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Reducing the recycle flue gas rate of an oxy-fuel utility power boiler

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HIGHLIGHTS

- Studied concepts to manage temperature at reduced recycle flue gas rate.
- Acceptable temperature could be achieved with 55% flue gas recycling.
- Used additional heat transfer surfaces, reduced inlet gas temperature.
- Also used novel firing strategies to manage furnace temperature.
- The heat transfer through the radiant areas is increased at reduced flue gas rate.
- The heat flux on the furnace walls is higher at reduced recycle flue gas rate.

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ABSTRACT

Oxy-fuel combustion is a technology for capturing CO_2 from coal fired power plants. One drawback of this technology is the need for a large quantity of recycled flue gas (RFG) to avoid excessively high temperatures inside the furnace. Instead of only using RFG to manage flue gas temperature, this paper presents and evaluates the concept of using additional heat transfer surfaces in the boiler furnace, reduced incoming gas temperature and combustion control technologies to manage the flue gas temperature in an oxy-fuel boiler with reduced RFG rate. A 1000 MW_e ultra-supercritical coal fired utility power boiler was modified using these concepts and studied using a computational fluid dynamics (CFD) model. The combustion, temperature, and heat transfer characteristics of the boiler were compared for three cases: (i) standard air combustion mode, (ii) conventional oxy-fuel combustion mode recycling 72% of the exhaust flue gas, and (iii) the novel oxy-fuel boiler concept recycling 55% of the exhaust flue gas. It is shown by the CFD results that the modified 1000 MW_e boiler could achieve an acceptable temperature level in its furnace while recycling 55% of total exhaust flue gas in spite of an increase in predicted temperature level. The predicted heat transfer through the radiant heat transfer areas of the modified boiler, including the furnace walls and platen super heater is significantly increased. Some heat transfer surfaces traditionally arranged in the convective heat transfer sections would need to be arranged inside furnace as radiant heat transfer surfaces for operation at oxy-fuel combustion mode with reduced RFG rate. The predicted heat flux on the furnace walls of this boiler is higher than that of a commercial air-fired boiler and of a conventional oxy-fuel boiler, although not higher than that of oil-fired utility power boilers.

1. Introduction

Oxy-fuel combustion is one of the promising technologies for CO_2 capture in the utility power industry. During oxy-fuel combustion, pulverized coal is burned in a mixture of nearly pure O_2 and the recycled flue gas (RFG, mainly CO_2), leading to an exhaust gas with very high CO_2 concentration, ready for storage after conditioning.

Burning coal in pure O_2 would result in an extreme temperature in a boiler furnace, which would cause tube failures of the furnace walls and serious slagging problems on the furnace walls when burning low ash fusion temperature coals. Burning coal in pure O_2 would also lead to an increase in furnace outlet temperature (FOT) at the entrance of the close-spaced convective heat transfer surfaces, which may cause excessive ash accumulations on the surfaces. Many researchers have been trying to achieve similar temperatures and heat transfer distributions in oxy-fuel combustion utility power boilers compared to commercial air combustion units [1–6], where the idea is to utilize the existing technologies and designs of conventional air combustion boilers and make the unit workable at both air combustion mode and oxy-fuel combustion

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mode. As a consequence, RFG rates from the exhaust gas stream for oxy-fuel combustion have been greater than 70%, in order to substitute for the nitrogen in air combustion conditions.

For existing boiler retrofit, a large amount of RFG is necessary to convert an existing air combustion unit to an oxy-fuel unit, workable in both air combustion and oxy-fuel combustion modes. However, it may not be necessary for a new oxy-fuel boiler designed to only operate in oxy-fuel combustion mode, since reducing RFG would lower not only the capital cost by the reduction of the size of the boiler and the flue gas recirculation fans, but also the operational cost by the reduced electricity consumption of the recirculation fans and reduced maintenance cost [7]. Reducing RFG ratio (total recycle flue gas mass rate divided by the total exhaust flue gas mass rate of a boiler or a test rig) or even entirely avoiding RFG may be a major target of new generations of oxy-fuel combustion utility power boilers.

Several authors have explored technologies to control flame temperature using less RFG in a combustion test rig. Among them, Becher et al. [7] showed that a reduction of the RFG ratio from 70% to 50% is possible in a combustion test rig while avoiding unacceptably high flame temperatures, when burning natural gas with pure oxygen using a non-stoichiometric burner (in which the oxygen provided to the burner is greater or less than the oxygen needed to exactly complete combustion of the fuel provided to the burner). Blume et al. [8] performed combustion tests burning natural gas and coal in the same facility and indicated that reducing the RFG ratio by staged combustion can maintain an acceptable temperature level in the furnace. Bohn et al. [9] burned lignite in the same combustion rig using two non-stoichiometric burners, and their results showed combustion performance comparable to an un-staged oxy-fuel combustion process using a large RFG ratio. These were useful proof-of-concept studies performed in a pilot-scale combustion rig. Few research papers can be found in the open literature examining the reduction of RFG in a full-size utility power boiler for oxy-fuel combustion.

In this paper, the concepts to moderate the furnace temperature are discussed, and these concepts are then integrated in a 1000 MW_e ultra super critical utility power boiler to manage the furnace temperature at reduced RFG ratio under oxy-fuel combustion conditions. The focus of this work is to evaluate the effectiveness of these methods on moderating the furnace temperature and to predict the heat transfer and temperature changes for the oxy-fuel combustion boiler with reduced RFG. This work is intended to provide design concepts for the development of next generation oxy-fuel boilers.

2. The reference boiler and the concepts to control furnace temperature

2.1. The reference boiler

To explore the technologies of moderating the gas temperature in boiler furnaces, a 1000 MW_e ultra super critical utility power boiler was selected as a reference boiler. As shown in Figs. 1A and 3A the boiler is equipped with 48 coal burners at 8 corners of the furnace, with 6 burners on each corner labeled as A/B/C/D/E/F. A burner (e.g., A in Fig. 1C) consists of two coal nozzles (e.g., A_L and A_C in Fig. 3A), 70% of the coal in the burner is discharged through concentrated coal nozzles (e.g., A_C) while the remaining 30% is discharged through a lean coal nozzle (e.g., A_L). The concentrated coal nozzles and lean coal nozzles are vertically arranged in such way that a concentrated coal nozzle would adjoin a concentrated coal nozzle (e.g., A_C adjoins B_C) and a lean coal nozzle would adjoin a lean coal nozzle (e.g., B_L adjoins C_L).

Together with the designed air distribution, this burner arrangement forms an alternating fuel rich region (oxygen lean) and fuel lean region (oxygen rich) in the vertical direction of the furnace, which can moderate furnace temperature without causing a significant delay in coal burnout. Also, over-fire-air (OFA) and separated-over-fire-air (SOFA) nozzles are used as air-staging nozzles through which part of the air/oxygen to complete combustion is supplied to the furnace.

The nozzles and burners of the 8 corners are designed to form two separate firing circles in the furnace as shown in Fig. 2. While Corners 1, 2, 7 and 8 create a clockwise swirling flow, Corners 3, 4, 5 and 6 create a counter clockwise swirling flow. The established swirling flow at the left and at the right of the furnace is relatively separate although there is not a division wall between the left and right sides of the furnace.

One reason to choose this boiler is its high efficiency, which would be also an objective of a next generation oxy-fuel boiler, the high efficiency partially compensating for the cost of using pure oxygen for combustion. Additionally, the combustion system in this boiler could be applied to control furnace temperature, since the alternating fuel-rich and fuel-lean regions along the furnace height are similar to the concept presented in the literature [8] to control temperature. Further, air staging nozzles used in this boiler provide an additional approach for temperature control. Finally, additional radiant heat transfer surfaces, which are essential for an oxy-fuel combustion boiler to operate at reduced RFG rate, can be arranged in the furnace by adding division walls inside the furnace to control furnace temperature without significantly impacting the flow characteristics so that there is no need to change boiler geometry.

2.2. The concepts and modifications to manage the furnace temperature

Assuming a boiler furnace volume as a volume shown in Fig. 4, the energy balance in the furnace volume can be described by

$$H_{in} + Q_f = H_s + H_{out} + H_{wall} \tag{1}$$

where H_{in} is the sensible heat of the incoming gas into the furnace volume, given by

$$H_{in} = (t_{in} - 25)\sum_i (m_i \cdot C_{pi}) \tag{2}$$

where t_{in} is the gas temperature entering the furnace volume (°C), m_i is mass flow rate of species i entering the furnace volume (kg/s), C_{pi} is average specific heat of species i from 25 °C to t_{in} (kJ/(kg °C)) and Q_f is the heat released inside the furnace volume (e.g., the heat released by combustion) (kW). H_s is energy transferred to the surrounding atmosphere (kW), H_{wall} is energy transferred to the heat transfer surfaces of the boiler furnace (e.g., furnace walls or platen super-heater surfaces) (kW) and H_{out} is the sensible heat of the gas exiting the furnace volume, given by

$$H_{out} = (t_{out} - 25)\sum_j (m_j \cdot C_{pj}) \tag{3}$$

where t_{out} is the gas temperature exiting the furnace volume (°C), m_j is the mass flow rate of species j out of the furnace volume (kg/s) and C_{pj} is average specific heat of species j from 25 °C to t_{out} (kJ/(kg °C)).

Eqs. (1)–(3) can be combined to find that

$$t_{out} = \varnothing \cdot (t_{in} - 25) + (Q_f - H_s - H_{wall})/\sum_j (m_j C_{pj}) + 25 \tag{4}$$

where $\varnothing = \sum_i (m_i C_{pi}) / \sum_j (m_j C_{pj})$.

When assuming no heat transfer to the surrounding atmosphere and no heat transfer to the heat transfer surfaces ($H_s = 0$ and $H_{wall} = 0$), the calculated gas temperature t_{out} is the adiabatic flame