DEC 0 3 1990

FINE PARTICLE COAL AS A SOURCE OF ENERGY IN SMALL-USER APPLICATIONS

DOE/PC/89904--T20

by

DE91 005883

S. Rajan
Department of Mechanical Engineering and Energy Processes
Southern Illinois University at Carbondale
Carbondale, Illinois 62901

Final Report
January 1, 1989–August 31, 1990

Submitted to
U.S. Department of Energy
Contract Number DE-FC22-89PC89904

November 1990

MASTER

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, mark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

ABSTRACT

The use of fine particle micronized coal as a source of energy for home heating applications has been explored in previous years under this program in a 150,000 Btu/hr pulse combustor. Experimental studies have been conducted on the combustion characteristics of micronized coal and combustion efficiencies have been measured. Emission levels of NO_x and SO_2 have been measured. In this final year of the program, the combustion and emissions characteristics of micronized coal were further explored in terms of the influence of stoichiometric ratio and frequency effects. Also, a model has been proposed which has potential for incorporating the unsteady mixing occurring in pulse combustors.

INTRODUCTION

Energy consumption in the residential and commercial sector is projected to grow at the rate of 1.7 to 1.8% per year between 1986 and 2000. Interestingly, the growth rates for the industrial and transportation sectors are lower, on the order of 1.2 to 1.6% and 0.1 to 0.6% respectively. The oil price hikes in recent years have given us warning to become self-sufficient in our energy resources and production. The residential and commercial sector uses about 72% gas, 26% oil, and 2% coal. Figure 1 shows a comparison of utilization of natural resources by various sectors. The current use and future projections indicate a strong need for developing suitable advanced technologies to increase the share of coal in this sector. Pulse combustors have shown promising trends of utilizing coal in a clean and efficient manner for small-scale applications. However, burner technologies for such combustors are not fully developed, and a lot of research and developmental work is required before small coal combustors will be available for residential use.

Pulse combustors gained prominence during World War II, when the Germans used them to propel V-1 rockets, more popularly known as the "buzz bombs." But it was after the energy crisis that serious efforts began to exploit this unique principle. In the early 1980s, two U.S. companies, Lennox and Hydrotherm, utilized this unique principle to design space and water heaters. This successful commercial application generated a lot of interest in pulse combustors and the concept gained credibility. Recent developments in the gas furnace industry (U.S. Department of Energy 1979, Gas Research Institute 1982) have demonstrated some of the inherent advantages of pulsating combustion, namely high combustion efficiency, high volumetric heat release rates, high heat transfer rates, and low levels of oxides of nitrogen among other features.

Figure 1. Utilization of natural resources by various sectors.

In the case of a solid fuel like coal, where the combustion sequence involves devolatilization and gaseous combustion followed by heterogeneous reaction at the particle surface, the benefits of pulsating combustion are even more readily apparent. Although while burning pulverized coal, the surface to volume ratio increases, the chemical reaction is not self accelerating and slows down as the oxygen concentration diminishes. The to-and-fro gas motion induced by the pressure fluctuations serves to remove the products of combustion like carbon monoxide and carbon dioxide from the coal particle surface, thus exposing the particle to fresh oxygen for reaction. This increases chemical reaction rates and heat transfer to coal particles, thus promoting better combustion.

While much of the experimentation and commercial applications of pulsating combustion has been with gaseous fuels, very little work has been done with coal. Investigators have demonstrated very high combustion intensities, on the order of two million Btu/hr ft³, and combustion efficiencies ranging from 91 to 97%. However, there is very little data available demonstrating the influence of air/fuel ratio and resonant frequency on the combustion efficiencies. In spite of the discovery that NO_x levels are inherently low in pulse combustors, very few actual levels have been reported in the literature, both for coal as well as gaseous fuel.

A pulverized coal or coal-water slurry fired pulse combustor promises to be a compact, efficient burner with high combustion intensities and good ash rejection characteristics. Since coal-water slurries, like commercial oils, can be cheaply and easily transported, this feature offers some advantages over burning micronized coal in dry solid form. However, burner technology for burning coal-water mixtures in pulsating mode is not well developed.

With the help of previous funding, a 150,000 Btu/hr pulse combustor suitable for small-scale applications has been designed and fabricated. Detailed experiments have been performed with gaseous propane fuel and with beneficiated micronized

coal. The effects of varying equivalence ratio and combustor frequency on the combustion efficiency and emissions have been studied. The results have been described in previous reports.

In this final year of DOE funding, the micronized coal experiments have been extended. In order to provide a better understanding of some of the fundamentals of pulse combustor operation, a literature search was conducted in both the experimental and pulse combustor modeling areas. A new approach towards pulse combustor modeling has been proposed and a computer program developed.

LITERATURE SURVEY

Pulse Combustion - Review

Combustion-driven oscillations were first observed by Higgens around 1777 (Tyndall 1987), and subsequently the phenomena was explained by Lord Rayleigh in 1878. Soon after this, efforts began to utilize the combustion-driven oscillations, instead of suppressing them as was the usual practice in conventional burners. The initial application of pulse combustors seems to be more as a thrust generating device rather than utilizing its various advantages for steam generation or other purposes. Propulsion devices were successfully developed based on the pulse principle (Thring 1961, Putman et al. 1986) and as mentioned earlier, similar devices were used by the Germans to propel V-1 jets in WW-II. Reynst has reported that after the war when the gasoline shortage became acute, H. Wahl tried to modify the V-1 to burn pulverized coal; and subsequently Wahl was also involved in designing of a gasification plant based on the pulse principle (Thring 1961).

Principle of operation. The fundamental principle which is necessary for the successful operation of a pulse combustor is the Rayleigh criterion (Rayleigh 1945). This principle states that the periodic heat addition to a standing wave will produce the strongest driving effect if the addition is in phase with the pressure wave and it takes place at the point of maximum pressure variation or the pressure antinode. This condition is satisfied by locating the combustion zone in a region where the magnitude of the pressure oscillations is significant and by allowing the pressure fluctuations to modulate the supply of air or fuel, or both.

Advantages of pulsating combustion. The major advantages of pulse combustion of interest regarding the combustion of gas, pulverized coal, and other similar solid fuels are as follows:

1. higher combustion intensities, which enable us to use smaller combustion chambers for the same heat release rate;
2. higher combustion efficiencies, resulting from better mixing and the periodic stripping of products of combustion from the surface of the coal particle. Because of higher combustion efficiencies, lesser amounts of fuel are used, which results in lowering of operating costs.
3. high heat transfer rates, leading to smaller heat transfer surfaces thereby reducing the equipment cost; and
4. low levels of oxides of nitrogen, carbon monoxide and hydrocarbons, which will enable the combustor to meet the air quality emission standards.

Further, the self aspirating action of the combustor eliminates or reduces the need for a forced-draft fan in certain designs. It has also been reported that the periodic pulses result in a scrubbing like action that helps to remove the products of combustion, and also reduce slagging, thus keeping the heat transfer surfaces clean.

Designs. Pulse combustors can be broadly classified into three major categories, depending on their design and operating cycle (Zinn 1985). These are the quarter wave or Schmidt type combustor, the Helmholtz combustor, and the Rijke combustor.

The quarter wave or Schmidt type combustor essentially behaves like a one end open and one end closed organ tube. As shown in Figure 2, the acoustic pressure oscillations excited within this tube have their maximum amplitude at the closed end and minimum at the open end. Initially, a charge of air and fuel is introduced in the combustion chamber and is spark ignited. Combustion takes place with a pressure rise, the one way flapper valve closes, and products of combustion tend to go out from the open end. The outflow of combustion products creates a low pressure

Figure 2. Operating cycle.

region near the closed end. This enables the one-way valve to open admitting fresh charge and bringing some of the hot gases back from the exhaust. The mixture of fresh charge and hot combustion products combine and explode, thus generating a pulse. The cycle repeats itself yielding a series of pulses whose frequency depends on the length of resonance tube or tail pipe. To satisfy Rayleigh's criterion the combustion must take place near the closed end where the pressure is maximum. Therefore, care must be taken to design the combustor in such a manner that the duration of mixing and reaction processes are such that heat is released near the end region. The German V-1 rockets operated on the same principle except that in some designs the mechanical flapper valve was replaced by an aerodynamic valve.

The Helmholtz combustor resembles the quarter wave combustor in principle. This combustor has a different geometry, as seen in Figure 3. It has a separate combustion chamber and a long resonance tube which is of a smaller diameter. The air and fuel valves are attached to the main combustor as shown. Commercial space and water heaters are mostly based on this type of design (Zinn 1985, Whitlock et al. 1982, Adams 1982).

The Rijke combustor is based on the Rijke tube (Tyndall 1987, Rayleigh 1945). It essentially consists of a fixed bed located at a distance of $L/4$ from the bottom of the tube (where L is the total length of the tube). The combustion process and the periodic oscillations are governed by the Rayleigh's criterion. Zinn et al., at Georgia Institute of Technology, have successfully burned unpulverized coal in pulse combustors based on this design (Zinn et al. 1982). Observation of the combustion zone by Zinn et al. has revealed that the burning coal particles move in a spout like motion that is almost like the motion of particles in a typical fluidized bed combustor. This enhances the mixing of fuel and air which results in very high heat release rates. Mixtures at very high equivalence ratios have been successfully burned in such combustors, which suggests that these combustors could be used for gasifying coal.

Figure 3. Types of pulse combustors.

Valve Designs

Typically three different types of valves have been used in pulse combustors. A brief description of each type follows.

Mechanical or flapper/reed valves. Schmidt was perhaps one of the earliest investigators to use this type of valve in pulse combustors designed for generating thrust. Figure 4 shows a few designs incorporating these valves. The valves may be freely moving flapper valves, or may be designed with a restoring force which keeps them normally closed. Reed valves employ springy metal leaves, fixed at one end and loose at the other. These valves have to be designed for proper performance at the combustor frequencies and to prevent fatigue failure. One of the major advantages of flapper valves is that these can be adjusted to get the desired air flow by simply shifting the spacer or tightening the nuts, whatever may be the mechanism. In certain designs mechanical valves were preferred over aerodynamic valves because working models were easy to fabricate and there were no backflow problems. In case of mechanical valves the closing of the valve during about half the cycle also reduces the amount of noise created compared to an aerodynamic valve. Severyanin (1965) has suggested the use of bell shaped flapper valves for better mixing characteristics. Most of the commercial small-scale heaters employ this type of valve.

Rotary valves. Rotary valved units are less complicated and are not as susceptible to fatigue failure and radiation as flapper valves. The design is similar to the flapper valve design in that rotary valves make a positive seal during the explosion part of the cycle. Kunsagi has done a number of tests using rotary valved pulse combustors (Putnam et al. 1986). The rotation of the valve fixes the frequency of fuel or air and the valve is generally rotated with the help of an electric motor. Kunsagi's design is shown in Figure 4. It consists of a perforated disk as the air inlet valve. Muller has also done some work comparing the performances of rotary and

Figure 4. Types of air valves.

aerovalved combustors (Putnam et al. 1986). This design did not gain much popularity and these days very few combustors incorporate this valve.

Aerodynamic valves. A lot of work on aerodynamic valves was done by SNECMA in the 1940s (Putnam et al. 1986). Figure 4 shows a typical aerovalved SNECMA pulse combustor. These valves are very simple in construction as they have no moving parts. The only problem that needs to be tackled is the backflow of products of combustion during the explosion part of the cycle. Essentially an aerovalve consists of an inlet passage which is intended to offer a much higher resistance to backflow than to inlet air. Kentfield (1982) has done a lot of work on single and multiple inlet combustors employing aerodynamic valves (Putnam et al. 1986, Kentfield 1982). As compared to other valves this type of valve is rather difficult to design and tune in order to get proper combustion efficiency.

Applications of Pulse Combustors

Pulse combustors have been used for various applications ranging from propulsion devices to space heating. Some of the applications are listed below:

Trust generation	Propulsion, vertical lift devices, surface cleaning and descaling
Fluid pumping	Pumping insecticides and fog oil, flue gas recirculation
Heating of liquids	Space heating, steam raising, water heating, petroleum processing
Indirect heating of air	Space heating of vehicles, homes, and industries
Direct heating	Drying in food industries; drying of blood, wood, and waste materials

Other uses

Refuse incineration, snow melting,
gasification, pilot burners, wood dust
incinerators, etc.

Combustion of Solid and Gaseous Fuels

While most of the experimentation and commercial applications of pulsating combustion has been with gaseous or liquid fuels (U.S. Department of Energy 1979, Gas Research Institute 1982), very little work has been done on the combustion of solid fuels.

F. H. Reynst was among the pioneers who advocated the use of pulse combustors for burning pulverized coal. Several of his papers dealing with combustion of pulverized coal are published in his collected works, edited by Thring (1961). The following section discusses in detail some of the operating parameters as investigated by workers experimenting with pulse combustors.

Combustion efficiencies and intensities. As mentioned earlier, among the major advantages of pulse combustion are the high combustion intensities and high combustion efficiencies. The handful of investigators who have experimented with combustion of coal in the pulsating mode have all demonstrated this fact. Based on the work done by Reynst, Sommers built and tested a pulverized coal pulse furnace capable of burning 4400 lbs/hr of Rhineland brown coal (Thring 1961). The combustor demonstrated high combustion intensities on the order of two million Btu/hr ft³.

In 1968, Hanby and Brown at the University of Sheffield, developed a small (0.5 ft³) conical, pulverized coal combustor, capable of firing pulverized coal at a feed rate of 50 lbs/hr (Hanley and Brown 1968). They measured combustion intensities on the order of 0.8×10^6 Btu/hr ft³ with carbon burnout ranging from 20 to 97%. In research sponsored by the Bureau of Mines, Belter et al. burned lignite dust in a 35

ft. long, 25 in. diameter pulse combustor, with combustion intensities of 0.5×10^6 Btu/hr ft³ (Belter et al. 1969). Zinn et al. (1982) investigated the combustion of large size coal particles (1/4 to 1/2 in.) in a Rijke type pulse combustor. The tests demonstrated that unpulverized coal could be burned stably and efficiently in a Rijke type pulse combustor. The investigators at Georgia Tech measured combustion efficiencies varying from 89 to 98.5%. Extensive work on pulse combustors has been going on in the USSR since WW-II. Severyanin (1982, 1965) and Babkin (1965), the foremost investigators, have also demonstrated very high combustion intensities and efficiencies in pulverized coal fired pulse combustors.

Heat transfer. Not many heat transfer studies have been done with combustors burning pulverized coal. Investigations done on gas fired pulse combustors have shown that there is a 15 to 35% increase in heat transfer rates. Hanby and Brown (1968) measured pressure, gas temperature, wall heat flux, and wall temperature at various locations and found that the heat transfer coefficients were significantly greater in pulse combustors than with steady flows. Blomquist et al. (1982) at Argonne have done extensive work on Helmholtz type pulse combustors, studying their heat transfer rates. They have reported a quasi-oscillatory pattern of increasing and decreasing heat transfer rate along the burner, which perhaps can be attributed to the pulsating nature of gas flow. Because of the pulsating flow, the heat transfer was observed to be increased by 15 to 25% at heat input rates varying from 100,000 to 54,000 Btu/hr. An increase of 20 to 35% was observed in the resonance tube. The use of a corebustor also increased the combustor heat transfer, and interestingly the increase was more in the combustion chamber rather than in the resonance tube. This may possibly be due to suppression of pulses by the corebustor. West and Taylor (1952) have also observed higher heat transfer rates in similar combustors. In another study Alhaddad and Coulman (1982) found that the

amplitude of pulsations was the main parameter influencing the heat transfer coefficient.

Gaseous emissions. With regard to the acid rain problem, the NO_x and SO_2 emission levels have become very important from the environmental standpoint. The discovery that pulse combustors produce very low levels of NO_x has not only increased its utility in small-scale applications like home heating, but has also created interest in other applications. A few investigators have reported data on gas fired pulse combustors, but it appears no levels of NO_x and SO_2 have been published for coal fired combustors. Putnam et al. (1986) have reported very low levels of NO_x in the range of 10 to 20 ppm, with natural gas and propane. It was also observed that the levels of NO_x increased with the increase in fuel feed rate, and the levels did not change appreciably with the change in frequency. The primary reason for low NO_x emissions in a pulse combustor is perhaps that the mixing rates are very high. High mixing rates lead to very rapid combustion which is followed by rapid quenching which suppresses NO_x formation. Typically, residence times in a pulse combustor can be as low as a few milliseconds depending on the frequency of operation.

Noise levels and vibrations. Possibly the only disadvantage associated with pulse combustors is the noise level and the vibrations due to the pulses. Noise levels of about 120 to 130 dB have been mentioned in the literature. It has been observed that aerovalved units produce more noise than mechanically valved units. The noise levels have also been linked with the pressure in the combustor; lowering the pressure rise in the combustor lowers the noise levels. Extensive work on controlling and understanding the noise levels produced was done by A.G.A laboratories in order to set guidelines for combustor design (Belles et al. 1982). Blomquist and Clinch (1982) measured sound levels using a Kjaer Model 2218 precision integrating sound level meter and observed levels ranging from 70 to 96 dBA. These levels were reduced with the help of inlet and outlet mufflers. Without the use of any muffling

devices, Belles et al. (1982) at A.G.A. have observed values of 77 and 91 dBA at the outlet and inlet, respectively. They also observed that the fuel input rate had a strong influence on the noise level, other factors which effected the levels were the exhaust and tail pipe areas, air valve spacing, and the volume of the exhaust decoupler. Zinn et al. (1982), while burning unpulverized coal in a Rijke type pulse combustor, measured noise levels from 145 to 165 dB at various air-fuel ratios. Maximum noise levels were measured at air-fuel ratios close to stoichiometric. The increase in noise levels quickly increased the CO₂ levels and decreased the CO levels, suggesting that the higher acoustic amplitudes which created more noise also helped to create more turbulence leading to a more efficient combustion process. Although higher noise levels are acceptable to defense and industrial users, the levels can be a problem with residential consumers. It has been observed that by using proper muffling devices the noise levels can be brought down to a great extent.

Factors Effecting Coal Firing in Pulse Combustors

As coal is a very slow burning fuel and it burns in a heterogeneous manner, certain factors and design parameters have to be given special attention for coal fired pulse combustors. Some of these factors are discussed below.

Air fuel mixing. For efficient combustion of coal, the mixing between coal particles and air should be as turbulent as possible. Suitable design of air valve(s), air inlet(s), and fuel feeding tube(s) will insure proper mixing. Hanby and Brown (1968) used two tangentially mounted pulverized fuel injectors to insure proper mixing. Air for combustion in their apparatus was introduced through a manifold, which aided in efficient mixing. Severyanin (1965) has used several techniques to increase mixing in his designs which include using a bell shaped flapper valve to introduce air, using pear shaped mixing chambers, and using four tangential tubes spaced 90

degrees apart tubes for introducing air. In a recent paper Severyanin (1982) has suggested other methods like floating plate damper, dipole devices, toroidal antechambers, etc.

Residence times. The length of the resonance tube should be such that coal particles have enough time to burn completely. This factor becomes all the more important as coal is a very slow burning fuel. Residence times are inversely proportional to the frequency of the pressure oscillations and the frequency is inversely proportional to the length of the resonance tube. A very high frequency combustor will give very little time for coal to burn. On the other hand, very low frequencies will yield low temperatures in the heat transfer areas and will also result in a very bulky design. Therefore optimum frequencies have to be selected to insure efficient combustion. Hanby and Brown (1968) burned pulverized coal efficiently at a frequency of 75 cycles/sec, which translates into a residence time of 0.013 sec. Such low residence times produce efficient combustion only because of the unique pulsating principle. Other investigators like Sommers (Thring 1961) and Severyanin (1965) have successfully burned pulverized coal in pulse combustors with frequencies ranging from 40 to 60 cycles/sec.

Particle size. The smaller the coal particle size, the greater the surface area to volume ratio. A larger surface area results in higher reaction rates leading to higher combustion intensities and better carbon burnout. For small-scale pulse combustors of the Helmholtz type, coal particle sizes varying from 73 to 200 mesh have been successfully burned. Coal size of 1/2 to 1/4 in. has been burned in a stationary bed Rijke type pulse combustor by Zinn et al. (1982) at Georgia Institute of Technology.

Coal type and rank. As with other conventional combustors, the coal type and rank are very important while designing combustion chambers and heat transfer areas. Sommers burned high ash Rhineland coals (Thring 1961) and Hanby and Brown (1968) designed a pulse combustor to burn high volatile British coals. Even

low Btu lignite has been combusted satisfactorily by Belter et al. (1969) at Baukol-Noonan Inc. mine.

Ash rejection characteristics. The flow pulsations in a typical pulse combustor result in a scrubbing like action which helps in removing coal ash from the heat transfer areas. The design of the resonance tube and its orientation should be such that the pulsating action is utilized to efficiently remove products of combustion. Ash removal spouts or bins should be located at convenient positions to remove ash periodically. While designing the collection bins care should be taken to insure that the acoustics of the resonance tube are not changed or disturbed in a way which would hinder efficient combustion.

EXPERIMENTAL SET-UP

Apparatus

In this section, the equipment used in the tests will be described in detail.

The schematic diagram of the complete set-up is shown in Figure 5. The main components of a pulse combustor are as follows:

1. air supply systems,
2. flapper valve,
3. combustion chamber,
4. fuel supply arrangement,
5. resonance tube, and
6. exhaust decoupler.

Each component is described in the following paragraphs. The operational procedures employed in the tests and other parameters are given in subsequent sections of the report.

Figure 5. Layout of pulse combustor test rig.

Air supply system. The inlet air is drawn through a 55 gal drum which serves to dampen the air flow pulsations. An ASME nozzle mounted at the inlet of the drum is used to measure the intake air mass flow rate by measuring the pressure drop across the nozzle. The pressure drop is measured with the help of a high precision Dwyer magnehelic gauge. When the pressure in the combustor falls below atmospheric, the air enters into the combustor via the one way flapper valve.

Flapper valve. A flapper valve similar in design to the one used by Severyanin (1965) is used to admit air into the combustor. The valve consists of a stainless steel bell-shaped flapper and a 12 in. long shaft as shown in Figure 6. The shaft is mounted on a web with adjustment nuts on both sides which can be loosened or tightened to increase or decrease the opening of the valve in order to control the amount of air going into the combustor. The valve sleeve is mounted on the top section of the combustor, this too is made adjustable to locate the valve at the best mixing position.

When firing micronized coal the air valve was slightly modified, the valve shaft was removed and the coal inlet tube served as the valve shaft as shown in Figure 6b. In this case, only the flapper moved up and down instead of the whole flapper-shaft assembly as in the earlier case. An adjustment sleeve was provided on the coal feeding tube to control the opening of the flapper valve. The details of the coal inlet tube and the coal feeder are discussed in the section "Coal Feeding Arrangement."

Combustion chamber. The combustion chamber was designed for a heat output of approximately 150,000 Btu/hr. Based on the volumetric heat release rate of one million Btu/hr ft³ as measured by other investigators working on pulse combustors, (Zinn et al. 1982, Severyanin 1965, Hanby and Brown 1968) the combustor volume was designed to be about 0.15 ft³. Figure 7 shows the details of the combustor cross section. The combustion chamber consists of 6 sections; the 6 in

Figure 6a. Air flapper valve.

Figure 6b. Flapper valve and coal feeding tube.

Figure 7. Cross section of combustion chamber.

top section; 4 middle sections of 5, 3, 2, and 2 in., respectively; and a bottom section of 5 in. The middle sections were designed such that they could be removed or inserted to study the effects of combustor volume on volumetric heat release rates and emissions. The inner profile of the first and the last section is made conical, so that fresh air can flow in and products of combustion can move out smoothly from the combustion chamber.

A Kistler 7061 water-cooled pressure transducer is mounted on the 5 in. section to monitor the pressure fluctuations in the combustion chamber. The signal from the pressure transducer is taken through a charge amplifier to a storage oscilloscope and it is also recorded in an IBM data acquisition system. Arrangements were made to take direct photographs of the pressure wave on a storage oscilloscope.

K-type chromel-alumel thermocouples (1/16 in. diameter) were mounted on each section of the combustor to study the temperature variation along its length. The thermocouple output was connected to a digital readout through a selector switch. The selector switch was used to read the various temperatures in the combustor and the resonance tube.

A standard spark plug attached on the 5 in. section of the combustor served to ignite the combustible mixture in the beginning of the test runs. During the later stages of the experiments, due to problems of removing the plug, it was replaced by a igniter made out of tungsten wire encased in a ceramic sleeve. On the same section two quartz glass portholes are provided to view the combustion process. An "L" shaped nozzle serves as the inlet for propane in the combustor, the details of which are shown in Figure 8. The entire combustor is lined with a 2 in. thick moldable ceramic-base insulation, the inside of which is painted with a hard ceramic paint.

Gas feeding arrangement. Commercial-grade propane was used during the experiments. The compressed propane tank was kept outside the building for safety

Figure 8. Cross section of 5 in. section showing "L" shaped gas inlet tube.

reasons. A 3/8 in. copper tubing from the tank served as the supply line. A variable area flowmeter with a regulating valve was used to measure and control the fuel feed rate. The fuel feed rate could be controlled accurately from 0 to 60 CFH. As mentioned earlier, an "L" shaped nozzle is used as the fuel inlet. The tip of the nozzle is perforated for better mixing of fuel with air. For safety purposes a feed check valve was installed in the supply line. Besides this valve, wire meshes were placed at a number of joints to prevent the flame from propagating backwards.

Coal feeding arrangement. Beneficiated coal is fed to the pulse combustor from a fluidized bed coal feeder which is a scaled-up version of the type used by Raghavan and Rajan in previously related coal combustion studies (Raghavan 1986). Similar feeders have been suggested and used by Hamor and Smith (1979), Milne and Beachey (1977), and Altenkirch et al. (1978). Figure 9 shows the details of this feeder with various rotameters and pressure gauges.

The coal storage column is made of two cylindrical sections of Plexiglas, each of internal diameter 3 and 5 in., respectively. The coal is fluidized with the help of compressed air admitted from the bottom of the feeder through a sintered porous plate. A filter is fixed at the top plate of the storage column to filter the exhaust air before it passes through the pressure gauge and the rotameter. The high pressure in the column serves to remove the mixture of coal and air through the off-take tube as shown in Figure 10. By changing the pressure in the bed the feed rate could be increased or decreased. The bed pressure was controlled either by varying the fluidization air flow or by varying the exhaust air flow or by varying both. A "Z" shaped perforated copper tube located below the off-take tube, served to create local turbulence to prevent straight channeling of air to the off-take tube.

The off-take tube similar to the one used by Rajan et al. (Raghavan 1986) was used for removal of coal from the feeder. Suspended coal entered through a 3 mm tube and was transported with the help of dilution air. Dilution air entered into the

Figure 9. Schematic of fluidized bed coal feeder.

Figure 10. Details of fluidized bed coal feeder.

off-take tube through a number of holes drilled on the 3 mm off-take tube as shown in Figure 11. Besides transporting the suspended coal, the dilution air also served to reduce or increase the coal feed rate. Increasing the dilution air decreased the coal feed rate. A decrease in the dilution air tended to increase the feed rate. To get the desired feed rates of 15 to 75 g/min various coal feed tubes were tried out before optimizing the diameter. An on-off brass valve was fixed at the end of the off-take tube to purge the tube in case any clogging occurred at the inlet.

Fluidizing air, exhaust air, secondary turbulence air, and dilution air were all monitored with the help of Cole Palmer variable area rotameters. Rotameters also were used to control and meter all the necessary air flows and the readings were used to calculate the air-fuel ratios.

The whole unit was suspended through a wheatstone bridge type load cell. The signal from the load cell was processed through an amplifier and taken to a strip chart recorder. The recorder continuously monitored the coal feed rate. During a typical run if the coal feed rate was constant a steady slope could be observed as shown in Figure 12. A small change in the coal feed rate could be quickly noticed on the strip chart and necessary changes were made in the bed pressure to bring the feed back to the same rate.

Teflon tubing was used to transport coal from the off-take tube to the coal injector. Teflon was selected because of its non-sticking properties with micronized coal. Coal enters into a 1/4 in. stainless steel tube, mixes with the secondary air and is injected into the burner through a stainless steel distributor. The distributor helps in better mixing of the fuel and air which enhances the combustion efficiencies. This design and number of holes were arrived at after extensive experimentation with different types of distributors and by studying the pressure amplitudes and composition of ash. The secondary injection air is also monitored and can be controlled if required. The complete injector is coaxial to the air inlet sleeve as

Figure 11. Details of off-take tube for feeding micronized coal.

Figure 12. Typical load cell signal output.

shown in Figure 8. The design is such that the position of the coal distributor can be changed to locate it at the best possible section, where maximum pressure amplitudes are achieved and carbon burnout are highest.

Resonance tube. The products of combustion are exhausted through a 2 in. diameter stainless steel resonance tube. The resonance tube is comprised of several small sections, to enable the length to be changed for studying the effects of frequency on combustion and emissions characteristics. Combustion tests were made using tube lengths of 4.5, 6.75, 7.16, and 9.5 ft, which corresponded to approximate frequencies of 28, 35, 36, and 47 cycles/sec. K-type thermocouples were fixed in each section to plot temperature profiles.

Exhaust decoupler. A baffled exhaust decoupler at the end of the resonance tube was used to reduce the noise levels and to collect ash particles. An induced draft fan was used to aid the removal of combustion gases from the decoupler, as the distance to the stack was about 30 ft.

Gas and ash sampling. Two stainless steel water jacketed probes are employed for particulate collection. One of them is located at the end of the vertical portion of the resonance tube (Figure 5) and the other is located at the end of the tube, at the entrance to the exhaust decoupler. The ash analyses from the two probes help to determine the extent of combustion taking place in the resonance tube. The temperature of the sampling probes was maintained at 150° C so that no water condensed in the tubes. An in-line Teflon filter with 10 µm Teflon membranes is used to collect ash particles. After each run the ash samples were collected and the filter membrane was replaced to prepare for the next run. The flow of gases through the probe is measured and regulated accordingly, to insure isokinetic sampling. A bypass valve on the sampling probe is connected to compressed air to purge the lines before and after each sample collection.

Gaseous emissions are sampled from a different tap located at the end of the decoupler. An in-line Teflon filter with Teflon membranes is used to prevent any particulates from entering the analyzers. Figure 13 shows the sampling system with the gas analyzer connections. The gas analyzers are connected in parallel and the sample gas is transported using heated 1/4 in. stainless steel lines. A Teflon diaphragm pump is used to deliver the gas sample to the analyzers. Another filter is located before the condenser which helps in delivering a clean gas sample to the bank of analyzers consisting of Beckman CO₂, CO, O₂, SO₂, and hydrocarbon analyzers. A separate heated line is connected to the NO/NO_x analyzer, which bypasses the condenser. The signals from all the analyzers are taken through an A/D converter to an IBM AT/PC data acquisition system which records all the readings. A schematic of the data acquisition system is shown in Figure 14.

EXPERIMENTAL PROCEDURE

The following paragraphs describe the experimental procedure for burning gaseous propane and beneficiated micronized coal.

Start-up on Propane Gas

Commercial-grade propane was used during these experiments. The procedure followed during a typical run on propane is given below.

1. The induced draft fan was switched on to purge the system of any left-over exhaust gases from the previous tests and to insure that the unburnt propane is quickly removed in case the combustion mixture fails to ignite. The cooling water to the pressure transducer was also switched on before igniting the propane in the combustor.
2. The second step was to switch on the spark plug which was located in the 5 in. section of the combustor as described in the previous sections.

Figure 14. Data acquisition system.

3. Next, propane was admitted by opening the control valve and the mixture was allowed to ignite. In the case that the mixture failed to ignite, the spark plug was switched off and the gas valve closed. The system was allowed to purge for 3 to 4 min. and then the second and third steps were repeated to ignite the mixture.
4. After stable pulsations were achieved and the combustor was sufficiently heated, the spark plug was switched off. The system was allowed to be operated on its own and as mentioned earlier the backflow of the hot exhaust gases were now responsible for ignition of the fresh charge.
5. On achieving continuous stable operation at a particular equivalence ratio, all the temperatures in the combustor and resonance tube were noted down, the pressure signals were recorded on the computer, photographs of the pressure signal were taken, air and fuel flows were recorded and the emission data was also collected on an IBM AT/PC data acquisition system.

Experiments on Coal

The coal used for tests was a subbituminous coal from the Illinois basin. Three coals were used which are designated as originating from mine nos. 1, 3, and 4. The beneficiated coal used in the tests was cleaned and micronized commercially to pass through a 325-mesh sieve. The initial ash content of 10.61% was decreased to 0.85% after the commercial coal beneficiation processes. The ultimate and proximate analyses of the raw coal, various intermediate stages and the final micronized coal are given in tables 1 through 3 for the three coals tested. The intermediate samples of physical beneficiation process included those from the preparation plant (PPO), the low ash middlings (LA), and the high ash rejects (REJECT). The physically cleaned sample was then chemically cleaned (CC) by treating with hydrofluoric acid and the

final micronized sample (micro) was used in the tests. The chemical cleaning was done by Hazen Research Corporation, Colorado.

Table 1
Proximate and Ultimate Analysis Mine 3

Parameter	PPO	LA	REJECT	CC	Micro
Proximate Analysis (%)					
Moisture	5.97	5.9	2.7	4.47	
Fixed carbon	48.22	51.71	57.61	54.76	
Volatile matter	35.20	37.19	36.80	39.75	
Ash	10.61	5.41	11.81	1.16	0.85
Ultimate Analysis (%)					
Carbon	67.33	71.52	65.53	71.85	72.77
Hydrogen	4.87	5.16	4.62	4.99	5.02
Nitrogen	1.78	1.39	1.38	1.48	4.18
Oxygen	26.02	21.39	28.47	21.68	20.73
Heating value, Btu/lb.	11,898	12,649	12,331	13,470	13,434
Total Sulfur (%)	3.43	2.47	3.66	2.97	2.69

Table 2
Proximate and Ultimate Analysis Mine 4

Parameter	PPO	LA	REJECT	CC	Micro
Proximate Analysis (%)					
Moisture	5.24	3.32	3.45	4.14	4.1
Fixed carbon	57.61	58.60	58.50	60.30	60.03
Volatile matter	32.11	34.29	32.87	34.87	35.22
Ash	5.04	3.75	5.31	0.69	0.65
Ultimate Analysis (%)					
Carbon	74.66	75.04	73.47	78.08	77.43
Hydrogen	4.81	4.94	4.82	5.03	5.09
Nitrogen	1.79	1.82	1.77	1.87	1.8
Oxygen	18.74	18.20	19.94	15.02	15.59
Heating value, Btu/lb.	12,984	13,417	13,231	13,810	13,804
Total Sulfur (%)	0.92	0.84	0.94	0.94	0.96

Table 3
Proximate and Ultimate Analysis Mine 1

Parameter	PPO	LA	REJECT	CC	Micro
Proximate Analysis (%)					
Moisture	2.22	2.29	2.39	3.37	
Fixed carbon	57.87	60.34	57.81	60.56	
Volatile matter	33.13	33.97	32.78	34.95	
Ash	6.78	3.45	7.41	0.76	0.68
Ultimate Analysis (%)					
Carbon	73.96	76.16	73.12	78.43	78.34
Hydrogen	4.88	4.99	4.89	5.21	5.16
Nitrogen	1.66	1.75	1.67	1.80	1.76
Oxygen	19.50	17.10	20.32	14.56	14.74
Heating value, Btu/lb.	13,130	13,798	13,056	14,016	13,944
Total Sulfur (%)	1.34	0.99	1.38	1.02	0.96

Before burning coal in the combustor the fluidized bed coal feeder was calibrated in order to control the feed rate by adjusting various air flows. The micronized coal was dried in a vacuum oven to remove the moisture. Dried coal was then sieved to make sure no oversize particles are present in the feeder to avoid any clogging problems. Dried and sieved coal was finally loaded in the coal storage column and calibration was carried out. Different diameter coal feeding tubes were used to achieve the proper coal feeding rates. A typical test procedure on combustion of coal follows.

1. The pulse combustor is started on propane gas by following the procedure given in the previous section.
2. After the combustor is heated sufficiently and steady temperatures are maintained in the combustor on the order of 900 to 1100° C, coal is slowly introduced in by controlling the bed pressure or by varying the dilution air of the fluidized bed coal feeder. It is allowed to co-fire with propane

until stable pulsations are achieved on coal. On stable operation with coal, gaseous propane is switched off.

3. Before collecting the data the water cooled ash collecting probes are purged, the elbow ash collection bin is cleaned and the stopwatch started. The ash samples are collected isokinetically; therefore, the collection flow rate is calculated beforehand. After the initial few runs, it was noticed that the probes were not getting heated enough; therefore, instead of water circulation, heating tapes were used to maintain the temperature at about 150° C. The probe pump and stopwatch are started simultaneously. Temperature, pressure, and various flows from the feeder are recorded. The secondary air flows are also recorded. Photographs of the pressure wave are taken and the exhaust gas emission data are recorded directly into a IBM PC/AT data acquisition system. After a sufficient run time of about 8 to 10 min., depending on the duration of the test, the ash from the main bin and from the two probes is collected in 20 mL vials for analysis. New Teflon membranes are put in the filters for subsequent runs.
4. To obtain the second set of data the fuel feed rate is changed by changing the bed pressure by either varying the fluidizing air or the exhaust air. Finer control on the fuel feed could be obtained by changing the dilution air. In all the tests the air mass flow rates were kept constant and only the coal feed rates were changed to get different equivalence ratios. Tests were made using equivalence ratios ranging from 0.6 to 2.3. After changing the equivalence ratio, data was collected in the same manner as described in step 3.

Data Collection

All necessary precautions and procedures were followed to get good data and analysis. The oxygen, carbon dioxide, carbon monoxide, sulfur dioxide and hydrocarbon analyzers were switched on at least 12 hr before the run. This was a general procedure recommended by the manufacturers to insure proper warming of the analyzers, before calibration. The NO-NO_x analyzer was switched on 2 to 3 hr before the experiments. All analyzers were calibrated using reference gases before the tests, and often times the calibration was rechecked during the experiments to insure good data.

The gas sample line and filters were heated with the help of electric heating tapes. This was done to make sure no water condensation took place before the gases reached the condenser. The heating tapes were switched on a couple of hours before the run was scheduled and the transformers were adjusted to heat the lines to about 80° C. Thermocouples measured the line temperatures at various sections and were monitored to insure proper heating. It was also made sure that the gas line condenser was filled with sufficient ice to remove water from the gas sample.

To insure uniformity of data, all the readings were noted down simultaneously. For a particular set at least 3 to 4 readings were recorded and later on, the average was taken for calculation and graphing purposes.

Ash samples collected were sent to the Coal Technology Laboratory at Southern Illinois University at Carbondale for heating value analysis.

RESULTS AND DISCUSSION

Coal Combustion Experiments

Tests were conducted using the equipment and experimental procedures described above. The tests were aimed at extending and providing a better understanding of the micronized coal experiments performed in previous years.

Temperature profiles and combustor frequency. Figure 15 shows the measured temperature profiles within the combustor and resonance tube for the three test coals. The equivalence ratio for the three coals is essentially the same and is very close to the stoichiometric value. The data shows the effect of (a) coal type and (b) pulsating frequency. Mine 3 and Mine 4 coals were burned with a resonance tube length of 9.5 ft while Mine 1 coal was tested at a resonance tube length of 7.16 ft. The corresponding frequencies of operation for the three coals as a function of equivalence ratios are shown in Figure 16. Mine 1 coal, because of its shorter resonance tube length, was operating at a slightly higher frequency than the other two coals. In Figure 15, the frequency for Mine 5 coal is higher than that of Mine 3 coal for the same resonance tube length and is the result of the influence of coal composition on combustion characteristics.

Coal composition influences combustion characteristics independent of pulse combustor operating conditions. Mine 4 coal, as seen from tables 1 through 3, is higher in carbon content, hydrogen and heating value than Mine 3 coal, comparing the chemically cleaned values in the tables. It has an ash content of 0.69% compared with 1.16% for Mine 3 coal. Although the volatiles content of Mine 3 coal is higher by about 5%, it does not appear to increase the overall reactivity of this coal. This is perhaps more so because the to-and-fro pulsating motion of the gas around the coal particle strips the boundary layer flame due to volatiles combustion from the coal particle and reduces energy feedback to the particle. On the other hand, this oscillating gas motion promotes better diffusion of oxygen to the particle interior and promotes more efficient heterogeneous char combustion reactions. The char reactivity is greater for low ash coals. The combined effect of these factors yields higher temperatures throughout for Mine 4 coal as depicted in Figure 15. Mine 3 coal exhibits the lowest temperatures throughout the combustion chamber and resonance tube.

Figure 15. Combustor temperature profile from mine nos. 1, 3, and 4 coals.

Figure 16. Pulse combustor frequencies when operating on the test coals.

As seen from tables 1 and 3, Mine 1 and Mine 4 coals have basically the same proximate and ultimate analysis and ash content. The heating value of Mine 1 coal is slightly higher, although for practical purposes it may be assumed that Mine 1 and Mine 4 coals have similar combustion characteristics based on the coal composition. The temperature profile in Figure 15 for Mine 1 coal is lower than that for Mine 4 coal even though their composition is essentially the same. This illustrates the influence of combustor operating frequency and residence time interaction. As seen in Figure 16, the shorter resonance tube length yields a somewhat higher frequency for Mine 1 coal. If the combustion characteristics of Mine 1 and Mine 4 coals are assumed similar, this shorter residence time appears to be the cause for the lower temperature at the combustor outlet for Mine 1 coal. The data of Figure 15 indicates that while higher mixing rates induced by higher frequencies may have a beneficial value, too high an oscillating gas motion may actually reduce oxidant diffusion into the porous coal matrix and thereby result in decreased conversion within the time frame of the shorter residence time.

Combustion efficiency. The combined influence of coal composition and frequency, as discussed above, on the measured combustion efficiency can be seen in Figure 17. The lower temperatures with Mine 3 coal result in slightly lower combustion efficiencies at the nominal equivalence ratio of 1. At the test frequency corresponding to the 9.5 ft resonance tube length, changes in equivalence ratio have a more pronounced effect on the combustion efficiency of the lower reactivity Mine 3 coal than on Mine 1 and Mine 4 coals. Overall, a high conversion efficiency of about 95% was measured for all the three test coals under stoichiometric conditions. This is promoted by the high surface to volume ratios of the micronized coal particles and the inherent advantages of pulse combustion.

Heat release rates. Although the combustor was designed for a maximum heat release rate of one million Btu/hour ft³, it was not operated at its maximum capacity.

Figure 17. Measured combustion efficiencies of the three test coals.

Figure 18 shows the variation of heat release rate with equivalence ratio. Generally, it was found that the heat release rate increased as the coal feed rate was increased, the relationship being quite linear. It is to be observed that the equivalence ratio was varied by keeping the air mass flow constant and by varying the coal feed rate, i.e., increasing equivalence ratio increases the coal feed.

Emissions. Gaseous emissions comprised of CO, CO₂, O₂, SO₂, and NO_x were measured during the tests. The CO, CO₂, and O₂ levels measured reflected the trends in temperature profiles and combustion efficiency as described above. Carbon dioxide levels ranged from 7 to 14% depending on coal composition, equivalence ratio, and combustor frequency. Carbon monoxide concentrations varied from 0.5 to greater than 3%.

The combined influence of coal nitrogen content, combustion temperatures, and residence times on NO_x formation in pulse combustors can be seen from the data of Figure 19. The nitrogen content of Mine 1, Mine 3, and Mine 4 coals are 1.8, 1.48, and 1.87, respectively. The influence of combustor frequency can be clearly seen by comparing the NO_x emission values for Mine 1 and Mine 4 coals (nitrogen content roughly the same). The higher frequency produced by the shorter 7.16 ft resonance tube length has a marked influence on the NO_x emissions, yielding lower NO_x values. Fuel nitrogen content by itself cannot be correlated directly with NO_x emissions. But fuel nitrogen content, combustor temperatures, devolatilization histories, and residence times all synergistically influence NO_x formation.

In the case of pulse combustors, the peculiar characteristics of the pulsating flow have a strong controlling influence on NO_x formation. Comparing Mine 1 and Mine 4 coal (same fuel N₂ content) at an equivalence ratio of 1.0, there is a substantial reduction from about 650 ppm to about 375 ppm in NO_x emissions. This is partly the result of lower peak combustion temperatures as depicted earlier in Figure 15. Another reason for these lower NO_x emissions is the enhanced mixing at

Figure 18. Variation of heat release rate with equivalence ratio.

Figure 19. NO_x emission levels from test coals.

higher frequencies of operation for Mine 1 coal which disperses hot combustion products from around the coal particle into the surrounding air. This reduces the residence time of the combustion products in the high temperature reaction zone surrounding the coal particle. It is felt that this combined influence of product dispersion and lower average temperatures promoted by the pulsating flow is largely responsible for the lower NO_x emissions at higher frequencies from Mine 1 coal. For the same frequency, Mine 3 coal which has a lower nitrogen content yields lower NO_x values than Mine 4 coal as seen in Figure 19.

Sulfur dioxide values for these three test coals ranged from about 800 ppm for Mine 1 and Mine 4 coals to about 2200 ppm for Mine 3 coal at an equivalence ratio of 1.0. This is in keeping with the higher fuel sulfur content of Mine 3 coal (2.69%) compared to that of Mine 1 and Mine 4 coals (0.69%). The increase in SO_2 emissions is almost threefold, in proportion to the fuel sulfur content. Frequency variations do not appear to influence the SO_2 emissions as much as they do the NO_x emissions. Equivalence ratio has a stronger effect, SO_2 emissions increasing with an increase in equivalence ratio, the increase being almost linear in the equivalence ratio range of 0.7 to 1.1.

Pulse Combustor Modeling

Combustion of coal in the pulsating mode is a very complex process, with the efficiency and emissions of NO_x and SO_2 being controlled by a number of variables. Tests conducted and reported above in our research program show that pulse combustor frequency, temperature, air-fuel ratio, mixing characteristics, and coal composition are all relevant factors. The fundamental mechanisms by which these factors control the combustion and emissions characteristics of coal in a pulse combustor are not clearly understood at the present time. To investigate these important basis mechanisms, a theoretical modeling study of the combustion of coal

In the pulsating mode was initiated and the following is a summary of the results with respect to the various modeling studies available in the literature.

The early studies by Reynst (Thring 1961) indicate the complexities of combust-
ing a solid fuel like coal in a pulse combustor. However, the analysis presented does
not deal with the parameters influencing the formation of NO_x and SO_2 . Gill and
Bhaduri (1978) have conducted an evaluation of the pressure and frequency in a
Helmholtz pulse combustor. They employ the spring-mass vibration analogy to the
phenomena occurring in the pulse combustor with combustion heat release.
Assumptions are made such as uniformity of distribution and combustion intensity
within the chamber and constant combustion rate. Temperature variations in the
gas column, heat transfer and other physical effects are excluded. The resulting non-
linear differential equation has been solved numerically to predict the combustion
pressure rise and frequency. This study follows an earlier effort by Bhaduri et al.
(1968) which involved integrating the unsteady heat transfer equation with the
assumption that the pulsating combustion could be modeled as a simple one-
dimensional wave propagating in the combustion chamber. This analysis enabled
the prediction of the temperature profile within the combustion chamber. These
studies again do not consider the basic mechanisms such as fuel-air mixing,
fluctuating reaction rates, non-steady heat transfer, etc., on the combustion process,
and are insensitive to fuel composition, and may be considered as global models.

In a more recent study, Dwyer et al. (1985) employ the one-dimensional, non-
steady mass, momentum and energy conservation equations to the pulse combustor.
The influences of wall friction, heat transfer, heat release and inlet valve dynamics
are included in the model. Integration of the governing equations employing finite
difference techniques has enabled the prediction of the combustion chamber
pressure and temperature and exhaust gas velocity. The heat release characteristics

of a solid fuel like coal are not included. The paper also contains additional references on zero and one-dimensional models of a pulse combustor.

The literature shows that efforts to understand the basic mechanisms of pulse combustion, especially with regard to a solid fuel like coal, have been very few and have not concentrated on the parameters relevant to NO_x and SO_2 formation. With this latter goal in mind, a model capable of describing the fundamental processes especially with respect to NO_x and SO_2 formation from the pulse combustion of coal was sought to be developed.

A stirred reactor type of approach has been taken in the model computations. The stirred reactor models permit the kinetics of the NO_x formation reactions to be easily incorporated into the model. However, two important features that need to be added in order that the model may represent pulsating combustion processes are (a) the unsteady nature of the combustion process and (b) the non-uniform, non-steady mixing processes in the pulse combustor. In the case of coal combustion, the heterogeneous nature of the combustion process must also be included.

In the present model, the unsteady nature of the combustion process is modeled, as a first trial, in the form of sinusoidal inflow of reactants and outflow of products. The mixing problem is somewhat more complicated, and two approaches were studied. These are the residence time description approach followed by Levenspiel (1962) and a multi-cell stirred reactor approach with backmixing, shown in Figure 20, as employed by Zeinalov et al. (1972).

Implementation of the programming has shown that the problem is quite complex. Including the various features of pulsating combustion processes, in order to evaluate their significance, made the model more complicated than initially conceived and produced some unstable computations initially. This was sought to be overcome by reformulating the problem. Test calculations were then performed using a global kinetics model for propane combustion to compare with our

T = temperature
C = concentration

v_f = forward flow
 v_b = backmixed flow

Q = heat loss

Figure 20. Schematic of Multi-Cell, backmixed stirred reactor model.

experimental propane results. Typical results of the computations are presented in Figure 20, which compares the experimentally measured temperature profile within the pulse combustor with the calculated profile. The fuel used in the test is propane.

The results indicate that there is good agreement between the calculated and measured temperature profiles at some distance downstream of the main burner section. However, the temperature predictions are lower in the rapid combustion section of the burner. This could be due to a variety of reasons, one of which may be the influence of computational cell size.

Since this primary combustion zone influences the prediction of combustion efficiency to a large extent, the calculation in this region was refined by incorp-

orating a smaller cell size into the computer program. The original mesh size was 0.5 in./cell. This was refined to 0.25 in./cell. Figure 21 shows the influence of the smaller cell size on the temperature profile within the combustor. The agreement within the high temperature zone of the combustor has improved.

At this time, the modeling work is being continued. The problem is a complex one because of the pulsating nature of the flow and the fundamental processes of mixing, chemical reaction, non-steady heat transfer, etc. On refining the computer program and incorporating the necessary kinetic models for NO_x formation, etc., it is planned to submit the results for publication.

CONCLUSIONS

Results of the experimental investigations show that the pulse combustor is a very efficient device in burning micronized coal. Combustion efficiencies as high as 95% were measured when burning beneficiated micronized coal with ash content less than 1%. Oxides of nitrogen emissions from coal-fired pulse combustors are essentially the result of fuel nitrogen conversion. Pulse combustor frequency is found to influence both the temperature profiles and NO_x formation. The influence of pulsation frequency on NO_x production is stronger than the influence of equivalence ratio. NO_x formation is found to be controlled by a combination of factors comprised of fuel nitrogen content, combustor frequency, residence time, equivalence ratio and combustor temperature profiles. For the three coals tested, SO_2 emissions were found to be generally in proportion to the fuel sulfur content.

We sought to formulate a numerical model to try to explain some of these pulse combustor features. Because of the complexity of the problem, model results are still considered preliminary. Comparison of measured and calculated temperature profiles appear to be in reasonable agreement for gaseous fuel. Further work on model development is continuing.

Figure 21. Comparison of measured and calculated combustor temperature profiles with propane fuel.

REFERENCES

- Adams, C. W. 1982. Performance results of Lennox pulse combustion furnace field trials. In Proceedings, Symposium on Pulse Combustion Applications, vol. 1. GRI Report 82/0009.2. Gas Research Institute.
- Alhaddad, A., and G. A. Coulman. 1982. Experimental and theoretical study of heat transfer in pulse-combustion heaters. In Proceedings, Symposium on Pulse Combustion Applications, vol. 1. GRI Report 82/0009.2. Gas Research Institute.
- Altenkirch, R. A., R. E. Peck, and S. L. Chen. 1978. Fluidized bed feeding of pulverized coal. Powder Technology 20:189-196.
- Babkin, Y. L. 1965. Pulsating-combustion chambers as furnaces for steam boilers. Thermal Engineering 12(9):23-27.
- Belles, F. E., P. S. Vishwanath, and J. E. Ives. 1982. Sound characteristics of a family of pulse burners at various heat release rates. In Proceedings, Symposium on Pulse Combustion Applications, vol. 1. GRI Report 82/0009.2. Gas Research Institute.
- Belter, J. W., L. Dockter, and R. C. Ellman. 1969. Operating experience with lignite-fueled pulse-jet engines. ASME Paper 69-WA/FU-4.
- Bhaduri, D., C. B. Baxi, B. S. Gill, and N. Mukhopadhyay. 1968. A theoretical study of pulsating combustion. Indian Journal of Technology 6(August).
- Blomquist, C.A., and J. M. Clinch. 1982. Operational and heat-transfer results from an experimental pulse-combustion burner. In Proceedings, Symposium on Pulse Combustion Applications, vol. 1. GRI Report 82/0009.2. Gas Research Institute.
- Dwyer, H. A., T. T. Bramlette, J. O. Keller, and B. R. Sanders. 1985. A numerical model of a pulse combustor. Paper WSS/CI 5-2B. Western States Section, The Combustion Institute.
- Gas Research Institute. 1982. Proceedings, Symposium on Pulse Combustion Applications. Report GRI-82/0009.2.
- Gill, B. S., and Bhaduri. 1978. Prediction of pressure and frequency in Helmholtz pulsating combustion systems. Indian Journal of Technology 16.
- Hamor, R. J., and I. W. Smith. 1979. Fluidizing feeders for providing fine particles at low stable flames. Fuel 394-403.
- Hanby, V. I. and D. J. Brown. 1968. A 50 lb./h pulsating combustor for pulverized coal. Journal of the Institute of Fuel (Nov.):423-425.
- Kentfield, J. A. C. 1982. Valveless pulse combustors with multiple inlets. In Proceedings, Symposium on Pulse Combustion Applications, vol. 1. GRI Report 82/0009.2. Gas Research Institute.
- Levenspiel, O. 1962. Chemical reaction engineering. New York: Wiley.

- Milne, T. A., and J. E. Beachey. 1977. The microstructure of polarized coal air flames, part 1. Stabilization of small burners and direct sampling techniques. Combustion Science and Technology 16:123-128.
- Okazaki, K., H. Shishido, T. Nishikawa, and K. Ohtake. 1984. Separation of the basic factors affecting NO formation in pulverized coal combustion. In Twentieth International Symposium on Combustion, pp. 1381-1389. The Combustion Institute.
- Putnam, A. A, F. E. Belles, and J. A. C. Kentfield. 1986. Pulse Combustion. Progress in Energy and Combustion Science 12:43-79.
- Raghavan, J. K.. 1986. Pulverized coal combustion in a one-dimensional burner. Masters thesis, Southern Illinois University at Carbondale.
- Rajan, S., and A. Gupta. 1988. Stoichiometric effects on the pulsating combustion of gaseous fuel. Paper presented at the Central States Spring Technical Meeting, The Combustion Institute, May, Indiana.
- Rayleigh, Lord. 1945. The theory of sound, vol. II. New York: Dover Publications.
- Severyanin, V. S. 1982. Application of pulsating combustion in industrial installations. In Proceedings, Symposium on Pulse Combustion Applications, vol. 1. GRI Report 82/0009.2. Gas Research Institute.
- Severyanin, V. S. 1965. The combustion of solid fuel in a pulsating flow. Thermal Engineering 16(1)6-8.
- Thring, N. W., ed. 1961. Pulsating combustion-The collected works of F.H. Reynst. New York: Pergamon Press.
- Tyndall, J. 1987. Sound. New York: D. Appleton & Company.
- U.S. Department of Energy. 1979. Proceedings, 1st Symposium on Pulse Combustion Technology for Heating Applications. DOE Report ANL/EES-TM-87.
- West, F. B., and A. T. Taylor. 1952. The effect of pulsations on heat transfer, turbulent flow of water inside tubes. Chemical Eng. Progress 48(1):39-43.
- Whitlock, D. J, S. F. Favors, and G. Reuter. Field tests on pulse combustion boilers for residential space heating and water heating. In Proceedings, Symposium on Pulse Combustion Applications, vol. 1. GRI Report 82/0009.2. Gas Research Institute.
- Zeinalov, M. A. O., M. Kuwata, and R. H. Essenhigh. 1972. In Proceedings, 14th International Symposium on Combustion, pp. 575-583. The Combustion Institute.
- Zinn, B. T., N. Miller, J. A. Carvalho, Jr., and B. R. Daniel. 1982. Pulsating combustion of coal in a Rijke type coal combustor. In Proceedings, 19th International Symposium on Combustion, pp. 1197-1202.
- Zinn, B. T. 1985. Pulsating combustion. Mechanical Engineering (Aug.):36-41.

- END -

DATE FILMED

02 / 15 / 91