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THE ANALYSIS OF RECIRCULATION ZONE AND IGNITION POSITION OF NON-PREMIXED BLUFF-BODY FOR BIOGAS MILD COMBUSTION

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ABSTRACT

Combustion ignition study is important due to the combustion process becoming more lean and efficient. This paper studied the recirculation zone and ignition location for the bluff-body non-premixed MILD burner with biogas used as fuel. The location of the ignition was critical to ensure the spark energy supply during the ignition process is successful ignite the mixture of air and fuel. The numerical calculations were done using the commercial code ANSYS-Fluent to simulate the furnace with bluff-body burner to determine the recirculation zone. The turbulence model used was the realizable k-ε model. The inner recirculation zone between the air and fuel nozzle is the best location for the ignition point since low velocity of air and fuel mixing will assist the ignition process. This is because the ignition energy will have time to ignite the mixture in the low speed of turbulent swirl flow. The most suitable location with the highest possibility of ignition is the centre of the recirculation zone.

Keywords: computational fluid dynamics, bluff-body MILD burner, recirculation zone, biogas, spark ignition

INTRODUCTION

The economic development and the increase of the population lead to the increase the energy demand. Currently, overall energy demand is highly dependent on the combustion of fossil fuel. The combustion of fossil fuel is projected to fulfill about 80% of these energy requirements (IEA, 2009; Maczulak, 2010). With the current consumption rate, the fossil fuel will be depleted by 2042 (Shafie and Topal, 2009). Improvement of the combustion process is crucial and will significantly impact the efficiency of energy generations. One of the techniques to improve thermal efficiency and reduce NO_x is Moderate or Intense Low-oxygen Dilution (MILD) combustion (Dally et al., 2002, 2004; Cavaliere et al., 2004, 2008). This technique is also known as Flameless Oxidation (FLOX) by Wünning (1991, 1996), High-Temperature Air Combustion (HiTAC) by Katsuki and Hasegawa (1998) and Tsuji et al. (2003) and Colourless Distributed Combustion (CDC) by Arghode and Gupta (2010, 2011). The main characteristics of MILD combustion are an elevated temperature of reactants and low temperature increase in the combustion process. To increase the reactant temperature, the Exhaust Gas Recirculation (EGR) concept and input air preheat is normally implemented. The hot exhaust gases are utilised to heat and dilute the oxygen in the injected fresh air.

In normal combustion systems, greater attention is given to the fully burning state, like combustion efficiency, heat release rates, flame stability, pollutant emission or flame extinction. Combustion research has also focused on these aspects (Mastorakos, 2009). The ignition process research receives less attention: especially spark ignition of non-premixed flames. In the experimental work by Birch et al. (1981), the probability of successfully ignition was correlated with the probability of finding mixture within the flammability limits. Mastorakos (2009) studied the ignition of non-premixed flames and the effect of turbulence models on fuel and oxidizer mixing process. This turbulent mixing process later affects the probability of ignition. The spark ignition has been studied experimentally and numerically by a few researchers (Birch et al., 1981; Ahmed et al., 2007; Marchione et al., 2009; Mastorakos, 2009 and Oldenholf et al., 2010, 2011) and still need more attention.

A Tungsten electrode was used by Ahmed et al. (2006, 2007) as ignition rod for the spark ignition because it can withstand up to 3200 K. They studied electrode diameters of 1.0mm, 0.7mm, and 0.5mm for ignition probability and found that the ignition probability was increased with the decrease of the electrode diameter and increase in spark energy. Ahmed (2006) also concluded that the ignition probability is nearly always decreased with increasing flow velocity. This is in line with the result of later studies (Ahmed et al., 2007) that the ignition probability consistently decreased with increasing bulk velocity. The ignition location should be in the recirculation zone where the velocity is very low so the energy supply by the spark ignition rod will be utilised to ignite the mixture and not flushed away by high velocity of air or fuel or both reactant.

This paper examined the location and shape of recirculation zone for MILD combustion bluff-body burner. The purpose of the recirculation zone study in this paper is to determine the best location for the spark ignition rod installation for the experimental MILD burner. The experimental test rig for MILD burner is in progress to carry out the experimental study on MILD combustion for open furnace. The ignition used in the experimental study is spark ignition type which need accurate location to ensure the mixture will properly ignite to start the flame.

BIOGAS AND ENERGY BALANCE

Biogas is a low heating value gas also known as low calorific value (LCV) gas. Biogas consists of the mixture of 50–75% methane and 25–50% carbon dioxide. The lower the methane content, the lower the heating value for the biogas. Table 1 shows a comparison of energy balance for biogas with 60% methane and 40% carbon dioxide and natural gas with 97% methane. The summary was made for a furnace that operates in flameless mode with biogas and natural gas, and the conventional mode with natural gas. The supply of thermal energy was constant at about 21 kW for all combustion modes. The energy calculated includes all the input to the combustion chambers which are fuel, air through the cooling tubes and air that will be preheated by the regenerative honeycombs.

The efficiency for the combustion with conventional mode is only 41.4% whereas for biogas and natural gas in MILD mode it is 68% and 70% respectively. The comparison for the efficiency of flameless mode for biogas and natural gas shows that biogas only 2% lower than natural gas. This is not a big issue compared to the benefit of biogas to the global warming and greenhouse gas effect.

Table 1. Biogas and natural gas energy balance (Colorado et al, 2010).

Combustion mode (fuel)	Flameless mode (biogas)	Flameless mode (natural gas)	Conventional mode (natural gas)
Energy input (including fuel + combustion air + cooling air) (kW)	21.13	21.31	21.02
Energy losses through the wall (kW)	3.00	3.07	3.20
Energy removed by the cooling tubes (kW)	14.39	14.99	8.71
Energy output through the chimney (kW)	2.72	1.39	8.25
Energy of the combustion products after the regenerative system (kW)	1.01	1.36	0
Efficiency (%)	68.0	70.0	41.4

In order to recover the energy losses through the exhaust gas, the EGR concept was applied to the combustion system. EGR behaves differently to heat regenerators and it work by recirculating a portion of the flue gas back to the combustion chamber through the EGR pipe. Lloyd and Weinberg (1974), Weinberg (1996) and Choi and Katsuki (2001, 2002) used the concept of heat recirculation combustion. Weinberg (1996) demonstrated it in his famous Swiss-roll burner by transferring the heat from burned products to the unburned fresh mixture. He used double walls that separate the products and the mixture and act as a heat regenerator.

EGR was also used as a solution to avoid NO_x and soot formation. EGR with MILD combustion was used by Wünning and Wünning (1997), Katsuki and Hasegawa (1998) and many other researchers have utilised EGR in their experiment and numerical studies (Tsuji et al., 2003, Cavaliere et al., 2004, 2008; Colorado et al., 2010; Noor et al., 2012a, 2012b, 2012c and Abtahizadeh et al., 2012). The EGR volume ratio is:

$$\text{EGR Ratio} = \frac{\text{recycled exhaust gas volume}}{\text{total exhaust gas volume}} \quad (1)$$

EGR will dilute the oxygen and increase the intake air temperature to the combustion chamber. The volume of hot exhaust gas to flow back into the combustion chamber depends on the level of oxygen dilution and air pre-heating needed. EGR will reduce NO_x emissions of the oxygenated fuels by more than 55% since it reduces both the pressure (Raj and Sendilvelan, 2010) and the maximum combustion temperature.

CFD MODELLING

Prior to this century, experimental work is an effective way for testing and optimisation due to the limited capacity to do huge data calculation. Modern technology means computational modelling of complex problems is feasible and preferable to expensive, comprehensive experimental studies (Chandrasekharan, 2013). Building the computational models are giving researchers deeper insights into problems than building the experimental setup. Despite the benefit of computational method, the experiment method still important step to compare and validate the computational result. This feedback can be used to improve the computational method.

Computational Fluid Dynamics (CFD) offers a cost-effective method especially in the beginning of the combustor design and parameter setting stage. It was therefore used here to study the recirculation zone and optimise the ignition location. The first CFD modelling work for MILD combustion was started by Japanese heating industry where a few researchers (Ishii et al., 1997; Zhang et al., 1997 and Hino et al., 1998) carried out simulations of continuous slab reheating furnace with emphasis on NO_x formation. The simulation work was successful and continued with experimental technique. In the current work, the biogas configuration of 60% methane and 40% carbon dioxide (molar base) was used. This ratio of biogas was also used by a few other researchers (Pomeroy, 2008; Colorado et al., 2010; Scholz and Ellner, 2011; Salunkhe et al., 2012; Noor et al., 2012a, 2012b, 2012c; Keramiotis and Founti, 2013). Table 2 shows the CFD setup and typical data for combustion chamber.

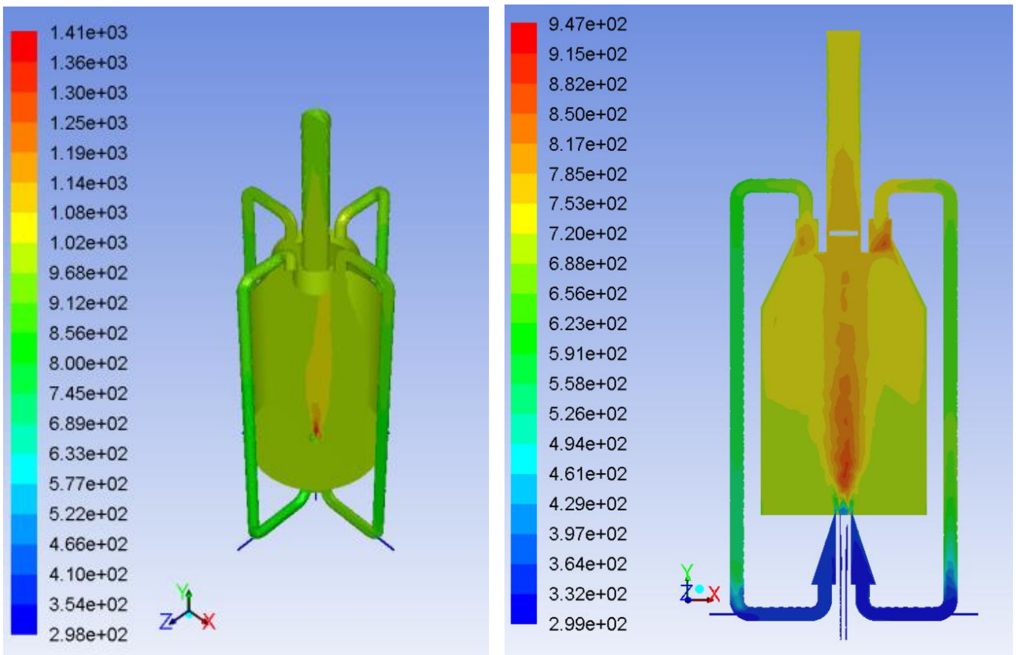
Table 2. Typical data for furnace and combustion chamber

Item	Data
Fuel	60% methane mixed with 40% carbon dioxide
Oxidiser	Atmospheric air and syntactic air at room temperature
Fuel Inlet	1 x 78.5 mm^2
Air Inlet	4 x 78.5 mm^2
Chamber size	Diameter 600mm, Height 860mm
EGR	4 EGR with 1962.5 mm^2 each inlet
Mesh method	Tetrahedrons (Patch conforming method) with 111,975 nodes and 501,831elements
Radiation model	Discrete Ordinate (DO) model. Absorption coefficient: Weighted Sum of Gray Gas (WSGGM) model.

The combustion chamber consists of four EGR pipe with inner diameter of 1962.5 mm^2 each. The MILD combustion simulation involved the solution of the chemical reactions, turbulent flows, heat transfer and species transport. In this work, the Reynolds-Averaged Navier–Stokes (RANS) equations together with a realizable k-ε turbulence model (Shih et al., 1995) [that developed based on standard k-ε turbulence model (Launder and Spalding, 1974)] are solved using commercial CFD software ANSYS Fluent 14.0 (Fluent, 2011). The discrete ordinate (DO) radiation model (Chui and Raithby, 1993) and absorption coefficient of weighted sum of gray gas (WSGGM) model is used in this work. Figure 1 shows early stage of the combustion process about 15 second after the ignition started. Figure 2 indicates that when MILD is achieved, temperature inside the combustion chambers will be homogenous.

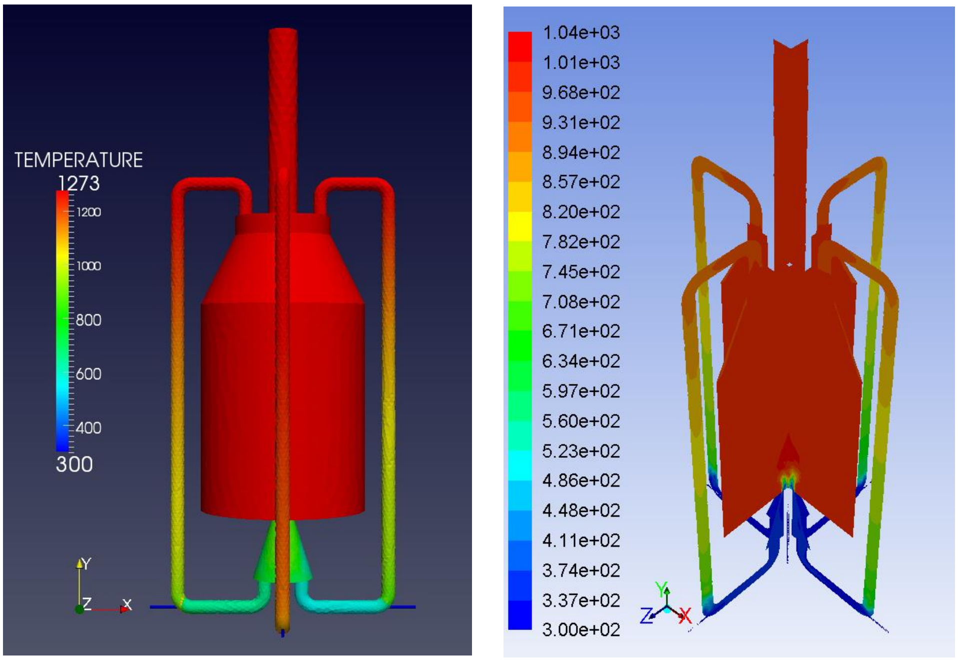
RECIRCULATION ZONE

Figure 3 shows the flame re-circulation zone on the schematic bluff-body burner diagram for 3.0 mm fuel nozzle and 10.0 mm annulus air nozzle as co-flow; this nozzle angle is 22°. The fuel velocity at exit is 75 m/s and air is at 5 m/s. The recirculation zone was formed and the centre of the recirculation zone was detected at x/D = 0.25 and r/D = 0.25. Figure 5(a) show that there other two zones in the flame schematic diagram are the flame neck zone and flame jet zone.



(a) (b)

Figure 1. Early stage of the of combustion process in open furnace, not MILD combustion state yet, (a) 3D image, (b) 2D image



(a) (b)

Figure 2. MILD combustion state was achieve, (a) furnace wall temperature at 1273 K, (b) inside chamber temperature 1040 K

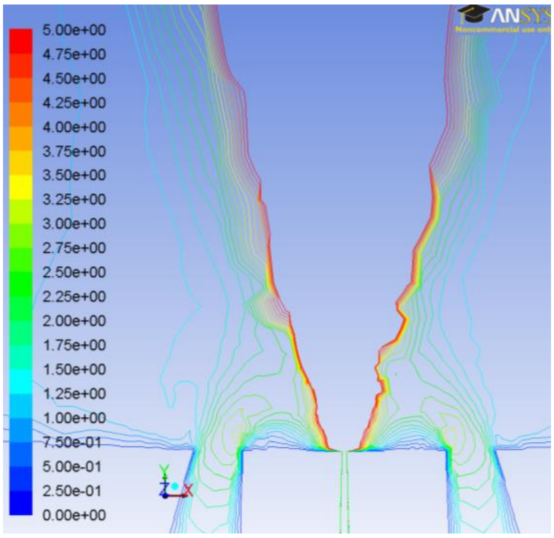
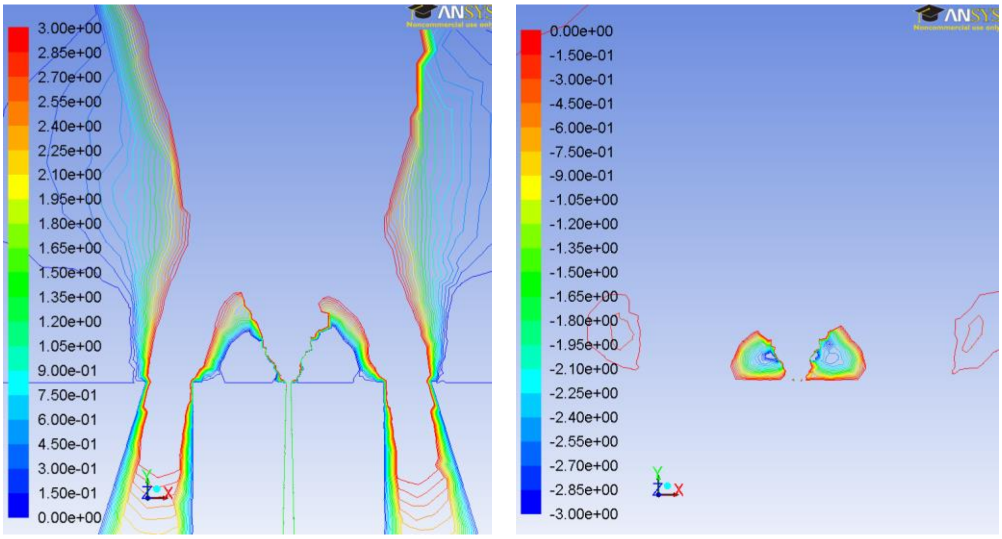


Figure 3. Contour of total velocity magnitude (0 to 5.0 m/s)

The inner recirculation zone was formed as two circles, a big and small circle of recirculation zone. From the analysis of Figure 3, 4(a) and 4(b), show that the recirculation zone can be divided into inner and outer recirculation zone as shown in Figure 5(b).

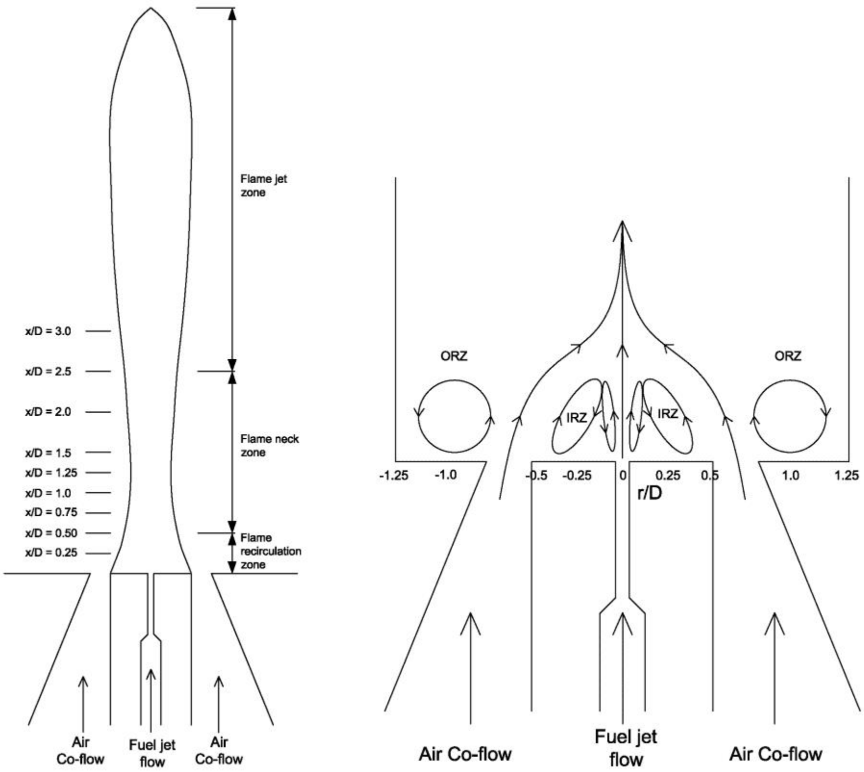


(a) (b)

Figure 4. Contour of Y velocity, (a) 0 to 3.0 m/s, (b) -3.0 to 0 m/s.

Figure 5 shows that there are two types of recirculation zone that are visualized in the swirl flow: the inner recirculation zone (IRZ) formed in between the air and fuel jet flow of bluff-body and the outer recirculation zone (ORZ) formed outside the annulus air flow. The recirculation zones were formed due to the bluff-body of the burner and create swirl flow around the air and fuel nozzle. The air velocity flow at 5 m/s and fuel at 70m/s, the width and the height of the recirculation zone were about 1 and 1.5 bluff-body diameters respectively. The recirculation of the mixture of fuel and air was important due to that process will create the turbulent flow of the mixture which will enhance the mixing process. The intensity of the IRZ is higher than ORZ due to the

IRZ is the recirculation formed in between the fuel and air jet flow. One cause of the higher intensity is the fuel jet velocity being much higher than the air jet velocity (Figure 3). In addition, IRZ is contained in a small volume within the air jet and has two vortices due to jets on both sides (Figure 5(b)), while ORZ occupies a bigger volume with a single vortex due to only one side being a jet.



(a) (b)

Figure 5. Schematic diagrams for bluff-body burner, (a) flame flow field with central fuel jet and annulus air co-flow, (b) flame re-circulation zone .

SPARK IGNITION LOCATION

Triantafyllidis et al. (2009) and Neophytou et al. (2012) concluded that the best location for ignition was in the centre of inner recirculation zone where the recirculation velocity is almost zero. This is important to ensure the spark energy supplied by the tungsten rod was not flushed away, thereby giving sufficient time for the spark energy to ignite the mixture of fuel and oxidant. Figure 6 shows the design of the ignition rod installation, using the values from the CFD study: $x/D = 0.25$ and $r/D = 0.25$ (vertically 10 mm from the nozzle base and horizontally 10 mm from centre of the nozzle).

CONCLUSION

A study on recirculation zone and the ignition location for the non-premixed MILD combustion bluff-body burner was done using CFD. The recirculation zone was formed due to the bluff-body, increasing turbulence of the flow of the fuel and air to make the mixture more homogenous and mix better than the flow without turbulent flow. The centre of recirculation zone was the best location to install the spark ignition rod. From the analysis, the most suitable location (with the highest possibility) of ignition is the centre of the recirculation zone. The centre of the inner recirculation zone for the current design is x/D = 0.25 and r/D = 0.25.

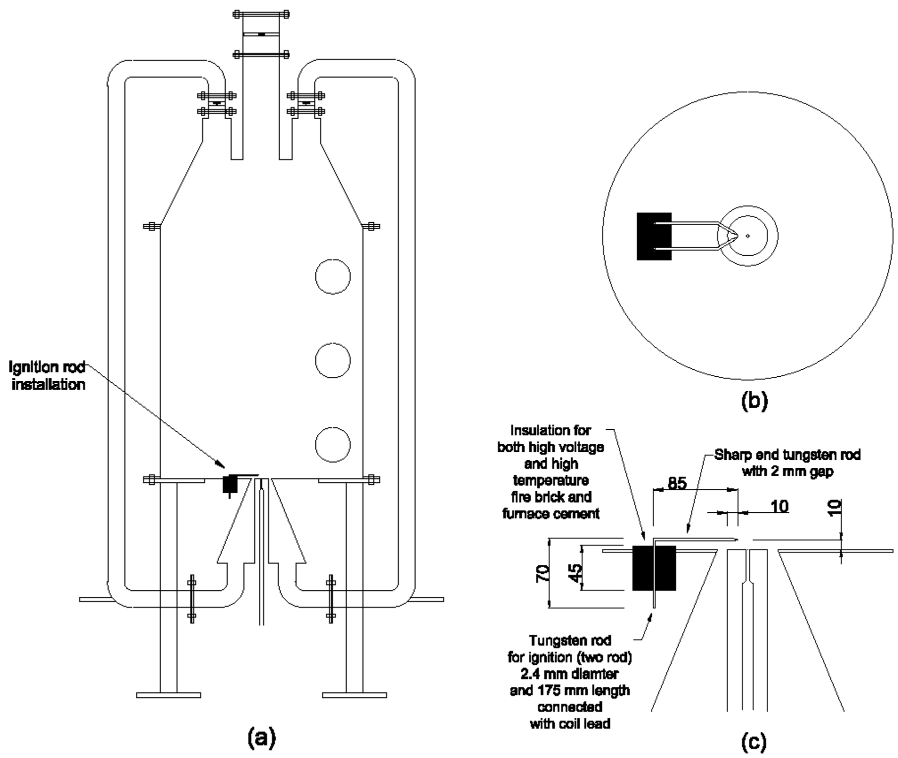


Figure 6. Schematic diagrams of combustion chamber, (a) ignition rod location, (b) location plan view and (c) location side view of ignition location and installation

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