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Development of a Coal Fired Pulse Combustor for Residential Space Heating

PHASE I
FINAL REPORT

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EXECUTIVE SUMMARY

This report presents the results of the first phase of a program for the development of a coal-fired residential combustion system. This phase consisted of the design, fabrication, testing, and evaluation of an advanced pulse combustor sized for residential space heating requirements. The objective was to develop an advanced pulse coal combustor at the ~ 100,000 Btu/hr scale that can be integrated into a packaged space heating system for small residential applications.

The strategy for the development effort included the scale down of the feasibility unit from 1 - 2 MMBtu/hr to 100,000 Btu/hr to establish a baseline for isolating the effect of scale-down and new chamber configurations separately. Initial focus at the residential scale was concentrated on methods of fuel injection and atomization in a bare metal unit. This was followed by incorporating changes to the advanced chamber designs and testing of refractory-lined units. Multi-fuel capability for firing oil or gas as a secondary fuel was also established. Upon completion of the configuration and component testing, an optimum configuration would be selected for integrated testing of the pulse combustor unit. The strategy also defined the use of Dry Ultrafine Coal (DUC) for Phases 1 and 2 of the development program with CWM firing to be a product improvement activity for a later phase of the program.

The residential unit (scaled-down feasibility unit) was fabricated and fired on gas, No. 2 fuel oil, and coal (micronized Pittsburgh No. 8). The maximum firing rate on gas was 180,000 Btu/hr with a turndown of 3:1 to 60,000 Btu/hr. The start-up runs with No. 2 oil was satisfactory but the oil feed system was inadequate for steady-state operations and the tests were converted to coal operation utilizing the modified MTCI biomass feeder. The combustor was operated on coal for short periods of time between 54,000 - 154,000 Btu/hr within the limits of the modified biomass feeder performance.

Bare metal and refractory-lined combustor tests were conducted with the original feasibility unit modified to incorporate the advanced chamber designs. The evaluation of the advanced chamber design performance and operations provided the design data required for the design and fabrication of the scaled-down units. The bare metal chamber achieved a sustained boost pressure of 16 inches of water, the refractory-lined chamber 18 inches of water when operated in the tuned combustor configuration.

A review and evaluation of the data collected from the prior tests resulted in the choice of a tandem configuration (two parallel units) for the evaluation testing. Early tests conducted with single pulse combustors, with and without phased coal injection, indicated that phased feeding of coal enhances the combustion efficiencies. A tandem combustor configuration resolved the issue of fuel phasing and provided 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) could be assembled. The residential unit consisted of a simple conical design. Tests were conducted with micronized Pittsburgh No. 8 coal in the new single units. The combustors burned coal with no apparent problems. 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.

The tandem pulse combustors were next 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 the best 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 operating in the tandem mode.

Development testing of the tandem units was completed and three configurations for coupling of the units were evaluated. The technique chosen for the extensive optimization development testing was one in which the tailpipes were coupled at the exhaust plenum. The refractory chambers were fabricated, the system assembled and operated for a series of tests. The feasibility of wet ash collection was also verified. Four long-duration tests were completed (1-3 hours). In the first test, stable performance over a 3:1 turndown ratio was achieved with a carbon burn-out efficiency of 97.7 percent (based on ash analysis). Performance during all tests in the series was both stable and ran without support gas or excitation air.

A comparison of the experimental results with the target goals are shown in the following table:

COMPARISON OF EXPERIMENTAL RESULTS WITH TARGET GOALS FOR COAL-FUELED RESIDENTIAL SPACE HEATERS

ITEM	TARGET GOALS	EXPERIMENTAL RESULTS
Fuel Selection	DUC or CWM	DUC
Ignition	Automatic	Achievable
Response Time	< 5 minutes to full load	5 minutes with continuous pilot
Combustion Efficiency	> 99 Percent	> 97 Percent
NO_x (as NO_2)	< .5 lb/MMBtu	< .5 lb/MMBtu
SO_2	< 1.0 lb/MMBtu	< 1.0 lb/MMBtu
Height	< 6 Feet	< 6-7 Feet
Floor Space	< 15 Ft^2	< 10 Ft^2

As seen, the experimental results obtained achieved the target values except for combustion efficiency. Although 99 percent combustion efficiency was not achieved, significant improvements in the level of carbon burn-out were progressively achieved through the test phase. An evaluation of the test results indicated the >99 percent combustion is achievable with properly controlled and adjusted conditions.

Based on the performance of the units during both the preliminary configuration testing and the long-term optimization evaluation testing, an optimum configuration for the residential heating system was chosen for the Phase 2 effort. The key features include the tandem configuration with tailpipe coupling, refractory-lined conical combustors, automatic fuel biasing and a wet cyclone particulate trap.

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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. These applications represent a significant market for coal utilization.

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:

- Availability of combustor technology in the size range required that is reliable, safe, and that can be integrated in a cost-effective system.
- 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.

Definitive data is presently lacking regarding the relative economic and logistic preference for CWM versus DUC fuels in residential applications. It is probable that the optimum fuel choice may be dictated by considerations specific to individual geographic regions. For this reason, MTCI believes that it is prudent to develop residential systems offering the flexibility to handle both types of fuels. The final selection of fuel choice for performance evaluation of the combustor will be made based on the results of development testing and preliminary analysis of optimum system configuration.

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 dry ultrafine coal fuels that can be readily conveyed by pneumatic means and 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. MTCI's pulse combustor-fired furnace will be designed for some fuel flexibility, allowing use of either CWM or DUC and an ability 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 75%
Combustion Efficiency:	Greater than 99%
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 $15 Ft^2$
Service Life:	Greater than or equal to 20 years
Fluid Reheat Temp:	20^oF for Water (From 180^o - 200^oF) 70^oF 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 provide the basic considerations used 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 that 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.

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 features, several design issues must also 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.

- Approach to achieving compliance with NO_x and SO_2 regulations (not applicable to the baseline residential case per the PRDA application requirements).

In formulating the system concepts, selections with regard to the type of combustor to be used, the characteristics of the air inlet valve design and the methods of ash rejection were considered. The selections made and the basis for these selections are specifically addressed in subsequent sections of this report.

The reliability, maintainability, and availability of this consumer-focused system must be commensurate with existing residential furnace history. It is projected that these levels of operability and service can be achieved by virtue of the pulse combustor system simplicity. This will result in maintenance requirements and service life equivalent to existing systems.

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 the SUR-LITE Corporation's experience in burner/ignitor systems.

Safety will be as good or better than conventional furnace safety since CWM is inherently safer to store and handle than natural gas or fuel oil and DUC is now being handled routinely in Europe in inerted and pressurized containers. 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. With the proper design of the controls which regulate air-to-fuel stoichiometry and system operations, safety standards can be easily achieved.

Steady-state efficiency of 80 percent is deemed to be attainable. However, long-term tests must verify this. Since others, Lennox and Hydro-Pulse, have developed highly efficient, commercial gas-fired 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. At the smaller size, the achievement of near-complete combustion within the combustor unit can be accomplished 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 6 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 and consumer expectations.

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.

1.2 RESIDENTIAL SPACE HEATING SYSTEM DESCRIPTION

Regardless of the type of equipment to which it is applied, a CWM or DUC-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 1-1 compares calculated values of combustion gas quantities and compositions for natural gas, DUC, and CWM fuels. Typical analyses of these fuels were assumed in the calculations. Combustion of all fuels was arbitrarily assumed to occur at 15 percent excess air. As shown in the table, combustion gas from the selected DUC or 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

TABLE 1-1:

COMPARISON OF COMBUSTION GASES PRODUCED BY GAS, DUC, AND CWM FUELS

	NATURAL GAS	DUC	CWM (70% SOLIDS)
TYPICAL ANALYSIS (wt. %):			
Sulfur	0	.69	.58
Hyrdogen	22.68	5.14	5.47
Carbon	69.26	85.19	84.30
Nitrogen	8.06	1.57	1.46
Oxygen	0	4.33	4.92
Ash, wt. %	0	1.16	1.35
Moisture, wt. %	0	1.92	1.9
	------	------	------
	100.00	100.00	100.00
HIGHER HEATING VALUE, Btu/lb	21,800	15,016	15,056
SLURRY WATER, lb/100 lb fuel	0	0	42.85
COMBUSTION GAS COMPOSITION* (volume %):			
SO_2	0	.04	.04
CO_2	8.33	14.85	14.03
N_2	71.58	75.01	75.52
O_2	2.47	2.60	2.47
H_2O	17.60	7.49	11.95
GAS VOLUME, lb-moles/10^6 Btu	31.74	31.77	33.80
GAS WEIGHT, lb/10^6 Btu	878	938	986
ASH, lb/10^6 Btu	0	.77	.90
SO_2, lb/10^6 Btu	0	.91	.78

*Combustion at 15 percent excess air.

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.

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 exchange will occur and will provide the required final temperature of the working fluid. 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 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 1-1, differences in gas weight, volume and composition produced by CWM or DUC 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 basic technical approach for integrating all combustor system components and formulating conceptual designs.

In the following Sections, a detailed account will be made of the technical work completed under this program in fulfillment of the design objectives. In Section 2.0, Technical Discussion, the design and fabrication of the pulse combustion apparatus is described. This includes a discussion of general design considerations for the development of residential coal combustors, scale-down issues, and an overview of the exploratory design evolution which was undertaken in this investigation. In Section 3.0, a description of the test facility is given. This includes a description of the physical plant and test stand, ancillary systems, instrumentation, and data acquisition. In Section 4.0, details of the development tests are given, including test procedures, operating history, test results, and data analysis. In Section 5.0, the optimized residential configuration is defined, and in Section 6.0, conclusions based on the experimental data and design are provided.

SECTION 2.0
TECHNICAL DISCUSSION

The overall objective of the Phase I test program involves the systematic development of coal-fueled combustion systems for residential space heating applications. The results of this work are to be used in defining an optimized residential configuration which is to be tested in an integrated Phase II activity.

In order to achieve the primary objectives of this program, technical activities in Phase I have been divided into several major elements as follows:

- o Design of 100,000 Btu/hr pulse combustor
- o Fabrication of pulse combustor
- o Development testing of the pulse combustor
- o Evaluation of Data and Selection of Optimum Configuration for Integrated Testing

2.1 DESIGN AND FABRICATION

In the following Subsections, an overview is given pertaining to the design, fabrication, and preliminary testing of pulse combustion systems investigated during this work. Discussion is directed toward establishing the evolution of system concepts which arose as a direct result of experimental testing, rather than detailing specific test results. A more detailed description of test results will be made in later sections.

The objective of this work was to develop coal-fueled combustion systems possessing specific design, operation, and performance attributes suitable for residential space heating applications. Therefore, discussion is initiated with a review of general design considerations relevant to the achievement of these objectives.

It will become evident in this section that numerous embodiments of the pulse combustion apparatus were tested prior to arriving at an optimized residential system design. Among the basic pulse combustors tested in this work included: 1) Lockwood/SNECMA, 2) modified Grand Forks, 3) modified Hanby, 4) advanced Quadratic Form Chambers, 5) bare metal conical chamber, and 6) refractory conical chamber. In addition, numerous configurational variants were investigated including: 1) phased coal injection, 2) jet pump induced second stage air injection, 3) tandem operation with both aerodynamic valve and tailpipe coupling, (4) fuel biasing, and (5) dry and wet particulate collection.

Since work on the residential combustor was closely coordinated with MTCI's development of an industrial retrofit application, design evolution relevant to both of these programs is shared in the following discussion.

2.1.1 General Design Considerations

The 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 entirely vanished since the introduction of natural gas, fuel oil, and direct electric space heating systems. In the past, coal-fired heating furnaces emitted undesirable quantities of smoke and ash, and they required daily attention for feeding of the coal and removal of the ash. The replacement of coal-fueled heaters with natural gas, fuel oil, and direct electric heaters is largely attributed to the low maintenance and emission characteristics offered by these latter systems. Therefore, if coal-fueled space heating is to compete in the current residential market, means must be devised to improve the logistical, operational, and performance characteristics for these systems.

Practical means for achieving these objectives will be available in the near-term with the advent of highly beneficiated coal fuels such as dry, ultrafine coal (DUC) that is easily conveyed in small-scale pneumatic apparatus, or coal water mixtures (CWM) that are easily stored and pumped in a manner similar to distillate fuels.

Although the availability of these new fuels will greatly facilitate the introduction of coal-fueled, residential, space heaters, additional design issues must still be addressed regarding the desired physical, mechanical, and thermal attributes of the integrated system. Implementation must therefore follow a carefully planned design development activity which encompasses all primary and ancillary system functions including fuel transfer, combustion, heat transfer, and particulate removal.

In the following paragraphs, a more detailed discussion of design considerations relevant to coal-fueled, residential, space heating systems is given. These design considerations have formed the basis for guiding the logical progression of development activities undertaken in this investigation.

A. Fuel Selection

Fuel selection is an issue which may have a profound influence on the ultimate marketability of coal-fueled, residential, space heating systems. Two primary fuels have been identified for this application; dry, ultrafine coal (DUC), and coal water mixtures (CWM). Each of these fuels must be delivered in a deeply beneficiated form in order to meet the operational and environmental targets for residential units (less than .8 lb/MMBtu mineral matter, and less than .5 lb/MMBtu sulfur).

Although the final selection between DUC and CWM may be dictated by economic and availability issues specific to a given geographic region, these two fuels can be distinguished by several important properties impacting fuel transport, storage, transfer, and combustion.

Most of the deep benefication processes currently under development involve aqueous processes. Therefore, the cleaned coal product is typically available in a wet form. Thus, in order to prepare DUC, the wet coal product must first be dried. In contrast, CWM fuels may be formulated in a more direct manner.

Drying of micronized clean coal is a relatively capital intensive process and consumes approximately 4-5 percent of the coal's heating value. Safety issues regarding the handling of highly ignitable DUC require resolution. However, the costs of transporting DUC to the end-user may be lower than that for CWM due to its higher energy density (15,000 Btu/lb versus 9,750 Btu/lb).

Off-loading and storage of CWM at the end-user site is generally thought to be simpler than for DUC. CWM is not prone to dusting, and can be easily pumped from a transport tanker directly to the storage tank much in the fashion of fuel oil. In contrast, DUC must be carefully conveyed to a storage bin blanketed with an inert gas.

Although storage of CWM in existing fuel oil tanks is an attractive proposition, several deficiencies in this approach are immediately apparent. Initial market entry for coal-fueled systems is likely to occur in the least temperate regions of the United States where high operating capacity factors more fully exploit the cost differential between coal and premium fuels. Therefore, CWM stored in unheated tanks is prone to the risk of freezing resulting in defluidization. This poses an extreme reliability issue since operating CWM handling equipment depends entirely on the maintenance of proper fluidity. Futhermore, aging, degradation, slime growth, and corrosion are recognized as major barriers to widespread residential usage of CWM. Although additives are being developed to better stabilize CWM, the cost of these additives are generally prohibitive. In contrast, DUC is much less sensitive to environmental factors. Consequently, transfer equipment designed to handle DUC may ultimately be more reliable than that which is proposed for CWM.

An additional thermal penalty results from CWM combustion at the end-user site due to the heat required for vaporizing the inherent water content of the CWM. Although this heat can theoretically be partially recovered using a condensing heat exchanger, condensation of an ash-laden flue gas stream in a heat exchange coil may be prone to fouling problems and may therefore be incompatible with residential end-use requirements. Thus, DUC offers the advantage of improved thermal efficiency compared to CWM. Furthermore, the combustor volumetric heat release rate is anticipated to be lower for CWM than for DUC necessitating more bulky combustor designs. The combustion performance of CWM is expected to be strongly related to the efficiency of atomizing the CWM in the combustor. Since atomizers are prone to plug, the long-term combustion stability of CWM in small-scale residential equipment may be questionable.

Furthermore, although no significant commercial experience exists to date in the United States for either fuel, a modest degree of commercial experience has been obtained in Europe on DUC.

Based on the considerations discussed above, it was determined that the properties of DUC appear at present to be more amenable to residential applications compared to CWM fuels. For this reason, developmental testing during this investigation focused on the use of DUC fuels.

B. Coal Feed and Injection System

Reliable and economic means must be devised for metering and conveying coal fuels from the storage bin to the heater in preparation for injection into the furnace combustion chamber. Several options are available for metering DUC fuels including: 1) screw feeders, 2) belt feeders, 3) disk feeders, and 4) fluidized bed feeders. Each of these feed systems are used to assist bin unloading and to provide a uniform coal feed to the conveyance system.

Although several different mechanical conveyance systems have found broad application in the solids handling industry (eg. belt, bucket, and screw conveyers), pneumatic conveyors appear to offer the only practical conveyance method for residential space heating applications. This is due to the lower maintenance requirements for the pneumatic systems and the difficulty of scaling mechanical systems to the size range of interest for residential applications.

A primary disadvantage of pneumatic conveyors involves the need for compressed carrier air. however, dense phase pneumatic conveyance of coal can be achieved with solids loadings as high as 10 pounds of coal per pound of carrier air. Since this represents less than one percent of the stoichiometric combustion air, it is apparent that the air supply compressor can be quite modest in size and thus is not anticipated to significantly impact the economics of the process.

As previously mentioned, several options currently exist for metering DUC fuels to the pneumatic conveyer. However, few, if any, of these systems have been developed specifically for the small-scale user. For this reason, off-the-shelf feeders are typically quite expensive, and generally offer functional options which exceed the needs of the residential end-user.

As in the case of a conventional, gas-fired, residential space heater, the coal-fueled unit is anticipated to operate in an on-off or modulated (high-low) load control mode. Therefore, sophisticated motor control panels, PID or sequential logic, variable speed control, etc., are largely unwarranted for the residential market.

Given the limited functional requirements of the coal metering system, it is expected that mass produced feeders can be inexpensively fabricated for residential DUC-field applications. Among the metering concepts mentioned, the disk feeder appears to offer perhaps the simplest mechanical approach.

However, since it was beyond the scope of the current investigation to develop novel subcomponent feed systems, an off-the-shelf variety, single screw feeder was employed in all experimental work. The screw feeder was used to meter coal from a storage bin to the pneumatic conveyance system. The pneumatic conveyor consisted of a receiving funnel connected to an air eductor. The DUC-laden carrier air is then transported directly to the coal feed injector.

C. Combustor

The combustor represents the heart of the coal-fueled, residential space heater. Since space limitations are a primary concern for residential applications, efficient coal combustion must be achieved at very high volumetric heat release rates. To achieve these high volumetric heat release rates for coal combustion, a highly turbulent combustor environment is a necessity. This objective is accomplished through the use of the pulse combustion apparatus investigated in this work.

Several different types of pulse combustors are available including the Helmholtz, Schmidt, and Rijke tubes. The Helmoltz type combustor was selected for this investigation due to its superior combustion performance relative to the other types. This superior combustion performance is attributed to the highly resonant nature of the Helmholtz configuration which tends to yield the highest pressure fluctuations per Btu/hr of firing. The resulting high level of turbulence with stable oscillations improves combustion efficiency and provides a level of pressure boost which is useful in overcoming pressure drop in the heat exchange and ash removal subsystems. Two types of air inlet valves can be used in pulse combustion; a mechanical flapper valve or an aerodynamic valve. Although mechanical valves provide somewhat higher boost pressures, the reliability of these valves is anticipated to be low, particularly in coal-fueled applications. The combustion of coal fuels is likely to result in ash deposits which will deteriorate valve seatings in mechanical systems. Erosion and corrosion further limits the application of mechanical valves. Aerodynamic valves therefore appear to be the most suitable for the residential application which require high reliability and low maintenance. Therefore, pulse combustion development work undertaken in this investigation considered the use of aerodynamic valves exclusively.

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. This attribute is anticipated to be particularly important for the development of simple, reliable and cost-effective residential systems. 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.

At the low firing rate, the heat release rate in Btu/ft^3-hr is lower due to the lower oscillating flow field in 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 for higher firing rates is compensated for by an 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 the 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. A combustor that normally operates with high excess air would, by virtue of employing a valve with high fluidic diodicity, operate at lower air stoichiometry by throttling the air intake at the 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 start-up characteristics of the combustor. With this design option, the turndown ration could be greater than 8:1. This however, may require an inlet air fan if the pressure drop for ash removal 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. Second, 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 effect 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 transition 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 dampened downstream of the flame, the region in which 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 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 pulse combustion system may be operated in two different primary modes: slagging and non-slagging. For slagging operation, the bulk of the coal ash is rejected from the combustor in a fluid state. For non-slagging operation, mineral matter is rejected as dry fly-ash. In order for slagging to occur, combustor temperatures must typically exceed 2600°F. The precise temperature for the onset of slagging operation is dictated by the ash characteristics specific to a particular coal feed. In contrast, slagging can be safely avoided if combustor operating temperatures are maintained below 2200°F to 2400°F.

Although ash rejection through slagging may be beneficial in certain industrial applications, slag rejection does not appear to offer a viable means of operation for the residential market. Firstly, hot slag represents a severe safety issue in residential applications. Secondly, the collection, storage, and removal of slag effluent is considerably less manageable than for fly ash. Finally, the materials of construction necessary to contain and handle the corrosive slag effluent is likely to negatively impact the overall system economics. In contrast, the rejection of fly ash does not appear to introduce significant logistic or safety issues. For this reason, this investigation has focused on the development of a dry, non-slagging combustor.

It is therefore evident that a careful balance must be made in order to achieve combustor temperatures which are high enough to ensure that an acceptable level of carbon-burnout occurs, but low enough to avoid slagging. Thus, both bare metal and refractory-lined combustors were tested in this investigation to explore the optimum operating temperature.

D. Particulate Collection

Collection of particulate fly-ash emitted from the combustion system can be accomplished using a variety of different apparatus including: 1) bag houses, 2) dry cyclones, and 3) wet scrubbers or cyclones.

The average coal fuel particle size envisioned for residential applications is less than 10 microns. Fly-ash resulting from the combustion of these fuels may include a significant portion of particles less than

1 micron. Due to the extremely small particle sizes of fly-ash generated from micronized fuels, dry cyclone systems are incapable of achieving the particle collection efficiencies necessary for residential use. Alternate dry collection means such as a bag house require frequent maintenance, are prone to plugging causing excessive pressure drops, and are likely to result in unacceptable dust emission levels, particularly during servicing and change out.

Futhermore, collection of fly-ash in a dry form necessitates storage in a container drums which must be periodically removed. For instance, a residential unit is anticipated to generated an annual quantity of approximately 200 lbs of particulate emissions.

In contrast, collection of fly-ash through wet methods may avoid many of the shortcomings inherent in the dry methods, and may allow continuous discharge of the collected solids into storm sewers, sanitary sewers, or septic tanks thereby minimizing maintenance requirements.

In the wet methods, a modest water spray is injected into the combustor exhaust. The atomized water droplets serve to agglomerate fly-ash particles into a more manageable form. The water effluent which is partially loaded with solids may then be discharged easily in a fluid stream. Since fly-ash is presently classified as a non-toxic substance, discharge in this manner may be a viable option for low-maintenance ash rejection in early systems.

For these reasons, particulate collection systems employed in this investigation have focused on wet systems.

2.1.2 Scale-Down Issues

Although small pulse combustors (100,000 Btu/hr) have been constructed for natural gas and oil fuels, none have been built specifically for coal fuels prior to this investigation. Work was previously performed by MTCI to develop pulse combustors which are capable of firing coal fuels in the range of 1 to 2 MMBtu/hr. Since this prior work formed the basis for the development of smaller residential units, design issues relevant to the scale-down of pulse combustors are discussed here.

In scaling down from 1 to 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:

$$F = c/2 \quad L_t \quad V_t/V_c \qquad \text{(Helmholtz)} \qquad (I)$$

Where: f = Frequency in Hz

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

L_t = Length of tailpipe (or resonance tube)

V_t = Volume of tailpipe

V_c = Volume of combustion chamber

The ratio V_t/V_c will not be varied 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 maximum frequency of operation must be compatible with the rate of heat release for a given fuel. For instance, dry ultrafine coal which offers an inherently high rate of heat release allows operation at higher combustor frequencies than standard grind pulverized coals or CWM. Thus there is a limit on the increase in frequency since the burn rate must be sufficiently high to "keep up" with the combustor cycle timing in order to effectively excite the combustion-induced oscillations. The scaled combustor frequency can be modified, however, by modifying the V_t/V_c ratio. This is accomplished by either increasing the combustion chamber volume or reducing the tailpipe volume or both.

Increasing the combustion chamber volume requires some additional considerations. This includes both chamber geometry modification and other combustor tuning constraints for stable operation and good turn-down. Increasing the chamber volume while retaining a cylindrical chamber shape and increasing chamber diameter will only 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 combustor transition region and, therefore, the mixing of fuel and air. The chamber geometry must be selected to accommodate both the volume increase and the required transition between the aerodynamic valve and the chamber. Tuning considerations, 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:

$$f = c/4(L_c + L_t) \qquad \text{(Quarter wave)} \qquad (II)$$

Where: f = Combustor frequency (Hz)

c = Speed of sound

L_t = Length of tailpipe

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

Thus, for proper combustor tuning, both equations I and II must be satisfied. This, in turn, imposes some constraints on chamber length and shape. Quadratic from generators can be employed to address these requirements while maintaining appropriate flow patterns in the chamber, in the transition from the aerodynamic valve to the chamber and from the chamber to the tailpipe. The transition region between the chamber and the tailpipe is responsible for shedding vortices on return of hot gases from the tailpipe during flow oscillations. These vortices of hot combustion products, together with the hot chamber wall temperature, are responsible for auto-ignition of the next combustion cycle.

In the case of the tailpipe, reduction in the tailpipe volume (to reduce the combustor frequency) can be affected by reducing the tailpipe diameter. This is necessary since reduction in the tailpipe length would disrupt the combustor tuning. This can be gleaned from examining both equations I and II above. The combustor frequency in equation I (Helmholtz) is proportional to $1/L_t$ while, in equation II (quarter wave) it is proportional to:

$$1/(L_t + L_c) \text{ or } 1/L_t \quad (\text{for } L_c \ll Lt) \tag{III}$$

There is, however, a limit to the reduction of the tailpipe diameter. For a small-scale size combustor, the pressure drop in a tailpipe 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 oscillations and impairs the combustor performance (burn rate, carbon conversion, pressure boost, etc.).

In addition to the potential for increased combustor damping, another constraint on the reduction of the resonance tube diameter is the limitation on maximum flow velocities in the tailpipe. Clearly, the maximum flow Mach number must be maintained comfortably less than 1.0 at the gas temperature 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/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 CWM fluids.

2.2 FABRICATION CONSIDERATION

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.

This approach, however, constrains the configurations and the combustor designs that can be explored since bare metal standard shapes are the building blocks. 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:

i In pulse combustion, the acoustic, fluidic and aerodynamic characteristics of the combustor configuration dictate performance. Therefore, constraints on design configurations are unacceptable.

ii The delivery times and costs for more complex off-the-shelf parts were not significantly less than for special-order items. Despite the fact that some suppliers list certain parts as catalog items, less standard parts are insufficiently inventoried. Thus, these items are treated in much the same manner as a specialty fabrication order, reflecting in both delivery time and cost.

iii Mass production of commercial residential units will eliminate this distinction as an area of concern.

Another approach to fabrication of flexible component and combustor configurations for the development effort 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 somewhat constrained and some limitations on material properties are important. For ease of fabrication, the material must be relatively thin and must have good ductility so it will flow properly during the forming process. If thick wall construction is desired, or if less ductile materials such as Inconel are to be used, then, 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.

Mandrels for metal spinning can be assembled in a composite mandrel for use in casting refractory 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 commercial manufacturing.

2.3 CHAMBER DESIGNS

2.3.1 Lockwood/SNECMA Design

In conjunction with prior CWM combustion studies, MTCI built combustors of the Lockwood/SNECMA design at the 1-2 MMBtu/hr scale. In order to build upon the data base generated in this prior work, an initial test unit of a similar design was built at the 100,000 Btu/hr scale. This approach allowed investigation of scale-down effects as an isolated variable.

In parallel with this effort, new chamber configurations were being developed at the larger scale. This allowed investigation of chamber modifications as a primary variable. Chamber configurations which performed in a manner suitable for residential applications could then be introduced at the 100,000 Btu/hr scale for further evaluation.

The scale-down Lockwood/SNECMA design is shown in Figure 2-1, and incorporates the following features.

- Tubular aerodynamic valves having a constant ID with flared inlet section which provides for an 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.

- Cylindrical Combustion Chamber with two conic (45^0) 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.

- A continuously tapered resonance tube which is formed in a U shape for compactness. The slight taper in the tailpipe biases forward flow in the tailpipe and enhances the combustion air recharge performance of the combustor.

Preliminary evaluation tests were conducted on the scale-down Lockwood/ SMECMA pulse combustor using natural gas, #2 fuel oil, and micronized coal. A maximum firing rate of 180,000 Btu/hr was achieved at a combustor frequency of 180 Hz.

Although preliminary tests indicated that micronized coal combustion was feasible in the scale-down Lockwood/SNECMA design, combustion stability was relatively poor. It was also apparent from qualitative observation that carbon-burnout was not as complete as desired. Although it is believed that

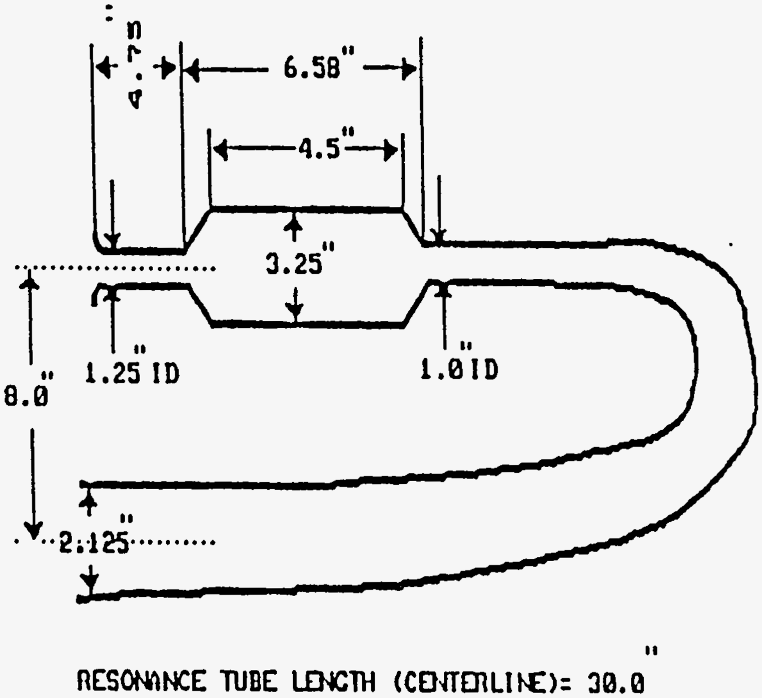


FIGURE 2-1:

SMALL PULSE COMBUSTOR UNIT
(WITH PERTINENT DIMENSIONS)

periodicity of the screw feeder negatively impacted the combustion performance, it was determined that alternative designs were necessary to achieve the target performance objectives for the residential unit. Therefore, additional development testing of alternative designs at the 1-2 MMBtu/hr scale was undertaken to arrive at a more suitable configuration.

2.3.2 Modified Grand Forks and Hanby Designs

Exploratory testing at the 1-1.5 MMBtu/hr firing range was undertaken using the modified Grand Forks and Hanby designs. These tests were intended to provide data and test experience which would be useful in developing an optimized advanced design for the residential unit.

During these tests, several different design concepts were investigated in order to monitor their impact on overall combustor performance. These concepts include jet pump induced second stage air and phased coal injection. A brief account of this work follows here.

A. Modified Grand Forks Design

A modified Grand Forks chamber design was fabricated at the 1-2 MMBtu/hr scale. The combustor design is shown in Figure 2-2.

In this design, the aerodynamic valve is orthogonal to the chamber/ tailpipe axis. The back end of the chamber is closed and the return flow must make a 90° turn to exit at the aerodynamic valve. This design benefits from high peak pressure fluctuations, per Btu/hr of fuel firing, in the combustion chamber.

The unit was also designed with a telescoping tailpipe and aerodynamic valve to allow improved tuning of the unit. In addition, the unit was outfitted with a phased coal injector consisting of a feed chamber with a volume of approximately one-tenth that of the combustor. The phased coal injector is shown in Figure 2-3.

The purpose of the phased injector is to feed coal only during the appropriate point in the combustion cycle. As a secondary benefit, the phased injector serves to condition and preheat the coal prior to injection.

As shown in Figure 2-3 the phased injection was coupled to the chamber through a 1" port located behind the aerodynamic valve. The resistance of the 1" port together with the small volume (capacitance) of the feed chamber did not noticeably affect the resonance of the Helmholtz combustor. Nevertheless, during the combustor exhaust stroke, hot gases from the combustion chamber rushed into the feed chamber and preheated the coal under substoichiometric air conditions. During the air intake stroke, the combustion chamber pressure drops causing the flow to reverse between the feed chamber and the combustion chamber. This provided for a phased coal injector which "inhales" coal with the feed air simultaneously during the intake portion of the combustor cycle.

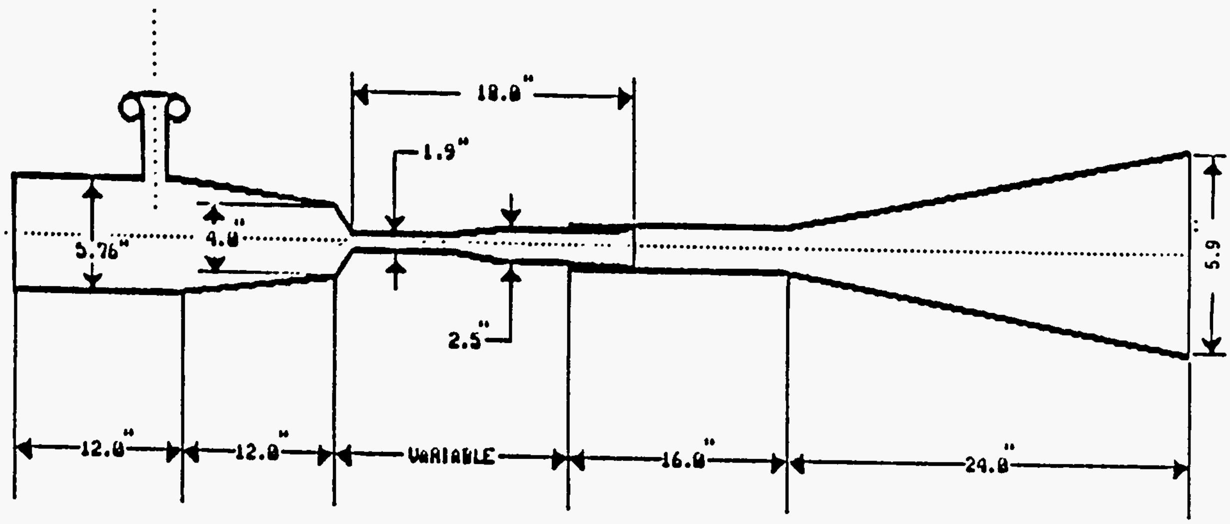


FIGURE 2-2:

MODIFIED GRAND FORKS DESIGN

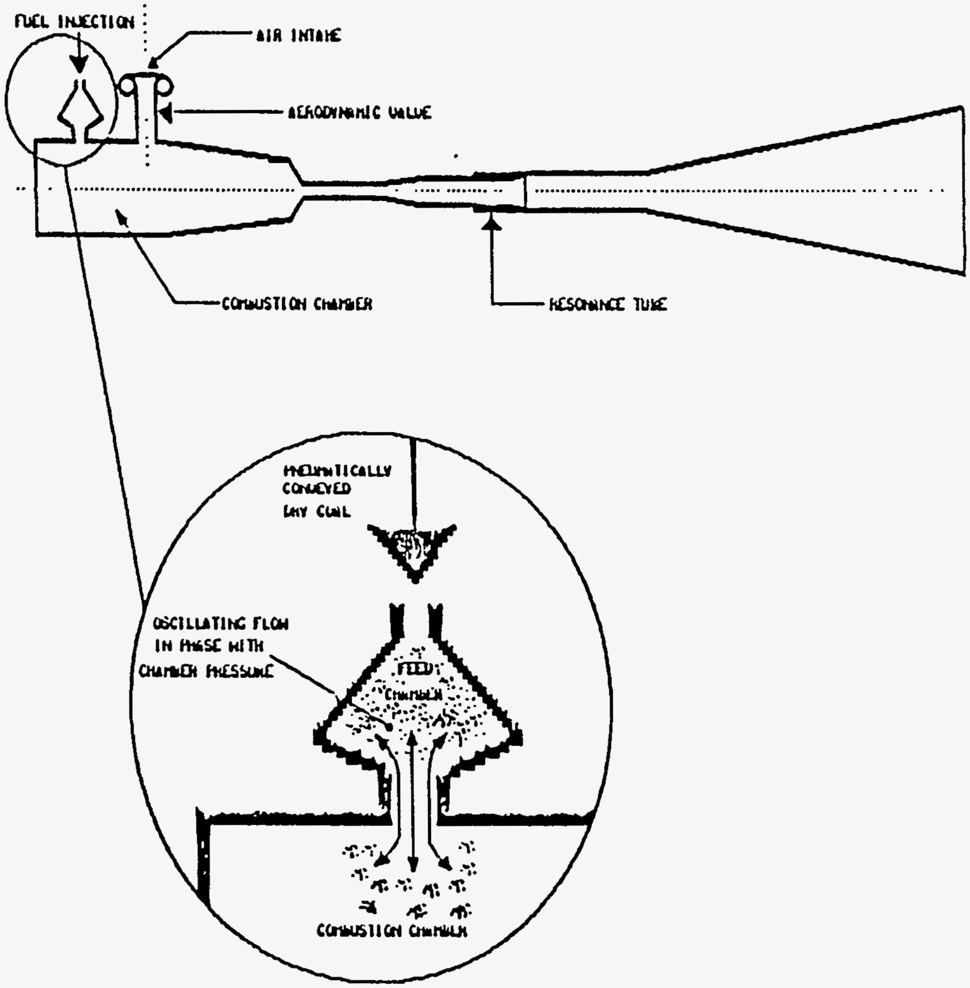


FIGURE 2-3:

PHASED COAL INJECTOR

Under certain test conditions, the phase injector appeared to intensify the pressure oscillations per Btu of coal fired. Although the relative importance of fuel preheat resulting from this method was not precisely known, it did appear evident that some form of fuel phasing is necessary for improving overall combustion performance.

B. Modified Hanby Design

Acquisition of further test experience was then gained using a modified Hanby pulse combustor with an expanding tailpipe as shown in Figure 2-4.

During the initial test period, the modified Hanby was operated on natural gas in order to tune the system, and a number of aerodynamic valve inserts were used to modify both the valve's fluidic diodicity and minimum flow resistance. In addition, the tailpipe 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/tailpipe interface during the initial test period. The failure was repaired before additional tests were made.

In subsequent test runs, a modified configuration was used to provide for tailpipe cooling by employing the pulsating flow field at the aerodynamic valve and the kinetic energy in the flow at the resonance tube exit as shown in Figure 2-5. In this configuration, air is induced, by the pulsating flow field at the aerodynamic valve, into both the combustion chamber (first-stage air) and the 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. Although air staging may be important for industrial systems, it is considered to be less important for residential systems operating at lower temperatures. Therefore, this configuration was no longer pursued at the smaller scale.

Preliminary testing of the modified Hanby on coal feeds was then undertaken. In this modified Hanby configuration, auto-aspiration of air was achieved with a 1.875" diameter venturi as the flow diode (Figure 2-6) in the aerodynamic valve, and the tailpipe length was varied between 63" to 70". When coal feed was started, the chamber temperature climbed from about 1978^{o}F to over 2400^{o}F (which was the limit of the Type K thermocouple that was used for measurement). The frequency was in the vicinity of 67 to 70 Hz. The small variation of frequency resulted from variations in the gas temperature in the combustor. The coal feed was a micronized Pittsburgh #8 with a nominal particle size of 10 microns.

Coal combustion increased the pressure boost and the dynamic pressure oscillation in the combustor indicating sufficient heat release in phase with the dynamic pressure oscillation in the modified Hanby combustion chamber. Coal feed was implemented using an eductor system (Figure 2-7) with a screw feeder metering the coal into the eductor system funnel. The coal flow

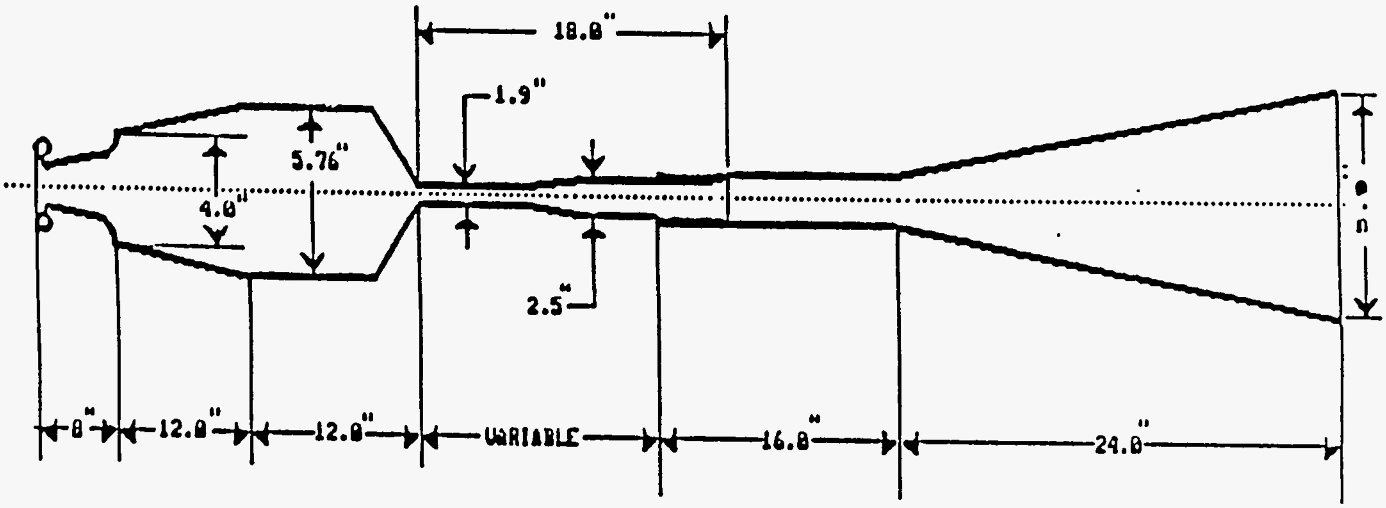


FIGURE 2-4:

MODIFIED HANBY DESIGN

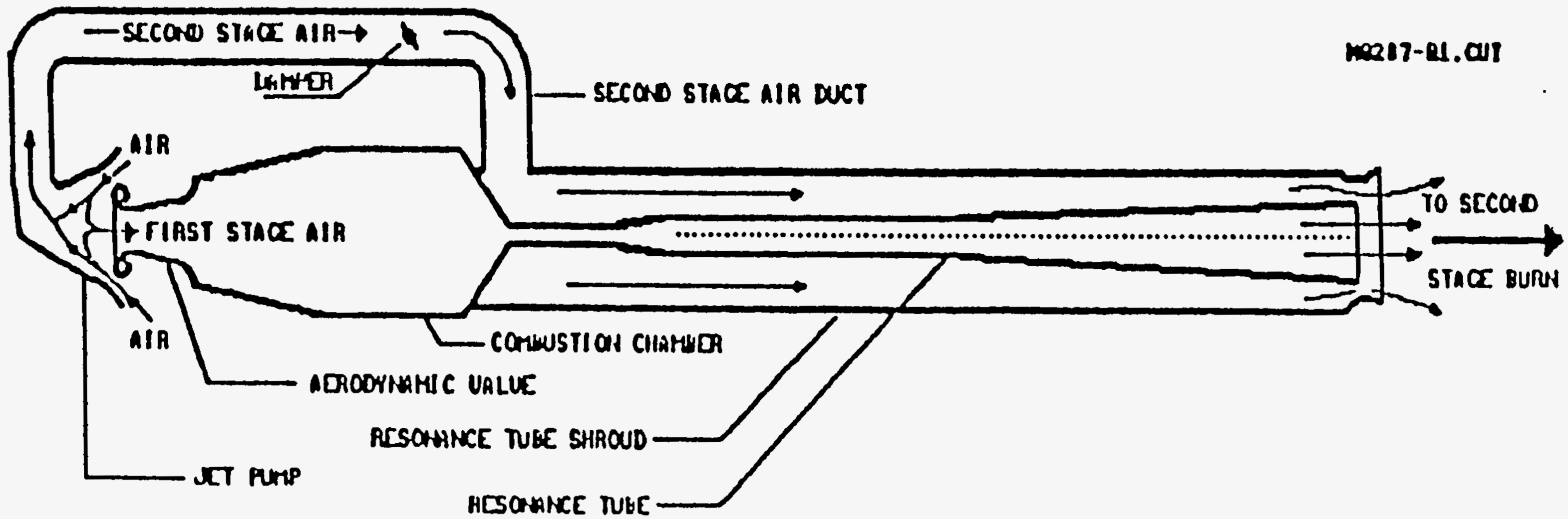


FIGURE 2-5:

RESONANCE TUBE COOLING CONFIGURATION

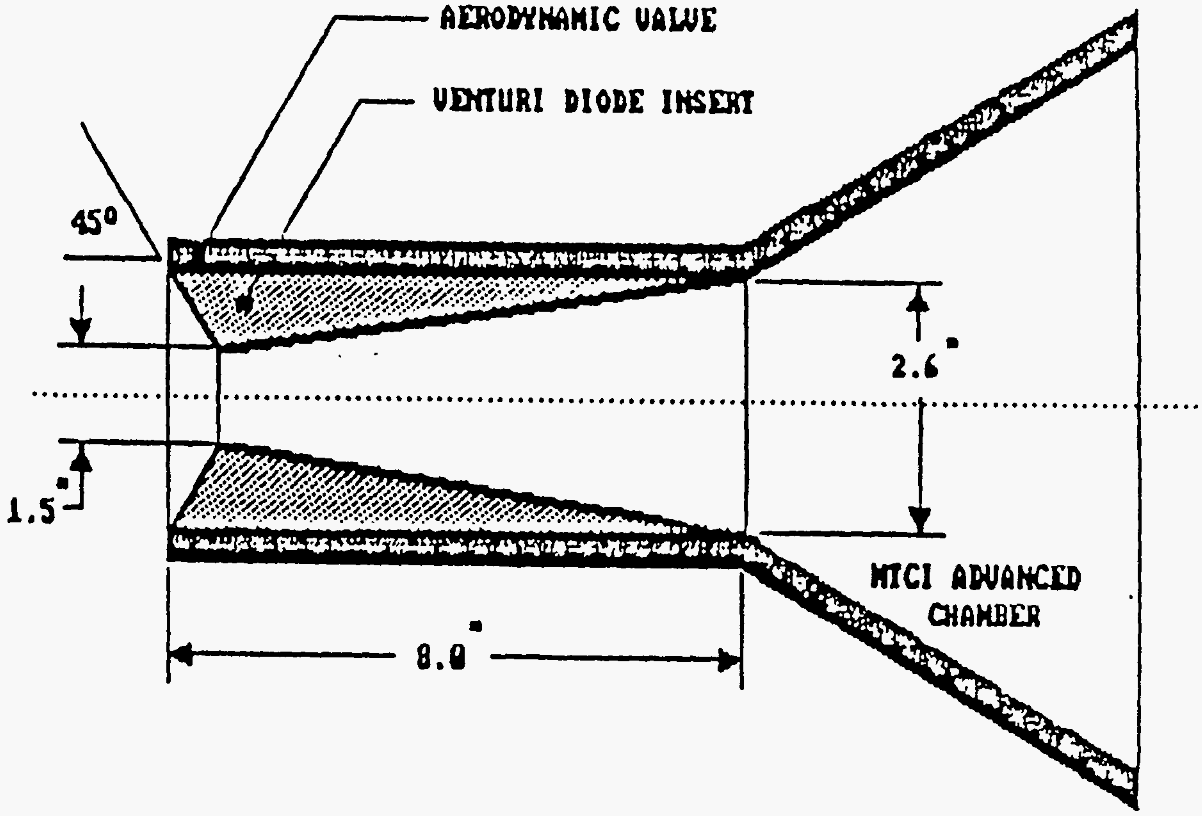


FIGURE 2-6:

VENTURI DIODE INSERT

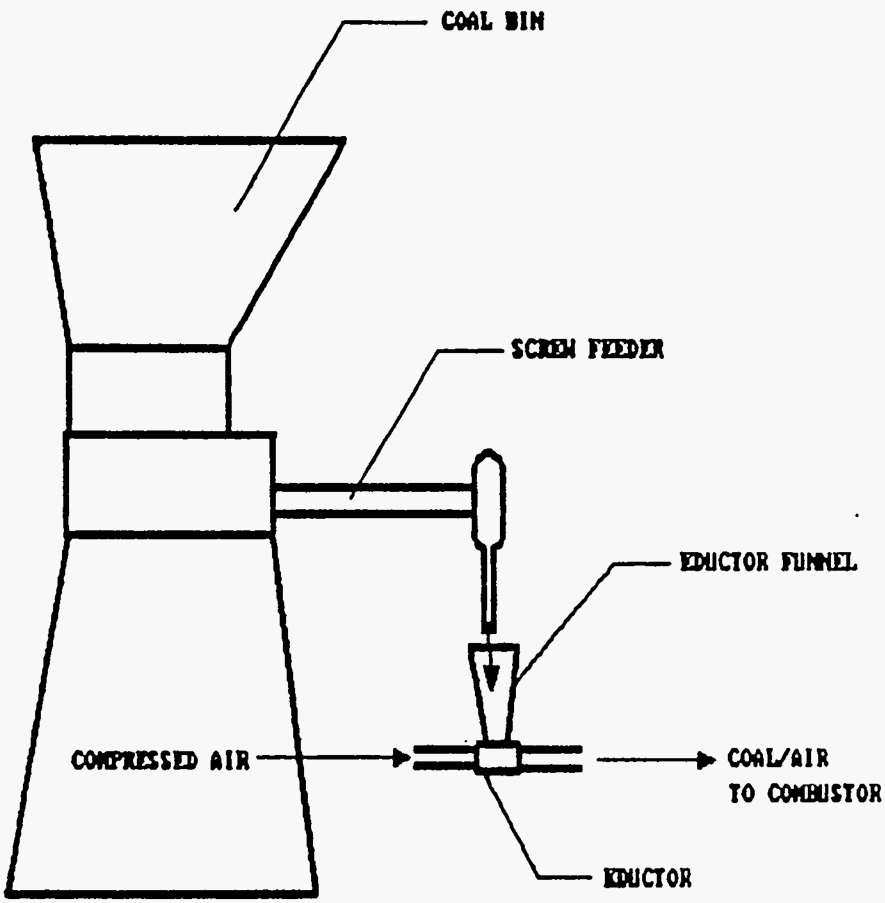


FIGURE 2-7:

COAL FEED SYSTEM

metered into the eductor funnel by the screw feeder possessed periodic changes in flow rate. This was reflected in the variation of the dynamic pressure amplitude. The frequency of the coal feed variation was in the order of two cycles per second; therefore, it did not couple with the resonant operating frequency of the combustor. This test condition, however, provided excellent verification of the combustor's load following capability. The heat release rate, as evident by the variation of the dynamic pressure amplitude variation and the combustor's sound pressure level, responded faithfully to the periodicity in the coal feed. The latter was due to the characteristics of the screw feeder used to feed the coal into the eductor funnel.

The Grand Forks and Hanby pulse combustor designs were primarily developed for combustion of gas and oil fuels. Although experimental results confirmed the feasibility of combusting coal fuels in these units, it was determined that the general performance levels exhibited by these units were insufficient to merit scale-down to the residential size.

Two separate advanced chamber concepts resulted from the test experience with the Lockwood/SNECMA, modified Grand Forks, and the modified Hanby designs. These concepts included some aspects of the prior designs but were optimized specifically for operation on coal-fired fuels. A description of these advanced design concepts follows here.

2.3.3 Advanced Chamber Designs

A. Quadratic Form Chamber

The first advanced chamber design concept employs quadratic form generators to define an axisymmetric chamber accommodating a number of design and chamber performance attributes. The quadratic form generators were chosen to be simple arcs of a circle as shown in Figure 2-8.

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 2-9 annd 2-10, respectively.

During air intake, the chamber volume is subdivided into three regions as shown in Figure 2-9. 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

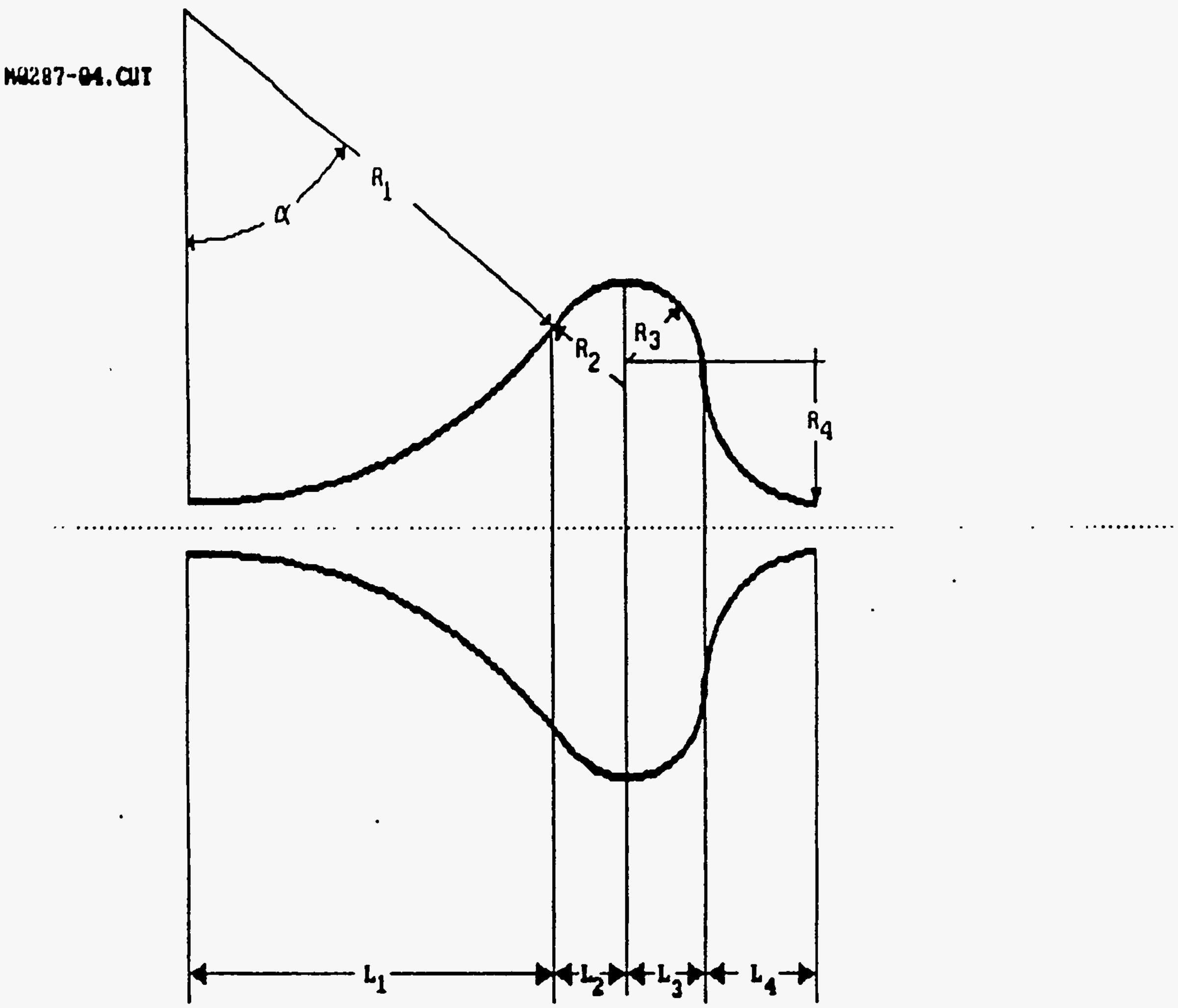


FIGURE 2-8:

MTCI ADVANCED CHAMBER CONFIGURATION

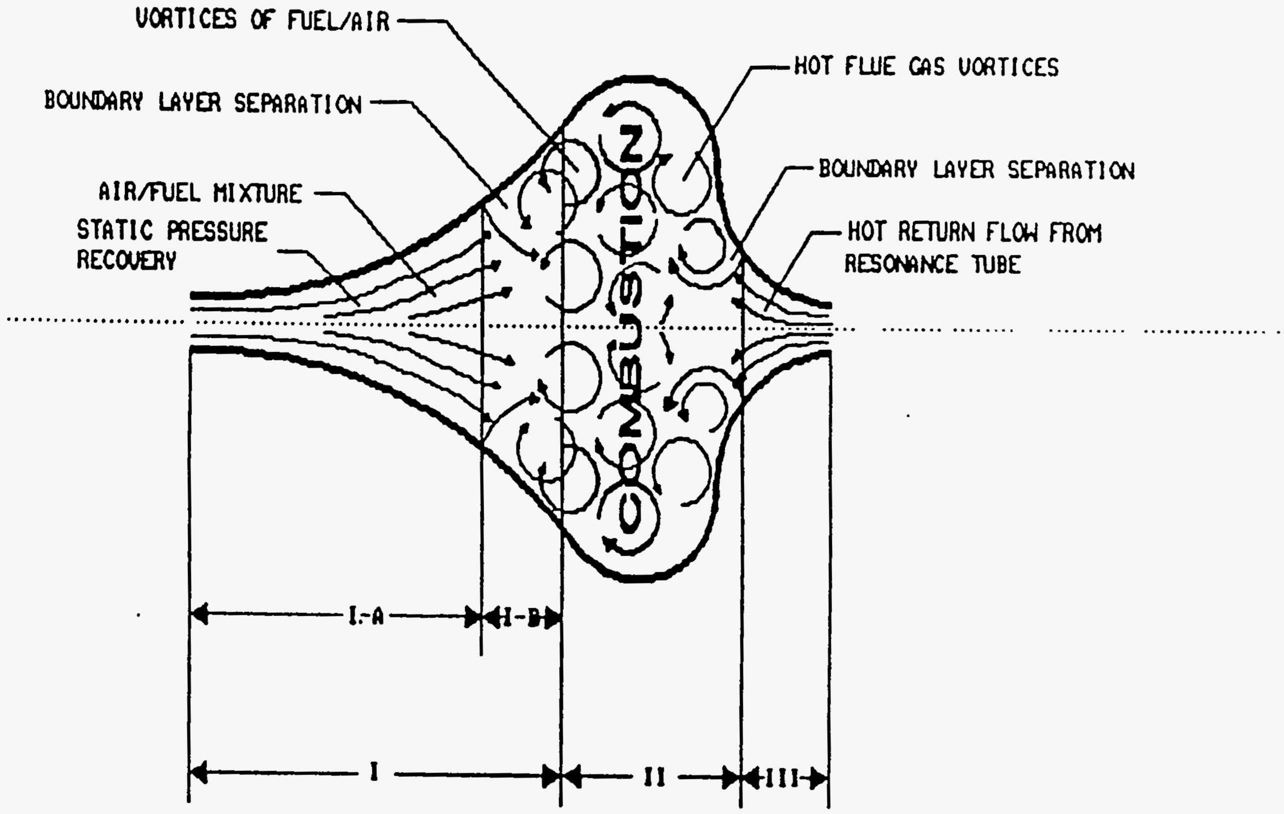


FIGURE 2-9:

FLOW CHARACTERISTICS DURING AIR INTAKE

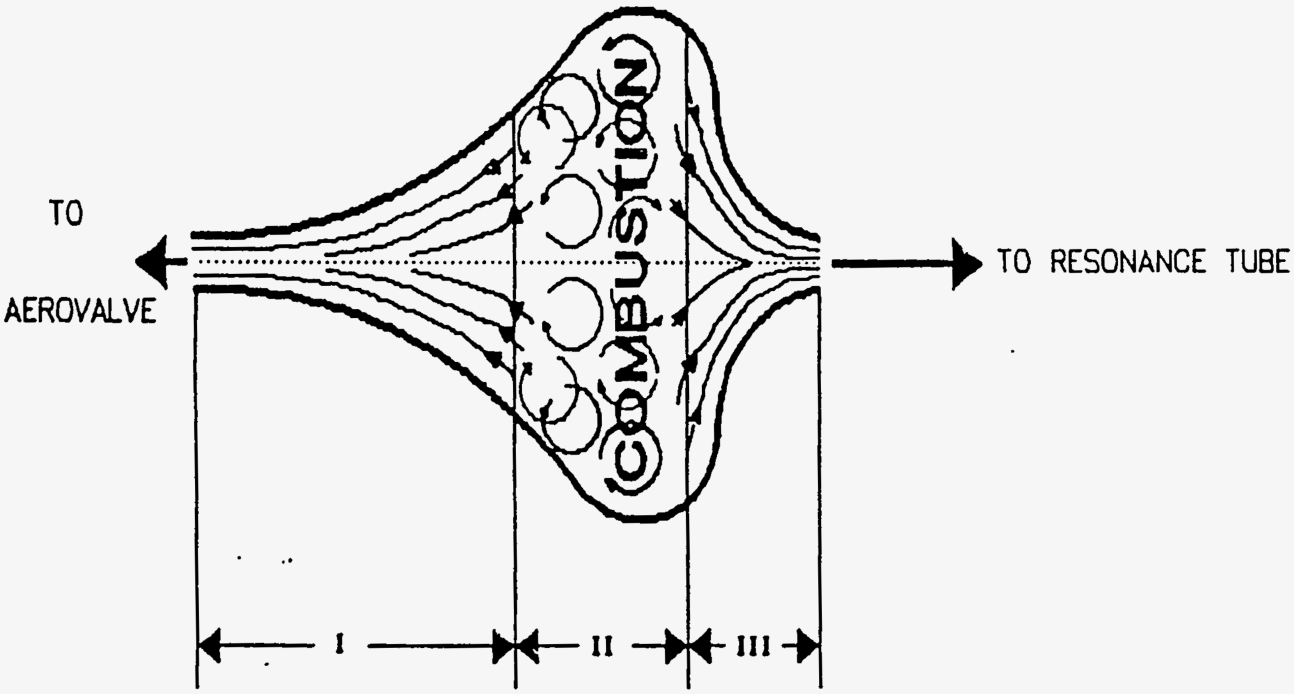


FIGURE 2-10:

FLOW CHARACTERISTICS DURING CHAMBER EXHAUST

becomes sufficiently large to cause flow separation and, in turn, gives rise to vortex shedding 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 behave 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 2-10.

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 form the chamber walls and the combustion zone (depicted as region II in Figure 2-9). This allows some fuel preheat and conditioning for rapid combustion downstream. 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 itself is employed to achieve mixing instead of the use of a bluff body or a flame holder. 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 characteristics. Variations in fuel affect the fuel preheat requirements and the flame speed. The 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, 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 MTCI 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. 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 ft^3 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. 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 generator or parts to suit the application requirements. These chambers would then be used with the appropriate tailpipes 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 due to the 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 auto-ignition. 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. However, excessive amounts of resonance tube return flow hinders the chamber's ability to recharge with combustion air and the gases in the combustion chamber.

To discuss the chamber design attributes during chamber exhaust, we now focus on Figure 2-10, which 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 montonically 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.

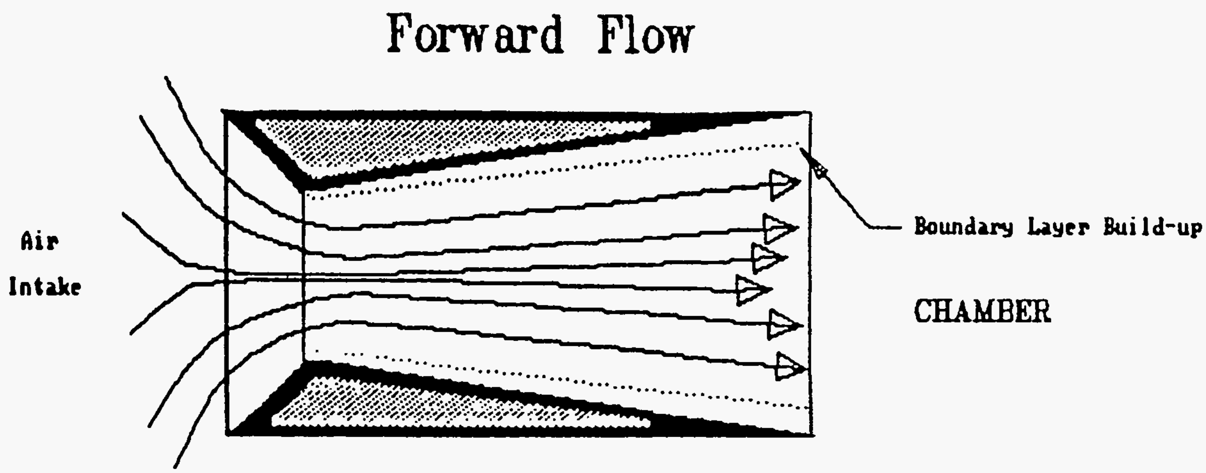
In region III, the hot gases are made to accelerate very rapidly due to the rapid reduction in the chamber's cross-sectional area in the direction of the flow. The diffuser length is very short and the flow acceleration is maintained high, thus causing the boundary layer thickness to remain small in that region reducing 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 tailpipe) 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 2-11. 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^{o} and 60^{o} (half cone angle). On the combustion chamber side, a generous shallow angle diffuser is used to provide for efficient pressure recovery (4^{o} to 7^{o}). 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 2-11, labeled Forward Flow. In both parts of Figure 2-11 the chamber is located on the right-hand side of the aerodynamic valve (not shown).

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 shown in the lower part of Figure 2-11, 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, the temperature of the air on intake is significantly lower than the temperature of hot gases from



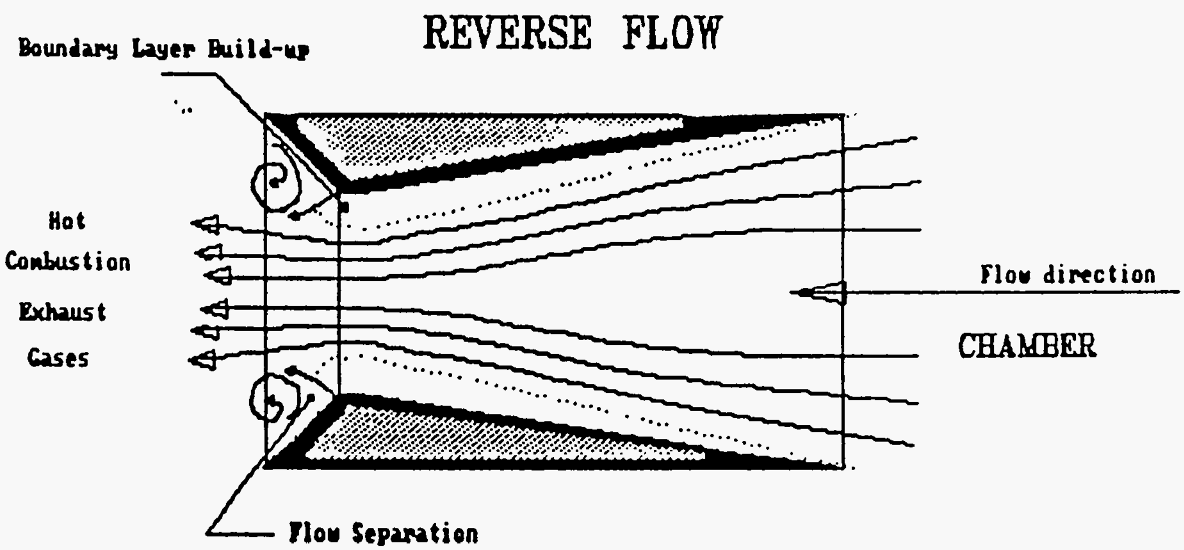


FIGURE 2-11:

MTCI DIFFUSER-BASED AERODYNAMIC VALVE

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 counterparts 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 in self-regulation of the net flow at the aerodynamic valve for intake of combustion air at a self-induced level of stoichiometry.

The flow characteristics in regions I and III of the chamber as, shown in Figures 2-9 and 2-10, are similar to those described for the aerodynamic valve shallow and steep diffuser sections, respectively. This similarity results in flow diodicity, which is in addition to that of the aerodynamic valve.

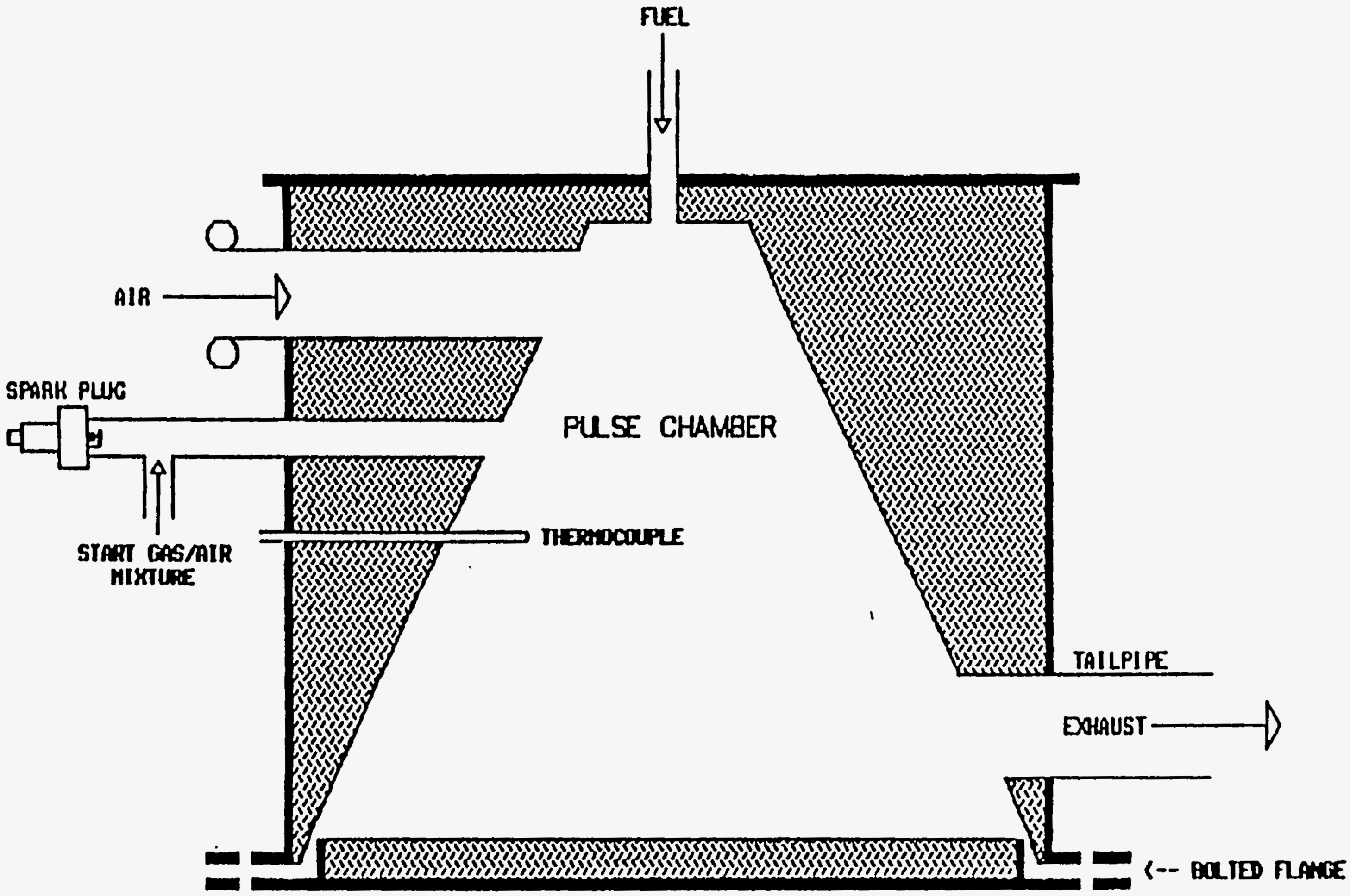
Extensive testing of the quadratic form chamber was undertaken at the 1 to 1.5 MMBtu/hr scale. This system was found to have performance attributes well suited to industrial-scale applications. A more detailed account or this work is reported under DOE Contract No. DE-AC22-87PC79654.

B. Advanced Conical Chamber

A second advanced chamber design which was considered to be more suitable for residential-scale apparatus was explored. This concept employs a simple conical design which can be economically constructed at the 100,000 Btu/hr capacity range. In this new design, the tailpipe and aerovalve were attached tangentially to the combustor. A schematic of the advanced residential chamber is shown in Figure 2-12.

The conical configuration of the advanced residential chamber was selected for several reasons. First, the residential chamber has a high surface-to-volume ratio and a small characteristic dimension. Therefore, the intensity and effective mixing length of vortices generated by the deceleration of influx gases during the suction and recompression cycle is expected to be diminished compared to larger industrial units. 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 heat up 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



(Refractory Design)

FIGURE 2-12:

RESIDENTIAL ADVANCED CHAMBER

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 dimensions, 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 the flow of combustion products 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.

Advanced conical chambers were constructed in both a bare metal form and in a refractory-lined form. The bare metal combustor typically operated with combustor temperatures of approximately 2000^{o}F. The refractory-lined combustor temperatures typically ranged from 2200 to 2300^{o}F. As expected, performance of the refractory-lined unit was significantly improved compared to the bare metal version. Extensive testing of the refractory unit on coal fuels was performed with promising results. These results are reported under Section 4.3, Development Testing and Results.

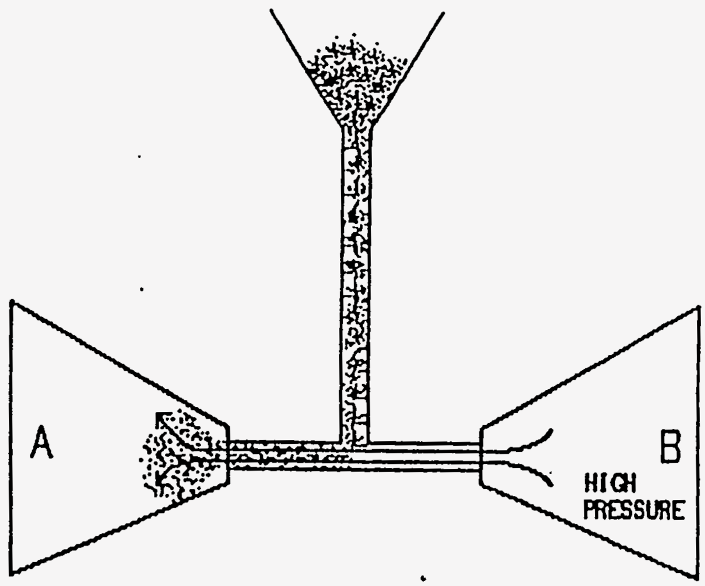
C. Tandem Units

In previous tests conducted under this program, only single pulse combustors were investigated. Since mechanical valves such as are commonly used in gas-fueled pulse combustors are impractical for coal-fueled units, the issue of proper fuel phasing is an important one.

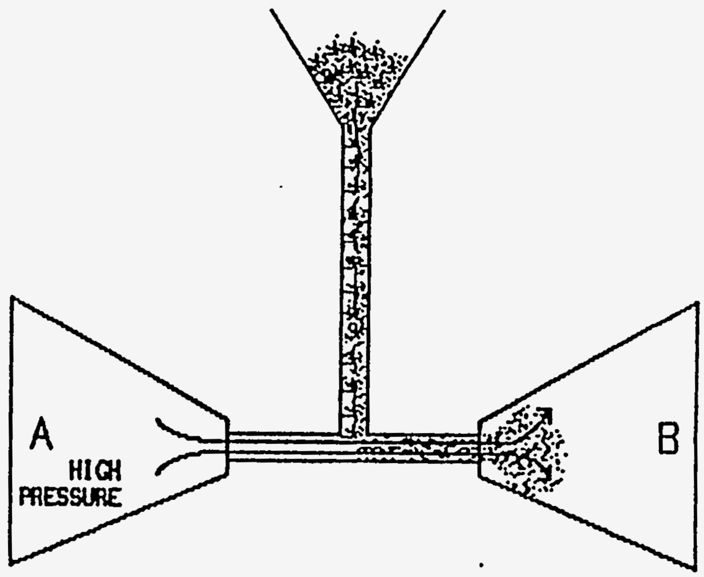
As previously described, attempts were made to improve the phasing of coal feed using a phase chamber attached directly to a single pulse combustor. The phase chamber served a secondary function of preheating and conditioning (partially volatilizing) the coal prior to injection. Although the phase chamber appeared to marginally improve the performance for a single combustor, it was evident that complete fuel phasing was not attained.

Therefore, it was determined that two pulse combustors operated in tandem may offer an improved means for fuel phasing. As a secondary benefit, operation of two combustors with a 180^{o} phase lag reduces total system noise emission through acoustic superposition damping.

The fuel system for the phased tandem units consists of a single feed line which intersects an injection tee. Each leg of the injection tee is directly connected to a separate combustion chamber. The fuel tee acts as a dynamic coupling allowing automatic fuel biasing between the combustors as shown in Figure 2-13.



FUEL INJECTION INTO CHAMBER A



FUEL INJECTION INTO CHAMBER B

FIGURE 2-13:

AUTOMATIC FUEL BIASING DURING TANDEM UNIT OPERATION

Efficient phasing of fuel in the coupling line is dependent on the ability to operate the tandem chamber 180^{0} 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^{0} 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^{0} 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 a 180^{0} phase lag.

However, in order to accelerate the hunt 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 can be envisioned: aerodynamic vale coupling, and tailpipe coupling. Since the displacement of gases is greatest at the tailpipe and tailpipe coupling is anticipated 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 expirates 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^{0} 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 the tandem 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 effective for particle scrubbing and thus accelerated combustion.

Several different schemes for coupling the tandem pulse combustor were attempted under this investigation. Some of these are depicted in Figures 2-14 through 2-16. Based on experimental evaluation, tailpipe coupling offered the greatest level of enhanced performance for the residential-scale unit. For this reason, extended evaluation tests were conducted for the tandem advanced, refractory-lined conical chamber operating with tailpipe coupling as reported in Section 4.2(c).

2.4 SUMMARY OF DESIGN DEVELOPMENT

The development of an optimized coal pulse combustion chamber and configuration involved the testing of dozens of alternate design configurations. In the preceding subsections, a discussion was made to present an overview of the design evolution and logic leading to the selection of an optimum system configuration. Table 2-1 provides a summary of the most significant test configurations investigated during this program.

In the following section, a detailed account of the experimental test results will be given.

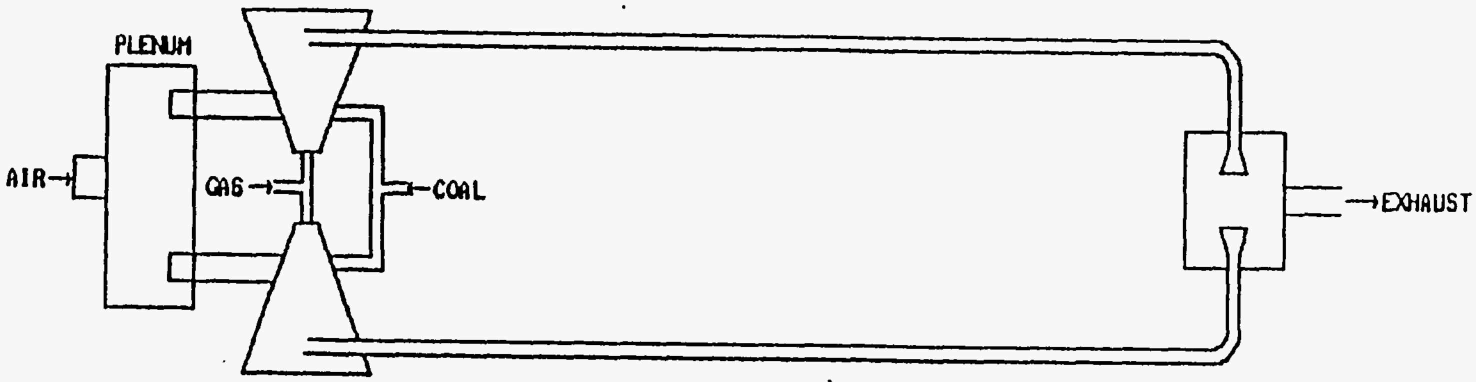


FIGURE 2-14:

TANDEM PULSE COMBUSTOR WITH COUPLED EXHAUST

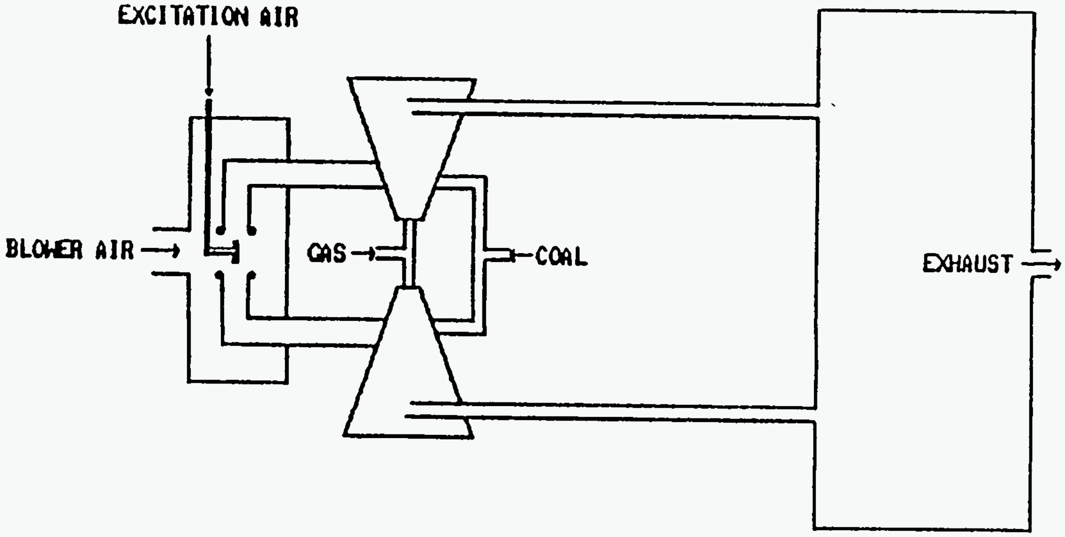


FIGURE 2-15:

TANDEM PULSE COMBUSTOR
(WITH COUPLED INLETS)

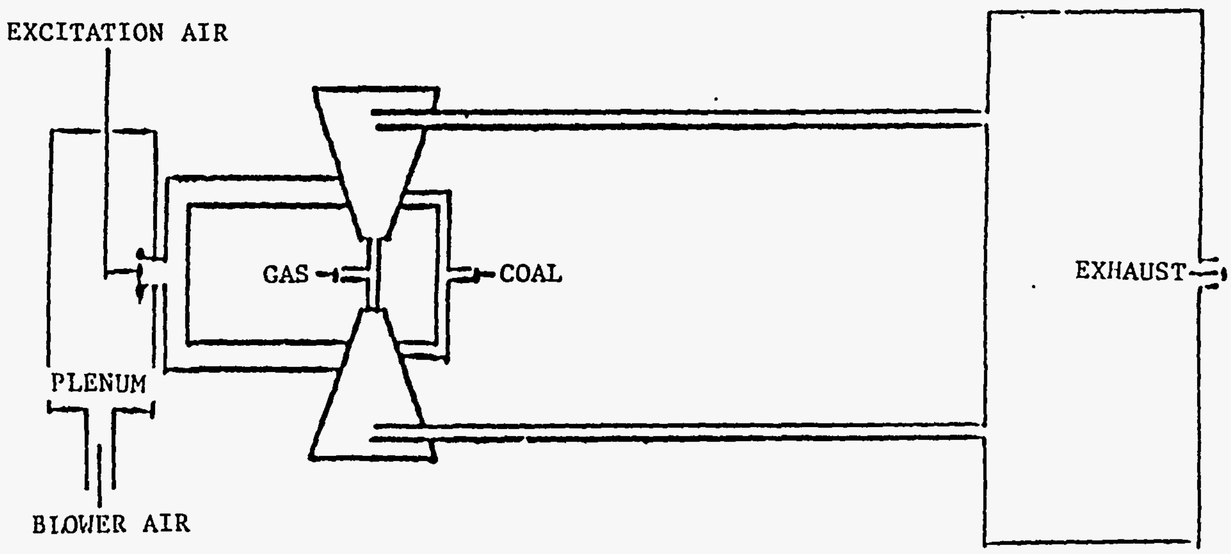


FIGURE 2-16:

**TANDEM PULSE COMBUSTOR
(WITH INLET TEE)**

TABLE 2-1:

SUMMARY OF SIGNIFICANT DESIGN CONFIGURATIONS TESTED

COMBUSTOR TYPE:	LOCKWOOD/ SNECMA	MODIFIED GRAND FORKS	MODIFIED GRAND FORKS	
CONFIGURATION :	Single Unit	Single Unit	Single Unit	
COUPLING METHOD :	None	None	None	
SCALE (Btu/hr) :	100,000	1,500,000	1,500,000	
FREQUENCY (Hz) :	180	80	80	
FUEL INJECTION :	Direct	Direct	Phase Chamber	
PRIMARY PARTICULATE COLLECTION:	Baghouse	Baghouse	Baghouse	

COMBUSTOR TYPE:	MODIFIED HANBY	MODIFIED HANBY	ADVANCED QUADRATIC FORM	ADVANCED QUADRATIC FORM
CONFIGURATION :	Single Unit	Single Unit with Jet Pump	Single Unit	Tandem
COUPLING METHOD :	None	None	None	Aerovalve
SCALE (Btu/hr) :	1,500,000	1,500,000	1,500,000	1,500,000
FREQUENCY (Hz) :	70	70	80	80
FUEL INJECTION :	Direct	Direct	Direct	Bias
PRIMARY PARTICULATE COLLECTION:	Baghouse	Baghouse	Cyclone/Scrubber	Cyclone/Scrubber

COMBUSTOR TYPE:	BARE METAL CONICAL	BARE METAL CONICAL	BARE METAL CONICAL	BARE METAL CONICAL
CONFIGURATION :	Single	Single	Tandem	Tandem
COUPLING METHOD :	None	None	Aerovalve	Tailpipe
SCALE (Btu/hr) :	100,000	100,000	200,000	200,000
FREQUENCY (Hz) :	100	55	55	55
FUEL INJECTION :	Direct	Direct	Bias	Bias
PRIMARY PARTICULATE COLLECTION:	Scrubber	Scrubber	Scrubber	Scrubber

COMBUSTOR TYPE:	REFRACTORY CONICAL	REFRACTORY CONICAL	REFRACTORY CONICAL	REFRACTORY CONICAL
CONFIGURATION :	Single	Tandem	Tandem	Tandem
COUPLING METHOD :	None	Aerovalve	Tailpipe	Tailpipe
SCALE (Btu/hr) :	200,000	200,000	200,000	200,000
FREQUENCY (Hz) :	55	55	55	55
FUEL INJECTION :	Bias	Bias	Bias	Bias
PRIMARY PARTICULATE COLLECTION:	Dry Cyclone/ Scrubber	Dry Cyclone/ Scrubber	Dry Cyclone/ Scrubber	Wet Cyclone

SECTION 3.0

TEST FACILITY

A description of the test facility including physical plant, ancillary systems, test stand, instrumentation and data acquisition is given in the following subsections.

3.1 PHYSICAL PLANT AND TEST STAND

All tests were conducted at MTCI's West Coast Laboratory, located in Santa Fe Springs, California. Tests were conducted indoors using proper ventilation, exhaust fans, and emission controls. The physical plant is supplied with all required utilities including a 15 psig natural gas main and a compressed air system.

Since numerous pulse combustors were tested under this investigation, a flexible test stand was used for mounting the test apparatus. The units were individually assembled and test and instrumentation connections were made as necessary to meet the technical objectives for each test run. A mobile coal feeder was employed for metering coal fuels into the combustors. Exhaust gases from the combustion apparatus were vented through overhead ducting to a fixed scrubber. A roof-mounted induced draft fan was employed for venting the gases to the atmosphere.

The MTCI facility contains a dedicated data acquisition room located in close proximity to the test stand. A general layout plan of the experimental bay which houses the residential test apparatus is shown in Figure 3-1. A more detailed description of individual test apparatus is made in the following section.

3.2 ANCILLARY SYSTEM

A. Coal Feed Systems

The coal feed system consisted of a Vibrascrew single-screw meter having a dry coal feed capacity of 30 lb/hr. The feeder transports coal to a funnel connected to an air eductor. The coal/carrier air mixture is then transported to the combustion test apparatus.

B. Scrubber

A scrubber was employed for final particulate polishing and SO_2 removal. The scrubber consisted of two parts; a quench tower and a scrubber column. The quench tower was a 2' diameter, 6' high stainless steel, packed vessel. Attached to the top of the vessel was an 10' diameter, 8' high steel pipe filled with stainless steel Raschig rings. The gases enter the quench tower through a 10" port and travel upward through the packing. A water spray is

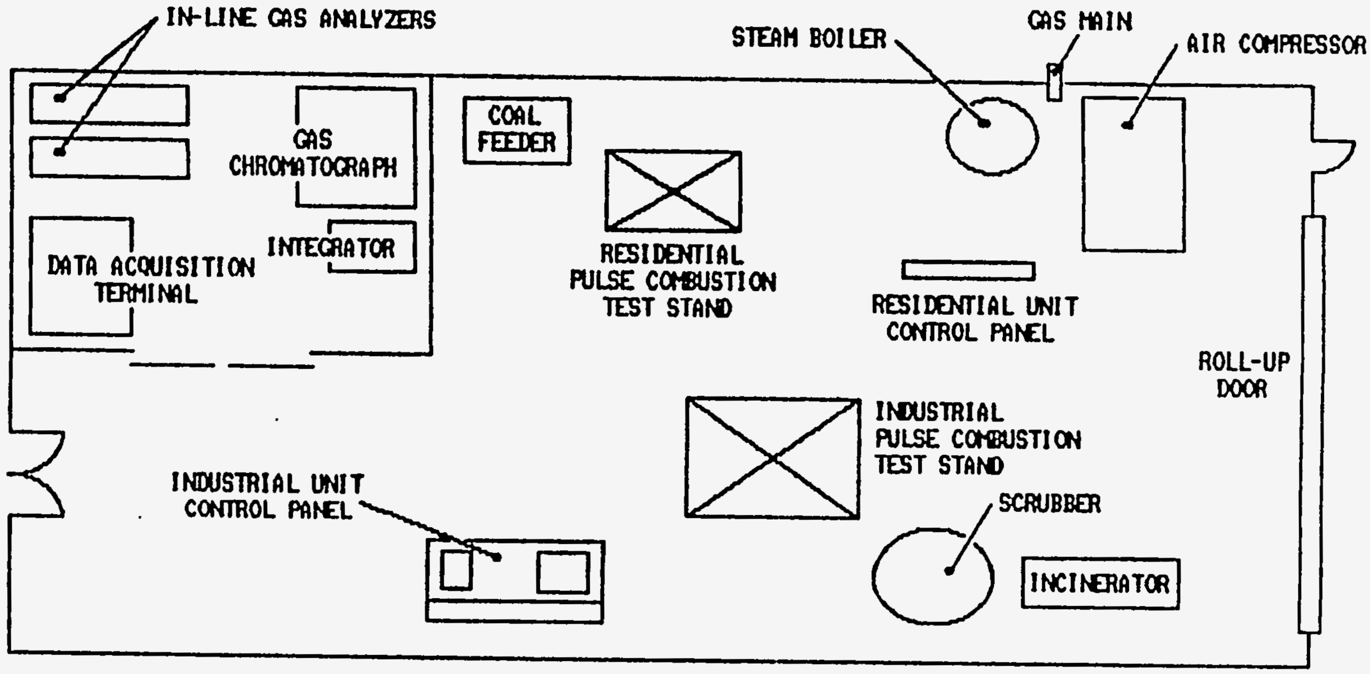


FIGURE 3-1:

PHYSICAL PLANT LAYOUT

introduced at the top of the quench tower. The cooled gases then enter the scrubber column where they contact a Na_2CO_3 solution which is introduced at the top of the scrubber packing. The circulation rate is approximately 9 gallons per minute. The unit includes a temperature and level control circuit. Finally, the clean gases exit through a roof-mounted induced draft fan to the atmosphere.

3.3 INSTRUMENTATION AND GAS ANALYSIS

Primary instrumentation and gas analyses utilized during the course of this experimental work are described in the following paragraphs.

A. Gas Chromatograph

The gas chromatograph was employed to provide a complete flue gas analysis in conjunction with additional on-line gas analyzers. The unit is a Carle model 4397 gas chromatograph equipped with seven columns:

1. 2 ft length of 2.7% Carbowa 1540 on Porasil C
2. 17 ft length; 27.5% BIS2 (EE)A on Chromosorb PAW
3. 14 ft length; 2.7% Carbowax 1540 on Porasil C
4. 4 in length; 28% DC 200/500 on Chromosorb PAW
5. 6 ft length; 80% Porapek N + 20% Porapek Q
6. 7 ft length; molecular sieve 13x
7. 3 ft length; molecular sieve 5A

All column switching is automated by computer-actuated valves. This arrangement permits accurate detection of hydrocarbons, oxides of carbon, and other gaseous effluents, while providing maximum system flexibility. Both flame ionization (FID) and thermal conductivity detectors (TCD) are employed for maximum peak resolution.

The gas chromatograph is calibrated using two pressurized cylinders, custom filled for MTCI's application. One cylinder contains a hydrocarbon mixture used to standardize the FID; the other cylinder, a special mixture for TCD calibration. Peak areas are numerically integrated, and response factors automatically calculated via a dedicated computer system.

B. Gas Analyzers

Two Horiba Model PIR-2000 Infrared Gas Analyzers were employed for continuous on-line monitoring of NO and SO_2 levels. The analyzers are based on a nondispersive infrared absorption principle. The unit has a range selectable switch and a response time of 1.2 seconds.

C. Pulse Analyzers

In order to characterize the frequency response and pressure fluctuations within the pulse combustion chamber, pressure transducers must be employed. In conjunction with a dual trace oscilloscope and an A/D converter, MTCI tested several different pressure transducers during the course of experimental work, including models from Kistler and Foxboro.

The Kistler 601 B3 is a quartz crystal piezoelectric pressure transducer. The output from the transducer is proportional to dynamic pressure. The transducer is exceptionally linear from 1 to 30,000 Hz. The output of the transducer is an electric charge which is converted to a voltage signal in a charge amplifier. The charge amplifier is a Kistler 500Y dual mode amplifier which is matched to the 601 B3. The voltage signal is then split to a dual trace oscilloscope and a data acquisition system consisting of an A/D converter as described in the following paragraphs.

The Foxboro pressure transducer generates a direct voltage signal of 1 to 5 volts corresponding to 0-15 psig. Both direct and preamplified Foxboro units have been employed. Monitoring of the Foxboro signal was performed in a manner similar to that which was described for the Kistler devices.

3.4 DATA ACQUISITION SYSTEM

The automated data acquisition system consists of a metrabyte DAS-8 analog to digital (A/D) converter which is multiplexed through a metrabyte EXP-16 expansion board. The data acquisition system allows real time monitoring and digitalization of all critical process variables including temperatures, pressures and pulse wave forms.

The DAS-8 is an eight channel, 12 bit, high speed A/D converter. It has a +/- 5 volt full scale input range with a resolution of 2.4 millivolts. The EXP-16 differentially inputs to a single output. This provides the capability to simultaneously monitor up to 128 separate channels.

The data acquisition interface boards are installed in an IBM-compatible computer offering hard disk storage capability. The data acquisition process is controlled by software developed by Laboratory Technologies Corporation. The software provides means for digital metering of all thermocouple inputs, real time tracing, graphing and data analysis.

The A/D converter is able to support sampling rates of up to 8000 Hz. This high speed sampling rate allows accurate analysis of wave forms which are generated during the pulse combustion. The acoustic wave forms are monitored from a pressure transducer or microphone. The waveform signal can be enhanced using an audio amplifier with a linear harmonic response for frequencies up to 20,000 Hz. The waveform is then digitalized using the DAS-8 A/D converter.

The digitalized signal is further processed using a Fast Fourier Transform (FFT) algorithm to decompose the waveform into a power spectrum. The power spectrum provides critical information on the relative intensity of frequency components comprising the pulse waveform. This data is useful for interpreting the performance characteristics of the pulse combustor as a function of fuel type, firing rate, hardware configuration, etc. All of the pertinent data can be easily tabulated, graphed, and saved in hardcopy form.

SECTION 4.0

DEVELOPMENT TESTING AND RESULTS

Development testing of the residential pulse combustion apparatus was conducted at MTCI's West Coast Laboratories from October, 1986 through October, 1987. The objective of these tests was to evaluate the technical viability of pulse combustion as a means for extending the applicable range of fuels for residential space heating to include coal.

In the following subsections, the results of these development efforts are reported. The discussion includes a description of test procedures, a summary of operating history, test results, and data analysis. The performance of the optimized test unit is also compared with target goals specified for coal-fueled, residential space heaters.

4.1 TEST PROCEDURES

Performance parameters were monitored in order to characterize the operation of each pulse combustor configuration. Parameters of particular interest included combustor temperature and pressure, flue gas composition, and fly ash carbon content. A discussion of test procedures relevant to the measurement of these parameters follows.

A. Temperature

Temperature measurements are commonly made at several different locations including the air inlet plenum, combustion chamber, tailpipe exit, exhaust plenum, and cyclone. A record of pertinent temperatures is made from a digital meter, or from the automatic data acquisition system.

B. Pressure

Static mean pressure measurements are made at several locations using a standard water column manometer including air inlet plenum, combustion chamber, and exhaust plenum. Dynamic pressure measurements are also made for the combustion chamber using a piezoelectric pressure transducer. Pulse waveforms are characterized using a dual trace oscilloscope or an analog to digital data acquisition system.

C. Flue Gas Analysis

Flue gas composition is determined using a combination of on-line continuous gas analyzers and a gas chromatograph. The Horiba analyzers provide accurate data on NO and SO_2 components at the ppm level. The Carle gas chromatograph is used to monitor CO_2, O_2, N_2, CO, and unburned hydrocarbons down to the .1 percent level. A portable effluent analyzer is used to measure CO levels below 1000 PPM. In some cases, Draeger tubes have also been used to confirm gas analysis measurements.

Gas sampling is typically performed at the tailpipe exit. An uncooled probe is inserted through the exit plenum and into the tailpipe to a depth of approximately 2 inches. The sample effluent is drawn through a collection filter, a secondary filter, and a knock-out pot using a vacuum pump. The vacuum pump discharges the gases to a water-cooled condenser. The gases are then split to the relevant analyzers for measurement.

D. Ash Analysis

Fly ash, which is contained in the sample effluent is deposited on a collection filter as it is drawn through the gas analysis circuit. After a sufficient collection period (which is typically determined by collection filter pressure drop), the filter is removed. The ash sample is collected, dried and weighed.

The carbon content of the ash sample is then determined using the loss-on-ignition test (700^oC, one hour). A carbon balance is then made to calculate the percent carbon burn-out for the given test condition.

A schematic of the analysis train for a typical test run is given in Figure 4-1.

4.2 OPERATING HISTORY AND TEST RESULTS

During the course of this investigation, four separate fuels were tested in the developmental combustors including natural gas, #2 fuel oil, micronized Pittsburgh #8 coal, and highly beneficiated, micronized coal from Upper Elkhorn #3 seam. The Pittsburgh #8 coal had a nominal size of 10 microns and contained 2.43 wt. % sulfur and 12.99 wt. % ash on an as received basis. The micronized Upper Elkhorn #3 coal was supplied by Energy International, Inc. and contained .69 wt. % sulfur and 1.16 wt. % ash. The mean particle diameter was 7 microns. Sample analysis for the two coal feeds used in this investigation are shown in Tables 4-1 and 4.2.

A. Lockwood/SNECMA Tests

Tests were performed on the Lockwood/SNECMA pulse combustor during the month of March, 1987. Natural gas, #2 fuel oil, and Pittsburgh #8 were tested at this time. Data was acquired on the combustor performance for sixteen separate test points. Tables 4-3 through 4-6 summarize the results of tests conducted during this period of development.

In addition to tests on natural gas, tests with #2 fuel oil were conducted to evaluate this fuel option for start-up. No combustor modification was required and the operation of the combustor was in general satisfactory. Nevertheless, problems were encountered with the oil feed system which did not permit steady-state operation. Based on the test experience in these runs, when the feed system operated satisfactorily for short periods of time, a decision was made to move to coal fired testing and a conclusion was reached that the combustor design is capable of operation on #2 fuel oil for start-up without modifications of the combustor design. Heat-up

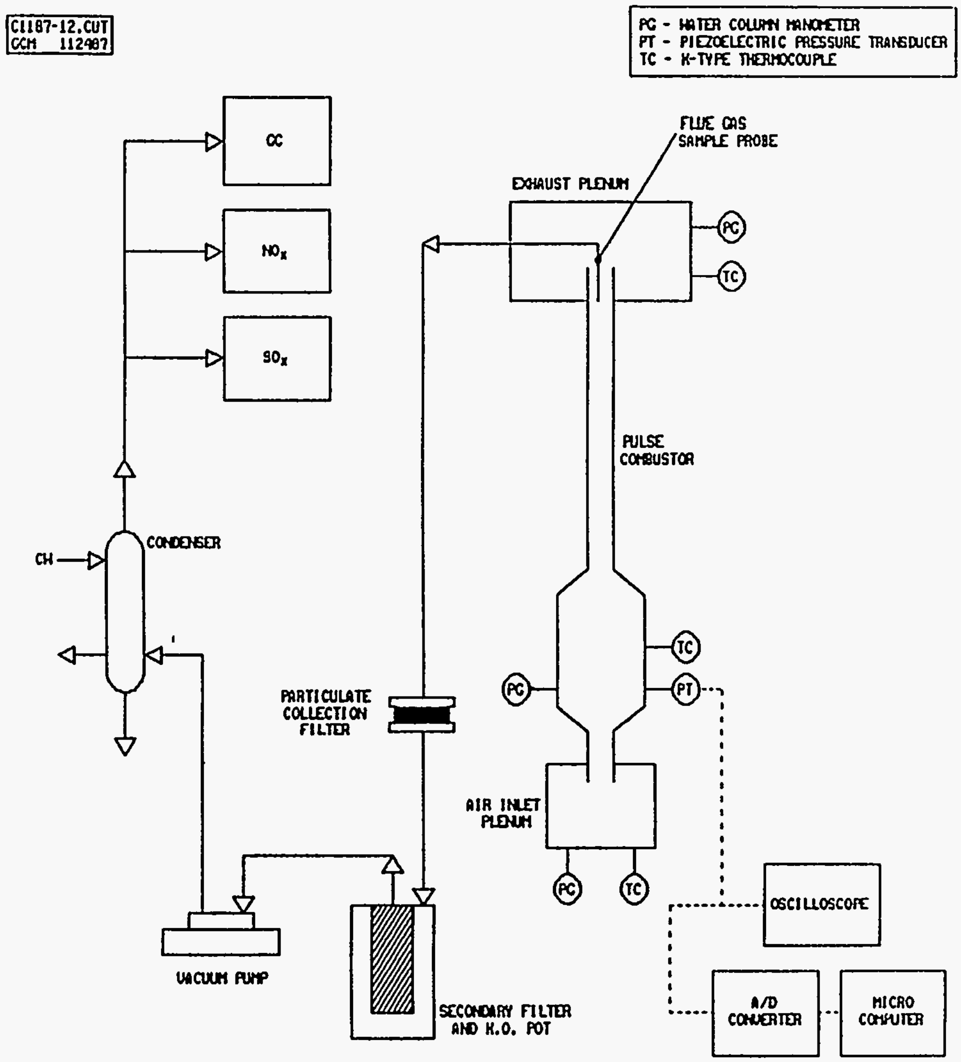


FIGURE 4-1:

TYPICAL ANALYSIS TRAIN

TABLE 4-1:

ANALYSIS OF MICRONIZED PITTSBURGH #8 COAL

PROXIMATE ANALYSIS	% WEIGHT AS RECEIVED	% WEIGHT DRY BASIS
Moisture	1.62	XXXX
Ash	12.99	13.20
Volatile	37.64	38.26
Fixed Carbon	47.75	48.54
	100.00	100.00
Btu/lb	12650	12858
Sulfur	2.43	2.47
Alk. as Na-O	----	0.22
SULFUR FORMS		
Pyritic Sulfur	1.56	1.59
Sulfate Sulfur	0.01	0.01
Organic Sulfur	0.86	0.87

FUSION TEMPERATURE OF ASH	REDUCING	OXIDIZING
Initial Deformation	2150^{o}F	2550^{o}F
Softening (H=W)	2550^{o}F	2700+^{o}F
Softening (H=1/2W)	2600^{o}F	2700+^{o}F
Fluid	2700^{o}F	2700+^{o}F
Silica Value	73.67	
Base: Acid Ratio	0.26	
T_2SO Temperature	2600^{o}F	

ULTIMATE ANALYSIS	% WEIGHT AS RECEIVED	% WEIGHT DRY BASIS
Moisture	1.62	XXXXX
Carbon	70.00	71.15
Hydrogen	4.90	4.98
Nitrogen	1.23	1.25
Chlorine	0.14	0.14
Sulfur	2.43	2.47
Ash	12.99	13.20
Oxygen (diff)	6.69	6.81
	100.00	100.00

TABLE 4-1:

ANALYSIS OF MICRONIZED PITTSBURGH #8 COAL
(Continued)

MINERAL ANALYSIS OF ASH	% WEIGHT IGNITED BASIS
Silica, SiO_2	50.37
Alumina, Al_2O_3	25.59
Titania, TiO_2	1.12
Ferric Oxide, Fe_2O_3	16.78
Lime, CaO	0.68
Magnesia, MgO	0.54
Potassium Oxide, K_2O	2.12
Sodium Oxide, Na_2O	0.29
Sulfur Trioxide, SO_3	0.56
Phos. Pentoxide, P_2O_5	0.09
Strontium Oxide, SrO	0.01
Barium Oxide, BaO	0.08
Manganese Oxide, Mn_3O_4	0.00
Undetermined	1.77
	100.00
Fouling Index	0.08
Slagging Index	0.64

TABLE 4-2:

DRAVO ENGINEERING COMPANIES, INC.
SAMPLE ANALYSIS SHEET

DRAVO ENGINEERING COMPANIES, INC.

SAMPLE ANALYSIS SHEET

COAL I.D. UE3-123-MCO-C

COAL DESCRIPTION: Upper Elkhorn #3 Seam - Deep Cleaned Coal Dry Micronized for HTCI

PROXIMATE:

MOISTURE	1.92%
VOLATILE MATTER	36.06%
FIXED CARBON	62.78%
ASH	1.16%
SULFUR	0.69%
BTU/LB	15016

ASH CHEMISTRY:

SiO_2	40.30%
Al_2O_3	27.80%
TiO_2	1.65%
Fe_2O_3	18.30%
CaO	3.45%
MgO	0.77%
K_2O	1.52%
Na_2O	1.52
SO_3	2.40%
P_2O_5	1.52%
SrO	0.48% Est.
BaO	0.45% Est.
Mn_3O_4	0.02% Est.
BASE ACID RATIO	0.37
T-250 TEMP.	2450°F

ULTIMATE:

CARBON	85.19%
HYDROGEN	5.14%
NITROGEN	1.57%

ASH FUSION: REDUCING

INITIAL	2090°F
SOFTENING	2230°F
FLUID	2370°F

FREE SWELLING INDEX:

6.5

HARDGROVE GRINDABILITY:

NA

Dravo

TABLE 4-2:

DRAVO ENGINEERING COMPANIES, INC.
SAMPLE ANALYSIS SHEET
(Continued)

Research Center

MICROTRAC PARTICLE SIZE ANALYSIS

Sample Identification: UE3 -123 -MCO

Test No.
Date 7-24-87

Division or Customer Name:

Tested by [signature]

Sample Preparation: ULTRASONIC & WETTING AGENT

RUN TIME: 573 Sec.

176	----------0
125	----------0
88	----------0
62	----------0
44	----------0
31	----------0
22	---------99
16	---------98
11	--------84
7.8	-----63
5.5	---43
3.9	-24
2.8	9
150	0
106	0
75	0
53	0
38	0
27	1
19	5
13	-----65
9.4	----------0
6.6	---------95
4.7	---------95
3.3	------71
2.4	---44

176	100.0	.0
125	100.0	.0
88	100.0	.0
62	100.0	.0
44	100.0	.0
31	100.0	.3
22	99.6	1.1
16	98.4	13.7
11	84.7	20.8
7.8	63.8	19.9
5.5	43.9	19.9
3.9	24.0	14.8
2.8	9.2	9.2

1.13	CS
7.05	MV
12.5	PH
6.09	PM
2.79	PS
.111	dV

2197

NOTES:

TABLE 4-3:

DATE	3/09/87	3/09/87	3/09/87	3/09/87
COAL TYPE		PITT 8 MICRO	PITT 8 MICRO	PITT 8 MICRO
BTU/HR GAS	252313	85342	29768	29768
BTU/HR COAL	0	38400	89600	89600
TOTAL BTU/HR	252313	123742	119368	119368
PERCENT COAL	0	31	75	75
INCHES BOOST	7	4.5		7
PSIG P-P	4	4	4	4
FREQUENCY	200	167	167	176
TEMPERATURES				
CHAMBER	2000+	1400	1910	1980
TAILPIPE	2000+	1850	1995	2000+
CYCLONE				
CYCLONE EXIT				
TAILPIPE SKIN	1640	1500		1650
SHROUD				
COMBUSTOR SKIN	1210	1250		1320
CARBON CONVERSION		77	77	77

TABLE 4-4:

DATE	3/10/87	3/10/87	3/10/87	3/10/87
COAL TYPE		PITT 8 MICRO	PITT 8 MICRO	PITT 8 MICRO
BTU/HR GAS	272500	89830	181300	42525
BTU/HR COAL	0	12800	89600	118400
TOTAL BTU/HR	272500	102630	270900	160925
PERCENT COAL	0	12	33	74
INCHES BOOST	5	5	4	7
PSIG P-P	4	4	4	4
FREQUENCY	200			
PERCENT OXYGEN	9		11.5	
TEMPERATURES				
CHAMBER	1700	2192	2053	
TAILPIPE	2276		2138	
CYCLONE				
CYCLONE EXIT				
TAILPIPE SKIN	1550		1715	
SHROUD				
COMBUSTOR SKIN	1193		1385	
CARBON CONVERSION		84	84	84

TABLE 4-5:

DATE	3/11/87	3/11/87	3/11/87	3/11/87
COAL TYPE			PITT 8 MICRO	PITT 8 MICRO
BTU/HR GAS	240000	240000	157300	0
BTU/HR COAL	0	0	44800	89600
TOTAL BTU/HR	240000	240000	202100	89600
PERCENT COAL	0	0	22	100
INCHES BOOST		5	6	
TEMPERATURES				
CHAMBER	1820	1840		
TAILPIPE	2012	2012	1872	1785
CYCLONE				
CYCLONE EXIT				
TAILPIPE SKIN	810	820	1590	
SHROUD				
COMBUSTOR SKIN	721		1380	
CARBON CONVERSION			94	94

TABLE 4-6:

DATE	3/12/87	3/12/87	3/12/87	3/12/87
COAL TYPE			PITT 8 MICRO	PITT 8 MICRO
BTU/HR GAS	219150	205300	85250	38120
BTU/HR COAL	0	0	51200	89600
TOTAL BTU/HR	219150	205300	136450	127720
PERCENT COAL	0	0	38	70
INCHES BOOST	6	8	8.5	4
PERCENT O2			9.2	
PPM CO			300	
PPM NOx			200	400
PPM SO2				600
TEMPERATURES				
CHAMBER	1869	2000	2066	1950
TAILPIPE	2000	2002	2165	1966
CYCLONE				
CYCLONE EXIT				
TAILPIPE SKIN	1880	1900	1374	1230
SHROUD				
COMBUSTOR SKIN	1700	1850	1292	1270
CARBON CONVERSION			88	88

rate of the combustor with oil firing was approximately half of that for natural gas. This was attributed to the higher flame luminosity with oil firing.

The maximum firing rate achieved on natural gas was only 272,000 Btu/hr, representing a combustor volumetric heat release of approximately 8 MMBtu/ft^3/hr. The maximum coal feed rate for these runs was 118,400 Btu/hr with support gas and 89,000 Btu/hr for coal firing only. Excitation air was required for all tests.

On March 9, 1987, the Lockwood unit was tested with natural gas alone, and with coal using a phased injector. The coal feed rates represented 31 percent and 75 percent of the total heat release. The frequency of operation was typically 200 Hz on gas only and approximately 170 Hz when gas and coal were fired in combination.

The unit attained boost pressures of up to 7 inches water. The peak to peak pressure appeared relatively constant at 4 psig despite changes in firing rate. The combustor temperature was typically in the range of 1900°F to 2000°F and exceeded 2000°F at the highest gas firing rate of 252,000 Btu/hr. The unit was operated with excitation air at all times.

Since coal was fed only for a short period of time, a single fly ash sample was collected. Carbon in ash analysis indicated a carbon burn-out level of 77 percent. It should be noted that the error margin in ash analysis for this particular run was high due to the small quantity of ash sample collected.

Tests were continued on March 10, 1987. The unit was operated with coal feed rates representing 12.5 percent, 33 percent, and 73.5 percent of the total heat release. Chamber temperatures of almost 2200°F were achieved. The unit was operated with relatively high excess air levels. Oxygen concentration in the flue gas varied from 9 percent to 11.5 percent (dry gas basis). The phased injector was not utilized for this run.

Carbon in ash analysis revealed carbon burn-out levels of 84 percent. It is significant to note that the ability to reduce the gas support level was diminished for this run due to the absence of the phased injector. The minimum allowable gas rate without the phased injector was 42,500 Btu/hr. This is probably due to the influence of the phase injector in preheating and partially devolatilizing the coal prior to injection thus, in effect, providing in situ support gas. Excitation air was required for stable operation during these runs.

Further tests were conducted on March 11, 1987 using a phased injector. In addition, an electric heater was used to preheat the lift air containing the feed coal. A maximum preheat temperature of 530°F was obtained.

Combustor performance significantly improved under these conditions. The unit was capable of operating on coal only, without support gas. Coal was fed at a rate of 89,000 Btu/hr. Measured carbon burnout levels of 94 percent were obtained. Excitation air was required for stable operation. Coal feed was eventually terminated due to plugging of the preheater.

These results reveal the importance of preheating and preconditioning the coal prior to injection. The phased injector serves this purpose to some degree, however, direct air preheat appears to be even more effective. It is evident that a more practical means are required for preconditioning the fuel in commercial residential systems.

A final set of runs were conducted on March 12, 1987. Due to the plugging problems experienced in the previous run, lift air preheating was not used for this run. Instead a vent was installed in the phased injector to allow flue gases to more effectively back-flow into the injection chamber. The vent gases were recirculated back to the eductor. This concept was employed in an attempt to improve in situ preheating of the coal without the use of external methods (i.e., electric preheater).

Coal was fed at a rate representing 70 percent of the total fired fuel. Performance of the unit did not appear to be as robust as for the prior run with lift air preheat. A carbon burn-out efficiency of 88 percent was measured.

Based on the test data, several important conclusions were drawn. First, the degree of mixing in the Lockwood type chamber was insufficient for coal combustion purposes. It should be noted that the Lockwood pulse combustor design was optimized for propane and natural gas fuels, and does not represent an optimum design for coal. This is evidenced by the fact that coal combustion could not be supported without the use of excitation air. Therefore, new chamber geometries must be developed for efficient coal combustion.

Secondly, preconditioning of coal appears to be extremely important for the proper functioning of the pulse combustor. This is probably due to the fact that pyrolysis products play a significant role in re-ignition of the combustor, and stabilization of the pulse oscillations. Therefore, new system configurations must be developed which consider the issue of fuel phasing and preconditioning.

B. Advanced Bare Metal Conical Combustor Tests

As described in the previous section, additional operating experience was gained from experiments performed on the Modified Grand Forks and the Modified Hanby designs. It was evident from this work that combustor designs specific to coal fuels were necessary for optimized performance. Another significant piece of information gained from this work was that fuel phasing was essential to achieve high carbon burn-out efficiencies.

Based on this evidence, development activities focused on the design of new advanced chamber configurations. As a result of these activities, two advanced chamber designs evolved; a quadratic form chamber, and a conical chamber. The quadratic form chamber appeared to be best suited to the industrial-scale market. The simple construction of the conical design lent itself more readily to the small-scale residential applications.

In conjunction with the development of the advanced conical chamber design, an improved means of fuel phasing was identified. This method involved the use of tandem pulse combustors to supercharge the coal fuel in phase with the combustion cycle. It was therefore determined that the advanced conical chamber would be ultimately tested in a tandem configuration. However, to provide initial baseline data, a single conical chamber was investigated.

In the initial version, the conical chambers were constructed of bare metal to avoid slagging. The conical chambers employed tangential aerovalves and tailpipes to enhance mixing in the combustion chamber. The design and performance characteristics of the conical design were discussed more fully in the previous section.

During the month of May, a single, bare metal conical chamber was tested. Test runs on both gas and coal were performed. During one of the initial tests, the combustor was fired on natural gas over the range from 90,000 Btu/hr to 210,000 Btu/hr. The unit continuously self-aspirated combustion air over the full range.

In this configuration, the tailpipe was 144" long and had a constant I.D. of 1.12". The combustor temperature was typically in the range of 2050^{o}F. The boost pressure in this unit while self aspirating was low (2 to 4 inches of water) increasing with the excitation air rate.

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 125 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" 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:

- Helmholtz frequency $= c/(2\ L_t) * (V_t/V_c)^{.5}$
- Quarter wave frequency $= c/[4(L_t+L_c)]$

C = speed of sound = 49 x T_{abs}

Lt = length of the tailpipe

Lc = length of the chamber

Vt = tailpipe volume

Vc = chamber volume

The results are shown in Table 4-7. Case B reflects the design configuration used in the current set of tests. It is evident that the experimentally measured frequencies are quite close to those which are predicted on theoretical grounds.

Tests were also 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 leakage 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 4 to 7 lbs/hr 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.

C. Tandem Unit

Following the single, bare metal, conical combustor tests, a tandem unit was assembled. The tandem unit was coupled at the tailpipes with a 4" gap between the exits. Initial testing of the coupled tandem unit was performed using 68" tailpipes.

The operating frequency of the 68" tandem unit was 100 Hz. The unit operated on Pittsburgh #8 coal with a small amount of support gas. The requirement for support gas was probably due to the high heat loss from the bare metal combustor, indicating that a refractory-lined unit might be necessary. It should also be noted that excitation air was require for stable operation during these runs.

The coal feed rate was varied during the course of the tests from 5 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 4-8 summarizes the data collected from the bare metal tandem advanced combustors. Note that the frequency for the tandem units was recorded using a microphone pick-up. Therefore, for tandem units operating 180^{0} out of phase, the recorded frequency is two times the true frequency for an individual combustor.

Additional tandem unit tests were made to investigate alternate coupling techniques. Preliminary results indicated promising performance for coupling through the aerodynamic valves. However, more extensive testing in late July and early August revealed that stable operation in the aerovalve coupled configuration could only be maintained for a limited range of firing rates. Furthermore, a consistent 180^{0} peak pressure phase lag could not be sustained for prolonged periods. Instead, the phase lag appeared to float in a

TABLE 4-7:

FREQUENCY CALCULATION

CASE	A	B	C	D
TAILPIPE STR LENGTH inches	55	144	55	30
TAILPIPE STR D inches	1.12	1.12	1.12	1.12
TAILPIPE CONE LENGTH inches	13.50	0	4	4
TAILPIPE CONE MIN D inches	1.12	1.12	1.12	1.12
TAILPIPE CONE MAX D inches	4	4	2	2
TAILPIPE LRG LENGTH inches	0	0	24	29
TAILPIPE LRG D inches	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 inches	1.20	1.20	1.20	1.2
CHAMBER MAX D inches	5	5	5	5
CHAMBER VOL cubic inches	66	66	66	66
TEMPERATURE ^{o}F	2460	2460	2460	2460
Frequencies				
QUARTER WAVE, HZ	96	48	80	103
HEMLHOLTZ, HZ	96	47	81	103

TABLE 4-8:

SUM OF RESULTS ON BARE METAL TANDEM
ADVANCED CONICAL COMBUSTORS

DATE	6/08/87	6/08/87	6/08/87	6/08/87
COAL TYPE			PITT 8 MICRO	PITT 8 MICRO
BTU/HR GAS	141000	202400	202400	87900
BTU/HR COAL	0	0	64000	243200
TOTAL BTU/HR	141000	202400	266400	331100
PERCENT COAL	0	0	24	73
INCHES BOOST	9	10	12	8
DECIBLES	110	111	110	
FREQUENCY	100	200	200	200
PERCENT O2	5.79	4.16	3.02	11.3
PERCENT CO	0.07	1.71	0.56	
PERCENT CO2	9.76	5.35	12.41	
TEMPERATURES				
CHAMBER	1957	2012	1976	1994
CARBON CONVERSION			87	92

transitory manner. This behavior was not observed for the tailpipe coupled configuration.

Based on these results, it was determined that a more comprehensive review of the existing data was necessary in order to better identify which system configuration offers the greatest promise for efficient coal combustion.

The following logical conclusions have resulted from experimental observations:

A) In order to achieve high combustion efficiency, coal fuel must be properly phased during the combustion cycle. Therefore, some form of fuel phasing is necessary. This can be accomplished using a single combustor with a phase injector or by using a coupled tandem unit with a fuel biasing injector. Based on experimental observation, the later option is more easily controlled, simpler in design, and offers the best performance.

B) Due to the combustion characteristics of coal, the optimum frequency of operation for coal fuels is less than that for gaseous fuels. This necessitates the use of longer tailpipes. The 140" tailpipe appears to result in an operating frequency which is consistent with the heat release rates for micronized coal. Furthermore, the additional tailpipe length offers viable means for efficient heat transfer.

C) Tandem combustor coupling can be accomplished by two primary means: (1) aerodynamic valve coupling, and (2) tailpipe coupling. Aerodynamic valve coupling is believed to be effective if sufficient thrust can be obtained as a result of high peak combustor pressure. However, it is now believed that for the small residential combustor which exhibits relatively low peak-to-peak pressures, tailpipe coupling is more effective over the entire range of turndown. Furthermore, since the residential unit operates with low ash coal in a non-slagging mode, ash erosion at the tailpipe couple is not expected to be problematic.

D) The chamber temperature for the bare metal unit is approximately 2000°F. In order to improve the combustion performance, a combustor temperature of 2200°F is desirable. This necessitates the use of a refractory-lined combustion chamber.

Based on the above conclusions which were drawn from experimental development testing, it was determined that the final design to be used for extensive data evaluation would include: 1) an advanced refractory chamber, (2) tandem unit operation, and (3) 140" tailpipes coupled at the exhaust plenum.

4.3 EVALUATION TESTING OF THE TANDEM REFRACTORY-LINED CONICAL CHAMBER TESTS

Duel refractory-lined conical chambers were fabricated in August, 1987. The shape and internal dimensions of the refractory units were unchanged from the bare metal version. The refractory-lined conical chamber is depicted in Figure 4-2.

As shown in Figure 4-2, the coal fuel biasing injector is located at the convergent side of the cone. The tailpipe and aerodynamic valve are located in opposing tangential fashion, with the tailpipe at the divergent end. The refractory chamber was sealed with a blind flange for inspection purposes.

140" tailpipes were attached to each combustor. The tailpipes terminated at a cylindrical exhaust plenum. The tailpipe exits were spaced at a 4" distance for optimal coupling.

In addition, a wet cyclone was incorporated into the system to investigate the feasibility of wet ash removal. Figure 4-3 illustrates the wet cyclone. Here, a modest quantity of water is atomized into the duct transition between the exhaust plenum and the cyclone. The spray droplets serve to agglomerate fly ash to a size range which is amenable to high particulate removal efficiencies using a standard cyclone.The pulse combustion apparatus was secured to the common test rig. Figure 4-4 shows the configuration of the final design.

A series of test runs were conducted from mid-September through early October to monitor the performance of the optimized configuration on deeply beneficiated DUC fuel from Upper Elkhorn #3 seam.

On September 21, 1987 the combustor was operated for a total period of approximately three hours. Of this period, DUC was fed for approximately two hours. The combustor operated in a stable manner without the benefit of support gas or excitation air over the entire coal feed period.

The coal feed rate was varied from 75,000 Btu/hr to 225,000 Btu/hr. Performance was stable over the full 3:1 turndown range. Typical combustor operating temperatures ranged from 2200^{o}F to 2300^{o}F with a mean combustor pressure of approximately 9" water column. A total of approximately 25 lbs of DUC fuel was fed during the test run.

Based on monitored oscilloscope waveforms, the two combustor chambers were running with a distinct 180^{o} phase lag, as desired. This confirmed the successful operation of the fuel biasing injector for the tandem unit. It was noted, however, that the pulse waveforms decayed on a low frequency cycle which was characteristic of the coal screw feeder (approximately .5 - 1 Hz). This appears to be aa result of non-uniform feeding by the screw feeder.

The non-uniform feed rate for the screw feeder is believed to negatively impact the performance of the unit. This is due to the fact that the pulse combustor is over-loaded with fuel during some periods of the cycle, while it is starved at other times. An improved feed system is necessary to overcome this problem.

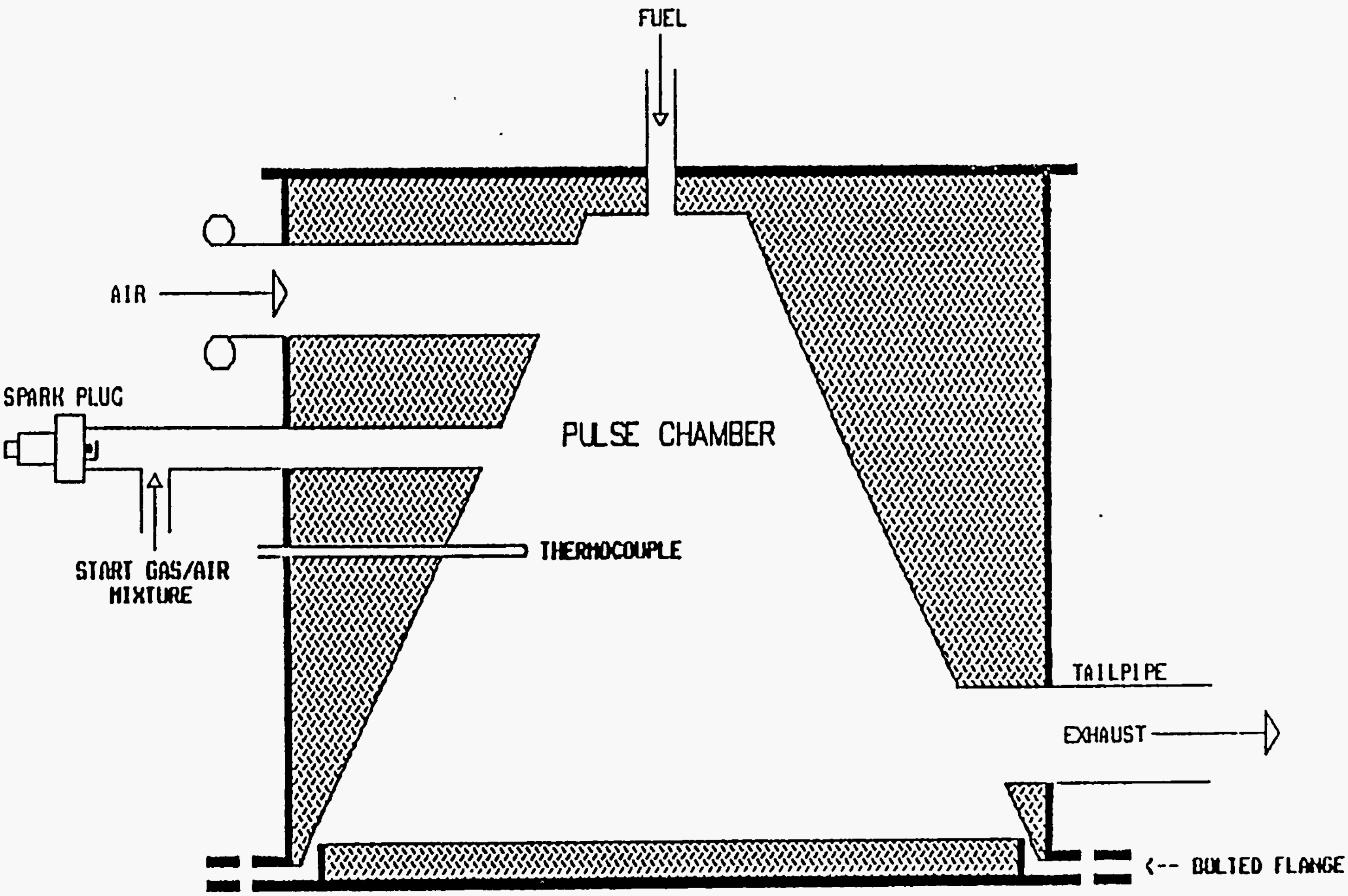


FIGURE 4-2:

RESIDENTIAL ADVANCED CHAMBER (REFRACTORY DESIGN)

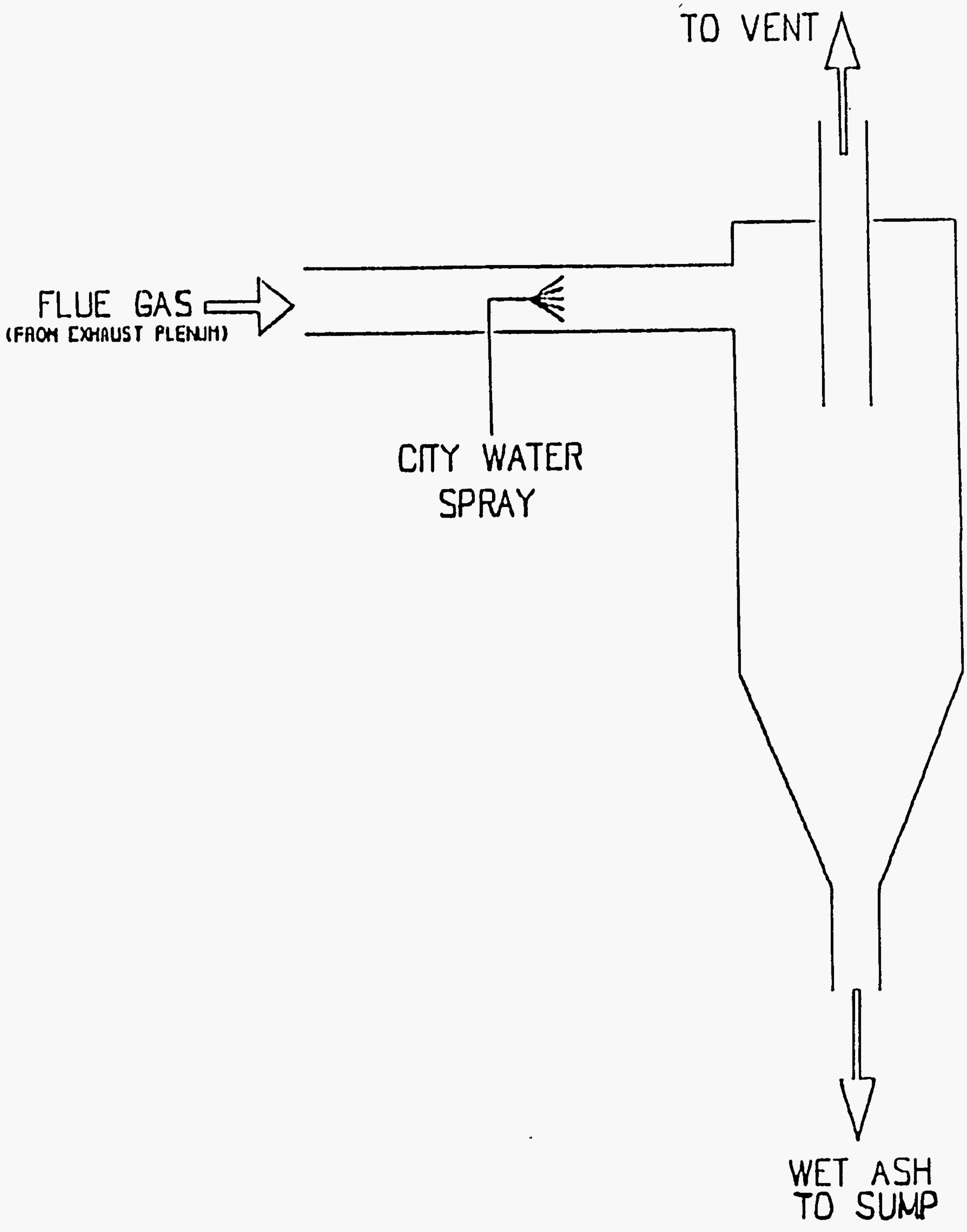


FIGURE 4-3:

WET CYCLONE PARTICLE TRAP

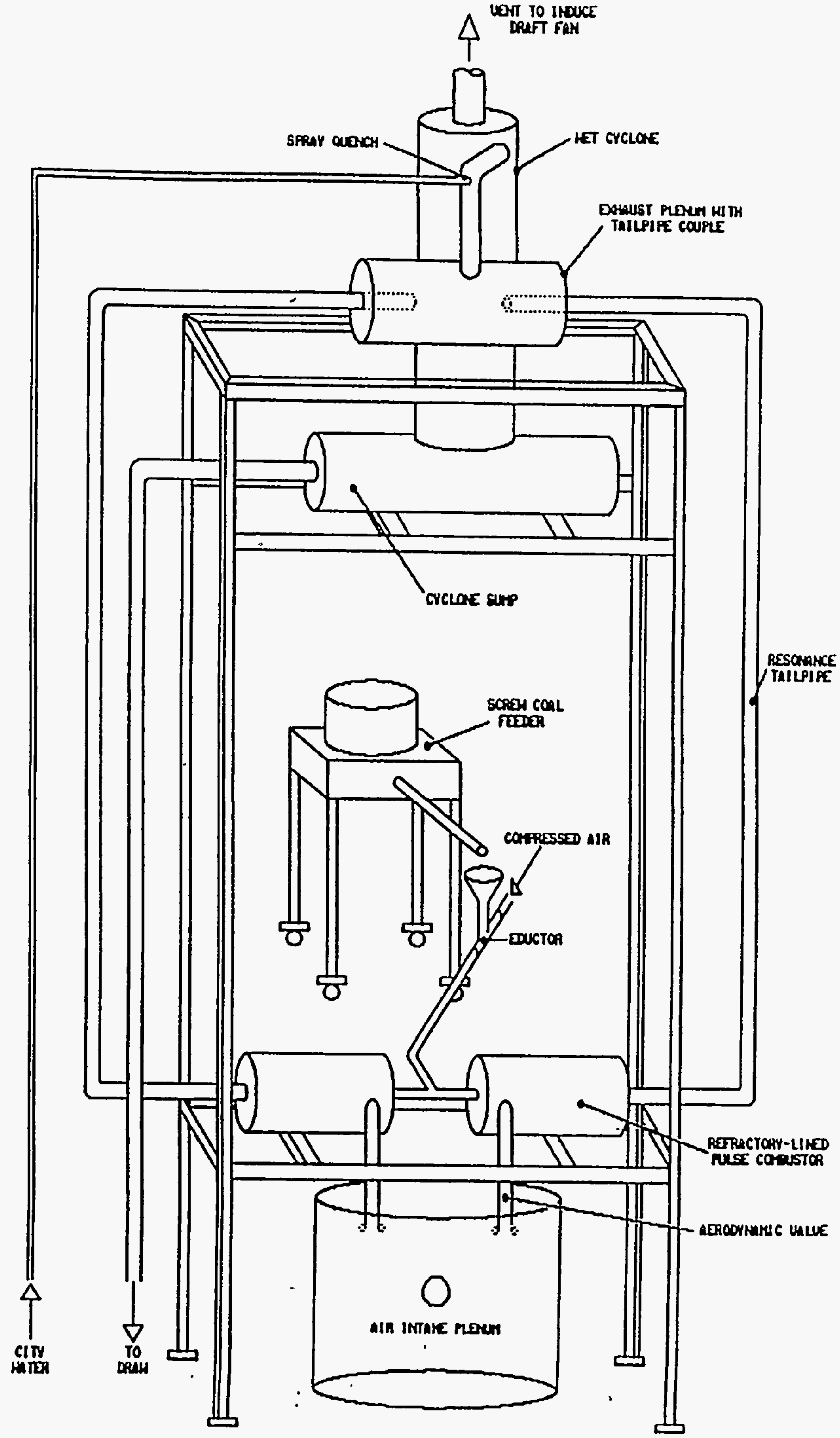


FIGURE 4-4:

OPTIMIZED RESIDENTIAL TEST APPARATUS

Fly ash samples were taken at two separate times during the run. The first sample was taken following prolonged operation at a firing rate of 150,000 Btu/hr. A second sample was taken after operation at 225,000 Btu/hr. Based on carbon in ash analyses, the carbon burnout efficiency at 150,000 Btu/hr was 97.7 percent. The carbon burnout efficiency at the 225,000 Btu/hr rate diminished considerably to 84.7 percent. The low burnout efficiency for the second sample seems to indicate that the maximum capacity of the pulse combustor was exceeded at the higher firing rate.

During this run, the pulse waveforms were monitored using both an oscilloscope and an A/D converter which digitalizes the pulse signal for analysis on a microcomputer. The pulse signal was digitalized with a 3000 Hz sampling frequency. The stored data was then processed using a Fast Fourier Transform (FFT) algorithm.

The FFT algorithm allows decomposition of the pulse waveform into its component frequencies. Figure 4-5 depicts the raw pulse signal over a .1 second sampling period. Figure 4-6 depicts the calculated frequency power spectrum for this pulse sample. The major peak in Figure 4-6 occurs at approximately 58 Hz. There also appears to be some lower frequency components. These lower frequency components may be introduced as a result of tailpipe coupling by increasing the effective length of the tailpipe. (i.e., a resonant mode which views both tailpipes as a single air column). The FFT analysis has proven to be useful in quantifying the frequency response of the combustor to an extent unattainable from oscilloscope readings alone.

Decibel sound measurements were monitored 3' from the air intake plenum. The decibel reading was 84 db. This reading is approximately equivalent to the background noise in the laboratory (coal feeder, air eductor, fans, etc.) indicating that the pulse combustor itself is operating well below 84 db.

An important observation made during the coal test was that the red-hot zone of the tailpipe extended only a very short distance from the combustor. This is conclusive evidence that the bulk of the coal combustion is occurring in the refractory-lined combustor. If hot incandescent coal particles were to have remained in the tailpipe, they would have radiated heat to the tailpipe walls thereby making them red-hot. Incandescent tailpipe regions of much longer lengths have been observed in tests performed by MTCI on other combustors where complete combustion was not attained in the combustion chamber. It is possible that the lack of unburned particles in the tailpipe for the current residential design may be a result of the efficient fuel phasing achieved by the tandem configuration.

A complete video recording of the test run was documented and is available for inspection.

A second coal run was performed on September 22, 1987. Coal was fed at a rate of 150,000 Btu/hr for a period of one hour. Observable combustor performance characteristics duplicate the previous run, however, the test was prematurely terminated due to the failure of the analytical gas sampling equipment.

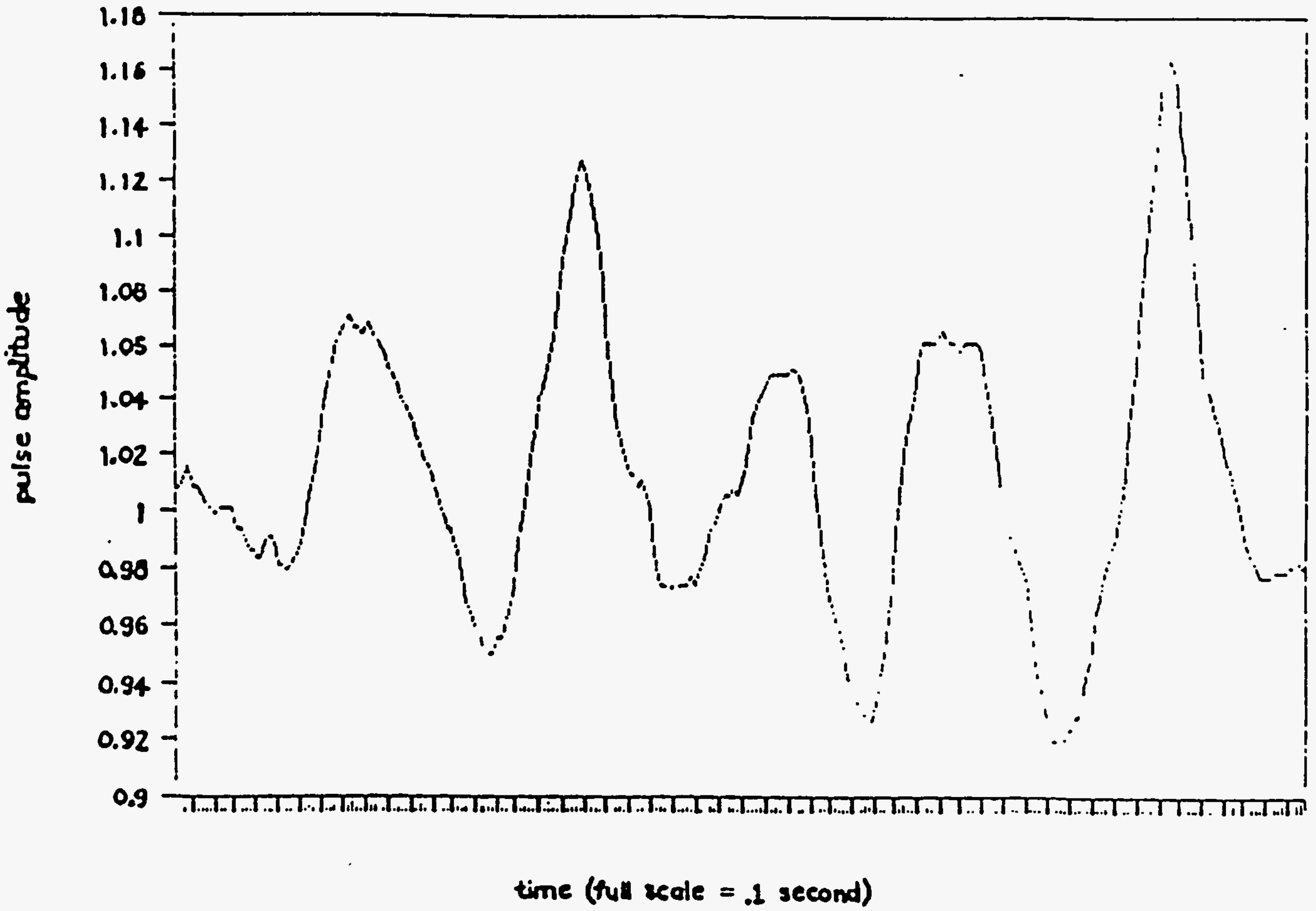


FIGURE 4-5:

PULSE COMBUSTOR ACOUSTIC SIGNAL

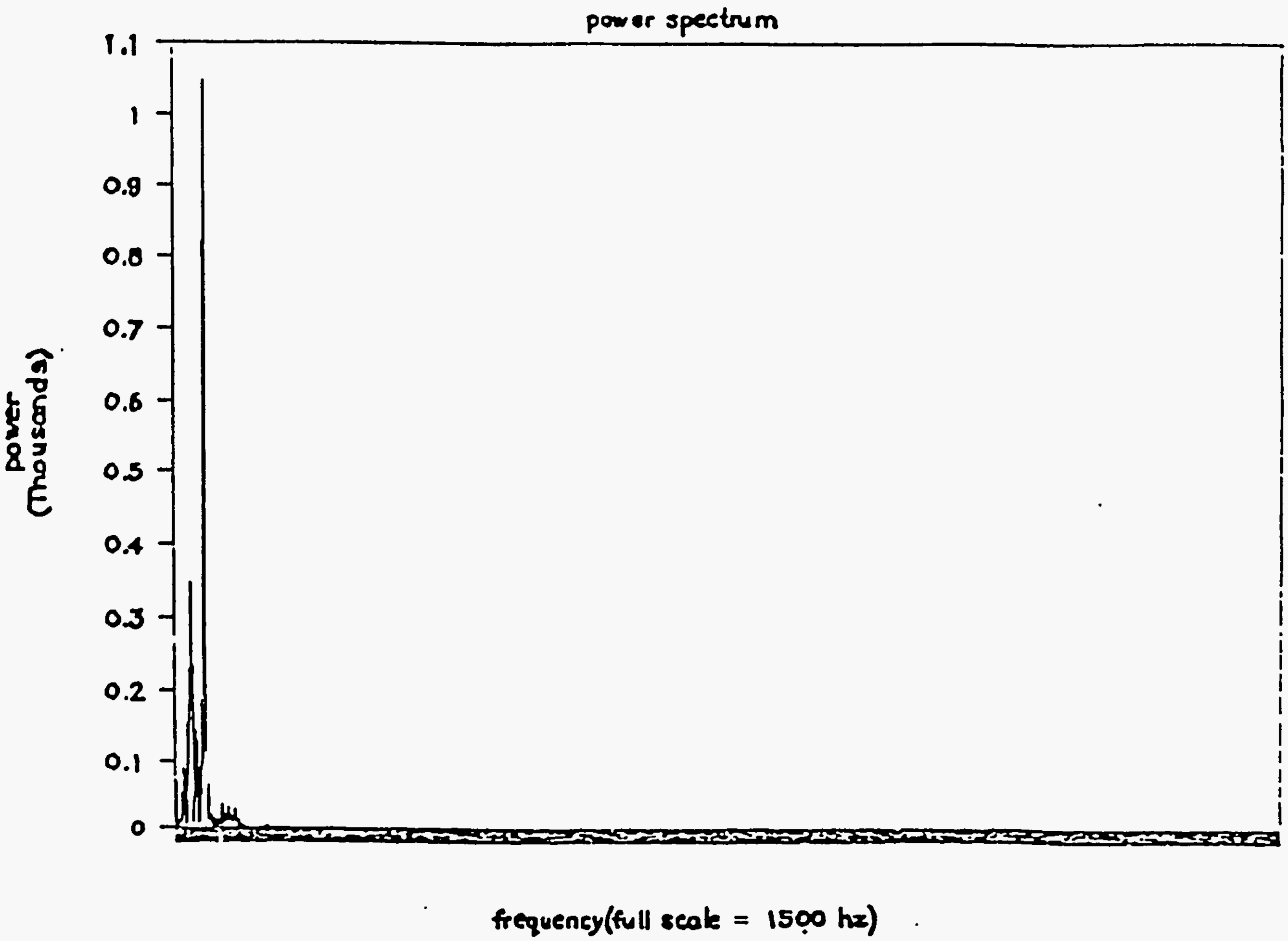


FIGURE 4-6:

PULSE COMBUSTOR ACOUSTIC SIGNATURE

A third test run was conducted on September 24, 1987. DUC was fed at a rate of 150,000 Btu/hr for a period of approximately one hour. The combustor temperature was typically 2300^{o}F and exhibited a mean pressure of 9 inches water column. 10,300 ppm of carbon monoxide was measured at the tailpipe exit with 2 percent oxygen on a dry basis. It should be noted that the oxygen concentration includes Argon since the gas chromatograph can not resolve these components. The Argon accounts for approximately one full percent of the 2 percent O_2 reading. Therefore, it is evident that the 10,300 ppm corresponds to near or sub-stoichiometric conditions. The measured carbon monoxide level at 3 percent oxygen was 6,570 ppm. A carbon burnout efficiency of 90.5 percent was measured for this run. The low carbon burnout level is clearly due to the sub-stoichiometric conditions for this test.

Another coal test run was conducted on September 30, 1987. The purpose of this test was to verify the feasibility of wet ash collection as described in the previous subsection.

A water quench spray was injected at the duct transition between the exhaust plenum and the cyclone. Water and ash is collected in a cyclone pump and then drained from an overflow tube. Fluid drained from the cyclone is then sampled to monitor the carbon burn-out efficiency.

Coal was fed at a rate of approximately 150,000 Btu/hr for a period of one hour. Measured carbon monoxide levels ranged from 2450 ppm to 3600 ppm during the course of the run. NO_x readings ranged from 200 ppm to 300 ppm (uncorrected).

Two separate ash samples were collected from the cyclone overflow tube. These samples were evaporated and then ashed to calculate the carbon burn-out efficiency. The two samples resulted in carbon burn-out values of 94.3 percent and 94.6 percent.

A final test run was made on October 8, 1987 during which DUC fuel was fed at a rate of approximately 131,000 Btu/hr for a period of two hours. The unit was operated with a wet cyclone condition as before. A fly ash sampling line was inserted downstream of the cyclone to estimate the cyclone particulate removal efficiency.

Virtually no detectable fly ash was collected in the downstream sampling port. This indicates that extremely high cyclone collection efficiencies were achieved. Four fly ash samples were collected from the tailpipe exit. Carbon burn-out efficiencies for these samples ranged from a low value of 89.8 percent to a high valve of 92.4 percent. Fly ash samples from the wet cyclone drain yielded 90.6 percent carbon burn-out efficiencies.

Table 4-9 summarizes test results obtained from the tandem, refractory-lined, conical chamber unit.

At the conclusion of the test runs, the conical chambers were disassembled for inspection. Although the interior refractory was generally found to be in excellent condition, the exit ports connecting to the tailpipes appeared to show signs of material deposition. The deposited material

TABLE 4-9:

SUMMARY OF RECENT EXPERIMENTAL RESULTS FOR RESIDENTIAL PULSE COMBUSTOR

Date	9-21-87	9-21-87	9-24-87	9-30-87	10-08-87
Chamber	conical refractory	conical refractory	conical refractory	conical refractory	conical refractory
Configuration	tandem/tailpipe	tandem/tailpipe	tandem/tailpipe	tandem/tailpipe	tandem/tailpipe
Gas feed, Btu/hr	none	none	none	none	none
Coal type	DUC	DUC	DUC	DUC	DUC
Coal feed, Btu/hr	150,000	225,000	150,000	150,000	131,000
Frequency, Hz	58	58	58	58	58
Mean Pressure, "H_2O	9	9.5	8.0	7.5	8.0
Chamber Temperature, °F	2300	2250	2350	N/A	N/A
Carbon Burnout, %	97.7	84.7	90.5	94.6	90.9
Flue Gas:(1)					
O_2%	6.5(2)	N/A	2.0/3.0	8.2(2)	11.4(2)
CO, ppm	4,860	N/A	10,032/6,570	3,220	>1,000
NO, ppm	N/A	N/A	N/A	200 - 300	200
Flue gas (corrected):(3)					
CO, ppm	6,026	N/A	10,032/6,570	4,527	>1,875
NO, ppm	N/A	N/A	N/A	281 - 422	375

1) Uncorrected Dry Gas Composition

2) Air Leak Suspected in Sample Line

3) Corrected To 3% O_2 (Dry Gas Basis)

appeared to be slag which formed during local high temperature excursions within the combustor. Note that the divergent end of the combustor where the tailpipes are attached is the hottest region in the combustor.

The material deposited in the entrance of the tailpipes resulted in a degree of flow restriction at this region. It was noted that for later runs conducted in early October, 1987 the combustors did not appear to perform as well as observed for earlier runs conducted in mid to late September. This is now believed to be due to the increasing degree of material buildup which occurred over the several week test period.

As previously discussed, the residential unit was intended for non-slagging operation. It is apparent that slagging conditions were prevalent at some time during operation of the refractory unit. The refractory thickness was approximately 1" for these combustors. It is believed that a small reduction in refractory thickness can be employed to avoid local slagging conditions.

SECTION 5.0

DATA ANALYSIS

Development testing undertaken in this investigation confirmed the feasibility of utilizing small scale pulse combustors (100,000 Btu/hr) for combusting micronized coal in the absence of support gas and excitation air.

Stable operation was obtained over the full 3:1 turndown range (75,000 Btu/hr to 225,000 Btu/hr). Out-of-phase (180°) tandem combustor operation was confirmed for the tailpipe coupled configuration. This was a significant achievement since it resolved the issue of phasing solid fuels in a pulse combustion apparatus.

In addition, the viability of a wet cyclone particle trap was verified as a means for efficient and low-maintenance fly ash rejection.

5.1 CARBON BURN-OUT

The bare metal conical pulse combustors achieved carbon burn-out levels of 92 percent. However, stable operation required that support gas be fired at a rate equaling approximately 25 percent of the total heat release.

For the refractory-lined, conical combustors, carbon burn-out levels of 91 percent to 97 percent were typical at a firing rate of 150,000 Btu/hr. As previously discussed the lower carbon burn-out efficiencies were obtained for later runs where it appeared that tailpipe restrictions may have diminished performance.

The volumetric heat release in the combustors is approximately 2 MMBtu/Ft3/hr at the 150,000 Btu/hr firing level. This represents a mean combustor residence time of approximately 24 milliseconds. Since the combustion cycle period for a unit operating at 55 - 60 Hz is 18 milliseconds, it is evident that on the average the coal particles remain in the combustion chamber for a time period which exceeds that of the combustion cycle.

It was noted that the burn-out efficiency diminished significantly at the 225,000 Btu/hr firing level. For this firing level, the mean combustor residence time is only 16 milliseconds. Therefore, the average coal particle remains in the combustor for a period less than the combustion cycle. This is reflected in the lower carbon burn-out efficiencies attained at the higher firing rate.

It is apparent that a relation exists between the characteristics of the combustor operating fuel, frequency, and firing rate for proper performance of the pulse combustors. The operating frequency must be compatible with the characteristic heat release rate of the specific fuel of choice. This is necessary to satisfy the Raleigh Criterion which states that heat release must occur during the interval of maximum combustor pressure. Thus, in the case of coal fuels which typically exhibit heat release rates which are less than that of gaseous fuels, a lower operating frequency is preferred.

Note that a primary means of supporting combustor pulsations in the case of coal fuels may be largely due to the devolatilization of coal particles leading to the generation of a rapidly combustable gaseous mixture. Under this hypothesis, the bulk of the carbon burn-out proceeds in a less rapid fashion and may occur over one or more combustion cycles. The benefits of using a phase chamber or a fuel biasing scheme, both of which expose the coal fuel to high temperature flue gas thus partially devolatilizing it prior to injection into the combustor, supports this hypothesis.

These insights reveal that the operating frequency should be selected for compatibility with the heat-up and devolatilization rates of the coal fuel for optimum support of combustor pulsations. By proper support of the combustor pulsations, the turbulent scrubbing action of the combustor can be effectively employed to complete carbon burn-out over one or more combustion cycles.

Thus, the firing rate must be sufficiently high to induce strong combustor pulsations, but should not significantly exceed a level which reduces average coal particle residence times to a value less than a combustion cycle period.

Note that Reynst has devised a coal combustion configuration which exploits the different time scales which exist for coal devolitilization and char combustion. In this concept, Reynst proposed the use of dual pulse chambers in series. It is evident that the first chamber serves principally to devolatalize coal in order to rapidly combust pyrolysis products, thereby generating intense combustion oscillations. With the second chamber acoustically tuned with the first, the intense oscillation field is then utilized to complete char burn-out over more prolonged periods covering one or several combustion cycles. It is now believed that this configuration merits further investigation in future work. Figure 5-1 shows a plot of carbon burn-out efficiency as a function of average combustor residence time for selected runs conducted using the refractory-lined conical chamber. A clear trend exists between the combustor mean residence time and the carbon burn-out efficiency. The only exception to this trend was exhibited by the run conducted on September 24, 1987. The measured oxygen content in the dry flue gas was 2 percent for this run. Included in this 2 percent value is a significant fraction of Argon which cannot be resolved by MTCI's gas chromatograph. Therefore, the uncharacteristically low conversion level is seen to be due to the prevailing substoichiometric conditions for this test.

Based on a modest extrapolation of the test results, it is evident that the pulse combustor is capable of achieving 99 percent carbon burn-out at 50 percent excess air and a mean combustor residence time of approximately 25 milliseconds or greater.

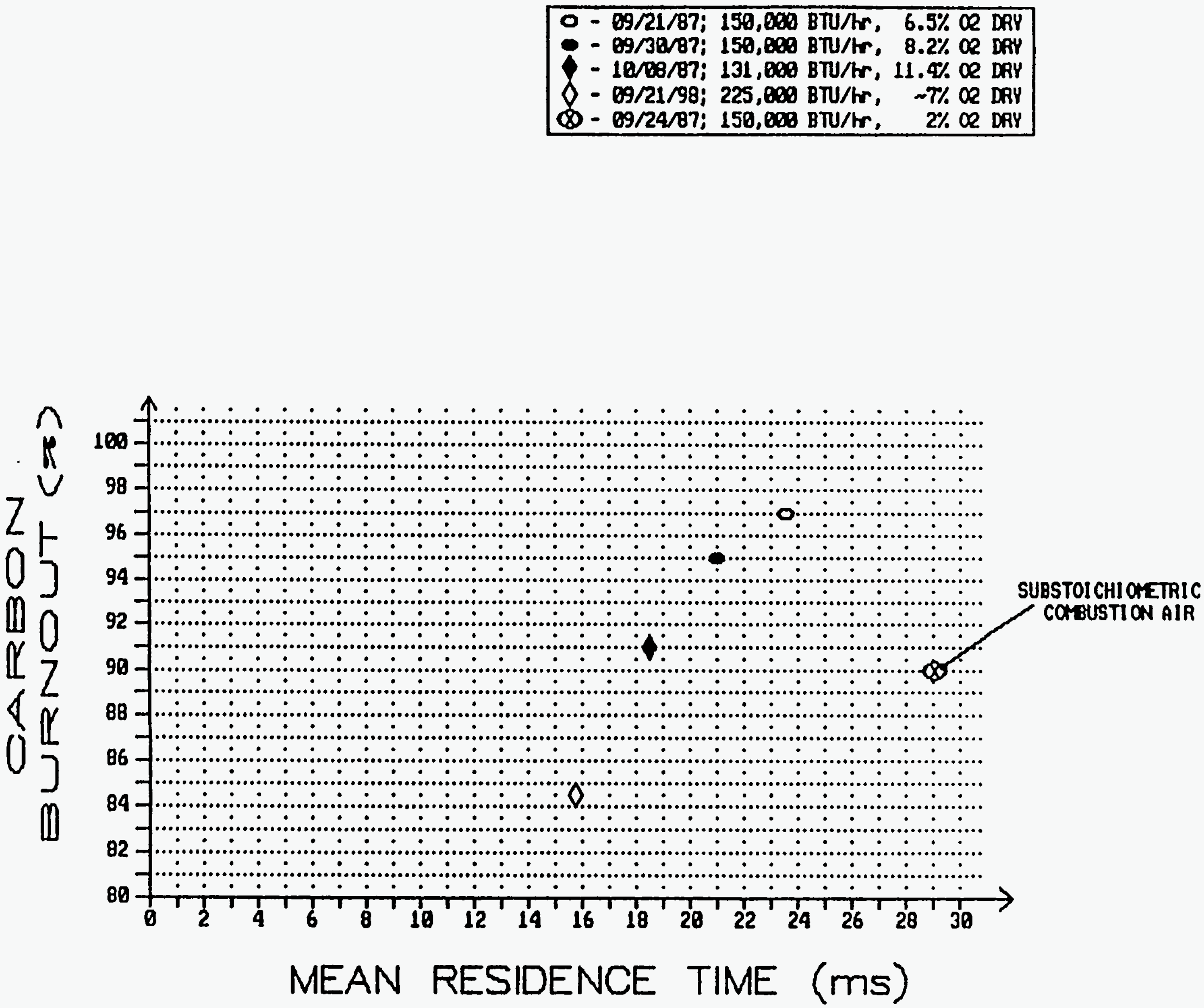


FIGURE 5-1:

CARBON BURN-OUT AS A FUNCTION OF COMBUSTOR MEAN RESIDENCE TIME

5.2 CARBON MONOXIDE EMISSIONS

The carbon monoxide emissions from the refractory-lined conical combustor ranged from 10,000 ppm to approximately 2000 ppm (corrected to 3% O_2) for runs having dry flue gas O_2 concentrations ranging from less than 2 percent up to 11.4 percent. As shown in Figure 5-2, the adjusted carbon monoxide levels correlate strongly with the oxygen content of the flue gas.

These relatively high carbon monoxide levels may be a result of insufficient mixing within the combustion chamber. It is probable that insufficient mixing may be limited only to localized zones within the combustor. The precise location of these zones has not yet been identified. It is also possible that the high heat transfer rates in the bare resonance tubes results in rapid kinetic quenching of the reaction between residual carbon monoxide and oxygen. It is thought that the incorporation of a radiation shield near the inlet portion of the resonance tubes may provide sufficiently longer residence times in a high temperature environment to significantly reduce the residual carbon monoxide levels.

Further development work is necessary to fine tune the pulse combustion apparatus to reduce carbon monoxide emissions to a more acceptable level. It should be mentioned that even conventional, gas or oil-fired, residential space heaters are quite sensitive to fine adjustments of the burner. It is not uncommon for emissions from these units to vary by an order of magnitude as a function of burner adjustment.

5.3 NITROGEN OXIDE EMISSIONS

Nitrogen oxide emissions from the pulse combustors were typically in the range of 300 ppm to 400 ppm corrected to 3% O_2 (approximately .45 to .6 lb/MMBtu as NO2). These low NOx emission values are consistent with the relatively low temperature of operation for the dry ash pulse combustors. However, insufficient data exists to predict the precise nitrogen oxide emission levels as a function of firing condition.

5.4 COMPARISON OF TEST RESULTS WITH TARGET GOALS

Operational, performance, and emission targets have been developed for coal fuel-based, residential space heaters. These targets reflect desired attributes which will allow significant market penetration for coal-based systems in competition with gas or oil-fired units.

Table 5-1 summarizes and compares the target goals with the experimental results obtained using pulse combustors under this investigation. As seen in the table, the experimental results achieve the target goals in every case with the exception of combustion efficiency. Although 99 percent combustion efficiency was not achieved during the course of this investigation, significant improvements in the level of carbon burn-out were progressively

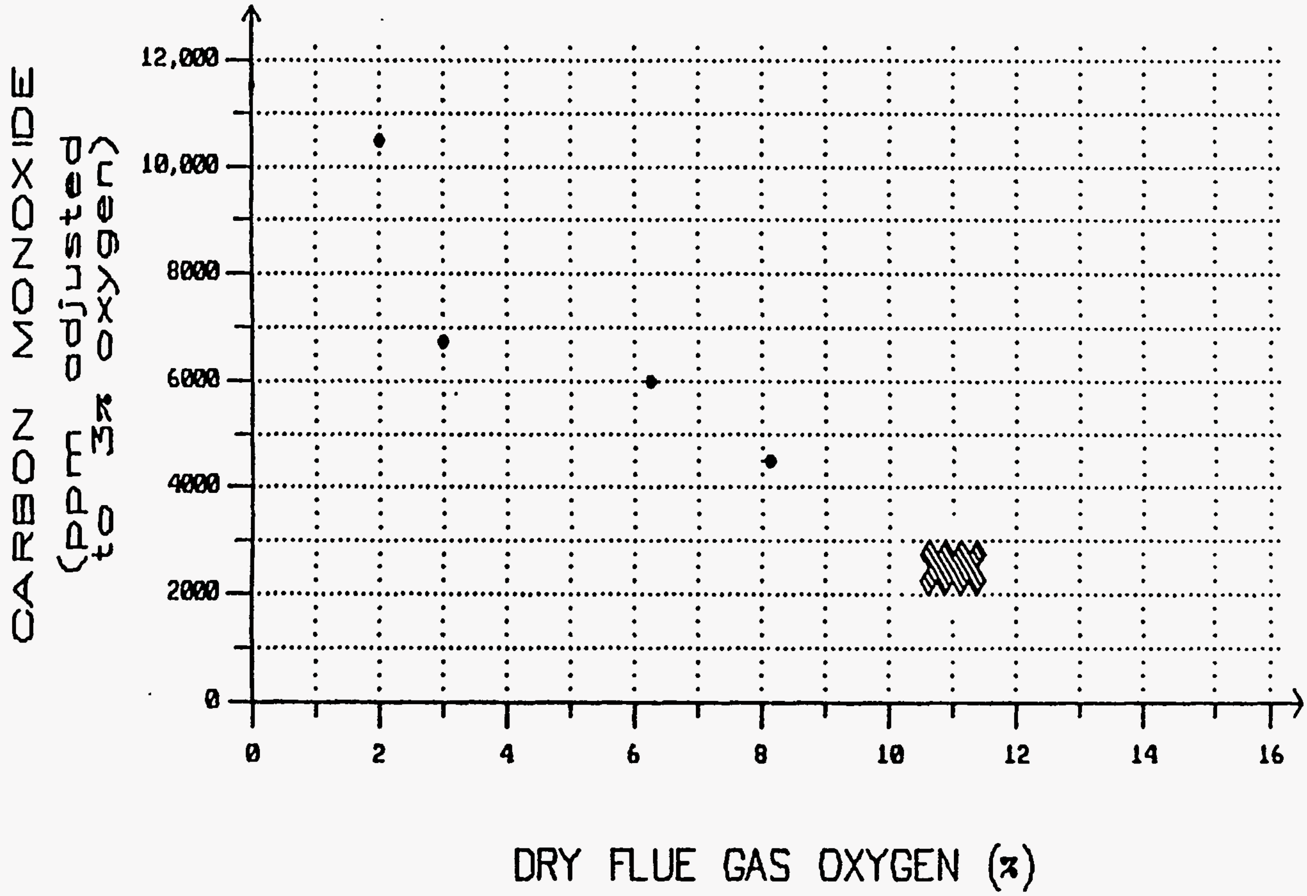


FIGURE 5-2:

EFFECT OF FLUE GAS OXYGEN CONTENT ON CARBON MONOXIDE EMISSIONS

made throughout the course of this investigation. Furthermore, the data resulting from this work indicates that >99 percent combustion efficiency is achievable under properly controlled conditions.

TABLE 5-1

COMPARISON OF EXPERIMENTAL RESULTS WITH TARGET GOALS FOR COAL-FUELED RESIDENTIAL SPACE HEATERS

ITEM	TARGET GOALS	EXPERIMENTAL RESULTS
Fuel Selection	DUC or CWM	DUC
Ignition	Automatic	Achievable
Response Time	< 5 minutes to full load	5 minutes with continuous pilot
Combustion Efficiency	> 99 Percent	> 97 Percent
NO_x (as NO_2)	< .5 lb/mmBtu	< .5 lbb/mmBtu
SO_2	< 1.0 lb/mmBtu	< 1.0 lbb/mmBtu
Height	< 6 Feet	< 6-7 Feet
Floor Space	< 15 Ft^2	< 10 Ft^2

SECTION 6.0

OPTIMUM SYSTEM CONFIGURATION

6.1 SYSTEM DESCRIPTION

Based on the development tests conducted in this Phase I effort, an optimized configuration has been selected for a prototype unit, coal-fueled, residential space heating system. Although the workscope under this contract does not include the preparation of detailed designs or projected economics for an integrated prototype unit, preliminary information has been developed and is presented here. Figure 6-1 depicts a potential commercial embodiment for the optimized residential configuration.

The optimized system incorporates the following key features:

- Tandem configuration
- Refractory-lined dual pulse combustors
- Intake air plenum
- Exhaust plenum with tailpipe coupling
- Automatic fuel biasing
- DUC coal feeder with pneumatic eductor
- Wet cyclone particle trap

The unit is designed to utilize natural gas for start-up. Under normal operating conditions, the unit consumes only dry, ultrafine coal fuel. The turndown ratio is 3:1 with a maximum firing rate of 225,000 Btu/hr and a maximum combustor heat release of 3 MMBtu/ft3/hr. The ignition system consists of a spark ignitor and a pilot flame.

The entire pulse combustion apparatus is enclosed in a thin gauge steel furnace housing. The unit is arranged in a compact format and is anticipated to stand approximately 6-7′ tall and require 10 ft^2, excluding coal storage.

6.2 SYSTEM PERFORMANCE

A forced draft fan supplies circulation air to the housing. The air first passes through a finned-tube heat exchanger and then, directed by internal baffles, passes over the resonance tube and combustor before exiting as hot air to the end-user. It is estimated that approximately 80 percent of the heat is released from the combustor and resonance tubes, while only 20 percent of the heat is released in the final heat exchanger. The unit is capable of achieving 70^oF reheat for circulation air flows from 800 CFM to 2000 CFM. Thermal efficiencies of greater than 80 percent are anticipated.

The unit incorporates a wet particle cyclone which is capable of 95 percent fly ash removal efficiency. Ash rejection occurs in wet form with discharge to the residential sewer lines.

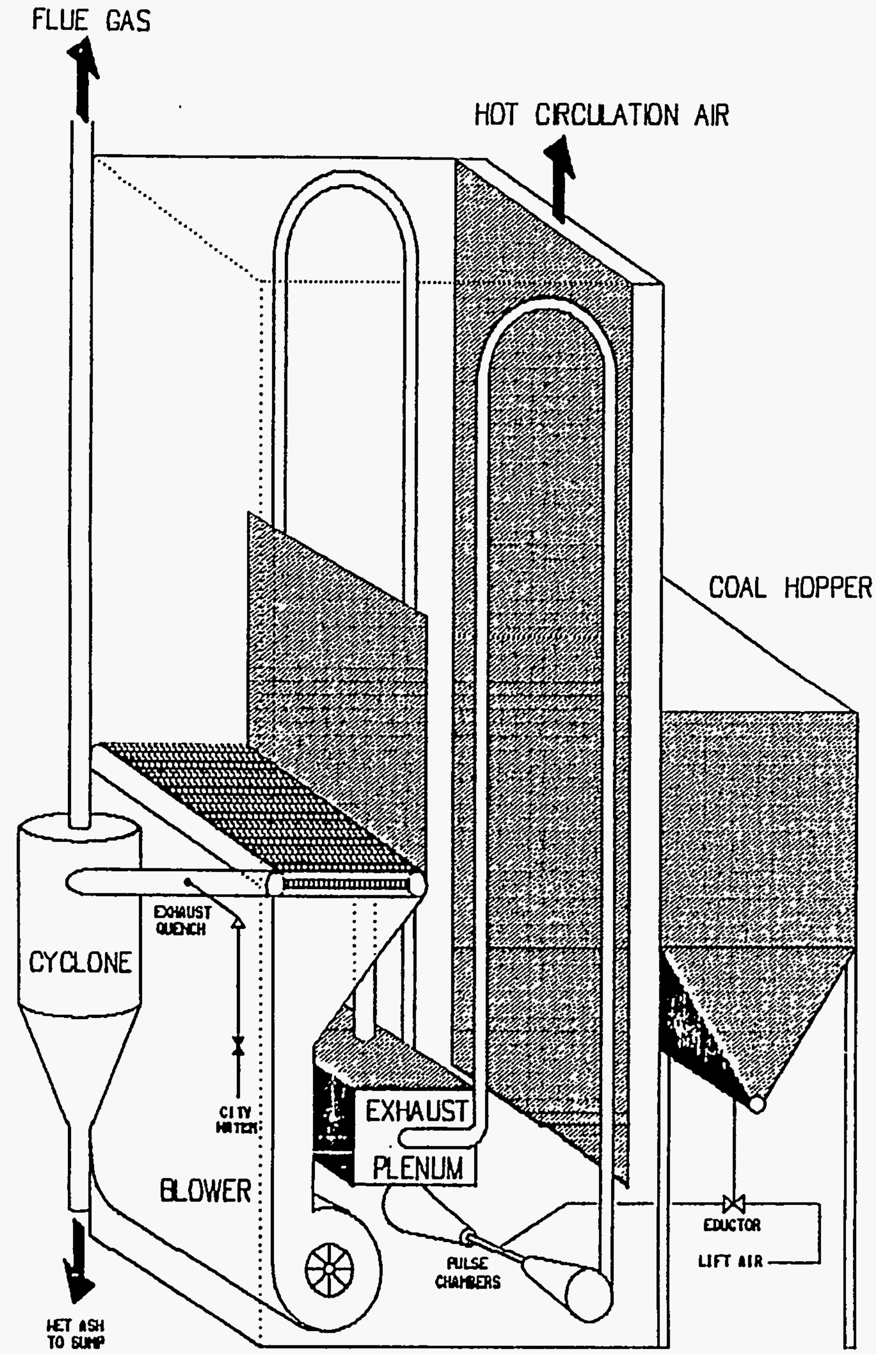


FIGURE 6-1:

EQUIPMENT LAYOUT FOR PHASE II SYSTEMS INTEGRATION TESTS

Since the coal feed employed in the unit is a highly beneficiated, micronized coal, SO_2 emissions of less than 1.0 lb/MMBtu are achieved. NO_x emission levels of less than .5 lb/MMBtu are anticipated.

A typical material balance for the prototypical, coal-fueled residential unit is shown in Table 6-1.

6.3 SYSTEM ECONOMICS

Capital costs for the prototypical, residential space heater are given in Table 6-2. The total projected materials and equipment cost is $1370/unit. The pulse combustor itself represents only 15 percent of the total equipment and material cost.

Assembly labor is estimated at $250/unit based on a breakdown of mechanical, structural, and electrical installation requirements. A 40 percent overhead is included as typical of the manufacturing environment resulting in a total manufactured cost of $2268/unit.

A more thorough analysis of shipping, installation, and dealer charges is necessary before the comparative economics between coal-based heaters and gas-based heaters can be made.

TABLE 6-1: MATERIAL BALANCE - RESIDENTIAL PULSE COMBUSTOR

Feed Type	:	DUC Coal (15,000 Btu/hr)
Nominal Firing Rate	:	100,000 Btu/hr
Capacity Factor	:	15 % (1315 hr/yr; 131.5 MMBtu/yr)
Coal Feed	:	8760 lb/yr
Total Rejected Solids	:	193 lb/yr (105 lb Ash, 88 lb Carbon)
Water Spray	:	600 Gallon/yr

TABLE 6-2: RESIDENTIAL UNIT - PROJECTED ECONOMICS

		COST $
Equipment and Materials:	Storage Bin	80.00
	Coal Feed System	350.00
	Pulse Combustor	200.00
	Heat Exchanger	40.00
	Ash Rejection System	120.00
	Furnace Enclosure	200.00
	Fans, Valves, Ignitors and Controls	380.00
	TOTAL MATERIALS	1370.00
Assembly Labor :	Mechanical (3 Manhours)	75.00
	Structural (3 Manhours)	75.00
	Electrical (4 Manhours)	100.00
	TOTAL LABOR	250.00
Overhead :	40% of Manufactured Cost	648.00
	TOTAL	2268.00

SECTION 7.0

SUMMARY AND CONCLUSIONS