**DEVELOPMENT OF A COAL FIRED PULSE
COMBUSTOR FOR RESIDENTIAL SPACE HEATING**

**TECHNICAL PROGRESS REPORT
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SECTION 1.0

INTRODUCTION

The United States has vast reserves of coal. If the coal can be used in place of oil and gas, foreign dependence on oil can be reduced and domestic oil reserved for transportation and strategic military uses. However, using coal instead of oil and gas requires that a number of combustion technologies, cost and environmental control technology-related issues be addressed.

Recent technological developments in coal beneficiation and coal liquid mixtures project that premium, coal-based fuels, such as coal-liquid mixtures, or dry ultrafine coal, when combined with advanced fuel handling, storage, and combustion technologies, can approach standards of performance expected by users of natural gas and petroleum in small-to-moderate-scale applications including residential applications.

Many of the coal beneficiation processes that are being developed require fine grinding of coal, and sometimes, slurring in a liquid medium. After fine-grinding of the coal, transportation and storage of the dry coal will be possible but more difficult than transportation of coal-liquid slurries. On the other hand, burning the coal-water mixture in a conventional burner is more difficult than burning fine-ground coal and is also moderately less efficient.

Another problem encountered in promoting coal as a residential fuel is related to the lack of compact, efficient, inexpensive burners for this application. These combustors will also be required to compete with oil and gas fuels in controlling pollution to meet the existing, regional environmental standards. While local standards do indeed vary in the United States, it is expected that contributions made by the fuel-bound nitrogen to NO_x formation will be a problem that must be addressed by the proposed combustor technology development program. In addition, particulate matter rejection and opportunities for sulfur capture from the combustion system effluent and gases must also be included as part of the development program to the degree that their reduction is integrally related to combustor operation.

In summary, the primary combustion technology-related issues which must be addressed before premium coal based fuels can be employed in residential heating applications are as follows:

- o Availability of combustor technology in the size range required that are reliable, safe, and that can be integrated in a cost effective system.
- o Control of NO_x , SO_x (where applicable), and particulate emissions to meet present and projected standards. (Note: Sulfur dioxide emissions control would not be required for premium coal grades defined for the proposed study).

- o Operational considerations regarding maintenance, responsiveness to demand, and methods of ash, and, where necessary, spent sulfur sorbent disposal.

For non-retrofit applications, such as those described in this program, DOE, has defined a combustion system to include the combustion unit, integrally coupled to a suitable heat exchange unit.

However, the complete system, including component and other supporting systems, will be considered to the extent that such considerations will impact the integration of the combustion technology into the overall system.

Coal-liquid slurries will be employed by MTCI as the fuel choice for the initial system development and commercial introduction into the residential end-use application. This selection was made for several reasons. First, MTCI anticipated that coal-liquid slurries will encounter less institutional resistance from both the end-user and the local authorities enforcing safety and health regulatory requirements. This is primarily because coal-water slurry fuels are perceived to be like oil fuels, if not even safer. Secondly, developing the technology for burning coal-liquid slurries for initial market introduction implies that the technology could employ ultrafine, dry coal fuels should the fuel handling and storage be developed and should it gain acceptance for residential use. This is particularly true for pulse combustors. Third, MTCI anticipates that the cost of coal slurries will ultimately be lower in comparison with other fuels as delivered to the combustor. This includes all cost contributions incurred at the mine, fuel formation, transportation to site, and the cost of on-site hardware to store and handle the fuel.

1.1 APPLICATION REQUIREMENTS

Use of coal in large utility boilers and medium sized industrial boilers is practiced widely; however, in small scale applications, such as for residential space heating, the use of coal has almost vanished since the use of natural gas, fuel oil, and direct electric space heating have become popular. In the past, coal-fired heating furnaces produced a lot of black smoke and ash, and they required daily attention for feeding of the coal and removal of the ash.

With the advent of coal slurry fuels (particularly highly loaded and beneficiated coal slurries) that are easily stored, pumped and used as fuel oils, the use of coal for space and water heating becomes a promising market.

The application assumes the availability of both a premium coal-based fuel and the equipment required to handle the fuel, including the equipment for delivery to the user-building, storage within the building (or in a tank adjacent to the building), and transportation from storage to the burner. The design fuel is to have a mineral matter content less than 0.8 lb/million Btu and a sulfur content less than 0.5 lb/million Btu. Although MTCI's pulse combustor-fired furnace can burn dry powder coal, MTCI feels that for safety reasons, coal water slurry is the fuel of choice for use in residential space

heating. MTCI's pulse combustor technology will also be developed to accommodate firing of a secondary fuel of both natural gas and No. 2 fuel oil. This is believed both achievable and beneficial since some parts of the nation may require natural gas as the secondary fuel and other parts of the nation may require fuel oil.

In addition, the space heater replacement market requires consideration of both forced air draft and hot water heating systems. The latter can be found in the eastern parts of the U.S. and many parts of Europe. Therefore, the combustor technology performance requirements must be mindful of such considerations. In particular, pulse combustor-fired systems which include condensate heat recovery must consider the level of pressure boost to be supplied by the combustor and the methods for ash rejection, particulate removal, and sulfur capture, if any, which can be employed with such systems.

The design target for the performance of these systems is as follows:

Primary Fuel:	Coal-liquid mixture or dry, ultrafine coal
Secondary fuel:	Natural gas or fuel oils
Fuel Characterization	
- Mineral matter:	Less than 0.8 lb/MMBtu
- Sulfur:	Less than 0.5 lb/MMBtu
Ignition:	Automatic
Response Time:	Up to 5 minutes to full load
Reliability and Safety:	Comparable to oil-fired residential heaters
Steady State Efficiency:	Greater than 80%
Combustion Efficiency:	Greater than 90%
Daily Maintenance:	None
Scheduled Maintenance:	Less than or equal to twice a year
Size:	Height less than or equal to 6 feet Floor space less than or equal to 15Ft ²
Service Life:	Greater than or equal to 20 years
Fluid Reheat Temp:	200°F for Water (From 180° - 200°F) 700°F for Air at 800 ft ³ /min

Integrated Accumulation
and Collection:

Desirable

Flue Gas Particulate
Removal:

Acceptable

The pulse combustor technology which has been developed at the laboratory scale by MTCI offers the potential of meeting these requirements for residential space heating applications in the size range of interest, 100,00 Btu/hr. The attributes of this technology include a compact combustor unit which aspirates its own air, has high heat transfer rates for indirectly heated units, and is self-adjusting with respect to combustion stoichiometry and turn down rates.

SECTION 2.0

TECHNICAL BACKGROUND

Pulse combustion can achieve high combustion efficiency, deep staging, high turndown and high sulfur capture efficiency. Sulfur removal is achieved by injecting dry sorbent. In addition, these combustors can aspirate their own combustion air and provide pressure boost and the associated exit gas kinetic energy. This energy can be employed to reject particulate matter from the combustor effluent with little to no system pressure drop penalty. The combustion-induced oscillations and aerodynamic design of a properly designed pulse unit can achieve these benefits in a simple, compact and self-regulating combustor without the need for a complex collection of subsystems.

2.1 PULSE COMBUSTOR DISCUSSION

The intrinsic stoichiometry of a pulse combustor can be fixed by design and can be maintained constant through a wide range of firing rates. At the lower end of this firing rate range, the combustion-induced pressure fluctuation in the chamber is lower. Therefore, the amount of air intake induced by the fluidic diode (the aerodynamic valve), coupled with the inertial effects of the hot gas column in the resonance tube, is lower. When the fuel feed rate is increased, the amplitude of the pressure fluctuations is increased due to the increase in the heat release responsible for excitation of the combustion-induced dynamic pressure. This, in turn, induces more air intake. The combustor operating stoichiometry is, therefore, automatically maintained over a range of firing without the need to actively control and coordinate the combustion air and fuel mass flow rates. The range of firing for which the operating stoichiometry can be maintained is a function of the combustor design, particularly the aerodynamic valve and the geometry of the chamber transition region from the chamber inlet to its maximum diameter. With deep staging (substoichiometric air), the firing range is reduced somewhat. With air stoichiometry between 0.7 and 2.0, the fuel firing rate range covers approximately a 4:1 turndown ratio.

At the low firing, the heat release rate in $\text{Btu/ft}^3\text{-hr}$ is lower due to the lower oscillating flow field on the combustor. As the fuel feed rate is increased, the increase in the pressure fluctuation causes a corresponding increase in the fuel burn rate and a correspondingly higher heat release rate. Therefore, the reduction in residence time due to the overall increase in the burn rate of the coal particles. This tends to maintain the combustion efficiency of a properly designed combustor substantially constant over the design operating range.

The primary function of the aerodynamic valve is to act as a fluidic diode which employs pressure fluctuations in the combustion chamber for inducing the intake of the combustion air. There are two engineering design parameters that dominate the design of an aerodynamic valve size; the minimum resistance to air intake and the fluidic diodicity of the valve. The latter

is a non-dimensional ratio between the resistance to flow out of the chamber to the resistance to flow into the chamber (intake). In general, the higher the fluid diodicity of the aerodynamic valve, the more air per Btu/hr of fuel firing is induced by the intake. We should note that a combustor which would normally operate with high excess air, by virtue of employing a valve with high fluidic diodicity, can be operated at lower air stoichiometry by throttling the air intake at a plenum inlet. With a fixed damper setting at the inlet into the plenum, the combustor firing rate can be varied with the induced stoichiometry remaining essentially constant for a range of firing.

It is also possible to reduce the lowest firing rate of a combustor by reduction of both the aerodynamic valve and the resonance tube minimum diameter. This also enhances the startup characteristics of the combustor. With this design option, the turndown ratio could be greater than 8.1. This, however, may require an inlet air fan if the pressure drop for ash removal and flow of the gases through the equipment being retrofitted (boiler, etc.) requires it. Nevertheless, the air intake (mass flow rate) remains dependent on the firing rate since the self aspiration and boost pressure contribution of the pulse combustor unit remains in effect. This system configuration tends to increase the maximum combustion intensity achievable for two reasons. First, with the higher flow resistance at both ends of the chamber, more dynamic pressure amplitude obtained. Secondly, on air intake, the presence of an air fan tends to allow "supercharging of the combustor to higher firing rates than are attainable under atmospheric aspirating conditions.

The geometry of the combustion chamber can be selected to affect the fraction of the fuel burn which contributes to inducing the pressure oscillations and the fraction which is burned downstream from the dynamic pressure peak region under the influence of the induced oscillatory flow conditions. The chamber geometry also affects the sensitivity to method and location of fuel injection as well as to changes in fuel characteristics. The burn rate in the combustion chamber is dominated by vortices which are shed from the transitions in the cross-sectional area of the chamber. In the resonance tube, however, the burn rate is dominated by the axial, oscillating, flow velocity component which tends to increase monotonically from the resonance tube inlet to the exit. The combustion process in the resonance tube is mostly responsible for completing the burn of char produced from the larger particles which are volatilized and partially burned upstream in the chamber. The increase in the oscillating velocity along the resonance tube maintains a high rate of char burn as the char particles become more prone to entrainment and as the O_2 partial pressure decreases. In all other combustion systems, the relative motion between the gases and the solids is dependent on swirl, turbulence, etc. These flow fields tend to get damped downstream in the flame where they are needed most, i.e., as the other particles become smaller, ash laden and entrainment-prone, and as the partial pressure of oxygen decreases.

The design of the resonance tube could involve more complex generators in principle, but in MTCI's experience, this is not necessary; i.e., a straight line generator forming a conical section is quite practical. This degree of freedom allows control of the gas exit velocity and the overall volume of the resonance tube for a given length. The gas exit velocity is selected such

that it is sufficiently high to induce second stage air or disentrain solids, and simultaneously, sufficiently low to allow good mixing for the second stage burn in staged combustion. The volume of the resonance tube affects both the resistance time available for completing the burn of char produced from larger particles as well as the resonant frequency of the unit.

The pulse combustor design flexibility is uniquely suited for small commercial applications such as residential heating. The degrees of freedom available permit the design and mass production of these simple-to-manufacture, compact burners for a wide spectrum of small end-use applications. The feasibility of the proposed technology has already been demonstrated in prior MTCI combustor development programs.

2.2 RESIDENTIAL SPACE HEATING SYSTEM DESCRIPTION

Regardless of the type of equipment to which it is applied, a CWM-fired pulse combustor must first of all meet the basic thermal requirements of the furnace and convective sections. The furnace will generally be designed for a specified heat flux profile, normally characterized by a maximum heat flux in the flame zone, with the flux relaxing to a lower value at the furnace outlet. Producing an acceptable heat flux profile in the furnace necessitates achieving a proper combination of initial gas temperature, radiant emissivity of the gas, and gas residence time in the furnace. Table 1 compares calculated values of combustion gas quantities and compositions for natural gas and CWM fuels. Typical analyses of the two fuels were assumed in the calculations. Combustion of both fuels was arbitrarily assumed to occur at 15 percent excess air. As shown in the table, combustion gas from the selected CWM is not substantially different from that produced by gas firing in terms of volume or composition and is also similar to residential fuel oil.

The initial temperatures of combustion gases from oil and gas firing are comparable; i.e., essentially the adiabatic flame temperatures. Oil flames are, however, much more luminous (higher emissivity) than gas flames, which results in their radiating a larger portion of the heat released to the furnace walls and producing a lower furnace-exit gas temperature than gas flames. CWM firing in a pulse combustor can be expected to produce a lower initial temperature at the furnace entrance than oil or gas firing since some of the heat is removed by water evaporation and heat loss (if any) from the combustor before the combustion gas reaches the furnace. Also, the CWM furnace-inlet gas may be essentially non-luminous if the combustion has been completed in the combustor (no combustion staging).

The thermal requirements of the convective section of the boiler or process heater are generally met if the combustion gas from the furnace enters at or near the design conditions of temperature, flow rate, and composition. The result of this will be that the correct amount of overall convective heat adsorption will occur and will permit the required final temperature of the working fluid to be met. It is particularly important that the gas temperature be near its design value in the vicinity of the finishing section of the heat transfer surface, where the final fluid temperature is determined. Too low a gas temperature at this location will result in the specified fluid

TABLE 1
COMPARISON OF COMBUSTION GASES PRODUCED BY
GAS AND CWM FUELS

	Natural Gas	CWM Fuel (70% Solids)
Typical Analysis (wt.%):		
Sulfur	~0	2.2
Hydrogen	22.68	4.8
Carbon	69.26	72.8
Nitrogen	8.06	1.5
Oxygen	~0	6.2
Ash, wt. %	~0	9.0
Moisture, wt. %	~0	3.5
	100.00	100.00
Higher Heating Value, Btu/lb	21,800	13,080
Slurry water, lb/100 lb fuel	0	42.85
Combustion Gas Composition* (Vol.%):		
SO ₂	~0	0.158
CO ₂	8.33	13.90
N ₂	71.58	70.78
O ₂	2.47	2.45
H ₂ O	17.60	12.71
Gas Volume, lb-moles/10 ⁶ Btu	31.74	33.33
Gas Weight, lb/10 ⁶ Btu	878	970
Ash, lb/10 ⁶ Btu	~0	6.88
SO ₂ , lb/10 ⁶ Btu	~0	3.38

*Combustion at 15 percent excess air. Air humidity 1.5 moles per mole dry air.

temperature not being met, and too high a gas temperature at this location can result in overheating and damaging the tube metal. These considerations are particularly important for residential design where either immersed or surrounding heat exchangers are employed to remove heat directly from the combustor.

As indicated in Table 1, differences in gas weight, volume and composition produced by CWM firing versus oil/gas firing are not major. Therefore, the thermal requirements of the convective section should be met if the furnace-exit gas temperature is sufficiently near the design point. Generally, achieving the design heat absorption to the furnace walls will produce a furnace exit gas temperature near design. However, some consideration must be given the heat used to evaporate water from CWM fuel, combustor heat losses or cooling requirements, if any, as well as the slight differences in gas weight and composition.

FIGURE 1: PROGRAM LOGIC FOR THE DEVELOPMENT OF AN ADVANCED PULSED-COAL COMBUSTOR FOR RESIDENTIAL SPACE HEATING

Task 1.2 Fabrication of a Pulse Coal Combustor

Pulse combustor components specified in the design in Task 1 will be fabricated and/or procured. Efforts will be made to utilize off-the-shelf components to the greatest extent possible. Kistler pressure transducer instrumentation will be used and a coal slurry pump and baghouse will be procured. During this stage of the program, arrangements will be made to obtain different types of coal slurries. Initial test configuration will be set-up after completing the procurement. The combustor hardware will be assembled and unit operability determined.

Task 1.3 Testing the Pulse Coal Combustor

This is the major and most important task of this program. After completion of the set-up of the test unit, shake-down testing will be done to ensure safety during the regular test. The test plan developed under Task 1.1 shall consist of three major elements: 1) combustor geometry (including the tail pipe and aerodynamic valve) variations, 2) fuel types, and 3) controls evaluation. For the combustor geometry variations, two aerodynamic valves, two chambers and two tail pipes will be evaluated. In the case of the fuel types, first the combustor will be tested with natural gas and then with No. 2 fuel oil. Combustion air/fuel ratio, pressure boost, frequency, peak pressure variations, noise levels, temperatures and analysis of exhaust gases for O₂, N₂, CO, CO₂, and NO₂ will be performed. Combustion efficiency will be calculated. Then the unit will be tested on three coal slurries, chosen in consultation with DOE. Table 2 provides the fuels test matrix. Particulate loading of the exhaust gases will be determined by the EPA method 5. Operational noise levels will be measured at various points in the laboratory.

In each test series, there will be a number of tests dictated by the variations in configurations to be tested. The above test matrix does not give the details of the individual tests. In early testing series, some of the configurational variation may be found useless to pursue and such combinations will be eliminated in the later series. When feeding coal-based fuel to the pulse combustor, the particulates collected and flue gases will be analyzed to determine unburned carbon and combustion efficiency. MTCI will use gas chromatograph and instrumentation equipment already on-hand for gas analysis.

TABLE 2: Test Matrix - Phase I

<u>TEST SERIES NO.</u>	<u>FEED</u>	<u>CONFIGURATION EVALUATION</u>	<u>CONTROL EVALUATION</u>	<u>EMISSIONS MONITORING</u>
1.	Natural Gas	Yes	No	Yes
2.	#2 Fuel Oil	Yes	No	Yes
3.	Slurry Fuel #1	Yes	Yes	Yes
4.	Slurry Fuel #2	No	Yes	Yes
5.	Slurry Fuel #3	No	Yes	Yes

Task 1.4 Evaluation and Selection of Optimum Configuration for Integrated Testing

The test data collected in Task 1.3 will be analyzed carefully. Based on the test results, an optimum configuration will be selected for further study as an integrated system in Phase II. This configuration will have the potential to meet or exceed the performance criteria and, in addition, will have the capability for fuel flexibility.

Whereas the emphasis in Phase I will be on the pulse combustor development, the emphasis in Phase II will be on testing and evaluating heat transfer/heat exchange equipment as integrated with the pulse combustor. Based on the level of success achieved during Phase I work, progressive development and testing of integrated combustor/heat exchanger subsystems will be performed. This second phase will be carried out in five technical tasks, each of which is described below.

TASK 2:

Task 2.1 Design of Integrated System

The optimum configuration of pulse coal combustor with proper aerodynamic valve, chamber, and tail pipe, as recommended by Phase I study, will be designed in detail to be applicable to the residential space heating market. The tail pipe will become part of the heat exchanger. Flexibility in the heat exchanger design will be provided to evaluate the following: a condensing mode and non-condensing mode, ash separation by cyclone before the condensing heat exchanger and ash separation by bag filter between the two heat exchangers.

Task 2.2 Fabrication of Integrated System

Making maximum use of the components from Phase I, a test unit for Phase II will be fabricated. The following equipment will be designed, fabricated and/or procured: New heat exchangers; mufflers, silencers, or decouplers at the air inlet exhaust; and an effective control system for fuel switching. This equipment will be added onto the best pulse combustor developed in Phase I. Instrumentation to monitor sound levels will also be incorporated.

Task 2.3 Testing the Integrated System

In this major task, the integrated combustor/heat exchanger subsystem will be tested in several modes so as to find the best possible mode, which meets the performance criteria at the least possible capital cost. Table 3 provides a test matrix that will be conducted for the condensing and non-condensing modes. Heat transfer rates and system thermal efficiencies will be calculated from the measurements of fuel firing rate, condensate and ash material balances, and temperatures of input/output streams. Performance of the ash collection systems (baghouse and cyclones) will be monitored.

TABLE 3: Test Matrix - Phase II

<u>TEST SERIES NO.</u>	<u>CONDENSING MODE</u>	<u>ASH REMOVAL</u>	<u>NO. OF STAGES OF OF HEAT EXCHANGE</u>
1.	Yes	Condenser	One
2.	No	Bag Filter	One
3.	Yes	Cyclone	One
4.	Yes	Bag Filter	Two
5.	No	Cyclone	One

Task 2.4 Data Analysis and Modeling

The data gathered in Task 2.3 will be analyzed and correlations developed to predict the performance of the integrated pulse combustor/heat exchanger under other conditions. This task will be carried out concurrently with Task 2.3 to maintain the proper experimental path.

Task 2.5 Technical and Economic Evaluations

Based on the test results (from Tasks 2.3 and 2.4), technical and economic evaluations will be made to determine the applicability of the pulse coal combustor for the residential space heating market. Environmental impact of this technology will also be evaluated for a few selected regions in the United States.

3.2 STATUS

During the testing last quarter with the 1.0 MM Btu/hr refractory lined combustor, it was noticed that slag was slowly flowing from the exit combustor and straight down the straight section of the tailpipe. It became apparent that slag would only form if combustion was fairly complete (low carbon in ash) and that the slag in the tailpipe and throat of the combustor affect of the combustor performance.

After a review and evaluation of the data collected from the prior tests, a system design for a tandem configuration (two parallel units) was chosen to improve fuel flow phasing. To date, tests were conducted with single pulse combustors, with and without phased coal injection. The phased coal injector was designed to stand-off the pulse combustor and to feed coal during the combustion portion of the cycle. It was determined that this method of feeding coal helps the combustion efficiencies although it was unclear whether the coal was being completely phased or just partially phased. A tandem combustor configuration should resolve this issue and provide a more effective phased feed cycle.

Two residential sized advanced chambers were fabricated with stainless steel 304 so that a tandem design system (two pulse units) can be assembled. The advanced chamber for industrial retrofit system (reported in an earlier monthly) had complex curvature surfaces, whereas the residential unit currently being investigated has a simple conical design. Another major difference between the two advanced chambers is that the inlet and exhaust pipes are attached tangentially for the residential unit where as these are attached axially for the industrial unit.

Tests were conducted with micronized Pittsburgh #8 coal in the new residential combustor. After some initial coal feeding problems, the combustor burned coal with no apparent problems. Due to leaking of the micronized coal out of the intake plenum, the test was stopped before a large enough ash sample was gathered to determine the extent of combustion. Coal was fed at a rate of four to seven pounds per hour with a support gas rate of 1 CFM. The boost pressure, peak to peak pressure, and frequencies were comparable to gas alone at this firing rate.

Some minor design changes were made to the tandem unit mostly in the aerodynamic valve. The aerodynamic valve length was changed to 12-1/2 inches to provide a tuning capability for inlet coupling. Also, a blower was attached to the intake plenum to eliminate unburned coal or gas from escaping.

The tandem pulse combustors were assembled and tested. At first, the units were tested on natural gas firing. The gas injector location was found to be critical. The injector location that gave good performance was at the narrow end of the combustor perpendicular to the aerovalve. This result is believed to be because of the high shear rates provided by orthogonal placement of fuel and air jets. The fuel supply system consisted of a single feed line connected to a tee situated close to the combustor injection points. The fuel tee acts as a dynamic coupling allowing automatic biasing of fuel between the two combustors. The coal injector was arranged in a similar manner and located somewhat distant from the gas injector. The scrubber components were acquired and installed in the facility and performed well during testing.

SECTION 4.0

TECHNICAL DISCUSSION OF WORK COMPLETED DURING THE REPORT PERIOD

The conical configuration, Figure 2, of the advanced residential chamber was selected for several reasons. Since the residential chamber has a high surface to volume ratio and a small characteristic dimension, the intensity and effective mixing length of vortices generated by the deceleration of influx gases during the suction and precompression cycle is expected to be diminished compared to the industrial unit. It is commonly held that the generation of intense vortices appropriately timed within the pulsation cycle is a prerequisite for robust oscillating combustion. Therefore, alternate means were necessary to improve the turbulent mixing of fuel, air, and activated reaction products during the positive pressure cycle in the small residential chamber.

One method for accomplishing this goal involves the application of a conical combustion chamber which connects to a tangential exhaust pipe located at the divergent side of the cone. During the suction phase of the pressure oscillation, exhaust gases in the tailpipe are accelerated in a cyclonic fashion toward the convergent side of the cone. At the convergent side of the cone, fuel and air are introduced into the chamber. The air and fuel parts are orthogonally positioned to enhance mixing and heatup of the fresh air and fuel mixture. The reaction products, which have been accelerated in a cyclonic fashion, maintain an active boundary layer envelope which surrounds the fresh air and fuel mixture. As the combustion chamber swings to the positive pressure phase, the chamber gases are rapidly decelerated resulting in a sudden and high degree of turbulence. The activated boundary layer breaks up into vortices which are carried into the fresh inflow thereby igniting the mixture during the desired phase of oscillation. Since the high kinetic energy of the reaction products, which are accelerated in a swirl of relatively large characteristic dimension, does not dissipate until the onset of turbulence, intense vortices are generated. The intense mixing offered by this arrangement is thought to enhance performance of the oscillating combustor with respect to volumetric heat release, peak pressure, and combustion efficiency.

The conical chamber also serves the secondary function of preferentially biasing combustion products flow in the direction of the tailpipe. In effect, the divergent section of the cone acts as a diffuser providing pressure recovery for effluent gases entering the tailpipe. Therefore, the chamber itself introduces a second order diodicity (the first order diodicity being the aerodynamic valve) which is thought to enhance pressure boost within the chamber.

The initial design of the tandem pulse combustor configuration combined many different ideas. The chamber shape is conical, designed after Hanby's pulse combustor. Hanby reported superior combustion with the conical shape as compared to a cylindrical shape. The aerodynamic valve and the tailpipe are

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FIGURE 2A: RESIDENTIAL ADVANCED CHAMBER

FIGURE 2B: INDUSTRIAL ADVANCED CHAMBER

FIGURE 2: ADVANCED CHAMBER DESIGNS

connected to the chambers to enhance the wave traveling through the chamber and tailpipe. A diagram can be seen in Figure 3. The rationale for utilizing the tandem configuration was based on the bases that a pulse combustor is designed to work in two basic cycles, inflow and outflow. During the inflow the fuel (being dry coal, CWM or natural gas) enters the chamber to a slight vacuum in the chamber along with the combustion air and return gases from the tailpipe. When the hot return gases hit the fuel mixed with the combustion air, the fuel explodes, creating a peak pressure. This peak pressure forces gas out the tailpipe and aerodynamic valve which creates a slight vacuum in the chamber.

When a slow burning fuel such as coal enters the chamber continuously, much of it will be entering as the combustor is exhausting and the coal will not burn but simply pass through the chamber. If any of this coal does burn, it will be adding heat and energy where none should be added, thereby reducing the pulsations.

Connecting two combustors through the coal feed line will help combustion in two ways. First, since the combustors will be running totally out of phase, the coal will only travel into the chamber with the lower pressure (which is desired). Also, as the coal enters the chamber, the hot exhaust gases from the higher pressure chamber will rush by the coal helping to devolatilize it as it enters the combustion chamber. Second, upon coupling the two chambers, the pressure waves will reinforce each other giving higher peak to peak pressures and better scrubbing of the particles. Another advantage of using two combustors in tandem is the noise reduction.

Efficient phasing of fuel in the coupling line is dependent on the ability to operate the tandem chambers 180° out of phase. Under these conditions, one chamber achieves a low pressure phase just as the other chamber simultaneously achieves a high pressure phase. Due to the pressure gradient existing in the fuel coupling line, combustion products are accelerated from the high pressure chamber to the low pressure chamber. The momentum of the accelerated gases biases a flow of fuel from the main fuel line into the tee and eventually into the low pressure combustion chamber. One half cycle later, a similar phenomena occurs in the opposing direction. by these means, fuel can be properly phased without the use of mechanical flapper valves or an independent phasing chamber.

As mentioned above, efficient fuel phasing through the use of a common fuel coupling line depends on operation of the tandem units with a 180° phase shift. It is possible that the natural instability of the tandem units employing a common fuel coupling line may be sufficient to automatically pull the two units 180° out of phase. This conclusion derives from the logical assumption that the units will inherently hunt for the most stable and robust operating mode. And of course, this mode is most probably that condition which results in efficient fuel phasing, i.e., the condition of 180° phase lag.

However, in order to accelerate the search for this state during start-up and to dampen excursions from this state during transients or natural system upsets, additional coupling means can be employed. Two such coupling means

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FIGURE 3: TANDEM PULSE COMBUSTOR

can be envisioned, aerodynamic valve coupling and tailpipe coupling. Since the displacement of gases is greatest at the tailpipe, tailpipe coupling is at present thought to be the most effective of these choices.

In tailpipe coupling, tailpipes from the two individual combustors are placed with suitably spaced opposing exits. The distance between the two tailpipe exits is critical since an excessively small gap interferes with the natural outflow of combustion products, while an excessively large gap diminishes the desired coupling effect. As one tailpipe expires high velocity exhaust gases, it supercharges the second unit by converting the momentum of these gases into pressure. One half of a cycle later, the reverse process occurs. Since the two pulse combustors will resist natural quarter wave oscillation behavior (i.e. fight each other) unless they are 180° out of phase, a significant driving force exists for achieving and stabilizing at this condition.

Aside from promoting out of phase operation and thus fuel phasing, the supercharging effect tends to augment the combustion induced oscillations thereby increasing peak-to-peak pressures achieved within tandem the combustion chambers. As an additional benefit, any unburned fuel particles remaining at the outlet of the tailpipe will most likely be entrained into the opposing tailpipe. Since these regions of the tailpipe have the greatest levels of velocity fluctuations, particles are afforded long residence times under conditions of effective particle scrubbing and thus accelerated combustion.

Finally, coupled operation of tandem units is thought to actively diminish the acoustic emissions through destructive acoustic interference.

In the earlier tests, the pulse combustor would burn all of the coal part of the time and somewhat less the rest of the time. Since these tests were run with a phased injector, it was concluded this was not totally effective and some other method of phasing the coal injection, such as a tandem configuration concept, should be employed.

Prior to assembling the tandem units, a residential advanced chamber was separately test fired. Testing of the residential advanced combustor design was conducted using both gas and coal with very encouraging results. During one of the initial tests, the combustor was fired on gas at a range from 90,000 Btu/hr to 210,000 Btu/hr while continuing self-aspirating air. In this configuration, the tailpipe was 144-inches in length at a constant 1.12 inch ID. The combustor quickly rose to 2066°F and maintained that approximate temperature throughout the test. The boost pressure in this unit while self-aspirating was low (two to four inches of water), increasing with the excitation air rate. The maximum boost pressure seen with the excitation air on high was eight inches of water. The lower boost pressure was also reported by Handy, whose design closely resembles MTCI's.

An important finding in this test was the range of frequency variations which occurred as a function of firing rate. The longer tailpipe was used to lower the frequency to 48 Hertz. It was found that under most conditions, the frequency was maintained about 111 to 124 Hertz; although when the gas was

turned up (over 200,000 Btu/hr), the frequency dropped to 48 Hertz. It is thought some higher order harmonic may be prevailing at lower firing rates.

In selecting a design operating frequency, calculations were performed using four tailpipes with three different sections. The first section is a 1.12-inch diameter tube section followed by a conical transition and finally a larger straight section at the end. From these, the Helmholtz and quarter wave frequencies were equated using the following equations:

- o Helmholtz frequency =
- o Quarter wave frequency = $C/[4(L_t + L_c)]$

where:

C = speed of sound = $49 \times T_{abs}$
 L_t = length of the tailpipe
 L_c = length of the chamber
 V_t = tailpipe volume
 V_c = chamber volume

The results are shown in Table 4. Case B reflects the design configuration using in the current set of tests.

TABLE 4: FREQUENCY CALCULATION

CASE	A	B	C	D
TAILPIPE STR LENGTH inches	55	144	55	30
TAILPIPE STR D	1.12	1.12	1.12	1.12
TAILPIPE CONE LENGTH	13.5	0	4	4
TAILPIPE CONE MIN D	1.12	1.12	1.12	1.12
TAILPIPE CONE MAX D	4	4	2	2
TAILPIPE LRG LENGTH	0	0	24	29
TAILPIPE LRG D	2	2	2	2
TAILPIPE VOL cubic inches	131	142	137	128
CHAMBER LENGTH inches	7.75	7.75	7.75	7.75
CHAMBER MIN D	1.2	1.2	1.2	1.2
CHAMBER MAX D	5	5	5	5
CHAMBER VOL cubic inches	66	66	66	66
TEMPERATURE °F	2460	2460	2460	2460
<u>Frequencies</u>				
QUARTER WAVE, HZ	96	48	80	103
HELMHOLTZ, HZ	96	47	81	103

The tandem unit tests implemented thus far are encouraging. The effect of increasing the tailpipe length so as to provide more heat transfer area for the eventual space or water heating application, as well as reducing the frequency of operation, will be investigated during the next period. The selection of fuel oil or natural gas as an alternate fuel to coal needs to be made after performing a few more experiments with both fuels. The choice of the form of coal fuel, whether to use the CWM slurry or the dry ultrafine (DUF) coal will also be made after obtaining more results with the different fuels. Among the different forms of coal fuel requested through the DOE fuel supply contractor, Energy International, the dry pulverized coal and the micronized coal water mixture have been received. The dry ultrafine (DUF) or micronized coal and the standard grind slurry have not been received. The fuel evaluation tests are also expected to begin during the next period.

The tandem advanced pulse combustors were coupled at the tailpipes with a four-inch gap between exits. The combustors were ignited using spark plugs located in each chamber. Initial testing of the coupled tandem units was performed using 68-inch tailpipes in contrast with the 144-inch tailpipes employed in the single advanced combustor previously tested. The decision to reduce the length of the tailpipe was made in order to investigate the relation between operating frequency and combustor performance, and to determine if more compact combustor dimensions are feasible. Tests will be conducted using longer tailpipes in the coupled tandem configuration at a later date.

The operating frequency of the 68-inch tailpipe tandem unit was 100 Hz. The unit was operated on coal with a small amount of support gas. The requirement for support gas is believed to be due in part to the heat losses which occur in the bare metal combustor. Operation of a refractory lined combustor is expected to significantly accelerate the rate of coal particle heat-up due to increased radiation from the hot refractory walls. This may obviate or diminish the need for support gas. It should also be noted that excitation air was required for stable operation. Proper tuning should diminish the need for this air.

The coal feed rate was varied during the course of the tests from five to 19 lbs/hr. At the higher feed rates, the support gas accounted for approximately 26 percent (87,000 Btu/hr) of the total heat input, while the coal feed accounted for a 74 percent fraction (243,000 Btu/hr). The tandem configuration ran smoothly and appeared to result in efficient combustion based on visual inspection of the coal ash. Measurement of the carbon in the collected ash resulted in a calculated combustion efficiency of approximately 92 percent. Table 5 summarizes the data collected from the tandem advanced combustors. Note that the frequency for the tandem units was recorded using a microphone pick-up. Therefore, for tandem units operating 180° out of phase, the recorded frequency is two times the true frequency for an individual combustor.

Slag and brown ash particles were found in the combustors and the flue gas plenum. The presence of slag in the bare metal unit suggests that the temperature at the center of the combustor may have significantly exceeded that near the chamber walls. The perpendicular inlet and tangential exhaust

TABLE 5: SUMMARY OF TEST RESULTS ON TANDEM
ADVANCED CONICAL COMBUSTORS

PULSE DATA				
DATE	06/08/87	06/08/87	06/08/87	06/08/87
TIME	12:18	14:28	17:35	18:38
NATURAL GAS DATA				
GAS PSI	1.18	2.50	2.50	1.00
GAS CFH	1.60	3.20	3.20	1.00
GAS SCFH	2.24	3.21	3.21	1.40
AIR DATA				
EXCITATION AIR PSI	70.00	90.00	90.00	95.00
EXCITATION AIR CFH	6.00	7.20	7.20	7.50
EXCITATION AIR SCFH	16.32	19.22	19.22	20.49
AUXILIARY AIR PSI	0.00	0.00	0.00	18.00
AUXILIARY AIR CFH	0.00	0.00	6.00	6.00
AUXILIARY AIR SCFH	0.00	0.00	7.62	8.93
COAL DATA				
COAL TYPE	NICHO	NICHO	NICHO	NICHO
DRY COAL FEED (LB/HR)	0.00	0.00	3.00	19.00
WET FEED (LB/HR)	0.00	0.00	0.00	0.00
Btu/LB	12000.00	12000.00	12000.00	12000.00
PERCENT WATER	1.62	1.62	1.62	1.62
PERCENT CARBON	70.00	70.00	70.00	70.00
PERCENT SULFUR	2.43	2.43	2.43	2.43
PERCENT OXYGEN	6.69	6.69	6.69	6.69
PERCENT NITROGEN	1.23	1.23	1.23	1.23
PERCENT HYDROGEN	4.90	4.90	4.90	4.90
PERCENT LIME/STONE	0.00	0.00	0.00	0.00
INCHES BOOST	9.00	10.00	12.00	0.00
OR	110.00	111.00	110.00	
FREQUENCY	100.00	200.00	200.00	200.00
PERCENT OXYGEN	3.39	4.16	3.62	11.30
PERCENT CO	0.07	1.71	0.56	
PPH H2S		-	-	
PPH SO2		-	-	
PERCENT CO2	9.76	5.35	12.41	
Btu GAS / HR	141079.39	202396.29	202396.29	87093.26
Btu COAL / HR	0.00	0.00	64000.00	243200.00
TOTAL Btu / HR	141079.39	202396.29	266396.29	331093.26
PERCENT COAL	0.00	0.00	24.02	73.43
TEMPERATURES				
CHAMBER	1937.00	2012.00	1976.00	1994.00
TAILPIPE				
CYCLONE				
CYCLONE EXIT				
TAILPIPE SKIN				
WALL TEMP				
COMBUSTION GASES				
FRACTION COMBUSTED	1.00	1.00	0.87	0.92
PERCENT CARBON IN ASH			49.00	36.00
CONFIGURATION	SINGLE	TANDEM	TANDEM	TANDEM
INLET DESCRIPTION				
INLET PROTRUSION (IN)	12.50	12.50	12.50	12.50
INSERT LENGTH (IN)				
INSERT MINIMUM ID (IN)				
INSERT MAXIMUM ID (IN)				
INSERT DEPTH OF STEP (IN)				
INJECTION				
FUEL TYPE	NATURAL GAS	NATURAL GAS	NATURAL GAS	NATURAL GAS
TUBE ID (IN)	0.31	0.31	0.31	0.31
INJECTION DISTANCE FROM T	DIRECT	DIRECT	DIRECT	DIRECT
INJECTION GUN TYPE	TEE	TEE	TEE	TEE
INJECTION ORIENTATION				
INJECTION DIRECTION	STRAIGHT	STRAIGHT	STRAIGHT	STRAIGHT
FUEL TYPE	COAL	COAL	COAL	COAL
TUBE ID (IN)	0.43	0.43	0.43	0.43
INJECTION DEPTH	SEMI-DIRECT	SEMI-DIRECT	SEMI-DIRECT	SEMI-DIRECT
INJECTION ORIENTATION				
INJECTION DIRECTION				
CHAMBER VOLUME (CU3)				
TAILPIPE VOLUME (CU3)	64.93	64.93	64.93	64.93
	80.40	80.40	80.40	80.40

stream configuration appears to promote excellent turbulent mixing in a manner which supports efficient combustion. This is evident since the mean combustion chamber residence time is only of the order of 10 milliseconds.

When inspecting the refractory lined tailpipe, it was noticed slag only built up on the straight section, and not the conical. It was decided to remove the straight section and test the pulse combustor. From this, three new pieces of information were found. First, slag did not build up on the sides of the conical tailpipe - it seemed to just fall through. Second, in eliminating the straight section of tailpipe, both the quarter wave and Helmholtz frequencies were changed and it was discovered the combustor followed the quarter wave frequency, not the Helmholtz. Third, although natural gas burns well at higher frequencies, coal seems to burn well at lower frequencies.

In selecting the optimum configuration, the total range and types of emissions must be addressed; 1) blow-back through aerodynamic valve, 2) submicron and micron-sized ash dust escaping from the baghouse, and 3) SO₂ in the flue gas. The first item can be avoided by adding a small blower to the air inlet plenum. The second and third items can be solved simultaneously using a caustic scrubber.

MTCI's tests are carried out indoors with proper exhaust fans. Nevertheless, the off-design testing optimization indicated that in addition to the fans, a scrubber was necessary to perform the test matrix safely. The use of the scrubber will eliminate the quench spray cooler and the baghouse. A schematic of the proposed scrubber is given in Figure 4.

The scrubber designed consists of two parts, lower packing and upper packing. The body of the scrubber is a six-foot high stainless steel tank with a two-foot diameter. The gases will enter the tank through a ten-inch port, two feet from the bottom, and then pass through a two-foot bed of stainless steel rasching rings with a spray of water and Na₂CO₃ solution on top. The bottom two feet is used to hold the recycle solution. Above the tank is an eight-foot length of ten-inch stainless steel pipe filled with rasching rings. An identical spray is set up above these rings too. The Na₂CO₃ solution will be sprayed above either of these two beds, or both, at a rate of nine gallons per minute. This will quench the gases and remove any SO₂ and ash. A level control will be installed on the holding section of the scrubber so that when the caustic solution gets low, a water spray will come on to maintain the liquid level.

The scrubber components have been acquired and installed. This unit will service both the industrial retrofit program (DE-AC22-87PC79654) and the residential program (DE-AC22-86PC90278).

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FIGURE 4: PULSE COMBUSTION TEST FACILITY WITH SO₂ SCRUBBER UNIT

SECTION 5.0

PLANNED ACTIVITY FOR THE NEXT PERIOD

Developmental testing of the tandem configuration will continue during the next period for optimum system configuration selection. Tests utilizing various fuels will also be completed, if they are received. The data will be evaluated and testing will be videotaped for presentations. An evaluation of the performance test data should be far enough along to permit the selection of a configuration for the Phase II effort on the design, fabrication, and testing of an integrated system.