Advances in Measurements and Simulation of Gas-Particle Flows and Coal Combustion in Burners/Combustors

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Abstract. Innovative coal combustors were developed, and measurement and simulation of gas-particle flows and coal combustion in such combustors were done in the Department of Engineering Mechanics, Tsinghua University. LDV/PDPA measurements are made to understand the behavior of turbulent gas-particle flows in coal combustors. Coal combustion test was done for the non-slagging cyclone coal combustor. The full two-fluid model developed by the present author was used to simulate turbulent gas-particle flows, coal combustion and NO\textsubscript{x} formation. It is found by measurements and simulation that the optimum design can give large-size recirculation zones for improving the combustion performance for all the combustors. The combustion test shows that the non-slagging coal combustor can burn 3-5mm coal particles with good combustion efficiency and low NO emission. Simulation in comparison with experiments indicates that the swirl number can significantly affect the NO\textsubscript{x} formation in the swirl coal combustor.

Keywords: PDPA Measurements, Simulation, Gas-Particle Flows, Coal Combustion

1. INTRODUCTION
Developing highly efficient and low pollutant coal burners/combustors remains to be important tasks of power engineering in China, since in many cases low-volatile and high-ash-content coals are used in power stations with serious problems of stabilizing coal flames at low load. In some cases coals are burned in small-size industrial and domestic furnaces with very low combustion efficiency and serious pollution, including high dust emission. Besides, low NO\textsubscript{x} burners should be developed for utility boiler furnaces in power stations. To solve these problems, systematic studies on developing coal combustors, measurement and simulation of gas-particle flows and coal combustion in such combustors were carried out by the present author’s group. Still in the years of 1983-1986, a coal combustor with high-velocity jets was developed for stabilizing anthracite and low-volatile coal flames, and measurements and simulation of gas-particle flows and simulation of coal combustion were done in optimization design of this kind of combustors. Then, in the years of 1989-1996 a non-slagging cyclone coal combustor was developed, and measurements and simulation of gas-particle flows were made for subsequent design of this kind of combustors and combustion experiments. Finally, in the years of 2000-2003, the development, measurements and simulation of gas-particle flows and coal combustion in a low-pollutant swirl coal combustor were conducted. The effect of fuel feeding location and swirl on NO\textsubscript{x} formation was studied.
2. DEVELOPED COAL BURNERS/COMBUSTORS

There are many types of low-grade coals produced in China, and in many power stations, high-moisture, high-ash-content low-volatile coals, even anthracite coals have to be used. For burning these types of coals the problem of flame stabilization at low load is crucial, frequently, it was obliged to burn oil for stabilizing the coal flame. To solve this problem, a coal burner/combustor with high-velocity jets, as shown in Fig.1, was developed (Zhou et al., 1994). Its principle is to open some tiny holes at the front end wall of a sudden-expansion chamber, from these holes high-velocity jets with velocities about 200~300m/s are injected into the combustor, while the coal-particle-laden main stream velocity is about 20m/s. Due to the entrainment effect of the jets a large-size recirculation zone connecting the corner part and the near-axis part is induced, in which the particle concentration is sufficiently high, so it is favourable to coal flame stabilization. The measurements and simulation are aimed to give optimal jet-hole location and flow-rate ratio.

![Fig.1 A coal Burner/Combustor with high-Velocity Jets](image)

In China, more than 20% of coals are burned in small-size industrial and domestic furnaces with very low combustion efficiency and serious pollution, including high dust emission. It is expected to achieve highly efficient and low-pollutant combustion in these furnaces. To solve these problems, a two-staged non-slagging cyclone combustor, organized in perpendicularly swirling flows, called “spouting-cyclone combustor” (SCC) was proposed (Zhou et al., 1992). The principle of SCC is shown in Fig.2. There is a two-staged combustion in two perpendicularly swirling flows. In the lower part of the combustor, called “spouting zone”, an opposed-jet system consists of one or a pair of primary-air jets laden with 3-5mm coal particles and one or a pair of bottom-air jets. The primary-air jet and bottom-air jet are in parallel and opposite directions. This opposite-jet system can create large-size recirculating flows around the horizontal axis as a primary combustion zone. The larger the particle sizes, the longer the residence time in which particles will multiply circulate in this zone. In the upper part of the combustor, called “cyclone zone”, one or two pairs of tangentially introduced secondary-air jets can create cyclonic flows around the vertical axis Here small particles or particles not yet burnout in the primary zone will complete their combustion. The particles collected on the wall can be led to return to the primary zone and burn again. The spouting zone is a fuel-rich zone for reducing the NO\textsubscript{x} formation. The secondary-air flow rate will keep the combustion temperature lower than the ash melting point, so as to maintain the coal combustion in a non-slagging mode. Subsequently it was found that the original shape of the spouting zone is not sufficiently good for particle recirculation. Then the shape of the spouting zone was changed from a cylinder to a rectangular cross-sectional channel (Wang et al., 1996), as shown in Fig.3, it is an improved spouting-cyclone combustor.
Furthermore, swirl coal burners are widely used in utility boiler furnaces of power stations in China. Its advantage is that it can stabilize the coal flame by itself. Recently, for designing such kind of combustors, not only the flame stabilization but also reducing pollution should be taken into account. It is expected that there is high coal concentration in the recirculation zone, but too high concentration will lead to reducing combustion efficiency. A typical swirl coal combustor was proposed (Li et al., 2000), as that shown in Fig.4. It includes a central tube, primary-air tubes for supplying coal-particle-laden flows, secondary-air tubes for supplying non-swirling and swirling flows. The measurements and simulation are aimed to study the effect of different flow rates and swirl angles of primary air and secondary air on the size of recirculation zone and coal concentration, in order to give good flame stabilization and simultaneous low NOx formation.

3. EXPERIMENTAL SET-UP & MEASUREMENT TECHNIQUE FOR ISOTHERMAL GAS-PARTICLE FLOWS
The experimental set-up for measuring isothermal gas-particle flows in Case 1 is shown in Fig.5. It consists of the test section 15, an air-supply system (compressor 9 and blower 12), a powder-supply system 10 and 11, and a LDV measurement system 1, 2, 3. The measurement instrumentation is a DISA-55X 2-D LDV system with forward scattering. The test section is given in Fig.1, where L=400mm, D=100mm, d=24mm, Dj=67mm and dj=1mm. The inlet flow parameters are: \(u_0 = 15\text{m/s}\); \(u_j = 260\text{m/s}\); \(G_p = 60\text{g/min}\); \(d_p = 60\mu\text{m}\).
The experimental set-up for measuring isothermal gas-particle flows in Case 2 is shown in Fig. 6. It consists of blowers 1, pipe lines and valves 2, flow meters 3, air compressor 4, secondary-air pipe 5, atomizer 6, powder feeder 7, spouting zone 8, cyclone zone 9, laser 10, transmitter 11, receiver 12, bottom-air pipe 13 and primary-air pipe 14. A TSI 2-D LDV system was used to measure the gas and particle velocities separately. A TSI atomizer was used to produce 1-5 μm liquid droplets tracking the gas flow and an electro-magnetically vibrating powder feeder was employed to supply 250-300 μm glass beads as the particle phase. The cylindrical test section (Fig. 2) of spouting zone and cyclone zone are made of plexiglass. There are one or two downward inclined primary-air inlets and one or two upward inclined bottom-air inlets in the spouting zone, and four tangential secondary-air inlets at two different heights of the cyclone zone. The test section was mounted on a traversing system for flow measurements at different cross sections.

The experimental set-up for measuring isothermal gas-particle flows in Case 3 is shown in Fig. 7. It consists of air compressors 18, 19, a Powder feeder 15, a test section 12, a PDPA system 1-11, and a cyclone separator 14. The test section is given in Fig. 4. A 3-D PDPA made by Dantec Inc was used to measure gas-particle flows, and a backward scattering arrangement was used. The particle velocity is measured based on the Doppler frequency shift. The particle size and concentration are measured based on the phase difference caused by Mie scattering and the number of particles passing the measuring volume during a certain time interval.
The test section is a cold model of swirl burner (Fig.4) together with a cylindrical furnace of 800mm diameter and 1500mm height. The measurement system is a DANTEC 3-D phase Doppler particle anemometer (PDPA). Glass beads of 0–10μm are taken as the gas tracer and glass beads of 50μm represent the particle phase. The inlet flow parameters are shown in Tab. 1. For each case there are 5 measurement cross sections along the axial direction and for each cross section there are 21 measurement points along the radial direction. At each measurement point the data rate is 500 per minute and each time of sampling lasts 2 minutes.

![Fig.7 Experimental Set-Up for Case 3](image)

Table 1 Flow Parameters for Case 3

| Burner  | Case | Primary Air | Swirler Angle | Q (m³/h) | q₁ (%) | q₂₁ (%) | q₂₂ (%) |
|---------|------|-------------|---------------|----------|--------|---------|---------|
| Type I  | 3-1  | No Coal Concentrator | 45°          | 1179     | 20.1   | 54.3    | 25.6    |
| Type II | 3-2  | With Coal Concentrator | 45°          | 1174     | 19.2   | 53.3    | 27.5    |

where Q——total air flow rate, q₁——primary air ratio, q₂₁——swirling secondary-air ratio, q₂₂——non-swirling secondary air ratio, air temperature 291K.

4. NUMERICAL MODELS AND METHODS FOR SIMULATION

For simulating turbulent gas-particle flows and coal combustion, a full two-fluid model was proposed (Zhou et al., 2000). The time-averaged equations of turbulent reacting gas-particle flows can be written as:

\[
\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}(\rho v_j) = \sum_p n_p m_p = S
\]  

(1)

\[
\frac{\partial}{\partial t}(\rho v_i) + \frac{\partial}{\partial x_j}(\rho v_j v_i) = \frac{\partial p}{\partial x_i} + \mu \left( \frac{\partial v_i}{\partial x_i} + \frac{\partial v_i}{\partial x_j} \right) + \rho g_i + \sum_p \frac{m_p}{\tau_r} \left[ \bar{n_p} (v_{pi} - v_i) + \bar{v_p} v_{pi} \right] + v_i S + F_{mi} - \frac{\partial}{\partial x_j}(\rho v_j v_i)
\]  

(2)

\[
\frac{\partial}{\partial t}(\rho h) + \frac{\partial}{\partial x_j}(\rho v_j h) = \frac{\partial}{\partial x_j} \left( \rho \lambda \frac{\partial T}{\partial x_j} \right) q_r + \sum_p n_p Q_p - \sum_k n_p m_p C_{pp} T_p + W_s Q_s - \frac{\partial}{\partial x_j}(\rho v_j h)
\]  

(3)

These equations describe the conservation of mass, momentum, and energy for gas and particle phases, as well as the energy equation for the gas and particle temperatures. The terms on the right-hand side of the equations represent various sources and sinks of mass, momentum, and energy, including buoyancy (\(g_i\)), inelastic collisions (\(F_{mi}\)), and heat transfer between phases (\(W_s Q_s\)).
\[
\frac{\partial}{\partial t} (\rho Y_s) + \frac{\partial}{\partial \xi_j} (\rho v_j Y_s) = \frac{\partial}{\partial \xi_j} \left( D \frac{\partial Y_s}{\partial \xi_j} \right) - W_S - \alpha_S \sum_p n_p m_p \frac{\partial}{\partial \xi_j} (\rho Y_s v_j) \]
\[(4)\]

\[
\frac{\partial n_p}{\partial t} + \frac{\partial}{\partial \xi_j} (n_p v_{pj}) = -\frac{\partial}{\partial \xi_j} (n_p v_{pj}) \]
\[(5)\]

\[
\frac{\partial (n_p m_p)}{\partial t} + \frac{\partial}{\partial \xi_j} (n_p m_p v_{pj}) = -m_p \frac{\partial}{\partial \xi_j} (n_p v_{pj}) + n_p m_p \]
\[(6)\]

\[
\frac{\partial (n_p m_c)}{\partial t} + \frac{\partial}{\partial \xi_j} (n_p m_c v_{pj}) = -m_c \frac{\partial}{\partial \xi_j} (n_p v_{pj}) + n_p m_c \]
\[(7)\]

\[
\frac{\partial (n_p m_w)}{\partial t} + \frac{\partial}{\partial \xi_j} (n_p m_w v_{pj}) = -m_w \frac{\partial}{\partial \xi_j} (n_p v_{pj}) + n_p m_w \]
\[(8)\]

\[
\frac{\partial}{\partial t} (n_p p_{pi}) + \frac{\partial}{\partial \xi_j} (n_p p_{pi} v_{pi}) = n_p g_i + \frac{1}{\tau_r} \left[ n_p (v_i - v_{k_i}) - n_p v_{pi} \right] + \frac{n_p m_p}{m_p} (v_i - v_{pi}) \]
\[- \frac{\partial}{\partial \xi_j} \left( n_p v_{pj} v_{pi} + v_{pj} n_p v_{pi} + v_{pi} n_p v_{pj} \right) \]
\[(9)\]

\[
\frac{\partial}{\partial t} (n_p h_p) + \frac{\partial}{\partial \xi_j} (n_p v_{pj} h_p) = \frac{n_p}{m_p} (Q_h - Q_p - Q_{rp}) + \frac{n_p m_p}{m_p} (h - h_p) \]
\[- \frac{\partial}{\partial \xi_j} \left( n_p v_{pj} h_p + v_{pj} n_p h_p + h_p n_p v_{pj} \right) \]
\[(10)\]

where Eqs. (1) through (10) are gas continuity, gas momentum, gas energy, gas species, particle number density, total particle mass, particle raw coal, particle moisture, particle momentum and particle energy equations respectively, and \( W_s \) is the time-averaged reaction rate. All correlations and the time-averaged reaction rate in these equations need to be modeled. The particle mass changing rate \( \dot{m}_p \) due to moisture evaporation, pyrolyzation and char combustion are determined by the algebraic expressions based on a surface evaporation model, one-equation pyrolyzation model and diffusion-kinetic model of char combustion. For two-phase turbulence modeling, a k-\( \varepsilon \)-kp model is used. For volatile and CO combustion in the comprehensive modeling of coal combustion, originally the conventional EBU-Arrhenius model is used (Zhou at al., 2000). The nomenclature for the above-stated equations can be found there.

The NO formed in coal combustion consists of mainly thermal NO and fuel NO. For the reaction kinetics of thermal NO formation, the well-known Zeldovich mechanism is used

\[
\text{N}_2 + \text{O} \rightarrow \text{NO} + \text{N} \\
\text{N} + \text{O}_2 \rightarrow \text{NO} + \text{O} 
\]
The total reaction rate is determined by

\[ W = 8.39 \times 10^{16} \rho^{1.5} T^{-0.5} Y_{N_2} Y_{O_2}^{0.5} \exp(-564.4 \times 10^3 / RT) \]

For the reaction kinetics of fuel NO formation, the DeSoete mechanism is used:

- HCN + O2 → NO (a)
- HCN + NO → N2 (b)
- C + NO → N2 (c)

Table 2 gives the kinetic constants in \( K = B \exp(E / RT) \), where \( K \) is the reaction coefficient.

| Reaction | B         | E (J/mol) |
|----------|-----------|-----------|
| a        | 1 \times 10^{11} (1/s) | 2.803 \times 10^5 |
| b        | 3 \times 10^{12} (1/s) | 2.5105 \times 10^5 |
| c        | 41.8 kg-mol/(atm.m^2.s) | 1.425 \times 10^5 |

For NO formation in turbulent flows, an algebraic second-order moment (ASOM) turbulence-chemistry model is proposed. The time-averaged reaction rate is (Zhou et al., 2003)

\[ \bar{W} = B \rho^{1.5} \left[ \bar{Y}_N \bar{Y}_O \bar{K} + \bar{Y}_N \bar{K} \bar{Y}_O + \bar{Y}_O \bar{K} \right] \]

\( K = \exp(-E / RT) \)

where, \( \bar{K} = \int \exp(E / RT) p(T) dT \), \( p(T) \) is the PDF of temperature. Assuming a top-hat PDF gives

\[ \bar{K} = \{ \exp[-E / R(T + g_{T1}^{1/2})] + \exp[-E / R(T - g_{T2}^{1/2})] \}, g_T = T^{1/2} \]

The correlations \( \bar{K} \bar{Y}_N, \bar{K} \bar{Y}_O, \bar{Y}_N \bar{Y}_O, \bar{Y}_N \bar{K}, \bar{Y}_O \bar{K} \) in Eq. (11) can be determined by a generalized form of the following transport equation:

\[
\frac{\partial}{\partial t} (\rho \bar{\psi} \bar{\Phi}) + \frac{\partial}{\partial \mathbf{x}_j} (\rho \mathbf{V_j} \bar{\psi} \bar{\Phi}) = \frac{\partial}{\partial \mathbf{x}_j} \left( \frac{\mathbf{e}_{\phi}}{\mathbf{e}_{\psi}} \frac{\partial \bar{\psi} \bar{\Phi}}{\partial \mathbf{x}_j} \right) + c \bar{g}_T \frac{\partial \Phi}{\partial \mathbf{x}_j} \frac{\partial \psi}{\partial \mathbf{x}_j} - c g_{T1} \frac{\partial \psi}{\partial \mathbf{x}_j} \]

\( \psi, \phi \) are the instantaneous species concentration and reaction coefficient \( K \); \( \Phi, \Psi \) are the mean species concentration and reaction coefficients.

In case of coal combustion, due to the complexity of the processes, for the sake of simplicity Eq. (13) is reduced to the following algebraic expressions by neglecting the convection and diffusion terms:

\[
\bar{Y}_N \bar{Y}_O = C_1 \frac{k^3}{\bar{e}^2} \frac{\partial \bar{Y}_N}{\partial \mathbf{x}_j} \frac{\partial \bar{Y}_O}{\partial \mathbf{x}_j}; K \bar{Y}_N = C_1 \frac{k^3}{\bar{e}^2} \frac{\partial \bar{K}}{\partial \mathbf{x}_j} \frac{\partial \bar{Y}_N}{\partial \mathbf{x}_j}; \bar{Y}_O \bar{K} = C_1 \frac{k^3}{\bar{e}^2} \frac{\partial \bar{Y}_O}{\partial \mathbf{x}_j} \frac{\partial \bar{K}}{\partial \mathbf{x}_j} \]

where \( k, \bar{e} \) are the turbulent kinetic energy and its dissipation rate. The temperature or enthalpy fluctuation correlation \( g_T \) can be determined by:
Equations (11), (12), (14), (15) constitute the algebraic unified second-order moment (ASOM) turbulence-chemistry model for NO formation in turbulent flows. The nomenclature in these equations is given by Zhou et al. (2003).

5. MEASUREMENT & SIMULATION RESULTS OF GAS-PARTICLE FLOWS

Figure 8 gives the predicted and measured gas and particle axial velocities for Case 1. It is seen that measurements and predictions are near to each other. Both of them show large-size reverse flow zones and particles keep forward motion in the central reverse flow zone. Figure 9 shows the measured and predicted particle mass fluxes. Predictions and measurements are in good agreement. Due to the particle inertia, a large amount of particles enter the central reverse flow zone, forming high particle concentration there, hence enhance flame stabilization at low loads and reduce NO formation. The measurements and simulation give the optimal jet-hole location and flow-rate ratio (not permitted to be reported here due to technical license) under which the size of recirculation zone and the reverse flow rate are maximal.
Figure 13 gives the predicted two-phase u-v velocity vectors in the spouting zone of the improved SCC for Case 2. It is seen that the gas and particle recirculation is much better than that in the original SCC, almost occupying the full space of the primary combustion zone.

Figure 14 shows the measured and predicted gas and particle axial velocities for Case 3 respectively. The agreement is quite good. It can be seen that a large reverse flow zone ranging from $x=0$ to $x=1.5m$ is formed at the near-axis region of the burner exit. The maximum width of the near-axis recirculation zone locates at $x=0.3$ m, and highest negative velocities of about -
5m/s are found at the location x=0.1m within the near-axis recirculation zone. Figure 15 shows the measured and predicted gas and particle tangential velocities for Case 3 respectively. At first and second cross sections (x=10mm, x=110mm), the predicted tangential velocities are higher than those measured, and further in the downstream region there still exists the discrepancy between predictions and experiments. Since the k-\( \varepsilon \)-kp model can well predict weakly swirling flows, and predicted tangential velocities show reasonable Rankine-vortex structure observed in swirling flows, it may be considered that the discrepancy is caused mainly by errors in measurement of tangential velocities. Because the radius of the furnace model is so large, it is impossible to guarantee the symmetry of measurements. The measured tangential velocities at the axis are not zero which is obviously wrong. Figure 16 gives the comparison between the predicted gas and the particle axial velocities for Case 3. It is found that the velocity slip between the gas and particle phases is evident only at the first, second and third cross sections. The particles initially have almost the same velocity as that of the primary-air flow. The secondary air flow has higher velocity and mass flow rate than those of the primary-air flow; therefore the velocity slip between gas and particle phases is large near the inlet region. On the other hand, due to higher inertia, the particles are not able to follow the rapid expansion and deceleration of the air flow, so that the particle velocity is higher than the gas one in some regions.
6. MEASUREMENT RESULTS OF COAL COMBUSTION
The combustion test for Case 2 was done with bituminous coal particles of 3-5mm. The proximate analysis of the Chinese Datong coal is: volatile 24.7%, ash 11.7%, moisture 3%, fixed carbon 63.3%. The temperature rise in the spouting zone and the cyclone zone is shown in Fig.17. The temperature in the spouting zone is in the range of 1273-1373K, which is below the ash melting point. The temperature in the cyclone zone is about 100K higher than that in the spouting zone. The NOx concentration is in the range of 100-320ppm (Fig.18), which is acceptable for a small-size furnace. The combustion efficiency reaches 92%, which is considered as a good performance for such small-size combustors burning ungrounded coal particles.

7. SIMULATION RESULTS OF COAL COMBUSTION
Figures 19 and 20 show the predicted gas and particle temperature maps for Case 1. It is seen that the high temperature is developed right at the stagnation point of the central reverse flow zone, and particle temperature is higher than the gas temperature. Figures 21 and 22 give the predicted thermal and total NO concentration for Case 1 respectively. Obviously, the total NO concentration is much higher than the thermal NO concentration. Simulation of coal combustion and NO formation was carried out also for a swirl combustor measured by Abbas et al. (1993), as shown in Fig.23. The geometrical sizes and inlet-flow conditions are given in Tables 3 and 4. The coal proximate analysis is given in Table 5. The pulverized coal is fed through the annular space “b”, the secondary air is supplied through the annular space “c”, and a bluff body is located in the central orifice “a”. The average initial size of the pulverized coal is 45 μm. The prediction results for cold flows are compared with the PDPA measurement results and for coal combustion are compared with the coal combustion experimental results (Abbas et al., 1993).
Table 3 The Geometrical Sizes of the Swirl Combustor

| D    | L    | d₁   | d₂   | d₃   |
|------|------|------|------|------|
| 0.6m | 3.0m | 0.022m | 0.035m | 0.104m |

Table 4 Inlet-Flow Parameters of the Combustor

|                | Air Flow Rate (kg/h) | Coal Feeding Rate (kg/h) | Swirl Number |
|----------------|----------------------|--------------------------|--------------|
| Primary Air    | 32.8                 | 14                       | 0.0          |
| Secondary Air  | 120.4                | 0                        | 0.5; 0.8; 1.0; 1.4 |

Table 5 Proximate Analysis of the Coal

| Volatile (%) | Char (%) | Water (%) | Ash (%) |
|--------------|----------|-----------|---------|
| 35.8         | 53.7     | 6.3       | 4.2     |

Fig. 23 The Coal Swirl Combustor

Fig. 24 Gas Axial Velocity (—Pred □ Exp)

Fig. 25 Turbulent Energy (—Pred □ Exp)

Fig. 26 Temperature (— Pred. □ Exp.)

Fig. 27 NO Concentration (— Pred. □ Exp.)
Figures 24 and 25 show the predicted and measured gas velocity and turbulent kinetic energy for cold flows and Figure 26 gives the predicted and measured temperature profiles, showing a good agreement between predictions and experiments. Figure 27 shows the predicted NO concentration profiles and their comparison with the experimental results. The agreement is also good. Figures 28 and 29 give the predicted averaged NO concentration at the exit and burnout rate vs. as the swirl number respectively and their comparison with experimental results. Both predictions and experiments show the common tendency: as the swirl number increases the NO concentration at first will decrease and then will increase, while the burnout rate at first will increase and then will decrease. There is a quantitative discrepancy between predictions and experiments. The predicted lowest NO emission and highest burnout rate occur at the swirl number of 0.8, but the measured ones occur at the swirl number of 1.0. This discrepancy may be caused by numerical errors and inaccuracies of the models. The overall NO formation in coal combustion should be determined by the temperature, coal concentration and turbulent fluctuation. With the increase of swirl number from 0.5 to 0.8, the temperature increases not so much, but the coal concentration in the inlet zone increases and the turbulent fluctuation decreases. Therefore, the NO formation decreases.

8. CONCLUSIONS

(1) LDV/PDPA measurements and simulation of isothermal gas-particle flows give the optimum design which can create large-size recirculation zones and high particle concentration for stabilizing coal flame and reducing NO formation in coal combustors.

(2) Combustion measurements show that a small-size non-slagging cyclone coal combustor can burn 3-5mm coal particles with high combustion efficiency and low NO emission.

(3) The full two-fluid model developed by the present author can well simulate turbulent gas-particle flows, coal combustion and NOx formation.

(4) Simulation in comparison with experiments gives the effect of swirl on NO emission in a swirl coal combustor

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