

**DEVELOPMENT OF A NOVEL RADIATIVELY/CONDUCTIVELY STABILIZED
BURNER FOR SIGNIFICANT REDUCTION OF NO_x EMISSIONS
AND FOR ADVANCING THE MODELING AND UNDERSTANDING
OF PULVERIZED COAL COMBUSTION AND EMISSIONS**

Final Report

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Abstract

The primary objective of the proposed study was the study and analysis of, and design recommendations for, a novel radiatively-conductively stabilized combustion (RCSC) process for pulverized coal, which, based on our prior studies with both fluid fuels and pulverized coal, holds a high promise to reduce NO_x production significantly. We have primarily engaged in continuing and improving our process modeling and analysis, obtained a large amount of quantitative information about the effects of the major parameters on NO_x production, conducted an extensive exergy analysis of the process, evaluated the practicalities of employing the Radiatively-Conductively Stabilized Combustor (RCSC) to large power and heat plants, and improved the experimental facility. Prior experimental work has proven the feasibility of the combustor, but slagging during coal combustion was observed and should be dealt with. The primary outcomes and conclusions from the study are: (1) we developed a model and computer program that represents the pulverized coal combustion in the RCSC, (2) the model predicts that NO_x emissions can be reduced by a number of methods, detailed in the report. (3) the exergy analysis points out at least a couple of possible ways to improve the exergetic efficiency in this combustor: increasing the effectiveness of thermal feedback, and adjusting the combustor mixture exit location, (4) because of the low coal flow rates necessitated in this study to obtain complete combustion in the burner, the size of a burner operating under the considered conditions would have to be up to an order of magnitude, larger than comparable commercial burners, but different flow configurations of the RCSC can yield higher feed rates and smaller dimensions, and should be investigated. Related to this contract, eleven papers were published in journals and conference proceedings, and ten invited presentations were given at university and research institutions, as well as at the Gordon Conference on Modern Development in Thermodynamics. The results obtained are very encouraging for the development of the RCSC as a commercial burner for significant reduction of NO_x emissions, and highly warrants further study and development.

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1. EXECUTIVE SUMMARY

We have primarily engaged in continuing and improving our process modeling and analysis, obtained a large amount of quantitative information about the effects of the major parameters on NO_x production, conducted an extensive exergy analysis of the process, evaluated the practicalities of employing the Radiatively-Conductively Stabilized Combustor (RCSC) to large power and heat plants, and improved the experimental facility. Prior experimental work has proven the feasibility of the combustor, but slagging during coal combustion was observed and should be dealt with.

The primary outcomes and conclusions from the study are:

- We developed a model and computer program that represents the pulverized coal combustion in the RCSC. Some of its main features are detailed accounting of the heat and mass transfer in the process, including the Radiative Transport Equation, a k -epsilon turbulence model for the gas, a Lagrangian model for the particle flow, the kinetics of 14 species, and the exergy field description.

- We predict that NO_x emissions can be reduced by using smaller coal particle sizes, by feeding the coal in a way that it is more concentrated near the combustor walls, by enhancing the NH_3 NO_x -reduction reaction, by increasing the excess air ratio, and most significantly, by increasing the inlet air/coal mixture temperature. The latter can be accomplished by using the RCSC in a recirculating heater configuration, also investigated in this study.
- While one may draw many conclusions from this analysis about possible ways to improve the exergetic efficiency in this combustor, a couple are already evident. One is to increase the effectiveness of thermal feedback: the thermal exergy is raised by this feedback upstream of the flame front to about 1/3 of its maximal value. Another is the possibility of the existence of an optimal combustor exit location for the mixture, where the exergy is higher than the one shown here and yet the reactions were completed to satisfactory extent.
- Because of the low coal flow rates necessitated in this study to obtain complete combustion in the burner, the size of a burner operating under the considered conditions would have to be much, up to an order of magnitude, larger than comparable commercial burners. However, different flow configurations of the RCSC can yield higher feed rates and smaller dimensions, and should be investigated.
- Related to this contract, eleven papers were published in journals and conference proceedings, and ten invited presentations were given at university and research institutions, as well as at the Gordon Conference on Modern Development in Thermodynamics

- Encouraged by the NETL project monitor, Dr. Philip Goldberg, we have made contacts with the two other groups engaged in DOE funded research on reducing NO_x emissions in coal combustion (Prof. Wendt, U. of Arizona with Prof. Sinclair, Purdue U., and Profs. Pershing, Lighty and Sarofim, U. of Utah) to explore ways of cooperation in this work. We have agreed among us on several specific cooperative efforts, and continued communications during the project.
- The results obtained are very encouraging for the development of the RCSC as a commercial burner for significant reduction of NO_x emissions, and highly warrants further study and development.

2. BACKGROUND

The Radiatively-Conductively Stabilized Combustor (RCSC) is a combustion method developed by the Principal Investigators and co-workers (cf. Chen and Churchill 1972A, 1972B, Tang, Churchill and Lior 1980, 1981, Goepf, Tang, Lior and Churchill 1980, Li and Lior 1990, Chan, Arai, Lior and Churchill, 1992, Kim and Lior 1993, 1996), which demonstrated the generation of extremely low levels of NO_x emissions (5-100 ppm with fluid fuels), mostly because of the rapid heating of the fuel and minimal back-mixing of the products of combustion in such combustors. This is a reduction by approximately an order of magnitude as compared to conventional burners. We have demonstrated that pulverized coal flames can also be stabilized in such a combustor by preheating with propane. The work done under this contract was to investigate this promising process of radiatively-conductively stabilized combustion in refractory tubes by thorough heat transfer and kinetic modeling and experimentation, with air-entrained pulverized coal as

the fuel, and to advance the development of a commercial/industrial low-NO_x burner based on this principle.

This report summarizes the work done and its conclusions. Information published by us in journals and conference proceedings, listed in Section 14 of this report, is only briefly summarized, and the details can be found in these papers.

3. OBJECTIVE

The primary objective of the proposed study is the rigorous modeling, analysis, experimental exploration, and validation, and design recommendations for a novel radiatively-conductively stabilized combustion (RCSC) process for pulverized coal, which, based on our prior studies with both fluid fuels and pulverized coal, holds a high promise to reduce NO_x production significantly.

4. THE ORIGINALLY PROPOSED WORK STATEMENT AND THE ACCOMPLISHMENTS

Task 1: Modeling and Analysis:

1.1: Improved chemical kinetics: as described above, a more comprehensive and complete free radical model (close to 200 reactions are expected) would be included to provide the proper description of the reaction product generation rates throughout the combustor, including the devolatilization and char combustion phases.

Accomplished: The model was improved as described in Section 5.3 below: the kinetic description of the major reactions was improved and their role was better clarified, and a better computational method for this very stiff differential equation system was developed and implemented. The free-radical model was not implemented because there were other more important parts of the project to do, and was planned to be done in the no-cost extension period denied by NETL. While a free radical model may have improved somewhat the prediction accuracy of the emitted species, the model we have used is entirely adequate for the thermal analysis of the combustor, and for acceptable

accuracy of the species evolution. In addition, a rather complete exergy analysis of the process was done.

1.2 Improved fluid mechanics: the current three-layer turbulent model would be replaced by the k- ϵ or similar model, and the particle motion equations which also take into consideration the size change of the particle during its passage through the combustor would be improved.

Accomplished: (1) The 3-layer turbulent flow model was replaced by an advanced k-epsilon model, (2) the no-slip coal particle flow model was replaced in the program by a much more appropriate particle flow module, based on a Lagrangian approach. Task 1.2 was thus fully completed.

1.3 Improved radiative transfer solution: The P-3 or another better approximation would be used to replace the currently used P-1 approximation.

Accomplished: The P-3 approximation was added to the model, but produces results only a few % different than those obtained from the P-1 approximation. Since it increases computation time significantly, it is not necessary to use it for the combustor configurations and conditions considered.

1.4 Co-combustion with gaseous fuel: the model would be modified to allow analysis of the effects of co-combustion of the coal with gaseous fuel, such as propane.

Accomplished: This modification of the model was started but not completed. It was planned to be done in the no-cost extension period denied by NETL.

1.5 Modeling and sensitivity analysis of combustor configurations: Guided by the results of the improved combustor model, the sensitivity of pollutant production and combustion efficiency would be investigated as a function of the main problem parameters described above (coal/air ratio, presence of a gaseous fuel, thermal properties and configuration of the combustor, coal particle size). The optimal combination of the parameters would be used to model and design the prototype combustor.

Accomplished: Performed and completed, as shown in Section 7 below.

Task 2: Experimental Studies and Validation months 3-24.

Experiments would be conducted to validate or develop some of the model equations for which conclusive confidence is unavailable in the literature, and also to validate our overall model with representative parameters which are within the overall practical parameter envelope.

Accomplished: (1) The experimental facility was improved and prepared as described in Section 8 below, and preliminary experiments were conducted, (2) The severe slagging described in the previous experiments (Section 9 below) has prevented continuation of the experiments, the Ph.D. student who has helped improve the facility and was going to

work on resolving this problem has unexpectedly (and unexplainably) left the University, and the no-cost extension period was denied by NETL, (3) Nevertheless, many of our computational model results have been validated by comparison with our experimental results and those from other researchers, as shown in our publications and Section 9 below.

Task 3: Design and Construction of the Prototype Combustor

3.1 Design: the design of the prototype would be based on the results of Task 1.5 and experience from Task 2.

Accomplished: A preliminary design was completed as described in Sections 8 and 11.

3.2 Construction We expect to construct a small version of the prototype in our laboratory, based on our past experience. In the construction of a larger industrial/commercial combustor we would be assisted by the power utilities and a combustor manufacturer (Cleaver Brooks) which have expressed interest in participating in this development.

Accomplished: Only the single tube RCSC was built and tested. The severe slagging described in the previous experiments (Section 9 below) has prevented continuation of the experiments, and the Ph.D. student who has helped improve the facility and was going to work on resolving this problem has unexpectedly (and unexplainably) left the University, and the no-cost extension period was denied by NETL,

Task 4: Combustor Testing and Development.

The prototype combustor would be tested within the full envelope of operating parameters, to determine emissions of all major pollutants (NO_x , SO_x , CO , CO_2 , HCN) as well as combustor efficiency. Furthermore, ease of ignition, operation, slagging, and reliability, would be monitored and recorded:

Accomplished: The single tube vertical combustor was constructed, ignition was successfully achieved and continuous combustion was maintained (Section 9 below). Severe slagging inside the combustor tube blocked it significantly after several hours of operation and further development of the experimental part of this work had to be stopped.

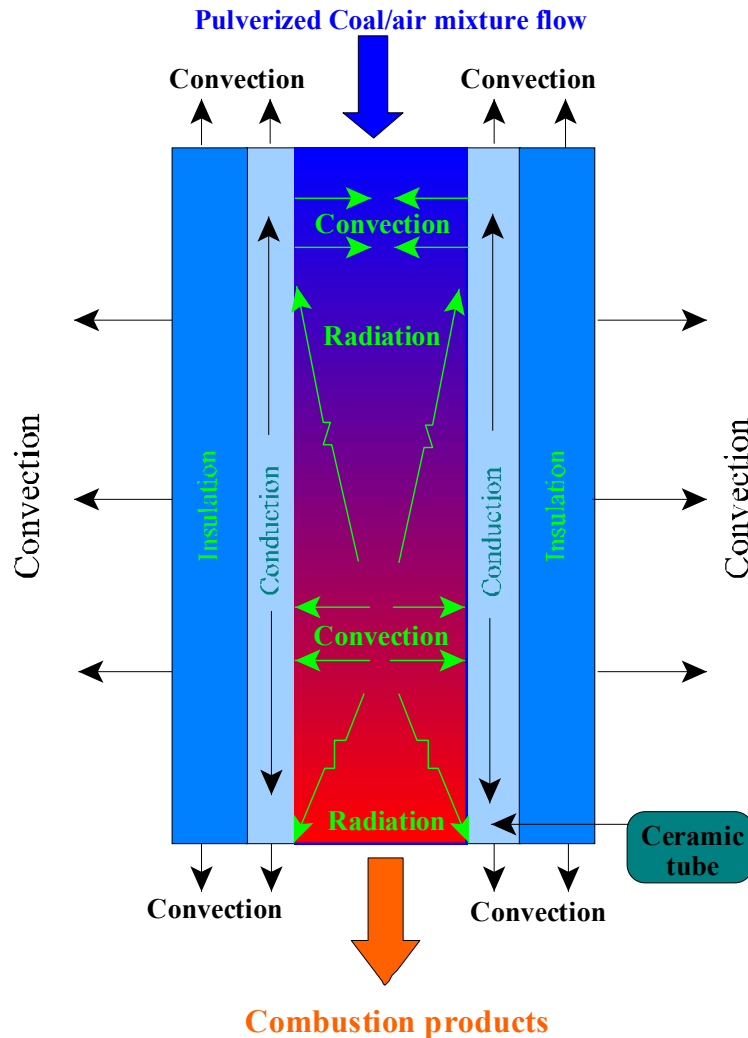
5. THE COMBUSTION MODEL

5.1 Basic Features

Our basic pulverized coal combustion model in the RCSC, which have developed in the past and with which we started this study is described in detail by Kim and Lior (1998). Its original version was successfully validated by comparison with experimental data (Kim and Lior 1993, 1996, 1998), but not in all aspects. For example, the actual comparison of the predicted NO_x emissions to experiments was not possible and is recommended for the future. Nevertheless, the validated aspects and the sensible predicted results give us, and the peer reviewers of our published papers, confidence in the model.

The basic principle of flame stabilization by radiation and conduction from the walls is depicted in Fig. 5.1.

Fig. 5.1 The Thermal Feedback Mechanism in the RCSC



Pre-mixed coal particles and air enter the tube and are heated to the ignition temperature primarily by convection from the tube wall and the radiantly-heated particles. The hot, burned gases and particles heat the down stream wall by convection and radiation. This stabilization mode produces a very stable, oscillation-free flame. The residence time of the reactants in the flame zone is much shorter than that in recirculation burners, therefore, the RCSC yields very low levels of thermal NO_x .

The basic features of the phenomena, as obtained from our computations, are shown in Figs 5.2 and 5.3 below. Figure 5.2 shows the temperature distribution in the RCSC, and

Figs. 5.3 shows the species concentration distribution at the centerline along the combustor.

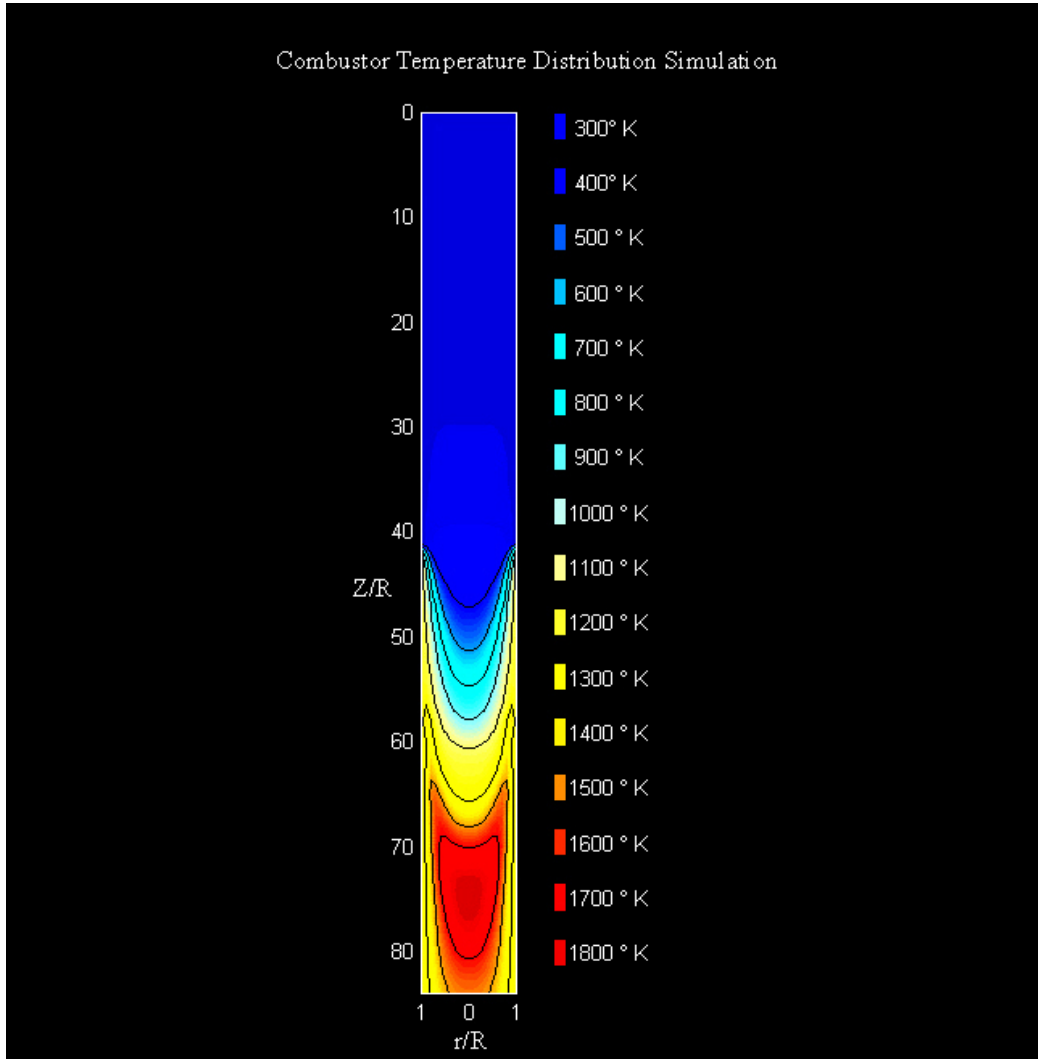


Fig. 5.2 A computed temperature distribution in the RCSC. Air/coal mixture at 300C enters at the top, and gradually warms up due to direct radiant heating from the flame, and feedback heating from the wall.

Gas Distribution in Combustor

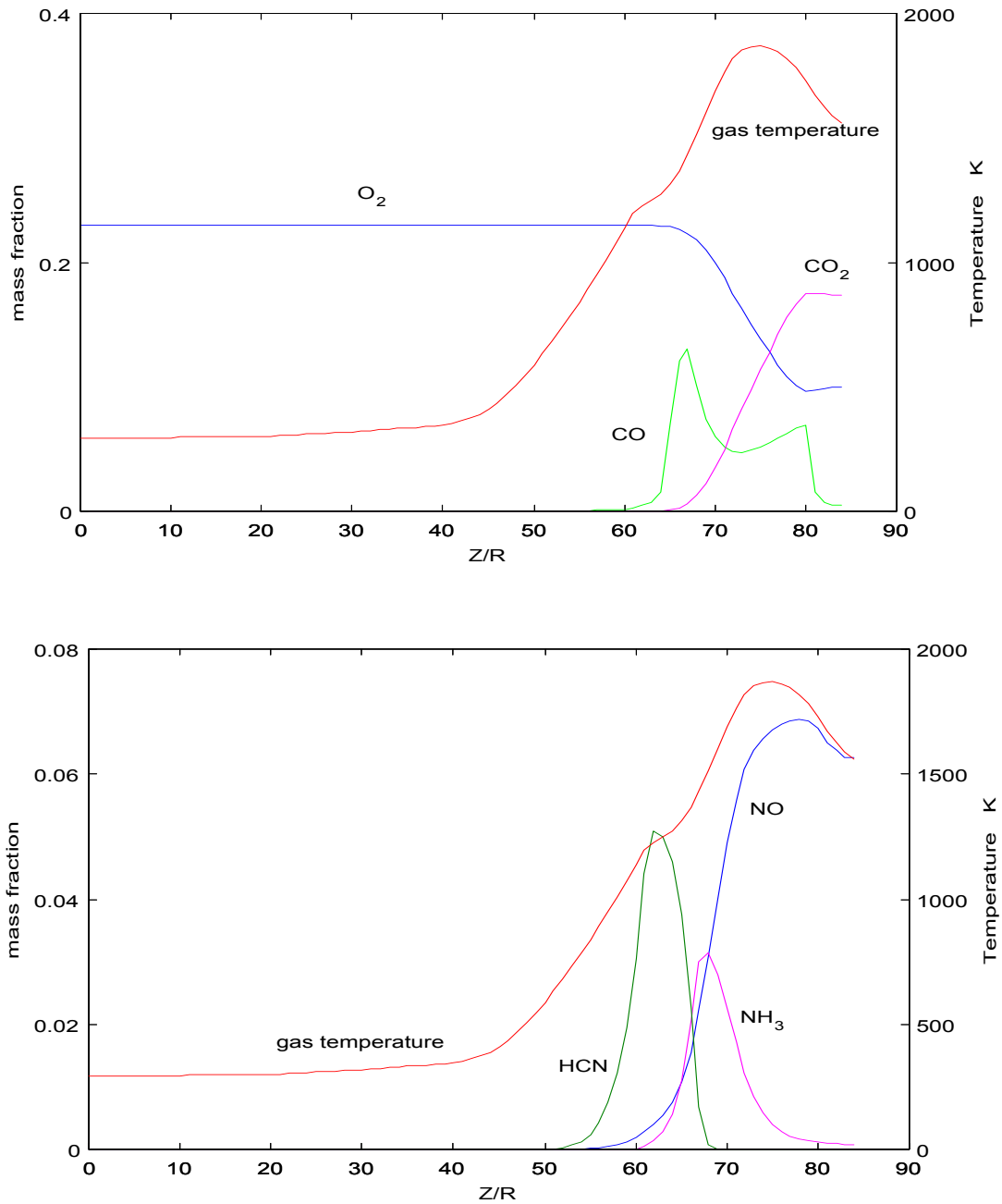


Fig. 5.3 The computed gas temperature and composition along the RCSC. NO production follows the rise in the gas temperature. Noteworthy is the formation of HCN and its conversion to NH_3 as the temperature rises, and then the partial reduction of NO by the NH_3 .

A number of important improvements in this model were made in the course of this study: (1) the 3-layer turbulent flow model was replaced by an advanced k-epsilon model, (2) the kinetic description of the major reactions was improved and their role was better clarified, and a better computational method for this very stiff differential equation system was developed and implemented, (3) the no-slip coal particle flow model was replaced in the program by a much more appropriate particle flow module, based on a Lagrangian approach

The computational scheme is shown in Fig. 5.4.

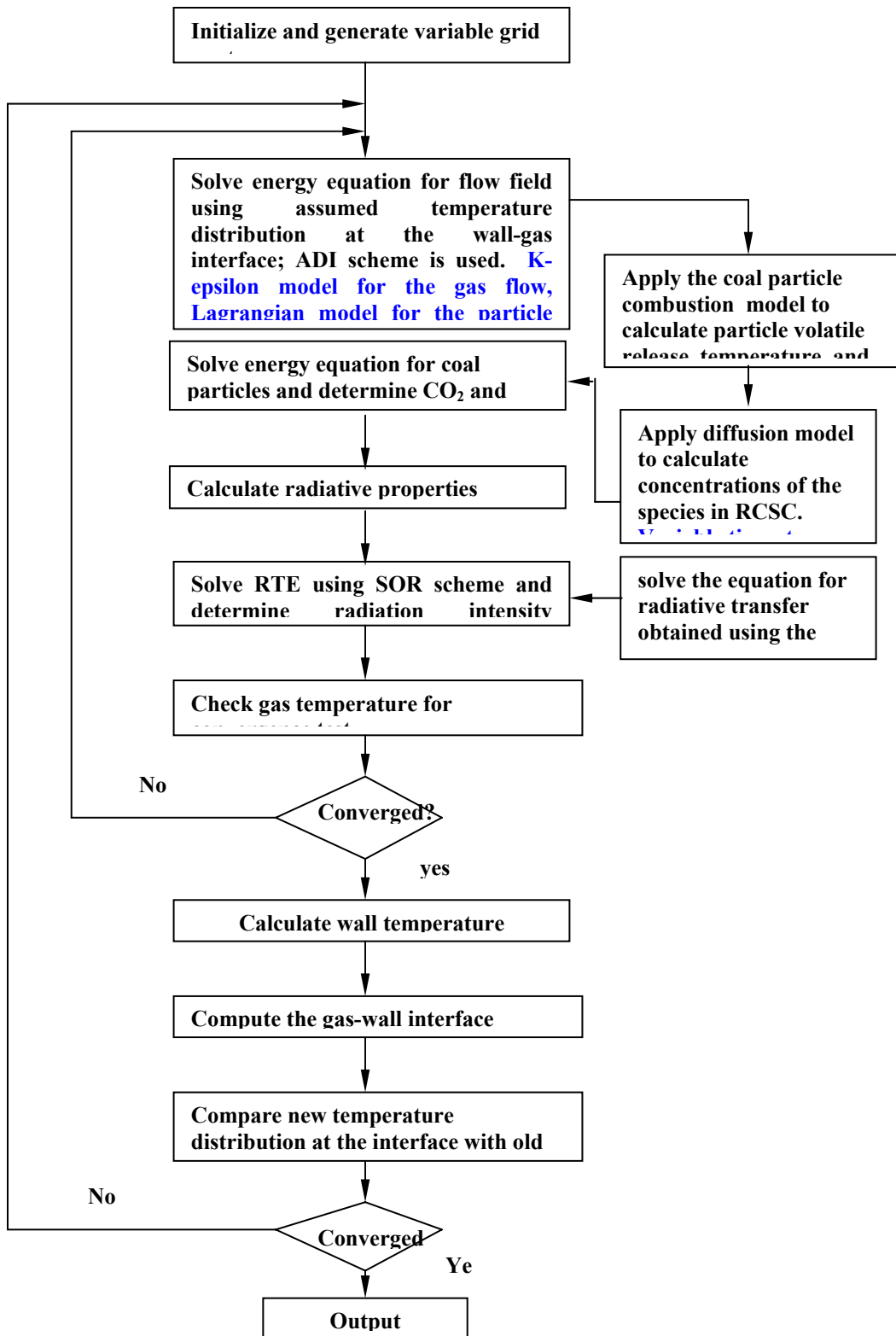


Fig. 5.4 The computational scheme

5.2 Coal particle motion modeling

5.2.1 Background. Gas-solid flow is a common problem in coal combustion research. During passage through the combustor the particles heat up, coal particles devolatilize, change size and even eventually break up into smaller ones. All this should be included in coal combustion model. Eaton [1] (the references to this Section are at its end, and the equation numbers are just for this Section) suggested that we can model the changing by equations of mass, momentum and energy along its trajectory as following equations

$$\frac{dm_p}{dt} = -R_p \quad (1)$$

R_p is the rate of change of mass due to devolatilization and combustion

$$m_p \frac{d\vec{U}_p}{dt} = \sum \vec{F}_p \quad (2)$$

$$\frac{d(mh)_p}{dt} = \sum Q_p \quad (3)$$

Q_p is the sum of the various sources of energy, such as heat conduction, convection, radiation, and reaction heat.

5.2.2 Single particle motion in gas-solid flow.

(1) Forces acting on particle

In gas-solid flow, one of the basic and important aspects is single particle motion in gas-solid two phases flow. The flow pattern of particle is affected by initial conditions, boundary conditions of domain and interacting forces acting on the particle. Base on the Newton law, we can get the general equation for particle motion as

$$m \frac{d\vec{U}_p}{dt} = \sum \vec{F} \quad (2)$$

These forces controlling the motion of particle could be classified as (1) forces through the interface between fluid and particle, (2) forces imposed by external fields. [9]

The major influence of fluid acting on a single particle could be expressed as being composed of four different forces. They are the Drag force, Basset force, Saffman force and Magnus force.

Drag force comes from the unbalanced pressure distribution and viscous stress on the particle surface.

$$F_D = C_D A \frac{\rho}{2} |\vec{U} - \vec{U}_p| (\vec{U} - \vec{U}_p) \quad (4)$$

A: exposed frontal area of particle

C_D : drag coefficient, function of Re_p

$$Re_p = \rho d_p \frac{|\vec{U} - \vec{U}_p|}{\mu} \quad (5)$$

Basset force

The Basset force comes from the acceleration and deceleration of particle in fluid. Basset force is negligible when the fluid-particle density ratio is small, the time change is longer than the Stokes relaxation time or the acceleration rate is low.[9]

$$F_z = \frac{M_a}{a^2} \frac{\partial}{\partial t} \left[U_p \left(4.5 \nu t + 9a \sqrt{\frac{\nu t}{\pi}} + 0.5a^2 \right) \right] + M_a g \quad (7) [9]$$

M_a is the mass of fluid displaced

U_p is the particle velocity

With constant acceleration, the rate of the Basset force to the Stokes drag,

$$R_{BS} = \sqrt{\frac{18}{\pi} \frac{\rho}{\rho_p} \frac{\tau_s}{t}} \quad (8)$$

Stokes relaxation time could be expressed as

$$\tau_s = \frac{\rho_p d_p^2}{18\mu} \quad (9)$$

The Saffman force

Saffman force develops from the velocity gradient field.

$$F_s = 0.25 K \mu \left| \vec{U} - \vec{U}_p \right| d_p^2 \sqrt{\frac{1}{\nu} \left| \frac{\partial(\vec{U} - \vec{U}_p)}{\partial y} \right|} \quad (10)$$

Saffman gave the $K = 6.46$ in 1968, [2]

Ratio of the Saffman force to Stokes drag force is

$$R_{ss} = \frac{K d_p}{12\pi} \sqrt{\frac{1}{\nu} \left| \frac{\partial(\vec{U} - \vec{U}_p)}{\partial y} \right|} \quad (11)$$

In a constant shear rate flow, it can be estimated by

$$R_{ss} = \frac{K}{12\pi} \sqrt{\text{Re}_p} \quad (12)$$

At low Re_p or small shear rate it can be negligible.

The Magnus force

The Magnus force develops from particle rotation. It is lift force that moves the particle toward high velocity region.

F_M is independent of the viscosity.

$$F_M = \frac{\pi}{8} d_p^3 \rho \Omega \times \vec{U}_p \quad (13)$$

Ratio of the Magnus force to the Stokes drag is

$$R_{MS} = \frac{d_p^2}{24} \frac{\rho}{\mu} \Omega \quad (14)$$

When the spin is slow or the particle is small, the Magnus force can be neglected.

Additional forces

There are additional forces acting on the particle in fluid; they are pressure gradient force, gravity and buoyancy force, added mass force, and the rotating coordinates force.

Pressure gradient force

$$F_p = -\frac{1}{4} \pi d^3 \nabla P \quad (15)$$

Gravity and buoyancy force

$$F_B = \frac{1}{6} \pi d^3 (\rho_p - \rho) g \quad (16)$$

Added mass force

$$F_A = -\frac{1}{12} \pi d^3 \rho \frac{du}{dt} \quad (17)$$

Rotating coordinates force

$$F_R = \frac{1}{6} \pi d_p^3 \rho_p (-2\omega \times u - \omega \times \omega \times x) \quad (18)$$

The pressure gradient force is negligible when particle much larger than fluid density.

The added mass force is always automatically included. The rotating coordinates force is included when the coordinate is rotating. The gravity and buoyancy force is optional.

(2) Different models for Drag force and lift force

The BBO equation: Many Lagrangian flow models of motion of discrete particles are based on the equation developed by Basset(1888), Boussinesq(1903) and Oseen(1927), (BBO). In low Reynolds number, the motion of a sphere particle in linear flow could be expressed by the BBO equation,

$$\begin{aligned} \frac{\pi}{6} d_p^3 \rho_p \frac{d\vec{U}_p}{dt} = 3\pi \mu d_p (\vec{U} - \vec{U}_p) - \frac{\pi}{6} d_p^3 \nabla p + \frac{\pi}{12} d_p^3 p \frac{d}{dt} (\vec{U} - \vec{U}_p) \\ + \frac{3}{2} d_p^2 \sqrt{\pi \rho \mu} \int_0^t \frac{d}{d\tau} (\vec{U} - \vec{U}_p) \frac{d\tau}{\sqrt{t-\tau}} + \sum_i f_i \end{aligned} \quad (19)$$

The modified BBO equation is used when the flow is beyond the Stokes regime. The effect of convective acceleration of gas surrounding the particle is included,

$$\begin{aligned} \frac{d\vec{U}_p}{dt} = 0.75 \frac{C_D \rho}{d_p \rho_p} |\vec{U} - \vec{U}_p| (\vec{U} - \vec{U}_p) + \frac{\rho}{2\rho_p} C_I \frac{d}{dt} (\vec{U} - \vec{U}_p) + \frac{\rho}{\rho_p} \frac{D\vec{U}}{Dt} \\ + \frac{9\rho}{d_p \rho_p} \left(\frac{\nu}{\pi}\right)^{0.5} C_B \int_0^t \frac{d}{d\tau} (\vec{U} - \vec{U}_p) \frac{d\tau}{\sqrt{t-\tau}} + \left(\frac{\rho}{\rho_p} - 1\right) \vec{g} \end{aligned} \quad (20)$$

C_D : drag coefficient

C_I : carried mass coefficient

C_B : Basset coefficient

For a simple harmonic motion at $Re_p < 62$,

$$C_I = 2.1 - \frac{0.132 An^2}{1 + 0.12 An^2} \quad (21)$$

$$C_B = 0.48 + \frac{0.52 An^2}{(1 + An)^3} \quad (22)$$

An is the Acceleration number

$$An = \frac{\left| \frac{d(\vec{U} - \vec{U}_p)}{dt} \right|}{\left| \vec{U} - \vec{U}_p \right|^2} d_p \quad (23)$$

Gouesbet [3] recommended the modified Riley's equation, which can handle small and large particle Reynolds number. The modifications introduce correcting factor for drag term, added mass term and the Basset term.

$$\begin{aligned} \rho_p \frac{\pi d_p^3}{6} \frac{d\vec{U}_p}{dt} = & -\frac{\pi d_p^2}{8} \rho_g C_D (\vec{U}_p - \vec{U}_g) \left| \vec{U}_p - \vec{U}_g \right| \\ & - \rho_g \frac{\pi d_p^3}{6} C_A \frac{d(\vec{U}_p - \vec{U}_g)}{dt} + \frac{\pi d_p^3}{6} (\rho_p - \rho_g) \vec{g} \\ & + \rho_g \frac{\pi d_p^3}{6} \frac{D\vec{U}_g}{Dt} - \frac{\pi d_p^2}{4} C_H \frac{\sqrt{\rho_g \mu}}{\pi} \\ & \times \int_{-\infty}^t \frac{d(\vec{U}_p - \vec{U}_g)}{dt} (t - \tau)^{-0.5} d\tau \end{aligned} \quad (24)$$

$$C_D = \frac{24}{Re_p} (1 + 0.15 Re_p^{0.687}), \quad Re_p < 200 \quad (25)$$

$$C_A = 1.05 - 0.0066 / (A_C^2 + 0.12) \quad (26)$$

$$C_H = 2.86 - 3.12 / (A_C^2 + 1)^3 \quad (27)$$

$$A_C = \frac{\left| \vec{U}_p - \vec{U}_g \right|^2}{d_p \left| \frac{d(\vec{U}_p - \vec{U}_g)}{dt} \right|}, \quad A_C \leq 60 \quad (28)$$

When $Re_p < 1$, above equation will be the original Riley equation.

In dilute flow, the drag force is the major factor affecting the particle motion. In most flow the turbulent intensities lower than 20%, when the $\rho_p \gg \rho_g$, and assume particle is sphere, the particle momentum equation could be simplified since all other forces are negligible except drag force and gravity force. It could be written as [4]

$$\frac{d\vec{U}_p}{dt} = \frac{\vec{U}_p - \vec{U}_g}{\tau} + \vec{g} \quad (29)$$

When $Re_p < 200$, Clift et al.(1978) the drag coefficient could be expressed as

$$C_D = \left(\frac{24}{Re_p}\right)(1 + 0.15 Re_p^{0.687}) \quad (30)$$

$$\tau_p = \frac{m_p}{3\pi d_p \mu_g} \frac{1}{(1 + 0.15 Re_p^{0.687})} \quad (31)$$

Above equations are the mostly used models in current research to calculate the particle trajectories.

A similar model is widely used by many people. Karki (1996) used the following equation to calculate the coal particle [6].

$$m_p \frac{d\vec{U}_p}{dt} = \vec{F}_D + m_p \vec{g} \quad (32)$$

$$\vec{F}_D = 0.5(\rho_g A_p C_D) |\vec{U}_p - \vec{U}_g| (\vec{U}_p - \vec{U}_g) \quad (33)$$

$$C_D = \frac{24}{Re_p} (1 + 0.15 Re_p^{0.687}) \quad (34)$$

$$\text{Re}_p = \frac{\rho_g d_p |\vec{U}_p - \vec{U}_g|}{\mu_g} \quad (35)$$

For the lift force, D. Hall (1988) suggested that the mean fluid lift force on a sphere on a smooth surface could be expressed as following

$$F^+ = (20.9 \pm 1.57)(a^+)^{2.31 \pm 0.02} \quad (36)$$

Where F^+ is the non-dimensional force and a^+ is the non-dimensional particle radius scaled on fluid boundary layer parameter. [5] Surface roughness can significantly change the mean fluid-induced lift forces acting on particles.

Saffman [2] gave the lift force of particle in flow as

$$L = 6.46(U_g - U_p) \frac{d_p^2}{4} \sqrt{k/\nu} \quad (37)$$

k is the velocity gradient

(3) Mass change of a particle during combustion

In coal combustion flow, coal particles devolatilize, react with surrounding gases and break up into smaller ash and soot. Its density, size and properties keep on changing during this process. By assuming the particle swelling is proportional to the extent of volatile release (Smoot and Pratt, 1979), the transient diameter of a particle during devolatilization can be calculated as

$$d_p = d_{p,0} [1 + (1 - \gamma)(V/V_\infty)] \quad (38)$$

is the ratio of the initial particle diameter to the final diameter

During combustion, the diameter and density of particle keep on changing. Using the overall reaction constant K , the carbon consumption rate is

$$\frac{dm_p}{dt} = \pi d_p^2 K P_{o_2} \quad (39)$$

The diameter and density of coal particle during combustion is given by Essenhigh (1988) as following

$$\frac{\rho_p}{\rho_{p,0}} = \left(\frac{d_p}{d_{p,0}} \right)^\alpha \quad (40)$$

The parameter α is determined experimentally and varies with different coal. With the assumption of sphericity in shape throughout the combustion process, the diameter of char particles can be calculated as

$$d_p = \left(\frac{6m_p}{\pi \rho_p} \right)^{1/3} \quad (41)$$

The initial density (after devolatilization) is given as

$$\rho_{p,0} = \frac{6(m_{p,0} - V_\infty)}{\pi d_{p,\infty}^3} \quad (42)$$

D_p is the diameter of a particle after devolatilization.

Gas-solid flow in the RCSC system

In the current RCSC model, the domain is in a vertical ceramic tube. Cold air and coal particle mixture flows in from top. Coal particles absorb heat and begin to ignite, then devolatilize and react with surrounding gas. After combustion these particles will break

down to smaller particles such as ash and soot. The following table gives the basic information of the current model,

Tube length = 10.5 feet,

Tube diameter = 2 inches,

Inlet flow velocity of coal/air mixture = 5.419 m/s

Mass of dry air = 2.022 g/s

Mass of coal feed rate = 0.2 g/s

Reynolds number at inlet = 1753

Coal particle diameter = 100um

Coal particles number at inlet = 5037460

Assumptions:

1. At entrance all particles are uniform size spheres
2. The gas-solid flow is dilute and the presence of particles has no effect on the velocity profile.
3. The volume occupied by the coal particles is neglected.
4. Coal particles are uniformly distributed throughout the tube cross-section
5. Coal particles maintain spherical shape throughout the combustion process, but the size is changing continuously.
6. Char particles consist of pure carbon and ash
7. Char burning does not occur until devolatilization is completed.

8. Particle fragmentation does not occur until carbon consumption is complete.

9. Carbon is consumed by the reaction $C+O \rightarrow CO$

The current particle motion model in RCSC is following [7]

$$\text{(Smoot and Pratt, 1979)} \quad \frac{dU_p}{dt} = \frac{U_g - U_p}{\tau} \quad (43)$$

$$\text{Relaxation time (Wallis 1969)} \quad \tau = \frac{\rho_p d_p^2}{18\mu_g C_D} \quad (44)$$

$$\text{Drag coefficient} \quad C_D = \left[2 \left(\frac{12}{|\text{Re}_g - \text{Re}_p|} \right)^{0.5} + 0.371 \right]^2 \quad (45)$$

Turbulent dispersion of particle in fluid

In most turbulent flows, groups of particles that start from the same position might have different trajectories because of the particle dispersion of fluid turbulence. Combining mean velocity with a contribution due to turbulence could include the effect of turbulence. The turbulence is assumed to be made up of a collection of randomly directed eddies. The variance of the eddies speed is taken to be twice the turbulent kinetic energy $2k$, the eddy lifetime t_E and eddy length l_E are given by

$$t_E = 1.5^{0.5} C_\mu^{0.75} \frac{k}{\varepsilon} \quad (46)$$

$$l_E = C_\mu^{0.75} \frac{k^{1.5}}{\varepsilon} \quad (47)$$

C_μ is a turbulence model constant

The particle motion model computation algorithms

In order to improve the algorithms and model of coal particle motion in RCSC system,

we should select different models to compare their results. Therefore, the modified

Riley's equation and simplified model are selected to calculate the forces acting on coal particle. We select the equation 36 (by D. Hall) to compare with the lift force equation 37 given by Saffman [2]. During the calculation we will use equation 41 and 42 to calculate the particle size and density. The algorithm of coal particle motion computation in the RCSC model is shown in Fig. 5.5.

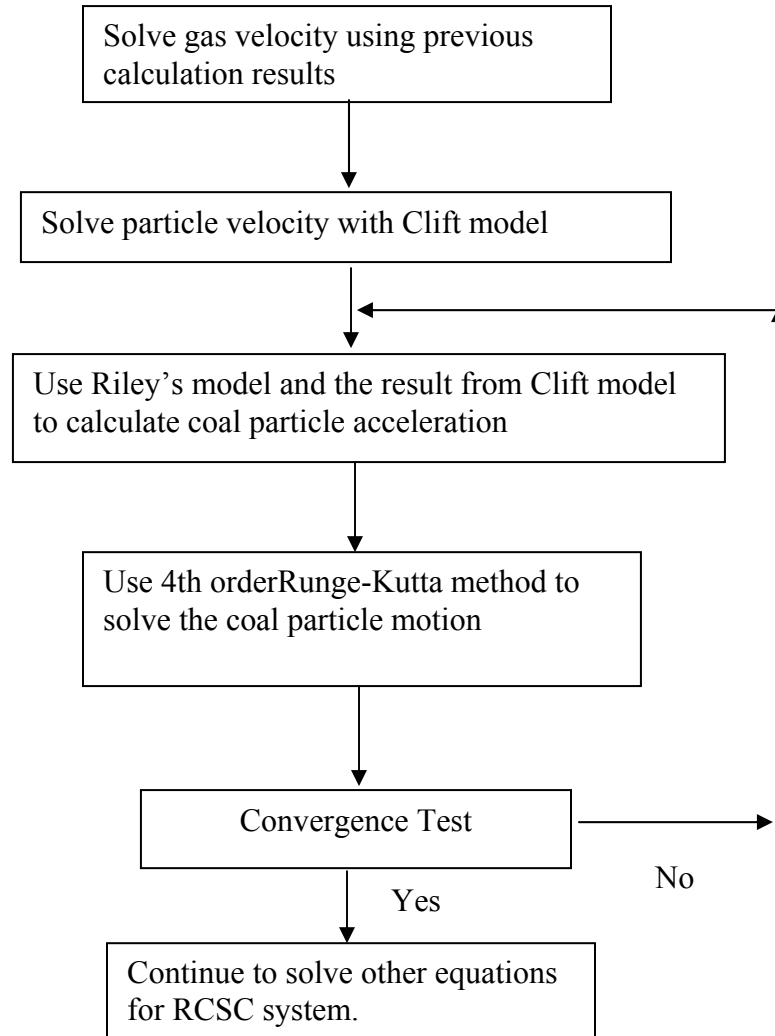


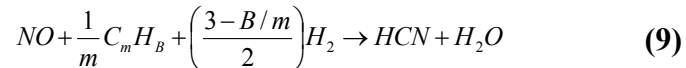
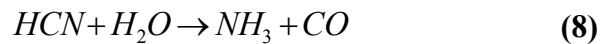
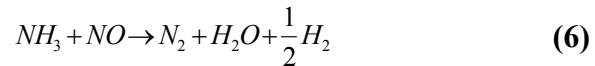
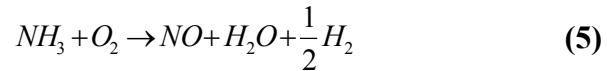
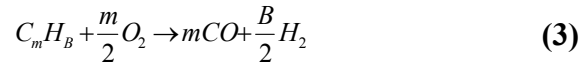
Fig. 5.5 The algorithm of coal particle motion computation in the RCSC model

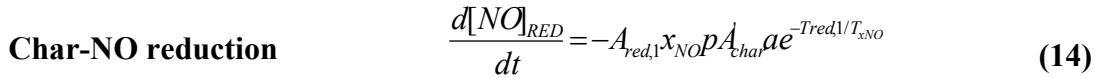
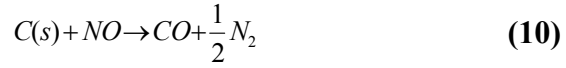
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- [7] Kim, “Conjugate heat transfer analysis in a radiatively/conductively-stabilized pulverized coal combustor”, 1994
- [8] David E. Stock, “Particle Dispersion in flowing gases – 1994 Freeman Scholar Lecture”, Journal of Fluids Engineering, Vol. 118, pp4—17 March 1996

5.3 The kinetic model for the chemical reactions, and the numerical algorithm for its solution

A 14-reaction model is used:





The kinetic equations (2), (5), (6), (8) were used to calculate the gas components reactions in combustor. Since fuel NO is the main source of NO formed in coal combustion processes, the reactions to reduce NO by (9),(10), (12), (13) and (14) were used in the computation model. Reaction (11) was removed because of its low reaction rate. Reaction (7) is added into the model.

6..RCSC CHARACTERISTICS

6.1 The coal properties

The following anthracite coal properties were used in computation,

Table 6.1

MOIST	VM	FIXED C	ASH	C	H	S	N	O	HEATING VALUE
%	%	%	%	%	%	%	%	%	kJ/kg
2.0	10.6	67.2	20.2	70.6	3.3	0.62	0.78	2.6	11925.

6.2 Transient analysis of flame stabilization (gas only) (Based on published papers by Kansuntisukmongkol et. al, 1997a, 1997b, 1998)

The RCSC, as stated earlier, results in virtually complete conversion of any fuel with virtually no formation of prompt NO_x and, if the flame front is near the exit, virtually no thermal NO_x. One unique characteristic of this burner with both favorable and

unfavorable aspects is its great thermal stability owing to the large thermal capacitance of the ceramic tube as compared to the flowing stream of air, fuel, and products of combustion. One favorable aspect is the dampening of all perturbations. One unfavorable consequence is the considerable period of time to establish a stable flame starting with a cold tube. Another is the time required to move the flame from one longitudinal location to another, which is essential in order to select the optimal stationary state among the as many as seven that are also unique characteristics of this burner.

These three papers represent a definitive study of the transient behavior associated with the thermal capacitance of the burner in order to guide its optimal operation.

6.3 Heat transfer and heat flow distribution in the RCSC wall

Since the RCS wall has a major role in the stabilization of the flame, the heat transfer and heat flow in RCSC were analyzed. The following table gives the equations we used to calculate the heat transfer in RCSC.

Table 6.2. Equations of Heat transfer in ceramic wall and insulations

Heat flow	Equations.
Incoming Energy	$Q_1 = E_{\text{air}} + E_{\text{coal}} = m_{\text{air}} C_{p,\text{air}} T_{\text{air}} + m_{\text{coal}} Q_{\text{coal}} + m_{\text{coal}} C_{p,\text{coal}} T_{\text{coal}}$
Outgoing Energy	$Q_2 = E_{\text{gas}} + E_{\text{char}} + E_{\text{ash}} + E_{\text{soot}} =$ $\sum_{CO, CO_2, N_2, \text{etc}} m_i C_{p,i} T + m_{\text{char}} C_{p,\text{char}} T_{\text{char}} + m_{\text{ash}} C_{p,\text{ash}} T_{\text{ash}} + m_{\text{soot}} C_{p,\text{soot}} T_{\text{soot}}$

Heat transfer along wall	$Q_3 = \int [q_{rad} + (-k_1 \frac{\partial T}{\partial r}) \times 2\pi R_1] dx$
Heat transfer at two ends	$Q_4 = \int_A q_{rad} dA$ (it also should equal $Q_1 - Q_2 - Q_3$) where $\nabla \cdot q_{rad} = \int_0^\infty \kappa_\lambda (4\pi I_{b,\lambda} - I_{o,\lambda}) d\lambda$ $\frac{\partial^2 I_o}{\partial r^2} + \frac{1}{r} \frac{\partial I_o}{\partial r} + \frac{\partial^2 I_o}{\partial z^2} = -3(1 - \omega_0)(\omega_0 g - 1) \tau_0^2 (4\pi I_b - I_0)$

Q₁: Incoming Energy with air/coal particle flow.

Q₂: Outgoing Energy with gas/ash flow.

Q₃: Heat transfer along wall

Q₄: Heat transfer at two ends by conduction, convection and radiation

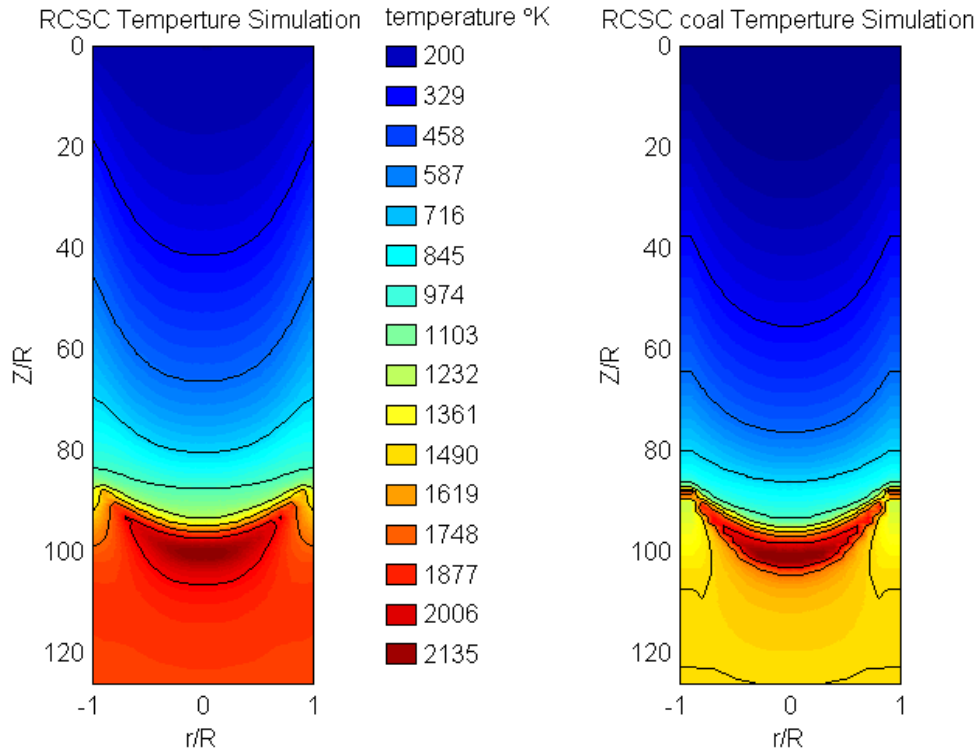
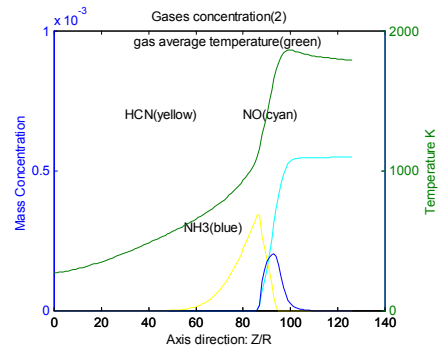
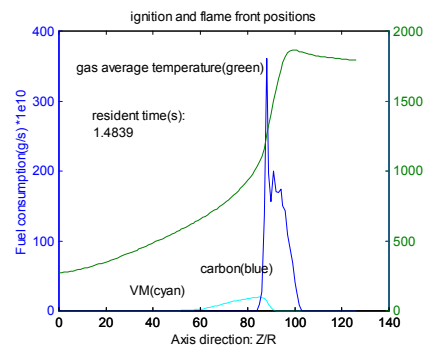
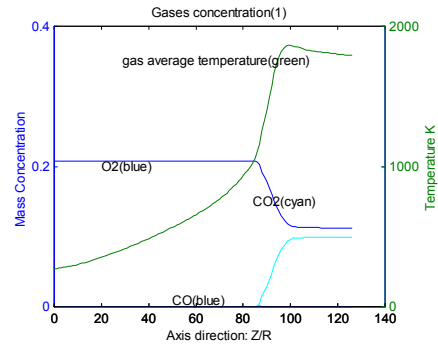
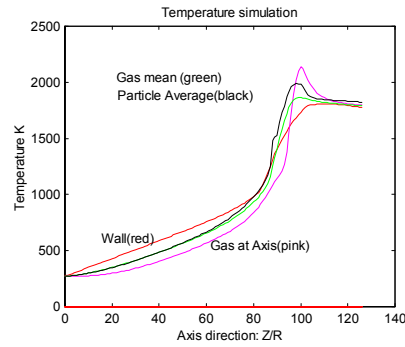
TABLE 6.3. HEAT FLOW IN THE INSULATION

Heat transfer from Gas to Ceramic wall	$Q_3 = \int [q_{rad} + (-k_1 \frac{\partial T}{\partial r}) \times 2\pi R_1] dx$ $q_3 = q_{rad} + (-k_1 \frac{\partial T}{\partial r}) \times 2\pi R_1 dx$
Heat transfer in Ceramic and insulation	$\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) + \frac{\partial^2 T}{\partial z^2} = 0$
Heat transfer from Ceramic wall to Fiberfrax	$Q = \int (-k_2 \frac{\partial T}{\partial r}) \times 2\pi R_2 dx$
Heat transfer to two ends from Ceramic wall	$Q = A \cdot h \cdot \Delta T$
Heat transfer from Fiberfrax to Fiberglass	$Q = \int (-k_3 \frac{\partial T}{\partial r}) \times 2\pi R_3 dx$
Heat transfer from Fiberfrax to air	$Q = A \cdot h \cdot \Delta T$

Heat transfer from Fiberglas to air	$Q = \int h \times \Delta T \times 2\pi R_4 dx$
-------------------------------------	---

One case of coal combustion in RCSC is analyzed and the heat flow in the wall and insulation material was calculated. Table 6.2 gives the heat flow calculation equations. The following Fig. 6.1 and calculations in Table 6.3 show the final result. It is found that less than 3% of heat flow passes the insulation material.

Such insulation is good for coal ignition and combustion. However, the thermal NO emission will be higher because of the high temperature in the RCSC combustor. We might need to transfer more heat from the RCSC to control the combustion temperature to reduce the final NO emission.



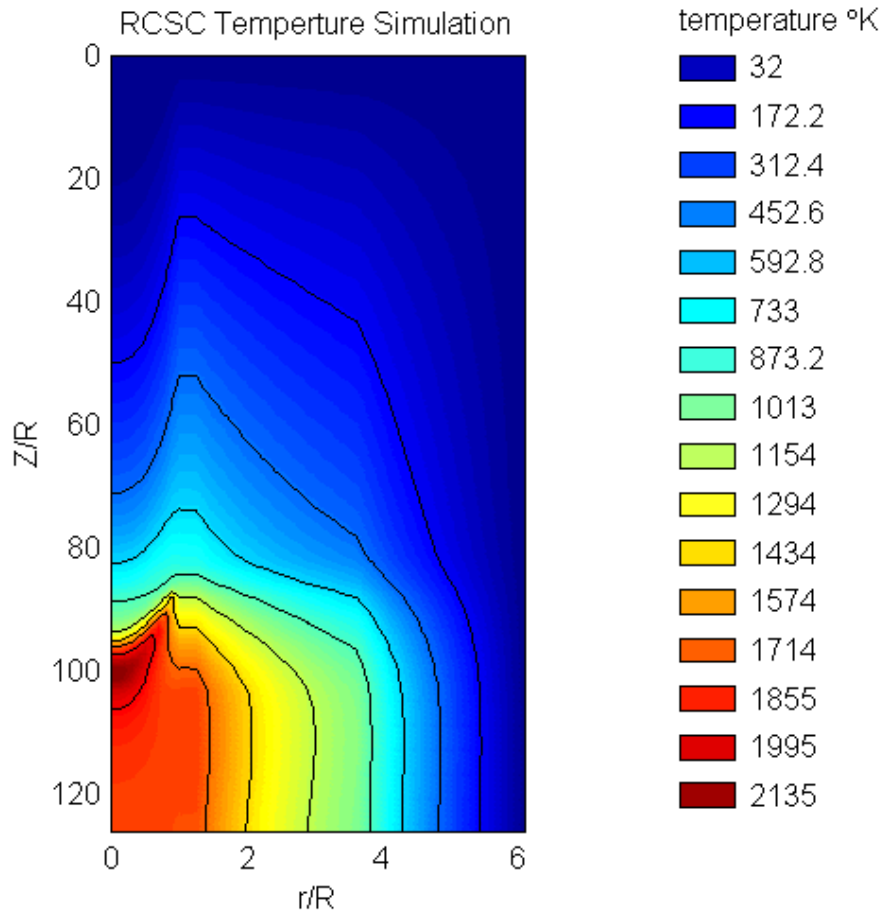


Fig. 6.1 The temperature and concentration profiles and the temperature distributions in the wall effect computations. Extra air ratio = 10%, Coal feed rate = 0.2g/s, Tube length = 10.5 feet, particle diameter $D=100\mu\text{m}$, Inlet Velocity = 1.652m/s, $K=2$ Btu/hr-ft-F, Inlet $T=\text{Ambient air } T=273\text{K}$

Table 6.3 Heat flow in the RCSC, kJ/h

Q_1	Radiation from gas to wall	2966.64
Q_2	Radiation from wall to gas	2879.0
Q_3	Net radiation between gas and wall	87.59
Q_4	Heat convection from gas to wall	-0.69
Q_5	Heat conduction from internal surface to wall	86.2
Q_6	Heat conduction from ceramic wall to insulation	14.2
Q_7	Heat convection from insulation I to air	7.46
Q_8	Heat flow into the RCSC(coal and air energy)	19971.5
Q_9	Heat flow out of the RCSC(coal, gas, ash, soot)	19382.4
Q_{10}	Radiation to outside of RCSC	83.98
Q_{11}	Heat convection from wall to air	7.66

7. NO_x PRODUCTION SENSITIVITY TO MAJOR PROCESS PARAMETERS

7.1 The Ammonia reaction

The NO concentration for the base case was found to peak at about 700 ppm at the axial location where the temperature peaked, and was reduced gradually towards the combustor exit, to about 600 ppm, primarily due to destruction by the NH₃ which was formed in the combustor. The results indicate that NH₃ injection into the flue gas along with the use of staged combustion or low-NO_x burner could lower NO_x emission level effectively.

7.2 Flame location

It is possible, mostly by flow manipulation, to position the flame stably nearly any place along the tube. The results of the sensitivity analysis show that flame location has little effect (~6%) and internal wall emittance an insignificant effect, on overall NO_x production.

7.3 Emittance of the combustor interior walls

While higher emittance helps stabilize the flame, the results of the sensitivity analysis show that it had an insignificant effect on overall NO_x production.

7.4 Excess air

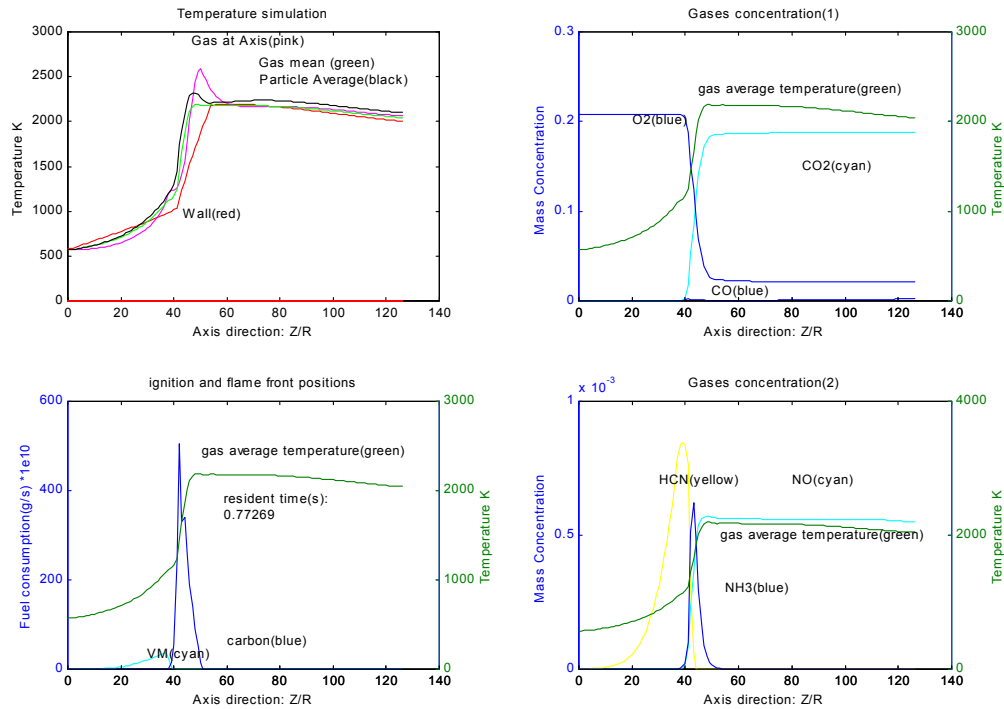
It was found that emitted NO mass fractions could be lowered by lowering the flame temperature, accomplished by increasing the excess air ratio. Nevertheless, the overall mass of the NO_x produced by the RCSC was still very significantly lower at the lower values of the excess air ratio which produce thinner flames of higher temperature. This

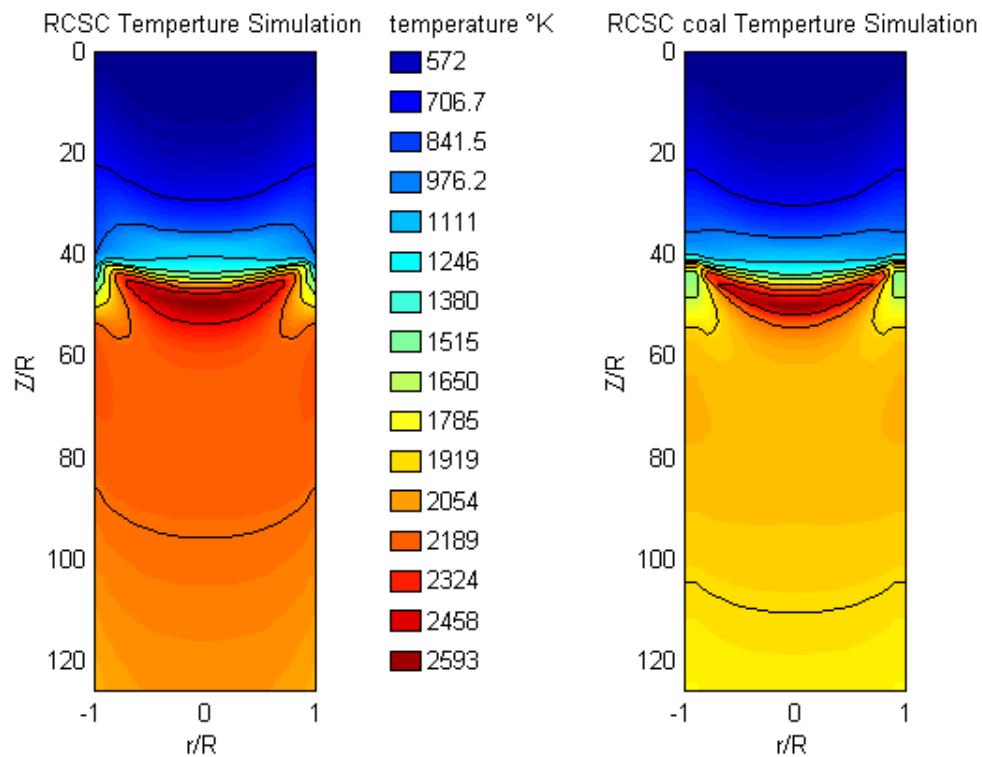
points towards the conclusion that the capability of the RCSC to produce such flames has a stronger effect on overall NO_x emissions reduction than the lowering of the flame temperature practiced in many contemporary NO_x reduction schemes.

7.5 Particle size

The coal particle size was varied between $90\mu\text{m}$ and $110\mu\text{m}$ in computational Case 1.

The results are shown in Figs 7.1 – 7.3 below.





RCSC System Temperature Profile

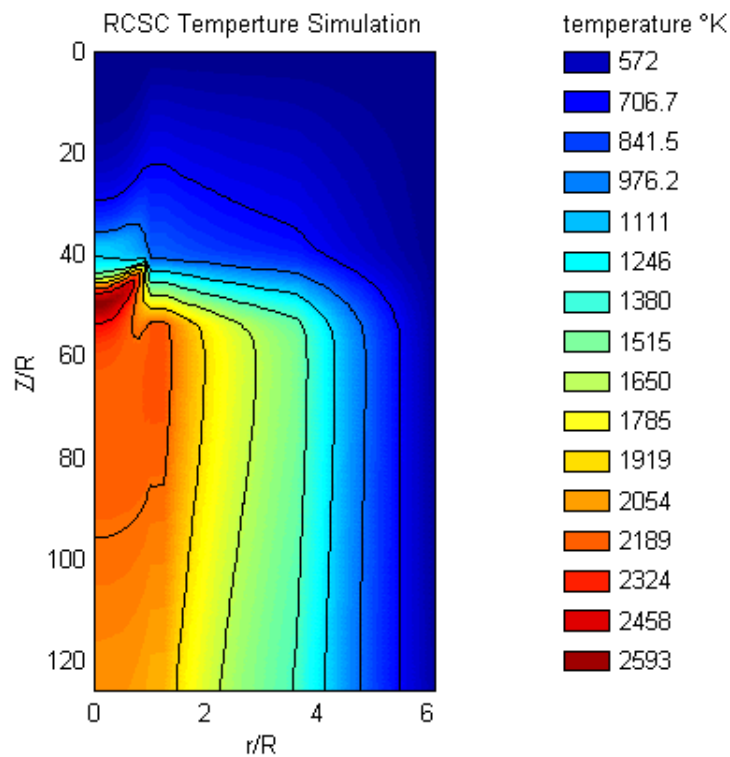


Fig. 7.1 Temperature and concentration distributions. Excess air ratio = 10%, Coal feed rate = 0.2g/s, Tube length = 10.5 feet particle Diameter=100 μ m, Re=1753, Inlet Velocity = 1.652m/s, K=2 Btu/hr-ft-F

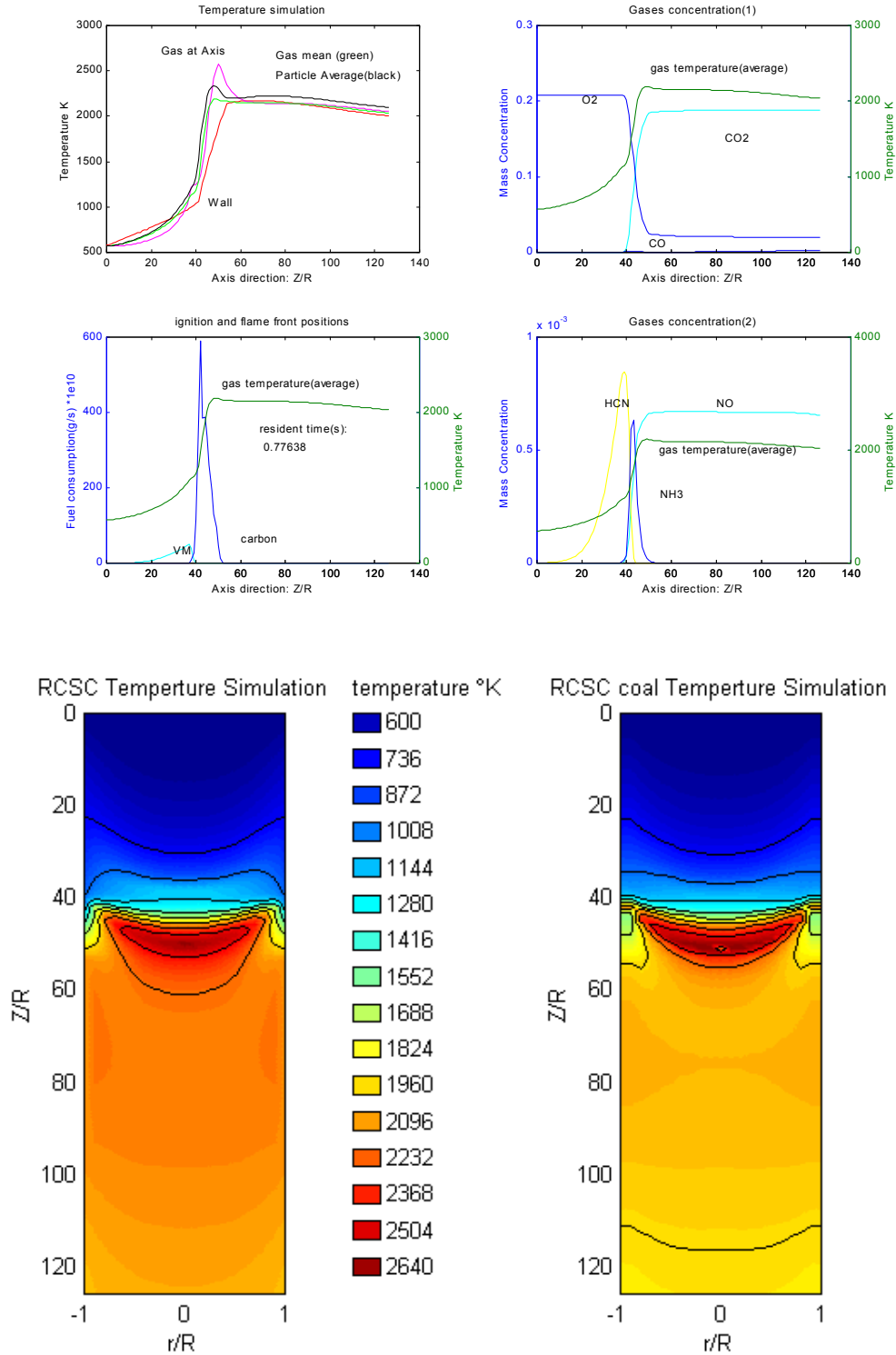
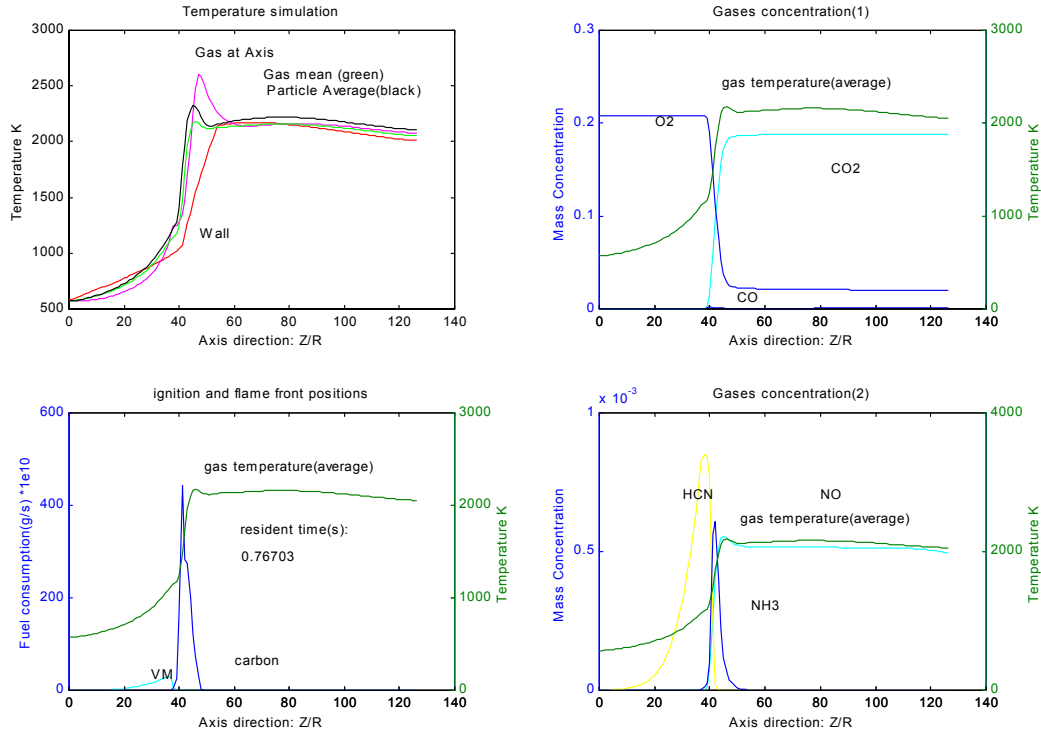


Fig. 7.2 Temperature and concentration distributions. Excess air ratio = 10%, Coal feed rate = 0.2g/s, Tube length = 10.5 feet Particle D=110 μ m, Re=1753, K_1 = 2 Btu/hr-ft-F, Ambient air =200K, inlet air=300K



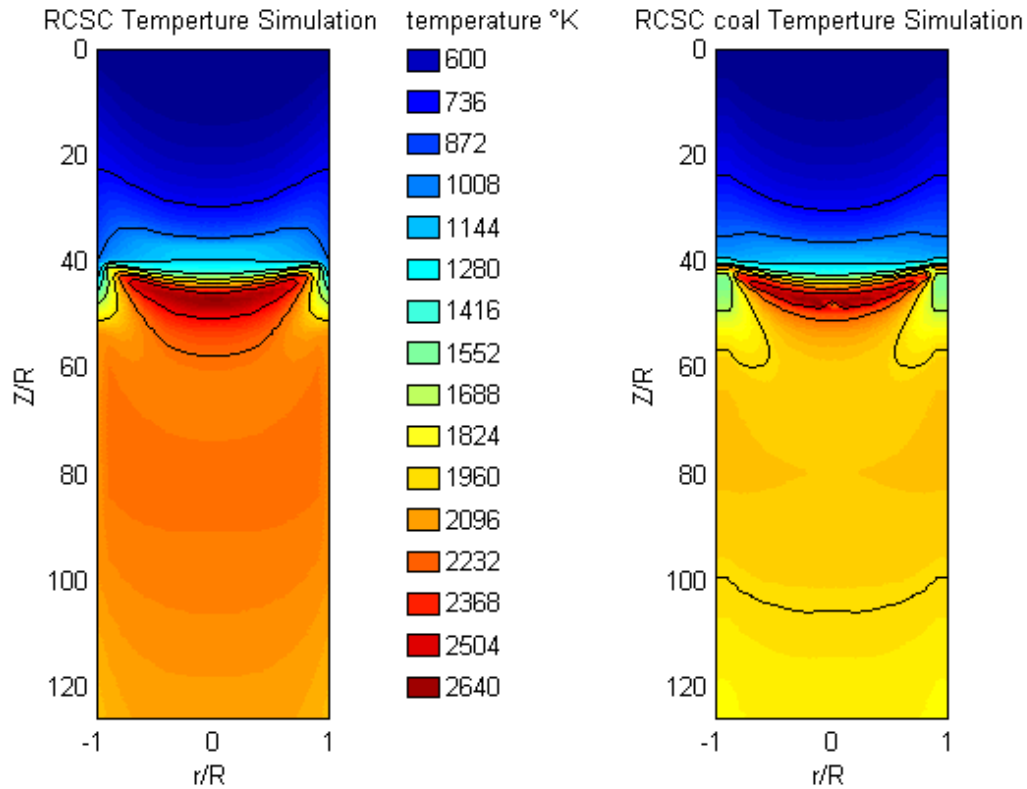


Fig. 7.3 Temperature and concentration distributions. Excess air ratio = 10%, Coal feed rate = 0.2g/s, Tube length = 10.5 feet Particle $D=90\mu\text{m}$, $Re=1753$, $K_1=2\text{ Btu/hr-ft-F}$, Ambient air = 200K, inlet air = 300K

Compared with case 1 where $D=100\text{ }\mu\text{m}$, it is noticed that when we use $90\mu\text{m}$ coal particle in combustion, the NO concentration at exit is 510ppm, 8% lower than the base case of $100\mu\text{m}$ (550ppm) and 40% lower than the case when we use $110\mu\text{m}$ coal particle. It indicates that NO emissions are reduced as the coal particle size is reduced.

Some other interesting conclusions from this analysis are (1) that direction of the coal particles towards the wall improves the heat transfer by convection between the wall and flow, and the particles would this be preheated and ignited in a shorter time and travel distance, thus also helping stabilize the flame, especially during ignition, (2) in particular,

we found that control of upstream heat transfer and particle concentration may be better for flame stabilization than manipulation of the flame position, as we have done so far, and (3) an explanation of the experimentally observed slagging near the combustor exit: based on our computations, the downstream wall temperature can reach the hemispherical and fluid temperature of ash. The ash particles there will therefore deform and melt, and, as observed, stick to the wall. Because of high particle concentration near the wall, slagging develops quickly. This has also allowed a proposal for a modified configuration of the RCSC that would reduce slagging.

7.6 Inlet temperature effects

The inlet temperature was varied from 573K to 973K and the results are shown in Figs 7.4 – 7.6.

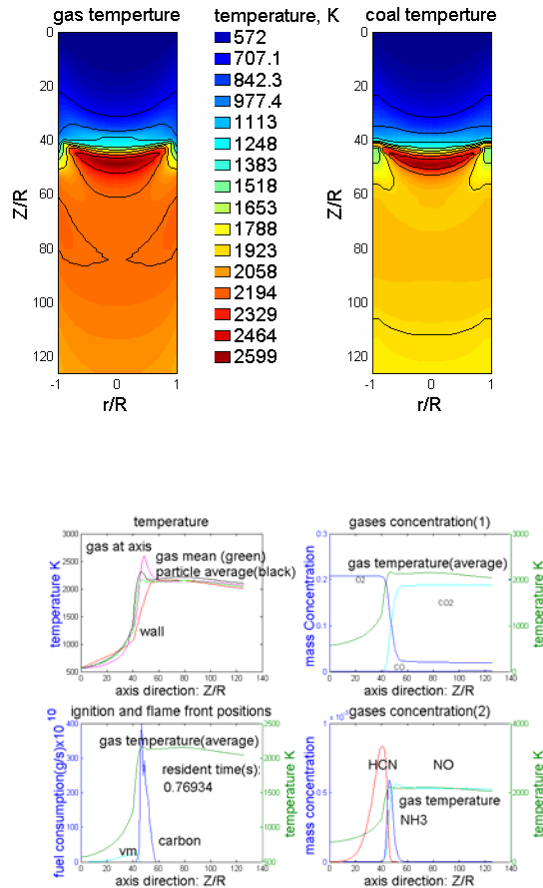


Fig. 7.4 Temperature and concentration distributions. Excess air ratio = 10%, Coal feed rate = 0.2g/s, Tube length = 10.5 feet, Coal particle Diameter=100 μ m, Inlet Velocity = 1.652, **Inlet air=573K**

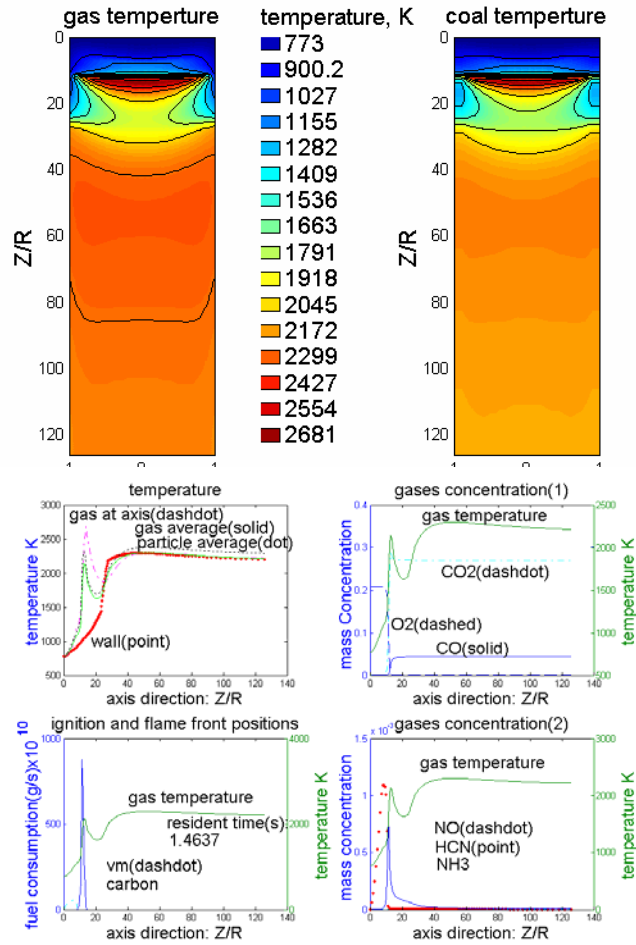


Fig. 7.5 Temperature and concentration distributions. Excess air ratio = 10%, Coal feed rate = 0.2g/s, Tube length = 10.5 feet, Coal particle Diameter=100 μ m, Inlet Velocity = 1.652 m/s, **Inlet air=773K**

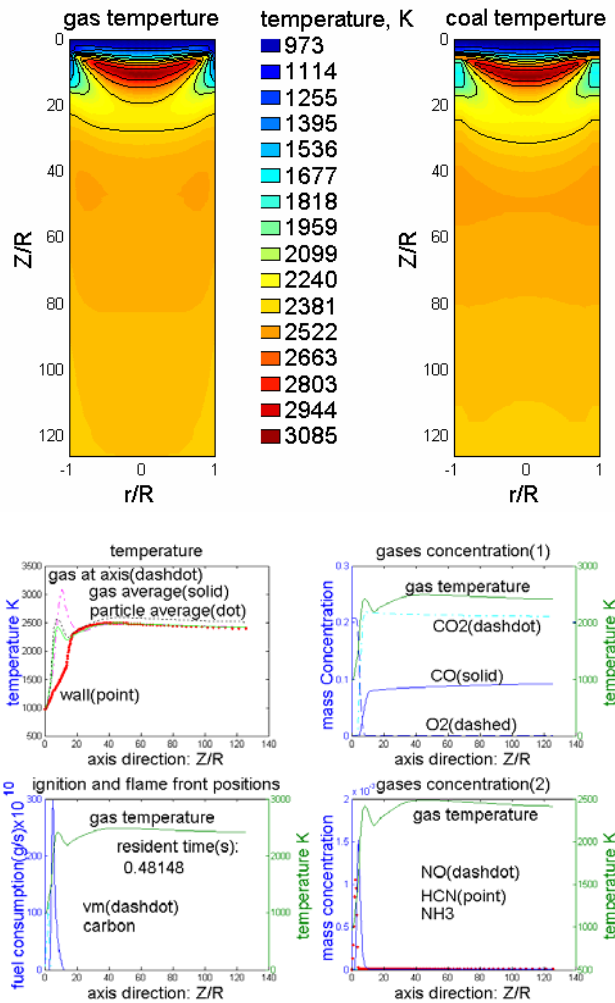


Fig. 17.6 Temperature and concentration distributions. Excess air ratio = 10%, Coal feed rate = 0.2g/s, Tube length = 10.5 feet, Coal particle Diameter=100 μ m, Inlet Velocity = 1.652 m/s, **Inlet air=973K**

The results indicate a significant reduction of NO_x production at the higher inlet temperatures. While this prediction requires experimental verification, it seems to be rational based on the following facts. Higher incoming air temperature (both primary and secondary air) require less heat transferred from the combustor to reach the ignition temperature. Quick ignition can keep the flame front stable by control of primary air flow rate. Because they lower the gas density, higher air temperature causes higher average air

velocity for the same coal feed rate. The viscosity increases with temperature, which in turn reduces the Reynolds number. The change of the flow from turbulent to laminar, tends to reduce NO_x emission due to the lower extent of mixing and shorter residence time.

Higher incoming air temperature is also helpful for shortening the flame zone. Since the heating time and length are shortened, the flame front temperature is increased and thus the coal particle burns quicker there. A higher temperature and thinner flame zone could be obtained by this way.

There is, however, a limit for increasing the input air temperature, since the mixture might ignite too early, and also because the excessive temperatures may damage the RCSC combustor, especially when high volatile material content coals are used, such as bituminous coal and lignite.

7.7 Preheat by a heat-recirculating burner, and decomposition of NO_x using a double-spiral catalytic reactor (Shinoda, Maihara, Kobayashi, Arai, and Churchill 1998, Doi, Shinoda, Matsunami, Arai and Churchill 2001, Beaver et al. 2001)

The concept of the RCSC has other applications, such as a gas preheater to allow combustion of low heat value gases, and for spiral catalytic reactors. A recirculating RCSC configuration, where some or all of the reaction products are circulated annularly along it, can indeed be used to provide the preheat of the input air which was found, as discussed above, to significantly reduce the production of NO_x. These systems were examined, but only with fuel gas, in above above-referenced publications. The spiral

heat-exchanger developed by Retallick, Strenger, Targett and Churchill as a catalytic incineration of pollutants in the air of living and working spaces may be used to convert N_2O , which is becoming recognized as a serious contaminant of the upper atmosphere, and which is emitted from the operating rooms of hospitals, automobile catalytic converters, agricultural fertilizers, etc. back to O_2 and N_2 .

8. THE RCSC COMBUSTOR: DETAILS OF DESIGN, OPERATION, AND BASIC RESULTS

8.1 Introduction

Experiments were conducted, mostly before 1997, for model validation and to determine appropriate ranges of experimental parameters, with an ultimate objective of designing a commercial combustor based on this principle. Special attention was given to pulverized coal flame stabilization in this combustor.

8.2 The Combustion Facility

The combustion facility consists of an air/gas delivery system, a coal feeding system, a ceramic combustor tube, a cooling unit, and a data acquisition system. A schematic diagram of the system is shown in Fig. 8.1. To obtain the high temperature condition for coal ignition, preheating is required. The preheating consists of two steps. First, a preheater which burns city gas is inserted into the combustor tube from the bottom through the base opening. Although there is a special opening installed just below the cooling unit, which was originally designed to be used for preheater insertion, the much wider opening in the base is used because of easy access. The heating is continued until the highest wall temperature reaches to about 800 K, which is high enough to establish a stable ethane flame: since heating

by city gas alone could not raise the wall temperature sufficiently high to burn coal, additional heating was necessary, and ethane was used as a secondary energy source. It was supplied through the top of the combustor after it was mixed with air in the mixing chamber.

Fig. 8.1 shows major components of the combustion facility.

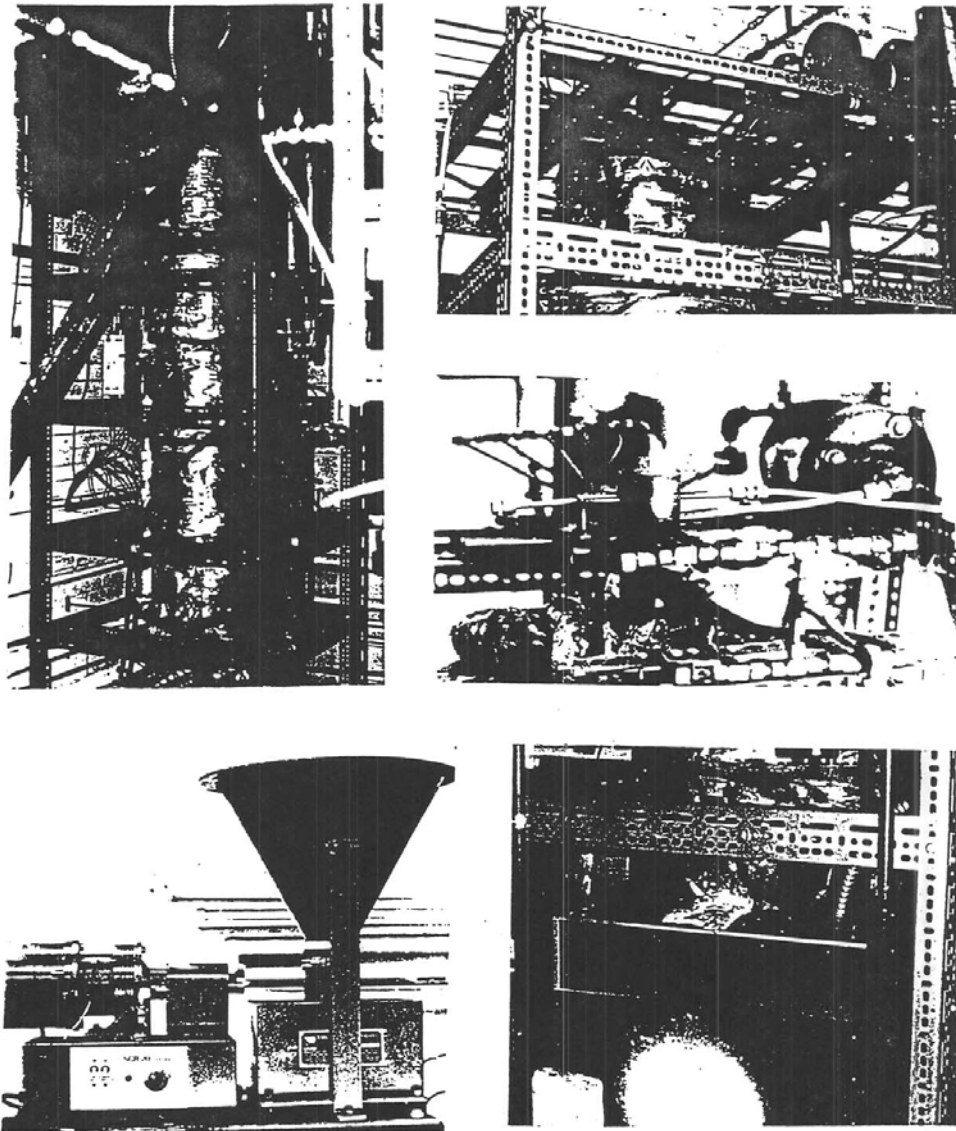


Fig. 8.1 Photographs of the combustor and coal feeder; from top to left, clockwise: upper section, combustor top, bottom section, and coal feeder.

8.2.1 The Combustor and the Temperature Instrumentation

A schematic of the combustor is shown in Fig. 8.2. The combustor tube is a 7-foot-long, 2 1/4-inch OD ceramic tube, with a 1/8-inch-thick wall (McDanel Refractory Type MV30, 85% Mullite and 15% glass with maximum working temperature 2025 K). It is surmounted by a stainless steel cone (Fig. 8.3) for distributing the coal-air mixture, followed downstream by a two-row, horizontal cooling tube bank to prevent flashback. As seen in Fig. 8.3, the coal is supplied upwards to the top of the cone in the primary air stream, with secondary air supplied concentrically just below but through a jet directed upward against the primary stream flowing downward to enhance mixing.

Seventeen type-S thermocouples (Pt/Pt-10%Rd), spring-loaded into 1/16-inch-deep indentations along the length of the tube are used to measure the mid-wall temperature. In the previous configuration, more thermocouples were located near the top of the combustor. As a result, the spacing of the thermocouples near the bottom half of the combustor was greater than at the top, yielding less accurate temperature profile in the flame region. To increase the resolution of the data reading in the flame region, therefore, more thermocouples were installed in the middle section of the tube in the new tube design.

Two kinds of insulation, surrounding the ceramic tube, were used to minimize the heat loss through the wall (Fig. 8.2). The inner layer of the insulation is high-temperature Fiberfrax (Carborundum Co., $k=0.116$ W/m-K) and the outer layer is lower-temperature Fiberglas (Carborundum Co., $k=0.034$ W/m-K). Although it has a lower thermal conductivity, Fiberglas cannot stand temperature beyond 600 K. Each layer is 6.35 cm thick.

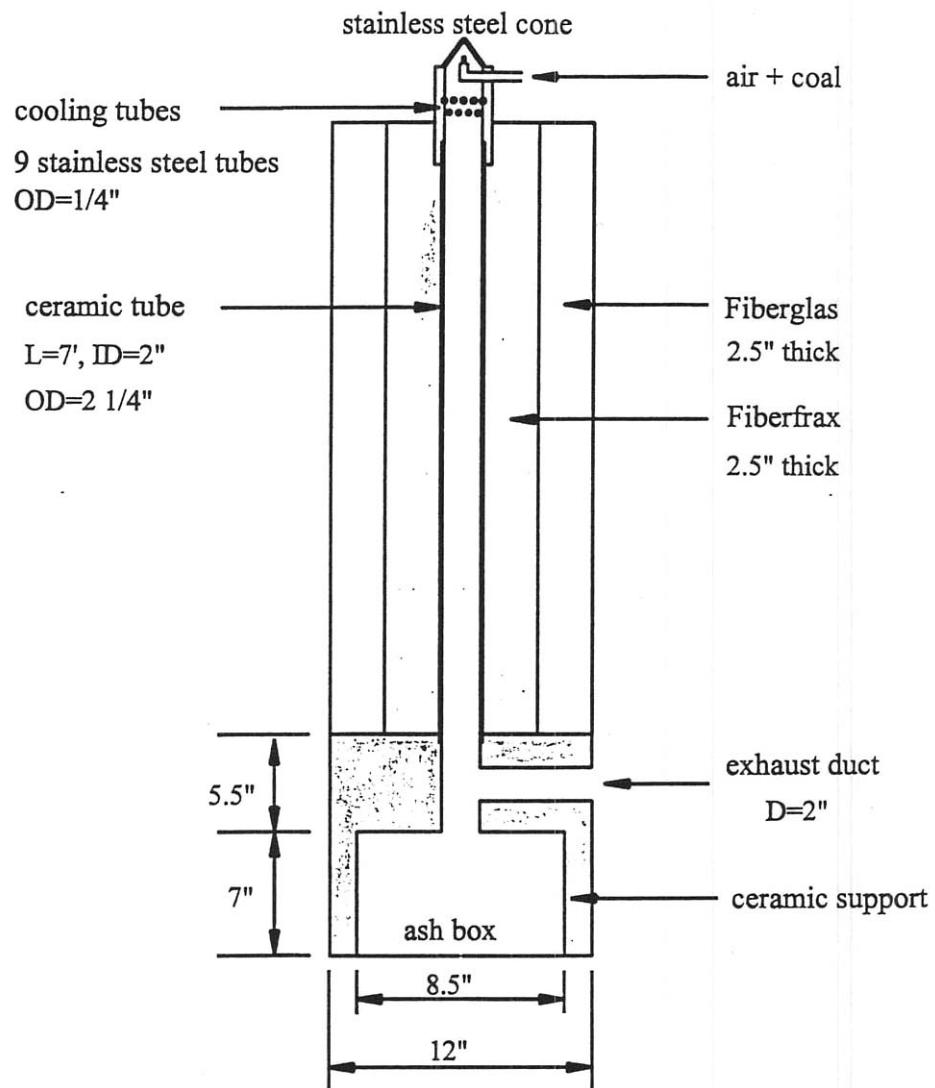


Fig. 8.2 The experimental RCSC

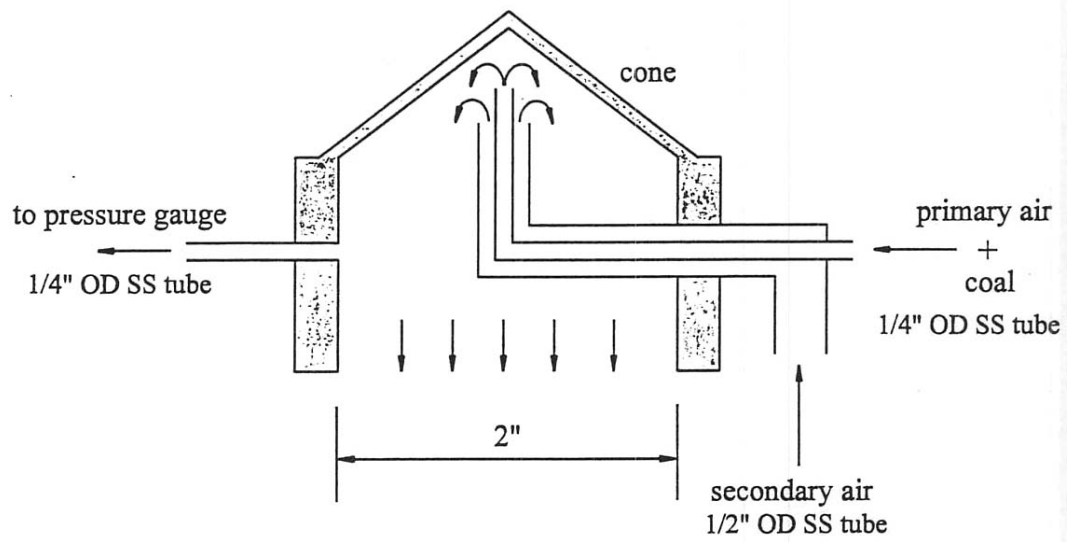


Fig. 8.3: The stainless steel top cone designed to improve coal/air mixing and downflow uniformity

8.2.2 The Preheater

Fig. 8.4 shows the preheater used for heating the combustor before ethane is supplied. The preheater consists of a welding blow pipe, a 60-inch-long flexible stainless steel hose (1/4 in. in diameter), and a tip made of 1/4-inch stainless steel tube. The annular corrugations of the hose are protected by a single braided stainless steel wire sheath and can be used at temperature as high as 1100 K. One end of the hose is attached to the outlet of the blow pipe which controls the air and city gas flow. To make preheating efficient, a stainless steel tip is attached to the other end of the hose. The 8-inch-long preheater tip was designed to make the gas flame stretch toward the combustor tube wall in four directions, not the same direction as the tube length. This tip design is believed to heat the combustor tube to a required temperature level more effectively and quickly. A short heating time is crucial for the protection of the preheater. The gas delivering tube deteriorates from the heat and eventually melts if it is repetitively exposed to such a high temperature environment for a long period of time.

8.2.3 Air/Gas Delivery System

Air is used in this experiment for two purposes: to burn fuel and to carry pulverized

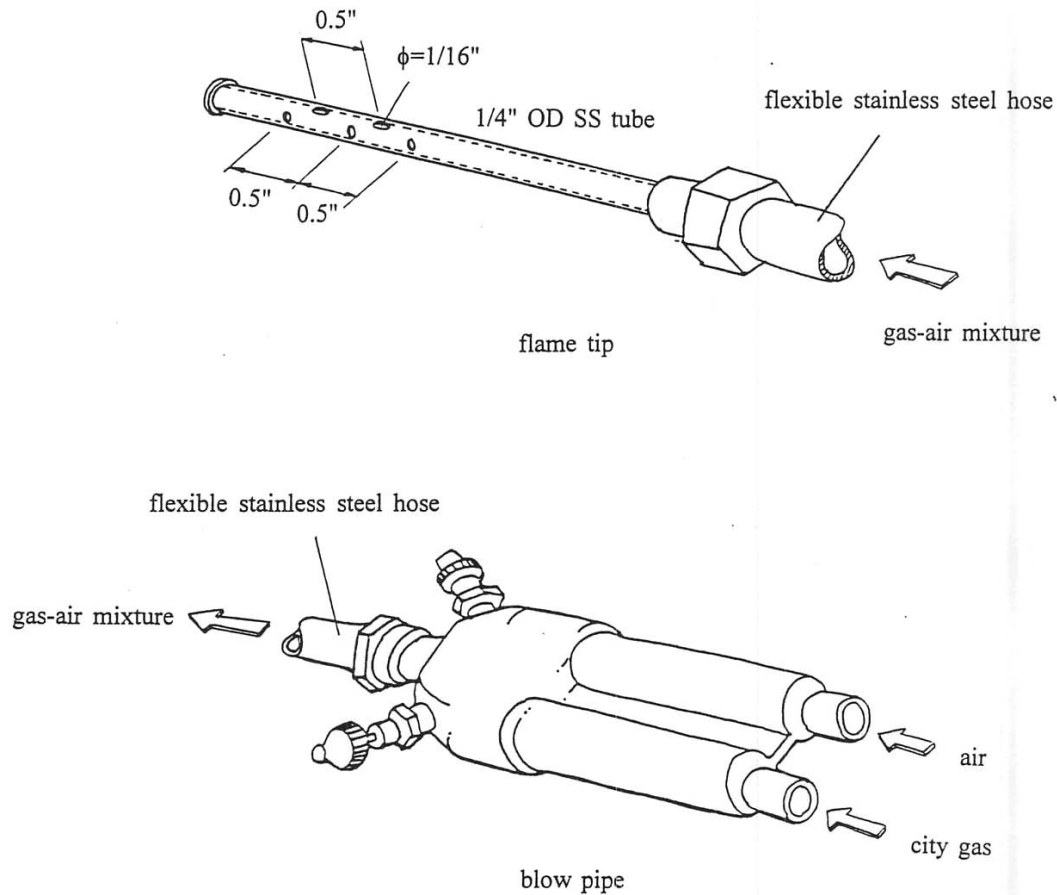


Fig. 8.4 The preheater: stainless steel flame tip (top) and blow pipe (bottom)

coal from the feeder to the combustor. Compressed air at 120 psig is used after reducing its pressure to about 20-40 psig. To remove dirt and moisture from the air, a filter unit and a dehumidifier (Ingersoll-Rand Model I Air Dryer, max. operating pressure 300 psig, capacity 5 CFM at 100 psig) were installed in the air path. The flow rate is controlled by two rotameters: Fisher and Porter serial no. 8206A9270R2 (max. rate 19.5 SCFM, for the primary air) and model no. 10A1755A, serial no. 8403A0411A1 (max. rate 2.44 SCFM, for the secondary air), respectively.

Grade 2 ethane is supplied from a 200 size cylinder (32 lbs, max. pressure 543 psig),

after the pressure is reduced to 20 psig by a pressure regulator (Matheson model no. 8-350). The flow rate is measured by a rotameter (Fisher and Porter model no. 10A2735A, serial no. 6803A4351A1, max. rate 1.1 SCFM). Before they are supplied to the combustor, the ethane and air are mixed together in a chamber filled with ceramic packing (Fig. 8.5). This mixing chamber also serves as a secondary quencher in the case of cooling unit failure.

8.2.4 Coal Feeding System

The coal supply system was designed to transport coal particles from the hopper to the combustor. It consists of a 1.5 cubic foot conical hopper and a screw feeder (Vibra Screw Model SCR-20), and a venture compressor. Pulverized coal poured in the hopper is delivered to the discharge tube by the vibrating screw feeder and picked up by the primary air at the venturi. The coal is then carried into the combustor.

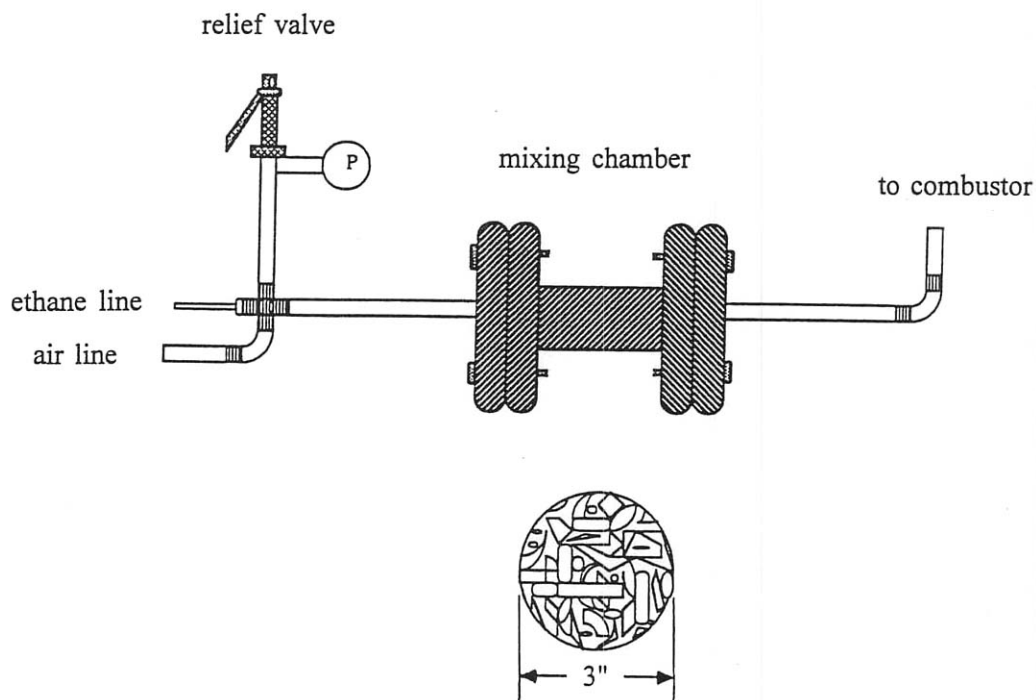


Fig. 8.5 The mixing chamber (top) and its cross section with the ceramic particles filling.

The feeder is basically a screw conveyor with feed rate controlled by changing the rpm of the screw. The base of the feeder vibrates at a certain frequency to facilitate feeding from the hopper to the screw. The feed rate can be varied by controlling the speed potentiometer in the range of 0.05-0.5 g/s. The level of coal in the hopper is recommended by the manufacturer to be kept within the range of 1/3 full to full.

8.2.5 Cooling Unit

To prevent a flashback near the entrance of the combustor, a cooling unit was installed just below the fuel/air nozzle which discharges coal and air into the combustor. The cooling unit, as shown in Fig. 8.6, consists of nine stainless steel tubes (1/4 in. in diameter), and cool tap water flows through the tubes. This also ensures that the entrance region remains sufficiently cool to prevent premature ignition of fuel. To prevent coal accumulation on the cooling tubes, a triangular stainless steel roof is welded on each tube.

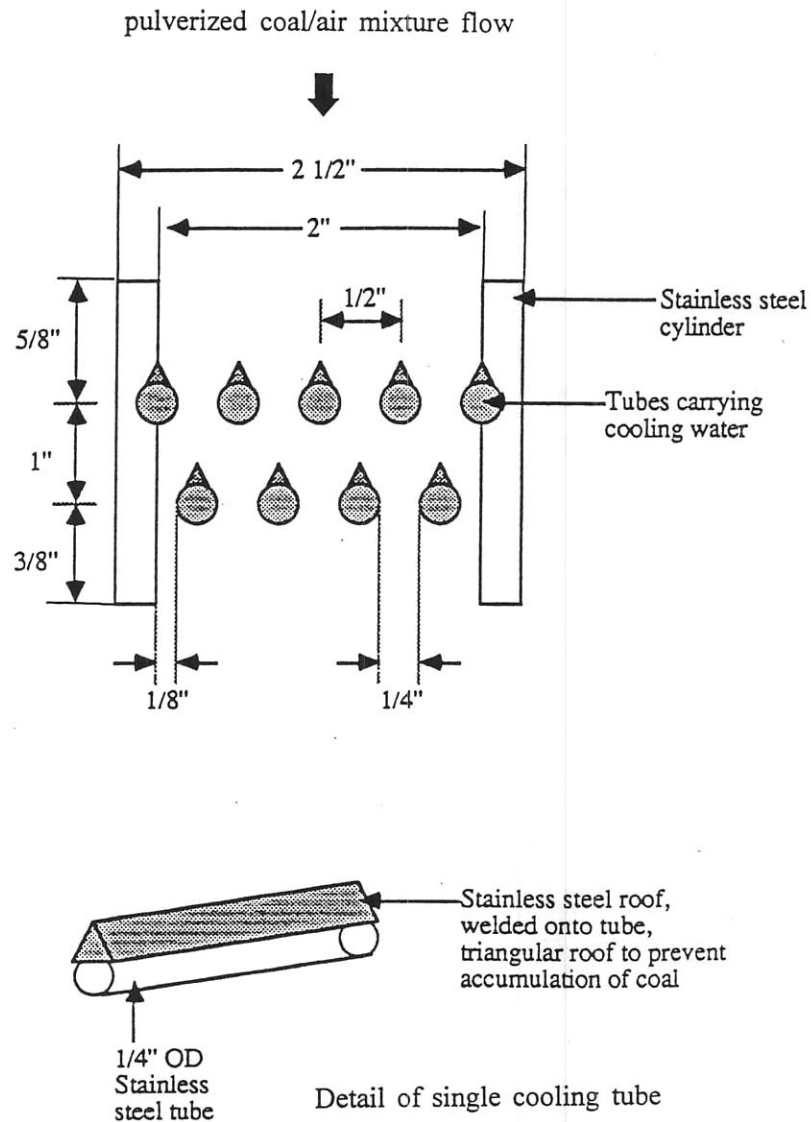


Fig. 8.6 Cross section of the cooling unit (top) and detail of a single tube in the unit (bottom)

8.2.6 Data Acquisition System

To monitor the wall temperature distribution, a computer-controlled data acquisition system, the last version of which is described in Section 9 below, was used. The acquisition unit contains boards to measure all of the thermocouples and gas composition analyzers. The recorded data is automatically analyzed and displayed with a computer program developed for that purpose.

8.3 Experimental Procedure

8.3.1 Startup

1. Start the data acquisition program.
2. Open the master valve to the air line located before the inlet of the air dryer and set the pressure regulator to 60 psig.
3. Turn on the air dryer.
4. Set the air pressures to 5 psig (primary air used for burning city gas) and 40 psig (secondary air), respectively.
5. Open the valve to the secondary air slightly to allow a very small amount of air to flow. This downward air stream keeps the upper portion of the combustor tube cool by preventing natural convection when city gas is burned.
6. Turn on the water flow to the cooling unit.
7. Turn on the switch of the fan installed at the outlet of the chimney.

8.3.2 Preheating

1. Open the city gas valve in the blow pipe to its maximum and ignite it using a striker.
2. Insert the preheater into the combustor tube from the bottom as far as possible. This will locate the flame around 40 in. from the tube inlet.
3. Adjust the air flow rate very carefully to obtain a blue flame, observing the flame color using a monitor mirror through the base opening.
4. Continue the heating until the highest temperature reaches to about 800 K.
5. While the tube is under heating, open the ethane tank and adjust the pressure to 20 psig. Keep the valve downstream of the rotameter closed.
6. When the highest wall temperature reaches to about 800 K, first increase the secondary air flow rate to a rotameter reading of 30, and then open the ethane valve

to a rotameter reading of 20.

As ethane ignites, a bursting sound will be heard. If the air flow rate is not high enough or ethane is supplied first, the ethane ignites and the flame propagates backward. Then the flame attaches to the cooling tube bank, producing a very dangerous situation. If this happens, shut the ethane valve immediately while keeping the air flow to its maximum. After the flame is extinguished, open again the ethane valve slowly. If the fuel/air ratio is not proper, i.e., too much or too little air, the flame will oscillate. Carefully adjust the flow rate.

7. Once a stable ethane flame is established, remove the preheater from the combustor. Keep the air flow through the preheater until it is cooled down sufficiently.
8. Increasing the air flow causes the flame move downward, and vice versa. Considering the movement of the flame, locate the flame front to the desired position by slowly controlling air and ethane flow rate, monitoring the wall temperature distribution.
9. Continue to burn ethane until the highest wall temperature reaches to 1200 K or higher temperature.

8.3.3 Coal Combustion and Flame Stabilization

1. Determine the primary air flow rate and coal feed rate.
2. Set the primary air pressure to 40 psig.
3. Set the coal feeder controller knob to the predetermined scale.
4. Pour pulverized coal into the hopper of the feeder.
5. Open the primary air valve and turn on the coal feeder motor.
6. Close the ethane valve slightly and burn both fuel simultaneously to avoid flame extinction.

7. Adjust the air flow rate (both the primary and secondary air) slowly to the target condition.
8. Close the ethane valve completely.
9. Monitor the temperature distribution to determine when the combustion is stabilized.
10. Once a stable flame is obtained, record operating conditions and prepare for another run.

8.3.4 System Shutdown

1. Turn off the coal feeder motor.
2. Close the air line after the tube cools down to about 500 K.
3. Turn off the exhaust fan.
4. Shut down the cooling water flow and all gas line.
5. Turn off the computer and data acquisition unit.

8.4 Results and Discussion

8.4.1 Thermocouple Construction and Calibration

Thermocouples were made from bare wires (Pt and Pt+10% Rd wires, Engelhard Corp.). The wires were welded together to make a junction and the tip was visually inspected by using a magnifier. The wires were then put into a double-bore ceramic tube, and a male connector (Type NMP miniature thermocouple connector, Omega Engineering Inc.) was attached to the end of the wires. After the thermocouples were assembled, each of them was calibrated in an electric crucible at three different temperatures, using a standard NIST-calibrated type-K thermocouple as reference before they were installed on the combustor tube. The temperature measurement error was about 1%.

8.4.2 Coal Preparation

Anthracite coal (VM 13.0%, FC 70.6%, ash 16.4%) obtained from PP&L (Philadelphia Power and Light Co.) was used in the experiments. A 6" screen mill (Raymond Laboratory Mill, A division of Combustion Engineering Inc.) was used for pulverizing the coal. The mill works on a rotating hammer principle and the resulting particle size can be varied by changing the screen. The pulverized coal was then sized in a sieve-shaker. Only coal particles in the desired size range were collected.

The pulverized coal sample was then dried to eliminate moisture after sieving. From preliminary experiments, it was found that the moisture contained in the coal powder caused feed-related problems. The moisture in the coal depreciates liquidity of the coal powder and thus prevents smooth feeding. The coal sample which was put in a metal pan was heated for 20 to 30 minutes while the powder was being stirred. To prevent devolatilization, the temperature of the coal was kept below 340 K.

8.4.3 Coal Particle Size Distribution

A particle size analyzer (Microscan II by Quantachrome) in the LRSM (Laboratory for Research on Structure of Matter, University of Pennsylvania) was used to determine the coal particle size distribution. The analyzer consists of an x-ray emitter, a sample cell, an x-ray detector, and a circulation system. A low-energy x-ray beam discharged from the emitter passes through the sample cell, where the beam is attenuated by suspended particles, and the weakened intensity of the x-ray beam is measured by the detector. Since the attenuation of the x-ray beam is proportional to the concentration and size of particles, size distribution can be determined by measuring the difference of the x-ray intensity.

To place coal particles in the path of the x-ray, the particles are mixed with distilled water in the phase separator and then circulated by the pump through the cell. 4 or 5 drops of

dispersant (triton x-100 or aqueous calgon) are added to the coal-water mixture to prevent particle sedimentation. The attenuation of the x-ray flux is less than 0.1% when only water is circulated. As particle concentration increases, however, naturally the amount of attenuation also increases. To obtain the best results from the analysis, coal should be added to the water continuously and well stirred until the attenuation level reaches to about 15%. The size distribution data of the pulverized coal sample used in the experiment, including the modal and mean sizes, as well the size distribution parameter α .

8.4.4 Preheating

Before feeding pulverized coal into the combustor, the flame front must be placed at a desired location by varying air and ethane flow rate during the preheating period. The location of the flame can be determined by monitoring the wall temperature distribution which is displayed on the computer screen. Considering the operating conditions and the length of the combustor tube, it is preferable to place the flame front as high as possible to provide a coal particle longer residence time in the combustor. The attachment of a flame tip which is made of 1/4-inch stainless steel tube to the metal hose was found to be very effective for fast heating. In less than 30 minutes of burning city gas, the peak wall temperature reaches about 800 K, providing a sufficient condition for burning ethane gas. Besides, the metal tip has several advantages over the originally designed ceramic tip. It is durable, easy to make, and, when it deteriorates from continuous use, it can be replaced with a new tip readily.

After igniting the city gas, the preheater was inserted into the combustor tube from the bottom. The air valve on the blow pipe (green color) was kept closed until the preheater was completely inserted into the combustor. Otherwise the flame would be easily

extinguished since the pressure of the city gas is low (about 2 psig) compared to that of the air (5 psig). Before inserting the preheater, the other opening in the base to the exhaust system was closed to protect the flame from extinction. The air valve was then open slowly to obtain a blue flame, monitoring the color of the flame using a mirror. As the tube inside became warm, the location of the flame was changed according to an experimental plan by slowly moving the tip up or down. In most cases the tip was located around the middle of the tube to place the flame as high as possible with the current preheater design. to obtain a blue flame, monitoring the color of the flame using a mirror.

Ethane was used as a secondary energy source for the preheating, since burning the city gas (mostly methane) alone was not sufficient to preheat the combustor for coal combustion. Air was supplied through the secondary air line to burn ethane by opening the valve wider, which was already open to prevent convective heating of the upper portion of the tube (Step 5 of the startup procedure). Although opening the ethane valve and increasing the air flow are almost simultaneous, the air valve should be open first, immediately followed by the opening of the ethane valve. If the air flow rate is too low or ethane is supplied first, the flame tends to travel backward after it is formed in the high wall temperature region. The flame then attaches to the cooling tube bank and the temperature of the stainless steel cone increases rapidly to a dangerous level. Once this happens, it is very difficult to control combustion. When the air flow was increased to detach the flame from the cooling tubes, the flame front was observed to oscillate up and down inside the combustor tube, and eventually the flame became extinguished (Experiment #1 and #2 in Table 8.1).

Figure 8.7 shows the wall temperature profiles during the preheating period. When city gas was burned alone, the wall temperature rose up to 800 K in a very narrow region while the other portion of the tube still remained cool. This is obviously due to the newly attached flame tip which directs the flame toward the tube surface perpendicularly. The low

thermal conductivity of the ceramic tube causes very low conductive heat transfer through the wall, keeping the temperature of other parts of the wall low. When ethane gas was burned for about an hour, peak wall temperature reached higher about 1200 K to 1300 K. Although the lower part of the tube was heated by hot gas stream, the wall temperature decreased by 400 K, from 1200 K to 800 K, which may be not sufficient for coal particles to complete combustion. In an effort to raise the temperature of the lower section and make the profile nearly uniform, the amount of the ethane flow was increased by up to 50%. The flame, however, either drifted downward and reduced the length of the hot tube section or became very unstable. From experience it was found that the ethane flame was easily stabilized at the flow rate of 0.225 SCFM and this rate was used in the experiments.

8.4.5 Flame Stabilization and Visual Observation

Unlike burning gaseous fuel, establishing a stabilized flame in this combustor requires special attention in preheating when coal is used as fuel. Since there is no backmixing of combustion products and reactants in the combustor, it is very important to provide a high temperature environment for the coal particles to complete all stages of the combustion processes, such as heatup, devolatilization, ignition, and char combustion. As seen in Fig. 8.7, however, the wall temperature in the region downstream the flame front becomes as low as 800 K, making char combustion difficult. If ethane supply is terminated too early after coal begins to be fed into the combustor, the temperature of particles becomes too low in the post-ethane-flame region to continue combustion. Cold gas temperature slows down the heterogeneous oxidation process, and most of the particles leave the combustor incompletely burned.

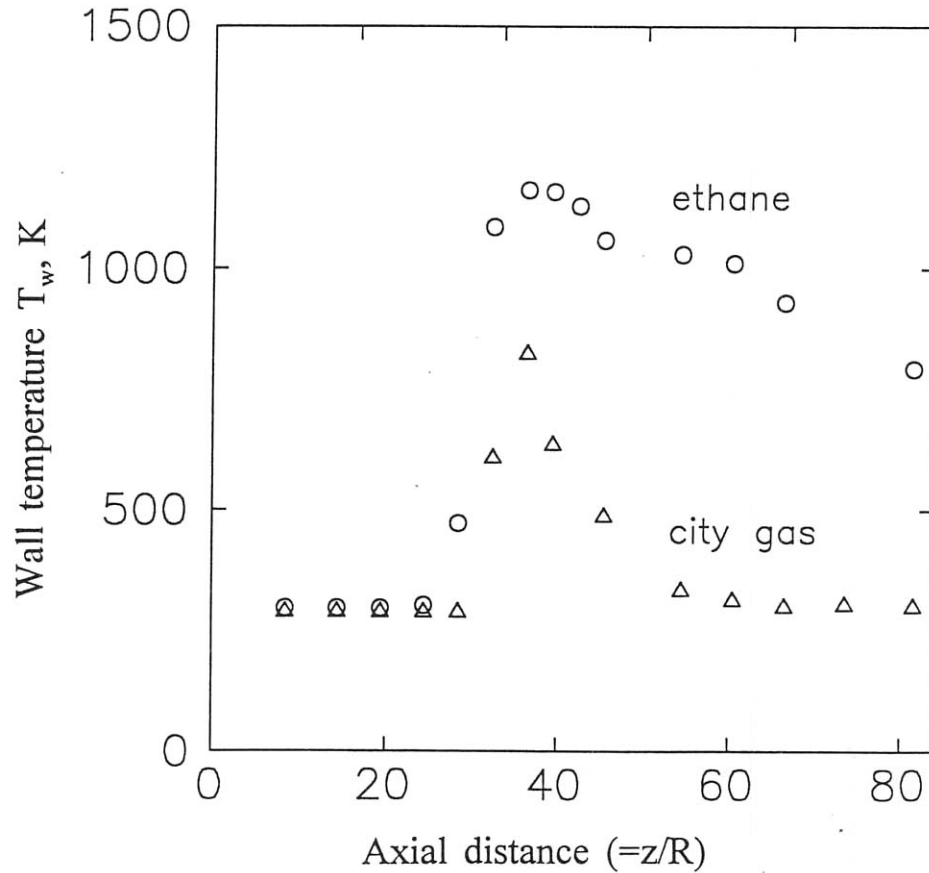


Fig. 8.7 Measured temperature distributions along the burner during the preheating period.

o: ethane gas; Δ : city gas.

Realizing the difficulty of sustaining a coal flame, ethane was burned with pulverized coal for a while rather than stopped immediately. When the wall temperature reached around 1300 K from preheating, the ethane was replaced by pulverized coal in a series of small steps: a small increase in the coal feed rate, followed by a decrease in the ethane flow almost simultaneously. Coal was supplied by opening the primary air valve and turning on the coal feeder. About 5 to 10 minutes were allowed for the flame to adjust itself to the new fuel condition. The ethane was replaced by coal slowly and completely shut off after 20 to 30 minutes. In this way, the wall temperature of the lower section of the combustor increased

and it was possible to obtain stabilized coal flames. Experiments were performed at several different coal feed rates, between 0.084 g/s and 0.122 g/s, and the corresponding temperature distributions in the combustor wall were measured. The experimental conditions and results are summarized in Table 8.1.

Visual observation through the base opening by using a mirror showed that a very bright region was formed inside the combustor. It took the form of a fire ball, and volatile matter was believed to be burned there. When the ethane supply was terminated too early (Experiment #1 and #2), numerous shining particles which were passing through the bright zone were observed. The particles, however, became dark at a short distance downstream of the flame and most of the carbon was not consumed during the particles' remaining travel to the tube exit. The color of the particles collected at the base was dark gray or black, and this indicated that very little oxidation has occurred. Insufficient energy supply to the combustor made the wall temperature even lower, and the flame eventually was extinguished.

Besides the low wall temperature of the lower part of the combustor tube, the length of the tube seems to be insufficient to provide enough residence time for a coal particle of the size used here to complete combustion. Assuming that a half of the tube (approximately 40 in. from the bottom) is heated to 1300 K and 50% excess air is supplied, the residence time is much shorter than 1 second in the hot region of the combustor. Supplying coal at a higher rate to compensate the insufficient energy supply for sustaining the temperature of the wall high was found not to be feasible with the current design because of inadequacy of the ejector system which resulted in clogging of the coal supply line.

Table 8.1 Operating conditions and results of the experiments.

Run No	feeder scale	primary air		secondary air		results **	reasons for failure or estimated slag buildup
		Q, %*	P, psig	Q, %*	P, psig		
1	N/A	N/A		25	40	F	flame flash back
2	N/A	N/A		25	40	F	flame flash back
3	3.0	50	40	0	40	F	severe clogging
4	2.0	50	40	0	40	F	severe clogging
5	1.2	50	40	0	40	F	insufficient preheating
6	1.2	50	40	0	40	F	insufficient preheating
7	1.1	50	40	0	40	F	heavy slag buildup (thickness ~ 1.0 in.)
8	1.2	50	40	0	40	S	light (< 0.5 in.)
9	1.1	50	40	0	40	S	light (< 0.5 in.)
10	1.0	50	40	0	40	S	light (< 0.5 in.)
11	0.9	50	40	0	40	S	light (< 0.5 in.)
12	1.0	50	40	0	40	S	moderate (~ 0.5 in.)
13	1.1	50	40	0	40	S	moderate (~ 0.5 in.)
14	1.2	50	40	0	40	S	moderate (~ 0.5 in.)
15	1.3	50	40	0	40	S	moderate (~ 0.5 in.)
16	1.0	40	40	0	40	F	long flame
17	1.2	45	40	0	40	F	unstable flame

* rotameter reading

** F: stabilization failed, S: flame stabilized

A much bigger and brighter flame was formed when ethane was burned with coal. After the ethane flow was replaced completely with coal according to the way described above, the flame was still visible, indicating that coal combustion was stabilized. Glowing particles were rarely seen at the tube exit, and therefore most coal particles were believed to complete combustion. At the same time, slag accumulation was also observed in the flame region. The slag deposit developed as the coal combustion continued, and after about an hour it almost reached the center of the combustor tube. Such observation of growing slag deposit indicates that the ash was melted and accumulated on the surface of the slag. The section of the combustor tube where heavy slag was covered was estimated to be less than 10 in. wide in the axial direction. Considering the narrow slag-covered zone and the observed evidence of complete combustion, the coal particles were believed to be burned at the surface of the slag.

Throughout the experiments, it was sometimes observed that burning coal agglomerate was falling down. Lumps, in the form of loosely packed coal agglomerate and approximately 5 mm or smaller in diameter, were seen in the ash box. Only a small fraction of carbon in such agglomerates was consumed on route to the observation mirror in the ash box. The number of lumps increased especially when flame stabilization failed. As the flame became stabilized and the slag deposit grew, agglomeration decreased significantly.

Currently, it is unclear why and where the agglomeration occurs. Since a triangular top was welded on each cooling tube in the combustor, pulverized coal is unlikely to accumulate on the cooling tubes. The unburned coal agglomerates can be a source of significant energy loss and should be avoided in future experiments.

8.4.6 Coal Feeding

Due to the vibration of the coal feeder, the coal powder poured into the hopper

becomes densely packed, especially when the coal is moist, and the screw encounters strong resistance against rotation. When more than half of the hopper was filled with coal, sometimes the screw stopped rotating completely, making a system reset necessary. To avoid this problem, the coal powder was dried and the depth of the coal in the hopper was kept around 1/3 full.

Clogging occurred under certain circumstances, somewhere between the venturi and the coal discharge nozzle in the stainless steel cone, then making coal feeding impossible. The problem appeared soon after coal feeding started when the feed rate was ≥ 0.2 g/s (Experiment #3 and #4 in Table 8.1). The feeder was originally designed to deliver up to 0.5 g/s of pulverized coal. At such high feed rates, however, the pressure of the primary air (max. 40 psig) is not sufficient to carry the coal to the discharge nozzle, causing the coal accumulated in the feed line. In the current study, therefore, the feed rate was limited between 0.084 g/s and 0.122 g/s, well below the maximum capacity of the feeder.

Various flow regimes can exist when particles are transported through a horizontal tube in the form of suspension. If the mean stream velocity is slower than a certain minimum transport velocity, the particles accumulate on the bottom of the tube, causing stationary layer of particles or sliding particles. The clogging is therefore believed to occur in the feed line due to insufficient momentum of the flow and the coal particles. The moisture contained in coal is likely to contribute to the clogging also. After the moisture was eliminated from the coal by heating the sample (for 20-30 minutes at around 340 K), the problem improved and disappeared completely at a moderate feed rate of 0.1 g/s.

8.4.7 Slag Buildup

Repeated experiments were accompanied by slag buildup in the combustor tube. The

accumulation of melted ash developed quickly in the flame zone especially when both ethane and coal were burned. Once the slagging starts, it develops fast and, after hours of running the experiment, the slag deposit almost blocked the entire tube. If allowed to occur, such heavy slag buildup keeps the air and coal mixture from flowing downward, forcing the stream flow out of the combustor through the gap between the combustor tube and the cooling tube assembly.

Slag is fused deposits or resolidified molten ash that adheres to surfaces exposed to high gas temperatures. Slagging and fusibility are affected by the percentages of iron, calcium, silica, alumina, etc. in the ash. The slag deposit in the RCSC was very loose and most of it could be easily removed by cleaning with a stick. Some of the slag, however, bonds strongly to the inner surface of the combustor tube, which consequently becomes very rough even after the cleaning. This makes the time interval for another slag buildup much shorter. At the coal feed rate of 0.1 g/s, the estimated time for the slag (density 15-30 lb/ft³) to grow and completely fill the 2-inch-ID tube in the flame zone (approximately 8 to 10 in. wide) is roughly 3 hours (for a tube without previous ash deposit; 10% ash was assumed). In the experiments, such heavy slag buildup was observed after about 2.5 hours of running.

It was observed that the slag developed mostly in the ethane flame zone, which suggested that coal combustion was completed there. High flame temperature expedite coal combustion and then consequently promotes slagging. The slag deposit was believed to help flame stabilization possibly by changing the flow pattern in the combustor. It increased the residence time of a particle in the flame zone and helped the particle complete combustion. However, such combustion behavior should be avoided to allow the coal particles to travel through the combustor tube while they burn. This can be achieved by terminating ethane supply early and prevent coal from burning with ethane. To this end, the wall temperature in the lower part of the combustor should be raised to the level of 1200 K to 1300 K to insure

continuous coal combustion.

8.4.8 Wall Temperature Distribution

Stabilized coal flames were obtained in the RCSC and wall temperatures were measured at various coal feed rate, from 0.084 g/s to 0.122 g/s (primary air=8.00 SCFM to 10 SCFM, P=40 psig, secondary air=0 SCFM). The measured temperature profiles in the combustor tube wall are shown in Fig. 8.8. As seen in the Figure, all the profiles have similar shapes, showing the location of the flame did not change much. This is believed to happen due to the existence of the slag deposit. After the slag deposit is established, combustion is completed in the slagged zone although the feed rate is changed. The slagging was observed throughout the experiments.

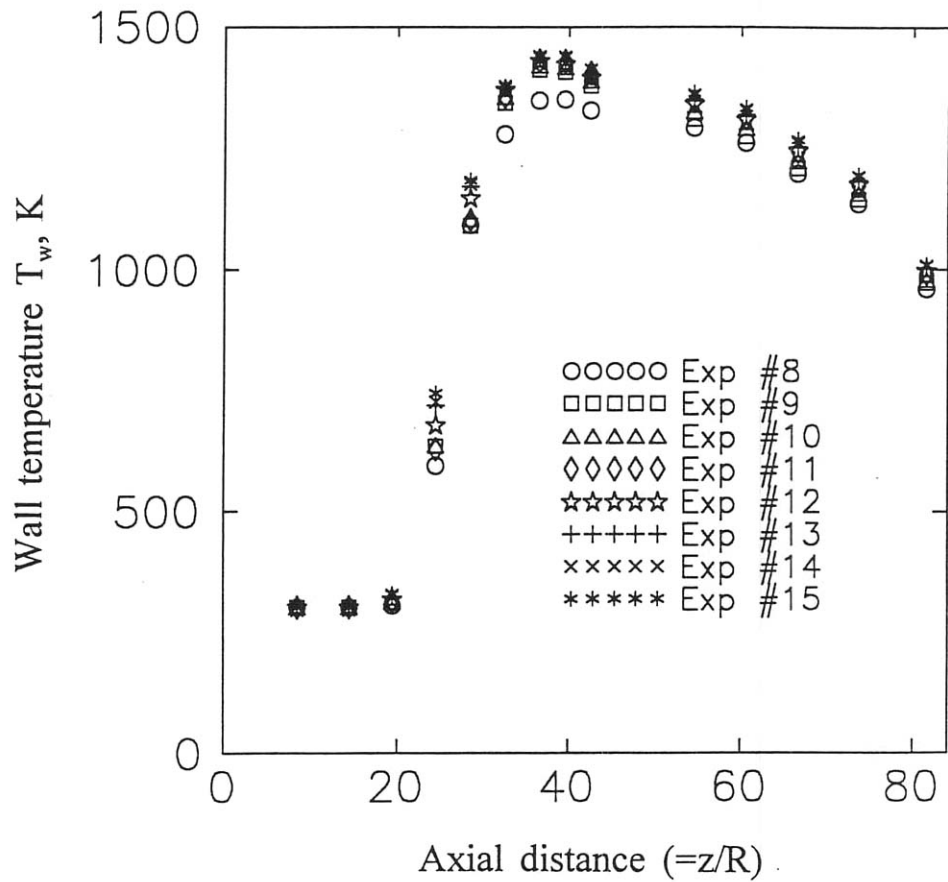


Fig. 8.8 The temperature distribution in the combustor tube wall at various operating conditions.

It is also noteworthy that the shape of the temperature profiles are similar to that of the ethane flame in Fig. 8.7. Considering that the ethane flame is very thin, it is also reasonable to conclude the coal combustion occurred in a very narrow zone. This indicates that the slag functions as a flame holder and helps combustion stabilized by increasing the residence time of coal particles in the flame region.

In the pre-flame region, temperature variation according to the different experimental conditions is negligible since no combustion energy is released from coal. In the flame

region, however, the variation due to the change of coal feed rate is wide as much as 200 K. Flame front was established at about 30 in. from the combustor inlet and the temperature increases sharply there. The peak temperature was observed at 40 in., reaching almost 1500 K. The wall temperature then dropped to about 1000 K at the exit of the combustor. This large temperature drop in the combustor wall suggests that much of the combustion energy is transferred to the wall by conduction through the slag layer.

8.4.9 Comparison with the results predicted by the first version of the model

The measured and predicted temperature distributions in the ceramic tube are given in Fig. 8.9. While the temperature profiles are similar in principle, the results show that there are differences in some aspects. As seen in the figure, the predicted result is in good agreement with the measured profile in the pre-flame region. The axial locations where the wall temperature rises sharply also agree, indicating that the location of the flame front was predicted well by the model. In the flame region, however, the temperature profiles show slightly different behavior. The measured peak wall temperature of 1430 K is 230 K higher than the predicted temperature at the same location. In the downstream of the flame front, the wall temperature drops notably, from 1430 K to 1000 K at the combustor exit. The predicted temperature, however, decreases within a narrow limit, about 110 K, maintaining almost uniform temperature in the same region. This suggests that the combustion behavior and the heat transfer mode in the flame zone are somewhat different from the theoretical model.

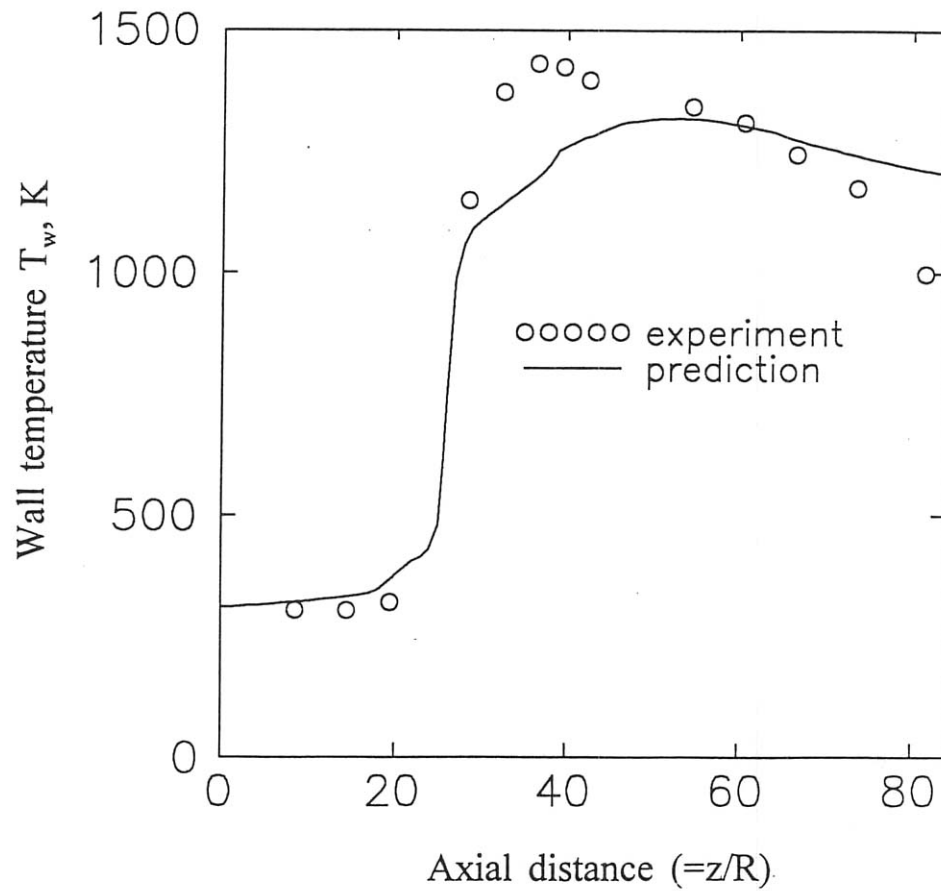


Fig. 8.9 Comparison of the measured (o) and computed (-) wall temperature distributions

In the model, the energy from burning coal particles is transferred to the surrounding gas by convection while the particles travel through the combustor tube without recirculation. Combustion energy is therefore released over a wide range of the combustor tube length and the peak wall temperature appears at some distance from the flame front. Slagging, however, changes combustion behavior of coal particles significantly in the RCSC, although it is believed to help flame stabilization. Because the particles are delayed in the slagged region, combustion is likely to be completed there, transferring most of the thermal energy to the

combustor wall. The flame downstream region of the combustor is then heated mostly by hot gas stream and flame radiation, not by the coal flame. The wall temperature therefore peaks in a relatively narrow range of the tube length at around $z=40$ in. and decreases thereafter.

The slag deposit changes the heat transfer mode in the flame zone, especially at the inner surface of the combustor tube. Since particles are believed to complete combustion in the narrow slagged zone, a substantial amount of combustion energy is transferred to the wall by conduction through the slag layer which continuously grows due to the accumulation of melted ash. Transferring combustion energy to the narrow section of the wall instead of releasing it over a wider range of the combustor length as in the theoretical model results in low gas temperature in the flame downstream. Consequently the wall temperature in the bottom section of the combustor decreases and becomes lower than the results predicted by the model. Due to the low thermal conductivity of the ceramic tube, the heat transfer from the hot tube section through the wall is very small and does not raise the wall temperature much.

To examine the change in temperature distribution when combustion is completed in such a narrow region, the residence time of the particles in the computer model was artificially varied in the flame zone to approximately emulate the experimental situation. Considering that the entire combustion may have occurred in the narrow slagged zone, the residence time of the particles in the slagging region was increased so the coal particles complete combustion between $z=28$ and 36 in. from the combustor entrance. In the theoretical model, the residence time in the flame is about 0.125 (at the tube center) to 0.6 s (near the wall) and the flame thickness is about 40 cm. By reducing the particle velocity by about half in the region, the combustion can be completed in the slag zone. Fig. 8.10 shows a temperature profile obtained from the modified model. Although the result was slightly improved, the peak temperature was still lower than the measured temperature, suggesting

that the conductive heat transfer through the slag layer was significant.

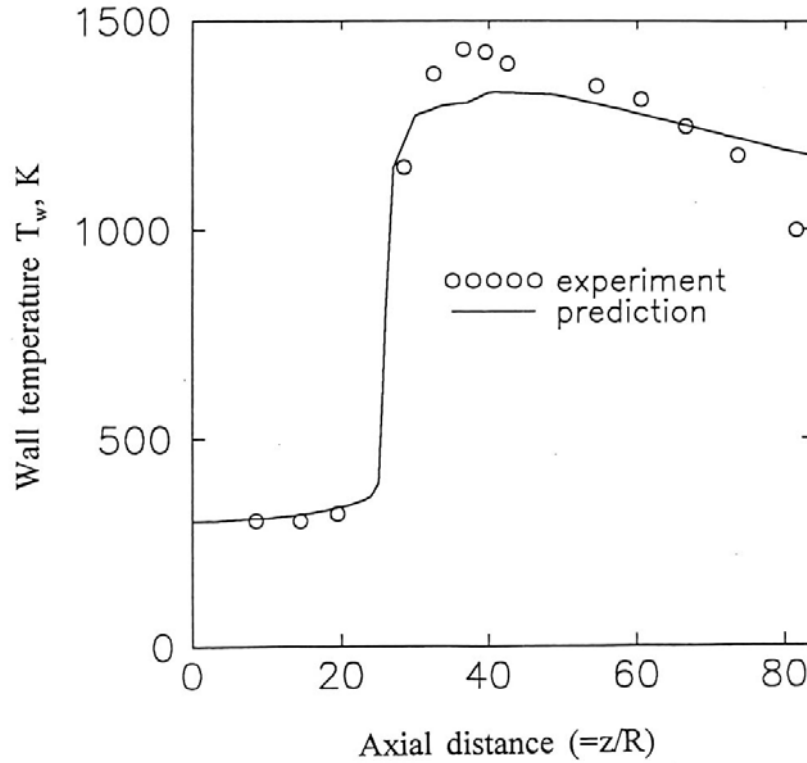


Fig. 8.10 Comparison of the measured (o) and computed (-) wall temperature distributions after modifying the velocity to approximately account for the slag.

8.5 Error Analysis

8.5.1 Sources of Error in the Model

(1) The flow model

In the RCSC, ignition of coal particles starts near the wall first and then penetrates inwards while the particles move along the combustor. Consequently, the shape of the flame front is convex in the flow direction, which results in a large radial variation in temperature. This suggests that the velocity component in the radial direction becomes important in the

flame region. While the k- ϵ turbulence flow model used in this study is usually adequate for these types of flows, it may potentially introduce a certain error in heat transfer analysis unless validated by actual velocity distribution measurements in the combustor.

(2)The heat transfer at the wall-gas interface

Heavy slag buildup on the inner surface of the combustor tube changes the heat transfer characteristics at the interface considerably. The slag, formed at the flame front first, developed into a thick, nonuniform layer as combustion continued. However, such slag deposit was not considered in this study, and no heat transfer to and from the slag was included in the model.

Growing slag implies that coal particles are believed to complete combustion in the slagged zone, transferring most of the combustion energy to the slag. Consequently, the wall temperature becomes higher than that in a combustor which has no significant slag deposit in the flame zone. Without any available information about slag deposit in the RCSC such as thickness and buildup speed, some differences between theoretical results and those from the measurement are to be expected.

(3) The P-1 approximation

To find the radiation intensity distribution in the RCSC, the Radiative Transfer Equation was solved by applying the spherical harmonics method. Governing equations were derived for both the P-1 and P-3 approximations in this study. To compare the relative accuracy of the two approximations, a comparison was made for a cylindrical geometry ($L=20$ in., $D=2$ in., $T_w=1366$ K, $\beta=3.28$ m⁻¹, $\kappa=0.98$ m⁻¹). The results shows that the P-1 approximation gave up results within 1.7% of those from the P-3 approximation in the

entrance at the middle of the combustor tube, increasing somewhat at the inlet and exit regions. Although the P-3 approximation is generally believed to yield more accurate results than the P-1 approximation, it requires 3 to 5 times longer computing time. The equation of the lowest order P-1 approximation was therefore actually used in the solution procedure.

(4) Radiative properties

Computing the radiative properties of particulates, using the Mie equations, in the RCSC would increase overall computing time considerably. Computationally very simple approximate expressions were therefore used for computing the extinction and scattering coefficients of coal, char, ash, and soot particles. The approximations, when used with the exponential wide-band model for CO₂ and H₂O, were found to introduce less than 1% error in radiation intensity calculation. The effect of the error on the temperature distribution was less than 1.5%, and, considering significantly reduced computing time, is acceptable.

(5) Devolatilization model

Predicted time-temperature history of coal particles at the early stage of devolatilization was found to agree generally well with the measured data by Fletcher (1989a, 1989b). The predicted results for 115 μm particles underestimated the particle temperature by as much as 70 K when the highest gas temperature at the centerline of the combustor was 1250 K. Lower particle temperature implies slower devolatilization, which in turn results in lower gas temperature at the flame front where volatile matter is burned. This also causes the flame to drift downward and consequently reduces the residence time of the particles in the combustor. The time delay was about 10 ms and this can cause a flame move approximately 0.2 m downward in the RCSC, which is significant.

8.5.2 Energy Loss in the Experiments

(1) Energy loss from unburned coal agglomerate

Incompletely burned coal agglomerates were observed throughout the experiments. The mass of the agglomerates, although not always constant, was estimated to amount approximately 5 to 10% of total coal supplied. Until they reached the tube exit, only a small fraction of carbon in the agglomerates was consumed, causing substantial energy loss.

A large number of unburned coal agglomerates were seen when slag deposit was insignificant. However, the agglomerates were observed much less when significant slag was built up, because they were likely to be trapped and completed combustion in the slag-deposited region. Since stabilization of the coal flame without slag buildup is desirable, agglomeration should be avoided in the future experiments.

(2) Heat loss through clamps

At the time of the combustor installation, 4 stainless steel clamps were included in the supporting structure to keep the ceramic tube in place. Each clamp consists of two 36-inch-long stainless steel bars (width = 1 in., thickness = 1/4 in.). The clamps were loosely fitted after Fiberfrax was wrapped around the ceramic tube so there is no direct contact between the tube and the clamps. The clamps, nevertheless, act like fins and transfer heat from the combustor to surrounding area. The temperature of the Fiberfrax in the flame zone, where the third clamp is located, was estimated as high as about 400 K based on the results obtained from the model. Considering that the surface area of the clamp exposed to air is approximately 0.078 m^2 , the energy loss to the surrounding air of 300 K through the third clamp alone is calculated roughly as 200 W, 8% of the supplied coal energy.

8.6 Section 8 Conclusions

Pulverized anthracite (mean diameter 46 μm) was burned in the RCSC and stable flames were obtained at various feed rate with the aid of preheating by ethane. Establishment of a stabilized flame in the RCSC was not successful when the ethane supply was stopped immediately after the preheating. To help sustain the flame and complete coal combustion, the ethane supply was continued for 20 to 30 minutes after coal feeding was started, making both fuels burn at the same time. While monitoring the flame visually, the ethane supply was slowly decreased and eventually terminated completely, leaving a stable pulverized coal flame. The wall temperature was measured and compared to the predicted results from the model. From this study, the following conclusions were obtained.

- 1) Combustion experiments were conducted by burning pulverized anthracite and stabilized flames of that fuel alone were obtained. The temperature distribution in the combustor tube wall was measured at various coal feed rate.
- 2) Throughout the experiments, heavy slagging was observed in the flame region. The slag deposit was believed to perform the function of a flame holder in the RCSC, helping flame stabilization.
- 3) Combustion was likely to be completed in a narrow slag-covered region (estimated as about 8 in. wide), and most of the combustion energy is transferred to the combustor wall through conduction.
- 4) The predicted temperature distribution is in good agreement with the measured data in the pre-flame region. Also the location of the flame front predicted by the model matches well with the measurements. The model, however, underestimates the temperature in the flame zone, probably because conductive heat transfer through the

- slag layer was not considered in the model.
- 5) Burning ethane and pulverized coal together was found to initiate slagging, though it was helpful to raise wall temperature in the lower section of the combustor and establish a stable coal flame. To prevent the formation of thick slag layers, however, the ethane supply should be terminated as early as possible. To this end, an improvement must be made in the preheating system so as to generate high wall temperature in the flame downstream region.

9. IMPROVEMENTS IN THE EXPERIMENTAL FACILITY

The facility for the RCSC experiments on pulverized coal combustion, used for validations, is described schematically in Fig. 9.1 below.

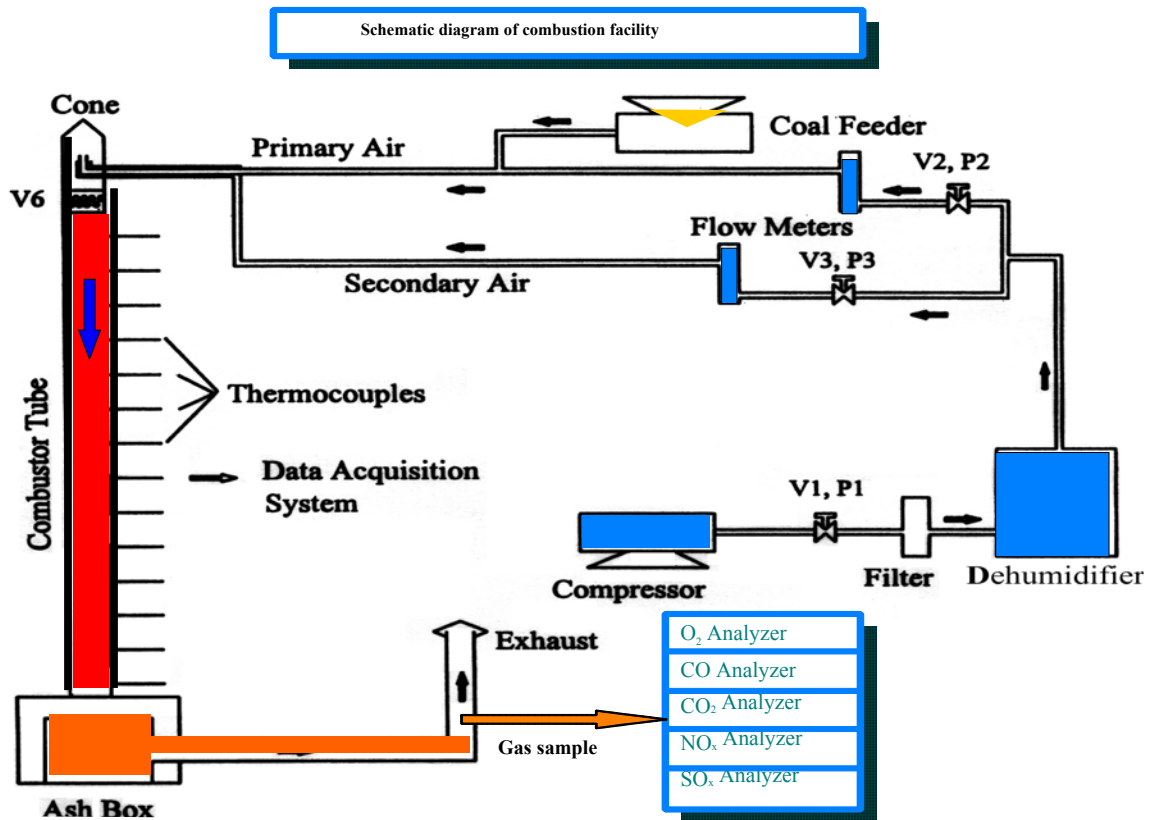


Fig. 9.1 The experimental facility

The primary improvement was the implementation of a computer-aided data acquisition system. Temperatures and the concentrations of CO_2 , CO , O_2 , NO_x , SO_x are the major data that need to be measured. The voltage output of the different instruments are shown in Table 9.1.

Table 9.1 Signal and Monitor list

Measured Parameter	Instruments	Signal voltage
O_2 concentration	Beckman Oxygen 7003 Monitor	0-100 mV
CO_2/CO concentration	Horiba PIR-2000 Gas Analyzer	0-500 mV
SO_x concentration	TE Pulsed Fluorescent Gas Analyzer Series 40	0-100 mV
NO/NO_x concentration	TE Model 10B NO/NO_x Analyzer	0-10 V
Wall temperatures	Type-S (Pt/Pt-10%Rd) thermocouples	0-10 mV

Data acquisition boards

The CIS_DAS801 and CIO_EXP32 boards of Computer Boards Inc. were chosen and purchased for analog signal acquisition. According to the temperature range, the 17 type-S thermocouples (Pt/Pt-10%Rd) were separated in two groups to measure high range temperature ($>750\text{K}$) and low temperature ($<750\text{K}$). They are connected to the CIO_EXP32 boards in two groups. The signals from CIO_EXP32 were sent to the CIS_DAS801 board and then input into the PC.

There are 8 channels in CIS_DAS801. The gain for each channel on the CIO_DAS801 can be independently set for each channel via software (HP VEE 5.0). Two channels were used for the inputs from CIS_EXP32 (thermocouples) and other channels were used for O_2 , NO/NO_x , SO_x and CO/CO_2 data acquisition.

Data from these channels were sent to a PC. Data acquisition software HP VEE 5.0 was installed and used to control data sampling process. The schematic of the DAS is shown in Fig. 9.2.

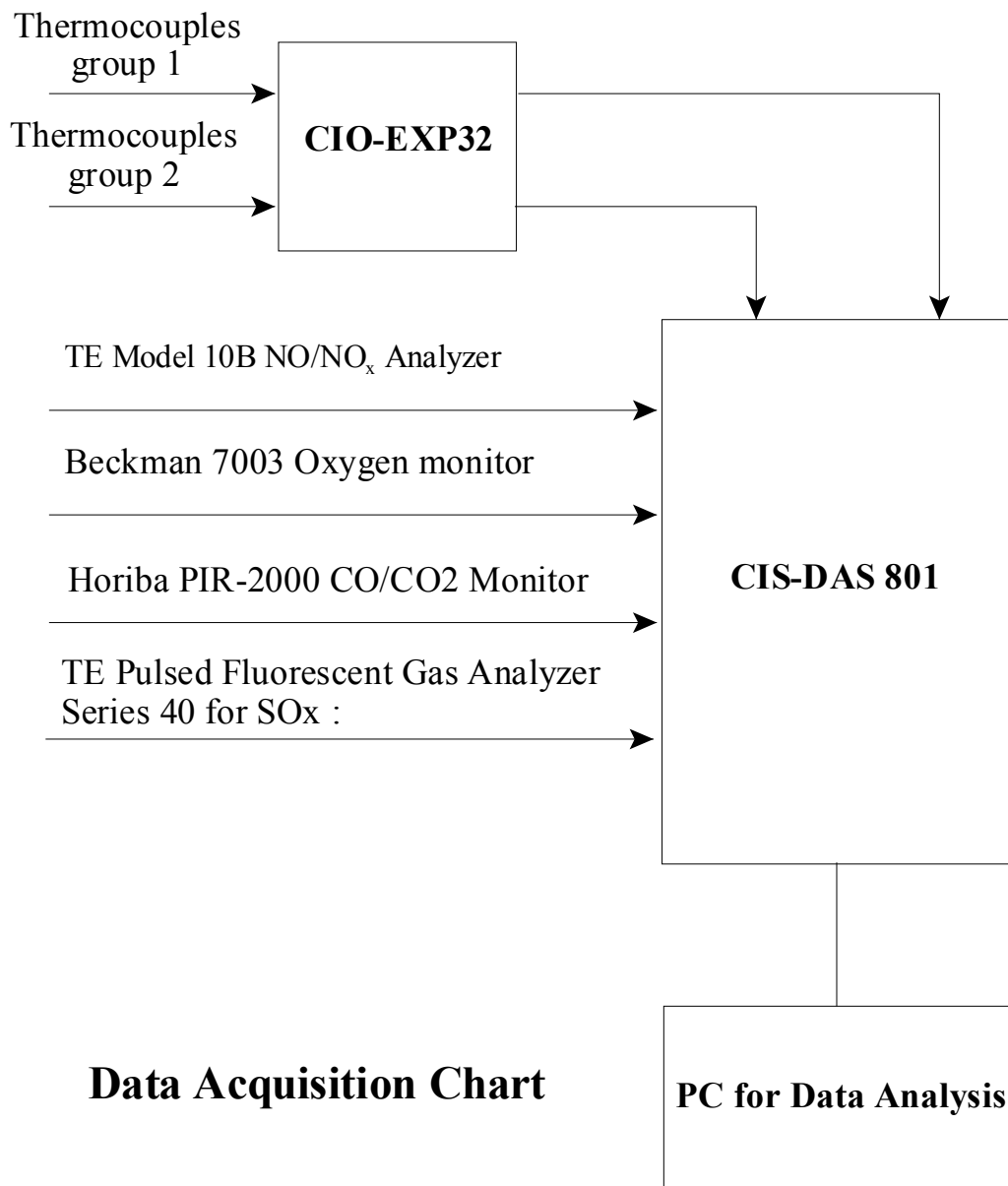


Fig. 9.2 The DAS

10. EXERGY ANALYSIS OF PULVERIZED COAL COMBUSTION (Lior 1999, 2001)

A local and global exergy analysis capability was implemented our model. It allows the computation of the total exergy as well as the exergy components distributions in the RCSC. This information is very important for fundamental understanding of the process with aim to minimize exergy destruction and ultimately increase the power production effectiveness of combustors. One of the results obtained is that 70% of the original coal exergy emerges with the hot gas, with 30% lost due to the combustion process. This loss is somewhat lower than that experienced in conventional boilers, where about 40% of the exergy is lost in the combustion furnace.

Exergy Loss Analysis

The major exergy comes from thermomechanical exergy and chemical exergy. The potential exergy and kinetic exergy could be ignored compared with thermomechanical and chemical exergy.

Thermomechanical Availability

$$A_{\text{thermal}} = (u - u_0) + p_0(v - v_0) - T_0(S - S_0) = (h - h_0) - T_0(S - S_0)$$

assuming all gases are ideal gases

$$A_{\text{thermal}} = c_p(T - T_0) - T_0 c_p \ln(T/T_0) \quad (T_0 = 298.15\text{K})$$

$$C_p = a + bT + cT^2 + dT^3 \quad \text{where } a, b, c, d \text{ are given in the following table}$$

Gas Component	Gas Properties c_p			
	a	$b \cdot 10^3$	$c \cdot 10^6$	$d \cdot 10^9$
(1) CO	6.48	1.566	-0.2387	0
(2) CO ₂	$18.036 - 0.00004474T - 158.08T^{-0.5}$			
(3) NH ₃	6.5846	6.1251	2.3663	-1.5981
(4) H ₂ O	6.97	3.464	-0.4833	0
(5) H ₂	6.424	1.039	-0.07804	0

(6) O ₂	6.732	1.505	-0.1791	0
(7) N ₂	6.529	1.488	-0.2271	0
(8) Air	6.557	1.477	-0.2148	0

C_p of NO, HCN, NH, H⁺, OH⁻ could be ignored because of low concentration

Figure 10.1 and 10.2 are thermal exergy distribution in RCSC tube, kJ/(kg of coal)

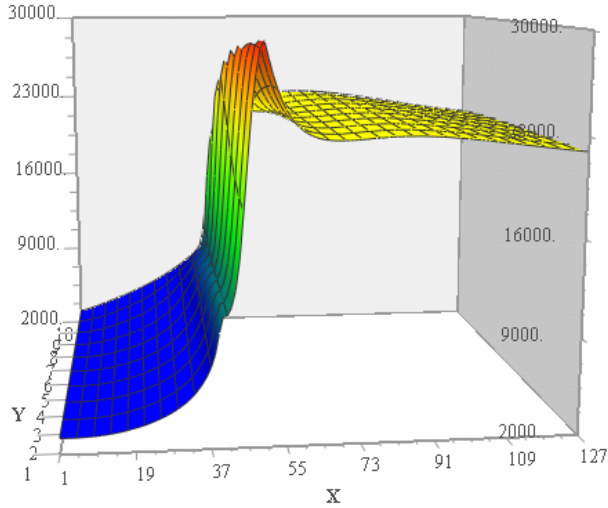


Figure 10.1

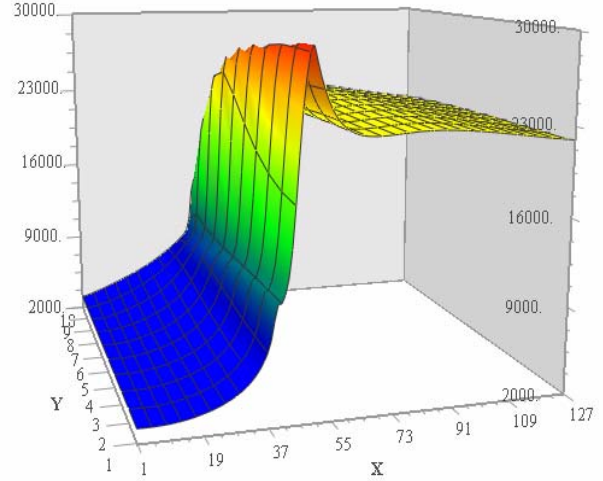


Figure 10.2

Chemical Availability in Fuel could be expressed as

$$A_{ch} = \sum_{k=1}^n N_k (\mu_k - \mu_{k0})$$

where μ_{k0} is the chemical potential of substance k within the mixture at the restricted dead state.

Values of Gibbs Function of Formation (From the Janaf Table, 1998)

		g_F , kJ/mol	Note(page number)
1	N ₂	0	P1621
2	O ₂	0	P1745
3	NO	87.193	P1603
4	HCN	124.725	P600
5	H ₂ O	-228.582	P1324
6	NH ₃	-16.367	P1343
7	CO	-137.163	P641
8	CO ₂	-394.389	P643
9	C	0	P550
10	H(g)	203.278	P1261
11	OH(g)	34.277	P1298
12	H ₂	0	P1310

Figure 10.3 and 10.4 are chemical Exergy distribution in RCSC tube, kJ/(kg of coal)

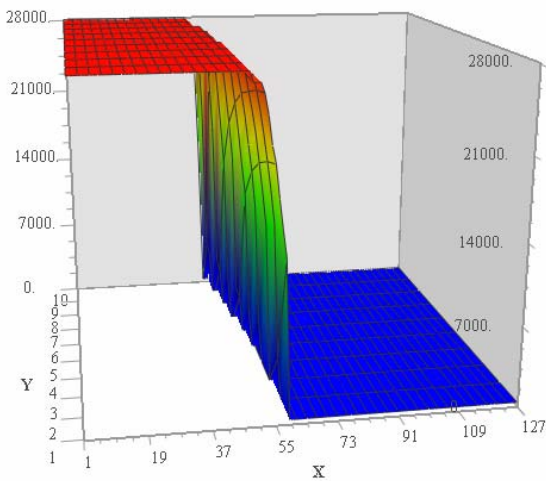


Figure 10.3

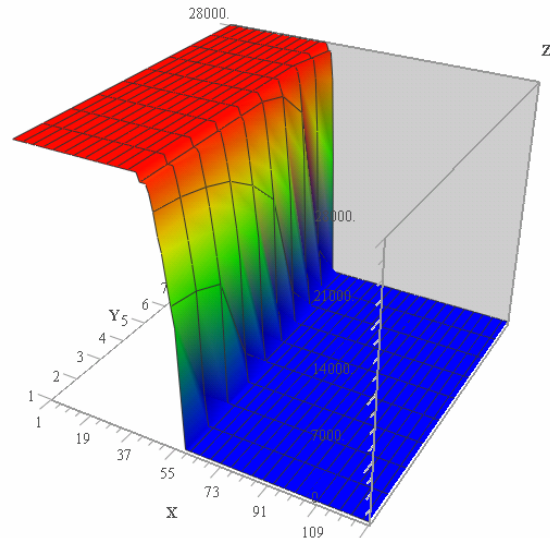


Figure 10.4

Total Availability could be expressed as

$$A_{\text{total}} = (h - h_0)V - (T - T_0)S + \sum_{k=1}^n N_k (\mu_k - \mu_{k0}) = A_{\text{thermal}} + A_{\text{ch}}$$

where N_k is the number of moles of substance k present.

Figure 10.5 and 10.6 are the total exergy distribution in RCSC tube, kJ/(kg of coal)

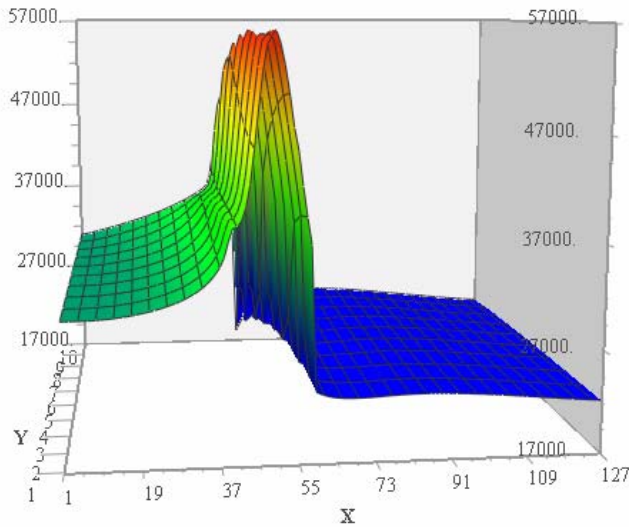


Figure 10.5

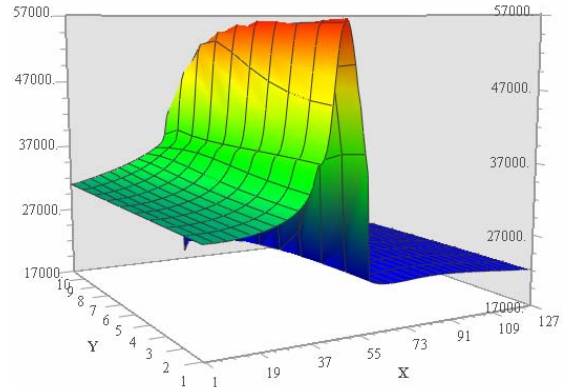
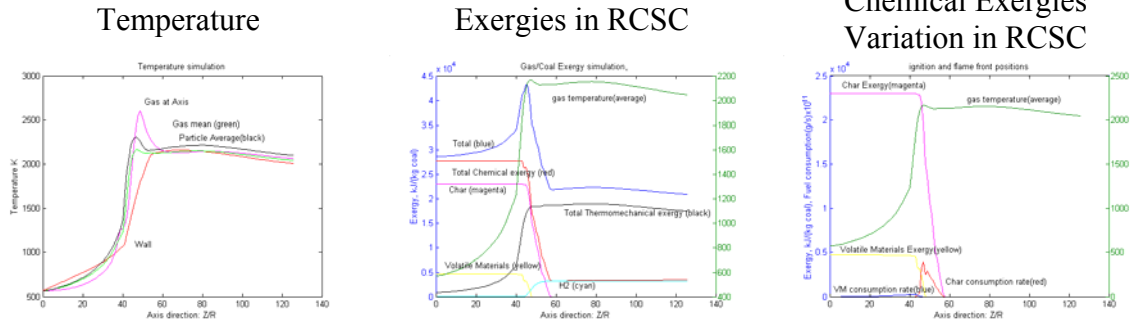


Figure 10.6

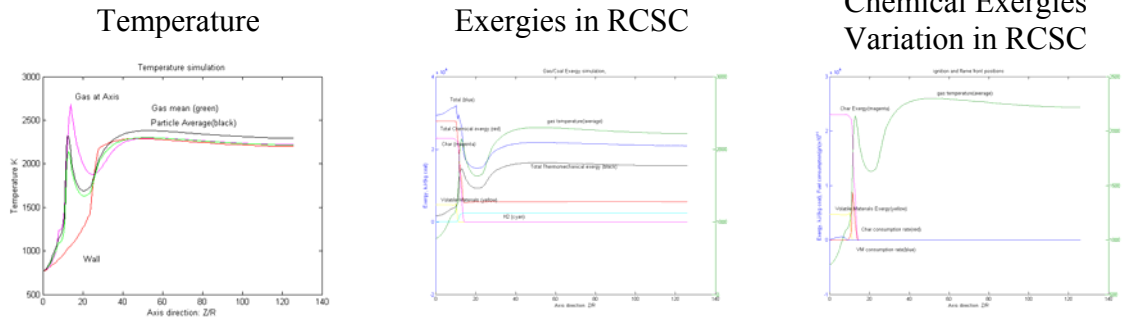
Figures 10.1 – 10.6 show that the exergy transferred from coal to air during the combustion process. At the exit 70% of coal exergy is transferred to the hot air and 30% is lost due to the combustion process. Because of the irreversible and non-isentropic process in combustion, it is impossible to convert all exergy in coal particles to the surrounding gas.

In a typical boiler, about 40% exergy is lost in the combustion in furnace. Therefore, the RCSC has a better exergy transfer efficiency and it is applicable in industry.

Inlet air/coal temperature = 300°C



Inlet air/coal temperature = 500°C



Inlet air/coal temperature = 700°C

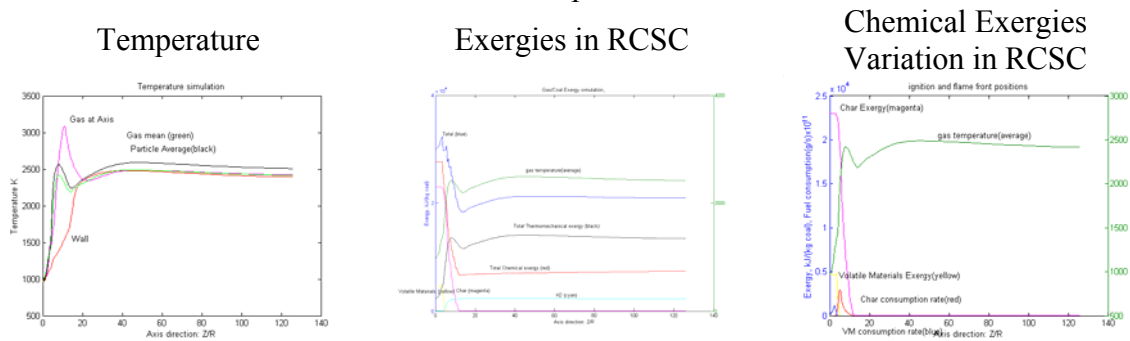


Fig. 10.7 Temperature, all exergies and chemical exergy variation along the combustor for 300, 500, and 700C inlet temperatures

Some of the conclusions from observing these distributions are (i) the thermal exergy follows rather closely the temperature field, (ii) the chemical exergy starts decreasing when the volatiles start burning, and then decreases precipitously when the char starts burning, and (iii) the total exergy rises to a peak at the upstream face of the flame front where the temperature is high due to thermal feedback yet the chemical exergy hasn't been completely used up yet.

For the conditions of this study, about 30% of the original fuel exergy is destroyed in the combustion. Over 90% of exergy destruction takes place in the thin flame zone, and less than 10% destroyed downstream of the flame zone, the latter due to the temperature drop and heat loss due to the radiative and convective transport to the surrounding wall and exit. So far, only the effects of inlet air temperature, in the range of 573-973K, were investigated; increasing it tended to slightly decrease the exergy efficiency, in part because of the increased heat losses.

While one may draw many conclusions from this analysis about possible ways to improve the exergetic efficiency in this combustor, a couple are already evident. One is the effectiveness of thermal feedback: the thermal exergy is raised by this feedback upstream of the flame front to about 1/3 of its maximal value. Another is the possibility of the existence of an optimal combustor exit location for the mixture, where the exergy is higher than the one shown here and yet the reactions were completed to satisfactory extent.

11. PRELIMINARY ESTIMATIONS FOR THE APPLICATION OF THE RCSC IN ADVANCED POWER PLANTS

Since the capacities of most advanced power plants are becoming larger. It is necessary to estimate the number of RCSC combustor base on present Model for power plant. The following table is a basic number of RCSC combustor required for a 300MW power plant.

Table 11.1. RCSC number for one typical 300 MW Boiler

ITEMS	UNIT	Eqs.	300MW Boiler
Flow rate of Water(Steam)	kg/s		284.7
Steam Pressure at exit	MPa		17.5
Steam Temperature at exit	°C		540
Incoming Water Pressure	MPa		4.9
Cool water Temperature	°C		170
Heat value of Coal	kJ/kg		11925
Gas temperature at exit	°C		148
Gas enthalpy at exit	kJ/s		1067
Cool air temperature	°C		30
Cool air enthalpy	kJ/s		193
Heat lost from gas	%		0.5
Heat lost from ash	%		1.5
Heat lost from exit gas	%		7.06
Heat lost by Outside cooling	%		0.67
Boiler efficiency	%		90.3
Saturated Steam Enthalpy	kJ/kg		3393
Saturated Water Enthalpy	kJ/kg		1370
Cool Water Enthalpy	kJ/kg	4.9MPa, 170 °C	721.65
Water drained for clean*	kg/s	1% of water flow rate	2.847
Heat transferred to boiler	kJ/s		760540

Coal feed rate	kg/s	70.65
Coal rate for per RCSC	g/s	0.2
number of RCSC		<u>353255</u>

* These reference values come from Robert Shannon, “ Handbook of Coal-Based Electric Power Generation”, Noyes Publications, 1982.

Combustor Area calculation

Base on the calculation in Table 11.1, one can find that the number of RCSC tube is very large for one advanced power plant. If these tubes were arranged in clusters and installed in the front wall and back wall of a boiler, and the diameter of each tube is 5.72cm, the total combustor area is

$$\frac{1}{4} \pi D^2 \times number = \frac{1}{4} \pi 0.0572^2 \times 353255 = 907.8 m^2$$

In many advanced power plants (such as of 300MW output) the total burners area is around 100m², therefore, it is necessary for the coal flow rate in each tube be increased near 2g/s for its commercial application.

For the RCSC combustor, the heat released by coal for per square meter can be calculated from:

$$q = \frac{W \cdot H}{A}$$

where W is coal feed rate, H is the Heat value of coal, A is the cross section of RCSC combustor.

At W=0.2 g/s, $q=0.0002 \cdot 11925 / (0.25 \cdot 3.1416 \cdot 0.0572^2) = 928 \text{ kW/m}^2$

At W=0.09 g/s, $q=0.00009 \cdot 11925 / (0.25 \cdot 3.1416 \cdot 0.0572^2) = 262 \text{ kW/m}^2$

Computations were made to examine the effect of tube diameter, length, excess air and inlet temperature, on the required burner area. In a cluster with a given coal flow rate, longer tubes, larger diameter, and preheating, and minimal excess air compatible with complete combustion seem to reduce the required total combustor area. Still, the main way to reduce the area is by increasing coal flow rate.

12. CONCLUSIONS AND RECOMMENDATIONS

- We developed a model and computer program that represents the pulverized coal combustion in the RCSC. Some of its main features are detailed accounting of the heat and mass transfer in the process, including the Radiative Transport Equation, a k-epsilon turbulence model for the gas, a Lagrangian model for the particle flow, the kinetics of 14 species, and the exergy field description.
- We predict that NO_x emissions can be reduced by using smaller coal particle sizes, by feeding the coal in a way that it is more concentrated near the combustor walls, by enhancing the NH_3 NO_x -reduction reaction, by increasing the excess air ratio, and most significantly, by increasing the inlet air/coal mixture temperature. The latter can be accomplished by using the RCSC in a recirculating heater configuration, also investigated in this study. This warrants further development.
- The wall temperature strongly influence the coal ignition position. It is near the wall where coal particles are ignited first, then the flame extends to the center of tube.
- Since the wall temperature profile is related to the ignition position and flame front, it can influence the nearby gas temperature. This feature can be used to control the combustion. A good wall temperature profile, for example, should make the flame front locate near the tube middle point, which will provide enough heat for incoming coal particles and residence time to burn out completely. It can also resist small disturbance, keep the flame in stable, and protect the combustor from overheating (e.g., flame front and ignition position near or at the combustor exit would be easily blow off by small disturbances; a flame front near the entrance may overheat the combustor.)
- While one may draw many conclusions from this analysis about possible ways to improve the exergetic efficiency in this combustor, a couple are already evident. One

is to increase the effectiveness of thermal feedback: the thermal exergy is raised by this feedback upstream of the flame front to about 1/3 of its maximal value. Another is the possibility of the existence of an optimal combustor exit location for the mixture, where the exergy is higher than the one shown here and yet the reactions were completed to satisfactory extent.

- The experiments conducted have demonstrated the methods for ignition and stable flame maintenance. Severe slagging has stopped the experimentation during the period of this contract, but ways to reduce or eliminate this problem can be found and should be explored.
- To be comparable in size to commercial combustors, the feed rate in the RCSC combustor geometry should be increased by about an order of magnitude (to 2 g/s coal feed rate in our experimental RCSC configuration). This should be accomplished while allowing full burnout of the coal, and should be made feasible by further work on configuration optimization. Other ways to decrease the required burner area are longer tubes, larger diameter, and preheating, and minimal excess air compatible with complete combustion.
- The basic attributes of the RCSC, which eliminates back-mixing and reduces residence time, and allows flame positioning along the combustor for finding a combination of minimal NO_x emissions with maximal burner efficiency, make it worthwhile to develop further till commercial applications

13. SIGNIFICANCE TO FOSSIL ENERGY PROGRAMS

The national goal expressed by the USDOE is to reduce NO_x and SO_x emissions in coal plants by at least an order of magnitude by the year 2030, and at least by four-fold by 2005. The NO_x emissions, currently at several hundred ppm, should be reduced to single-digit levels. Using fluid fuels, the RCSC has been proven in the laboratory to indeed produce NO_x at levels below 10 ppm. Using coal, our analysis and experiments conducted so far indicate that this novel approach to combustion is highly likely to reduce NO_x levels significantly.

14. ARTICLES, PRESENTATIONS, AND STUDENT SUPPORT

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S. W. Churchill, "Thermally stabilized combustion", Plenary lecture at International Gas Turbine Conference, Baltimore, MD, 1998

N. Lior, "Radiatively-conductively stabilized combustion for reducing NO_x emissions", Luleå University of Technology, Sweden, 1998

N. Lior, "Radiatively-conductively stabilized combustion", MEFOS (Foundation for Metallurgical Research), Sweden, 1998

N. Lior, "Radiatively-conductively stabilized combustion for reducing NO_x emissions", Åbo Akademi University, Finland, 1998

N. Lior, "Some advances in energy conversion to power", University of Lund, Sweden, 1998

N. Lior, "Radiatively-conductively stabilized combustion and its effects on reducing NO_x emissions", Chalmers University of Technology, Sweden, 1998

S. W. Churchill, "Some projections on energy technology" RAN Conference on Energy and Environment for the 21st Century", Nagoya University, , Nagoya, Japan, 2000

S.W. Churchill, "Thermally stabilized combustion - flow, heat transfer, and chemical kinetics in the fast lane" Academy of Chemical Engineers Lectureship, U. Missouri, Rolla, 2000.

S. W. Churchill, "Some projections on energy technology" RAN Conference on Energy and Environment for the 21st Century", Nagoya University, , Nagoya, Japan, 2000

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N. Lior, "Some second law aspects of fuel use and its effects environment" invited presentation, the Gordon Research Conference on Modern Developments in Thermodynamics, Ventura, CA, March 2001.

Students Working, and Who Worked, on This Grant

Fanfan Xiong, graduate (Ph.D.) student at the University of Pennsylvania

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Richard Chung undergraduate student at the University of Pennsylvania (work-study student)

Julian Dimery undergraduate student at the University of Pennsylvania (work-study student)

Joe Corcoran, undergraduate senior project student at the University of Pennsylvania (graduated).

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