Performance of High Temperature Air Combustion Boiler with Low NO_x Emission*

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Thermal performance in the experiments and three-dimensional numerical simulations for a high temperature air combustion boiler where fuel can be efficiently combusted by high temperature preheated air (800°C–1 000°C) is examined. The boiler can burn not only natural gas but also low calorific gas (e.g. full gasification gas obtained from coal or wastes). In the boiler, four regenerative burners are installed. This boiler has new features that not only air but also gasification gas is heated up to 900°C, and combination of burners is switched every 15 seconds where two burners are used as inlets of fuel and air and the other two burners are used as outlets of exhaust gas. Natural gas and syngas obtained from coal are burned. The NO_x emission for each fuel is less than 50 ppm. The heat transfer of three-dimensional calculation is predicted higher than that of experiment.

Key Words: Boiler, Heat Transfer, Waste Heat Recovery, Burner, Diffusion Combustion

1. Introduction

A new concept boiler where fuel is combusted by highly preheated air (800°C–1 000°C)^{(1),(2)} was developed and tested in Tokyo Institute of Technology^{(3),(4)} and the experiment is successfully achieved. Uniform heat transfer to boiler tubes is realized, and NO_x emission is less than 50 ppm even for various combustion loads⁽⁵⁾. This boiler is also demonstrated for low calorific gas. With the high temperature air, even the low calorific gas can be combusted without any ignition or extinction problems. This boiler combusts not only various calorific gas but also gasification gas obtained from coal. The boiler is included in MEET I facility with gasifier for coal or wastes installed in

Tokyo Institute of Technology⁽⁶⁾. The scale-up facility, what we call, MEET II starts up. The large-scale MEET II boiler combusts full gasification gas from coal or wastes. The MEET II boiler has new features that the calorific value of the gasification gas is so low that not only air but also gasification gas is preheated to 900°C. The natural gas and syngas combustion are experimented in MEET II boiler.

In the present paper, the experiment and three-dimensional simulation for the MEET II boiler are examined.

2. Concept of High Temperature Air Combustion Boiler

Figure 1 shows the operating principle of high temperature air combustion boiler. The operation principle is very similar to the industrial furnaces with regenerative burners⁽⁷⁾. Fuel gas is combusted with high temperature preheated air and the thermal energy contained in generated combustion gas is delivered to water/steam in boiler tubes by mainly radiative heat transfer process. Then the combustion gas with the temperature of 1 200°C passes through a

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Fig. 1 Concept of the high temperature air combustion boiler

honeycomb bed to be cooled down to about 150°C and is exhausted to atmosphere. The combustion of each side (side A and side B) takes place alternatively. Switching period is about 10 - 30 seconds. The honeycomb bed is functioning as a heat sink and heat source during the switching of the combustion.

This boiler is expected to have following advantages.

- 1) Low NO_x combustion is done by in-furnace exhaust gas recirculation.
- 2) Heat transfer augmentation occurs due to flattening of temperature profile of combustion gas.
- 3) Hard ware downsizing is possible by heat transfer augmentation and elimination of convective heat transfer section.
- 4) High temperature/high pressure steam is easily obtainable because there is no pollutants such as acids or dust which attack boiler tubes.
- 5) Low calorific syngas can be easily combusted.
- 6) Decomposition of dioxin is promoted in the high temperature field, and resynthesis of dioxin is suppressed due to rapid cooling of combustion gas within a honeycomb heat storage bed.

3. Numerical Method

The governing equations to be solved in the calculation domain are the three-dimensional steady-state equations. The code "FLUENT 5" has been adopted to perform the numerical calculation. In this study, the flow field, the temperature field, the concentrations of the species of reactants and the products inside the boiler have been simulated. The turbulence was model by standard $k-\varepsilon$ model⁽⁸⁾ and the instantaneous full chemical equilibrium by using PDF is assumed for the 18 species (CH₄, C₂H₆, C₃H₈, C₄H₁₀, H₂, O₂, N₂, CO, CO₂, H₂O, H, CH, CH₂, CH₃, O, OH, C₂, C(S)). DO model⁽⁹⁾ is adopted for radiative heat transfer. The absorption coefficient is determined as weighted-sum-of-gray-gases modeled for H₂O and CO₂⁽¹⁰⁾. Governing equations are following.

Continuity equation :

$$\frac{\partial}{\partial x_i}(\rho \bar{u}_i) = 0$$

where ρ is the density, u_i is the velocity for x, y, z direction.

Navier-Stokes equation :

$$\frac{\partial}{\partial x_i}(\rho \bar{u}_i \bar{u}_j) = -\frac{\partial p}{\partial x_i} + \rho g_i + \frac{\partial}{\partial x_j} \left[\mu_e \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) - \left(\frac{2}{3} \mu_e \frac{\partial \bar{u}_l}{\partial x_l} \right) \right]$$

where p, g and μ denote the static pressure, the gravity acceleration and the molecular viscosity, respectively. The turbulent viscosity μ_t is defined by $0.09 \rho k^2/\varepsilon$, and the effective viscosity μ_e is given by $\mu + \mu_t$.

Enthalpy equation :

$$\frac{\partial}{\partial x_i}(\rho \bar{u}_i \bar{H}) = \frac{\partial}{\partial x_i} \left(\frac{\mu_t}{\sigma_h} \frac{\partial H}{\partial x_i} \right) + \left[\mu \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) - \left(\frac{2}{3} \mu \frac{\partial \bar{u}_l}{\partial x_l} \right) \right] \frac{\partial \bar{u}_i}{\partial x_j} + S_h$$

where $H, \sigma_h (=0.85)$ and S_h denote the total enthalpy, the energy Prandtl number and the Radiation source terms, respectively.

Turbulent kinetic energy equation :

$$\frac{\partial}{\partial x_i}(\rho \bar{u}_i k) = \frac{\partial}{\partial x_i} \left(\frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_i} \right) + \mu_t \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \frac{\partial \bar{u}_j}{\partial x_i} - \rho \varepsilon$$

where k and ε denote the turbulent kinetic energy and the turbulent kinetic energy dissipation rate, and σ_k is the turbulent Prandtl number for k which is given by $\sigma_k = 1.0$.

Turbulent dissipation rate equation :

$$\frac{\partial}{\partial x_i}(\rho \bar{u}_i \varepsilon) = \frac{\partial}{\partial x_i} \left(\frac{\mu_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_i} \right) + C_1 \frac{\varepsilon}{k} \mu_t \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \frac{\partial \bar{u}_j}{\partial x_i} - C_2 \rho \frac{\varepsilon^2}{k}$$

where σ_ε is the turbulent Prandtl number for ε . In this study, the model constant is given by $\sigma_\varepsilon = 1.3, C_1 = 1.44$ and $C_2 = 1.92$. In our calculation, recirculation flow is occurred in the boiler, but the recirculation is not so strong as swirl flow. It has been confirmed that the results using the Reynolds Stress Model is in good agreement with that using the standard $k-\varepsilon$ model⁽⁸⁾. Therefore, the coefficients of standard $k-\varepsilon$ model⁽⁸⁾ are adopted.

4. Experimental and Numerical Results

Figure 2 shows the schematic view of MEET II boiler used for the experiment. In the boiler, four regenerative burners are installed. Figure 3 shows the operation of switching of burners for a cycle. The combination of burners is switched every 15 seconds where two burners are used as inlets of fuel and air

Fig. 2 Schematic view of MEET II boiler

Fig. 3 Operation of switching of burners for a cycle temperature air combustion boiler

Fig. 4 Detail of regenerative burner

and the other two burners are used as outlets of exhaust gas. Figure 4 shows the detail of the regenerative burner. For start-up of the boiler, natural gas and air are combusted. Natural gas is introduced from the center of the burner, and air is introduced into the upper honeycomb in the figure. Then flue gas passes through the upper honeycomb installed in the

Fig. 5 Measurement points of the heat flux

Fig. 6 Heat fluxes for natural gas (left) and syngas (right) combustion in experiment

Fig. 7 Temperature measured in the boiler

opposite side. Figure 5 shows the measurement points of the heat flux. The heat fluxes transferred to the boiler tubes are measured at 6 points for a side, and are measured at 6 points on the opposite side in the same way. Thermo Gage 1 000 (water cooling type) by Vatel Co. Ltd. is used for measuring the heat flux. The temperature in the boiler is measured at two points labeled as 2 in Fig. 5.

Firstly, the experiment and the three-dimensional simulation fueled by natural gas are examined. The steam pressure and temperature are 0.7 MPa and 165°C, respectively. The flow rate of the pilot air, the primary air and natural gas are 60 m³/h, 490 m³/h, and

Fig. 8 CO and NO_x emissions for natural gas (left) and syngas (right) combustion in experiment

Fig. 9 Calculation region

33.5 m³/h. The heat fluxes and temperatures measured for natural gas and syngas combustion in experiment are shown in Figs. 6 and 7. The heat fluxes to the boiler tubes (60–80 kW/m²) are uniform except for one point. This is why one of heat flux sensor was trouble. The heat flux of a scale-up MEET II boiler is the almost same as that of MEET I about 70 kW/m². The average of heat flux for syngas combustion is a little higher than that in natural gas combustion, because the temperature in the boiler is high in syngas combustion as shown in Fig. 7. Figure 8 shows the CO and NO_x emissions for natural gas and syngas combustion in experiment. The averaged CO and NO_x emissions are 23 and 48 ppm for natural gas, and 178 and 39 ppm for syngas, respectively.

A schematic view of calculation region for MEET II boiler is shown in Fig. 9. Mole fractions of fuel components and calculation conditions are shown in Tables 1 and 2. The syngas is obtained by gasifying UP coal. The Ultimate analysis of UP coal is shown in Table 3. These conditions are same as the experi-

Table 1 Mole fractions of fuel

	Natural Gas	Syngas
CH ₄ [%]	88.2	0.13
C ₂ H ₆ [%]	5.2	0.0
C ₃ H ₈ [%]	5.1	0.0
C ₄ H ₁₀ [%]	1.5	0.0
H ₂ [%]	0.0	15.14
CO [%]	0.0	20.37
CO ₂ [%]	0.0	6.92
N ₂ [%]	0.0	57.44

Table 2 Calculation conditions

	NG Comb.	Syngas Comb.
Air Flow Rate [m ³ /h]	550	534
Air Temperature [°C]	1000	900
Fuel Flow Rate [m ³ /h]	33.5	304
Fuel Temperature [°C]	20	900
Thermal Input [kW]	386.4	384.7
Heating Value [MJ/m ³]	41.5	4.56
Boiler Tube Temp. [°C]	180	180

Table 3 Ultimate analysis of UP coal

C [wt %]	74.5
H [wt %]	5.4
S [wt %]	0.36
O [wt %]	9.25

ment. The temperature distribution of natural gas combustion for left side combustion is shown in Fig. 10. Maximum flame temperature is 1785°C, and the average temperature in the boiler is about 1058°C. The total heat transfer rate to the boiler tubes is 293 kW. The average heat flux to the boiler tubes is about 63 kW/m², and this result is lower than that of experiment. For the detail comparison, the heat fluxes to the boiler tubes obtained from calculation is shown in Fig. 11. The measured points are same as the experiment. In the experiment, the average heat flux is about 65 kW/m². The average heat flux obtained from calculation is almost same as that from experiment, although each heat flux in the experiment is scattered. The upper heat fluxes are higher than the lower ones for natural gas combustion because of buoyancy in Fig. 11. For syngas combustion the effect

Fig. 10 Temperature distribution for natural gas combustion

Fig. 11 Heat fluxes by calculation to the boiler tubes

Fig. 12 Temperature distributions for left side combustion (left) and diagonal combustion (right) on the symmetry plane parallel to boiler tubes

is small. Figure 12 shows the temperature distributions for syngas combustion. The temperature distribution for syngas combustion is more uniform than that for natural gas combustion. The radiative heat

Fig. 13 Path lines for left side combustion (left) and diagonal combustion (right) on the symmetry plane parallel to boiler tubes

transfer is about 86% of the total heat transfer to the boiler tubes. Figure 13 shows path lines for each case on the symmetry plane paralleled to boiler tubes. For left side combustion, recirculation of flue gas occurs at the center of boiler, mixing of fuel and air will be promoted. On the other hand, for diagonal combustion, a large recirculation in the boiler is produced. As a result, high recirculation of flue gas will be realized because MEET II boiler is combusted by combination of four steps. Total heat transfers and heat fluxes to boiler tubes of 300 kW, 62 kW/m² (left side combustion) and 305 kW, 62 kW/m² (diagonal combustion) are obtainable, respectively. About 85% of total heat transfer is covered with radiative heat transfer.

5. Conclusions

The experiment fueled by natural gas and syngas obtained from coal, and the three-dimensional simulation have been examined for the MEET II boiler. Several conclusions are drawn.

- (1) The experiment of natural gas and syngas combustion is successfully done. The NO_x emissions for both case are less than 50 ppm, and uniform heat flux from 60 to 80 kW/m² in the boiler is obtained.
- (2) For the syngas combustion by calculation, temperature distribution is uniform for both left side and diagonal combustion. The average heat flux to the boiler tubes in calculation is almost same as that in experiment, although each heat flux in the experiment is scattered.

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