Effect of excess air coefficient on the combustion characteristics of a multi-stage dual swirl burner

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Abstract. This study proposed a multi-stage dual swirl burner for heating furnaces to achieve low nitrogen oxide (NOx) emissions based on swirling combustion and staging combustion technology. The effect of excess air coefficient on combustion characteristics and NOx emissions was studied by numerical simulation. The flow field, temperature field, oxygen concentration, and the NOx concentration distributions of the burner were discussed and analysed in detail. The simulated results showed that two recirculation zones for entrained flue gas formed in the reaction process by swirl blades. Meanwhile, increasing the excess air coefficient can improve the fuel/air mixing rate and diffusion effect in the combustion chamber, reducing the maximum combustion temperature and NOx emissions. Furthermore, when the excess air coefficient increased from 1.1 to 1.4, the maximum flame temperature was reduced from 1918.3 K to 1819.6 K, and the mole fraction of NOx at the outlet decreased from $1.67 \times 10^{-6}$ to $0.77 \times 10^{-6}$, which revealed the potential of the burner for the clean combustion.

1. Introduction

Industrial heating furnace is an essential piece of equipment in mechanical manufacturing industries, and they play a prominent role in the steel production process. However, various pollutants are also produced by them, especially nitrogen oxides (NOx) pollutants [1], which are harmful to the human body, environment, and ecosystem. Many provinces and cities have carried out a series of policies and standards for industrial air pollutants in China. For instance, Hebei issued the severe “Boiler Air Pollutant Emission Standards” (DB13/5161-2020) in 2020, requiring that the NOx emissions of all gas-fired boilers must be limited to 50 mg/Nm³ after June 1, 2021 [2]. In the face of stringent environmental protection requirements, it is essential to develop new combustion methods to improve combustion quality and achieve low-NOx emissions.

At present, various low-NOx combustion technologies are used to meet the emission standard, such as staging combustion [3], flue gas recirculation [4], MILD combustion [5], and swirl burners [6]. Swirl burners are widely used to stabilize the flame independently and improve the combustion performance [7]. In a swirling scheme, a high-speed swirled flow is usually generated by swirl blades or tangential jets injectors. Due to the entrainment of the swirled flow, the swirl burner usually can form flue gas recirculation zones during the combustion process [8], which can dilute the local oxygen concentration, thereby decreasing the maximum flame temperature and NOx emissions. Scholars have reported many experimental investigations and numerical simulations on low-NOx swirl burners in recent years. Zeng et al. [9] studied the influence of outer secondary-air vanms on the combustion and
NOx emissions characteristics of a 300 MWe utility boiler. The results showed that when the vane angle was 35°, the NOx concentration reached the lowest (420.2 ppm at 6%O₂). Li et al. [10] confirmed that the lower swirl air angles were better to reduce NOx emissions but usually lead to an increase in CO. Zhou et al. [11] comparatively studied the combustion characteristics of two swirl burners and found that NOx emissions were significantly reduced by about 39.8% when the optimized swirl burner replaced the prototype swirl burner. Amiri et al. [12] performed an experimental study on the effect of air swirler with different vane angles on the emission characteristics in a cylindrical furnace. They found that the air swirler with a vane angle of 60° could minimum the emission pollutants. Hosseini et al. [13] investigated a methane-air diffusion flame using simulations. The results showed that when the swirl number of the inlet air gradually from 0.0 to 0.6, the fuel-air mixing rate and the NOx reduction were indeed improved. Belal et al. [14] reported the effect of different swirl angles (40° and 35°) on the flame stability under the same swirl number. The results showed that the high swirl combustion (swirl angle of 35°) had better flame stability but produced a higher flame temperature, resulting in more significant NOx emissions.

According to the above pieces of literature, it turned out that using swirl burners is an effective strategy to enhance combustion performance. However, few studies have considered the influences of air swirling and fuel swirling on combustion characteristics so far, which directly affects the amount of NOx generated. Therefore, in this paper, based on the swirling combustion and staging combustion technology concepts, we proposed a multi-stage dual swirl burner for the heating furnace to improve combustion quality, which used blast furnace gas (BFG) and coke oven gas (COG) as its fuel gases. Then, a 3D calculation model was established. We adopted four representative excess air coefficient levels (1.1, 1.2, 1.3, and 1.4) to analyse the combustion effect in the current work comparatively. The flow field and temperature field were analysed and discussed. The oxygen concentration and NOx concentration distributions were also studied deeply. Hopefully, this research can provide a reference for the development of low-NOx burners for industrial heating furnaces.

2. Model and calculation method

2.1. Physical model

The combustion chamber of the dual swirl burner was simplified to a cylinder with an 1100 mm diameter × 3000 mm long dimension. The final physical model is shown in Fig. 1. The dual swirl burner located at the left end of the combustion chamber, and the exhaust gas outlet located at its right end. In order to stabilize the flames and form staging combustion, the fuel divided into four stages, and the air divided into three stages. The fuel and air were arranged alternately to achieve uniform gas mixing. Due to the high calorific value of COG, a small amount of COG was injected into the central pipe as the primary fuel to facilitate ignition. Only the BFG was remained and then be divided into three portions, as shown in Fig. 1. Therein, eight gas injection pipes were evenly distributed in the circumferential direction for the tertiary BFG injection.
Meanwhile, using the co-swirling combustion technology to adjust the swirling intensities of air and fuel. The internal coaxial swirling blades were designed in the secondary air channel with ten blades (a swirling angle of 30°). The external coaxial swirling blades designed in the secondary BFG channel with twelve blades (a swirling angle of 45°). The internal and external swirling blades contributed to forming a high turbulent combustion zone, which increased the amount of flue gas circulation near the nozzle reduced the maximum combustion temperature. The specific design parameters of the dual swirl burner are shown in Table 1.

Table 1. Design parameters of the dual swirl burner.

| Gas type     | Flow rate (m³/h) | Circulation area (×10⁻³ m²) | Diameter (mm) |
|--------------|------------------|----------------------------|---------------|
| COG          | 25               | 0.27                       | 18            |
| Primary air  | 112              | 1.25                       | 44            |
| Primary BFG  | 1000             | 11.12                      | 130           |
| Secondary air| 781              | 8.79                       | 166           |
| Secondary BFG| 3000             | 33.00                      | 264           |
| Tertiary air | 1578             | 17.50                      | 304           |
| Tertiary BFG | 2000             | 2.78                       | 60            |

2.2. Mathematical model

The numerical simulations were carried out by Fluent version 16.0. The continuity equation, momentum equation, energy conservation equation, realized k-ε turbulent equation, discrete ordinates (DO) radiation model, and finite-rate/eddy-dissipation combustion model were adopted in the calculation process. The thermal NOx and prompt NOx model were used to calculate the NOx generation rate. The main control equations used in the simulation are as follows:

(1) Continuity equation
\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{v}) = 0
\]

(2) Momentum equation
\[
\frac{\partial}{\partial t} (\rho \mathbf{v}) + \nabla \cdot (\rho \mathbf{v} \mathbf{v}) = -\nabla p + \nabla \cdot (\mathbf{F})
\]  \hfill (2)

(3) Energy conservation equation

\[
\frac{\partial}{\partial t} (\rho H) + \nabla \cdot (\rho \mathbf{v} H) = \nabla \cdot \left( k_{\text{eff}} \nabla H \right)
\]  \hfill (3)

(4) Turbulent equation (Realized k-\( \varepsilon \) two equation)

\[
\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_i} (\rho k u_i) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_s} \right) \frac{\partial k}{\partial x_j} \right] + G_k + G_\varepsilon - \rho \varepsilon
\]  \hfill (4)

\[
\frac{\partial}{\partial t} (\rho \varepsilon) + \frac{\partial}{\partial x_i} (\rho \varepsilon u_i) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_s} \right) \frac{\partial \varepsilon}{\partial x_j} \right] +
\]

\[
\rho C_v \varepsilon - \rho C_z \frac{\varepsilon^2}{k + \sqrt{\varepsilon}} + C_{\mu} \frac{\varepsilon}{k} C_{\varepsilon} G_\varepsilon
\]  \hfill (5)

(5) Radiation equation (Discrete Ordinates model)

\[
\nabla \cdot (I(r, \hat{s})\hat{s}) + \left( a + \sigma_t \right) I(r, \hat{s})
\]

\[
= an^{\sigma_t} \frac{\sigma_t}{4\pi} \left[ I \left( r, \hat{s} \right) \Phi \left( \hat{s}, \hat{s} \right) \right] d\Omega
\]  \hfill (6)

(6) NOx formation models (Thermal and Prompt NOx)

\[
\frac{d[NO]}{dt} = k_{f_1}[O][N_2] + k_{f_2}[N][O_2] + k_{f_3}[N][OH]
\]

\[
k_{r_1}[NO][N] - k_{r_2}[NO][O] - k_{r_3}[NO][H]
\]

\[
\frac{d[NO]}{dt} = k_{r_f} [O_2]^f [N_2][\text{FUEL}] e^{-k_{sr} \frac{r_f}{RT}}
\]  \hfill (7)

2.3. Mesh division and boundary conditions

The pre-processing software ICEM was used to generate the hybrid mesh. The combustion chamber was meshed using hexahedral structured grids, while the dual swirl burner was meshed using tetrahedral unstructured grids. The grid of the burner was locally encrypted because of its complex geometry. Fig. 2 shows the entire meshing grid with a local magnification. To verify the grid independence, the number of grids gradually increased from 800,000 to about 1,800,000, and the calculation showed that the outlet’s temperature and mass flow rate had no significant change. Finally, the total number of grid elements was 885,871, and the grid quality was all above 0.35, which ensured the reliability of the simulation and method.
The total heat load of the heating furnace was set to 5.64 MW. The boundary conditions for air and fuel were set to velocity inlet (25 m/s, 300 K), and the outlet was set to pressure outlet (-100 Pa). The non-slip adiabatic wall boundary conditions were adopted for furnace walls. The fuel consisted of COG and BFG, and their compositions and calorific values were shown in Table 2. Besides, the SIMPLE algorithm was selected to solve the flow field. The convergence criterion was that the residual errors of each equation were less than $10^{-5}$, and the gas temperature and mass flow at the outlet were unchanged.

Table 2. The composition and calorific value of used gases.

| Fuel | Composition (vol. %) | Calorific value (kJ/m³) |
|------|----------------------|-------------------------|
| COG  | 5.22% CO, 60.29% H₂, 21.02% CH₄, 1.31% C₂H₄, 0.37% C₂H₆, 1.88% CO₂, 0.84% O₂, 9.08% N₂ | 15703.81 |
| BFG  | 20.40% CO, 1.30% CH₄, 0.20% C₂H₄, 0.03% C₂H₆, 17.10% CO₂, 6.60% O₂, 49.70% N₂ | 3348.54 |

2.4. Model Verification

To verify the accuracy of the calculation models, the simulation results were compared with the experimental data of a dual swirl burner in the literature [15].
The literature took a rectangular area of 110 mm × 83.2 mm (radial × axial) as its research scope. The experiment results were compared with the simulation results, as shown in Fig. 3 and Fig. 4. Notably, the relative errors were less than 5%, which proved the accuracy of the calculation model.

3. Results and discussion

The combustion characteristics of gaseous fuels are affected by the excess air coefficient, which directly impacts combustion temperature and NOx emissions and determines whether it is energy-saving. Hence, in this section, the numerical solutions were performed by comparing the velocity field, temperature field, oxygen concentration, and NOx concentration distributions under four different excess air coefficients, which were 1.1, 1.2, 1.3, and 1.4.

3.1. Analysis of velocity field

Fig. 5 shows the velocity field distributions on the Z cross-sections (Z=-0.25 m, Z=-0.75 m, Z=-1.25 m and Z=-1.75 m) under different excess air coefficients.
Fig. 5. Velocity distributions on different Z planes.

It can be seen that the velocity fields of the four excess air coefficients in the same cross-section were similar in shapes, showing typical swirling flow field distributions. By adopting dual swirl coaxial blades, two low-velocity zones on the same cross-section were located in the centre of the combustion chamber and near the furnace wall. The swirling flow of oxidizer and fuel caused vortexes, forming two recirculation zones for entrained flue gas, and this was determined by the turbulence intensity of the rotating jet and the shear force at the backflow boundary. The internal recirculation zone was formed by the secondary air, while the entrainment of secondary BFG greatly caused the external recirculation zone. As the excess air coefficient increased, the average velocity of the cross-section also increased, increasing the area of the internal recirculation zone. However, these changes were very slight because the swirl intensity was affected by the angle of swirl blades and the radius ratio of the swirling channel. Moreover, when the excess air coefficient was constant, there was a significant radial expansion downstream of the flow field, and the range of high-velocity along the gas flow became larger, indicating that the fuel and air were mixed in the combustion chamber and gradually diffused to the outlet to form a diffusion flame.

Fig. 6. Axial velocity curves of the furnace’s centre line.

Fig. 6 shows a graph of the axial velocity along with the axial distance with a partial enlargement to more clearly highlight the difference in flow field under different working conditions. As shown in Fig. 6, the changing trend of the axis velocity of the furnace’s centre line under different operating
conditions was the same. When the excess air coefficient increased from 1.1 to 1.4, the average speed on the centre line increased from 6.6 m/s to 7.2 m/s. There were two low-speed valleys on the velocity curve, and it can infer that there were two vortexes along the axial direction. In the transition range from the first low-speed valley to the second low-speed valley, when the excess air coefficient was 1.4, the axial velocity reached its maximum value of 9.2 m/s at \( Z = -1.1 \) m. From the partial view, we found that increasing excess air coefficient would increase the local velocity, especially since the axial distance was in the range of -0.2 m to -1.8 m, which was conducive to the development of fuel/air mixing rate. After \( Z = -1.8 \) m, the smaller the excess air coefficient, the greater the axial velocity along the negative \( Z \) direction, which promoted the spread of flue gas. When the gas flow approached the exit of the combustion chamber, the axial velocity slowed down, resulting in a gradual flattening of the speed.

3.2. Analysis of \( O_2 \) concentration field

In the combustion process, only the oxygen in the air participates in the combustion reaction, and nitrogen as the diluent will absorb a large combustion heat, so the oxygen concentration directly influences the combustion quality. If oxygen is sufficient, it is beneficial to achieve complete combustion and low-NOx emissions. However, if the excess air coefficient is too large, the oxygen concentration increases simultaneously, causing an increase in flue gas and heat loss.

Fig. 7 shows the oxygen distributions on the \( Y = 0 \) plane. Due to the diffusion effect, the oxygen concentration decreased as the axial distance increased, indicating most \( O_2 \) was consumed near the nozzle. When the excess air coefficient increased from 1.1 to 1.4, the oxygen concentration in the combustion chamber increased. Additionally, as the excess air coefficient increased, the low-oxygen zone in the furnace’s centre gradually shrank, and the high-oxygen concentration zone moved closer to its centre from the furnace walls. This phenomenon was because of the swirling effect; the excess air diffused outward, rebounded after encountering the wall, and merged with the central gas flow to form a near-wall flow. On the one hand, due to the fuel contained a small amount of oxygen, the actual combustion degree was more thorough than the design state. On the other hand, the excessive air did not only participate in the combustion reaction but also took away some combustion heat, resulting in a decrease in maximum combustion temperature. Thus, it can predict that the NOx emissions would be relatively low.

![Fig. 7. Oxygen distributions on the Y=0 plane.](image)
3.3. Analysis of temperature field and NOx concentration field

Fig. 8 depicts the temperature distributions of the multi-stage dual swirl burner on the Y=0 plane. The temperature distribution shape was similar to the oxygen distribution, as illustrated in Fig. 7. There was a local conical high-temperature zone in the furnace, and it gradually shrunk with the increase of excess air coefficient. As mentioned above, due to the increase of air velocity, the expansion angle of flame spreading near the secondary air nozzle increased, and the flame spread radially in the radial direction to avoid the accumulation of fuel, thereby reducing the flame temperature. When the excess air coefficient was 1.1, the maximum flame temperature was 1918.3 K, and the average temperature at the outlet was 1780.5 K. As the excess air coefficient increased, the unburned low-temperature components gradually diffused into the furnace, exchanging heat with the high-temperature flue gas, which was good for reducing the combustion temperature and restraining the NOx formation. In the recirculation zone, the high-temperature flue gas mixed with the main gas flow and participated in the combustion reaction to help improve the flame stability. When the excess air coefficient increased to 1.4, the maximum flame temperature reduced to 1819.6 K, and the average temperature at the outlet dropped to 1664.7 K.

[Fig. 8. Temperature distributions on the Y=0 plane.]

Fig. 8. Temperature distributions on the Y=0 plane.

Reasonable control of excess air coefficient is an important measure to suppress the NOx formation, and the combustion temperature was the primary factor. Fig. 9 shows the NOx concentration distributions on the Y=0 plane. Due to the main component of the fuel was BFG, the combustion temperature was relatively low compared with full coke oven gas as the fuel. We can observe that the distribution of NOx concentration was approximately axisymmetric. With the increase of excess air coefficient, the NOx concentration distribution tended to homogenize, and its gradient reached the minimum of 10^-7. When the excess air coefficient increased from 1.1 to 1.4, the maximum mole fraction of NO was reduced significantly from 3.52×10^-5 to 1.89×10^-5. In addition, the NOx concentration at the outlet decreased linearly, from 1.67×10^-6 to 0.77×10^-6. The reason was that the increase in excess air coefficient increased the surface area of the flame and intensified the mixing of fuel and unburned air. Then, the excessive air diluted and reduced the flame temperature, causing a remarkable decrease in the thermal NOx. However, although the increase of excess air coefficient improved the temperature uniformity and NOx reduction, the cost of production also increased. A more significant excess air coefficient in terms of energy utilization is not an excellent choice. Therefore, it should control the excess air coefficient reasonably to reduce NOx emissions.
4. Conclusions

In summary, numerical simulations of a multi-stage dual swirl burner were carried out under different excess air coefficients. According to this research, the main conclusions obtained are as follows:

1. The difference in excess air coefficient in the multi-stage dual swirl burner had an important influence on its combustion performance. Due to swirl blades, two recirculation zones for entrained flue gas formed in the reaction process.

2. As the excess air coefficient increased, the gas flow velocity increased, especially in the front of the furnace, which was conducive to the development of fuel/air mixing rate and diffusion effect.

3. Within the scope of the research conditions, the excessive oxygen concentration in the combustion chamber significantly increased with the increase of excess air coefficient, which absorbed part of the combustion heat, thereby reducing the flame temperature.

4. When the excess air coefficient increased from 1.1 to 1.4, the maximum flame temperature reduced from 1918.3 K to 1819.6 K, diminishing the high-temperature zone. Correspondingly, the mole fraction of NOx at the outlet reduced from 1.67×10⁻⁶ to 0.77×10⁻⁶.

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