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SLAGGING RETROFIT PULSED COAL COMBUSTOR
Final Report

Work Performed Under Contract No. AC22-85PC80261

For
U. S. Department of Energy
Pittsburgh Energy Technology Center
Pittsburgh, Pennsylvania

By
Avco Research Laboratory, Inc.
Everett, Massachusetts

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FINAL REPORT

prepared for

U.S. DEPARTMENT OF ENERGY
WORK PERFORMED UNDER CONTRACT DE-AC22-85PC80261

prepared by

AVCO RESEARCH LABORATORY, INC.
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ABSTRACT

A concept for a novel form of slagging retrofit pulsed coal combustor was tested in the laboratory. The combustor is based on controlled use of a form of high pressure amplitude combustion instability. The approach adopted was to resolve, in single pulse experiments, the basic technical issues arising in the development of the combustor.

In a cold flow device, the issues of coal spatial distribution were addressed and a combustor and solids disperser configuration was developed to give uniform coal distribution in the combustor. Single pulse ignition experiments were conducted to determine the pressure rise in combustor, pressure rise/decay times, and coal conversion as function of various operating variables. Up to 70 percent coal carbon conversion was obtained for a bituminous coal, with projected potential to convert more than 90 percent within the combustor volume in a multiple pulse situation; higher values are expected with higher-volatility coals. Coal injection, flame propagation, and blowdown times leading to potential combustor size reduction of three times over steady flow combustors were demonstrated. The results give high pressure exhaust leading to potentially improved downstream heat transfer and reduced boiler size. Finally, zero-, one-, and two-dimensional mathematical models were developed in support of the experiments and also to provide design capability.

The results of this program validate the concept and demonstrate the feasibility of developing a highly attractive, high pressure gain, and a very compact overall combustor system.

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1.0 INTRODUCTION AND BACKGROUND

1.1 PROGRAM OBJECTIVE

The objective of this program was to resolve basic technical issues arising in the development of a slagging retrofit pulsed coal combustor. The combustor is based on controlled use of a form of high pressure amplitude combustion instability (L-star design). The approach adopted was to address, in single pulse experiments, the issues of pulsed coal and air injection, coal spatial pattern in the combustor, ignition, flame propagation, burn times, pressure rise and decay times, and exhaust flow rate on a time resolved basis. The scope of work was as follows:

- Task 1 - Reactor Design and Fabrication
- Task 2 - Coal Dispersion Characterization
- Task 3 - Combustion Characterization and Optimization
- Task 4 - Modeling Support
- Task 5 - Analysis and Design Capability

1.2 PRINCIPLE OF OPERATION

In this section, first the traditional acoustic approach to pulsed combustion and then the L-Star approach are discussed. Both are analyzed in a preliminary, back-of-the-envelope fashion to introduce the difference between the two approaches. A more detailed analysis is contained in Section 4 for the L-Star approach.

1.2.1 Acoustic-Pulsed Combustor

Figure 1-1 is taken from a recent report.⁽¹⁾ It illustrates the principle of operation of a quarter-wave (i.e., smallest possible) acoustic pulsed combustor. In A., ignition and combustion of the reactants take place. In B., these hot products expand into the resonator volume; a valve is required to minimize (or prevent altogether) backflow into the feed line. In C., the rarefaction wave returning from the end of the resonator produces a low pressure in the combustion volume, which allows purging and recharging. In D.,

A. IGNITION AND COMBUSTION

B. EXPANSION

C. PURGE AND RECHARGE

D. RECHARGE AND COMPRESS

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Figure 1-1. Pulsed Combustion Principle of Operation

the compression wave from the end of the resonator returns, leading to a high pressure within the combustion volume, at which point the cycle returns to the beginning at A.

Figure 1-2 shows schematically the relevant dimensions of a conventional pulsed combustor using an acoustic resonator. It is possible to carry out a very general calculation for the volume of a conventional pulsed combustor; the requirements are simply that (a) the resonator time scale for a quarter wave resonator must have a certain relationship to the characteristic combustion time of the fuel, and that (b) the rarefaction wave return must coincide with combustion product removal so that the system can refill with a fresh charge. These two conditions specify an acoustic-resonator pulsed combustor and are therefore essential requirements.

Condition (a) can be described by:

$$\begin{aligned} \tau_{\text{RESONATOR}} &> \tau_{\text{COMB}} \\ \text{or, } L/a_R &> \tau_{\text{COMB}} \end{aligned} \quad (1-1)$$

where L is the resonator length and a_R is the sound speed in the resonator, and Condition (b) by:

$$\tau_{\text{COMB, OUTFLOW}} \sim 2\tau_R \sim \frac{2L}{a_R} \quad (1-2)$$

$$\text{or, } \frac{\rho_0 V_C}{2\tau_R} \approx \frac{\pi}{4} D^2 U_R \rho_R \quad (1-3)$$

where ρ_0 is the cold-gas density, V_C is the combustion volume, D is the resonator diameter, U_R is the flow speed in the resonator and ρ_R is gas density in the resonator. From combining these two requirements, one can arrive in a straightforward way at the fact that the combustion volume must have a certain relationship to the resonator volume:

Since

$$V_R = \frac{\pi}{4} D^2 L,$$

$$\frac{V_C}{V_R} \approx 2 \frac{\rho_R}{\rho_0} M_R \approx \frac{M_R}{2} \approx 0.2 \quad (1-4)$$

CONVENTIONAL PULSED COMBUSTOR
(ACOUSTIC RESONATOR)

Figure 1-2 Conventional Pulsed Combustor (Acoustic Resonator)

where M_R is the Mach number in the resonator. Equation (1-4) shows that the resonator volume is five times as large as the combustion volume; this is an unavoidable consequence of the requirements of operation. In most operational acoustic-pulsed combustors, the amount of fresh charge taken in is ~ 10-20 percent, and not ~ 100 percent as assumed here. That has two consequences: a) the resonator volume is reduced, which is desirable; and b) the peak pressure rise becomes very low, which is very undesirable.

The volumetric heat release rates for an acoustic pulsed combustor versus a steady flow combustor can be calculated as:

$$\left(\frac{\dot{Q}}{V_C + V_R} \right)_{AP} \approx \underbrace{\frac{\rho_o V_C q_m}{2\tau_R}}_{\text{peak heat flow}} \underbrace{\frac{2\tau_R}{2\tau_R + \tau_{C,P}}}_{\text{"duty cycle"}} \frac{1}{V_C + V_R} \approx \frac{\rho_o q_m}{6(2\tau_R + \tau_{C,P})} \quad (1-5)$$

where q_m is the heat release per unit mass.

Equation (1-5) illustrates the fact that, in an acoustic pulsed combustor there is high peak heat flow during the combustion part of the cycle, but during a large part of the cycle, there is no heat release; there is, then, a duty cycle factor. Both the combustion volume and the resonator volume are also taken into account. This calculation is, in fact, optimistic; a more precise calculation would yield even lower values of volumetric heat release. The steady flow relation incorporates the density in the combustion region, the combustor volume, the heat release per unit mass, and the combustion time under steady flow conditions:

$$\left(\frac{\dot{Q}}{V_C} \right)_S \approx \frac{\rho V_C q_m}{\tau_{C,S} V_C} \approx \frac{\rho q_m}{\tau_{C,S}} \quad (1-6)$$

Figure 1-3 shows the volumetric heat release for acoustic pulsed versus steady flow combustors Eqs. (1-5) and (1-6). The ratio of volumetric heat release for the two systems versus the ratio of the combustion time to the

characteristic acoustic time is plotted. As the pulsed system approaches the region to the right of the plot, the pressure gain drops and therefore performance drops off. Because of higher combustion intensity, the characteristic combustion time for coal under pulsed conditions will be reduced, therefore two comparisons are shown: one where the ratio of combustion times is unity, and one where the pulsed combustion time is one-third of the steady flow combustion time. As can be seen from Figure 1-3, even under the most optimistic conditions, the steady flow combustor will be smaller, having a higher volumetric heat release.

1.2.2 L-Star Pulsed Combustor

Figure 1-4 shows the Avco concept, labeled the "L-Star" pulsed combustor. In this configuration, there is no acoustic resonator. The combustion volume and exhaust orifice are chosen such that the time for pressure decay is greater than or equal to the time for combustion, in order to achieve high pressure gain. The configuration shows air being fed from a blower and entering through a vortex diode. The function of the diode is to act as an aerodynamic valve with low pressure drop in the forward direction (to allow the air to enter the combustion chamber easily), and with very high pressure drop in the back direction, (so as to prevent the hot high pressure combustion products from flowing back through the inlet line). The flow is given a swirl as it enters the chamber. This swirl is utilized to stratify the coal charge so as to place the small particles in the middle of the chamber and the large particles towards the walls of the chamber. The small particles will burn faster; with this arrangement, the mass that exits the combustor first will have been fully combusted. Once the exhaust jet leaves the combustion chamber, it impacts the slagging plate where slag separation occurs. The clean hot gas flow then proceeds to the boiler.

For the case of the L-Star pulsed combustor, the duty cycle is related to the combustion time of the fuel in a pulsed mode, the time to exhaust the combustion gas, and the time to refill with a fresh charge, and is given by:

$$f_{L-P} \approx \frac{\tau_{C,P}}{\tau_{C,P} + \tau_{EXH} + \tau_{FILL}} \quad (1-7)$$

Figure 1-3 Volumetric Heat Release, Acoustic-Pulsed Versus Steady State Combustors

- KEY REQUIREMENTS:

- NO ACOUSTIC RESONATOR
- CHOOSE COMBUSTION VOLUME AND EXHAUST ORIFICE SO:

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$$\tau_{\text{PRESSURE DECAY}} > \tau_{\text{COMB}} \left. \vphantom{\begin{matrix} \tau_{\text{PRESSURE DECAY}} \\ \tau_{\text{COMB}} \end{matrix}} \right\} \begin{matrix} \text{HIGH} \\ \text{PRESSURE} \\ \text{GAIN} \end{matrix}$$

Figure 1-4 L-Star Pulsed Combustor Concept

The ratio of filling time to exhaust time is given by:

$$\frac{\tau_{\text{FILL}}}{\tau_{\text{EXH}}} \approx \frac{\rho_c U_c A_{\text{EXH}}}{\rho_o U_o A_{\text{FILL}}} \approx 1 \quad (1-8)$$

where the fill and exhaust areas are taken to be equal. Equation (1-8) shows that the ratio of the filling time to the exhaust time is roughly one, so that the duty cycle can be related simply to the combustion time and to the exhaust time:

$$f_{\text{L-P}} \approx \frac{\tau_{\text{C,P}}}{\tau_{\text{C,P}} + 2\tau_{\text{EXH}}} \quad (1-9)$$

The heat flow then is made up of the peak heat flow and the duty cycle:

$$\left(\frac{\dot{Q}}{V_c}\right)_{\text{L-P}} \approx \underbrace{\frac{\rho_o V_c q_m}{\tau_{\text{C,P}}}}_{\text{PEAK HEAT FLOW}} f_{\text{L-P}} \frac{1}{V_c} \approx \frac{\rho_o q_m}{\tau_{\text{C,P}} + 2\tau_{\text{EXH}}} \quad (1-10)$$

Figure 1-5 shows the ratio of volumetric heat release for the L-Star pulsed combustor versus the steady flow combustor and versus the ratio of combustion time to characteristic exhaust time. The figure shows that very large gains are possible in volumetric heat release (approaching an order of magnitude) by using the L-Star combustor.

A zero-dimensional model, based on weak wave theory, can be used to assess pressure trends in the L-Star combustor. It is inaccurate for large pressure gains, but is indicative. Figure 1-6 shows a three-region zone model: region one is the combustor, two is the slag separation region, and three is the boiler.

Figure 1-7 shows typical model predictions for the pressure, and for the fraction of the total mass that has left the combustor. The left side of the figure shows the pressure rise relative to the ideal of instantaneous heat addition; the ideal pressure rise would be 6 atm for these conditions. As can be seen, for a ratio of combustion time to exhaust time of one, pressures in

Figure 1-5 Volumetric Heat Release for L-Star Pulsed Combustor Versus Steady and Acoustic-Pulsed Combustors

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Figure 1-6 Regions of L-Star Pulsed Combustors for Model

Figure 1-7 Typical Model Predictions ($V_1/V_2 = 2$, $A_e/A_n = 2$)

excess of 3 atm are achieved. In addition to that, the peak pressure drops off to 1 percent of the peak (~ 12 in. of water) at about 50 msec, allowing the refill cycle to start at that point. The right side of the figure shows the fraction of total mass efflux for the same condition. At the end of the combustion time, roughly 40 percent of the mass has left the combustor. This indicates a requirement for very good control of the stratification of the coal in the combustor to avoid loss of combustion efficiency. The results of this model show that the pressure rise in the combustion chamber for slagging is about 50 psig versus about 1 psig normally available from blowers. For improved heat transfer, it is desirable to have high time averaged p'_g . The model predicts 2 psig. It is also possible to use a muffler to tailor the spectral content of the pressure into the boiler. The requirements for this muffler will be set by the boiler structural dynamics.

2.0 EXPERIMENTAL FACILITY AND PROCEDURES

The objective of the program was to verify a novel concept of pulsed coal combustion based on L-Star design, discussed in Section 1.2.2. While many experiments on coal combustion^(2,3) and coal devolatilization⁽⁴⁾ have been carried out in closed bomb reactors, in this situation rapid injection and mixing, along with blowdown through an opening, are crucial. The approach adopted was to address, in single pulse experiments, the issues of pulsed coal and air injection, coal spatial pattern in the combustor, ignition, flame propagation, burn times, pressure rise and decay times, and exhaust flow rate on a time-resolved basis. Initially an experimental system was set up to achieve this aim. As the program progressed, increased understanding of the issues involved warranted changes in the initial experimental setup. In the following sections this progressive improvement in the experimental system and procedures is discussed, leading to a final configuration suitable for single pulse combustion experiments.

2.1 INITIAL SYSTEM CONFIGURATION FOR COAL DISPERSION CHARACTERIZATION

2.1.1 System Configuration

A transparent combustor cold-flow device was constructed to study the coal dispersion characteristics. Figure 2-1 shows a schematic of the experimental setup.

(a) Combustor. A plexiglass cold-flow device was used to allow observation and characterization of the coal particles flow pattern. It was made from a 10-inch-diameter, quarter-inch plexiglass cylinder. The overall geometry of the device is shown in Figure 2-2. The device is divided by a swirl plate into two distinct chambers; the plenum chamber and the combustion chamber.

Main air was introduced into the plenum chamber by means of two air inlet lines placed diametrically opposite on the cylindrical wall. Pulverized coal was injected directly into the combustion chamber through a centrally located injection tube. A provision was made for installation of a centrally located target that could be used as a dispersion-inducing agent.

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Figure 2-1 Initial Single Pulse Experimental Setup Schematic

Figure 2-2 Single Pulse Cold Flow Device

The function of the swirl plate is to impart a swirl to the air as it flows from the plenum chamber to the combustion chamber. The amount of swirl to be imparted to the air is an important variable to be studied and therefore required the fabrication of a number of swirl plates. Figure 2-3 is a drawing of the first swirl plate used. An average hole inclination angle (θ) of 45 degrees was selected in this case. This plate attempts to keep the initial swirl profile close to that of a solid body rotation by selecting $\tan \theta$ proportional to the radius. Figure 2-4 is a photograph of the plate.

The combustion volume was adjustable by means of a movable plate at the exhaust end. Nominal combustion volume was 20 liters. The movable exhaust plate had a centrally located exhaust orifice. The size of this orifice determines the exhaust time of the combustion products, and therefore the time available for combustion.

(b) Air Injection System. The main air was stored in a 12-liter reservoir, connected to the plenum chamber by twin 1-inch lines that terminate just upstream of the swirl plate. Air injection from the reservoir was initiated by actuation of two pneumatically actuated ball valves. The ball valve assembly was set up such that the valves go from a closed-to-open-to-closed position in one motion. The rate at which this motion occurs was controlled by the helium pressure on a solenoid valve that activates the ball valves. This set-up allows the injection of a precise amount of air per pulse, by manipulating the helium pressure on the solenoid valve and the initial pressure in the air reservoir.

(c) Solids Injection System. A system was designed to deliver pulverized coal to the reactor in well defined "slugs" suitable for pulsed operation. Figure 2-5 shows the initial injection system. It consisted in part of a 1-liter reservoir for the carrier gas (air) that is filled and locked off from its supply tank. A pneumatically actuated ball valve controls the flow of air out of the reservoir downstream to manually loaded coal particles. This fast-acting ball valve, operating under the same principle as the ball valves in the air injection system, can be used to vary the quantity of air delivered. Specifically, the valve actuation speed, port size, and pressure differential are parameters that can be varied to fine-tune the solid fuel transport conditions. For a given set of parameters, a precisely controlled puff of air is the result. The particles were carried in a slug through the delivery tube into the reactor.

Figure 2-3 Swirl Plate Drawing

Figure 2-4 Swirl Plate Photograph

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Figure 2-5 Initial Single Pulse Solids Injection Subsystem

(d) Expansion Tank. A 200-liter aluminum tank was located at the exhaust end of the reactor. This tank was used as a means of expanding and cooling combustion products. Gas samples could be taken from this volume to determine the degree of combustion in the reaction. The tank can be closed off from the exhaust line by means of a 4-inch ball valve. The tank was equipped with several ports to accommodate diagnostic instruments such as pressure gauges.

(e) Diagnostics. To characterize the extent of solids dispersion, the following diagnostics were being used on the reactor.

- o High speed photography: To provide a general indication of the particle distribution patterns in the cold flow experiments.
- o Pressure transducers: Two pressure transducers are used on the combustor: one in the main combustor volume, and the other transducer was mounted immediately downstream of the combustor exhaust port to monitor the pressure at which combustion products leave the combustor.

2.1.2 Reactor Components Sizing

A zero-dimensional model was used to facilitate the sizing of the various components of the pulsed combustor experiment. The experimental apparatus was assumed to consist of three separate volumes: high pressure storage tank, an intermediate chamber connecting the storage tank and the combustion chamber, and the combustion chamber. The combustion chamber is assumed to exhaust to the ambient. The open areas connecting two chambers are A_1 , A_2 , A_3 . A schematic is shown in Figure 2-6.

The mass flux from volume i to volume j depends on the instantaneous properties in chambers i and j .

$$\dot{m}_{ij} = \rho_i v_i A_i \quad (2-1)$$

where

$$v_i = \sqrt{\frac{2(p_i - p_j)}{\rho_i}} \quad \text{for } p_i > p_j \quad (2-2)$$

$$= 0 \quad \text{for } p_i < p_j \quad (2-3)$$

The latter equality excludes the back flow. The maximum value of v_i , under choked conditions is equal to $C_D \sqrt{RT_i/M}$, where C_D is the discharge coefficient, R is the universal gas constant, and M is the molecular weight.

- ① HIGH PRESSURE RESERVOIR
- ② COMBUSTOR PLENUM CHAMBER
- ③ COMBUSTION CHAMBER

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Figure 2-6 Zero-Dimensional Model Regions

The conservation equations for mass and energy for three volumes can be expressed as:

$$\frac{d}{dt} \rho_1 V_1 = - \dot{m}_{12} \quad (2-4)$$

$$\frac{d}{dt} \rho_2 V_2 = \dot{m}_{12} - \dot{m}_{23} \quad (2-5)$$

$$\frac{d}{dt} \rho_3 V_3 = \dot{m}_{23} - \dot{m}_{3a} \quad (2-6)$$

$$\frac{d}{dt} \rho_1 h_1 V_1 = - \dot{m}_{12} h_1 \quad (2-7)$$

$$\frac{d}{dt} \rho_2 h_2 V_2 = \dot{m}_{12} h_1 - \dot{m}_{23} h_2 \quad (2-8)$$

$$\frac{d}{dt} \rho_3 h_3 V_3 = \dot{m}_{23} h_2 - \dot{m}_{3a} h_3 + \dot{Q}_c \quad (2-9)$$

Here, subscript "a" refers to the ambient conditions. Specific enthalpy is denoted by $h (= c_p T)$ and $\dot{Q}_c$ is the rate of heat release due to combustion. For the sake of simplicity, c_p and M are assumed constant, wall heat losses are neglected, and $\dot{Q}_c$ is assumed to have exponential dependence.

$$\dot{Q}_c = Q_t \exp(-(t-t_{\text{ign}})\tau_c/\tau_c) \text{ for } t > t_{\text{ign}} \quad (2-10)$$

$$= 0 \quad \text{for } t \leq t_{\text{ign}} \quad (2-11)$$

Here t_{ign} is the time for the onset of ignition, Q_t is the total heat released in the combustion process, and τ_c is the characteristic time of combustion.

The above equations were solved subject to the following initial conditions:

$$\text{At } t=0, p_2 = p_3 = p_a = 1 \text{ atm}, \quad (2-12)$$

$$T_1 = T_2 = T_3 = T_a = 300\text{K}, \quad (2-13)$$

$$\text{and } p_1 = p_1(0). \quad (2-14)$$

The volume of the combustion chamber (V_3) is set at 20 liters in these calculations. In the experimental design, this volume is variable. The valve between the reservoir and combustor is kept open to allow 20 liters of air (based on 1 atm pressure) to pass through the tank and afterwards is closed instantaneously. At this time, the ignition in the combustion chamber takes place.

The objective of the computational results was to select volumes V_1 , V_2 , areas A_1 , A_2 , A_3 , pressure $p_1(0)$ subject to some physical constraints and the overall objective of achieving pressure gain in the combustion chamber, and a cycle time of about 100 ms. Through a trial-and-error procedure, the following parameters were selected: $V_1 = 12$ liters, $V_2 = 3$ liters, $A_1 = 1.2 \text{ in}^2$, $A_2 = 6.63 \text{ in}^2$, $A_3 = 3.14 \text{ in}^2$, and $p_1(0) = 6.5 \text{ atm}$.

The computational results for the cold flow case are shown in Figure 2-7. The pressure in the storage tank (chamber 1) decreases from 6.5 atm to 4.88 atm in 25 ms. There is a corresponding increase in the plenum chamber (chamber 2) pressure, which reaches a maximum value of 1.8 atm. The combustion-chamber (chamber 3) pressure lags behind and reaches a maximum value of 1.5 atm. After the closing of the main valve, the pressures in two chambers equilibrate quickly and decrease to 1 atm in a total time of 70 ms.

The computational results for the ignition case are shown in Figure 2-8. It is assumed that ignition occurs at 25 ms after start of air injection, with a characteristic heat release time of 10 ms. The pressure in the combustion chamber increases because of combustion and reaches a maximum value of 4.2 atm at 40 ms and decreases to a value of 1.5 atm at about 70 ms, giving a blowdown time of 30 ms. In this duration, the chamber 2 pressure remains constant. After 70 ms, pressure in both chambers is equal and decreases as exhaust to the ambient occurs. The total time for a complete cycle is about 90 ms.

In this design, the swirl plate separating volumes 2 and 3 requires a total area of 6.63 in^2 , or 60 holes of 3/8-in. diameter. Leaving about a 60-percent margin for the covered area, 60 holes can be drilled in 3 rows at 3.5, 5.5, and 7.5-in. diameters. Figure 2-3 illustrates the symmetry of the hole configuration.

An additional parameter affecting the performance of the combustion chamber is the initial swirl. In an attempt to impart a medium level of swirl, an average hole inclination angle θ of 45° was selected. It is also

M2515

Figure 2-7 Cold Flow Computational Results (Zero-Dimensional Model)

M2514

Figure 2-8 Ignition Case Computational Results (Zero-Dimensional Model)

attempted to keep the initial swirl profile close to that of a solid body rotation by selecting $\tan \theta$ proportional to the swirl plate radius, where θ is 45° at $r=2.75$."

2.2 COAL DISPERSION EXPERIMENTATION

With the initial experimental system in place, experiments were conducted to study the effects of solids dispersion target, swirl plate design, and relative time delay between air and coal injection, on the coal spatial distribution.

2.2.1 Effect of Solids Dispersion Target

Two different configurations were used to determine the effect of the solids dispersion target on the coal dispersion. In the first configuration no solids dispersion target was used and the solids were injected directly into the combustion volume. Expectations were that, for the cases where the air injection started before coal injection, enough backflow would be set up to disperse the solids without the aid of the target. The delay times were varied from an air delay of 15 ms (denoted by negative sign) to a coal delay of 30 ms (denoted as positive). High-speed films were taken and, for each case, an estimate of the percentage of coal that exited before dispersing in the combustor volume was made. The aim was to determine conditions such that all the coal injected mixed with the gas before exiting the combustor.

Figure 2-9 is a plot of the percentage of coal that left the combustor before mixing with the gas as a function of the delay time. The results were quite contrary to initial expectations. Up to 75 percent of the total coal injected went straight through the exhaust orifice before mixing, when the air injection preceded the coal injection by about 10 to 20 ms. This indicated that a center low-pressure core is formed without appreciable backflow. Figure 2-10 is a photograph of the reactor taken at 8 ms after coal injection when the air injection preceded the coal injection by 17 ms. From Figure 2-9, it is clear that over the entire acceptable range of delay times, without the center target, at least 30 percent of coal will exit uncombusted.

In the second configuration we installed a 2" spherical target. The experiments showed good coal mixing for all time delays.

M3028

Figure 2-9 Effect of Coal/Air Delay Time on Coal-Air Mixing
- No Solids Dispersion Target

Figure 2-10 Coal Distribution at 8 ms after Coal Injection
- Coal Delay 17 ms - No Solids Dispersion Target

2.2.2 Effect of Relative Time Delay Between Air and Coal Injection.

Several runs utilizing pulverized coal were made and recorded on film to determine the effect of swirl plate design on the coal spatial distribution. In all these runs the other parameters were set as follows:

- o A 3"- diameter solids dispersion target
- o Swirl plate No. 1 (Figure 2-4)
- o 20-liter combustion chamber volume
- o 2"- diameter centrally located exhaust port

Runs were made in which the coal was injected first, followed by air, as well as runs in which the coal lagged the main air. The time lags ranged from 20 ms of air delay to 30 ms of coal delay.

In all cases, good coal-air mixing was observed but swirl patterns were formed which had discernible bands of coal particles interspersed with void areas. Figure 2-11, a single frame taken from a high-speed film sequence, depicts a typical swirl pattern. The runs in which coal preceded the main air were characterized by less well defined swirl, more noticeable backflow and slower clearing times. More unidirectional flow, with clearly defined swirl bands and faster clearing times, were observed when the coal lagged the air.

In all the runs, it was clear that most of the coal was concentrated close to the walls of the chamber. This suggested that we alter the swirl-plate hole configuration in a fashion that will provide more uniform particle distribution.

2.3 INITIAL SYSTEM CONFIGURATION FOR SINGLE PULSE COMBUSTION EXPERIMENTS

At the start of the program, the expectations were that a plexiglass model of the reactor would be used for cold flow coal dispersion studies, while a metal reactor would be built for the combustion experiments. The experience, with cold flow experiments and the advantage of being able to view and record on film the combustion process in a transparent reactor, prompted us to consider using a plexiglass reactor for combustion studies as well. Stress calculations showed that a 1/2" plexiglass reactor would easily withstand the pressure pulses. Therefore, a new 1/2" plexiglass combustor was fabricated which was essentially similar to the one described above. Figure 2-12 is a photograph of the new combustor.

In order to better accommodate the requirements of the combustion experiments, the new unit was fabricated with numerous ports to allow varied positioning of the ignition source and diagnostic instruments.

Figure 2-11 Typical Coal Swirl Pattern - Spherical Dispersion Target

M3068

Figure 2-12 Plexiglass Combustor for Combustion Experiments

An additional change that was made was the fabrication and installation of a new swirl plate. The difference between this plate and its predecessor is that the holes in the outermost row are aligned with the longitudinal axis of the combustor and therefore do not participate in imparting swirl to the coal/air mixture. This change was implemented in order to decrease the disproportionately high concentration of coal previously encountered along the combustor wall.

2.3.1 Injection Repeatability Studies

A study was undertaken to investigate the accuracy and repeatability of the programmed delays in the combustion events sequence. In this study, high speed films taken during cold flow injection runs at a frame rate of 500 per second indicated the exact moments (to within 2 ms) when air and coal enter the combustor and when the arc occurs. Films of several runs were taken. The reduced data from these runs is summarized in Table 2-1. It can be seen from the data that the time required for the coal to begin to exit the inlet tube once the 180° actuator was activated varied by as much as 78 ms, indicating that the original equipment configuration (shown in Figure 2-5), under a given set of conditions, coal was not being reproducibly injected into the reactor. The standard deviation for coal delivery times was 30.74 ms as compared with only 1.13 ms for the arc.

This irregularity was attributed to the fact that in each run the coal was being manually loaded into a curved inlet line in which the coal did not always settle in the same position and was therefore being accelerated from different initial conditions (see Figure 2-5). To remedy the situation the coal injection system was modified as shown in Figure 2-13. In this new configuration, coal is loaded into a straight, vertically oriented inlet tube. It accumulates compactly and repeatably at the inlet side of a closed 90° actuated ball valve. Pressurized air is then introduced behind the coal charge. When the ball valve is actuated, the coal is blown through it into the combustor in reproducible fashion. The reduced data from the modified configuration is presented in Table 2-2. The standard deviation for coal delivery times dropped to 1.64 ms, indicating a highly repeatable coal injection process.

TABLE 2-1
INITIAL CONFIGURATION EVENT DELAYS

RUN NO.	PROGRAMMED DELAY, ms			ACTUAL DELAY, ms		INDICATED INHERENT EQUIPMENT DELAY,* ms	
	AIR (CH#1)	COAL (CH#2)	ARC (CH#3)	COAL	ARC	COAL	ARC
1	0	0	0	95	-5	82	-31
2	0	0	0	85	No Arc	72	-
3	0	0	0	95	-5	82	-31
4	0	20	20	68	34	35	-32
5	0	20	20	70	35	17	-31
6	0	10	30	68	34	15	-32
7	0	-15**	30	-3	13	25	-30
8	0	-25**	5	-3	-11	35	-29
9	0	-17**	8	63	-8	93	-29
10	0	-0**	0	9	-17	22	-30

Avg. 47.8, Avg. -30.55,
σ = 30.74 σ = 1.13

For actual delays, zero time is defined as the instant when air first enters the combustor.

*Inherent equipment delay (t_e) (the time required for events such as ball valve actuation, gas/solids transport through a fixed length of tubing, etc.) equals actual observed delay (t_a) less the sum of programmed delay (t_p) and computer interchannel delay (t_c) (13 ms per channel):

$$t_e = t_a - (t_p + t_c)$$

EXAMPLE: Case 1 above

$$t_{e_{\text{coal}}} = 95 - (0 + 13) = 82 \text{ ms}$$

$$t_{e_{\text{arc}}} = -5 - (0 + 2 \times 13) = -31 \text{ ms}$$

** Indicates a channel switch, i.e., the coal signal is sent via channel 1 and the air signal via channel 2.

M4093

Figure 2-13 Final Single Pulse Solids Injection Subsystem

TABLE 2-2
MODIFIED CONFIGURATION EVENT DELAYS

RUN NO.	PROGRAMMED DELAY, ms			ACTUAL DELAY, ms		INDICATED INHERENT EQUIPMENT DELAY,* ms	
	AIR (CH#1)	COAL (CH#2)	ARC (CH#3)	COAL	ARC	COAL	ARC
1	Same Signal			21	No Arc	21	-
2	Same Signal			20	-29	20	-29
3	Same Signal			20	-30	20	-31
4	0	0	0	32	-3	19	-29
5	0	0	0	32	-3	19	-29
6	0	-20**	31	-12	16	21	-28
7	0	-25**	25	-18	6	20	-32
8	0	-50**	19	-46	1	17	-31
9	0	-15**	19	-11	1	17	-31
10	0	-15**	19	-11	0	17	-32
11	0	-20**	27	-16	8	17	-32
						Avg. 18.9, $\sigma = 1.64$	Avg. - 30.55, $\sigma = 1.51$

For actual delays, zero time is defined as the instant when air first enters the combustor.

*Inherent equipment delay (t_e) (the time required for events such as ball valve actuation, gas/solids transport through a fixed length of tubing, etc.) equals actual observed delay (t_a) less the sum of programmed delay (t_p) and computer interchannel delay (t_c) (13 ms per channel):

$$t_e = t_a - (t_p + t_c)$$

** Indicates a channel switch, i.e., the coal signal is sent via channel 1 and the air signal via channel 2.

2.3.2 Disperser Configuration

With reliable coal, methane, and air injection subsystems in place, the next goal was to determine the coal disperser configuration that provides the most globally uniform coal dispersion in the reactor. The results of coal flow experimentation show that a centrally located, axially symmetric coal dispersion target provides the most globally uniform coal dispersion in the reactor. Within this group, it was found that a conical target does not contribute to the formation of swirl bands (a nonuniformity) as does the sphere. This determination was made from viewing high speed films taken during cold flow injection runs. A pictorial comparison of the dispersion patterns observed is shown in Figure 2-14.

2.3.3 Ignition Source

The ignition source to be used between pulses is natural gas (methane) coupled with an electric spark. The first methane injection apparatus used a 75 cc reservoir in line with a ball valve coupled to a 180° pneumatic actuator. This setup resulted in a short duration, high velocity puff of methane directed at the spark gap. Repeated experimentation on ignition of methane alone revealed that, for the above setup, the relative time delay between the valve actuation and electric spark was very critical. For a given set of conditions (natural gas pressure in the 75 cc reservoir and helium pressure on the solenoid valve actuating the ball valve), the range of time delay over which combustion of methane could be obtained was less than half a millisecond. This small operating range made it extremely difficult to obtain reproducible ignition of methane. This effect was attributed to the short time and high velocity (sonic) at which methane exited the ball valve. Therefore the 180° actuator was replaced by a 90° actuator. In this new setup the ball valve remains open after actuation, until it is manually shut off. A gate valve was also introduced downstream of the ball valve to permit control over the rate at which methane is injected in the combustor. With this new setup, reproducible combustion of methane was achieved. Figure 2-15 shows the final methane injection setup.

Figure 2-14 Effect of Disperser on Solids Spatial Distribution

Figure 2-15 Natural Gas Injection Subsystem

2.4 FINAL SYSTEM CONFIGURATION

As discussed in Sections 2.2 and 2.3, the experimental system was progressively improved. A final addition was the installation of an external recirculating loop between the combustor and the expansion tank. This modification was undertaken to permit determination of coal carbon conversion via gas analysis. The procedure adopted was to analyze representative post-combustion gas samples and determine the total amount of carbon containing gases (CO_2 , CO , and CH_4). The results could then be used to give direct indication of coal carbon utilization.

The recirculation system consisted of a blower and an external line connecting the combustor to the expansion tank.

Mixing efficiency of the blower was determined by introducing a known quantity of methane into the combustor and by circulating the gases via the blower for times varying from 5 minutes to 20 minutes. Gas samples were then taken from the combustor and the expansion tank and analyzed on a gas chromatograph. After a series of runs and with some minor modifications, it was clear that the gases close to the eight corners of the expansion box never got entrained in the circulating flow.

To circumvent the problem, a gas recirculation manifold was welded at the suction end of the blower inside the expansion tank. Different lengths of 3/8" copper tubing were then run from the manifold to the eight corners of the tank and also to some other locations where dead zones could occur. With this modification, good mixing was achieved between the combustor and expansion tank.

With all the modifications described above, the assembly was ready for combustion experiments. The final system configuration is shown in Figure 2-16.

MS2870

Figure 2-16 Final Single Pulse Experimental Setup Schematic

3.0 SINGLE PULSE COMBUSTION EXPERIMENTS

3.1 SHAKEDOWN EXPERIMENTS

The two diagnostics used to characterize the performance of the combustor are the pressure pulses and the degree of coal carbon utilization, as determined by gas analysis. In the shakedown runs, only the pressure traces were monitored. With all other parameters held constant (see Table 3-1), different amounts of coal were injected into the combustor.

TABLE 3-1
EXPERIMENTAL CONDITIONS OF SHAKEDOWN RUNS

Coal:	Pittsburgh #8 (-400 Mesh)
ϕ_{coal}^* :	Varying between 0-2.3
Air:	10 liters over 25 ms
Natural Gas:	$\phi_{\text{CH}_4}^* = 0.26$ as ignition gas and
	$\phi_{\text{CH}_4} = 0.34$ as carrier gas for coal
Time Delays:	Air and coal start entering combustor simultaneously. Spark ignited when coal front reaches exhaust plate.

* ϕ_{coal} and ϕ_{CH_4} defined relative to stoichiometric oxygen available in combustor volume.

Figure 3-1 shows the peak pressure at the position of the exhaust plate as a function of the total fuel (coal+methane)-to-oxygen stoichiometry. Coal and methane-to-oxygen stoichiometry (ϕ_{coal} and ϕ_{CH_4}) are defined relative to stoichiometric oxygen available in the combustor volume. Starting from a $\phi_{\text{coal}} = 0$ (i.e., $\phi_{\text{total}} = 0.6$), addition of coal shows increase in the peak pressure, indicating coal burning. However, since the maximum peak pressure is not at $\phi_{\text{total}} = 1$ (i.e., stoichiometric), the coal and/or methane

Figure 3-1 Effect of Coal Loading on Peak Pressure

combustion is not complete. Finally, above $\phi_{\text{total}} = 2.2$, there is a sharp decline in the peak pressure, indicating quenching effect due to too much fuel.

Figure 3-2 shows the oscilloscope pressure traces for the runs. Since for each run only one pressure peak is observed, it is concluded that both methane and coal combust at the same apparent rate. The combustion time (pressure rise time) is about 20 ms with a blowdown time (pressure decay time) of about 30-35 ms. This blowdown time is consistent with 30 ms predicted by the zero-dimensional code (see Section 2.1.2). A typical combustion sequence is shown in Figure 3-3. The figure shows ignition, flame propagation, and final complete filling of the combustor volume by the flame. When comparing Figures 3-2 and 3-3 to determine the event times (e.g., pressure rise time), it should be noted that a completely luminous combustor may be a result of slow heterogeneous oxygen-char combustion, which does not contribute significantly to the pressure profile.

In another set of experiments the time delays and amount of air and coal ($\phi_{\text{coal}} = 1.0$) were held constant and the amount of methane was varied. Methane, as ignition gas, was held constant at $\phi_{\text{CH}_4} = 0.26$ and as carrier gas was varied from ϕ_{CH_4} of 0 to 0.33. Figure 3-4 shows the effect methane amount and injection location on the pressure rise due to coal combustion alone. The pressure rise due to coal combustion alone is determined as the difference in pressure peaks for coal plus methane run and only methane (base-line) run. Figure 3-4 shows that for coal combustion, methane is more effective as ignition gas rather than carrier gas.

3.2 SYSTEMATIC EXPERIMENTS

A number of parameters affect the performance of the combustor. A total of five primary parameters were identified and systematically varied to assess the effect of each of them. These parameters are:

- A) Natural Gas (Methane)-to-Oxygen Stoichiometry (ϕ_{CH_4})
- B) Coal/Air Time Delay
- C) Coal Particle Size
- D) Coal-to-Oxygen Stoichiometry (ϕ_{coal})
- E) Ignition Source Location

PRESSURE TRACE
 $\phi_{\text{CH}_4} = 0.6 ; \phi_{\text{COAL}} = 0.33$

M3969

Figure 3-2 Typical Experimental Pressure Traces

Figure 3-3 Flame Propagation and Combustion

Figure 3-4 Effect of Natural Gas Loading and Injection Location on Coal Combustion Pressure Peak

3.2.1 Statistical Design of Experiments

In order to assess the effects of varying individual parameters, a two-level fractional factorial design of experiments was adopted. A complete factorial design requires the performance of $2^5 = 32$ experiments for a five-parameter system. An assumption was made that four factor interaction effects were not significant. Under this assumption, defining a contrast $ABCDE=1$ allows performance of 16 experiments instead of 32. However, in this design the main effects are aliased with four factor interaction effects, and two factor interaction effects are aliased with three factor interaction effects.

Under this experimental design a "high" and a "low" state was assigned to each of the five major parameters. These levels are shown in Table 3-2. Next, based on the defining contrast $ABCDE=1$, the 32 experiments were divided into two blocks of 16 experiments each. Table 3-3 shows the experimental conditions with various combinations of the high and low states. Performance of 16 experiments under either block would then give the same information. Experimental conditions under Block I (principal block) were selected for the present study.

Error estimates were obtained by repeating five experiments. The significance of the main effects were determined using the statistical F test at 95 percent confidence levels. The calculation procedure is described in Appendix I.

3.2.2 Selection of Parameter Levels

The rationale used in selection of the parameter levels, as shown in Table 3-2, is discussed below.

A) Natural Gas-to-Oxygen Stoichiometry

It was expected that using high amounts of natural gas would result in more total fuel (coal + natural gas) combustion. Therefore, the approach adopted was to use high amounts of natural gas and then back down from that value. Therefore, ϕ_{CH_4} of 0.6 and 0.26 were selected as the levels of this parameter. In all the experiments, ϕ_{CH_4} of 0.26 was used as ignition gas and any additional natural gas was introduced as carrier gas for coal.

TABLE 3-2
FRACTIONAL FACTORIAL DESIGN

<u>Parameter Designation</u>	<u>Parameter</u>	<u>"1" or "High" Value</u>	<u>"0" or "Low" Value</u>
A	ϕ CH ₄	.6	.26
B	Timing Delays - Coal - Air	69 ms 57 ms	22 ms 42 ms
C	Coal Particle Size	-325 mesh	70% below 200 mesh
D	ϕ coal	.4	1.0
E	Ignition Source Location	Radially Offset	Central

TABLE 3-3
FRACTIONAL FACTORIAL DESIGN OF EXPERIMENTS

BLOCK I (Principal Block)	BLOCK II
(1)	a
ab	b
ac	c
ad	d
ae	e
bc	abc
bd	abd
be	abe
cd	acd
ce	ace
de	ade
abcd	bcd
abce	bce
abde	bde
acde	cde
bcde	abcde

Note: Experimental conditions are according to statistical nomenclature, i.e., 'ab' represents higher levels of A and B and lower levels of C, D, and E (see Table 3-2).

B) Time Delays

The two levels of time delays were chosen such that for the "high" case, the coal front just reaches the ignition source location (electrodes) at the time of spark discharge and in the "low" case the coal front has reached the exhaust plate at time of spark discharge. Expectations were that if the flame propagation pushes some unburnt coal through the exhaust plate, the "low" case would show lower levels of combustion. This parameter would indicate how critical time sequencing would be in the multiple pulse case.

C) Coal Particle Size

Pittsburgh #8 coal was used in the experiments. The low level was the as received coal, which was specified to 70 percent below 200 mesh coal (standard power plant grind coal). To study the effect of the coal particle size, part of this coal was sifted and the fraction below 325 mesh (44 microns) was used as the high level. The proximate and ultimate analyses of these coals are given in Table 3-4.

D) Coal-to-Oxygen Stoichiometry (ϕ_{coal})

The levels of parameter ϕ_{coal} were selected in conjunction with the levels of ϕ_{CH_4} , such that the total fuel-to-oxygen ratio ($\phi_{\text{total}} = \phi_{\text{coal}} + \phi_{\text{CH}_4}$) covered fuel lean, stoichiometric, and fuel-rich conditions.

E) Ignition Source

It was anticipated that the swirling air would stratify the coal in the radial direction. The optimum location of the ignition source would be where the local fuel-to-oxygen stoichiometry is close to unity; therefore, two locations of the ignition source were tested. A central location of the ignition source was selected due to the symmetry of the combustor. A radially offset location, midway between the combustor centerline and the wall was selected as the second test position. Although it was not apparent which position was optimum, the second location was selected as a representative radially offset location.

3.2.3 Results of Statistical Experiments

Experiments in the principal block (see Table 3-3) were conducted and the coal carbon conversion and pressure peak at the position of the exhaust hole were recorded for each run. The data are given in Table 3-5.

TABLE 3-4
COAL ANALYSIS

Coal: Pittsburgh No. 8
(Bituminous Coal)

	70%-200 Mesh		-325 Mesh	
	<u>As Received</u>	<u>Dry Basis</u>	<u>As Received</u>	<u>Dry Basis</u>
% Moisture	1.52	-	1.20	-
% Ash	4.77	4.84	5.35	5.41
% Volatile Matter	37.50	38.08	36.92	37.37
% Fixed Carbon	56.21	57.08	56.53	57.22
% Sulfur	1.27	1.29	1.30	1.32
% Chlorine	0.10	0.10	0.10	0.10
% Carbon	77.82	79.02	77.80	78.74
% Hydrogen	5.20	5.28	5.08	5.14
% Nitrogen	1.70	1.73	1.72	1.74
% Oxygen (By Diff)	7.62	7.74	7.45	7.55
Heating Value (Btu/lb)	14044	14261	13961	14131

TABLE 3-5
EXPERIMENTAL DATA FOR FACTORIAL EXPERIMENT

<u>Experimental Conditions</u>	<u>Coal Carbon Conversion</u> (percent)	<u>Peak Pressure</u> (psig)
(1)	19	9.0
ab	30	11.4
ac	48	20.8
ad	50	9.7
ae	29	14.2
bc	34	11.7
bd	38	7.3
be	25	7.5
cd	53	11.2
ce	41	21.9
de	15, 12	6.7
abcd	67, 71	11.3, 11.9
abce	32, 35	8.2, 9.4
abde	53	11.6
acde	36, 49, 36	14.5, 17.4, 15.6
bcde	54	13.7

Note: Experimental conditions are according to statistical nomenclature, i.e., 'ab' represents higher levels of A and B and lower levels of C, D, and E (see Table 3-2).

Experimental

Five experiments were repeated to obtain an estimate of the experimental error. Statistical F test with 95 percent confidence limits is used to determine the significance of the effects. The sum of squares of the main effects and that of the error terms were calculated and the resulting F values (F_{exp}) are tabulated in Table 3-6. For any effect to be statistically significant, the absolute value of F_{exp} should be greater than the F value obtained from standard statistical tables (F_{tab}).⁽¹⁾

Discussion

A) Effect of Natural Gas-to-Oxygen Stoichiometry (ϕ_{CH_4})

The stoichiometry of natural gas to available oxygen in the combustor (ϕ_{CH_4}) was changed from 0.6 (high) to 0.26 (low). From Table 3-6, it is clear that factor A (ϕ_{CH_4}) has a positive effect on both coal carbon conversion and the pressure peak. It can be argued that it is obvious that an increase in natural gas concentration would increase the coal carbon conversion because of resulting higher temperature from natural gas combustion. However, since the total amount of oxygen available for both the fuels (natural gas and coal) is fixed, an increase in ϕ_{CH_4} would result in lesser oxygen available for coal conversion. Therefore, the positive effect seen from the experiments is of significance, as it indicates that in the range studied, increase in ϕ_{CH_4} does increase the coal carbon conversion.

B) Effect of Time Delay

The two levels of time delays were chosen such that for the "high" value (see Table 3-2), at the instant of spark discharge, the coal front just reaches the spark location. For the "low" conditions, the coal front has already reached the exhaust plate before the instant of spark discharge. The purpose of choosing this parameter was to determine how critical the relative timings were in the processes. The expectations were that for the "low" case as the coal front has gone beyond the spark location before sparking, some coal would be pushed through the exhaust orifice before combustion, thereby resulting in lower conversion.

TABLE 3-6
RESULTS OF STATISTICAL DESIGN

<u>DESIGNATOR</u>	<u>COAL C FACTOR</u>	<u>F_{exp} CONVERSION</u>	<u>PRESSURE PEAK</u>
(A)	ϕ_{CH_4}	10.5	7.8
(B)	Time Delay	4.3	-30.8
(C)	Coal Particle Size	27.1	62.0
(D)	ϕ_{coal}	28.1	-16.8
(E)	Arc Location	-7.0	0.1
Mean Square Error		25.9	1.3
Degrees of Freedom		5	4
F _{tab} for 95% Confidence Level		6.61	7.71

Note: For any factor to be statistically significant, absolute value of F_{exp} should be greater than F_{tab}.

The results show that the time delay had no significant effect on the coal carbon conversion. This suggests that the time delays are not very critical as far as the coal conversion is concerned.

The time delay has a strong negative influence on the peak pressure. The negative effect is understandable because in the "low" case, since coal, carrier gas (natural gas for "high" ϕ_{CH_4} cases), and air have longer time to mix, the pressure peak is higher.

C) Effect of Coal Particle Size

From Table 3-6, it is clear that a decrease in the coal particle size increases both the coal carbon conversion and the pressure peak. Smaller coal particles react faster than large particles, resulting in more conversion and pressure rise.

D) Effect of Coal-to-Oxygen Stoichiometry (ϕ_{coal})

A strong positive effect indicated for factor D on coal carbon conversion indicates that with a decrease in ϕ_{coal} , the conversion increases. This can be attributed to the increase in available oxygen for low coal loading.

A significant negative effect of ϕ_{coal} on pressure peak is indicated in Table 3-6. Pressure rise is dependent on the amount and rate of fuel consumption and not on the level of conversion. Therefore, opposing effects of ϕ_{coal} on coal carbon conversion and pressure peak are not surprising because of the way in which the levels of ϕ_{coal} are selected (see Table 3-2). For the "low" level ($\phi_{coal} = 1.0$), more coal is available for combustion than in the "high" level ($\phi_{coal} = 0.4$). This results in a higher pressure peak for the former case, but a higher conversion for the latter case, as discussed above.

E) Effect of Arc Location

A negative effect of arc location on coal carbon conversion suggests that coal burns more effectively when the ignition source is located centrally, rather than radially offset. The arc location has no significant effect on the pressure peak.

Summary of Results

Assemblage of the effects of the five parameters studied leads to the following conclusions.

1. In all experiments, only one combustion pressure peak is observed. This indicates that both CH_4 and coal react at apparent comparable rates.
2. An increase in ϕ_{CH_4} concentration increases the coal conversion. The purpose of natural gas is primarily to provide the initial ignition and secondly to increase the gas temperature to a level where coal devolatilization can occur. However, in satisfying the latter requirement, the natural gas consumes oxygen, thereby decreasing the oxygen available for coal combustion. In a multiple pulse situation, the temperature of the gases in the combustor will result in the mixing of cold fresh air and hot combustion products from the previous pulse. The use of natural gas will then be limited to its requirement as ignition source which will greatly decrease its amount. Consequently, more oxygen will be available for coal conversion. Therefore, for the single pulse experiments, a coal conversion of 70 percent is substantial and has the potential of increasing further in the multiple pulse case.
3. No significant effect of time delay on coal conversion is encouraging, as it suggests that in the multiple pulse case the timing of coal and air injection relative to the ignition are not critical.
4. In the time available for combustion, small particle size coal converts to a higher degree than larger particle size coal. Conversely, if more time is provided, more coal would combust. This suggests that we decrease the exhaust port area and increase the residence time of the coal particles.
5. Centrally located ignition source gives higher conversion than radially offset ignition source.

3.2.4 Effect of Coal Residence Time

The results of statistical experiments were instrumental in determining that one of the critical parameters, restricting the coal conversion is the

time available for combustion. In all the experimental runs reported above, 2" diameter exhaust hole was used. This resulted in a blowdown time of about 35 ms.

In the next set of experiments, the exhaust hole was reduced to 1" diameter. In all these experiments, the centrally located ignition source was used. Typical oscilloscope pressure traces are reproduced in Figure 3-5. The blowdown time was increased from about 35 ms to 100 ms, when going from 2" to 1" exhaust hole.

Table 3-7 shows the coal conversion data obtained with the reduced exhaust area. Except as noted, the experimental conditions are to be inferred from Table 3-2. Coal conversion data with 2" exhaust hole are also shown for comparison.

Run No. 1 in Table 3-7 was conducted with 70 percent below 200 mesh size coal. As seen from the results, a gain of 20 to 24 percentage points in conversion is obtained with decreased exhaust area. This can be explained on the basis that larger size coal requires more time to combust and therefore is favorably affected by an increase in the residence time. In Run No. 2 smaller particle size coal (-325 mesh) was used. The results show no improvement with decrease in exhaust area, suggesting that the residence time was sufficient for combustion of small particles of coal even with 2" exhaust hole.

Run No. 3 was also conducted with finer coal (-325 mesh), but with a higher natural gas-to-oxygen stoichiometry (0.43 versus 0.26 for Run No. 2). About 28 percentage points gain is shown with the smaller exhaust orifice. The rate at which coal combusts depends on the oxygen available for its combustion. In Run No. 3, since more natural gas is used, less oxygen is available for coal combustion. Hence, higher residence time helps increase the coal conversion.

In Run Nos. 4 and 5, the total fuel-to-oxygen stoichiometry is maintained at unity, with an increase in natural gas-to-coal ratio, an increase in coal conversion is seen. This result suggests that a lower rank coal (e.g., lignite or subbituminous), which contains more volatile fractions, would perform better than the bituminous coal (Pittsburgh No. 8) used in the present experiments.

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Figure 3-5 Effect of Exhaust Orifice Size on Blowdown Time

TABLE 3-7
EFFECT OF COAL RESIDENCE TIME

RUN NO.	EXPERIMENTAL CONDITIONS	CONVERSION WITH 2" EXHAUST HOLE (%)	CONVERSION WITH 1" EXHAUST HOLE (%)
1	bd	38	58. 62
2	cd	53	51
3	bcd with $\phi_{CH_4} = 0.43$	43, 42.5	71
4	bcd with $\phi_{coal} = 0.74$ $CH_4/coal = 0.35$	-	59
5	bcd with $\phi_{CH_4} = 0.43$ $\phi_{coal} = 0.57$ $CH_4/coal = 0.75$	-	65

Note: Experimental conditions are according to statistical nomenclature, i.e., 'ab' represents higher levels of A and B and lower levels of C, D, and E (see Table 3-2).

4.0 MATHEMATICAL MODELING

Two basic modeling requirements were identified for the program. The first objective was to support and understand the single pulse experiments. The second objective was to develop modeling tools for combustor design capability (multiple pulse). The modeling effort progressed through a zero-dimensional model, used for combustor sizing and prediction of gross effects, to one-dimensional transient model, to setting up the basis structure for a two-dimensional, transient, axisymmetric, viscous, swirling flow code. These models are described in the following sections.

4.1 ZERO-DIMENSIONAL MODEL

The basic structure for the zero-dimensional model was described in Section 2.1.2. The model represented by Eqs. (2-1) through (2-14) was extended to take into account the individual species. The mass balance equations for four species, viz. coal, oxygen, nitrogen, and combustion products, were solved in addition to the total mass and energy balance equations. The conversion of the coal to the combustion products is described on the basis of an overall second order global rate.

$$\frac{d}{dt} \rho_i Y_{ic} = - k (\rho_i Y_{iO_2}) (\rho_i Y_{ic}) \quad (4-1)$$

where Y_i 's refer to the mass fraction in the volume i , and k is the reaction rate constant, given by

$$k = \tau_c (\rho_o Y_{cs})^{-1} \quad (4-2)$$

Here, τ_c is the characteristic reaction time at the atmospheric pressure and the stoichiometric coal concentration. The value of τ_c is taken as a free parameter. Additional extensions of the model include the consideration of

some nonideal situations. The reverse flow between the combustion chamber and the plenum chamber is allowed; the discharge coefficient for the reverse flow is assumed to be a free parameter. The desirable departure from the perfect mixing in the combustion chamber is accounted for in an indirect manner. It is assumed that the composition and temperature of the exhaust products differ from the average composition and temperature of the combustion chamber. The exhaust products are assumed to consist of a mixture with a fixed fraction of stoichiometric combustion products and average combustion chamber composition. In addition, the simulation procedure is also extended to include multiple pulse situations.

The simulations were carried out for the first five pulses, each of 100 ms duration, starting from initially cold gas. The plenum chamber, p_2 , and the combustion chamber, p_3 , pressure histories are shown in Figure 4-1. The steady state (periodic) behavior is achieved after two pulses. The predicted maximum pressure is 2.7 atm for the combustion chamber and 1.5 atm for the plenum chamber.

If one allows for the reverse flow from the combustion chamber to the plenum chamber with a small value of C_D ($C_D = 0.1$), the two pressures equilibrate quickly and both reach a value of 2.6 atm. The equilibration rate is related to the relative volumes of two chambers and, in the present case of small volume of the plenum chamber, the equilibration rate is fast. The maximum temperature in the combustion chamber is reduced only slightly from 1700K to 1600K (see Figures 4-2 and 4-3).

The effect of nonideal mixing in the combustion chamber was also examined by assuming that the exhaust products consist of 90 percent combustion products, and 10 percent products at the average combustion chamber composition. The ability to expel the hot combustion products without mixing with the incoming cold products is desirable and leads to a substantial increase in the combustion chamber pressure (see Figure 4-4). In contrast to the previous situation the reduced mixing requires more pulses to attain the steady state periodic behavior.

The zero-dimensional model was used to calculate the pressure histories under the cold-flow conditions. A top-hat velocity profile in time, for the gases leaving the chamber 1, was inadequate in predicting the initial pressure $P_1(0)$ required to inject 20 liters of air at 1 atm in 25 ms. The predicted

Figure 4-1 Pressure Histories for Combustion and Plenum Chambers; $C_D = 0$ for Backflow (Zero-Dimensional Model)

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Figure 4-2 Pressure Histories for Combustion and Plenum Chambers;
 $C_D = 0.1$ for Backflow (Zero-Dimensional Model)

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Figure 4-3 Combustion Chamber Temperature History
(Zero-Dimensional Model)

Figure 4-4 Pressure Histories for Combustion and Plenum Chambers;
Ninety Percent of Exhaust Products at Flame Conditions
(Zero-Dimensional Model)

value of $P_1(0)$ was 6.5 atms, while the experimental value was 11.5 atms. The opening of the ball valves was modeled more accurately by assuming sine square dependence of the open area. The predicted pressure histories are shown in Figure 4-5. The predicted value of $p_1(0) = 10.3$ atm is in reasonable agreement with the experimental value of 11.2 atms, and the pressure drop Δp_1 is in reasonable agreement with the experimental value.

4.2 ONE-DIMENSIONAL MODEL

A one-dimensional transient model for the pulsed combustor was developed on the assumption of a plug flow in the axial direction. At this level of description all the hydrodynamic quantities at any cross section are assumed uniform. The resulting set of conservation equations for mass, species mass, momentum, and energy are given by:

$$\frac{\partial}{\partial t} \rho + \frac{\partial}{\partial x} \rho u = S_t \quad (4-3)$$

$$\frac{\partial}{\partial t} \rho Y_i + \frac{\partial}{\partial x} \rho u Y_i = \frac{\partial}{\partial x} \left(\rho D \frac{\partial Y_i}{\partial x} \right) + R_i + S_i \quad (4-4)$$

$$\frac{\partial}{\partial t} \rho u + \frac{\partial}{\partial x} \rho u^2 = - \frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left(\mu \frac{\partial u}{\partial x} \right) + S_m \quad (4-5)$$

$$\frac{\partial}{\partial t} \rho C_v T + \frac{\partial}{\partial x} \rho u C_v T = \frac{\partial}{\partial x} \left(\lambda \frac{\partial T}{\partial x} \right) - p \frac{\partial u}{\partial x} + \dot{Q}_c + S_e \quad (4-6)$$

Here S_t , S_i , S_m , and S_e refer to the contributions to the total mass, species mass, momentum, and energy of the gas phase from interactions with the particle phase. $\dot{Q}_c$ is the volumetric heat release rate due to the gas phase combustion and R_i is the reaction rate for i^{th} species. Y_i is the mass fraction of the i^{th} species. Momentum and energy losses due to wall interactions are neglected. Transport properties, μ , D , and λ refer to the turbulent properties and are related to each other by the assumption of unit turbulent Prandtl and Schmidt numbers.

A Lagrangian description of the coal particles is adopted here. Each coal particle is characterized by its radius, position, velocity, temperature, and its chemical composition. The chemical composition of a coal particle is

Figure 4-5 Cold Flow Pressure Histories (Zero-Dimensional Model)

characterized by the mass fraction of unburnt dry ash free coal, residual char, and the ash. A representative sample of coal particles is injected according to the size distribution and their motions followed with time. The position and velocity are given by.

$$\frac{d}{dt} x_p = v_p \quad (4-7)$$

$$\frac{d}{dt} v_p = \frac{-3\rho_g |v_p - u|}{8\rho_p r_p} C_D (v_p - u) \quad (4-8)$$

and the temperature is given by

$$\frac{d}{dt} \rho_p v_p c_p T_p = \dot{q}_c + \dot{q}_s + \dot{q}_v \quad (4-9)$$

$\dot{q}_c$ is the rate of conductive heat transfer, $\dot{q}_s$ is the heat release rate due to the surface reactions, and $\dot{q}_v$ is the heat release rate due to the devolatilization of coal. A two-step devolatilization mechanism⁽⁵⁾ is assumed.

Here v_1 and v_2 are volatiles and R_1 and R_2 are chars formed by the two reaction steps. k_1 and k_2 are kinetic constants and α_1 and α_2 are stoichiometric parameters. The surface reaction rate of residual chars is governed by the diffusion of oxygen and the surface reaction. Heat released due to the char reaction is absorbed completely by the coal particles.⁽⁶⁾

The gas phase conversion of the volatiles is governed by the turbulence dissipation rate. Instead of solving for the turbulence properties by an appropriate turbulence model, e.g., two-equation $k-\epsilon$ model, it is assumed that the turbulence level is constant and that the turbulence length scale is also a constant fraction of combustor diameter. These assumptions allow the determination of turbulence diffusivity and time scale.

The coupled set of Eulerian and Lagrangian equations are solved by a finite difference scheme. Convective terms are discretized implicitly in time while the diffusion terms and the source terms are discretized explicitly in time. An iterative predictor-corrector algorithm based on ICE technique⁽⁷⁾ is used for the pressure iteration.

Improvements in the one-dimensional model were made to include the influence of swirling air motion. The swirling air motion causes the axial velocity profile to depart from the plug flow profile. The average gas velocity affecting the axial motion of coal particles can be different from the mass average axial velocity. This effect can be empirically accounted for by assuming that the coal particles interact with a gas velocity which is fraction B of the mass average velocity. The value of B will be determined from experiments.

The one-dimensional model was also modified to allow use of time dependent initial conditions for the mass flux and pressure of the incoming gas. The previously described zero-dimensional model was used to obtain the values for the mass flux and pressure (hence velocity and density) of the incoming gas under the cold flow conditions. These values were used as the input for the one-dimensional model.

The motion of the ball valve controlling the pulsed coal injection was also simulated by assuming the constant angular velocity and it was concluded that the coal injection starts 7 ms later than the air injection, primarily because of the different size and aspect ratio of the valves. A top-hat mass flux profile in time for the coal injection was assumed. The total time over which the ball valve remains open was measured experimentally.

The predicted axial movement of coal front was monitored and is shown in Figure 4-6. Similar information was also obtained experimentally by looking at frames of high speed movies. In experiments the coal front is not straight, and the indicated band corresponds to the time for the minimum and maximum axial location of the front. The agreement between the predicted and experimental values is reasonable.

The minimum time spent by a coal particle in the combustion chamber is found to be 35 ms, while the average residence time for the air is 25 ms. The increased residence time for coal is partly due to the inertial delay required for the coal to accelerate to the air velocity. The model can be used to

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Figure 4-6 Comparison of Model Predicted and Experimental Cold Flow Coal Front Movement (One-Dimensional Model)

suggest the means of modifying the residence time of coal particles. Use of ultrafine coal particles with the average radius of 6 μ m leads to a decrease in the residence time as shown in Figure 4-7.

4.3 TWO-DIMENSIONAL MODEL

A transient axisymmetric two-dimensional code has been developed to simulate the flow in a pulsed combustor. The approach adopted for simulating the pulsed coal combustor is a combined Eulerian-Lagrangian approach. The gas phase conservation equations are described by a set of Eulerian equations while the coal particle phase is described by a set of Lagrangian equations. The coupling between the two phases occurs through interphase exchange terms. The gas phase equations for the mass, species balance, momentum, and enthalpy are given by

Mass Balance:

$$\frac{\partial}{\partial t} \rho + \frac{\partial}{\partial x} (\rho u) + \frac{1}{r} \frac{\partial}{\partial r} (\rho v r) = S_\rho \quad (4-10)$$

Species Balance:

$$\frac{\partial}{\partial t} (\rho Y_i) + \frac{\partial}{\partial x} (\rho u Y_i) + \frac{1}{r} \frac{\partial}{\partial r} (\rho v r Y_i) = D_{Y_i} + S_{Y_i} + R_{Y_i} \quad (4-11)$$

Momentum Balance:

$$\frac{\partial}{\partial t} \rho u + \frac{\partial}{\partial x} (\rho u^2) + \frac{1}{r} \frac{\partial}{\partial r} (\rho u v r) = D_u + S_u - \frac{\partial p}{\partial x} \quad (4-12)$$

$$\frac{\partial}{\partial t} \rho v + \frac{\partial}{\partial x} (\rho u v) + \frac{1}{r} \frac{\partial}{\partial r} (\rho v^2 r) = D_v + S_v - \frac{\partial p}{\partial r} + \frac{\rho w^2}{r} \quad (4-13)$$

$$\frac{\partial}{\partial t} \rho w + \frac{\partial}{\partial x} (\rho u w) + \frac{1}{r} \frac{\partial}{\partial r} (\rho v w r) = D_w + S_w \quad (4-14)$$

Enthalpy Balance:

$$\frac{\partial}{\partial t} \rho h + \frac{\partial}{\partial x} \rho u h + \frac{1}{r} \frac{\partial}{\partial r} (\rho v r h) = -\nabla \cdot \underline{q} - (\underline{r} : \underline{\nabla v}) + \frac{Dp}{Dt} + S_h + R_h \quad (4-15)$$

Figure 4-7 Effect of Coal Particle Size on Coal Front Movement (One-Dimensional Model)

In Eqs. (4-10)-(4-15), D's refer to the diffusion contribution, S's refer to the particle contributions, and R's refer to the gas phase reaction contributions. ρ is density, Y_i is the mass fraction of species i ; u , v , and w are axial, radial, and azimuthal velocity components, h is specific enthalpy, p is pressure, q is convective heat flux, and τ is shear stress tensor. The total number of molecular species included in the code is eight, viz. CH, CH₂, O₂, N₂, CO₂, H₂O, CO, and H₂.

The diffusion contributions are given by:

$$D_u = - \left(\frac{1}{r} \frac{\partial}{\partial r} (r \tau_{rx}) + \frac{\partial}{\partial x} (\tau_{xx}) \right) \quad (4-16)$$

$$D_v = - \left(\frac{1}{r} \frac{\partial}{\partial r} (r \tau_{rr}) - \frac{\tau_{\theta\theta}}{r} + \frac{\partial}{\partial x} \tau_{rx} \right) \quad (4-17)$$

$$D_w = - \left(\frac{1}{r^2} \frac{\partial}{\partial r} (r^2 \tau_{r\theta}) + \frac{\partial}{\partial x} \tau_{\theta x} \right) \quad (4-18)$$

The stress tensor components are given by:

$$\tau_{rr} = - \mu^t \left[2 \frac{\partial v}{\partial r} - \frac{2}{3} \nabla \cdot \underline{v} \right] \quad (4-19)$$

$$\tau_{\theta\theta} = - \mu^t \left[2 \frac{v}{r} - \frac{2}{3} \nabla \cdot \underline{v} \right] \quad (4-20)$$

$$\tau_{xx} = - \mu^t \left[2 \frac{\partial u}{\partial x} - \frac{2}{3} \nabla \cdot \underline{v} \right] \quad (4-21)$$

$$\tau_{r\theta} = \tau_{\theta r} = - \mu^t \left[r \frac{\partial}{\partial r} \left(\frac{w}{r} \right) \right] \quad (4-22)$$

$$\tau_{\theta x} = \tau_{x\theta} = - \mu^t \left[\frac{\partial w}{\partial x} \right] \quad (4-23)$$

$$\tau_{xr} = \tau_{rx} = - \mu^t \left[\frac{\partial u}{\partial r} + \frac{\partial v}{\partial x} \right] \quad (4-24)$$

where

$$\nabla \cdot \underline{v} = \frac{1}{r} \frac{\partial}{\partial r} (rv) + \frac{\partial}{\partial x} u \quad (4-25)$$

Here $\nu^t = \nu_{\text{lam}} + \nu_{\text{turb}}$ is the total viscosity. The turbulent viscosity is calculated from the solution of a two-equation $k-\epsilon$ model.

$$\nu_{\text{turb}} = C_\nu \rho k^2 / \epsilon \quad (4-26)$$

While the laminar viscosity is assumed to be that of air with

$$\nu_{\text{lam}} = A_1 T^{3/2} (T + A_2) \quad (4-27)$$

where A_1 and A_2 are constants.

Source terms, S , are related to the particle dynamics and the interphase transport rates for mass, momentum, and energy. In a given gas volume, S is equal to the sum of contributions from all the particles in that volume. The individual particle contributions will be described later.

The conservation equations for k and ϵ are given by:

$$\frac{\partial}{\partial t} \rho k + \frac{\partial}{\partial x} \rho u k + \frac{1}{r} \frac{\partial}{\partial r} (\rho v r k) = D_k + S_k \quad (4-28)$$

$$\frac{\partial}{\partial t} \rho \epsilon + \frac{\partial}{\partial x} \rho u \epsilon + \frac{1}{r} \frac{\partial}{\partial r} (\rho v r \epsilon) = D_\epsilon + S_\epsilon \quad (4-29)$$

where

$$D_k = \frac{\partial}{\partial x} \left(\frac{\nu^t}{\sigma_k} \frac{\partial}{\partial x} k \right) + \frac{1}{r} \frac{\partial}{\partial r} \left(\frac{\nu^t}{\sigma_k} r \frac{\partial}{\partial r} k \right) \quad (4-30)$$

and

$$D_\epsilon = \frac{\partial}{\partial x} \left(\frac{\nu^t}{\sigma_\epsilon} \frac{\partial}{\partial x} \epsilon \right) + \frac{1}{r} \frac{\partial}{\partial r} \left(\frac{\nu^t}{\sigma_\epsilon} r \frac{\partial}{\partial r} \epsilon \right) \quad (4-31)$$

$$S_k = G - C_D \rho \epsilon \quad (4-32)$$

$$S_\epsilon = (C_1 \epsilon G - C_2 \rho \epsilon^2) / k \quad (4-33)$$

$$G = \nu_{\text{turb}} \left[2 \left(\frac{\partial u}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial r} \right)^2 + \left(\frac{v}{r} \right)^2 + \left(\frac{\partial u}{\partial r} + \frac{\partial v}{\partial x} \right)^2 \right. \\ \left. + \left(\frac{\partial w}{\partial x} \right)^2 + \left(\frac{\partial w}{\partial r} - \frac{w}{r} \right)^2 \right] - \frac{2}{3} \left(\frac{v}{r} \right) \left(\nu_{\text{turb}} \left(\frac{v}{r} \right) = \rho k \right) \quad (4-34)$$

The turbulent constants are $\sigma_k = 1.0$, $\sigma_\epsilon = 1.3$, $C_D = 1.0$, $C_1 = 1.44$, $C_2 = 1.92$, $C_\nu = 0.09$. Equations (4-28)-(4-34) are the standard equations for use in the gas phase turbulence models. It is assumed that the contributions to k and ϵ from the particle phase are negligible.

The gas phase reactions are modeled by use of an eddy breakup model. Two global reactions, corresponding to two kinds of volatiles, are assumed to occur.

where V_1 and V_2 are the volatiles of CH and CH_2 type, respectively. The rate of disappearance of volatile V_1 is given by

$$\frac{d}{dt} m_{V_1} = -k_1 \quad (4-35)$$

$$\text{where } k_1 = C_{V_1} \cdot \frac{\rho \epsilon}{k} \cdot \min \left(Y_{V_1}, \frac{Y_{O_2}}{S_1} \right) \quad (4-36)$$

$$\text{and } \frac{d}{dt} m_{V_2} = -k_2 \quad (4-37)$$

$$\text{where } k_2 = C_{V_2} \cdot \frac{\rho \epsilon}{k} \cdot \min \left(Y_{V_2}, \frac{Y_{O_2}}{S_2} \right) \quad (4-38)$$

C_{V_1} and C_{V_2} are the constants of the eddy breakup model.

The formation of CO and H₂ in the gas phase is included through two equilibrium reactions

The forward rate of the reaction is assumed to be arbitrarily large, while the backward rate is calculated from the equilibrium constant. Under coal rich conditions, the gas phase decomposition of volatiles is accounted through following two global reactions

The motion of the coal particles is followed in a Lagrangian manner. A small number of particles are injected in a statistical manner according to their properties so that each simulated particle represents a large number of actual particles. Each simulated particle is characterized by its radius, r_p , temperature, T_p , three velocity components, u_p , v_p , and w_p , its chemical composition comprised of dry ash free coal content, amount of char and ash, and the number of actual particles it represents, N_p .

The position and velocity history of the particle is followed by solving the set of linear equations:

$$\frac{d}{dt} x_p = u_p \quad (4-39)$$

$$\frac{d}{dt} y_p = v_p \quad (4-40)$$

$$\frac{d}{dt} u_p = F_{xp} \quad (4-41)$$

$$\frac{d}{dt} v_p = F_{yp} + w_p^2/y_p \quad (4-42)$$

$$\text{and } \frac{d}{dt} w_p = F_{\theta p} - v_p w_p/y_p \quad (4-43)$$

$$\underline{F} = \frac{3}{8} \frac{\rho}{\rho_p} \frac{(\underline{u} + \underline{u}' - \underline{u}_p)^2}{\rho_p} (\underline{u} + \underline{u}' - \underline{u}_p) C_D \quad (4-44)$$

Here $\underline{F}$ represents the drag force exerted on the particle by the surrounding gas, and $\underline{u}$, $\underline{u}'$, and $\underline{u}_p$ represent gas velocity, turbulent eddy velocity, and particle velocity vectors, respectively. C_D is the drag coefficient which depends on the particle Reynolds number.

The particle temperature history is given by

$$\frac{d}{dt} (\rho c T)_p = \dot{q}_c + \dot{q}_r \quad (4-45)$$

where $\dot{q}_c$ is the convective heat transfer rate, $\dot{q}_c = + 4\pi r_p^2 h_c (T_g - T_p)$, and $\dot{q}_r$ is the heat flux contribution due to the devolatilization of coal and the heterogeneous char oxidation. Convective heat transfer coefficient is related to the gas phase Nusselt number by $Nu = 2 h_c r_p / \lambda_g$. $\dot{q}_r$ is calculated by assuming that O_2 reaches the particle at the gas temperature and CO_2 and volatiles leave the particle at the particle temperature. As a result of this assumption, the heat liberated by the heterogeneous char oxidation is absorbed by the particle and the heat required for devolatilization is given by the particle. The individual devolatilization and char oxidation rates are calculated as follows.

The rate of devolatilization is calculated by assuming two competing reaction mechanisms shown in Eq. (4-a) where dry ash free coal breaks into two volatiles, V_1 and V_2 , (of type CH and CH_2) depending on the particle temperature. The rate of disappearance of dry ash free coal is given by:

$$\dot{m}_{daf} = - (k_1 + k_2) 4\pi r_p^2 \dot{m}_{daf} \quad (4-46)$$

The effective rate of char oxidation is governed by the rate of oxygen diffusion to the surface and subsequent reaction of O_2 at the surface. The overall stoichiometry of reaction is

In absence of any information, the effective rates of both reactions are assumed equal and are given by

$$\frac{d}{dt} (\rho Y)_{O_2} = -k_{\text{eff}} (4 \pi r_p^2) (\rho Y_{O_2}) \quad (4-47)$$

$$\text{where } k_{\text{eff}} = \frac{k_{\text{diff}} \cdot k_{\text{ch}}}{k_{\text{diff}} + k_{\text{ch}}} \quad (4-48)$$

$$k_{\text{diff}} = \frac{D_g M_c}{R (r_p) (T_p + T_g)} \quad (4-49)$$

$$\text{and } k_{\text{ch}} = A_{\text{ch}} e^{-E_{\text{ch}}/RT_p} \quad (4-50)$$

Here D_g is the binary diffusion coefficient at the gas temperature, M_c is atomic weight of carbon, R is universal gas constant. A_{ch} and E_{ch} are frequency factors and activation energies for chemisorption process.

4.3.1 Sample Results

The idealized combustor geometry for initial computations is shown in Figure 4-8. The combustor inlet has three annular openings and a central exhaust hole. The computations were started by injecting air in the initially stagnant air. The axial velocity of incoming air was specified as 10^3 cm/s, the radial and azimuthal components were set to zero. The simulations were carried out for 5 ms of real time and required 350 computational steps. A computational grid of 19×19 was used. The computed velocity vectors are shown in Figure 4-9. The initially strong radial gradients in the axial

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Figure 4-8 Schematic of Combustor Geometry for Two-Dimensional Model

M4098

Figure 4-9 Computed Gas Velocity Vectors at 12 ms (Two-Dimensional Model)

velocity disappear after approximately 10 cm. The pressure is uniform throughout the combustor as the expected equilibration time based on the sonic velocity is approximately 1 ms.

The two-dimensional code was tested further by running it in a zero-dimensional mode. The number of interior cells is selected as unity, and the appropriate wall boundary conditions are selected so that the code simulates a constant volume adiabatic combustor. Initial pressure was chosen as 1 atm and temperature as 300K. Different amounts of coal were injected and the conservation equations were integrated in time. Because the reaction rates are extremely small at room temperature, the devolatilization reaction rates were increased artificially to achieve the combustion. The steady state temperature is shown in Figure 4-10 for different values of equivalence ratios. The calculated values of equilibrium temperatures are in good agreement with the values calculated from the NASA code. The neglect of radicals, e.g., H, O, and OH, has little effect on the equilibrium temperature under the present conditions.

Two-dimensional code was also used to study the transient simulation of the particle heatup, devolatilization, and combustion. For these studies monosized particles were injected in a constant volume chamber of hot air. The temperature and pressure of gas and the particle temperature were followed with time. The turbulence equations were not solved, but were specified by $\tau_t = 0.2$ ms and $u' = 100$ cm/s. All the turbulence eddies were characterized by the average turbulence properties. The calculated pressure and temperature of gas and the particle temperature histories are shown in Figures 4-11 and 4-12 for different sized particles. The ignition delay times are approximately 1 and 3 ms for 10 and 20 μ m radius particles, respectively. All monosized particles show an overshoot in the particle temperature. The plateaus in the particle temperature curves correspond roughly to the end of the devolatilization and heterogeneous char oxidation process. The integrated conservation equations were monitored for the overall conservation of quantities; both mass and internal energy were conserved with error less than 0.1 percent.

An important difference in the ignition behavior is observed when poly-disperse particles are used. The results for 20 particles with the number average radius of 20 μ m are shown in Figure 4-13. Pressure rise is gradual

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Figure 4-10 Comparison of Thermodynamic Equilibrium Temperatures for Constant Volume Combustion

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Figure 4-11 Pressure and Temperature Histories for Stoichiometric Combustion of Monosized Particles; $R = 10 \mu m$ (Two-Dimensional Model)

M4096

Figure 4-12 Pressure and Temperature Histories for Stoichiometric Combustion of Monosized Particles; $R = 20 \mu\text{m}$ (Two-Dimensional Model)

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Figure 4-13 Pressure and Temperature Histories for Stoichiometric Combustion of Coal Particles of Average $R = 20 \mu m$ (Two-Dimensional Model)

and slow, with no indication of a sudden ignition. The weight average particle temperature exhibits no overshoot. The oscillations in the average gas temperature are believed to be statistical due to the interaction of particles with the turbulent eddies. The turbulent eddy lifetime was assumed to be 2 ms. Once again, mass and internal energy were conserved in the integration process.

The two-dimensional code was tested under one-dimensional mode by neglecting radial gradients. The influence of an impulsively started air stream on the coal particles was examined, and the computed results at 4 different times of 2 ms interval are shown in Figure 4-14. These results clearly indicate the carrying over of the fines by the air stream and their subsequent removal from the domain. The code was also tested under different time-dependent (periodic) inflow boundary conditions, and found to give satisfactory results. For nonreacting, nonswirling flow cases, the model predictions have been compared with the experimental data of Chaturvedi⁽⁷⁾ and Owen,⁽⁸⁾ and show good agreement as shown in Figures 4-15 and 4-16.

Figure 4-14 Location of Coal Particles in an Impulsive Started Air Stream at Different Times

Figure 4-15 Comparison of Centerline Profiles for the Turbulent Flow in Sudden Confinement Expansion: Symbols - Exp. Data of Chaturvedi; Line - Calculated Value

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Figure 4-16 Comparison of Centerline Profiles for Two Coaxial Jets Entering a Sudden Confined Expansion: Solid Line Exp. Data of Owen; Dotted Line - Calculated Value

5.0 CONCLUSION

5.1 PROGRAM CONCLUSIONS

The basic issues arising from the development of the L-Star pulsed combustor were addressed in single pulse experiments. The results of the program validate the L-Star combustor concept and demonstrate the feasibility of developing a highly attractive combustor system.

In a cold flow device the issue of coal spatial distribution was addressed. It was determined that coal injection through a straight tube did not give acceptable coal dispersion. The experiments with a dispersion target showed that a centrally located, axially symmetric cone-shaped target gave the most globally uniform dispersion in the combustor. In a multiple pulse situation, a hollow cone injector would serve the same purpose as the central conical target in the experimental design. Therefore, the use of a central disperser in the experiments should not be constituted as a requirement in the multiple pulse case.

The coal conversion and pressure rise were determined for various combinations of five major operating parameters. The purpose of these experiments was to study the effect of each of these parameters to aid in optimizing the combustor configuration and operating conditions. In all these experiments, natural gas coupled with an electric spark was used as the ignition source. The results of these experiments are summarized below.

Natural gas and coal react at apparent comparable rates. An increase in the natural gas concentration from natural gas-to-oxygen stoichiometry of 0.26 to 0.6 increases the coal carbon conversion. The timing of coal and air injection relative to the ignition are not critical. A centrally located ignition source gives higher conversion than a radially offset ignition source. Increase in the blowdown time from 35 ms to 100 ms, increased the coal carbon conversion by 20-25 percent points.

In summary, the experimental accomplishments are as follows:

- Addressed issues of coal spatial distribution and developed a configuration to give good solids dispersion.
- Demonstrated up to 70 percent coal carbon conversion in single pulse. This conversion could be increased by using a higher-volatility coal than the bituminous coal (Pittsburgh No. 8) used in the experiments.

- Has potential to convert more than 90 percent coal carbon in multiple pulse situation within the combustor volume with a bituminous coal.
- Demonstrated coal injection, flame propagation, and blowdown times leading to potential size reduction of three times over steady flow combustors.
- Demonstrated high pressure exhaust leading to potentially improved downstream heat transfer and reduced boiler size.
- Developed zero-, one-, and two-dimensional mathematical models in support of experiments and also to provide design capability.

5.2 RECOMMENDATIONS FOR FURTHER WORK

- Single pulse experiments should be conducted with a high volatile coal, e.g., lignite or subbituminous, to determine the effect of coal rank.
- Multiple pulse experiments should be conducted as they have potential to convert more coal.
- Experiments should be conducted with micronized coal to increase the rate of combustion.

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APPENDIX I

STATISTICAL ANALYSIS OF EXPERIMENTS

The mean square error for coal carbon conversion and pressure peak were obtained in the following manner. (10)

Sum of squares of the error (SSERR)

$$= \sum_{i=1}^n \sum_{j=1}^{m_i} (x_{i,j} - \bar{x}_i)^2$$

where $x_{i,j}$ = experimental data point at condition i with j replicates

$\bar{x}_i$ = mean of m_i experimental data obtained under identical conditions

m_i = number of data points under identical conditions

n = total number of replicate sets.

The degrees of freedom (df) for the error term is given by:

Degrees of freedom (df) = Total number of points - Total number of means.

$$\text{Mean Square Error} = \frac{\text{SSERR}}{\text{df}}$$

To obtain the mean square of the main effect of factors, the following procedure was adopted.

Let S = sum of response of all factorial experiments at the higher level of a factor minus sum of response of all factorial experiments at the lower level of the same factor.

Then sum of squares of the main effect of the factor

$$= \frac{S^2}{\text{Number of Factorial Experiments}}$$

Degrees of freedom for each main effect is equal to one.

$$\text{Mean Square} = \frac{\text{Sum of Squares}}{\text{Degrees of Freedom}}$$

For example, the mean square of factor A (ϕ_{CH_4}) on coal carbon conversion is obtained as:

$$SSA = (ab + ac + ad + ae + abcd + abce + abde + acde - (1) - bc - bd - be - cd - ce - de - bcde)^2 / 16$$

(here 'ab' represents the coal carbon conversion for experiment conducted at higher levels of A and B and lower levels of C, D, and E).

Degrees of freedom for main effect of A = 1

$$\text{Mean Square for A} = \frac{SSA}{\text{Degrees of Freedom}}$$

The F_{exp} is next calculated as:

$$F_{exp} = \frac{\text{Mean Square of Main Effect}}{\text{Mean Square of Error}}$$

This value of F_{exp} is compared to the tabulated values of F (F_{tab}) obtained from standard statistical F tables. (11)

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SLAGGING RETROFIT PULSED COAL COMBUSTOR

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