} = 83285.7\ kJ/m^3$$

3. RESULTS AND DISCUSSION

The focus of this study is to proffer techniques and measures aimed at optimum thermal efficiency improvement, through proper burner design, proper burner positioning and reduction of excess air.

Summary of the design parameters of the LPG burner for an oil refinery boiler been investigated in this study have been listed in Table 2, where they have been compared with those of approximately the same burner rating but fired by biogas carried out by [9].

The 22KW LPG burner whose design parameters are given in Table 2, is a pre-aerated burner and air is supplied at atmospheric or slightly higher pressure. Air is entrained by gas issuing from the jet into the chamber, which is simply a short length of tube. The mouth of the burner is designed so that the linear velocity of the gas-air mixture leaving the tube is slightly higher than the flame speed of the mixture involved. If the gas velocity is less than the flame speed of the mixture, then the flame will flashback down the tube and seat on the neat gas nozzle, thus acting as a diffusion flame.

If the velocity exceeds the flame speed, the flame lift off the burner rim and takes up some position above the burner, at which it is very unstable. A stable flame is produced when the velocity is equal to the flame speed. This is because the gas velocity at the rim is zero (0). The parabolic form of the flame front is due to the velocity distribution across the burner tube of the gas laminar flow [10].

Flame may travel in one of three ways in a combustible mixture depending on the condition of the ignition. It may travel with a relatively slow uniform velocity, or with a faster accelerating velocity or the later may reach such proportion that the velocity becomes very high and the combustion mixture is said to "detonate". However, this work is primarily concerned with the slow uniform velocity of flame propagation [11].

3.1 Proper Burner Position and Reduction of Excess Air

The position of the burner to the kiln axis, contribute to the effectiveness of the heat generated. In this work, the position of the burner is assumed parallel to the kiln axis with a distance of 0.5 m to 1.3 m, and if a proper horizontal distance within this range is achieved, significant changes in the amount of gas used can results positively in the direction of fuel utilized efficiency.

Reduction in the amount of excess air while keeping the gas supply constant leads to higher temperature and increase in available heat. In other words, with the reduction of excess air, there will be less amount of gas consumption, required for maintaining the desired temperature. A 3.5% Oxygen in flue is about 18% excess air while the lowest desirable minimum excess air of 8% is detected by 1.5% Oxygen in the flue gas, with LPG as fuel [7].

Oxygen enrichment improves the efficiency of the fuel used. However, it has a disadvantage of increasing the cost of steam production in a boiler.

Table 2. Summary of design results for LPG burner compared with a similar biogas burner

S/No.	Parameters	Units	Designed Values	
			LPG Burner	Biogas Burner
1	Burner Rating	KW	22	22.6
2	Wobbe Index	kJ/m^3	83285.7	47998.9
3	Volumetric Flow rate of fuel	m^3/sec	1.89x10^{-4}	4.85x10^{-4}
4	Size of Burner Nozzle	mm	1.9	2.6
5	Gas Supply Pressure	psi	0.80	0.80
6	Length of Burner Slot for air entrainment	mm	137.61	87.74
7	Size of Burner Pipe	mm	46.48	27.86
8	Total Orifice Diameter	mm	400.53	641.29
9	Number of 3mm Orifices needed for flame Stability	-	134	214

4. CONCLUSION

This study has been able to proffer techniques and measures aimed at optimum thermal efficiency improvement, through proper burner design. These values of the design parameters have been compared with those of biogas with the same method, and approximately, the same thermal output obtained elsewhere. The study also suggests that proper burner position and reduction of excess air can lead to a significant reduction in the amount of gas used, leading to fuel utilization efficiency, thereby saving the cost of steam production in an LPG fired refinery boiler. Technology by its very nature is a discipline which only takes root when specific problems arise from human and industrial needs are addressed. Energy technology provides a basis for technological development and as such can lead to the achievement of a foundation from which local technology can find support.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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APPENDIX 1

Evaluation of Calorific Value of LPG

Peter D. Osborn (1985) has given the calorific values (vapourized) of commercial butane to be 122 MJ/m^3, and 95 MJ/m^3 for propane. Therefore, the calorific value of LPG which is made of 80% butane and 20% propane will be

(122 x 0.8 + 95 x 0.2) MJ/m^3 = 116.6 MJ/m^3 = 116600 KJ/m^3

APPENDIX 2

Evaluation of Density of LPG

The relative densities butane and propane compared to air at 15oC and 1016 atm are 2.07 kf/m3 and 1.55 kg/m3 respectively (Energy Management Handbook, 1985).

Therefore, density of LPG (80% butane and 20% propane) is estimated as follows:

$$Density \ of \ LPG = \frac{0.8(2.07)}{0.2 + 0.8} + \frac{0.2(1.55)}{0.2 + 0.8} = 1.97 \ kg/m^3$$
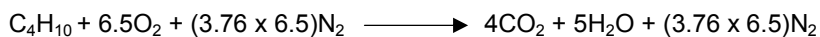
APPENDIX 3

Evaluation of Volumetric Air Requirement for LPG

For Propane (C_3H_8) only:

$$C_3H_8 + 5O_2 + (3.76 \times 5)N_2 \longrightarrow 3CO_2 + 4H_2O + (3.76 \times 5)N_2$$

Stoichiometric Volume of air required = 5 + (3.76 x 5) = 23.8

For Butane only:

$$C_4H_{10} + 6.5O_2 + (3.76 \times 6.5)N_2 \longrightarrow 4CO_2 + 5H_2O + (3.76 \times 6.5)N_2$$

Stoichiometric volume of air required = 6.5 + (3.76 x 6.5) = 30.94.

Therefore, for LPG (80% butane and 20% propane)

Stoichiometric volume of air required $\frac{V_e}{V_0}$ is

(23.8 x 0.2) + (30.94 x 0.8) = 29.5

At 10% excess air, volumetric air requirement for LPG,

$$\frac{V_e}{V_0} = 29.5x1.10 = 32.45$$