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ADVANCED COAL-FUELED COMBUSTOR FOR RESIDENTIAL SPACE HEATING APPLICATIONS

Prototype Design Report

TOPICAL REPORT

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Prepared for:

F. W. Steffgen
J. C. Winslow
U.S. Department of Energy
Pittsburgh Energy Technology Center

Prepared by:

Y. Kwan
S. L. Chen
W. R. Seeker
Energy and Environmental Research Corporation
18 Mason
Irvine, CA 92718

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SUMMARY

This report reviews the design of the prototype coal-fired space heating system for residential applications. The entire unit comprises the following key components, namely: a combustor, heat exchanger, and ash collector.

The combustor was developed in Phase I of this program and is optimized in Phase II. The design of this optimized combustor consists of a short refractory quarl, swirl stabilized burner and a double shell combustion chamber made of stainless steel. The burner was designed to burn dry ultrafine, high volatile bituminous coal at a nominal firing rate of 100 x 10^3 Btu/hr and 30 percent excess air. The combustor was designed to utilize the combustion air as cooling air for the chamber. The combustion chamber was designed to yield a gas phase residence time of approximately 1 second.

The heat exchanger is a commercial design, by the Will-Burt Company. The unit is engineered for minimum pressure drop and maximum heated surfaces for heat transfer.

Particulate laden flue gas clean-up is achieved by an ash collector. This collector is designed especially for this application since no commercial ash collector is available for this reduced scale. The ash collector consists of primarily a bag filter and is designed for easy ash removal and cleaning.

1.0 INTRODUCTION

The objective of this program is to develop an integrated combustor/heat exchanger which burns coal either as dry ultrafine coal or as a coal water slurry and which is suitable for use in a range of residential space heaters. The program has been divided into two phases. To date, Phase I has been completed. In Phase I, combustor concepts were developed and evaluated by means of mathematical modeling and experimental testing. The most promising concept, an air-cooled combustion chamber, was fabricated and tested. This first generation combustor system was subject to further testing and optimization in Phase II. The optimized combustor configuration then will be integrated with a commercially available heat exchanger to complete the prototype residential space heating system.

This report documents briefly the design of this 100 x 10^3 Btu/hr coal-fueled space heating system. Figure 1-1 illustrates schematically the major components for a coal-fired warm air system. There are three key components:

- Combustor which is responsible for converting coal energy to sensible remote heat
- A heat exchanger that transfer sensible heat to circulating air
- An ash collector subsystem for final particulate control

The performance goals of this space heater are specified as follows:

Primary Fuel:	Coal/liquid mixture or dry ultrafine coal
Ignition:	Automatic
Response Time:	$\leq$ 5 minutes to full load
Reliability/Safety:	Comparable to oil-fired residential heating systems
Steady-State Efficiency:	> 80 percent
Combustion Efficiency:	> 99 percent
Daily Maintenance:	None

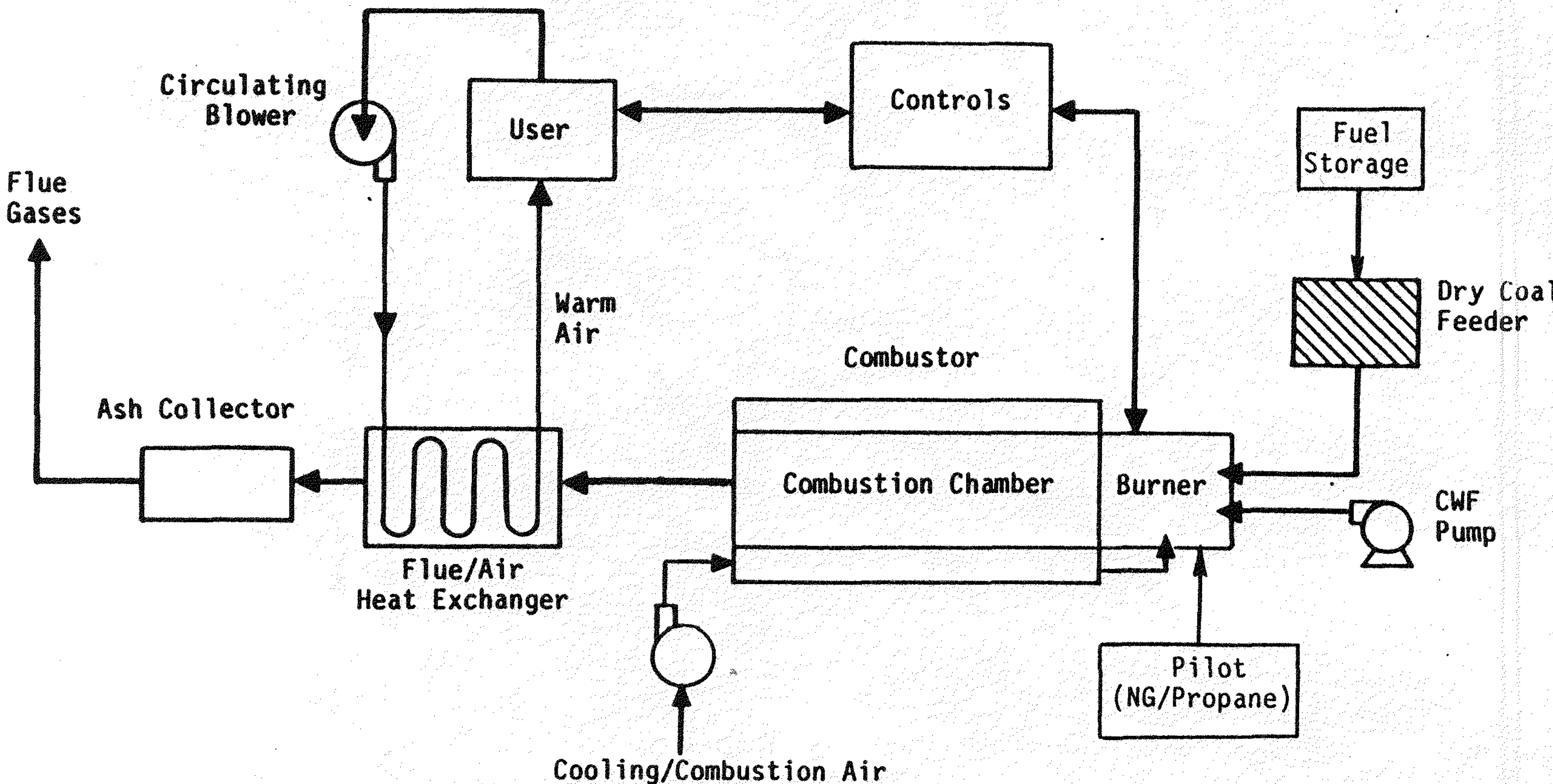


Figure 1-1. Schematic of a coal-fueled, forced-air furnace system for residential applications.

Scheduled Maintenance: $\leq$ twice a year

Size Constraints: Height - 6 ft, Floor Space - 15 ft^2

Service Life: $\geq$ 20 years

In the next section, the report describes the detailed design of each of the three components of the coal-fired space heating system.

2.0 PROTOTYPE SPACE HEATING SYSTEM DESIGN

Figure 2-1 shows a cross-sectional view of the integrated coal-fired space heater as currently envisioned. The combustor is configured in a down-fired mode with the burner mounted at the top of the combustion chamber and the flue gas exiting from the bottom. Upon leaving the combustor, the hot flue gas passes through a shell heat exchanger and then through a high efficiency particulate collector in which most of the ash particles, together with other particulate matter (i.e., unburned carbon) is collected. The entire unit, except for the fuel storage unit, is designed to fit within the 15 ft^2 x 6 ft. height space constraint. It is expected that the fuel storage unit will be placed elsewhere in the house not only because of the space requirement, but more importantly for safety reasons, especially when utilizing dry ultrafine coal. The design of the coal storage and its capacity is beyond the scope of this project. It is agreed, in general, that the in-house storage capacity equivalent to about 250 hrs. of firing would appear to be reasonable (Bartis, 1985).

Figure 2-2 shows an example of a material balance for a 100,000 Btu/hr combustor burning dry ultrafine coal. The material balance shows that the feed rates to the combustor are roughly 7 lb/hr coal and 106 lb/hr (1400 SCFH) air at 50 percent excess. Using air at 60^oF as combustor air, the maximum temperature which could be reached is about 2880^oF; however, due to heat losses, the actual temperature would be considerably lower. For the indicated coal sulfur content of 0.7 percent, or about 0.45 lb/10^6 Btu, the flue gas SO_2 content is about 380 ppm. At the ash level of 1.3 percent, or about 0.85 lb/10^6, the monthly ash production at 50 percent capacity factor would be about 30 lbs.

2.1 Combustor Design

In developing the combustor design, various design parameters were considered, along with their impacts on combustor performance. The key design parameters were:

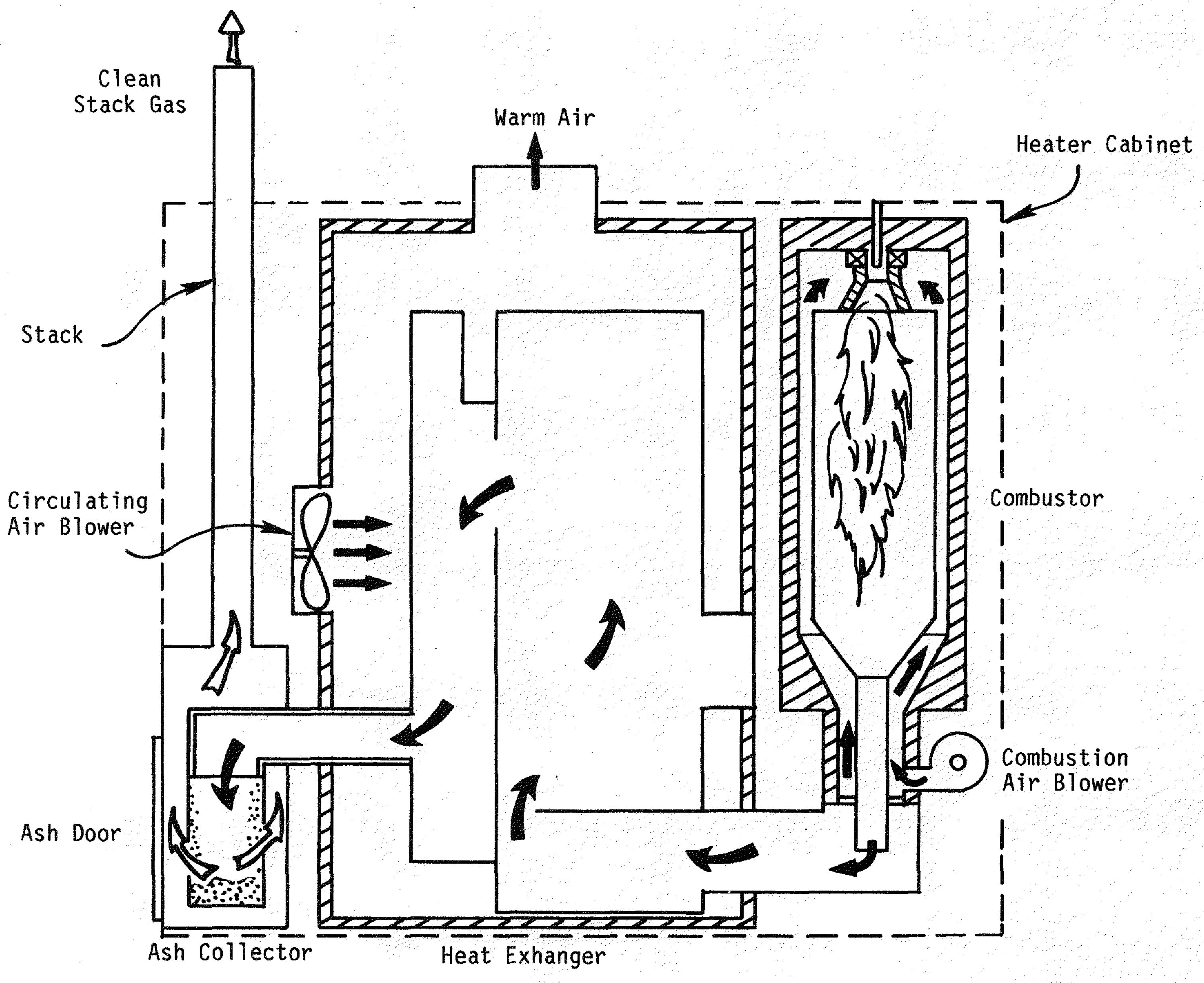
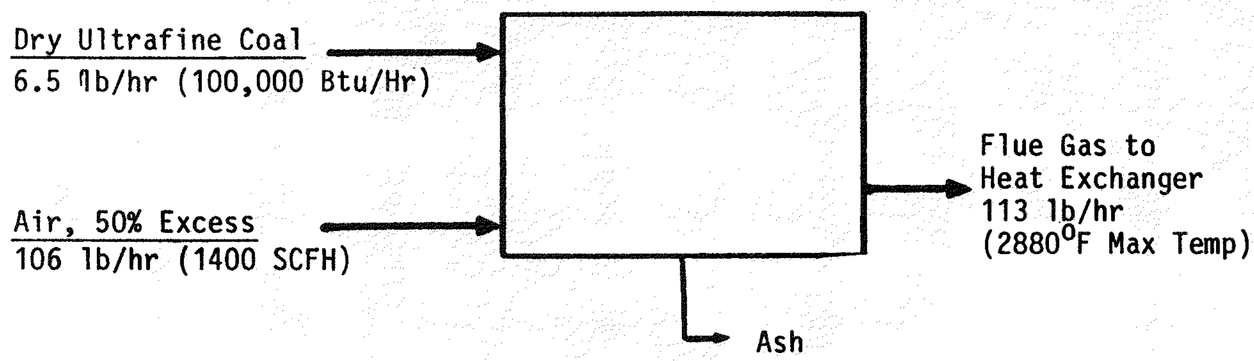


Figure 2-1. Integrated coal-fueled prototype space heating system.



0.0845 lb/hr Ash (30 lb/month @ 50% Cap Factor)

DUF Composition

FC 61 Wt Percent
VM 37
MM 1.3 (0.7% S)
HV 15,300 Btu/lb
C 82.3
H 5.1
N 1.5
S 0.7
O 8.7
Ash 1.3

Flue Gas Composition (Vol. Basis)

CO_2 11.7%
H_2O 4.4%
N_2 77
O_2 7
SO_2 380 ppm

Figure 2-2. Combustor material balance.

- Burner configuration - The chief function of this parameter is to produce a stable flame of the desired size and shape.

- Combustion Chamber Volume - This parameter is dictated primarily by residence time requirements of a given fuel, and its shape is determined by the flame shape produced by a given burner configuration.

- Material of Construction - The selection of a given type of material of construction influences significantly the combustor's combustion performance and service life as well.

In addition to the hardware considerations, other design parameters such as operating parameters were also addressed. For example, what amount of excess air would be appropriate to achieve good carbon burnout while maximizing thermal efficiency? A detailed discussion of the combustor design criteria and the development of the conceptual combustor design can be found in the Phase I Final Report (Kwan et al., 1988). In this part of the report, emphasis is given to the engineering designs of the optimized combustor which is to be integrated with other components to form the prototype space heater. Detailed design drawings with dimensions and material specifications for the optimized combustor are presented in the Appendix.

2.1.1 Burner Design

The primary function of a burner is to produce a stable flame. Flame stabilization is achieved by a combination of reduced fuel and air velocities and recirculation or back mixing of combustion products and heat to the inflowing fuel and air. There are several techniques for stabilizing a flame, among them, the use of a swirl generator to create a recirculation flow pattern near the burner zone, or the use of a hot refractory quarl to stabilize a flame by re-radiating to the inflowing fuel and air. Both flame stabilization methods were evaluated extensively through testing in Phase I. The results of that series of tests suggested that the long refractory-line quarl provided excellent flame stability as well as carbon burnout.

Figure 2-3 shows the carbon burnout produced with this burner with the short quarl swirl burner firing dry ultrafine coal. The long quarl burner also gave the best performance with coal-water fuel (CWF); however, since carbon burnout did not exceed 93 percent, CWF was not employed in Phase II testing. It is obvious that the long quarl burner is superior to the short quarl burner as far as carbon burnout is concerned. The long quarl burner was not chosen for optimization because it has two major drawbacks. First, it requires a long heat-up time to reach proper operating temperature during start-up due to high thermal inertia. This also implies that the consumption of a large quantity of auxiliary fuel is necessary before the coal flame can be stabilized. Secondly, due to the nature of the cycling operation in most space heating applications, refractory materials subjected to frequent heat-up and cool-down are at high risk of failure. With these concerns in mind, the short quarl swirl burner was selected to be optimized.

Figure 2-4 shows a cross-sectional view of the short quarl burner and its design specifications are listed in Table 2-1. Two swirl generators were evaluated during the optimization testing. The 60 degree swirl generator which provides a swirl intensity equivalent to a swirl number of 0.9, was the one used during Phase I characterization tests. The second swirl generator, a 75 degree radial vane type, provides an estimated swirl number of approximately 1.5. Figure 2-5 shows the carbon burnout results of these two burners tested under higher preheated air temperature condition. The carbon burnout result from Phase I was also shown in the same figure for comparison. It should be noted that the operating conditions for Phase I tests were different from the current phase. In Phase I, the excess O_2 was set to 2 percent while the maximum preheated air temperature was about 450^{o}F, which was heated through an electric air heater. For the current phase, tests are conducted in a recuperative mode, in which the preheated air temperature generally exceeds 1000^{o}F. More discussion of the recuperative operation can be found in Section 2.1.2

From Figure 2-5, it can be seen clearly that the carbon burnout goal of 99 percent can be achieved and this goal can be even surpassed with higher preheated air. This observation applies to both swirl designs, that is,

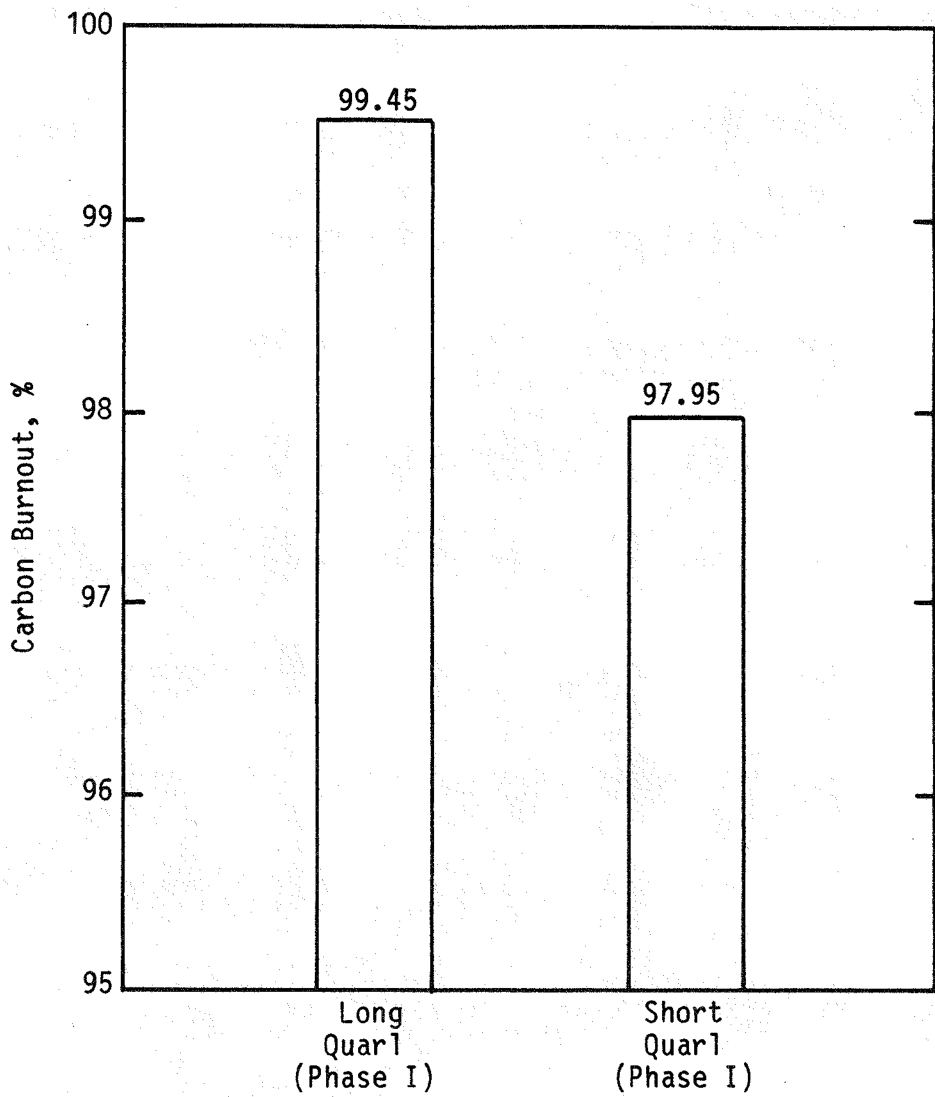


Figure 2-3. Impact of burner design on carbon burnout for dry ultrafine coal - Phase I Results.

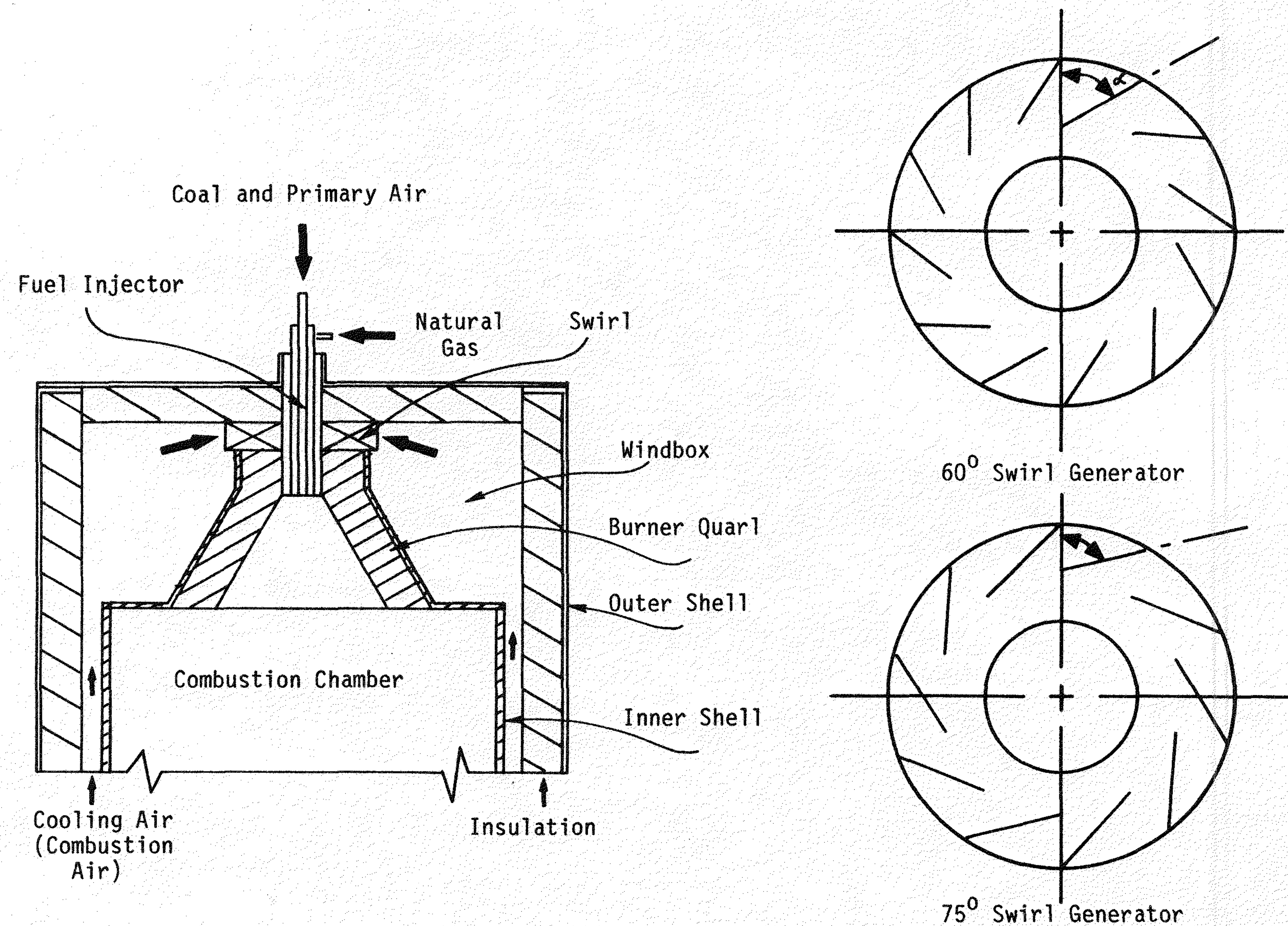


Figure 2-4. Cross-sectional view of optimized burner.

TABLE 2-1. PROTOTYPE BURNER SPECIFICATIONS

Parameters	Short Quarl
Firing Rate (Btu/hr)	100,000
<u>Primary Design</u>	
Stoichiometry (% TA)	35
Coal Nozzle Diameter (in)	0.495
Velocity @ $70^{o}F$ (ft/s)	70
<u>Coal Nozzle</u>	
Open Area (in^2)	0.15
Velocity @ $70^{o}F$ (ft/s)	93
Nozzle Position	Adjustable
<u>Secondary Design</u>	
Velocity (ft/s)	100
(SR_T = 1.4, $1000^{o}F$)	
ΔP (Inch WC)	5
<u>Swirl Generator (Removable)</u>	Fixed Radial Vanes
Vane Angle (Degree)	60,75
Swirl Number (Estimate)	0.91, 1.54
<u>Burner Exit</u>	Refractory-Lined
Exit Diameter (in)	5.25
Half Angle (Degree)	40
L/D	0.66
Configuration	Air-Cooled

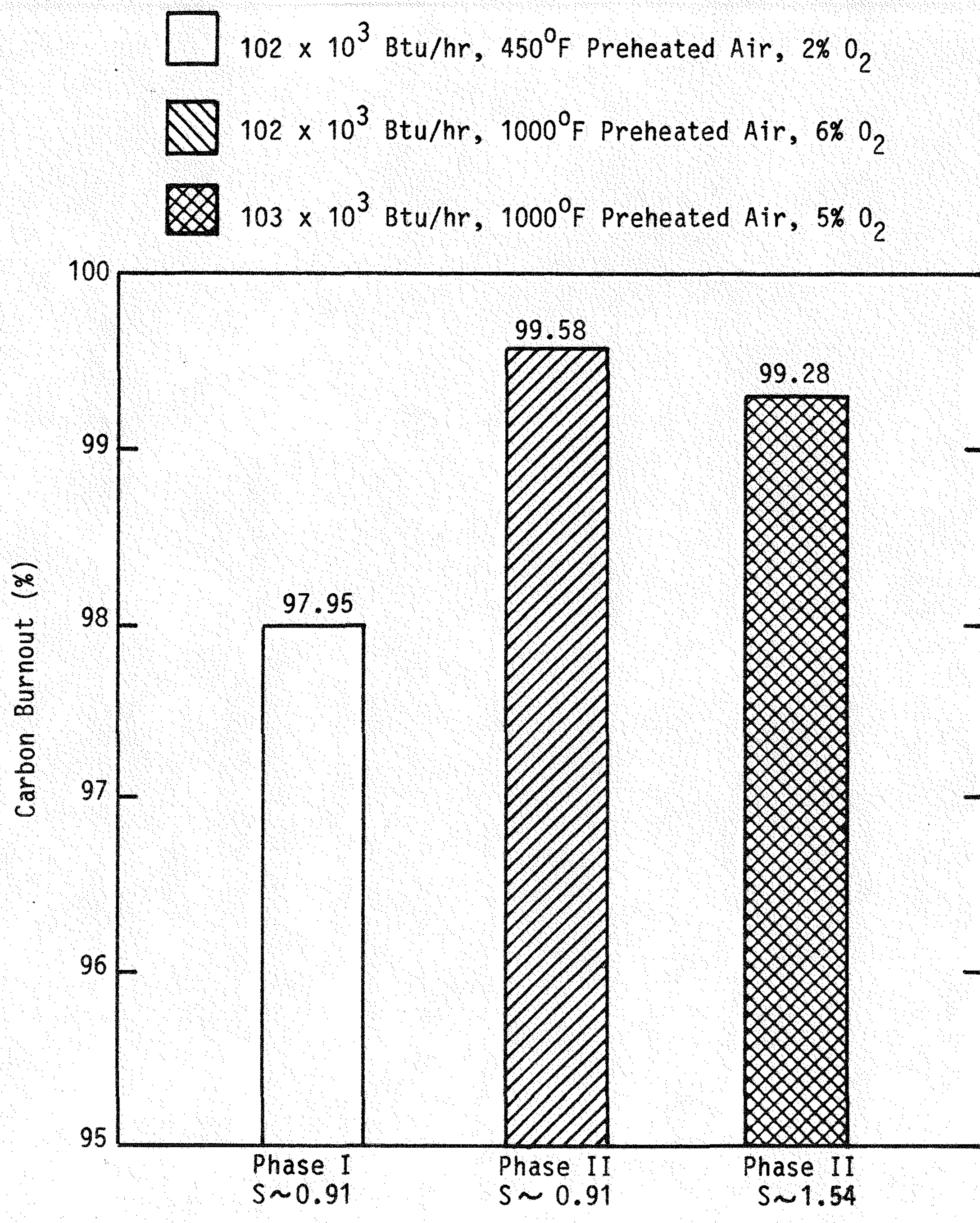


Figure 2-5. Carbon burnout for different burner designs and operating conditions - Phase II Burner Optimization.

99.58 and 99.28 percent carbon burnout for swirlers producing swirl number of 0.9 and 1.5, respectively. From the carbon burnout point of view, it appears that the low swirl intensity design seems to be more desirable. However, in evaluating both burner configurations, the higher swirl design showed better performance as far as flame stabilization is concerned, which is the subject of discussion in the next subsection.

2.1.1.1 Response Time

As mentioned earlier, the primary function of a burner is to produce a "stable flame". The question of how effective a burner is in stabilizing a flame was addressed in Phase II. The question is addressed as response time. Two modes of response time are defined as part of the burner development:

- Cold start-up response time, τ_{cs}
- Hot cycle restart response time, τ_{rs}

The cold start-up response time is defined as the elapsed time from the moment a cold combustor is energized to the moment when no support fuel is needed to sustain a full load coal flame. Figure 2-6 compares the cold start-up characteristics of the two burner designs as well as the amounts of auxiliary fuel (natural gas) consumed during the start-up period. With the low swirl burner ($S \sim 0.9$), full load flame stability was achieved in about an hour. The corresponding auxiliary fuel consumption during the start-up was about 48 cu. ft. As for the high swirl burner, cold start-up time was reduced to half, and about 20 percent of natural gas was saved. Tests were also conducted to see whether higher firing rates could reduce response time. Results from this series of tests are also given in Figure 2-6. By using the low swirl burner, cold start-up response time was reduced from an hour to about 43 minutes when firing rate was raised from 103×10^3 to 122×10^3 Btu/hr. Auxiliary fuel consumption was also reduced by 16 percent approximately.

The second type of response time is for cycling operation. A forced-air furnace or space heater commonly operates in a cycling mode, depending on heat demand, which in turn, depends on many factors. Among them, for

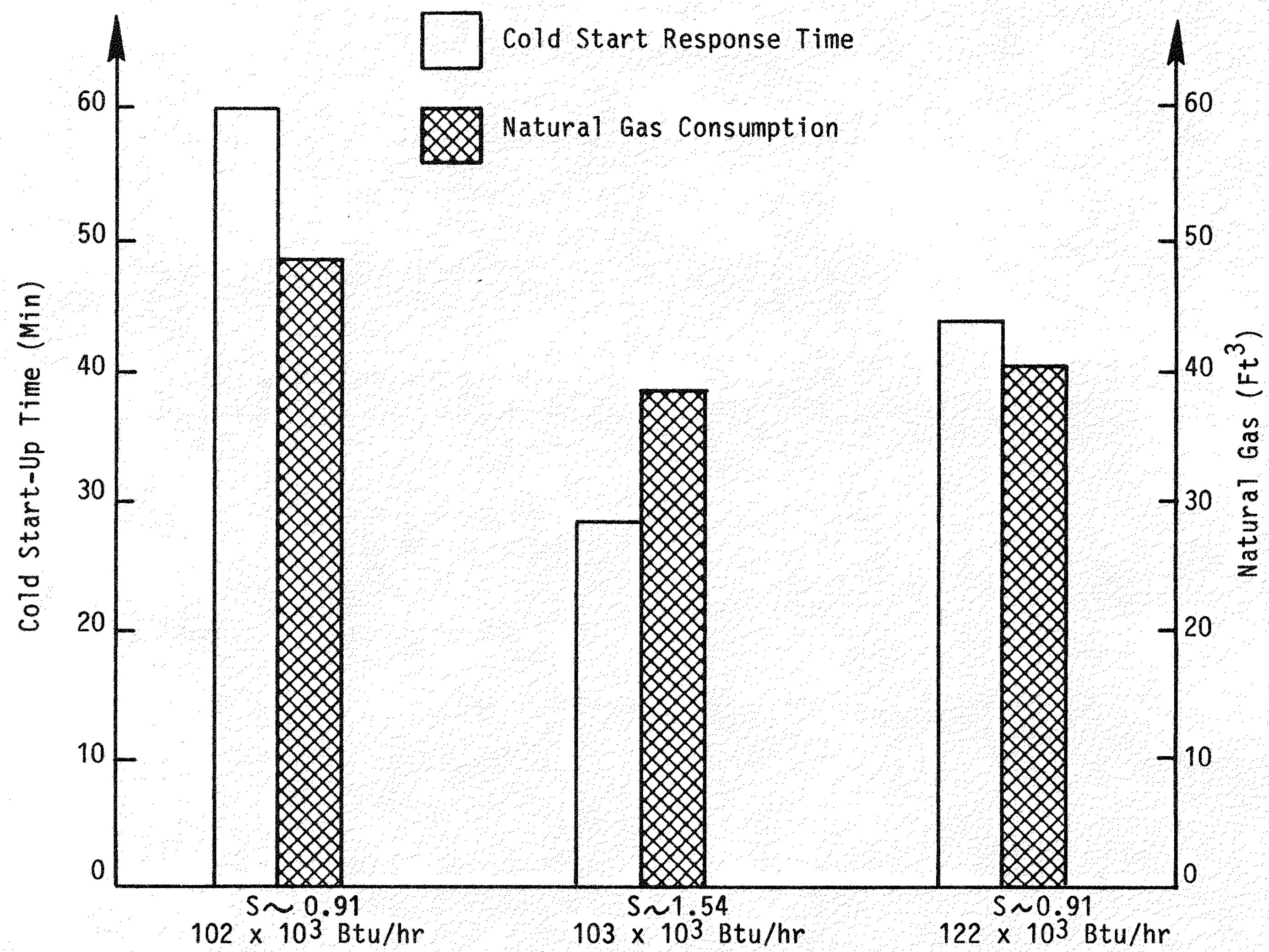


Figure 2-6. Cold start-up response for different burner designs and operating conditions.

example, are outdoor weather, type of insulation of a house, thermostat setpoint, etc. The cycling response time is defined as the elapsed time for the combustor to achieve a stable, full load flame from the time the combustor is re-energized to the point where no auxiliary fuel is needed.

Cycling response time for the low-swirl burner was briefly characterized. Because a variety of conditions can affect the cycling response time, a set of fixed conditions was established to conduct this series of tests, namely:

- Combustor fired on full load (100 x 10^3 Btu/hr)
- Combustor provided stable coal flame (no pilot)
- Combustor reached a thermal equilibrium condition (i.e., steady-state surface temperature at a given location of the combustor) before shutdown
- Pilot size was about 10 x 10^3 Btu/hr
- Coal feed energized upon confirmation of a pilot flame

Test results for this series of experiments are presented in Figure 2-7. Cases (a) and (b) represent the combustor shutdown for 10 and 15 minutes, respectively. In both cases, the burner produced a stable coal flame after 5 minutes of pilot-assisted ignition. For Case (c) where the shutdown time was 20 minutes, the burner required about 7 minutes of pilot flame to restore stability of coal flame. As the time of shutdown was increased to 25 minutes (Case d), about 22 minutes of continuous pilot support was needed to restore flame stability. It is interesting to see that this response time (Case d) can be cut to half by increasing the nominal firing rate from 100 x 10^3 Btu/hr to 120 btu/hr (Case e). This improvement of pilot-assist time is probably attributed to higher firing density which offsets the huge heat losses due to extended cool-down period during shutdown. Again, it is expected that the cycling response time will be a function of shutdown time. If the typical

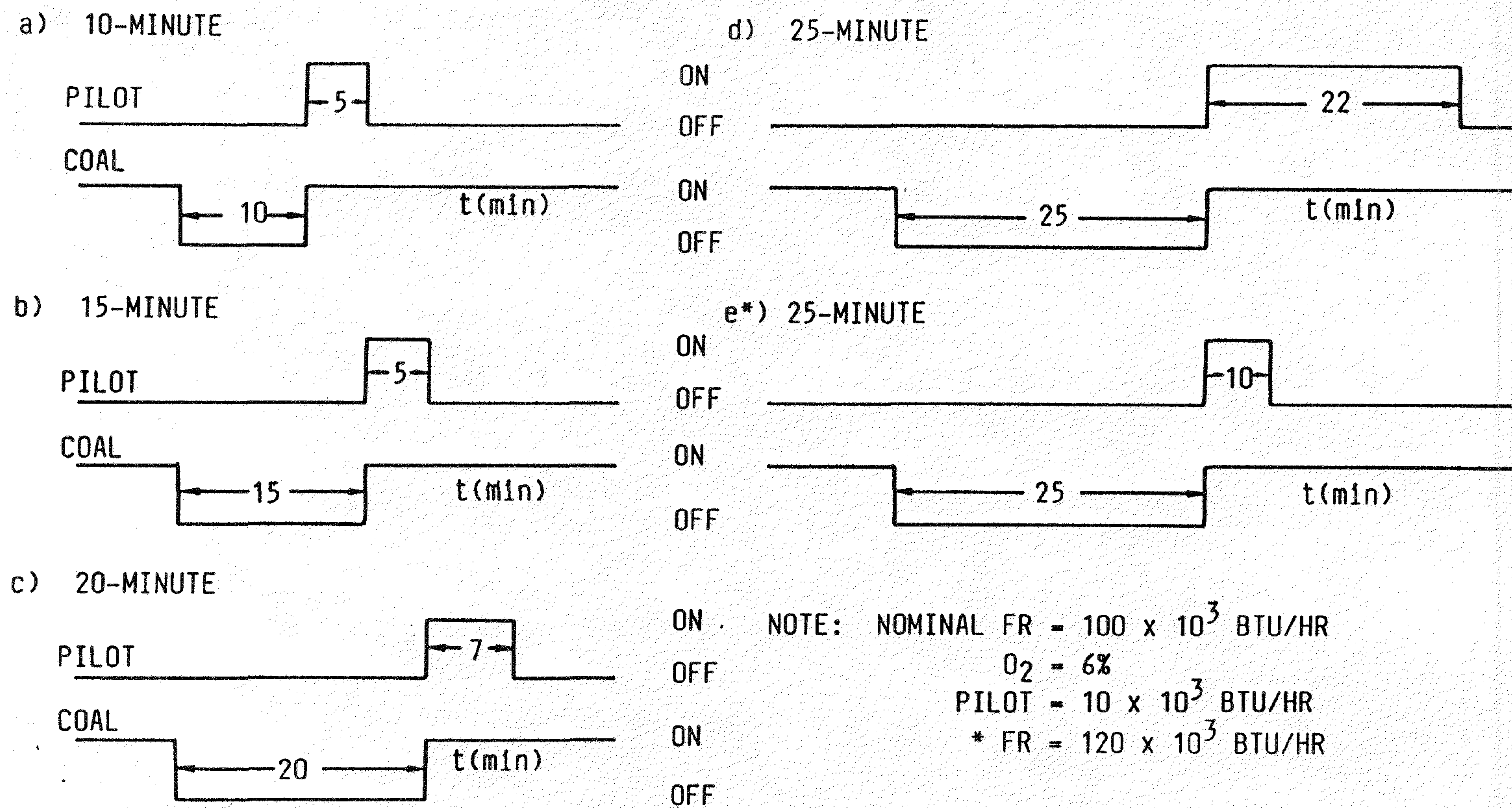


Figure 2-7. Characteristics of cycling response time for several operating conditions.

cycling mode (down time) of a space heater is between 10 and 15 minutes, 5 minutes response time will be easily achievable with this low swirl burner. Based on the performance of the cold start-up response time, the high swirl burner appears to perform much better. Full evaluation of this response time characteristics for both cold and cycling modes will be carried out for the integrated prototype combustor.

2.1.2 Combustion Chamber Design

The combustion chamber for the prototype combustor consists of a double-walled design using a jacket of cooling air surrounding the combustion chamber. The combustion chamber, cylinder made of 316 stainless steel, has an internal diameter of 12 inches and a length of 26 inches. The corresponding chamber volume is about 1.3 ft^3, including the transition section of the furnace. The outer shell has an overall length of 40 inches and is made of thin gauge carbon steel. The chamber is configured in such a way that the burner is mounted at the top end, and the flue gases exit at the bottom. To maintain the wall of the combustion chamber below the maximum tolerable level (~ 1500^oF), cooling air circulates upward through the annulus from the bottom of the chamber toward the burner. In the current phase, the inner shell is cooled using the combustion air. This "recuperative" concept has the advantage of transferring heat from the inner shell to the combustion air, which has been shown to be beneficial to carbon burnout and flame stabilization as well. A number of longitudinal fins are included in the design of this inner shell to ensure enough surface area for heat transfer. Furthermore, to minimize heat loss from the outer shell to the surroundings, the inner wall of the outer shell is lined with insulating materials.

2.1.2.1 Thermal Analysis

Heat transfer modeling was applied to predict wall temperatures of the combustion chamber and describe how it is affected by combustion gas temperature, cooling air flowrate, and combustion gas emissivity. Figure 2-8 shows how heat is transferred from the flame to the entire combustion chamber. Heat is transferred from the flame and combustion gases to the

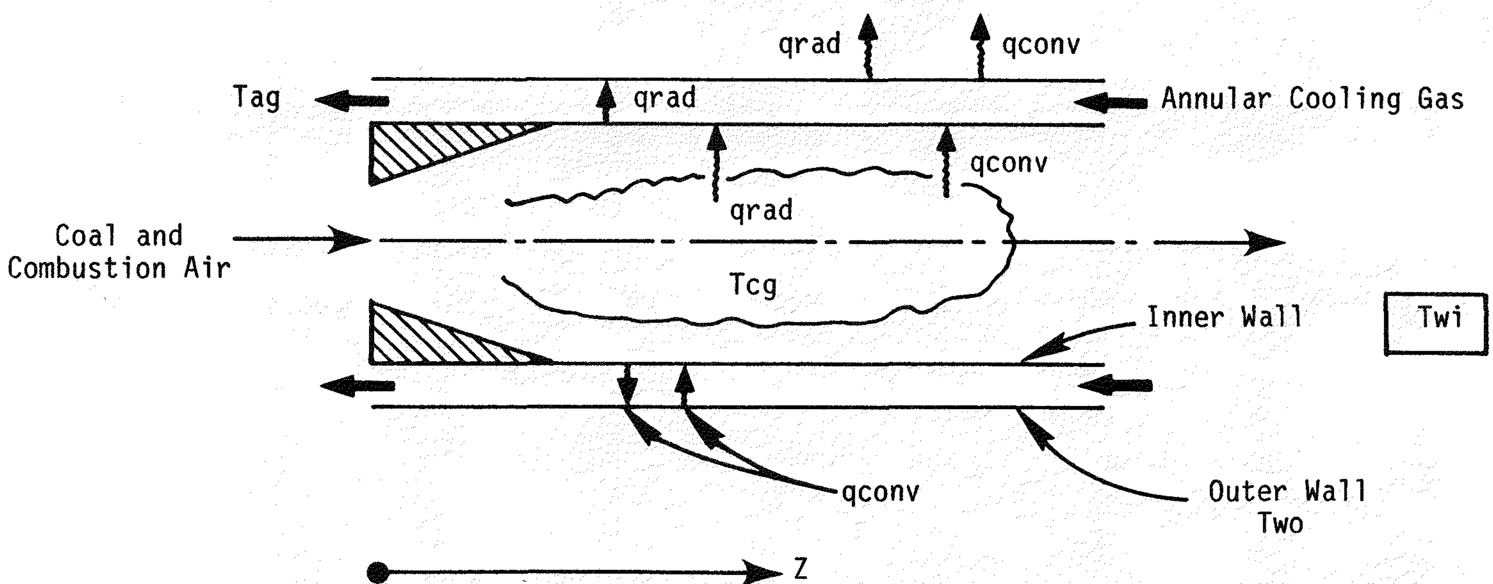


Figure 2-8. Heat transfer analysis.

inner wall chiefly by radiation. Then, there is radiation exchange between the two shells. Heat extraction is achieved by circulating cooling air through the annular section and the heat transfer is primarily convection. Finally, heat loss to the surroundings from the outer shell is by radiation and natural convection.

To model this process, a zone radiation model was applied to describe the heat transfer in this combustion chamber configuration. For computational purposes, the chamber was divided into 10 zones: The assumptions used in developing the model include:

- Surface within a given zone is isothermal
- Gas within a given zone is isothermal
- Radiative exchange between adjacent zones neglected
- Axial conduction in steel liner neglected
- Only the straight wall portion of the combustion chamber considered

To solve the problem, an electric circuit analogue was developed to account for all modes of heat transfer in the system. To obtain the unknown heat transfer rates and temperatures in the analogue, energy balances are made about each node at which the potential is unknown (Kwan et al., 1988).

A summary of typical predicted temperature profiles is shown in Figure 2-9. The stainless steel surfaces are assumed to have an emissivity of 0.65. Temperatures are plotted versus fractional length and include the combustion gas temperature (i.e., initially at 1480 K, 2200^{o}F), the inner wall or liner temperature (T_{wi}), the outer wall temperature (T_{wo}), and the annular cooling gas temperature (T_{ag}). The annular gas flows countercurrent to the combustion gas. The combustion gas has an emissivity of 0.4, which includes the luminosity contribution, and the gas mass flowrates are both 0.0126 kg/s (100 lb/hr), which corresponds to about 30 percent excess air. Notice that the temperature of the cooling gas (air) predicted was clearly below the actual measurement. This is because the model was set-up for a bare inner shell. With added surface area (fins), the actual cooling gas temperatures were measured over 800 K (1000^{o}F) at the windbox.

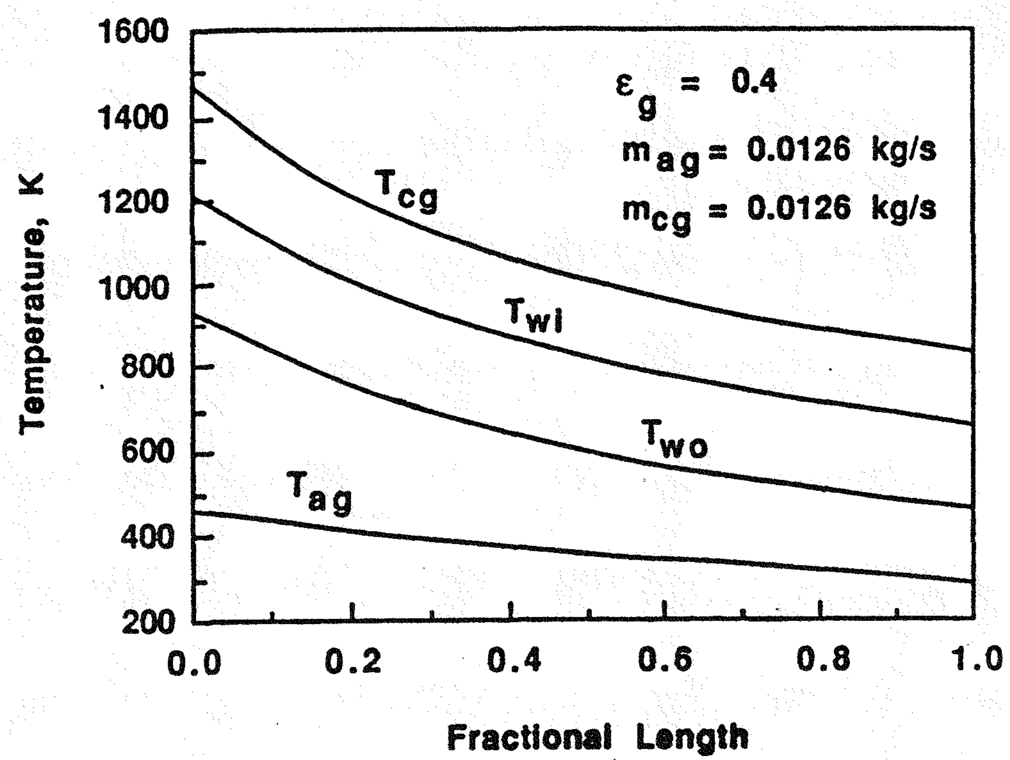


Figure 2-9. Zone model prediction showing temperature profiles for the combustion gas, the inner wall, the outer wall, and the annular cooling gas.

2.2 Heat Exchanger Design

The approach taken to develop a coal-fired space heater is to develop a new coal-fired combustor and integrate with an existing heat exchanger. This approach is taken because it allows the development of a commercial product for the least cost. The commercially-available coal-fired heat exchanger selected for this task is supplied by the Will-Burt Company of Orrville, Ohio. Will-Burt has been manufacturing heating products, including solid fuel heaters, furnaces and stoves, for 50 years.

The heat exchanger selected for the prototype space heater is the Will-Burt "Combustioneer" Model C-10 as shown in Figure 2-10. This heat exchanger is selected because of the simplicity of design, which consists of a two-pass configuration, low pressure drop and minimum passage blocking due to fouling. Additional design specifications of this heat exchanger and circulating blower capacity are listed in Table 2-2. The hot flue gases enter the heat exchanger from the ash pit entrance. Then, it flows upward entering the first cavity of the heat exchanger and turns toward the second cavity before exiting. The forced air (circulating air) is passed over the maximum area of the heated surfaces, including the surface of the ash pit itself. Air passes under the ash pit, scrubbing both sides, front and rear, and top and bottom of the heat exchanger.

2.3 Dust Collector Design

The final key component of the prototype coal-fired space heater is the ash collector. The primary function of this device is to remove particulates of the flue gases before their release to the environment. A commercial dry, bag type collector was originally proposed to be integrated to the design of the prototype space heater. However, due to the diminutive scale of this application, no commercial bag-type collector is available. Therefore, a preliminary design was made and the design criteria are:

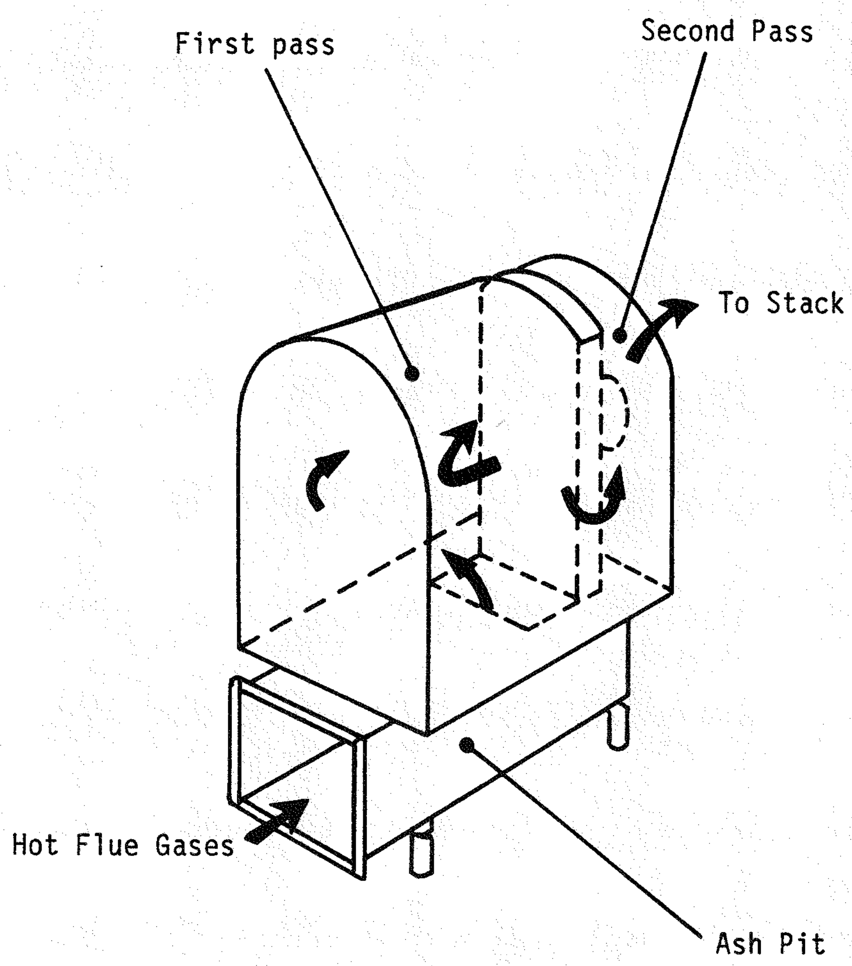


Figure 2-10. Commercial grade heat exchanger.

TABLE 2-2. DESIGN SPECIFICATION OF C-10 HEAT EXCHANGER AND CIRCULATING BLOWER

Material		Carbon Steel
Thermal Output	(100,000 Btu/hr)	95
Heating Surface	(ft^2)	43.5
Length	(Inches)	23
Width	(Inches)	21
Height	(Inches)	39
Blower Motor	(HP)	0.25
Blower Output	(CFM @ 2" WC S.P.)	1400

- Efficiency - It is expected that most particulate (ash particle) generated will be near submicron size. Thus, selection of filter medium must meet this specification.

- ΔP - Pressure drop across the bag filter is another key parameter to consider. Proper flue flow to cloth ratio can minimize this parameter.

- Convenience - the size of the filter bag and the convenience are inter-related. For the current application it is expected that about 2.5 lbs of dry ash will be produced per 24 hours of firing time. The size of the filter will be determined by how often particulates need to be unloaded per heating season.

Figure 2-11 depicts a proposed design of the ash collector for the prototype coal-fired space heater. The collector uses either a disposable bag or removable bag for easy cleaning. An induced draft fan will be installed at the stack to facilitate the gas cleaning process.

2.4 Instrumentation and Controls

Commercial grade instrumentation and control equipment will be used for the integrated system testing. For this particular application, simple system control (On-Off) will be applied.

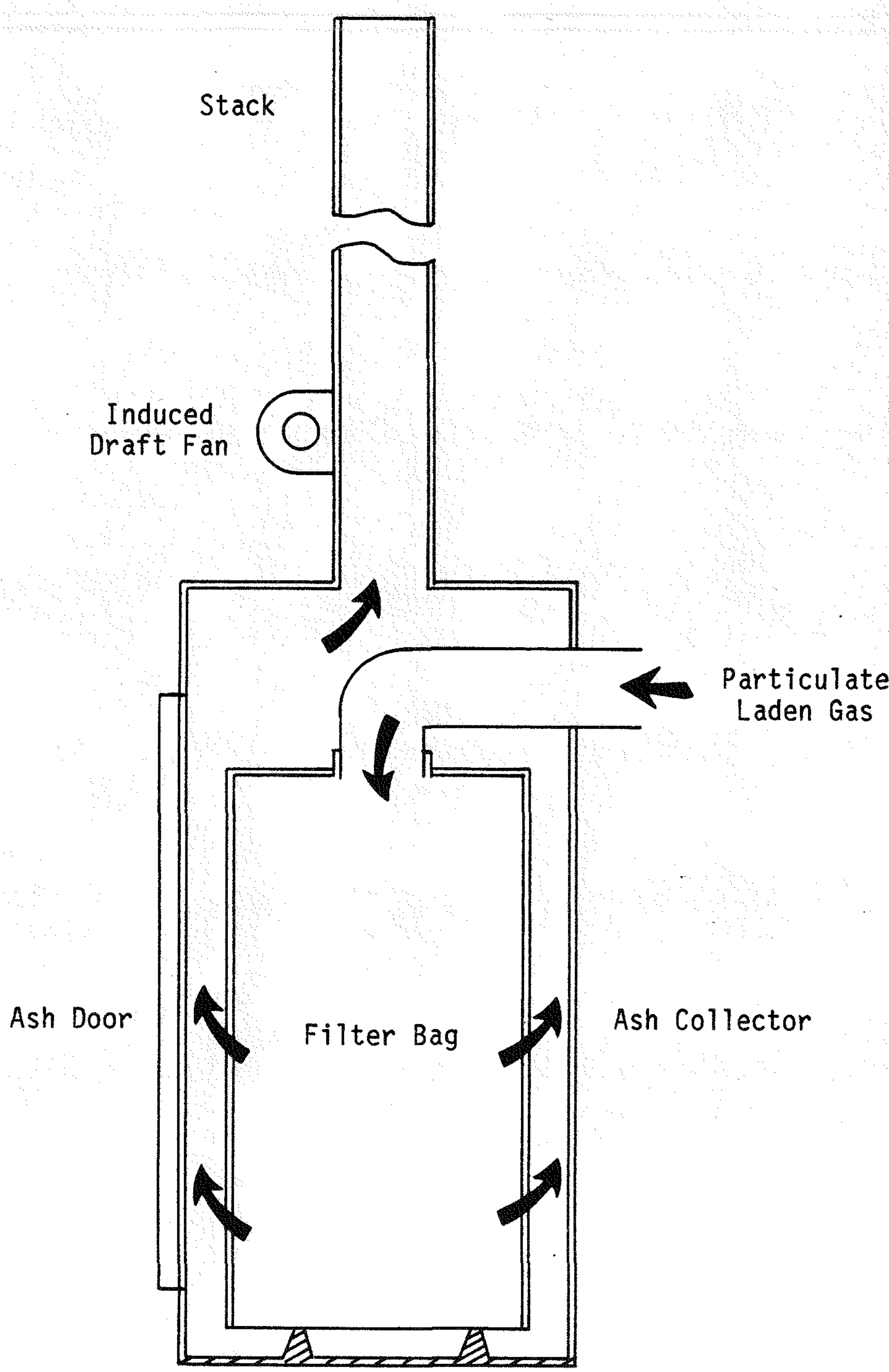


Figure 2-11. Ash collector design.

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