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

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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, slurrying 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 in place of 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 include 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 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) 70°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, 100,000 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.

The technical analysis and design concepts that have been formulated for the residential end-use application utilize the unique features of pulse combustion as the general technical approach for integrating all combustor system components.

These unique features of pulse combustion are also the basic considerations in formulating conceptual designs. The most important of these features are:

- o High combustor heat release rate
- o Capability to produce pressure boost
- o Availability of a pulsating flow field
- o Ability to deeply stage the combustion process

The high-release rate can be provided with pulse combustion represents a major element in formulating the design concepts. Due to their volumetric heat release rate capabilities, residential units can be sufficiently small to be substituted for existing home furnace and water heaters.

The ability of the pulse combustion system to be self aspirating and to develop a pressure boost was also considered in formulating conceptual designs. The pressure boost capability will be utilized to overcome the pressure drop associated with the use of a cyclonic or bag filter device for ash separation. In combustor concepts where two-stage combustion is implemented for NO_x control or enhanced efficiency (Option No. 3), the high velocity at the discharge of the resonance tube can be utilized to aspirate air used in the second combustion stage. The aspiration of the second stage air will allow a high degree of combustion staging without the need for auxiliary fans or blowers.

The pulsating flow field established within the resonance tube of a pulse combustion system can be utilized advantageously. These pulsations can be used to enhance ash agglomeration and collection within the resonance tube. Also, if it should become advantageous to burn high sulfur coal in the home,

dry sorbent material such as limestone, dolomite or Trona can be injected into the resonance tube to be calcined in the flue gas.

A design concept for small space and water heaters considers the size of the combustor, geometry of the resonance tube, bag filter, second stage cyclone and heat exchanger and their integral, system configurations optimization. This will ensure that the entire combustion furnace, as a unit, will fit within the available space typical of such equipment in residential environments.

In addition to the above preliminary checks, several major design issues must be considered in formulating the design concepts. The most important of these are the following:

- o Method of integration of the pulse combustor with peripheral and integral equipment
- o Approach to achieving compliance with NO_x and SO₂ regulations (not applicable to the baseline residential case per the PRDA application requirements).

In formulating the combustor concepts, a selection was made with regard to the type of combustor to be used, the characteristics of the air inlet valve design and the method of ash rejection. The selections made and the basis for these selections are discussed below.

1.1.1 Combustor Type

The use of the Helmholtz type combustor was selected in view of its superior combustion performance relative to the Schmidt type. In addition, thermal efficiencies are greater with the Helmholtz combustors than with the Schmidt tube combustors. This is attributed to the highly resonant nature of the Helmholtz configuration which tends to yield higher pressure fluctuations per Btu/hr of firing. This, in turn, improves the combustion efficiency for solid fuels.

In addition, the Helmholtz combustor was shown to provide greater pressure boost, which may be necessary to overcome pressure drops associated with ash rejection. The Helmholtz combustor has also demonstrated a high volumetric heat release rate, which is necessary to achieve combustor compactness.

1.1.2 Air Inlet Valve Design

Two types of air inlet valves can be used in a pulse combustion system; a mechanical or an aerodynamic valve. Based on the literature review, the mechanical valve can provide high pressure boost and, with rotary drive valves, some control over pulsating frequency. However, valve reliability over an extended period of time was shown to be poor. Further deterioration in performance and reliability of a mechanical valve was anticipated during

the combustion of solid fuels (like CWM) since ash deposits can cause damage to valve components. For these reasons, a selection was made to use an aerodynamic valve in our laboratory-scale development program. This type of inlet valve has proved to be both flexible and satisfactory and has been used almost exclusively in all combustors surveyed in the literature that were fired on coal.

1.1.3 Method of Ash Rejection

Two options of ash rejection are available, condensate or dry. The simplest approach would be to employ a system in which the exhaust gas is routed to a bag filter without further condensation (above 120°F exit gas temperature), the achievable thermal efficiency for the overall system is predicted to be comparable to conventional home units which burn natural gas or fuel oil. For a non-condensing MTCI system, capital cost will be significantly reduced when compared with a condensing system, but fuel costs will be approximately six percent higher.

Higher reliabilities that are commensurate with existing residential furnace histories can be achieved by virtue of pulse combustor system simplicity, especially for the non-condensing system design.

Automatic ignition with a 5-minute or less response time to full load (100,000 Btu/hr) is attainable based on MTCI's experience in pulse combustor design and testing, and SUR-LITE Corporation's experience in burner/ignitor systems.

Safety will be as good or better than conventional furnace safety since CWS is inherently safer to store and handle than natural gas or fuel oil. This can be expected to hold true for efficient combustor designs where carryover or entrainment of unburned particles to downstream equipment units does not occur. Clearly, natural gas and oil-fired systems have the advantage that unburned particles do not pose a safety hazard. By proper design of the controls which regulate air-to-fuel stoichiometry and trigger alarms and shutdown switches, safety can be enhanced.

Steady-state efficiency of 80 percent is deemed to be attainable. However, long-term tests must verify this. Since others, Lenox and Hydro-Pulse, have developed highly efficient, commercial pulse combustor home furnaces for the residential market, and since the peripheral components such as heat exchangers are state-of-the-art, MTCI is confident that the steady-state efficiency requirement can also be demonstrated.

Combustion efficiency of greater than 99 percent will be one of the primary technical challenges during Phase I and II engineering and testing. MTCI has already achieved efficiencies in the environment of 95 percent at the 1 MMBtu/hr level. At the smaller size, the achievement of near-complete combustion within the combustor unit can be well achieved by the use of correct, and conservative (sufficient size, frequencies, velocities and residence time) scaling parameters.

Semi-annual, scheduled maintenance can be readily achieved by correct process design selection. For condensing systems, condensate accumulation would be approximately 345 gallons every six months. For the more desirable, non-condensing system, even this requirement would vanish. The quantity of ash gathered from the bag filters or cyclone would be nearly trivial at 126 pounds every 6 months or a monthly accumulation of 21 pounds during the heating season which is easily handled by disposable storage bag filters. Nevertheless, the requirement of changing filter bags or cloths, if a filter is chosen, will impose additional maintenance during semi-annual clean-outs.

Based on existing combustor unit sizes and that of current commercial pulse combustor furnaces, the size constraint of 6 feet in height and 15 square feet of floor space appear to be readily attainable. Admittedly, current commercial units do not burn coal, and thus, should be more compact. Still, by integrating space and water heater exchange equipment into the combustor, operating in the non-condensing mode, and selecting filter/cyclone units that conform to size constraints, the specified physical envelope can be achieved.

Service life of the pulse combustor system can be expected to be as good as that which is commonly experienced by other coal combustor systems. One important difference is that higher volumetric heat release rates are obtained in pulse combustors than in conventional units. However, 40 years of prior testing experience by others, although mostly at the experimental stage, suggest that materials are available which guarantee a service life that is consistent with industry standards.

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 Btu/ft³-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 2-1 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 2-1
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 2-1, 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.

The technical analysis and design concepts that have been formulated for the residential end-use application utilize the unique features of pulse combustion as the general technical approach for integrating all combustor system components and formulating conceptual designs.

SECTION 3.0

PROGRAM DESCRIPTION AND WORK STATUS

3.1 PROGRAM DESCRIPTION

The systematic development of the residential combustion system is divided into three phases. Only Phases I and II are detailed here. Phase I constitutes the design, fabrication, testing, and evaluation of a pulse combustor sized for residential space heating. Phase II is an optional phase to develop an integrated system including a heat exchanger. Phase III is projected as a field test of the integrated coal-fired residential space heater. The program logic is depicted in Figure 3-1.

The objective of Phase I is to develop an advanced pulse coal combustor at the 100,000 Btu/hr scale which can later be integrated with a heat exchanger and controls to form a residential space heater. Phase I is comprised of four technical tasks which are described below.

TASK 1:

Task 1.1 Design of 100,00 Btu/hr pulse combustor

Although small pulse combustors (100,000 Btu/hr) have been built for natural gas and oil, none have been built for coal firing. Coal combustion requires more residence time than oil and gas. This is more of a factor for coal-mixtures, since the water needs to be evaporated first before the coal can burn. The design of the small pulse coal combustor should allow for the longer residence time requirements of slurry fuels, as well as the need for proper atomization of the slurry into the combustion chamber. In this regard, micronized coal slurries, such as the OTISCA coal slurry, may be better for small combustors since smaller coal particles would require less residence time for complete combustion.

Similarly, two chamber geometries will be designed and tested to arrive at a suitable combustor. With regards to the tail pipe, a single-tube tail pipe and multi-tube exhaust system will be designed for evaluation. A cyclone will also be designed to remove the particulate matter. In one case, the cyclone may be used as a second-stage combustor as well. All the components shall be made of metal and shall be capable of high temperatures. All accessories such as the slurry feeding system for the development unit will be specified.

An ignitor, pressure fluctuation monitoring sensor, temperature sensors, and other necessary control and monitoring instrumentation will be specified. SUR-LITE Corporation will be employed on a consulting basis for their expertise in ignitors and control instrumentation.

FIGURE 3-1: 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 3-1 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 3-1: 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 3-2 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 3-2: 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

Design of a scaled-down combustor was completed and a design of new chamber configurations at the present scale was also completed. The scaled-down design employed cylindrical chamber configuration maintaining the same shape of the initial feasibility unit such that scale-down is the only change. The new chambers designed for the present scale (nominal 1.5 MMBtu/hr combustor) are refractory lined and provide modular design to allow testing of two configurations having variable, conical sections, and cylindrical sections. Thus, the combustion chamber configuration change is maintained at the same scale, to initially evaluate this variable without scale-down.

Both the scaled-down and the new chamber configurations were fabricated. The scaled-down unit is a bare metal unit with a divergent tail pipe. The new chamber configuration unit was assembled maximizing the use of components fabricated under the previous contract (DE-AC22-83PC60419).

Testing was initiated on both units mentioned above using both CWM and dry coal conveyed with air. Some fuel feed problems were encountered. Tests

are exploratory at this stage with emphasis on combustor tuning and fuel injection.

Technical program direction and kick-off was held in Santa Fe Springs, California. A technical strategy was developed that calls for simultaneous scale-down with no change in chamber and combustor configuration and change in chamber configuration(s) at the present scale. This was adopted to isolate the effect of scale-down and new chamber configuration design separately. The results would be factored into the scale-down design and would also be available should scale-up to a pilot combustor unit (15 MMBtu/hr) become desirable. Initial focus at the residential scale unit will be on methods of fuel injection and atomization in a bare metal unit. This will be followed by incorporating changes to advanced chamber designs and testing of refractory lined units at the residential scale. Multi-fuel capabilities will also be established. A draft management plan was also generated.

SECTION 4.0

TECHNICAL DISCUSSION OF WORK COMPLETED DURING THE REPORTING PERIOD

This section discusses the technical results of the work completed during the period September 29 through December 28, 1986. The technical discussion is primarily associated with the design and fabrication of the advanced combustion chambers. Although testing was initiated during this period, the evaluation and data reduction were considered to be explanatory with emphasis on system shakedown, combustor tuning and fuel injection.

4.1 ADVANCED CHAMBER DESIGN

Scaling down the pulse combustor from 1 MMBtu/hr to 100,000 Btu/hr introduces some new design issues. In addition, the residential end-use introduces a number of operational and performance requirements as well as constraints.

4.1.1 Scale-down Issues

In scaling down from 1 - 2 MMBtu/hr to 100,000 Btu/hr, the combustor volume and dimensions can, in the first order, be reduced proportionately. This assumes that heat release rates per unit volume/hr could be maintained the same. A three dimensional geometric scale-down, however, will impact the frequency of operation since the device normally operates at a frequency approximated by:

$$f = \frac{c}{2 L_t} \sqrt{\frac{V_t}{V_c}} \quad (\text{Helmholtz}) \quad (I)$$

Where: f = Frequency in Hz

c = Effective speed of sound in the combustor (function of gas gas species and temperature profiles)

L_t = Length of tail pipe (or resonance tube)

V_t = Volume of tail pipe

V_c = Volume of combustion chamber

The ratio V_t/V_c will not be varied by the geometric scale-down; however, L_t will be reduced and the combustor frequency will therefore increase. There are constraints, however, on the increase in combustor frequency for a number of reasons.

The optimum range for particle size distribution of the coal (in CWM or dry coal) for combustor operation is related to the operating frequency of the combustor. Fine coal particles would be advantageous with combustors designed for high frequency operation. High frequency units operating on fine coal as fuel would achieve higher heat release rates, pressure boost and, in turn, high exit kinetic energy. Units scaled down in this manner would have essentially the same residence time at various fractions of full firing rate when compared to the existing combustors.

There is a limit, however, on increase in frequency since burn rate must be sufficiently high to "keep up" with the combustors cycle with effective excitation of the combustion induced oscillations.

The scaled combustor frequency can be modified, however, by modifying the V_t/V_c ratio. This can be achieved by either increasing the combustion chamber volume or reducing the tail pipe volume or both.

Increasing the combustion chamber volume requires some additional consideration. This includes both chamber geometry modification and other combustor tuning constraints for stable operation and good turn-down and combustion staging capability. Increasing the chamber volume while retaining a cylindrical chamber shape and increasing chamber diameter only, will affect flow patterns, particularly at the transition between the aerodynamic valve and the maximum chamber diameter. This will, in turn, influence the vortex shedding pattern in the transition region (Figure 4-1) and, therefore, mixing of fuel and air. It is most likely that the chamber geometry must be modified to accommodate both the volume increase and the required transition between the aerodynamic valve and the chamber. Tuning consideration, on the other hand, stem from the need to satisfy the quarter wave frequency, constraint in addition to that given in equation I above. The quarter wave frequency equation requires that the operating combustor frequency must be in the vicinity of:

$$f = \frac{c}{4 (L_t + L_c)} \quad \text{(Quarter Wave)} \quad \text{(II)}$$

Where: f = Combustor frequency (Hz)

c = Speed of sound

L_t = Length of tail pipe

L_c = Combustion chamber effective length (includes part of the aerodynamic valve)

Thus, for proper combustor tuning, both equations I and II must be observed. This, in turn, imposes some constraint on chamber length and shape. Quadratic form generators can be employed to address these requirements while

FIGURE 4-1: TRANSITION BETWEEN THE AEROVALVE AND CHAMBER

FIGURE 4-2: CHAMBER WITH QUADRATIC FORM GENERATORS

maintaining appropriate flow patterns in the chamber in general, and in the transition between the aerodynamic valve/tail pipe and the chamber in particular (Figure 4-2). The transition region between the chamber and the tail pipe is responsible for shedding vortices on return of hot gases from the tail pipe during flow oscillations (Figure 4-3). These vortices, of hot combustion products, together with the hot chamber wall temperature, are responsible for auto-ignition of the following combustion cycle.

In the case of the tail pipe, reduction in the tail pipe volume (to reduce the combustor frequency) can be affected by reducing the tail pipe diameter. This is necessary since reduction in the tail pipe length would only throw the combustor off tuning. This can be gleaned from examining both equations I and II above. The combustor frequency in equation I (Helmholtz) is proportional to $\frac{1}{L_t}$ while, in equation II (quarter wave

it is proportional to $\frac{1}{L_t}$ (for $L_c \ll L_t$).

There is, however, a limit to the reduction of the tail pipe diameter. With the small size combustor, the pressure drop in a small tail pipe having a small diameter would compromise the combustor performance because of the increase in gas viscous forces with respect to inertial forces. This introduces excessive damping in the resonant mode of the combustor which, in the limit, inhibits the combustion induced oscillation and impair the combustor performance (burn rate, carbon conversion, pressure boost, etc.).

In addition to the potential problem with increased combustor damping, another constraint on the reduction of the resonance tube diameter is the limitation on maximum flow velocities in the tail pipe. Clearly, the maximum flow Mach number must be maintained comfortably less than 1.0 at the gas temperatures in the tube.

The second most important issue with down-scaling the combustor relates to fuel injection. At a firing rate of 100,000 btu's/hr, the fuel flow rate, in the case of CWM, would be on the order of 10 lbs/hr or (1/360) lb/sec. This flow rate is very small and gives rise to difficulties in scaling down the fuel atomizer design particularly because of the non-Newtonian nature of the CWM fluids. It is conceivable, however, that a micronized CWM fuel with appropriate additives could be employed and successfully atomized at this low flow rate. Thus, the atomizer scale-down is expected to require "scale-down" of the CWM particle size distribution as well. Standard grind CWM fuels will still be utilized.

4.1.2 Application Issues

In the residential applications, additional consideration must be addressed in both the system and combustor design. These considerations include noise reduction, methods of ash rejection, pollution control, system reliability, safety, ease of operation and maintenance, and overall life cycle costs.

FIGURE 4-3: VORTEX SHEDDING ON RETURN OF HOT GASES FROM THE RESONANCE TUBE

Noise reduction is an important issue with residential applications of pulse combustion. There are two methods of noise reduction available; namely, passive and active. In passive noise reduction (Figure 4-4), an inlet plenum and exit expansion plenum can be employed to act as mufflers of sound. The expansion chamber shown here is a cyclone. In other design concepts, it is sufficient to have an inlet plenum and multiple resonance tube enclosed in a hot water heater (Figure 4-5). Such a hot water heater could be employed to supply both hot water and space heating needs in certain residential applications. The insulated hot water will be employed to damp the noise emanating from the resonance tubes.

In active suppression of noise, two tandem combustors have been very successfully used in large burners (U.S.S.R. fired such a unit at 170 MMBtu/hr; Figure 4-6). In the case of the residential application, however, this would mean that we would have two 50,000 Btu/hr units operating in tandem, which would be too small. Therefore, the active approach may only be suited for larger residential systems (apartment buildings, etc.) fired at rates higher than 300,000 or 500,000 Btu/hr. However, in large residential applications, the whole system could be housed in a remote room and active noise suppression may become less of a requirement.

The potential complexity of active noise suppression and related operational reliability issues, suggest that the initial focus should be placed on passive noise suppression techniques using a single combustor to enhance reliability and improve maintainability, etc. of the small residential systems (<300,000 Btu/hr). This approach is further substantiated by the Lenox gas-fired pulse combustor space heater experience. The Lenox system employs one combustor and passive noise suppression schemes.

All methods of ash rejection which are being considered focus on dry ash rejection. A slagging approach is not believed appropriate for small residential applications. One of the concepts involves the use of a cyclone at the combustor's exit (Figure 4-7). This design approach is amenable to evaluation during the combustor development tests and does represent a simple configuration for initial introduction of this technology to the residential end-use sector. Methods for ash collection downstream the heat exchange surfaces can meaningfully be tested later within the total system tests in Phase II.

As for pollution control, other than ash rejection, a number of design requirements and constraints were considered. The nitrogen oxide pollution aspects are dependent upon the following considerations.

First, pulse combustors tested by MTCI and by others, as reported in the literature, tend to be low NO_x devices due to the high rate of mixing and short burnout times. High mixing rates result in a more uniform temperature in the combustion zone, for a given stoichiometry. This in turn results in lower peak temperatures and less thermal NO_x production. In addition, the high heat release rates and the associated lack of oxygen diffusion impedance in reaching the surface of the burning coal particle tend to shorten the particle burnout time. This, in turn, reduces the residence time of nitrogen with oxygen at the combustion temperature and in turn reduces the rate of

FIGURE 4-4: PASSIVE NOISE REDUCTION

FIGURE 4-5: INTAKE PLENUM/MULTI-RESONANCE TUBE DESIGN

FIGURE 4-6: USSR TANDEM UNITS (170 MBTU/HR)

FIGURE 4-7: DUAL-STAGE COMBUSTOR WITH SINGLE-STAGE HEAT RECOVERY

production of nitrogen oxides. NO_x production will also be influenced by the characteristics of the coal-based fuel which is employed. The higher the fuel bound nitrogen content, the more NO_x production.

In general, the fuels that may ultimately be provided for residential applications are expected to have low amounts of fuel bound nitrogen. Nevertheless, it is prudent, at this stage, to develop a combustor design which has some combustion staging capability for NO_x reduction.

Sulfur removal, if required, can be implemented through injection of sorbents in the appropriate location in the combustor's resonance tube. The spent sorbent removal would be downstream the combustor with the removal of ash.

Other considerations; including system reliability, safety, and system life cycle costs; were not fully examined at this stage. However, consideration of these issues will be initiated as Phase II is approached.

4.1.3 Conceptual Designs of the Commercial Combustor Unit

There are two fundamental pulse combustor designs that are planned by MTCI for the residential applications, namely: 1) single resonance tube independent combustor and, 2) multiple resonance tube integrated combustor.

MTCI has built both types of combustors in conjunction with CWM combustion (single resonance tube) and biomass gasification (multiple resonance tube). In the case of biomass gasification, the multiple resonance tube combustor was fired with natural gas. The system configurations that are envisioned by MTCI for each type of combustor are provided in Figures 4-5 and 4-7.

The scaled-down combustor design and the new chamber design are shown in Figures 4-8 and 4-9. The scaled-down unit did not employ chamber geometry modifications to allow the investigation of the effect of scale-down as the only variable. New chamber configurations, however, were maintained at 1 - 1.5 MMBtu/hr combustor size to evaluate the modification in chamber geometry as the independent variable.

The scale-down combustor design shown in Figure 4-8 incorporates the following features:

- o Tubular aerodynamic valve having a constant ID with flared inlet section -- this provides for inexpensive and simple aerodynamic valve. The fluidic diodicity of this valve is derived from two effects. The first being the difference in density and viscosity between air being induced in the chamber and the products of combustion attempting to exit the chamber through the aerodynamic valve. The second effect is due to flow separation at the flared segment as the product of combustion flow backward in the aerodynamic valve.

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FIGURE 4-8: SMALL PULSE COMBUSTOR UNIT
(WITH PERTINENT DIMENSIONS)

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A. MODIFIED GRANDFORKS DESIGN

B. MODIFIED HANBY DESIGN

FIGURE 4-9: NEW CHAMBER DESIGN

- o Cylindrical Combustion Chamber with two conic (45°) transition sections at the chamber inlet and exit — this chamber has the same configuration as the Lockwood/SNECMA design used in the previous program. Thus, the only change is the combustor scale-down.
- o A continuously tapered resonance tube which is formed in a U shape for compactness. The slight taper in the tail pipe biases forward flow in the tail pipe and enhances the combustion air recharge performance of the combustor.

The dimensions of the small unit are shown in Figure 4-8. The new chamber design configurations included a modified Grandforks chamber (Figure 4-9a), and a modified Hanby chamber (Figure 4-9b) design.

In the Grandforks chamber design, the aerodynamic valve is orthogonal to the chamber/tail pipe axis. The "back-side" of the chamber is closed and the return flow must make a 90° turn to exit from the aerodynamic valve. This design also provides for higher peak pressure fluctuations, per Btu/hr of fuel firing, in the combustion chamber.

In the modified Hanby chamber design, the chamber expands from the inlet to a straight cylindrical section of 5.75" diameter. This diffuser section provides for pressure recovery and flow deceleration (mean flow).

In both cases, the chamber volume is approximately the same. The tail pipe was made to telescope so as to provide for combustor tuning. In addition, both aerodynamic valves can be modified by using inserts and in the case of the Grandforks unit, the aerodynamic valve can also telescope to change its length.

Based on the test experience with these chamber configurations, MTCI will further modify the design to provide the chamber configuration for the commercial unit which would be used in total systems tests in Phase II.

4.2 COMBUSTION FABRICATION

Methods of fabrication of advanced chamber configurations having quadratic form generators were investigated during this reporting period. This included fabrication of both refractory lined and bare metal components for the residential combustors. The approaches evaluated included:

- o Bare metal units fabricated from off-the-shelf parts available in stainless steel supply catalogs.
- o Metal spinning of specially designed quadratic form shapes.
- o Machining of mandrels having specially designed quadratic form generators to be used for casting refractory lined components.

Considerations were made to the needs of both the laboratory developmental test program and the manufacturing issues for the commercial

configuration system, once the development work was completed. A primary consideration for the test program is the speed by which modifications could be made and the cost of implementation. The primary considerations for manufacturing are the cost of manufacturing and the ability to maintain optimum design configurations without compromise to performance.

4.2.1 Bare Metal Units Fabricated From Standard Parts

With the small size of the residential pulse combustor, the possibility of using standard, off-the-shelf stainless steel parts represents an option. Standard parts which include tubing, bell reducers, conical reducers, etc., could be welded and, as appropriate, flanged to fabricate the pulse combustor units. This approach is quite attractive for laboratory test activities, particularly at the exploratory stage of a wide spectrum of design configurations and when one or two units need to be fabricated in a short time for test purposes. The cost of such components is moderate and most parts should be readily available.

This approach, however, will constrain the configurations and the combustor designs that can be explored since bare metal standard shapes are the building blocks. This may also result in constraining the scope of the developmental test program to bare metal components (chambers, tail pipes, etc.) which are also limited in configuration flexibility. A review of the available stainless steel off-the-shelf components was made to assess the availability of appropriate shapes, material properties and thickness, etc., and to explore the costs. As a result, this approach was rejected (as the primary approach for combustor fabrication) for the following reasons:

1. In pulse combustion, the acoustic, fluidic and aerodynamic characteristics of the combustor configuration dictate performance. Therefore, constraints on design configurations at this stage are unacceptable.
2. Tests with refractory lined chambers and tail pipes are required to complete the scope of investigation.
3. The cost of the off-the-shelf parts was not all that attractive and delivery times for complex geometries from some suppliers was unacceptably long, i.e., it seemed that these parts were made as special orders even though they were in the catalog, which reflected in both delivery time and cost.
4. More design modifications and tests may be required to converge on the commercial configuration units which would ultimately be specially manufactured anyway.

It was, therefore, decided to explore the following two approaches for fabrication of advanced combustor configurations.

4.2.2 Metal Spinning

Another approach to fabrication of flexible component and combustor configuration is metal spinning. In this approach, a mandrel is machined and metal is spun to form the desired axisymmetric shapes. In this approach, significant flexibility is available for the component shapes including quadratic form generators (circular arc, parabolic, hyperbolic, etc.). The metal thickness, however, is limited and some limitations on material properties come into play. The material must be relatively thin and must have good ductility so it will flow properly during the forming process. The thickness of the material can be increased somewhat and materials with relatively low ductility (such as Inconel) can be used; however, heat must be applied during the spinning processes. Depending on the extent of material flow requirement to fabricate the part, stainless steel including RA330 (which has oxidation and strength characteristics for high temperature applications similar to Inconel but with higher ductility) can be spun with thickness of 0.10 to 0.15".

Metal spinning is appropriate for both test activities and mass production of combustor parts, particularly chambers and aerodynamic valves of optimum configurations. Tail pipes may also be made using metal spinning methods for the small residential combustors, depending on the final design.

For test activities, hard-wood mandrels can be made quickly and at a relatively low cost. These mandrels can be used to spin a few parts. For mass production, however, hard-metal mandrels are required. Thus, for laboratory tests, heating the metal during the spinning process is not a viable option, nevertheless, conventional stainless material with small thickness is adequate for short duration testing of new design configurations.

This approach was, therefore, selected for fabrication of advanced configuration chambers and aerodynamic valves (bare metal). A set of fabrication test runs were made using shapes with quadratic form generators similar to what MTCI anticipates for the combustor chamber. The results were excellent.

4.2.3 Refractory Lined Components

In our discussions with the metal spinning shop personnel, it became evident that the mandrel design could be made with both metal spinning and casting of refractory-lined components in mind.

Mandrels for metal spinning could be assembled in a composite mandrel for casting chambers and aerodynamic valves. This is also possible for small tail pipes. The convenience of using the metal spinning mandrels for refractory-lined casting is valuable for the test program; however, this would not be a major consideration for manufacturing. Therefore, for the time being, every effort will be made to design the mandrels used for metal spinning such that they can also be used for casting refractory-lined components to the extent possible.

MTCI believes that it is important to address the above mentioned issues early in the program so as to evolve a commercial configuration combustor technology that is also manufacturable at a reasonable cost with sufficient design flexibility for optimized performance.

4.3 DEVELOPMENTAL TESTING

Exploratory testing at 1-1.5 MMBtu/hr on the modified Grandforks combustor was completed and tests using the modified Hanby chamber design were initiated. These configurations were shown in Figure 4-9. These tests are intended to provide data and test experience with these new chamber designs and to provide the basis for an optimized chamber design for the commercial configuration units.

During the initial test period (modified Hanby design) using natural gas to tune the system, a number of aerodynamic valve inserts were used to modify both the valve's fluidic diodicity and minimum flow resistance. In addition, the tail pipe was telescoped with the various aerodynamic valves while monitoring the dynamic pressure oscillations in the chamber and the static boost pressure. During the combustor tuning period, gas injection location and injector design were also varied.

A weld failure developed in the chamber/tail pipe interface during the initial test runs, which was repaired. The combustor performance shown increased peak dynamic pressure and boost, suggesting increased heat release in the chamber upstream of the tail pipe.

In the subsequent test runs, the method used to provide for tail pipe cooling employing the pulsating flow field at the aerodynamic valve and the kinetic energy in the flow at the resonance tube exit shown in Figure 4-10. In this configuration, air is induced, by the pulsating flow field at the aerodynamic valve, into both the combustion chamber (first-stage air) and a jet-pump (second-stage air). The amount of air flow in the second-stage air duct could be regulated by a damper in the second-stage air duct. The second-stage air is employed within the resonance tube shroud to cool the resonance tube. At the exit of the shroud, the second-stage air flow is further induced by the kinetic energy in the flow at the pulse combustor's exit.

The temperature of the tail pipe material is maintained sufficiently low ($< 2100^{\circ}\text{F}$) to avoid the tube failure, but also sufficiently high to achieve complete carbon conversion and low CO content in the flue gas. Therefore, in this combustor design scheme, which would allow a bare metal resonance tube, a refractory-lined chamber is used and the aerodynamic valve must be designed so as to control the stoichiometry of the pulse combustor with the second-stage cooling air in mind. The second-stage in this instance would be a cyclone which would also be employed for particulate matter rejection.

As a result of the initial test experience with both the modified Grandforks and the modified Hanby combustor, the following technical issues and design concepts were brought forth.

FIGURE 4-10: RESONANCE TUBE COOLING SCHEME

4.3.1 Fuel Feed

Controllable dry coal feeding remains to be a persistent problem and is compromising the pace of progress in the combustor's developmental test program.

To explore solutions, a number of coal feeders made by Vibrascrew, Acrison and K-Tron were considered. These range in price (verbal quotes) between \$17,500 to \$25,000 depending on the model and the options included. Our options at the present time are:

i) Purchase one of these reliable feeders that has the capability of delivering the required range of dry coal feed rates with both pulverized and micronized coal and with reasonable variation in actual versus nominal feed rate settings.

ii) Modify the old Vibrascrew unit, that was obtained from JPL under the MTCI/EPA program, to obtain higher pressure operation (from presently 1.5 psi to 5.0 psi) and wider range of coal feed rates. The higher pressure capability is required for pneumatic conveyance of coal against combustor boost pressure and line pressure drop. The range of feed rates could potentially be expanded by using a number of feed screws having different pitch/diameter.

iii) Further modify the MTCI biomass feeder; the biomass feeder which was used by MTCI in the DOE/SBIR program (Figure 4-11) was modified and used to feed dry coal; however, further redesign and modification is necessary. The additional modifications considered include: 1) reducing the gap between the metering screws and the holes in the feed plate or installing short pipes with small clearance between the pipe and the screw, 2) changing both pitch and gearing ratios for the metering screws, 3) reducing clearance between the main feed screw and the pipe, and 4) installing high frequency vibrators for both the main and the secondary feed bins.

The first option (option i) is the most reliable; however, MTCI did not include an estimate for such feeders in the initial program funding. We planned to use both of the feeders we currently have; the Vibrascrew, which was obtained from JPL, for lower feed rates, and modify the biomass feeder for higher feed rates. Further modification of both the Vibrascrew and the biomass feeders represent not only significant cost but also risk and potential for program delays.

4.3.2 Chamber Design

An advanced chamber design optimized for pulse coal combustion evolved as a result of the developmental test experience to date. The general design configuration is shown in Figure 4-12. The objective of this design is to employ quadratic form generators that are used to define an axisymmetric chamber in order to accommodate a number of design and chamber performance attributes. At this stage, the quadratic form generators were chosen to be simple arcs of a circle.

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FIGURE 4-11: MTCI BIOMASS FEEDER

NOTICE: MTCI INTENDS TO FILE FOR PATENT RIGHTS TO THIS CHAMBER DESIGN AND TO SECURE THE RIGHTS TO THE PATENT TO THE EXTENT ALLOWABLE UNDER THE CONTRACT PROVISIONS WITH DOE. IT IS REQUESTED THAT THIS DESIGN INFORMATION BE TREATED ACCORDINGLY.

FIGURE 4-12: MTCI ADVANCED CHAMBER CONFIGURATION

In order to discuss the design and performance attributes of the advanced chamber configuration, chamber flow characteristics during the relevant portions of both air intake and chamber exhaust are depicted in Figures 4-13 and 4-14, respectively.

During air intake, the chamber volume is subdivided into three regions as shown in Figure 4-13. Region I is further subdivided into subregions I-A and I-B. In subregion I-A, the air flow entering the combustion chamber is retarded in a shallow diffuser section. The reduction in the mean flow velocity is, therefore, achieved with efficient static pressure recovery. This persists until the rate of change in the chamber's cross-sectional area becomes sufficiently large to cause flow separation and, in turn, give rise to vortex shedding as shown in Figure 4-13 at the start of subregion I-B. Retarding the flow with efficient pressure recovery in subregion I-A is aimed at serving two functions. First, the static pressure at the boundary between subregions I-A and I-B will be higher than the static pressure at the inlet to the chamber. Therefore, for a given negative pressure in subregion I-B and region II (the mechanism by which a pulse combustor recharges with air), the negative pressure at the inlet during air intake will be even lower. The net effect is that the recharge with combustion air will be as if the negative pressure in the chamber is inducing combustion air flow from an intake which has a cross-sectional area equal to that of the boundary between subregions I-A and I-B. This would enhance the inducement of combustion air in the chamber on intake but with high resistance to exhaust flow, as will be discussed below using Figure 4-14.

Second, the fuel which is usually injected near the chamber air intake, is provided with more residence time in the flow which is being retarded in subregion I-A and exposed to back radiation from the chamber walls and the combustion zone (depicted as region II in Figure 4-13). This allows some fuel preheat and conditioning for rapid combustion down stream. In the case of CWM fuels, this is particularly useful since it is necessary to vaporize the water in the slurry, in addition to the coal devolatilization and char preheat.

In subregion I-B, where shedding of vortices is initiated, vigorous mixing of the fuel and the induced combustion air takes place. In this design, the chamber configuration is employed to achieve mixing instead of the use of a bluff body or a flame holder in the chamber that were used in some conventional and pulse combustors by others in the past. Using a bluff body or a flame holder is not attractive due to the attendant material, life and reliability problems which may be encountered at the chamber operating conditions (high temperature and intense acoustic field). Achieving the proper mixing through proper design of the chamber configuration, instead of using bluff bodies and flame holders, also simplifies the combustor design and reduces the manufacturing costs.

An added advantage to the design configuration of region I is the potential for the reduction in the combustor performance sensitivity to variations in fuel specifications. Variation in fuel specification affect the fuel preheat requirements and the flame speed. These differences between oil, gaseous and coal-based fuels are significant. But even within coal-based fuels, variables such as the type of coal, volatile content, char reactivity,

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FIGURE 4-13: FLOW CHARACTERISTICS DURING AIR INTAKE

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FIGURE 4-14: FLOW CHARACTERISTICS DURING CHAMBER EXHAUST

(particle size distribution and moisture content will affect the required fuel preheat and flame speed at the ignition region. The diverging chamber cross-section between the inlet and the combustion zone provides for monotonic decrease in the mean flow velocity. This, in turn, provides for a zone in which the flame front can stabilize, where it must as a result of fuel characteristics, in a short distance along the combustion chamber axis (subregion I-B). This expectation is based on our test experience with conic and quadratic transition regions between the aerodynamic valve and the old cylindrical chamber designs used in our previous contract (DE-AC22-83PC60419). In these tests, the sensitivity to fuel injection location and method of injection was reduced with quadratic form generators. We should also note that similar design considerations enter into the design of a burner throat in the case of conventional burners.

The bulk of the chamber volume is comprised of region II, the combustion region. In this region, the mean flow velocity is at its lowest level within the chamber due to the large cross-sectional area. This is intended to force the release of most of the heat near the dynamic peak pressure zone in the combustor and hence enhance the pressure fluctuations (which incidently further intensifies the heat release rate) per Btu/hr ft³ of firing rate. The objective of this is to improve combustion efficiency over a wider range of turn down or deep combustion staging conditions with complete carbon conversion. This design is also intended to provide for an expanded product line for less tooling and manufacturing costs. The combustion chamber volume can be increased significantly by adding a short cylindrical spool section at the maximum chamber diameter without significantly changing the general flow characteristics in the chamber. Therefore, a range of firing rates can be accommodated readily with the same quadratic form generated parts to suit the application requirements. These chambers would then be used with the appropriate tail pipes and aerodynamic valves.

The shape of the chamber in region III and near the region II/region III interface is selected to enhance auto-ignition and control the magnitude of flue gas return from the resonance tube. The low pressure within the chamber, which induces the combustion air intake, ultimately causes a reverse flow to occur in the resonance tube, with some hot flue gases returning to the chamber.

The returning hot flue gases are rapidly retarded in region II with a very rapid increase in the chamber cross-section (in the returning flow direction). This intentionally inefficient diffuser section results in rapid flow separation and the shedding of hot flue gas vortices that travel upstream in the combustion chamber (toward the chamber's air inlet). This vortex pattern of hot flue gases meets the forward moving vortex pattern of combustion air and fuel mixture within the combustion zone. This, together with the back radiation from the chamber walls, provides for autoignition. The design of the portion of the chamber near the exit is also important to the combustor performance in turn-down and deep staging while maintaining complete carbon conversion. We should also note that excessive amounts of resonance tube return flow hinders the chamber's ability to recharge with combustion air and dilutes the gases in the combustion chamber.

To discuss the chamber design attributes during chamber exhaust, Figure 4-14 depicts the flow characteristics during the exhaust part of the chamber operation.

As the chamber pressure rises due to combustion, the products of combustion move towards both the resonance tube and the aerodynamic valve. In region I the flow is monotonically accelerated due to the monotonic reduction in the chamber's cross-sectional area in the direction of flow. This, in turn, causes the static pressure to drop until it reaches a minimum at the chamber's air inlet. The aerodynamic valve further impedes the return flow as discussed below.

In region III, the hot gases are made to accelerate very rapidly due to the rapid reduction in the chamber's cross-sectional area in the direction of the flow. The diffuser length is very short and the flow acceleration is maintained high, thus causing the boundary layer thickness to remain small in that region with reduced resistance to forward flow. The stream-lines in region III poses significant curvature as the flow accelerates towards the resonance tube exiting the chamber. The centrifugal and large linear acceleration of the flow in this region causes larger size coal particles to lag the flow due to their inertia. Ash particles and small coal particles would preferentially be entrained with the flow to the resonance tube. This is due to the larger surface-to-mass ratio of the ash and smaller size coal particles. The smaller size coal particles are further burned downstream in the resonance tube in the monotonically increasing oscillating velocity amplitude. Sufficiently large coal particles lag behind and are retained in the chamber until the next recharge and are hence exposed to the next combustion cycle. Thus, one of the objectives of the rapid transition between the large maximum chamber diameter and the significantly smaller chamber exit (to tail pipe) diameter, is to provide a preferential dynamic trap for the larger coal particles and retain them in the chamber until they burn to a smaller size.

In order to further illustrate the diodic effect of the chamber's inlet and exit diffusers, we now present the attributes of the diffuser-based aerodynamic valve design employed by MTCI. The MTCI diffuser-based aerodynamic valve design is delineated in Figure 4-15. In this design, two simple (conic sections) diffuser sections comprise the aerodynamic valve. At the inlet, a steep diffuser angle is used which can be between 40° and 60° (half cone angle). On the combustion chamber side, a generous shallow angle diffuser is used to provide for efficient pressure recovery (40° to 70°). The length of the diffuser sections and the minimum aerodynamic valve diameter are selected to meet the combustor integration and performance requirements. Through these variables the overall fluidic diodicity and minimum recharge resistance for a given mean flow rate can be modified.

Upon air intake, the flow characteristics are shown in the upper part for Figure 4-15 labeled Forward Flow. In both parts of Figure 4-15 the chamber is located on the right-hand side of the aerodynamic valve (not shown).

FIGURE 4-15: MTCI DIFFUSER BASED AERODYNAMIC VALVE

During air intake, the boundary layer build-up, which is monotonic in the direction of the flow, is compensated for by the diverging cross-sectional area of the shallow diffuser section (right-hand side). The intake stream lines also draw from a large area near the valve intake since there is no flow separation on intake from the steep diffuser because of the flow acceleration from a large to a narrow cross-section.

On exhaust of hot gases from the chamber, the flow characteristics are labeled Reverse Flow. As depicted in the figure, the boundary layer build-up over the length of the shallow angle diffuser in the direction of flow, together with the diffuser angle, cause the effective minimum diameter to be small. Flow is then separated from the steep angle diffuser with reverse flow causing the stream lines to remain within a small cross-sectional area for exhaust.

In addition to the above, the temperature of the air on intake is significantly lower than the temperature of hot gases from the chamber during reverse flow. This in turn causes the mass density of the incoming air to be higher and the viscosity to be lower than their counter parts for the hot gases leaving the chamber.

Both the flow characteristics and the difference in temperature between the intake air and the chamber reverse flow gases give rise to the aerodynamic valve fluidic diodicity.

With the fluctuating pressure in the chamber, the fluidic diodicity of the aerodynamic valve causes the net flow at the aerodynamic valve to be intake of combustion air at a self-induced level of stoichiometry.

The flow characteristics in regions I and III of the chamber are similar to those described for the aerodynamic valve shallow and steep diffuser sections of the aerodynamic valve, respectively. This similarity results in flow diodicity, which is in addition to that of the aerodynamic valve.

SECTION 5.0

PLANNED ACTIVITY FOR THE NEXT PERIOD

The technical activities planned for the next reporting period are focused upon the Developmental Testing of the Residential Unit (Lockwood SNECMA Design) and the advanced bare metal and refractory lined chambers. It is expected that the residential unit will be fired on gas, No. 2 fuel oil and micronized coal.