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Temperature and emissions characteristics of a micro-mixing injection hydrogen-rich syngas flame diluted with N_2

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ABSTRACT

Micro-mixing injection combustion is an effective way to reduce the NO_x emissions of gas turbine, but it can also cause high temperature near the burner's nozzle. In this paper, micro-mixing injection combustion and N_2 dilution were combined to resolve this problem. The results indicate that this method not only reduces NO_x emissions effectively but also protects the nozzles by lowering the temperature of the flame and that near the nozzle. The larger the dilution amount, the lower the temperature and NO_x emissions could be. Under various experimental operating conditions, NO_x emissions could be reduced to less than 6 ppm (@15% O_2), which is lower than the emissions of most conventional gas turbine burners. According to the characteristics of temperature and NO_x emissions, it seems the Zeldovich route is not the control route for the flame. With most of the flames, the CO emission hardly changed and remained at a relatively low level within almost 6 ppm (@15% O_2).

Introduction

Integrated gasification combined cycle (IGCC) technology is a competitive clean coal power generation technology when carbon capture and storage is considered [1,2]. The syngas of the gas turbine for this technology mainly consists of H_2, CO and a small amount of CH_4, etc. [3]. Although the heat value of syngas is lower than that of natural gas, its combustion equilibrium temperature is higher due to its hydrogen-rich characteristics. Thus, problems such as flame flashback and excessive emissions of NO_x occur when hydrogen-rich syngas is combusted in a traditional gas turbine directly [4]. Therefore, it is very necessary to design and develop a new type of gas turbine for the low calorific value hydrogen-rich syngas.

H_2 is the main component in syngas. Due to its high laminar burning velocity and low lean flammability limit, hydrogen tends to shift the combustor operating conditions towards flashback regime [5]. For combustion security considerations, a diffusion flame may conveniently be used in IGCC hydrogen-rich syngas gas turbines. However, disadvantages of the diffusion flame include its high flame temperature and high NO_x emissions. Compared with a traditional natural gas turbine, because of the change of components and heat value, the mechanism of the reaction rate and combustion products are also affected [6]. For thermal NO_x, it is sensitive to temperature changes [7]. Work has been performed on lean premixed flames at gas turbine relevant conditions for higher calorific value syngas; NO_x measurement results showed that

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there is a linear dependence on the adiabatic flame temperature, hinting at the dominance of the temperature-dependent thermal NO_x pathway [8]. For the diffusion flame, Sahu et al. compared five chemical kinetic mechanisms to predict NO_x characteristics, and they found that the NNH and N_2O intermediate pathways were the major contributors to NO formation in all reaction mechanisms except GRI 3.0 [9].

The micro-mixing injection combustion method can reduce NO_x emissions and lower the combustion temperature by scattering the flame surface [10]. However, this combustion method shortens the flame, resulting in an increase in the burner wall temperature near the nozzle exit that may damage the nozzle [11,12]. Alternatively, to reduce NO_x emissions, the dilution technique has been studied widely. This method reduces pollutant emissions by lowering the flame temperature via lowering the concentration of the participate reactants [13–16]. In our study, the micro-mixing injection combustion and dilution methods were combined. Thus, the NO_x emissions will be reduced theoretically; at the same time, with the assistance of dilution, the influence of the flame on the nozzle exits will be weakened, the flame temperature and the wall temperature near the nozzles will be lower, and the nozzle will be protected from damage.

Usually, N_2, CO_2 or water vapor acts as the diluent [17–19]. In terms of reducing the NO_x emissions, water vapor and CO_2 dilution are more effective than N_2 dilution [20,21]. However, there are some disadvantages for water vapor dilution; for example, the presence of water vapor reduces overall efficiency because heating the water to steam consumes energy, the presence of water vapor may cause corrosion, etc. [22]. For CO_2 dilution, the major disadvantage with its application lies in the lack of a suitable CO_2 source with high pressure and high concentration in the whole IGCC system. Considering availability, the ideal dilution for IGCC may be N_2, which can be obtained from its air separation system easily. Moreover, high pressure N_2 can also be applied to increase the work capacity of the whole IGCC system.

In this paper, micro-mixing injection combustion and N_2 dilution were combined, and experiments were conducted to study the temperature and emissions characteristics of these flames.

Experiments

Experimental setup and micro-mixing injection burner

The experimental system is shown in Fig. 1, mainly consisting of a charging and distribution system, a micro-mixing fuel injection burner, a test and analysis system, and data acquisition and processing systems. A gaseous mixture of H_2 and CO with a certain ratio was used to simulate hydrogen-rich syngas. The syngas was mixed with the diluent N_2 and was sent to the micro-mixing fuel injection burner. In the burner, the gaseous mixture from the gas nozzles and the air from the air nozzles diffuse into each other, forming a diffusion flame after ignition.

The structure of the micro-mixing fuel injection burner is shown in Fig. 2 (a). An air chamber and a fuel chamber were designed at the bottom of the burner to ensure the uniform distribution of the air and fuel before they enter the combustion chamber. The burner nozzles, set up at the bottom of the combustion chamber, are arranged as a 5 × 5 matrix as shown in Fig. 2(b). Each suite of nozzles consists of a central fuel nozzle and eight peripheral air nozzles. There are 25 fuel nozzles and 200 air nozzles in the burner. To strengthen the mixture of fuel and air, the fuel nozzles were designed in vertical distribution and the air nozzles were distributed at a certain tangential angle. The specific parameters are shown in Ref. [23]. Each suite of nozzles conducts combustion independently. In the whole combustion chamber, the flame surfaces are dispersive and the local heat load is reduced. Thus, the flame temperature can be reduced and the NO_x emissions can be lowered by this method. In addition, observation windows made of transparent quartz glass are placed in the tripartite wall to observe the flame conveniently.

Flame measurement

In this paper, the Testo 350 XL flue gas analyzer was used to analyze the components and the temperature of the flue gas. The K-type fine sheathed thermocouple was used to measure the combustion chamber temperature. The S-type sheathed thermocouple was used to measure the flame temperature.

The flame temperature is an important thermodynamic parameter in combustion. It is related not only to the combustion efficiency and combustion stability but also to the formation and emission of pollutants. In the combustion chamber, the thermocouple is in a heat transfer equilibrium state, and the thermocouple with high temperature will conduct radiant heat to the ambient chamber walls with low temperature. This will lead the measured temperature to vary from the real flame temperature. It is necessary to eliminate the measurement error considering the heat transfer. The equilibrium equation is as follows [24]:

$$a(T_g - T) = \sigma\varepsilon(T^4 - T_w^4) \tag{1}$$

In equation (1):

a — the convective heat transfer coefficient between the flame and the thermocouple junction, $W/(m^2 \cdot K)$;

ε— the emissivity of the thermocouple junction;

σ — the blackbody radiation constant, 5.67×10^{-8} $W/(m^2 \cdot K^4)$;

T — the temperature of the thermocouple junction, °C;

T_g — the real temperature of the flame, °C;

T_w— the chamber wall temperature, °C.

In equation (1), the main error is caused by the convective heat transfer coefficient a. The following equation was used to calculate coefficient a.

$$N_u = 0.8R_e^{0.25} = \frac{ad_j}{\lambda} \tag{2}$$

$$R_e = \frac{wd_j}{\upsilon} \tag{3}$$

In these equations:

N_u — a dimensionless number, the ratio of convective to conductive heat transfer across the boundary;

Re — a dimensionless number, the ratio of inertial force and viscous force;