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A PRELIMINARY STUDY OF CONTROL PARAMETERS FOR OPEN FURNACE MILD COMBUSTION USING CFD

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ABSTRACT

Pollution regulation and demand for efficient energy have driven the combustion community to work on combustion improvement. Moderate or Intense Low oxygen Dilution (MILD) combustion is one of the best alternative new technologies for clean and efficient combustion. MILD is proven to be a promising combustion technology for industrial applications. This paper studies the design stage for an open furnace with exhaust gas recirculation (EGR) captured from the flue gas. This study uses ANSYS Fluent to simulate and predict the parameters. The study started with 3D furnace with two EGR. Due to incorrect flow, modifications have been made to the EGR inlet and outlet. Finally 3D model with four EGR was developed and was successful in producing the desired flow in the EGR.

Keywords: MILD combustion, computational fluid dynamics, exhaust gas recirculation, turbulent, open furnace

INTRODUCTION

Energy supply for heating, cooking, transportation is one of the basic needs for mankind. Combustion of fossil fuel up to 2030 is projected to fulfil about 80% of world energy needs (Maczulak, 2010). Demand for reliable, efficient and clean energy is increasingly important with concerns over climate change and the reduction of pollution emissions from human activities (IEA, 2002, Jonathan, 2006, Pacala and Socolow, 2004, IPCC, 2007, Ghoniem, 2011). Climate change is the effect of rising concentrations of greenhouse gases. Carbon dioxide, nitrous oxide, chlorofluorocarbons and aerosols are the main greenhouse gases produced by human activities. Environmental issue and concerns are motivating factors for innovation in combustion technology employed in transportation and stationary power-generation applications.

Among fossil fuel, natural gas combustion is the most attractive since it produces less harm to the environment because it releases less carbon dioxide, nitrogen oxide, sulphur dioxide, particulate and mercury per unit energy compared to oil and coal (EIA, 1999). In 2009, the estimated natural gas reserves is 187.5 trillion cubic meters, which can supply up to 7×10^{15} MJ of energy, and the petroleum reserves can supply up to 1383 billion barrel which can supply 8.4×10^{15} MJ of energy [BP, 2010, 2011]. Table 1 show the comparison of pollutants for natural gas, oil and coal. The use of natural gas will reduce the impact of fossil fuel combustion on climate change. In order to reduce further on NO_x and other harmful pollutant, lean mixture will reduce the combustion temperature and reduce the formation of NO_x. Beside fuel NO_x and prompt NO_x, the thermal NO_x is the main NO_x formation will increase rapidly after the combustion temperature reach 1573 K (EPA, 1999) and 1810 K (AET, 2012). Figure 1 shows the formation of NO_x. In order to achieve the low NO_x emission and high thermal efficiency combustion, MILD combustion technology is the proper selection which will give low NO_x emission and high thermal efficiency combustion.

Table 1. Pollutant from fossil fuel (EIA, 1999)

No.	Pollutant	Gas	Oil	Coal
		(kg of pollutant per 109 kJ of energy input)		
1.	Carbon dioxide	273,780	383,760	486,720
2.	Carbon monoxide	94	77	487
3.	Nitrogen oxide	215	1,048	1,069
4.	Sulphur dioxide	2.34	2,625	6,063
5.	Particulate	16.4	197	6,420
6.	Mercury	0.00	0.016	0.037

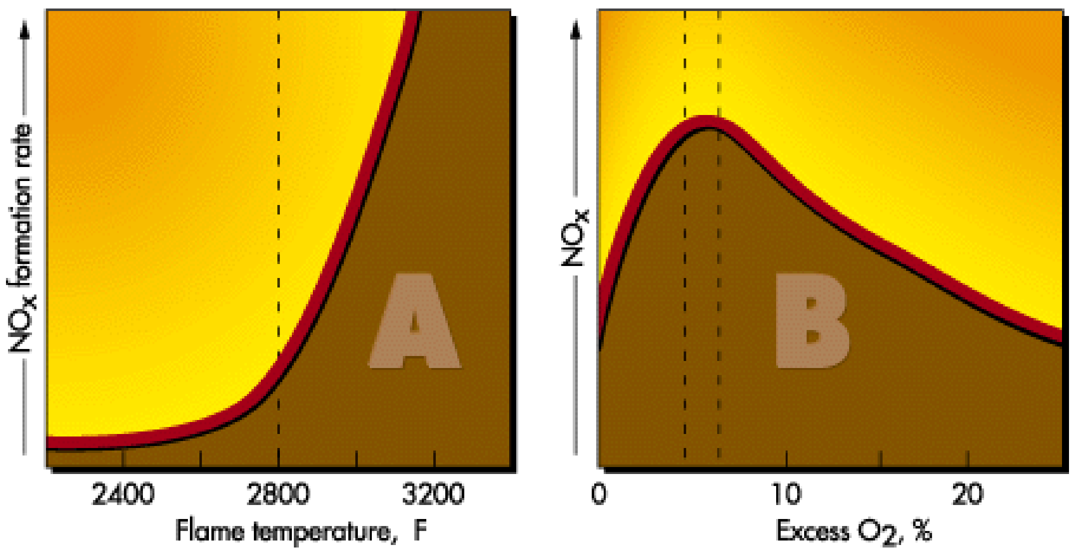


Figure 1. The rate of NO_x formation, (a) flame temperature in Fahrenheit (2800 F is equal to 1810 K) (b) percentage of oxygen level in the oxidiser (AET, 2012).

MILD combustion technology comes from the concept of excess enthalpy combustion (Hardestry and Weinberg, 1974). This combustion is also called flameless oxidation or FLOX (Wünning, 1991, 1996, Wünning and Wünning, 1997 and Milani and Wünning, 2007), low NOx injection (Orsino et al., 2001), Moderate or Intense Low-oxygen Dilution (MILD) combustion (Dally et al., 2002, Cavaliere and de Joannon, 2004, Christo and Dally, 2004) and high-temperature air combustion (HiTAC) (Katsuki and Hasegawa, 1998 and Tsuji et al., 2003). MILD combustion technology utilizes the heat and exhaust gas recirculation (EGR) or flue gas recirculation (FGR) to achieve stable low temperature combustion under a hot oxidant diluted condition. MILD combustion has been achieved experimentally (Dally et al., 2008, Li and Mi, 2010, Mi et al., 2010 and Li et al., 2010a, 2010b) and numerically (Scharler R and Obernberger, 2000, Giammartini et al., 2000, Awosope et al., 2006, Galletti et al., 2007, 2008, 2009, Mollica et al., 2009 and Szegö et al., 2003, 2009, 2010) in premixed and non-premixed combustion modes.

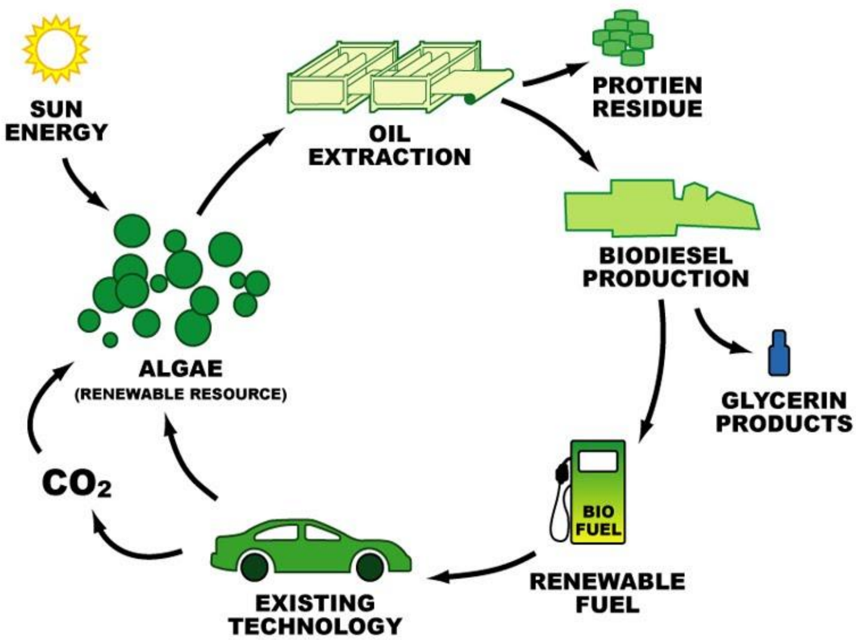


Figure 2. Carbon dioxide close cycle for biofuels (www.knol.google.com)

Beside NO_x emission and efficiency issue, another important issue in combustion is CO_2 emission which will also impact on greenhouse gasses (GHG) (Volk, 2008). Energy Information Administration (EIA, 2002, 2007 and 2011) reported that CO_2 emissions from combustion account for about 80% of anthropogenic GHG. In order to reduce CO_2 emissions, biogas will be the best alternative since biogas produced from biomass uses CO_2 in the photosynthesis stage; hence this will reduce the CO_2 in the atmosphere (Figure 2). The production of biofuels is normally based upon locally-available feedstocks including soybean, rapeseed, jatropha seed, palm oil, sunflower, cottonseed, tallow (animal fat) or even waste cooking oil. In the Australia and US, most biofuels are derived from soy beans while in Europe rapeseed is the largest source. Biogas normally consist of about 50% methane with the heating value of 21MJ/Nm^3, the density is 1.22 kg/m^3, which is similar to air: 1.29 kg/m^3 (Al-Seadi et al. 2008). Biogas also suitable for the internal combustion engine (ICE) used. Huang and Crookes (1998) and Borjesson and Mattiasson (2008) studied the efficiency of biogas for ICE, Caresana et al. (2011) used it for energy production, Colorado et al (2010) and Effuggi et al. (2008) use for MILD furnace combustion. The computational work was carried out using Fluent 13.0 (ANSYS 2010) in a preliminary study of the control parameters for open furnace combustion. The parameters involved are air and fuel velocity injected, nozzle design, chamber design, EGR design and exhaust design. The purpose of the study is to analyse and optimise the parameters and predict the flame behaviour.

GOVERNING EQUATIONS

Fluid flow governing equations consists of continuity equation, density, enthalpy, temperature, species mass fraction, turbulent kinetic energy (k), turbulent dissipation rate (ε). For the axisymmetric flow in low Mach number (M < 0.3) (Rehm and Baum, 1978 and Majda and Sethian, 1985), the transport equations are: mass (the continuity equation)

$$\frac{\partial \rho}{\partial t} + \nabla.\rho U = 0 \tag{1}$$

Velocity

$$\frac{\partial \rho (u,u,w)^T}{\partial t} + (\nabla.\rho U u, \nabla.\rho U v, \nabla.\rho U w)^T + \left(-\frac{\rho w^2}{r}, \rho g, \frac{\rho u w}{r}\right)^T = -\nabla \pi + \nabla.\tau \tag{2}$$

Enthalpy

$$\frac{\partial \rho h}{\partial t} + \nabla.\rho U = \nabla.\lambda_e \nabla T - \nabla.q_{rad} + \nabla.\sum_l \rho h_l(T) D_e \nabla m_l \tag{3}$$

Temperature

$$\rho c_\rho \frac{DT}{Dt} = \nabla.\lambda_e \nabla T - \nabla.\sum_l \rho h_l(T) D_e \nabla m_l - \rho \sum_l \frac{Dm_l}{Dt} h_l(T) \tag{4}$$

Species mass fraction

$$\frac{\partial \rho m_l}{\partial t} + \nabla.\rho U m_l = \nabla.D_e \rho \nabla m_l - R_l \tag{5}$$

The most common turbulent model is k-ε model (Jones and Launder, 1972, Launder and Sharma, 1974, Bardina et al., 1997 and Wilcox, 1998). The equation for turbulent kinetic energy (k) is (6) and turbulent dissipation rate (ε) is (7).

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + P_k + P_b - \rho\epsilon - Y_M + S_k \quad (6)$$

$$\frac{\partial}{\partial t}(\rho \epsilon) + \frac{\partial}{\partial x_i}(\rho \epsilon u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\epsilon}\right)\frac{\partial \epsilon}{\partial x_j}\right] + C_{1\epsilon}\frac{\epsilon}{k}(P_k + C_{3\epsilon}P_b) - \rho C_{2\epsilon}\frac{\epsilon^2}{k} + S_\epsilon \quad (7)$$

where turbulent viscosity, $\mu_t = \rho C_\mu \frac{k^2}{\epsilon}$, production of k, $P_k = -\rho\overline{u_i' u_j'}\frac{\partial u_j}{\partial x_i}$, effect of buoyancy, $P_b = \beta g_i \frac{\mu_t}{Pr_t}\frac{\partial T}{\partial x_i}$ and $\beta = -\frac{1}{\rho}(\frac{\partial \rho}{\partial T})_p$. In the effect of buoyancy, g_i is the component of the gravitational vector in the ith direction and Pr is turbulent Prandtl number. Pr is 0.85 for the standard and realizable k-epsilon model. Other model constants are $C_{1\epsilon}, C_{2\epsilon}, C_{3\epsilon}, C_\mu, \sigma_k$ and σ_ϵ.

The computational and simulation method to improve combustion process has been rapidly expanding over the last decade. Computational fluid dynamics (CFD) is an important tool to simulate and predict the behaviour of flame and all the parameters before the experimental work take place. CFD work will reduce the massive experimental cost especially during the beginning stage. Galletti et al. (2007) reported that recently the combustion and furnace industry shows the interest on CFD modeling. CFD may help in optimizing burners' performances such as injection nozzles and flue gas recirculation. CFD results must be validated with experimental work. Different scale of MILD combustion setups has been simulated using CFD software over the last decades (Danon, 2011). Turbulent flow is one of the important points in this study. Turbulent flow is needed for the oxidiser and fuel to mix before the combustion take place. Turbulent flow occurs at high Reynolds numbers and is a very complex process: even more complex when involved with combustion reaction or other chemical reaction. Tennekes and Lumley (1972) characterised the nature of the turbulence as irregularity, large Reynolds numbers, diffusivity, three-dimensional vorticity fluctuations and continuum phenomenon. Sensitivity to turbulence k-ε model (Jones and Launder, 1972, Launder and Sharma, 1974, Bardina et al. 1997 and Wilcox, 1998) was investigated but not reported in this paper. The model was initialised as follows, reference frame is relative to cell zone, gauge pressure is 0 Pa, x, y and z velocity are 0 m/s, turbulent kinetic energy is 16.8 m^2/s^2, turbulent dissipation rate is 40,751 m^2/s^3, pollutant NO, N_2O and HCN mass fraction is 0, Temperature is 600K, mean mass fraction is 0.5 and mixture fraction variance is 0.

RADIATION MODEL

The radiation model used in this work is Discrete Ordinate (DO) model (Chui and Raithby, 1993). Discrete ordinate model is applicable to a wide range of optical thicknesses. The optical thickness for MILD flames is not well defined or well known, making DO model seems like a good selection. The model solves a radiative transfer equation. The absorption coefficient used is weighted sum of gray gas model (WSGGM) which was conceptually developed by Hottel and Sarofim (1967) and used for spray combustion (Choi and Baek, 1996) and gas furnace (Liu et al. 1998). The

WSGGM is selected due to the reasonable compromise between the oversimplified gray gas model and a complete model.

CHEMICAL REACTION

The mass ratio of fuel and air was estimated by using the general equation of combustion (Equation 8). Assuming the combustion is using pure methane (CH_4) and air ($0.21O_2$ and $0.79N_2$), the general equation for hydrocarbon and air stoichiometric combustion is shown in equation (8) and the equation for methane and air combustion is shown in equation (9). From equation (9), the fuel to air ratio is 1:9.5. This ratio was used in early design stage for the air and fuel inlet and nozzle size (Table 2).

$$C_n H_m + \left(n + \frac{m}{4}\right)(O_2 + 3.76N_2) \rightarrow nCO_2 + \frac{m}{2}H_2O + 3.76\left(n + \frac{m}{4}\right)N_2 \quad (8)$$

$$CH_4 + (2)(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + 7.52N_2 \quad (9)$$

For more detailed stoichiometric combustion, equation for low calorific value (LCV) gas consists of 50% methane, 20% hydrogen and 30% carbon dioxide by mass fractions and EGR ratio is 50% is shown in equation (10) and (11).

$$(0.5CH_4 + 0.2H_2 + 0.3CO_2) + (1.1O_2 + 4.1N_2) + 1.0(0.8CO_2 + 1.2H_2O + 4.1N_4) \rightarrow 2.0(0.8CO_2 + 1.2H_2O + 4.1N_4) \quad (10)$$

$$(0.5CH_4 + 0.2H_2 + 0.3CO_2) + (1.1O_2 + 4.1N_2) + (0.8CO_2 + 1.2H_2O + 4.1N_4) \rightarrow (1.6CO_2 + 2.4H_2O + 8.2N_4) \quad (11)$$

RESULTS AND DISCUSSION

The computational work begin with normal combustion without EGR to check the combustion of LCV gas (biogas) consisting of 50% methane, 20% hydrogen and 30% carbon dioxide by mass fractions (Figure 3(a)). The gas enters the combustion chamber through 20 mm diameter fuel inlet at 40 m/s to 50 m/s into the combustion chamber. Air was injected at 80 to 100 m/s through an annulus gas inlet at the bottom of the chamber. The exhaust on the top of the combustion chamber is called stack. The stack consisting of damper acts as a stopper for the flue gas flow. If the damper was totally closed, the flue gas will not flow out of the chamber thru the stack. The damper can be adjusted to control the volume of flue gas flowing out from the chamber and flowing through into EGR pipe. A more detailed burner and chamber specification is shown in table 2. Once the combustion was stable, then two EGR pipes were added to the system to test the combustion with oxidiser diluted by flue gas through EGR (Figure 3(b) and 3(c)). At the beginning of the simulation, the EGR pipe was connected perpendicular and straight to the air inlet and exhaust outlet. Then a problem was encountered in the fluent, the fluid did not smoothly flow into the EGR pipe and out from the EGR pipe (Figure 3(b)). This is due to the hot flue gas flowing straight to the exhaust on the top of the chamber and also the opening of the exhaust being very small. Improvement was made to the input of the EGR. The smooth corner to help the fluid flow into the EGR pipe was added (Figure 3(c)). The result was encouraging with the flow showing significant improvement. This method was then implemented at the EGR outlet at the bottom of the burner figure 4(b).

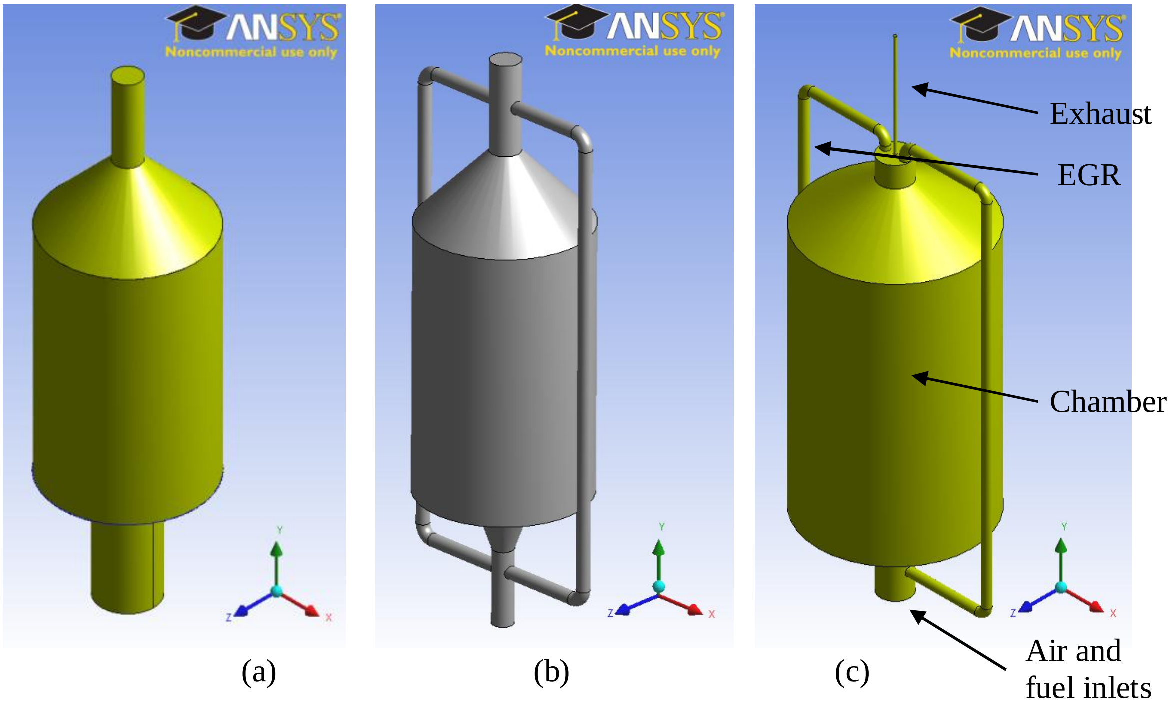
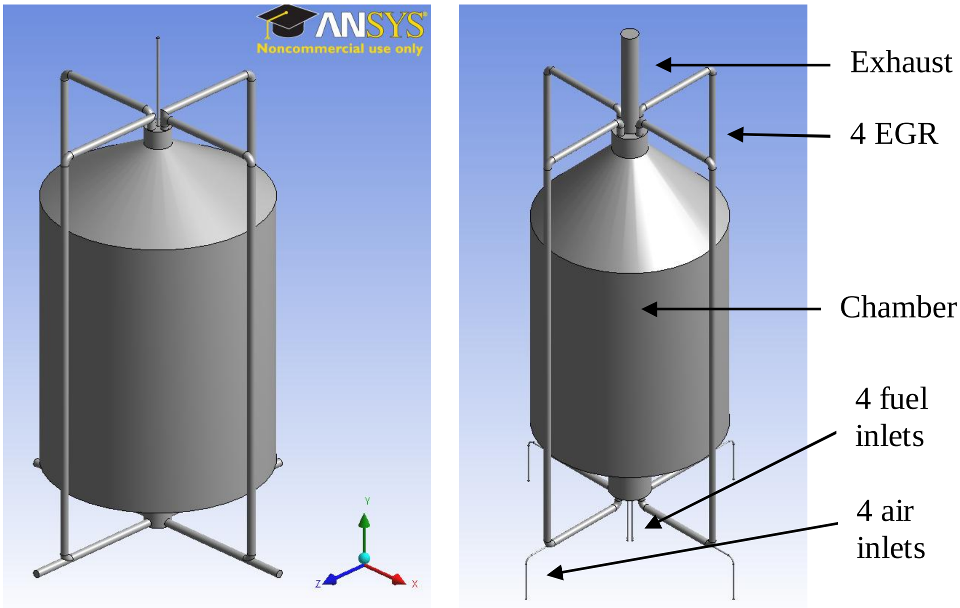


Figure 3. First combustion chamber model (a) No EGR (b) with 2 EGR pipe (c) with 2 EGR pipe and EGR inlet modified

Table 2. Typical data for furnace and burner in Figure 3(c)

Item	Data
Fuel	$0.5CH_4 + 0.2H_2 + 0.3CO_2$
Oxidiser	Atmospheric air, heated to 800 K
Fuel inlet	Round 1,256 mm^2, 40~50 m/s each
Air inlet	Annulus 5,140 mm^2, 80~100 m/s each
Chamber size	Diameter 375mm, Height 650mm
EGR	2 EGR with 386.9 mm^2 each inlet
Mesh method	Tetrahedrons (Patch conforming method) with 92,034 nodes and 421,172 elements
Radiation model	Discrete Ordinate (DO) model. Absorption coefficient: Weighted Sum of Gray Gas (WSGGM) model.

The next improvement was the addition of two more EGR pipes to become four. This new design can be seen in Figure 4. The gas entered the combustion chamber through four fuel inlet small pipes; each of them was 5 mm in diameter at 20 m/s to 40 m/s depending on the need for lean or rich combustion. The gas then flew through to the centre of the bluff body burner and jetted in through the bluff body burner with the annulus opening of 24.3 mm^2. Fresh air was injected at 80 to 100 m/s through 5 mm diameter at the side of each EGR pipe to induct the EGR to flow downward. The injected fresh air then mixed with the EGR gas. A more detailed burner and chamber specification is shown in Table 3.



(a) (b)

Figure 4. Final model with 4 EGR, (a) Air inlet internal diameter is 22 mm, (b) Air inlet internal diameter is 5 mm

Table 3. Typical data for furnace and burner in Figure 4(b)

Item	Data
Fuel	$0.5CH_4 + 0.2H_2 + 0.3CO_2$
Oxidiser	Atmospheric air, heated to 800 K
Fuel Inlet	4 x 19.6 mm^2, 20 m/s each
Air Inlet	4 x 19.6 mm^2, 80 m/s each
Chamber size	Diameter 600mm, Height 860mm
EGR	4 EGR with 386.9 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.

After the computational testing and analysis, the bluff body burner with 24.3 mm^2 fuel inlet and 97.3 mm^2 oxidiser inlets was chosen. The maximum temperature of the combustion is 1,540 K which is below the limit of the temperature for the rapid formation of NO_x. The temperature of the combustion zone is 1200 to 1540K (Figure 5). The inside wall temperature is about 750 to 800 K and the EGR flow is about 700 to 750 K. The flow for EGR is below 5 m/s since the opening for the chamber outlet is 176.6 mm^2. This opening is relatively big in size compared to the total inlet size of 486.8 mm^2. The ratio of inlet to outlet is 2.8. In order to increase this EGR flow, this opening can be reduced by increasing the EGR ratio. EGR ratio of 50% means half of the flue gas will flow back to the combustion chamber.

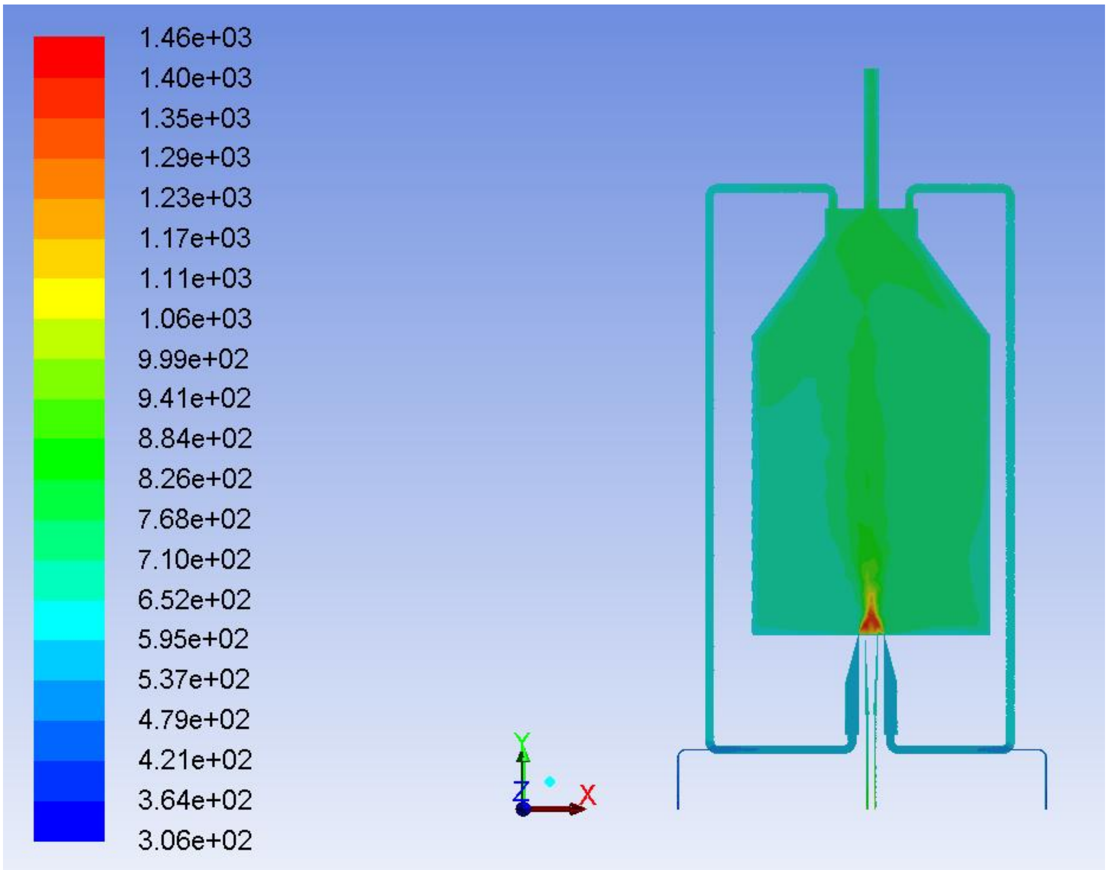


Figure 5. Combustion temperature in the chamber for Figure 4(b)

EGR flow is very important to achieve MILD combustion. Flue gas from the EGR will preheat the reactant and dilute the oxygen in the fresh air (Tsuji et al., 2003). The oxygen content in the oxidiser will be reduced from 21% to the required level. Figure 6 shows the velocity magnitude only showing the range of between -5.0 m/s to 5.0 m/s. The flow magnitudes are the highest at the centre of the chamber and dramatically reduce to very low speed far from the centre of the chamber.

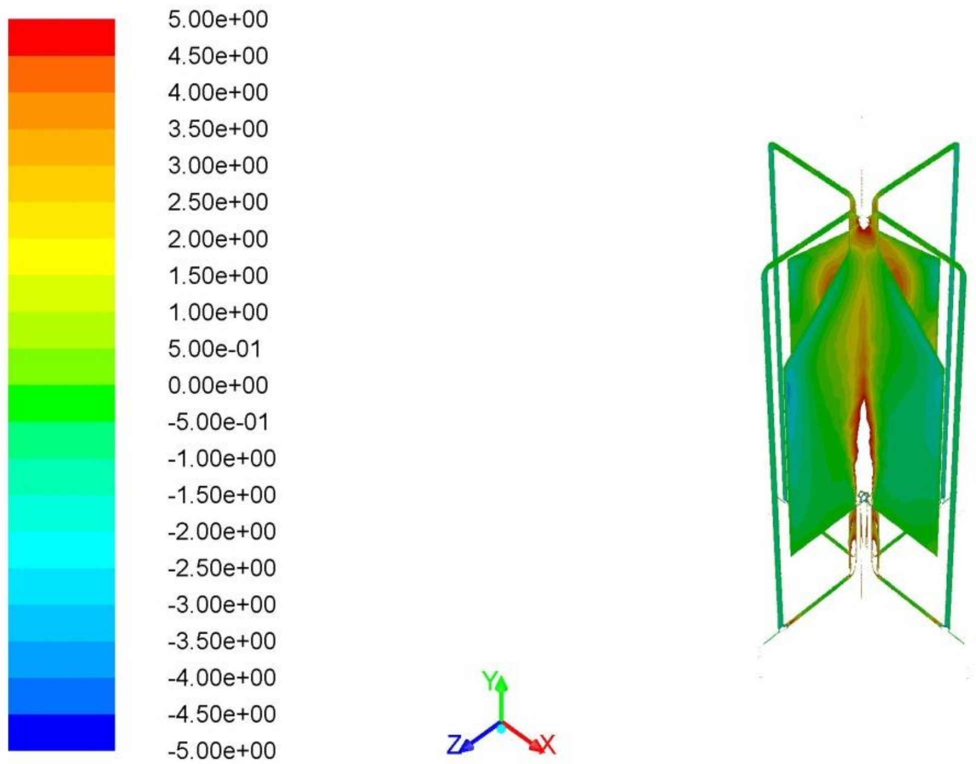


Figure 6. Velocity magnitude between -5.0 m/s to 5.0 m/s for figure 4(b) with gas inlet at 20 m/s and air inlet at 80 m/s.

There are four air inlets, one at each EGR. Every air inlet diameter is 5 mm with 100 m/s air velocity input. The total volume flow rate for the air is 7850 cm^3/s. If air is injected at 100 m/s and inlet diameter bigger than 5 mm, then EGR will flow in the reverse direction. The increase of inlet diameter or air velocity injected will increase the volume flow rate. If volume flow rate for the air inlet is higher than 7850 cm^3/s, EGR will flow upward instead of flowing downward. This is because the volume flow rate is higher than the allowed volume for the air nozzle at the combustion chamber. The air flow will reverse at the EGR pipe and this is an unwanted condition. Figure 7 shows the EGR flow in reverse direction. EGR should flow from top to bottom but the setting made the EGR flow in the reverse direction. Positive flow rate in Y direction means the flow is upward. Each EGR inlet is 379.9 mm^2 and the total for four EGR is 1519.8 mm^2. EGR should be flowing from top to bottom and mixed up with fresh air before combusting in the chamber. Figure 8 shows the correct flow of EGR. Negative flow rate in Y direction means the flow is downward. This flow can be achieved when the air inlet is control below 100 m/s for 5 mm air inlet.

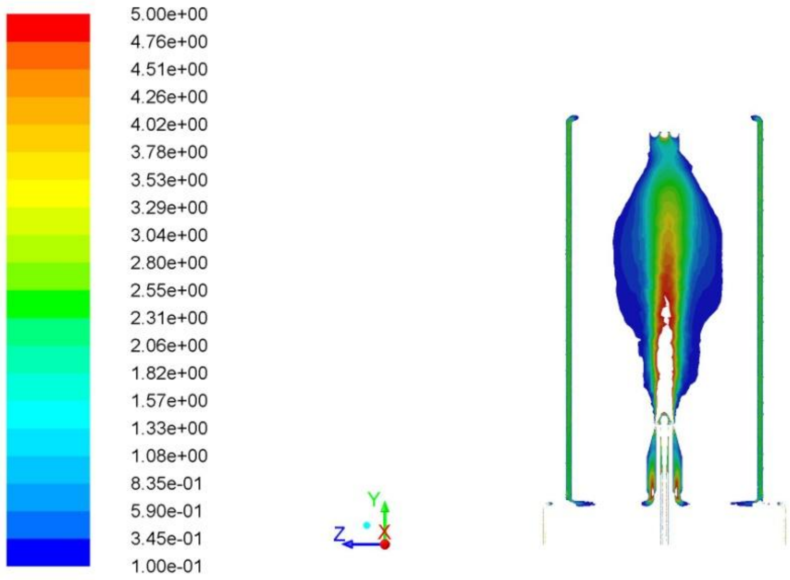


Figure 7. Velocity in Y direction between 5.0 m/s to 0.1 m/s for figure 4(b) with gas inlet at 40 m/s and air inlet at 120 m/s and EGR is reverse flow.

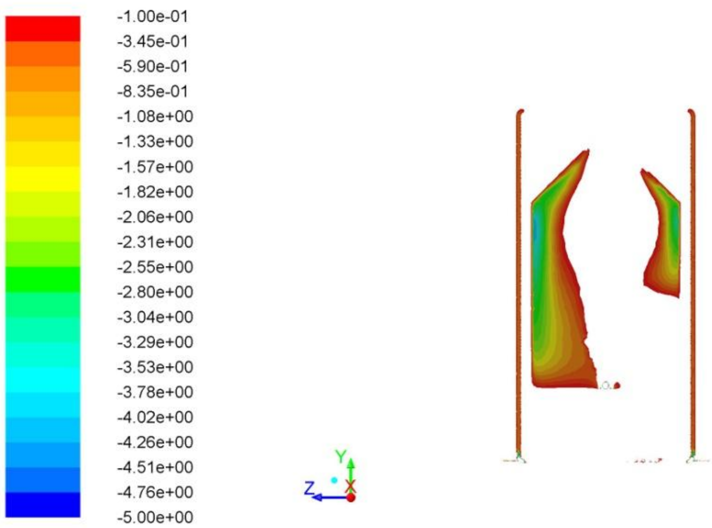


Figure 8. Velocity in Y direction between -0.1 m/s to -5.0 m/s for figure 4(b) with gas inlet at 20 m/s and air inlet at 80 m/s and EGR is correct flow.

CONCLUSION

The open furnace model was developed and numerically studied using FLUENT 13.0 to optimise parameters toward achieving MILD combustion in the chamber. The chamber design was finalised with four EGR pipes and the downward flow for flue gas was achieved. The LCV gas consisting of 50% methane, 20% hydrogen and 30% carbon dioxide by mass fractions was used in the study. The bluff body burner with 24.3 mm^2

fuel inlet and 97.3 mm^2 oxidiser inlet was chosen. From the CFD result, the temperature of the combustion zone is 1200 to 1500K and the temperature inside the wall chamber and EGR pipe flow is about 750 to 800 K. The flow for EGR is below 5 m/s since the opening for the chamber outlet is relatively big: 15 mm in diameter. This EGR flow rate can be increased by reducing the exhaust flow rate.

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NOMENCLATURE

CCS	Carbon capture and storage	HC	Hydrocarbon
CFD	Computational fluid dynamics	HTOC	High temperature combustion
CO	Carbon monoxide	IEA	International Energy Agency
CO_2	Carbon dioxide	LCV	Low calorific value
EGR	Exhaust gas recirculation	NO_x	Nitrogen oxides
FGR	Flue gas recirculation	OH	Hydroxyl
GHG	Greenhouse-gas	SO_x	Sulphur oxides
H_2O_2	Hydrogen peroxide	UHC	Unburned hydrocarbons