Aerodynamic Characteristics of a Stoker Furnace with Staged Combustion: Comparison of Cold Modeling Experiments and Numerical Simulations

Xinwei Guo, Hao Bai, Zhongxiao Zhang, Juan Yu, Degui Bi, and Zhixiang Zhu

ABSTRACT: A three-dimensional trial bed is established for a staged combustion boiler, and a modeling method based on similarity theory is proposed. The aerodynamic field of the 35 t/h layer combustion—composite combustion chamber—in the stoker boiler with staged combustion was evaluated. Further, a three-dimensional calculation model based on computational fluid dynamics (CFD) was used to simulate the aerodynamic field of the reformer furnace under normal operation, which facilitated convenience in the boiler design. Hot-wire anemometer and other instruments were used for the characteristic test of damper, a velocity field test in the furnace, wall wind test, temperature balance test at the outlet of the furnace, etc., and the law of motion for the flow field in the furnace was obtained. By analyzing the structure of staged combustion, the emission of nitrogen oxides and the combustion stability of a novel layer-fired boiler were studied. The calculated results are in excellent agreement with the experimental data. The results revealed that the combustion efficiency of the boiler and the reduction of nitrogen oxides were significantly improved by the staged combustion technology. There was no erosion on the water wall, and the flow velocity at the outlet of the furnace was uniform. This modeling method exhibits good adaptability to the combustion of stratified combustion boilers and is potentially useful for optimizing furnaces in a variety of applications.

1. INTRODUCTION

China is one of the biggest consumers of coal resources in the world. In 2017, the total energy consumption was 4.49 billion tons of standard coal, with coal accounting for the largest part of the energy structure. The furnace is essential equipment for transforming chemical energy into heat. The quality of combustion in the furnace not only influences the safety and economic efficiency of the unit but also determines the environmental impact of combustion-induced pollutants. The quality of combustion is primarily governed by the distribution pattern of the airflow field in the furnace. In several industrial furnaces, the proportion of stoker furnace is about 65%. During the operation of a conventional stoker furnace, coal particles move and combust simultaneously on the furnace grate. Coal particles pass through fire on one side and distinct regions are evident on the fuel layer along the direction of the length of the furnace of the furnace. The combustion efficiency is relatively low in view of the combustion reaction speed, space–time efficiency of combustion, and quality of combustion. Consequently, the thermal efficiency of industrial furnaces on the stoker furnace grate is not high and ranges between 60 and 80%.

The poor quality of combustion in a bed-fired boiler leads to poor coal adaptability, low combustion efficiency, and insufficient output in the boiler. To this end, Wang and Qu proposed a novel method to improve the combustion efficiency of chain boilers called the staged combustion method. The main characteristic of this method is that the swirl burners are arranged opposite to each other on both sides of the furnace wall, and two different combustion modes are realized in the furnace, which are laminar combustion and suspension combustion. The two combustion modes complement and promote each other, thus providing an efficient combustion mode for laminar combustion boilers. Zhao and Huang deduced the material balance and flue gas heat balance equations for a staged combustion chain boiler, thereby establishing the basic equation for furnace heat transfer. Du and Shen retrofitted a 2 t/h laminated-fired boiler by staged combustion. Because the aerodynamic field test of the system was not performed in the previous stage, the water wall was severely scoured after the coupled combustion of room-fired pulverized coal and large granular coal, and the flue gas temperature at the outlet of the furnace was unstable, which reduced the reliability.
of the boiler. The combustion characteristics, heat transfer process, and flow state of a boiler with staged combustion mode are complex, and the uniformity of heat load is higher than a traditional chain boiler. Experiments show that the boiler model for a cold aerodynamic field is similar to that for the hot aerodynamic field, which facilitates the understanding of the combustion characteristics of the hot state. However, this fact has never been examined through modeling or numerical simulation in the existing literature. In this study, a scale-down test system for a 35 t/h layer combustion boiler is established in a factory, which is reformed by staged combustion. The characteristics of airflow and the cold aerodynamic field in the furnace are evaluated. Numerical simulations are performed to validate the aerodynamic field test, and accordingly, a staged combustion layer furnace is explored. We believe that the systematic study provides a ready reference for the actual design and renovation engineering of boilers.

2. MODELING

2.1. Physical Model. We used an UG-35/5.3-M335t/h furnace in a plant as a prototype in the test bench. This furnace was a typical single-boiler and secondary high-pressure furnace. The length, width, and height of the furnace were 8.5, 4.52, and 13 m, respectively. The height of the front arch was 3.55 m (30% of the total length of the furnace grates) and the dip angle was 20°. The height of the rear arch was 1.078 m (50% of the total length of the furnace grates) and the dip angle was 10°. The coal seam was 100−150 mm high and the speed of the furnace grates was 2.5 cm/min. After stage combustion reconstruction, the furnace was scaled down with a ratio of 1:7 to construct an isothermal test system for similarity modeling of cold aerodynamic field. The entire test system was composed of a combustion chamber, an air supply system, and a combustor, as shown in Figure 1.

For easing the analysis, the cold test bench (Figure 2) was constructed by organic glass. Measurement holes were made on the sidewalls of the combustion chamber to measure test data, which were sealed by rubber plugs. The stage combustion technology was implemented in the furnace, which included block coal grate firing and coal powder firing in a turbulent burner. Turbulent burners were placed at the two sidewalls of the combustion chamber. This test device sent air into the combustion chamber through a draught fan. Primary air entered the combustion chamber from four air chambers at low regions,
and the secondary air entered from the turbulent burner. The primary air in the wind chamber mixed with the wind in the turbulent burner through the furnace grate and flew upward. It exited from the upper exit of the combustion chamber by the negative-pressure-induced draught system. The exit was kept at a negative pressure of 200–300 Pa, and the air entering the combustion chamber was adjusted by controlling the valves in each air duct.

### 2.2. Principle of Cold Modeling.

Cold modeling of the furnace simulates the airflow in the combustion chamber under cold test conditions and under a combustion-induced temperature rise. Cold modeling can provide qualitative laws for the furnace. Several Chinese and global studies have reported that under general conditions, the relationship among wind speed distribution in the field, pressure distribution, and operating conditions, which was obtained by the cold model after airflow entered the self-modeling regions, was basically applicable to hot equipment. This modeling technique is relatively simple and highly efficient. In this study, a universal cold modeling method is established for stage combustion technology in the stoker furnace. This model exhibited the following attributes. (1) The model structure maintained geometric similarity with the physical structure. (2) In both the model and the physical furnace, it was ensured that air entered the self-modeling regions. (3) The ratio of airflow momentum in the model was kept equal to that in the physical furnace.

Airflow in the combustion chamber can be viewed as the forced movement of viscous fluid. Some similarity criteria parameters such as the criterion number of dynamic uniformity \((H_0)\), Froude criterion number \((Fr)\), Prandtl constant \((Pr)\), Peclet number \((Pe)\), Euler criterion number \((Eu)\), and Reynolds number \((Re)\) can be deduced from the continuity equation, momentum equation, and the energy equation of fluid.

Under isothermal flow conditions, the airflow resistance is determined by \(Re\). Under general conditions, the ratio between pressure and inertia force is expressed by \(Eu\) in the flow process of the fluid. If the cross section is constant, the resistance coefficient of the airflow process is generally twice \(Eu\). When the airflow state enters the self-modeling region, \(Eu\) is unaffected by \(Re\) and remains constant. Under this circumstance, inertia force determines the airflow process and the effect of viscosity force can be ignored.

The aerodynamic test in the self-modeling region has many advantages. \(Re\) of the stoker furnace with staged combustion is very high. Therefore, wind speed distribution in the furnace is generally steady after the cold airflow enters the self-modeling region. This implies that the \(Re\) in the cold modeling test need not be equal to the actual \(Re\). In other words, the cold modeling test can be conducted under relatively low \(Re\), thereby facilitating a consistent flow field distribution. According to previous studies, \(Eu\) is nearly constant when \(Re \geq 4.5 \times 10^4\), indicating that the airflow in the furnace has entered the self-modeling region. The combustion model in the stoker furnace was staged combustion including suspension firing of coal powder in turbulent burners at the two sidewalls and the grate firing of coal blocks. Therefore, critical \(Re\) in the self-modeling region could be chosen as \(1 \times 10^5\). The scale ratio of the furnace model and the physical furnace was set to 1:7. The calculated results based on the above criteria parameters as well as the parameters of the model and the physical furnace and relative data are listed in Table 1.

#### Table 1. Calculated Results of the Model and the Physical Criterion Number

| no. | criterion | formula | boiler results | mode results |
|-----|-----------|---------|----------------|--------------|
| 1   | dynamic homogeneity number \((H_0)\) | \(\frac{\rho f_1 \omega_1}{\rho f_2 \omega_2} = H_0 = \text{const}\) | \(6.94 \times 10^4\) | \(1.38 \times 10^1\) |
| 2   | Froude number \((Fr)\) | \(\frac{f}{a} = Fr = \text{const}\) | 2.23 | 1.34 |
| 3   | Flow Prandtl number \((Pr)\) | \(\frac{\omega}{a} = Pr = \text{const}\) | 4.82E–01 | 6.38E–01 |
| 4   | Peclet number \((Pe)\) | \(\frac{\omega}{a} = Pec = \text{const}\) | 6.94E+04 | 5.29E+04 |
| 5   | Euler criterion number \((Eu)\) | \(\frac{f}{a} = Eu = \text{const}\) | 1.38E+01 | 1.38E+01 |

It is evident from Table 1 that \(H_0\), \(Fr\), \(Pr\), and \(Pe\) have a negligible effect on the uniform, forced airflow process. When the airflow enters the self-modeling region, the flow state and the speed distribution of the fluid remain constant and similar to each other. Moreover, \(Eu\) is kept constant. In other words, the cold modeling test can be conducted.

#### 2.3. Cold Modeling.

In the cold modeling process, one needs to ensure the entrance of airflow into the self-modeling region as well as the geometrical similarity and comparable boundary conditions between the model and physical furnace. The relationship of different airflows in the combustion chamber is a crucial factor to implement the cold modeling technique. Momentum is the primary airflow parameter in the combustion chamber. Therefore, momentum must be equal in the physical furnace and the model during the cold modeling test. Here, the subscripts O and M represent the physical furnace and the model, respectively. The subscripts 1, 2, 3, and 4 represent the primary wind (combustion air on the furnace grate), secondary wind (wind speed at the coal powder nozzle of the burner), reburning wind (secondary wind nozzle in the burner), and the over fire wind (secondary wind nozzle outside the burner), respectively. \(f\), \(\omega\), \(\rho\), and \(m\) are the nozzle area, mean flow speed, density, and sprayed mass rate, respectively. For example, \(m_p\) denotes the sprayed mass rate of coal powder. As the ratio of primary and secondary wind momentum is equal in the model and physical furnace, they can be rewritten as

\[
\frac{(m_{1M} \omega_{1M})}{(m_{2M} \omega_{2M})} = \frac{(m_{1O} + m_p) \omega_{1O}}{(m_{2O} \omega_{2O})}
\]

The airflow momentum of the primary wind in the physical furnace is composed of two parts

\[
(m_{1O} + m_p) \omega_{1O} = \left( \rho_{1O} f_{1O} \omega_{1O} \right) \left( 1 + \frac{m_p}{\rho_{1O} f_{1O} \omega_{1O}} \right) = \left( 1 + k \mu \right) \rho_{1O} f_{1O} \omega_{1O}^2
\]

where \(\mu\) is the mass concentration of block coals and fine particles on the furnace grate and \(k\) is a coefficient related to the flow rate of fine particles and wind speed, and it is approximately equal to 0.8. Therefore

\[
\frac{\rho_{1M} f_{1M} \omega_{1M}^2}{(\rho_{2M} f_{2M} \omega_{2M}^2)} = \left( 1 + k \mu \right) \rho_{1O} f_{1O} \omega_{1O}^2 / (\rho_{2O} f_{2O} \omega_{2O}^2)
\]

To maintain the equality of the momentum ratio between the physical furnace and the model, the ratio between the primary
wind speed and the secondary wind speed in the model is as follows

$$\frac{\omega_{2M}}{\omega_{2O}} = \frac{\omega_{2O}}{\omega_{2O}} \sqrt{\frac{\rho_{2M} \times \rho_{2O}(1 + \kappa \mu)}{\rho_{2O} \times \frac{f_{2M}}{f_{2O}} \times \frac{f_{1O}}{f_{1O}}} \times \frac{f_{2O}}{f_{2O}}}$$

Using similarity theory for wind nozzles $(f_{1O}/f_{2O} = f_{1O}/f_{2O})$, we have

$$\frac{\omega_{2M}}{\omega_{2O}} = \frac{\omega_{2O}}{\omega_{2O}} \sqrt{\frac{t_{2M} + 273}{t_{2O} + 273} \times \frac{t_{2O} + 273}{t_{1O} + 273} \times (1 + \kappa \mu)}$$

According to eq 5, the ratio of secondary and over fire wind speeds can be written as

$$\frac{\omega_{4M}}{\omega_{4O}} = \frac{\omega_{4O}}{\omega_{4O}} \sqrt{\frac{t_{4M} + 273}{t_{4M} + 273} \times \frac{t_{4O} + 273}{t_{4O} + 273}}$$

Generally, for the cold modeling test and the cold furnace test, $t_{1M} = t_{2M}$. Hence, eqs 5 and 6 can be further simplified. The absolute value of speed can be calculated from $Re = \omega d / \nu \geq 1 \times 10^5$ in the self-modeling region, which is used as the average speed in the cold modeling test.

$$\omega = Re \cdot \nu / d = \nu \times 1 \times 10^5 / d$$

Further, the necessary air volume is as follows

$$Q \geq 3600 \omega_{2M} \times \frac{273}{273 + t} = \text{Nm}^3 / h$$

where $t$ is the wind temperature in the cold modeling test and $F_M$ is the cross-section area above the combustion chamber of the cold model.

$Eu$ is equal in the model and the physical furnace. For example, the wind speed at the coal powder nozzle of the burner is

$$Eu_{2M} = Eu_{2O}$$

Expanding the above equation, we get

$$\frac{\Delta P_{2M}}{\rho_{2M} \omega_{2M}^2} = \frac{\Delta P_{2O}}{\rho_{2O} \omega_{2O}^2}$$

Therefore, the secondary wind speed can be determined as

$$\omega_{2M} = \omega_{2O} \sqrt{\frac{\rho_{2O} \Delta P_{2M}}{\rho_{2M} \Delta P_{2O}}}$$

The ratio of flowing resistance between the model and the physical furnace $(\Delta P_{2M}/\Delta P_{2O})$ can be randomly chosen as long as the airflow in the furnace is in the self-modeling region. To increase the measurement accuracy, it should be as large as possible according to the tolerance range of the fan capacity. For example, in the cold modeling test, it was calculated according to eq 7. Under the cold conditions, the wind speed in the combustion chamber was kept within 0.5 m/s. In other words, the wind speed must only meet 35% of the thermal air supply volume. Generally, to facilitate convenient observation and measurement in the cold modeling test, the fan delivery is adjusted to the rate of the fan. Finally, the wind speeds of different airflows in the cold modeling test were calculated according to eqs 5 and 6.

The calculated values of the minimum air supply volume in the self-modeling region, wind speed in the model, total air volume in the model, air volume distribution at storehouse and nozzles, and momentums in the model and the physical furnace are shown in Table 2.

### Table 2. Cold Modeling Parameters and Calculated Results

| no. | name                          | unit | boiler results | mode results |
|-----|-------------------------------|------|----------------|--------------|
| 1   | S.A. pulverized coal          | kg/kg| 0.6            | 0.6          |
| 2   | correction coefficient of coal and air velocity |       | 0.8            | 0.8          |
| 3   | cold/hot air pressure ratio   | m²/s | 0.6            | 0.6          |
| 4   | kinematic viscosity of air    | m²/s | 15.2E - 6      | 15.2E - 6    |
| 5   | Reynolds number in self-mode region | | 1E + 5           | 1E + 5       |
| 6   | furnace cross section length  | m    | 8.40           | 1.20         |
| 7   | furnace cross section width   | m    | 4.48           | 0.64         |
| 8   | furnace cross-sectional area  | m²   | 37.63          | 0.77         |
| 9   | equivalent diameter of furnace cross section | m    | 5.84           | 0.83         |
| 10  | primary air temperature       | °C   | 100.00         | 20.00        |
| 11  | secondary air temperature     | °C   | 120.00         | 20.00        |
| 12  | reburning air temperature     | °C   | 150.00         | 20.00        |
| 13  | over fire air temperature     | °C   | 220.00         | 20.00        |
| 14  | primary air velocity          | m/s  | 10.00          | 4.94         |
| 15  | secondary air rich-phase velocity | m/s | 25.00          | 12.69        |
| 16  | secondary air lean-phase velocity | m/s | 15.00          | 8.21         |
| 17  | rich-lean ratio               |      | 1.6            | 1.55         |
| 18  | reburning air velocity        | m/s  | 45.00          | 19.95        |
| 19  | over fire air velocity        | m/s  | 55.00          | 24.40        |
| 20  | total air volume              | m³/h | 81 648.60      | 4758.40      |

### 2.4. Numerical Simulations

The size ratio of the boiler to the model boiler was 7:1. To validate the accuracy of the results obtained using the cold model test, they were compared with numerically simulated results obtained using computational fluid dynamics (CFD).

The present study focuses on the coal seam surface of the chain grate stoker to the stoker outlet. Moreover, considering the complicated structural features of the grate boiler, tetrahedral unstructured tetrahedral meshes were adopted. The local meshes were used to refine the regions with large variations in the flow field. Further, meshes were refined at the lower stoker inlet, pulverized coal nozzle, and the stoker outlet. The boundary conditions for the numerical simulations are shown in Table 2. It indicates that velocity—inlet boundary conditions were adopted for lower stoker entrance with pressure of $-25 \text{ Pa}$, while pressure—outlet boundary conditions were used for the hearth exit.

The standard $K-\varepsilon$ model, which is a double-equation model, was used to numerically simulate the turbulent flow. Moreover, the EDM/FR model was employed for the turbulent gas phase, the two-competing-rates model for volatile gas, kinetics/diffusion-limited model for the turbulent gas phase, volatile combustion, and coke combustion, and P-1 model for radiative heat transfer. The SIMPLE algorithm was used to solve the coupled pressure and velocity equations. The discretization of pressure terms was conducted in accordance with the pressure staggering option (PRESTO) scheme, and all other terms were discretized using second-order upwind scheme. The convergence criteria were as follows: when the calculated residual errors for energy and radiation heat transfer terms were less than...
10^{-6}, and technological residual errors of other terms were less than 10^{-3}, it was assumed that the simulation converged to the solution.

2.5. NOx Reduction Principle. 2.5.1. Reduction of Overall Oxygen Concentration in the Furnace to Control NOx. Through the modification of the composite combustion technology, a large amount of pulverized coal was injected into the sidewall for suspension combustion, and the combustion speed was accelerated. Due to the increase of disturbance in the furnace, the mixture of fuel and air became more uniform, the excess air coefficient in the furnace decreased, the oxygen concentration at the outlet decreased, and the overall NOx emission was reduced.

2.5.2. Reducing Partial High Temperature to Control NOx in the Furnace. The injection of pulverized coal into the sidewall decreased the temperature of the main combustion zone of the grate with the highest initial temperature, and the high-temperature zone of the burned pulverized coal diffused evenly to the upper and middle parts of the furnace. This reaction mode decreased the local high temperature in the furnace and consequently reduced the production of thermodynamic NOx.

2.5.3. Control of NOx by Low Nitrogen Swirl Burner. The pulverized coal burner was injected into the furnace by a low nitrogen swirl burner to form a large recirculation zone. The low nitrogen burner is shown in Figure 11. The dense gas flow generated fuel-rich combustion inside the flame and inhibited combustion in the fuel-rich environment during the initial stage of pulverized coal combustion. A large amount of reductive volatile compounds was precipitated, and a large amount of NOx generated on the grate was reduced in a reductive atmosphere. The dilute gas flow on the outer region delayed the premature mixing of pulverized coal and oxygen, reduced the formation of NOx, and prevented coking.

2.6. Trial Data and Techniques. The test data mainly included flow characteristics in the aerodynamic field during stage combustion of the stoker furnace, which were as follows: (1) secondary wind speed at the nozzles of the wind storehouse, wind volume calibration, and leveling. (2) Under a constant wind volume, the speed of the secondary wind, reburning wind, and the over fire wind at nozzles of different opening degrees (100, 75, 50, 25, and 0%) was measured to obtain the damper characteristic curve. (3) The aerodynamic field in the furnace was analyzed. Wind speed in the furnace, closing-to-wall wind speed, and the wind speed in the horizontal flue before and after the stage combustion reconstruction were compared.

In this test, the wind speed was evaluated by hot-wire anemometer. Measurement holes were created on the walls of the combustion chamber, and measurement points were in the network distribution on cross section. Thirteen layers of measuring points were set from top to bottom along the Z-axis of the combustion chamber. Ten (five) rows of measuring points were set at an equal interval along the X-axis (Y-axis) of the combustion chamber. Measurement holes above the front and rear arcs were in cross distribution along the height direction. The speed direction was assessed by the band in the furnace.

The air volume needed on the test bench was calculated as 4758.4 m³/h by modeling. Delayed air distribution was applied to the downwind chamber. During practical operation, the combustion time of flame on the furnace grate was prolonged, thus reducing the local high temperature and production of nitric oxides. The air distribution in the cold modeling test of stage combustion is shown in Table 3.

### Table 3. Air Distribution in Stage Combustion in the Stoker Furnace

| Name          | Air Rate 1 | Air Rate 2 | Air Volume (m³/h) |
|---------------|------------|------------|-------------------|
| Overall       | 80%        | 5%         | 190.34            |
| Air Chamber 1 | 35%        | 45%        | 1332.35           |
| Air Chamber 2 | 15%        | 20%        | 571.01            |
| Air Chamber 3 |            |            | 1723.02           |
| Air Chamber 4 |            |            | 285.50            |
| Secondary Air |            |            | 190.34            |
| Reburning Air | 30%        | 50%        | 475.84            |
| Over Fire Air |            |            |                   |

3. RESULTS AND DISCUSSION

3.1. Characteristic Trial of Damper. The damper characteristics are analyzed under the situation when the wind speed varies with the damper opening. In this study, damper openings of the primary wind, secondary wind, reburning wind, and the over fire wind were examined. During measurement, the pressure at the exit of the combustion furnace was maintained between −200 and −300 Pa. The damper opening was set to 100, 75, 50, 25, and 0%. The wind speeds for different nozzles were measured by the hot-wire anemometer. At each measurement point, five measurements were conducted and the mean value was obtained. The damper characteristic curves obtained according to the corresponding relationship are shown in Figures 3–6.

![Figure 3. Damper characteristic curve at the lower storehouse.](https://dx.doi.org/10.1021/acsomega.9b03389)

The wind speed at the nozzles of different chambers in the primary wind storehouse is shown in Figure 3. The damper opening of each wind chamber under full load was adjusted. The wind speed in each chamber increased with the increase in the damper opening of the primary wind. Under a constant damper opening, the maximum wind speed deviation was 3.2 m/s.

Figure 4 shows that the two burners (thick-phase and thin-phase burners) of the secondary wind adjust the wind speed close to the desired value, i.e., the wind speed is in a reasonable range. At leveling, the speed of the thick-phase and thin-phase...
secondary winds at the nozzles on the two sidewalls was adjusted to about 12.62 and 8.21 m/s, respectively. The ratio between thick-phase and thin-phase wind speeds was reasonable and the error was controlled within 5%. Wind speeds at the two sides were relatively uniform, which was beneficial for the combustion in the furnace.\(^{46,47}\)

Figure 5 reveals that reburning wind speeds at the two sidewalls increase with the damper opening. The wind speed at the left sidewall is relatively higher when the damper opening is smaller than 75%, and it is higher than that on the right sidewall when the damper opening is larger than 100%. Meanwhile, it is slightly higher than the desired value. Under this circumstance, the recirculation zones at the two sides of the burners in the furnace may change differently. During practical operation, the wind speed at the two sides is adjusted to a reasonable error range.

It is evident in Figure 6 that the speeds of over fire wind at the two sidewalls are positively correlated with the damper opening, and the two speeds are nearly equal. During practical operation, the actual damper opening of over fire wind is adjusted according to the variation of unburned carbon in the flue dust. Meanwhile, the overall combustion in the stoker furnace is adjusted according to damper characteristic curves.

3.2. Measurement of Closing-to-Wall Wind Speed. Due to the strong mixing of rotating jets of turbulent burners and the coupling effect of grate and chamber firing, it is impossible to decide the direction of the closing-to-wall wind.\(^{48}\) Hence, the arithmetic mean of measured closing-to-wall wind speed in 10 s was selected without directional difference. The closing-to-wall wind speeds on the two adjacent upper and lower testing sections (CS-1 and CS-2) at the elevation height of burners on the cold modeling test bench were measured (Figure 1). Fifteen measuring points were set on each sidewall. The actual closing-to-wall wind speed at each measurement point after modeling is shown in Figure 9.

It can be seen from Figure 9 that the maximum wind speeds at the front, back, left, and right walls are 3.41, 3.0, 3.24, and 3.2 m/s, respectively. The closing-to-wall wind speeds for different layers are in the range of 1.5–3.5 m/s with small errors. Moreover, the airflow did not wash the sidewalls of the combustion chamber.

3.3. Measurement of Wind Speed in the Horizontal Flue at the Exit of the Combustion Chamber. Poor uniformity of the flue gas speed in the exit of the combustion chamber (horizontal flue) can easily lead to poor smoke temperature difference, resulting in the large stream temperature difference between two sidewalls, thereby affecting the safe and economic operation of the unit.\(^{49}\) Additionally, the extremely low flue gas speed in the horizontal flue can easily cause severe ash accumulation in the horizontal flue and the heated area at the tail, which is also a significant threat to the safe operation of the unit.\(^{50}\) Consequently, the flue gas speed at the exit cross section of the combustion chamber in the stoker furnace based on staged combustion was assessed in the cold modeling test. The flue gas speed in the upper and lower layers (25 measuring points in each layer) was measured at an interval of 120 mm to examine the uniformity of the airflow at the exit. The test results are shown in Figure 10.

Based on the calculations, the relative standard deviation of speeds between the upper and lower layers was 15.56 and 14.56%, respectively. The relative deviation of speeds between the upper and the lower layer in the horizontal flue was relatively large, especially in the right region of the furnace. Consequently, a dead angle could be formed to accumulate ashes during the actual operation of the furnace. It is suggested to strengthen soot blowing in this region during actual operation to prevent severe ash accumulation in the horizontal flue. Overall, the relative deviation of speeds between the upper and the lower layers in the horizontal flue was large, but the distribution of absolute wind speed was relatively uniform. As front arch and rear arch are present in common stoker furnaces, recirculation zones appear above these arches, accompanied with abundant eddies and complicated speed fields. A residual eddy still exists when the airflow in the furnace rises to the entrance of the furnace arch, resulting in the deviation of flue gas speeds at the left and right sidewalls in the horizontal flue. In the stoker furnace based on staged combustion, the deviation of flue gas temperature at...
the exit of the combustion chamber is eliminated to some extent by adjusting the proportion of reburning wind and over fire wind, which assures the uniform distribution of wind speed in the horizontal flue.

### 3.4. Numerical Simulation and Verification of Trial Results

Staged combustion of the stoker furnace involves two-phase (gas–solid) turbulent flow, gas-phase combustion, solid-phase combustion, radiation heat transfer, and other complicated processes. To simplify the investigation, the stoker furnace was divided into eight equal parts along the direction of stoker movement, and the component transportation model was used to calculate the average concentration, temperature, and velocity of gas components like O₂, CO, CH₄, NO, CO₂, and H₂ in the combustion-generated flue gas, which were actually measured on the bed seams corresponding to the eight parts, and these parameters were considered as the inlet boundary conditions at the lower side of the stoker. Standard $K-\varepsilon$ double-equation model was used to simulate turbulent flow. P-1 model was used to calculate the radiative heat transfer, and the discrete phase model (DPM) was used to simulate the combustion of pulverized coal particles. SIMPLE algorithm was used to solve the discrete equations for coupled pressure and velocity. The discretization of pressure terms was conducted using PRESTO scheme, and the remaining terms were discretized using the second-order upwind scheme. The convergence criteria were as follows: calculated residual errors of energy and radiation heat transfer items were smaller than $10^{-6}$, and technological residual errors of other items were smaller than $10^{-3}$.

The effect of stage combustion on the airspeed field was analyzed by measuring the wind speed field in the stoker furnace and the chain grate furnace. The variation of wind speed field in the furnace was observed by varying the positive rotation (along the movement of the furnace grate) and negative rotation (opposite to the movement of the furnace grate) of the turbulent

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**Figure 7.** Numerical simulation diagram. (a) Cross section flow field. Flow field profile of numerical simulation and velocity distribution diagram of burner layer section in grate-fired furnace via staged combustion; the nozzle spouting velocity of burner was 20 m/s. (b) Longitudinal section flow field. Flow field profile of numerical simulation and velocity distribution diagram of longitudinal section in grate-fired furnace via staged combustion; the nozzle spouting velocity of burner was 20 m/s.

**Figure 8.** Average velocity of grate-fired furnace via staged combustion along the altitude direction of the furnace.
burner. The wind speed at each layer of the combustion chamber was evaluated at the measurement points in the network distribution from bottom to top. The distribution of measurement points is shown in Figure 1. Accordingly, the mean of the wind speed at each layer was calculated. Furthermore, the modeling test stand was numerically simulated to compare and verify the data of the cold test stand. It is evident from the velocity distribution in the numerically simulated furnace in Figure 7a,b that the arrangement of different swirls in pulverized coal burner is basically similar to the overall trend of airflow in the furnace along the altitude direction of the furnace. Meanwhile, the airflow velocity was maximum near the throat under the above two situations. No obvious signs of disturbance were observed in the ordinary grate-fired furnace and on the central section of the furnace. However, the central section of the furnace exhibited distinct disturbance when the burner was in a positive-rotation arrangement. On the contrary, the central section of the furnace had a relatively weak disturbance in the negative-rotation arrangement. In conclusion, when the nozzle had different swirl directions, the disturbance in the furnace with positive rotation was higher than that with negative rotation at the same spouting velocity. Therefore, the disturbance in the furnace was reinforced, which was undoubtedly favorable for burning pulverized coal.

Figure 8 shows that the mean speed along the height of the stoker furnace increased initially near 300–400 mm, then dropped quickly at 400–600 mm, and finally increased steadily after 600 mm. This is because the height range of 300–400 mm was located between the front arch and rear arch of the combustion chamber. The wind speed section at the throat position decreased, while the wind speed value increased. Subsequently, a sharp reduction in the wind speed was observed as a collaborative effect of the sharp increase in the cross section of the rear arc, formation of a recirculation zone at the upper position, and the production of eddies due to the inconsistency in speed directions. The steady increase in the wind speed after 600 mm was caused by negative pressure in the combustion chamber.

In addition, the speed field in the common stoker furnace was consistent with that in the stoker furnace using stage combustion under positive and negative rotations of burners. The wind speed initially increased, then decreased, and finally increased slowly along the height of the combustion chamber. Under the three working conditions, the largest variation of wind speed was observed at the throat position. The wind speed at the throat position under positive rotation of the turbulent burner was higher than that under negative rotation. This can be explained as follows. As the wind speed is consistent with the direction of airflow movement under the positive rotation of burners in the entire combustion system, the wind speed at the throat position is the largest. As the wind speed is opposite to the direction of airflow movement under the negative rotation of burners, the wind speed is decreased. Nevertheless, the wind speeds under both positive and negative rotation of burners were higher than that in common stoker furnace because the added burners increased the disturbance of airflow. The results obtained through numerical simulations and experiments were basically consistent, and the model was comparatively better (see Figures 9–11).

4. CONCLUSIONS

A scaling-down test system of stage combustion reconstruction at the same proportion was established based on a 35 t/h stoker furnace in a plant, and a method for similarity modeling of staged combustion reconstruction in stoker furnace was proposed. Airflow characteristics and cold aerodynamic field in the furnace were examined. Besides, the flow field in the furnace was numerically simulated to validate the aerodynamic field test and
experimental results. The main results of the study are summarized as follows:

(1) In the cold state simulation test, the velocities of primary wind, reburning wind, and over fire wind were positively correlated with the damper opening. However, under the same dampering opening, the secondary wind speed of both walls exhibited a slight deviation. Therefore, the wind speed at both the wall nozzles should be adjusted according to the actual damper characteristic curve.

(2) After leveling the wind speed at the secondary air nozzle, the deviation of the wind speed decreased, and the thick—thin wind ratio was between 1.5 and 1.6, which is a reasonable range. A reduction atmosphere was formed on the side of the concentrated phase to reduce the concentration of nitrogen oxides, and oxidation atmosphere on the side of the dilute phase prevented the surface arc, back arc, and water wall coking.

(3) Under modeling conditions, the primary wind speed, secondary wind speed, reburning wind speed, and over fire wind speed were controlled to about 4.9, 12.6, 20.0, and 24.4 m/s, respectively. No wind speed deviation was detected at the nozzle, and the closing-to-wall wind speed was too low to wash the sidewall. The airflow speed in the horizontal flue was basically uniform, and staged combustion technology reduced the smoke temperature difference. Based on the above analysis, the aerodynamic field test based on cold similarity modeling of stage combustion furnace can serve as a ready reference for actual reconstruction engineering and design.

(4) Experiments showed that the NOx emissions were reduced if the reagent injectors were located near the corners above the secondary air nozzles, where the deep permeation of injection flow into the furnace. However, a recirculation zone was formed near the corners above the secondary air nozzles, where the injected reagent could be pushed forward by the recirculated flue gas to realize a good mixing with the flue gas.

(5) In the conventional combustion furnace and the proposed staged combustion furnace, the average wind speed increased initially, then decreased along the height, and finally increased gradually. The throat wind speed of the turbulent burner with positive rotation was higher than that with negative rotation in the staged combustion stoker. This is because when the turbulent burner was rotating, the mixture of fuel and air was more uniform, and the combustion and heat transfer were better.

**AUTHOR INFORMATION**

**Corresponding Author**
Zhongxiao Zhang — School of Environment and Architecture, University of Shanghai for Science and Technology, Shanghai 200093, China; orcid.org/0000-0003-2636-3312; Email: zhzhx222@126.com

**Authors**
Xinwei Guo — School of Environment and Architecture, University of Shanghai for Science and Technology, Shanghai 200093, China; orcid.org/0000-0002-3626-947X

**Hao Bai** — School of Environment and Architecture, University of Shanghai for Science and Technology, Shanghai 200093, China

**Juan Yu** — School of Mechanical Engineering, Shanghai Jiao Tong University, Shanghai 200240, China

**Degui Bi** — School of Environment and Architecture, University of Shanghai for Science and Technology, Shanghai 200093, China

**Zhixiang Zhu** — School of Mechanical Engineering, Shanghai Jiao Tong University, Shanghai 200240, China

Complete contact information is available at: https://pubs.acs.org/10.1021/acsomega.9b03389

**Notes**

The authors declare no competing financial interest.

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