The Effects of the Size and Location of the Recirculation Windows of Combustion Chambers in a Coke Oven on Emissions of NO and Unburnt Fuels by CFD Numerical Simulation

Yinping Cao¹, Zhengang Du¹, Zhen Hu², Shuangling Jin², Wangzhao Lu² and Rui Zhang¹,*
¹Baoshan Iron&Steel Co., Ltd., 655 Fujin Road, Shanghai 201900, P. R. China
²School of Materials Sciences, Shanghai Institute of Technology, 100 Haiquan Road, Shanghai 201418, P. R. China

*Corresponding author email: zhangrui@sit.edu.cn

Abstract. NOx emission from combustion chambers of coke ovens is receiving renewed attention in recent years in China. Combustion process optimization and post combustion treatments are used to reduce NOx emission. In this paper, the effects of size and location of recirculation windows in the heating flues of combustion chambers in a coke oven from Baoshan Iron & Steel Co., Ltd. on emissions of NO and unburnt fuels were investigated by CFD numerical simulation, targeting for redesign of recirculation windows. Results indicate that with increasing the window size, the amount of recycled flue gases increases, maximum flame temperature decreases and temperature uniformity is improved, leading to a reduction of NO emission. The emission of unburnt fuels increases with the window size at higher heat flux, but this is less obvious at lower heat flux. With moving the windows towards inlets of fuels and air, the average temperatures increase slightly, maximum flame temperature and temperature uniformity keep unchanged, the emission of unburnt fuels decreases apparently, the NO emission increases slightly at lower heat flux, but keeps unchanged at higher heat flux. The unburnt fuel emission is dominated by the average temperature while NO emission by both the average temperature and temperature uniformity. The effect of the window size on NO emission is more apparent than that of the location of the windows while the latter is more obvious than the former to reduce the unburnt fuel emission.

Keywords: CFD; NOx emission; Flue gas; Combustion.

1. Introduction
Anthropogenic NOx emissions have negative impacts on environment and human health such as acid rain, stratospheric ozone depletion and particulate matter formation. NOx emitted from coke ovens in China accounts for around 7% of total, ranking the fourth largest NOx source. Ministry of Environmental Protection of the People’s Republic of China has stipulated in 2014 that NOx content in flue gas from coke ovens in cities has to be less than 150 mg/L, and 500 mg/L for old ovens and 200 mg/L for newly erected coke ovens in rural areas. Selective catalytic reduction is used to reduce NOx contents in flue gas from old coke ovens in cities, which has a high operation cost. For newly erected coke ovens, methods such as the staging combustion and external recirculation of combustion gases have been used to control NOx emissions. Although an internal flue gas circulation is applied in most of the coke ovens in steel industries such as the Baoshan Iron & Steel Co., Ltd. in China, it is not
as efficient as the external recirculation and the NOx content in flue gas in Baoshan Iron & Steel Co. Ltd. is still higher than 400 mg/L. The catalyst used in selective catalytic reduction with ammonia to convert NOx to N2 in most of the iron and steel companies in China has a carcinogenic V2O5 component, which pose a threat to human health. And ammonia with an unpleasant smell is emitted in this method, causing another environment problem.

In a coke oven battery, the thermal energy generated by burning gaseous fuels in combustion chambers heats up adjacent coking chambers, where the coal-to-coke conversion is completed. The temperature required for completing the coal-to-coke conversion in the coking chambers is over 1273 K. Therefore, the temperature in combustion chambers has to be higher than 1273 K so that suitable conditions for generating large amounts of harmful NOx are guaranteed. The formation of NOx in combustion chambers of coke ovens is complex, depending on temperature, concentration, flow fields of gas, unmixedness of fuel and air, NOx reburning and postflame reactions. Numerical simulations based on computational fluid dynamics is the most powerful and appropriate method to investigate heat and mass transfer, flow patterns and chemical reactions in combustion chambers, which can be traced back to as early as 1990s[1-3]. With the development of computation software and the improvement of computation power, the NOx formation in combustion chambers has been simulated in great details in recent years and much progress has been achieved in understanding the complex process and NOx formation mechanism, which is helpful for NOx emission control in combustion chambers.

Many efforts have been made to develop the models and methods in numerical simulation of coking and combustion chambers to describe and understand the complex coking and combustion process, some of which are focused on NOx emission control. Heat transfer occurs between coking chambers and combustion chambers. So the simulation strategy has two types, the coupled simulation of both combustion and coking chambers, and the decoupled simulation on combustion chamber alone. Weiss et al carried out a decoupled simulation on combustion chamber, focusing on NOx and CO emission control using heat flux into coking chamber as the boundary condition[4]. They found that the air excess and the geometry of gas supply channels under a nonpremixed combustion mode had an influence on the NO formation and that the double-stage combustion produced lower NO emissions than the single-stage combustion, due to the uniform temperature distribution for the former. A. Q. Zhang, et al simulated the thermal processes in combustion and coking chamber of a coke oven by two decoupled methods, a parallel decoupled method using the heat flux obtained by the energy balance method as the boundary condition for the heating walls, and a serial decoupled method with no need of heat flux of the heating walls as the boundary condition[5]. They found that both methods were in good agreement with the measured temperature evolution in coking chamber, but the parallel method was time-saving. Heat flux averaged in space and in one-hour periods in literature was defined as the unsteady boundary conditions for the heating walls of the combustion chamber by J. Smolka et al[6] to perform a decoupled simulation on temperature variation on the external flue wall of the combustion chamber in a coke oven using an accurate heating flue geometry in a real unit. These authors investigated further the coupled CFD simulation on the combustion and coking chambers[7], which reflected the variation of boundary condition with coking time and along the height of heating walls. Jin K, et al proposed a three dimensional transient model for coupled coking and combustion and focused on the influence of air-staging on the NOx emissions[8]. They found that the staging combustion played an important role in improving temperature uniformity and reducing NOx concentration of exhaust. The efficiency of the air-staging method for NOx emission reduction was also verified by B. Staiger in solid-fueled boilers[9]. Adams BR and Wang DH found that reburning leads to reduced final NOx emissions[10]. Hermann W, et al[3] carried out an investigation on the influence of entrance angle of air inlet on the flow patterns and NO formation in the combustion chamber of a coke oven with a three-staged air supply and found that an angle of 30° in the third stage intensified a good mixing behavior between fuel gas and air by vortex, and as a consequence, leading to a reduced formation of NO. Wessipe K, et al. indicated that a very uniform flow pattern and temperature distribution along the flue height was obtained by placing honeycomb bodies in heating flues, resulting in a decrease of NOx emission[11]. The effect of flue gas recirculation
was investigated by J. Baltasar et al. in a gas-fired lab furnace and concluded that a marked decrease of NOx emissions with this method without significant effects on flame stability, overall combustion efficiency and emissions of CO and unburnt hydrocarbons[12]. The similar conclusions were drawn by F. Qian et al., who found that flue gas recirculation allowed NOx to enter the combustor with the chance of being reduced and that flue gas contained more CO2 to suppress NOx formation[13]. Gamrat et al. investigated the influence of external flue gas recirculation on NOx formation in the combustion chamber of a coke oven[14]. They used averaged heating flux of 2000, 3000, 4000 and 5000 W/m² as the boundary conditions of the heating walls and found that the NOx concentration dropped from near 1000 to around 200 ppm under an external recirculation ratio from 0 to 0.3, indicating that the external recirculation was a crucial method to reduce NOx emissions in the coke industry. Compared with the external flue gas recirculation, however, the importance of the internal flue gas recirculation in combustion chambers of coking ovens was focused much less in literatures.

In this paper, internal flue gas recirculation in combustion chambers of coking ovens was investigated, focusing on emission reduction for both NO and unburnt fuels. For this purpose, a real coking oven in Baoshan Iron & Steel Co. Ltd. was used as the starting point, and the size and location of the recirculation windows were varied and their effects on emissions of NO and unburnt fuels were investigated by a decoupled simulation method using different heat flux averaged along height of heating walls as the boundary conditions.

2. Numerical Models and Simulation Method

2.1 Physical Model

In a coke oven battery, combustion chambers and coking chambers are alternately arrayed. The physical model includes a combustion chamber only in the decoupled simulation. As shown in Figure 1, there are a upward heating flue, where air and fuel are supplied, a downward heating flue, where the flue gas is emitted, a bridge window in the upper part, where the gases change their flow direction, and two symmetrical windows in the lower part, where internal recirculation of flue gas occurs in each combustion chamber. The gaseous fuels, air inlets and the flue gas outlet are located on the bottom walls. In each flue, two parallel vertical walls, noted as heating walls, are in contact with the two coke oven chambers while the other two, named as dividing walls, are connected to the neighboring flues. Gaseous fuels and air are preheated to 1373K through the regenerative chamber before entering the combustion chamber. The combustion inside the combustion chamber lasts for approximately 30 min. Then, the flow direction is reversed, i.e., the air inlet becomes the flue gas outlet and vice versa. The time required for the coal-to-coke conversion in the coking chamber in one coking cycle is approximately 20 h. So, the coking and combustion are unsteady. However, the combustion can be divided into a finite number of steady state process, so the combustion is in a steady state in a certain coking moment. During the coking cycle, heat is transferred from the combustion chamber to the coking chamber through the heating walls. The up wall, bottom walls and dividing walls are considered to be adiabatic while heat flux from the combustion chamber to the coking chamber occurs through the heating walls, which varies from charging coal (0 h) to discharging coke (20 h) and along the height direction of the heating walls.
2.2. Governing Equations

The general conservation equations of combustion in combustion chamber are described as follows:

2.2.1. Energy equation.

\[
\frac{\partial (\rho H)}{\partial t} + \frac{\partial (\rho u_j H)}{\partial x_j} = \frac{\partial}{\partial x_i} \left( \lambda \left( \frac{\partial H}{\partial x_i} + \frac{\partial H}{\partial x_i} \right) \right)
\]

(1)

Where \( \lambda \) is fluid thermal conductivity (\( W \cdot m^{-1} \cdot K^{-1} \)), \( C_p \) is fluid specific heat (\( J \cdot kg^{-1} \cdot K^{-1} \)) and \( H \) is fluid enthalpy (\( J \cdot kg^{-1} \)).

2.2.2. Momentum equation.

\[
\frac{\partial (\rho u_j)}{\partial t} + \frac{\partial (\rho u_j u_i)}{\partial x_i} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \mu \left( \frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j} \right) \right)
\]

(2)

Where \( u_j \) is fluid velocity in \( j \) direction (\( m \cdot s^{-1} \)), \( \mu \) is viscosity (Pa⋅s) and \( p \) is pressure (Pa).

2.2.3. Continuity equation.

\[
\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0
\]

(3)

Where \( \rho \) is fluid density (\( kg \cdot m^{-3} \)) and \( u_i \) is fluid velocity in \( i \) direction (\( m \cdot s^{-1} \)).

2.2.4. Two-equation \( k-\varepsilon \) realizable turbulence.

\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho u_j k)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \mu + \frac{\rho C_k k^2}{\varepsilon} \right) \frac{\partial k}{\partial x_j} + G_k + G_b - \rho \varepsilon - Y_M
\]

(4)

\[
\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho u_j \varepsilon)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \mu + \frac{\rho C_k k^2}{\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} + C_{1e} \frac{\varepsilon}{k} (G_k + C_{3e} G_b) - C_{2e} \rho \frac{\varepsilon^2}{k}
\]

(5)

where \( k \) is turbulence kinetic energy (\( m^2 \cdot s^{-1} \)), \( \varepsilon \) is turbulence kinetic energy rate of dissipation (\( m^2 \cdot s^{-1} \)), \( G_k \) represents the generation of turbulence kinetic energy due to the mean velocity gradient (kg ⋅ m\(^{-1} \cdot s^{-1} \)), \( G_b \) is the generation of turbulence kinetic energy due to buoyancy (kg ⋅ m\(^{-1} \cdot s^{-1} \)), \( Y_M \) represents the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate (kg ⋅ m\(^{-1} \cdot s^{-3} \)), \( C_{1e}, C_{2e}, C_{3e} \) are constants, \( C_{1e} = 1.44, C_{2e} = 1.92, C_{3e} = 0.09 \), and \( \sigma_k \) and \( \sigma_\varepsilon \) are the turbulent Prandtl numbers for \( k \) and \( \varepsilon \), \( \sigma_k = 1.0, \sigma_\varepsilon = 1.3 \).

2.3. Combustion Model

2.3.1. Reactions.

\[
CH_4 + 2O_2 = 2H_2O + CO
\]

\[
CO + 0.5O_2 = CO_2
\]

\[
H_2 + 0.5O_2 = H_2O
\]

\[
C_2H_4 + 2O_2 = 2H_2O + 2CO
\]

2.3.2. Equations.

\[
\frac{\partial (\rho f)}{\partial t} + \frac{\partial (\rho u_j f)}{\partial x_j} = \frac{\partial}{\partial x_i} \left( \frac{\rho C_k K^2}{\varepsilon \sigma_t} \frac{\partial f}{\partial x_i} \right) + \iint B p f_f f_{ox} \times \exp \left( -\frac{E}{RT} \right) p(f_f, f_{ox}, T) df df_{ox} dT
\]

(6)

Where \( E \) is the activation energy, \( f \) is mixture fraction (%), \( \sigma_t \) is constant. \( R \) is the Molar gas constant, the subscript \( f \) is the fuel, the subscript \( ox \) is the oxidizing agent.
2.4. Radiation Model

\[ q_r = -\frac{1}{3(\alpha + \sigma_s) - \sigma_s} \cdot \nabla G \]

\[ \nabla \left( \frac{1}{3(\alpha + \sigma_s) - \sigma_s} \cdot \nabla G \right) + aG + 4a\sigma T^4 = S_G \]

Where \( q_r \) is radiation flux (W), \( \alpha \) is absorption coefficient (m\(^{-1}\)), \( \sigma_s \) is scattering coefficient (m\(^{-1}\)), \( G \) is incident radiation, \( C \) is linear phase singularity phase function coefficient, \( \sigma \) is Stephen-Boltzmann Constant, \( W \cdot m^{-2} \cdot K^{-4} \) and \( S_G \) is radiation source term.

2.5. NO Formation Model (Thermal NO Model)

NOx formation reactions:

\[ O + N_2 \rightarrow N + NO \]
\[ N + O_2 \rightarrow O + NO \]
\[ N + OH \rightarrow H + NO \]

The thermal NOx mechanism assumes that the rate of formation and consumption of N atoms are equal, so that the NOx formation can be defined as:

\[ \frac{d[NO]}{dt} = 2k_{f,1}[O][N_2] \frac{(1 - k_{r,2}[NO])^2}{k_{r,1}[NO]} \frac{k_{f,2}^2}{(1 + k_{r,2}[NO] + k_{f,3}[OH])} \]

Where \( k_{f,1}, k_{f,2} \) and \( k_{f,3} \) are the rate constants for the forward reactions, and \( k_{r,1} \) and \( k_{r,2} \) are the rate constants for the reverse reactions. To solve this equation, the concentrations of \( O \) and \( OH \) must be determined. In this study, the \( O \) and \( OH \) radicals were modelled using a partial-equilibrium model.

2.6. Boundary Conditions and Parameters

2.6.1. Boundary conditions.
- Gas inlet: \( m_{gas} = m_0, T_{gas} = T_0 \);
- Air inlet: \( m_{air} = m_1, T_{air} = T_1 \);
- Flue gas outlet: \( P_{out} = P_0 \);
- The top and bottom surfaces as well as the dividing walls are assumed to be adiabatic. The heat flux through heating walls of the upward and downward flues is varied from 2000 to 5000 W/m\(^2\) with reference to the heat flux in real coke ovens.

2.6.2. Component of gases.
- The fuel gas is a mixture gas of coke oven gas (COG) and blast furnace gas (BFG) with a volume ratio of 9.39%, whose composition is listed in Table 1. The flow rates and inlet temperatures of air and fuel gas are tabulated in Table 2.

Table 1. Composition of fuel gases.

| Components | H\(_2\) | O\(_2\) | N\(_2\) | CH\(_4\) | CO | CO\(_2\) | C\(_2\)H\(_4\) |
|------------|--------|--------|--------|--------|----|--------|----------|
| Volume, %  | 8.44   | 0.09   | 46.11  | 2.22   | 21.07 | 21.84  | 0.26     |

Table 2. The inlet boundary condition of the combustion chamber.

| Compositions | Temperature (K) | Mass flow rate (kg.s\(^{-1}\)) | k (m\(^2\).s\(^{-2}\)) | \( \varepsilon \) (m\(^2\).s\(^{-3}\)) |
|--------------|----------------|-------------------------------|----------------|----------------|
| Air          | 1373           | 0.02214(\(\alpha = 1.1\))    | 0.38            | 0.39            |
| Fuel gas     | 1373           | 0.02155                       | 0.43            | 0.44            |

The above models were solved numerically with the FLUENT package. The spatial domain was divided into 300,000 finite elements with structured non-uniform grids in Cartesian coordinates. The
discretization of governing equations employed the central difference scheme for diffusion terms and the second-order difference scheme for advective terms. The SIMPLE series algorithm was applied to handle the pressure-velocity coupling. The calculation precision was less than $10^{-5}$ for relative error.

In this investigation, the size of the recirculation windows was varied by decreasing the $L$ value from 200 mm for the original windows in Baoshan Iron&Steel Co. Ltd. to 0 mm, where two windows are merged into one. The location of the windows was varied by decreasing the $H$ value from 150 mm (the initial location corresponds to 122 mm) to 0 mm, where the recirculating flue gas is in the same level as the inlets of air and fuels. The effects of the size and location of the recirculation windows on the average and maximum temperature of the heating flue, concentrations of NO and unburnt fuels were investigated under different heat flux from 2000 to 5000 W/m².

3. Results and Discussion

3.1. The Volume-averaged Temperatures and Maximum Flame Temperatures

Figure 2 shows the changing trends of the volume-averaged temperatures and maximum flame temperatures in the combustion chamber with the locations of the recirculation windows under different recirculation window sizes and heat flux. It is found that with the moving of the recirculation windows away from the bottom of the combustion chamber (increasing $H$ values from 0 to 150 mm), the volume-averaged temperatures decrease and level off for all window sizes ($L_0$, $L_{50}$, $L_{100}$, $L_{150}$ and $L_{200}$) under all heat flux from 2000 to 5000 W/m². The average temperatures decrease with increasing heat flux under the same window size and location as expected. The average temperature is lower for a smaller window size ($L_{200}$) than for a larger window size ($L_0$) when the $H$ value is 0 mm under all heat flux. But this trend is gradually reversed when the $H$ value is increasing to 150 mm under the higher heat flux of 4000 and 5000 W/m². With the moving of the windows towards the bottom of the combustion chamber, the mixing between fuel gases, air and flue gases becomes well, which increases the combustion temperature as the temperature of flue gas is higher than that of the fuel gases and air in the inlets. As a result, the combustion reactions become fast and the conversion of fuels to $\text{H}_2\text{O}$ and $\text{CO}_2$ becomes high, which account for the increased average temperatures with moving the windows towards the bottom of the combustion chamber. With increasing window sizes ($L$ decreases from 200 to 0 mm), the amount of flue gas recirculated increases. As a result, the combustion temperature increases, but fuel gases and air are diluted. Under well-mixed cases where the $H$ values are small, the former is dominated, and as a result the average temperature is higher for a larger window size. Under a high $H$ value of 150 mm where the mixing of flue gas, air and fuels is the worst, the latter is dominated gradually, this phenomena is more obvious at higher heat flux.

The changing trend of maximum flame temperature with the location ($H$ values) of the recirculation windows is not obvious, fluctuates and remains unchanged roughly under all heat flux and window sizes. The higher is the window size, the lower is the maximum flame temperature. This could be explained by the fact that at a larger window size, the amount of flue gases recirculated is higher, which causes a dilution effect for fuel gases and air, and as a result the combustion temperature is lower. The higher is the heat flux, the lower is the maximum flame temperature under the same window size as expected.
temperatures keep mostly unchanged with H values, as a result, the NO concentrations keep unchanged with H values for all window sizes under the highest heat flux of 5000 W/m². For NO formation rates are lowest, indicating NO formation rate is least sensitive to temperatures. As the combustion chamber temperatures are less uniform at smaller window sizes, which is favorable for NO formation. For the larger window sizes of L 0, L 50 and L100, the combustion chamber temperatures are more uniform, which is unfavorable for NO formation. As a compromise result, the NO concentrations keep unchanged for the cases of larger window sizes of L 0, L 50 and L100. At the highest heat flux of 5000 W/m², both the average and maximum flame temperatures are lowest, and NO formation rates are lowest, indicating NO formation rate is least sensitive to temperatures. As the variation of average temperatures with H values is quite small (less than 5 K) and the maximum flame temperatures keep mostly unchanged with H values, as a result, the NO concentrations keep unchanged with H values for all window sizes under the highest heat flux of 5000 W/m².

Figure 2. The volume-averaged temperatures (left column) and maximum flame temperatures (right column) of the combustion chamber with H values (locations of the recirculation windows) at different L values (the recirculation window sizes) under different heat flux. The changing trend is quite similar to that of maximum flame temperatures. As Figure 2 shown, the average temperature changes are smaller with H values and window sizes (within 5 K), but the maximum flame temperature changes are larger at different window sizes. The differences between the maximum and average temperatures, which reflect temperature uniformity, depend on the changing trend of maximum flame temperature with window sizes. This indicates that the temperature in the combustion chamber become uniformity with increasing window size under all window locations and heat flux. The differences between the maximum and average temperatures increase slightly with increasing heat flux under the same location and size of the windows, indicating that the lower is the heat flux, the more is the uniformity of the temperature in the combustion chamber.

3.2. The NO Concentrations of Flue Gases in the Outlet of Combustion Chamber

Figure 3 shows the differences between the maximum and average temperatures in the combustion chamber versus H values (locations of the recirculation windows) at different window sizes and heat flux. Figure 3 shows the differences between the maximum and average temperatures in the combustion chamber versus H values (locations of the recirculation windows) at different window sizes and heat flux. The changes between the maximum and average temperatures in the combustion chamber become uniformity with increasing window size under all window locations and heat flux. The differences between the maximum and average temperatures increase slightly with increasing heat flux under the same location and size of the windows, indicating that the lower is the heat flux, the more is the uniformity of the temperature in the combustion chamber.

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3.2. The NO Concentrations of Flue Gases in the Outlet of Combustion Chamber

Figure 4 shows the variations of the NO concentrations in the outlet of the combustion chamber with locations of the recirculation windows at different window sizes and heat flux. It is found that at the heat flux of 2000, 3000 and 4000 W/m² the NO concentrations decrease slightly with increasing H values at the smaller window sizes of L150 and L200, keep almost unchanged at the larger window sizes of L 0, L 50 and L100. At the highest heat flux of 5000 W/m² the NO concentrations keep almost unchanged with H values for all window sizes. With increasing H values, the average temperatures decrease, causing a decreasing of NO formation rate and therefore a decrease of the NO concentrations. This is true at heat flux of 2000, 3000 and 4000 W/m² for the smaller window sizes of L150 and L200 as the combustion chamber temperatures are less uniform at smaller window sizes, which is favorable for NO formation. For the larger window sizes of L 0, L 50 and L100, the combustion chamber temperatures are more uniform, which is unfavorable for NO formation. As a compromise result, the NO concentrations keep unchanged for the cases of larger window sizes of L 0, L 50 and L100. At the highest heat flux of 5000 W/m², both the average and maximum flame temperatures are lowest, and NO formation rates are lowest, indicating NO formation rate is least sensitive to temperatures. As the variation of average temperatures with H values is quite small (less than 5 K) and the maximum flame temperatures keep mostly unchanged with H values, as a result, the NO concentrations keep unchanged with H values for all window sizes under the highest heat flux of 5000 W/m².
The NO concentrations decrease with increasing heat flux at the same location of the recirculation windows for all window sizes. This is reasonable in that both the average and maximum flame temperatures decrease with increasing heat flux and therefore the NO formation rates follow the same changing trend. The NO concentrations decrease with increasing window size under the same location of the recirculation windows for all heat flux. With increasing window size, the amount of the recirculated flue gases increases, leading to a decrease of maximum flame temperatures and therefore the NO formation rates with increasing window size under all heat flux. Although the average temperatures are also affected by the window size, this effect is not dominant as the variation of average temperatures is quite small (within 5 K).

Figure 4. The NO concentrations in the outlet of the combustion chamber with H values (locations of the recirculation windows) at different L values (the recirculation window sizes) under different heat flux.

3.3. The Concentrations of Unburnt Hydrocarbons (CH₄, C₂H₄, Denoted as CₙHₘ in Figure 5), CO and H₂ of the Flue Gases in the Outlet of Combustion Chamber

Figure 5, 6 and 7 show, respectively, the concentrations of unburnt hydrocarbons, CO and H₂ in the outlet of the combustion chamber versus the locations of the recirculation windows at different window sizes and heat flux. It can be seen from Figure 5 that the concentrations of unburnt hydrocarbons increase and level off with increasing H values at all window sizes and heat flux, indicating that the increased mixing extent at low H values increases the completeness of hydrocarbon combustion. This is understandable in that the average temperatures decrease and level off with increasing H value at all window sizes and heat flux. As can be seen in Figure 3, the temperature uniformity increases with increasing window size. But the concentration of unburnt hydrocarbons does not as shown in Figure 5. The temperature uniformity has little effect on the combustion reactions of fuel gases. This indicates that the concentrations of unburnt hydrocarbons are dominated by the average temperatures. However, as discussed above, the NO concentration is dependent on both the average temperature and temperature uniformity. This could be accounted for by the fact that the activation energies for the formation of NO are higher than those for the combustion of hydrocarbons. The concentrations of unburnt hydrocarbons increase with increasing heat flux under the same window sizes and locations since the average temperatures are lower at higher heat flux under otherwise identical conditions. At a high heat flux of 5000 W/m², the concentrations of unburnt hydrocarbons increase with increasing window sizes at the same location of the windows (the same H value). With decreasing the heat flux, this changing trend becomes less obvious. This could be ascribed to the fact that at lower heat flux, the average temperatures are higher, and therefore its effect for the extent of combustion of hydrocarbons become less significant.

The changing trends of the concentrations of unburnt CO (Figure 6) and H₂ (Figure 7) with the locations of windows are quite the same as that of unburnt hydrocarbons, indicating that a dominant role of the average temperature for the combustion reactions of CH₄, C₂H₄, CO and H₂. The concentration of unburnt CO is highest, followed by H₂ and hydrocarbons under the same heat flux, window size and location, which has the same concentration order as the inlet fuel gases.
heat flux increase with the window size, but this trend is less obvious at lower heat flux. Moving the windows towards the bottom of the combustion chamber increases the average temperature, by which NO emission is alleviated. The emissions of fuel gases at higher temperature uniformity in the combustion chamber. The concentration of NO is determined both by the average temperature, maximum flame temperature and temperature uniformity in the combustion chamber. The emissions of NO and unburnt fuel gases (CH₄, C₂H₄, CO and H₂) are determined in turn by the average temperature and maximum temperatures. The emissions of NOx and unburnt fuels, the layouts of the recirculation windows should be redesigned.

4. Conclusion
Starting from the original size (L200) and location (H122) of the recirculation windows, we change their values to see how they affect the emissions of NO and fuel gases by CFD simulation with the FLUENT software package. It is found that both the size and location of the recirculation windows have effect on the volume-averaged temperature and maximum temperatures. The emissions of NO and unburnt fuel gases (CH₄, C₂H₄, CO and H₂) are determined in turn by the average temperature and temperature uniformity in the combustion chamber. The concentration of NO is determined both by the average temperature, maximum flame temperature and temperature uniformity in the combustion chamber. But the emissions of the unburnt fuel gases are mainly dependent on the average temperature. Increasing the window size leads to an increase of amount of flue gases recirculated and an improved temperature uniformity, by which NO emission is alleviated. The emissions of fuel gases at higher heat flux increase with the window size, but this trend is less obvious at lower heat flux. Moving the windows towards the bottom of the combustion chamber increases the average temperature, by which NO emission increases a little bit at lower heat flux, but remains unchanged at higher heat flux. The emissions of fuel gases decrease with moving the windows towards the bottom of the combustion chamber. To optimize the emissions of NOx and unburnt fuels, the layouts of the recirculation windows should be redesigned.

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